Analysis of Deep Drawing, Ironing and Backward Can Extrusion

With main emphasis on residual stresses and process robustness

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Analysis of deep drawing, ironing and backward can extrusion with main emphasis on residual stresses and process robustness

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Department of Production
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Denmark
2005
Analysis of deep drawing, ironing and backward can extrusion

This thesis has been accepted by The Academy Council at the Faculty of Engineering and Science at Aalborg University for public defence in fulfillment of the requirements for the doctoral degree in technology. The defence will take place at Aalborg University in lecture room 1-2008, Fibigerstræde 16, on Thursday, February 2, 2006 at 12.30 punctually.

Aalborg, November 16, 2005.

Lars Døvling Andersen
Dean
PREFACE

This thesis is submitted to Aalborg University in order to meet the requirements for the doctoral degree (doctor technices) in accordance with the Ministerial order BEK nr 750 of 14/08/1996 and in accordance with rules of Aalborg University.

The work presented in this thesis constitutes some of the research work carried out by the author at the Department of Production, Aalborg University, Denmark in the period 1992 – 2005.

The author wants to thank all the people at Department of Production, Aalborg University and a special thank to the members of the Sheet Metal Forming Group.

The backward can extrusion experiments were carried out in collaboration with Grundfos A/S, Bjerringbro, Denmark. A special thank to Ole Schiøler Sørensen, Erik Madsen and Verner Hansen at Grundfos A/S, Bjerringbro, Denmark.

Aalborg

Joachim Danckert
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<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>semi-die angle of the conical die inlet</td>
</tr>
<tr>
<td>a₁</td>
<td>angle of the die land</td>
</tr>
<tr>
<td>C</td>
<td>constant in Hollomon’s expression</td>
</tr>
<tr>
<td>d₁</td>
<td>diameter</td>
</tr>
<tr>
<td>d₂</td>
<td>diameter of punch shaft</td>
</tr>
<tr>
<td>Dₚ</td>
<td>punch diameter</td>
</tr>
<tr>
<td>E</td>
<td>Young’s modulus</td>
</tr>
<tr>
<td>Eₜₐₙ</td>
<td>tangent modulus</td>
</tr>
<tr>
<td>Fₜₒₜ₉</td>
<td>radial force on the conical part of the ironing die</td>
</tr>
<tr>
<td>Fₙ_die land</td>
<td>radial force on the die land</td>
</tr>
<tr>
<td>Fₙₜₒₜ</td>
<td>total radial force</td>
</tr>
<tr>
<td>H</td>
<td>hardness of tool</td>
</tr>
<tr>
<td>h</td>
<td>height of punch land</td>
</tr>
<tr>
<td>K</td>
<td>constant</td>
</tr>
<tr>
<td>k</td>
<td>shear yield stress</td>
</tr>
<tr>
<td>K’</td>
<td>constant</td>
</tr>
<tr>
<td>l</td>
<td>length of die land</td>
</tr>
<tr>
<td>m</td>
<td>friction factor</td>
</tr>
<tr>
<td>n</td>
<td>number of time intervals</td>
</tr>
<tr>
<td>nₙ</td>
<td>deformation hardening exponent</td>
</tr>
<tr>
<td>nₙ</td>
<td>number of cans</td>
</tr>
<tr>
<td>P</td>
<td>radial force on punch land</td>
</tr>
<tr>
<td>p</td>
<td>surface pressure</td>
</tr>
<tr>
<td>q</td>
<td>normal surface pressure</td>
</tr>
<tr>
<td>qₐₐₙ</td>
<td>pressure on area A</td>
</tr>
<tr>
<td>qₐₐₙ,ₙ</td>
<td>average pressure on area A in time interval n</td>
</tr>
<tr>
<td>R</td>
<td>radius</td>
</tr>
<tr>
<td>R₁</td>
<td>inside radius of the ironing die</td>
</tr>
<tr>
<td>s</td>
<td>sliding length</td>
</tr>
<tr>
<td>sₙₜₐₙ</td>
<td>penalty factor</td>
</tr>
<tr>
<td>t</td>
<td>thickness</td>
</tr>
<tr>
<td>t</td>
<td>time</td>
</tr>
<tr>
<td>t₀</td>
<td>cup wall thickness before ironing</td>
</tr>
<tr>
<td>t₁</td>
<td>cup wall thickness after ironing</td>
</tr>
<tr>
<td>tₙₘₐₓ</td>
<td>maximum can wall thickness</td>
</tr>
<tr>
<td>tₙₘₐₓ,i</td>
<td>maximum can wall thickness in can number i</td>
</tr>
<tr>
<td>tₙₘᵢₙ</td>
<td>minimum can wall thickness</td>
</tr>
<tr>
<td>tₙₘᵢₙ,i</td>
<td>minimum can wall thickness in can number i</td>
</tr>
<tr>
<td>vₐₐₙ</td>
<td>sliding velocity on area A</td>
</tr>
<tr>
<td>vₐₐₙ,ₙ</td>
<td>average sliding velocity on area A in time interval n</td>
</tr>
<tr>
<td>z</td>
<td>distance from can rim</td>
</tr>
<tr>
<td>Z</td>
<td>wear volume</td>
</tr>
<tr>
<td>Zₐₐ</td>
<td>wear volume on area A</td>
</tr>
<tr>
<td>α</td>
<td>angle</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>----------------------------------------</td>
</tr>
<tr>
<td>$\beta$</td>
<td>angle</td>
</tr>
<tr>
<td>$\Delta t$</td>
<td>time interval</td>
</tr>
<tr>
<td>$\Delta t_{ave}$</td>
<td>average wall thickness</td>
</tr>
<tr>
<td>$\varepsilon_{plastic}$</td>
<td>equivalent plastic strain</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>equivalent strain</td>
</tr>
<tr>
<td>$\Theta$</td>
<td>angle</td>
</tr>
<tr>
<td>$\mu$</td>
<td>friction coefficient (Coulomb friction)</td>
</tr>
<tr>
<td>$\mu_{die}$</td>
<td>friction coefficient in the cup – die interface</td>
</tr>
<tr>
<td>$\mu_{punch}$</td>
<td>friction coefficient in the cup – punch interface</td>
</tr>
<tr>
<td>$\rho_{after}$</td>
<td>curvature after spring back</td>
</tr>
<tr>
<td>$\rho_{before}$</td>
<td>curvature before spring back</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>true stress</td>
</tr>
<tr>
<td>$\sigma_{hoop,max}$</td>
<td>maximum hoop stress</td>
</tr>
<tr>
<td>$\sigma_{m}$</td>
<td>hydrostatic stress (mean stress)</td>
</tr>
<tr>
<td>$\sigma_{\text{v.Mises}}$</td>
<td>equivalent v. Mises stress</td>
</tr>
<tr>
<td>$\sigma_y$</td>
<td>yield stress</td>
</tr>
<tr>
<td>$\tau$</td>
<td>friction stress</td>
</tr>
</tbody>
</table>
1 Introduction

The research work described in this thesis constitutes some of the research carried out by the author at the Department of Production, Aalborg University during the period 1992-2005. The thesis is primarily based on some of the author’s research reports, where more detailed descriptions than given in this thesis can be found; these research reports are not part of the thesis but have for reference only been included on the CD, which is enclosed in the thesis. The main results have been published in papers and patent applications of which 14 are included at the end of the thesis (when reference in the thesis is made to one of the included papers the reference is marked with a *, and when reference is made to a research report included on the CD the reference is marked with a **).

The metal forming processes dealt with in this thesis are deep drawing, ironing and backward can extrusion. The tools used in these processes are commonly designed according to guidelines, which have been established through many years practical experience in the metal forming industry. The “red thread” in the thesis is to show that a detailed understanding of the basic process mechanisms makes it possible to improve the processes, e.g. with regard to tool life and to the process robustness and with regard to the quality of the formed parts, and that the improvements, at least in some cases, can be achieved by very minor changes in the tool geometry.

The possibility of obtaining a detailed understanding of the basic process mechanisms has been greatly increased during the last decade due to the development, which has taken place within non-linear FEM and within computers. The greatly improved possibilities for carrying out detailed analyses and obtaining detailed knowledge about the basic process mechanisms in metal forming processes are in the opinion of the author not yet reflected in the commonly used guidelines for tool design.

The main topics of this thesis are:

- **Chapter 4**: An analysis of the residual stresses distribution in a deep drawn cup made in a two stage deep drawing process and an analysis of how the residual stress distribution in the cup wall is affected by a subsequent ironing of the cup wall.
- **Chapter 5**: An analysis of why residual stresses are induced in the deep drawn cup wall and how the residual stresses can be reduced by modifying the shape of the deep drawing die.
- **Chapter 6**: An analysis of how the residual stresses in the cup wall and the shape of the cup wall is influenced by the shape of the draw dies in a two stage deep drawing process.
- **Chapter 7**: An attempt to use FEM-simulations to predict the wear distribution on a deep drawing die.
- **Chapter 9**: An analysis of how the ironing process is influenced by the die design and by the main material and process parameters.
- **Chapter 10**: An analysis of the influence, which a small misalignment between punch and die has in the ironing process and how the influence from such a misalignment can be reduced.
- **Chapter 12**: An analysis of the influence, which a small misalignment between punch and die has in a combined deep drawing and ironing process and how the influence from such misalignment can be reduced.
- **Chapter 13-19**: An analysis of the influence, which a small misalignment of the punch land has in the backward can extrusion process when the punch is designed according to commonly used guidelines (ICFG recommendations) and how the
influence from such a misalignment can be reduced.
The Sheet Metal Forming Group, Department of Production, Aalborg University was the first research group in Denmark to start using a commercial FEM code to analyze and simulate metal forming processes. The FEM code LS-Dyna has since 1991, when The Sheet Metal Forming began licensing LS-Dyna, been the Group’s main FEM code. The Group has also used the FEM code LLNL-Nike2d [1] extensively.

The codes licensed in 1991 were the FEM code LS-Dyna3d [2], the pre-processor LS-Ingrid [3] and the postprocessor LS-Taurus [4]. In 1992 the license was extended to also include the 2D version of LS-Dyna, LS-Dyna2d [5] the pre-processor LS-Maze [6] and the postprocessor LS-Orion [7].

LS-Dyna dates back to Dyna3d [8], a public domain code, which development started in the mid-seventies at Lawrence Livermore National Laboratory (LLNL), University of California, USA. John Hallquist was one of fathers of Dyna3d. In 1989 John Hallquist resigned from LLNL and founded the company Livermore Software Technology Corporation (LSTC), and since 1989 there has been two versions of Dyna, the commercial code LS-Dyna from LSTC and the public domain code LLNL-Dyna from LLNL (public domain up to the end of the mid-nineties, at which time LLNL decided to limit the public access to the code).

LS-Dyna3d and LS-Dyna2d were in 1992 general purpose non-linear FEM codes based on the explicit time integration scheme. Since 1992 LS-Dyna has undergone a tremendous development; LS-Dyna2D and LS-Dyna3D has been merged into a single code named LS-Dyna and LS-Dyna is today a multi purpose – multi physics code making it possible to carry out thermal – fluid – mechanical coupled FEM analyses using an arbitrary combination of Lagrangian, Eulerian, ALE and meshless methods and using either implicit or explicit time integration.

Before 1991 the Sheet Metal Forming Group’s research within metal forming, primarily sheet metal forming, was mainly experimentally based. Since 1991 there has been a gradual shift towards a more extensive use of FEM, and today the metal forming processes are mainly analyzed using FEM and experiments are mainly used to verify the FEM results. This shift has not meant that the importance of the availability of good experimental facilities and the importance of carrying out experiments has become less important. In the opinion of the author the main reason why The Sheet Metal Forming Group has been able to establish itself as an international recognized research group within the analysis and simulation of metal forming processes is that the Group has a solid background in experimental research and from the experiments carried out has a deep knowledge about the processes, which are the subject of analyses and FEM simulations.

The ability and possibility to carry out FEM-simulations has increased very much since 1991 for the following reasons:

- The Sheet Metal Forming Group has gained extensive experience regarding how to make a robust FEM simulation and obtain reliable results.
- The development in LS-Dyna (as mentioned above).
- The development and availability of computer power.

In 1991 the number of elements in a FEM model was limited by the computer power available at that time and by money, because computers were extremely expensive. In Table 1 are listed some of the computers, which the Sheet Metal Group has bought since 1992. From the Table it can be seen that there has been a very big increase in RAM capacity and in disk capacity. Up till 2000 all simulations were carried out on so called “workstations”. To day the simulations are carried out on standard PCs and this has
reduced the price for computer power, RAM and disk capacity dramatically. The bottleneck to day is neither computer power, RAM capacity nor disk capacity, but capacity for backing up all the hard disks.

<table>
<thead>
<tr>
<th>Year</th>
<th>Type</th>
<th>RAM</th>
<th>Disk capacity</th>
<th>Price (DKr)</th>
</tr>
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<tr>
<td>1991</td>
<td>DecStation 200</td>
<td>16 MB</td>
<td>426 MB</td>
<td>107.000</td>
</tr>
<tr>
<td>1992</td>
<td>SGI Indigo</td>
<td>48 MB</td>
<td>512 MB</td>
<td>160.000</td>
</tr>
<tr>
<td>1996</td>
<td>SGI Indigo2</td>
<td>128 MB</td>
<td>4 GB</td>
<td>225.000</td>
</tr>
<tr>
<td>2000</td>
<td>SGI Octane</td>
<td>256 MB</td>
<td>18 GB</td>
<td>200.000</td>
</tr>
<tr>
<td>2002</td>
<td>Beowulf super computer with 10 CPUs</td>
<td>10*512 MB</td>
<td>&gt;200 GB</td>
<td>100.000</td>
</tr>
<tr>
<td>2004</td>
<td>PC with 2 CPUs</td>
<td>2 GB</td>
<td>&gt;250 GB</td>
<td>15.000</td>
</tr>
<tr>
<td>2005</td>
<td>Super computer with 28 CPUs</td>
<td>28*2 GB</td>
<td>&gt;14.000 GB</td>
<td>250.000</td>
</tr>
</tbody>
</table>

Table 1 Some of the computers bought by The Sheet Metal Forming Group since 1991.

The specimen shown in Figure 1 is forged from a round bar and illustrates the development, which has taken place during the last 15 years.

![Figure 1 A rotational symmetric forged specimen. To the left with the specimen made from multi colored plasticine [9] and to the right the equivalent strain distribution obtained from a FEM-simulation.](image)

The forging of this specimen was analyzed in the authors Ph.D.-project [9] in 1977. The internal deformation pattern was analyzed using model material technique and the left half of Figure 1 shows a forged model material specimen. This specimen was made up of different coloured plasticine, which in the plane of symmetry of the undeformed bar formed a pattern consisting of squares. By measuring the deformation of the squares it was possible to determine the equivalent strain distribution in the forged specimen. It was an extremely delicate task to make the model material bar, fairly time consuming to carry out the experiments and very time consuming to measure the deformed squares, but this way of analyzing the material flow in metal forming processes was the state of art up to 1990; with the classical theoretical methods it is not possible to determine accurately the equivalent strain distribution in a complicated forming process as this forging and one had
to rely on very time consuming experiments. To the right is shown the equivalent strain distribution determined from a FEM-simulation, which, with a very fine mesh both in the specimen and the tools and using adaptive remeshing, took approximately 6 min to carry out. The agreement between the experimental results (the shape of the specimen (made from aluminium) during the forging, the force displacement curve, the equivalent strain distribution obtained from the model material specimen) reported in [9] and the results obtained from the corresponding FEM simulation is very good.

When The Sheet Metal Group began using FEM to simulate metal forming processes in 1991, the use of FEM was confined to fairly few universities and to the major car manufacturers, who investigated the potential of using FEM for crash simulations. To day all universities and many metal working companies are using FEM routinely for the simulation of metal forming processes.

LS-Dyna is by no means restricted to the simulation of metal forming processes; the Sheet Metal Forming Group has used LS-Dyna to analyze a variety of processes, not only metal forming processes but also so different processes as plastic injection moulding, injection of insulin in human flesh and the manufacturing and performance of a fishing trawl.

Lately much of the research in the Sheet Metal Forming Group has focused on the combination of FEM simulations and optimization; a combination, which has become possible due to the vast increase in computer power and decrease in price. Combining FEM-simulations with optimization makes it possible not only to analyze a deformation process but also to optimize the forming process, e.g. to find the optimal tool design with regard to the shape of the formed part or to optimize the tool design with regard to the robustness of the forming process [10 - 18,48].

The combination of FEM-simulations and optimization has also been applied very effectively in the determination of friction and material parameters (when used in this regard the combination of FEM-simulations and optimization is usually call inverse modelling) [19-22]. When analyzing a sheet metal forming process the minimum requirements in order to obtain reliable results are that the constitutive model used accurately describes the elasto-plastic behaviour of the blank material and that the friction conditions are modelled accurately. However, with classical testing methods it is only possible to determine the material parameters, e.g. the true stress strain curve and the anisotropic properties, in a small strain range compared to the magnitude of the strains in many sheet metal forming processes; e.g. in a hydroforming process the maximum equivalent strain can easily exceed 1. If uniaxial tension tests are used to determine the material parameters, these can only be determined in the strain range from 0 to n, where n is deformation hardening exponent. For deep drawing quality steel n is typically in the range from 0.2 to 0.25, which means that in a simulation of a hydroforming process, the material behaviour in the strain range beyond n has to be extrapolated from the known behaviour in the strain range from 0 to n; and this drastic extrapolation can give rise to deviation between the real hydroforming process and the FEM simulation.

The idea behind inverse modelling is to carry out an experiment, which resembles the process in question. During the experiment some process characteristics e.g. the punch force–punch displacement curve, are measured accurately. In the FEM model of the experimental process the material parameters are adjusted using an optimization strategy until the best fit between the characteristics measured experimentally and the same characteristics predicted from the FEM simulation is obtained; the material parameters determined are those that yield the best fit.
Explicit versus implicit time integration

With the FEM code LS-Dyna one can chose between implicit and explicit time integration; in the later versions of LS-Dyna it is also possible in the same simulation to switch back and forth between implicit and explicit time integration.

The explicit finite elements methods were originally developed for the analyses of the transient response in highly dynamic non-linear problems of short duration, e.g. the impact of a bullet on an armour plate. The explicit time integration scheme is only conditionally stable and the stability criterion is that the maximum time step size must meet the Courant criterion, which in words states that the maximum time step size must be lower than the time it takes for a stress wave to propagate though the smallest element in the FEM mesh. In problems where the time to simulate is of the order milliseconds the Courant criterion usually does not pose a problem, but when used for the simulation of metal forming processes, which typically takes somewhere from 0.1 sec to a couple of seconds, it is necessary to artificially increase the loading rate or artificially increase the mass in order to keep the number of time steps within an acceptable limit. The simulation of a metal forming process is typically carried out using from 10.000 to 100.000 time steps.

The implicit time integration scheme can be made unconditionally stable and the time step size is primarily chosen on the basis of the desired accuracy (sometimes it is also necessary to limit the time step size to achieve convergence). When implicit integration is used, the simulation of metal forming process is typically carried out using from 100 to 1000 time steps.

There are advantages and disadvantages with both explicit and implicit time integration.

The main advantages using explicit time integration are:
- Strong non-linearity (both geometric and material non-linearity and non-linearity due to contact) is handled very efficiently and usually does not cause convergence problems.
- Little storage capacity is needed. Simulations with many degrees of freedom can be carried out on a standard PC.

The main disadvantages using explicit time integration are:
- The many time steps which are required may lead to an unacceptable CPU time.
- It may be necessary to use scaling (either increased loading rate or mass scaling) in order to maintain an acceptable time steps size, and scaling may introduce noise, e.g. in the contact forces.
- When carrying out a thermal mechanical coupled simulation and using load or mass scaling, it is necessary to scale the thermal parameters in order to obtain the correct thermal response.
- The maximum time step size is controlled by the smallest element in the FEM mesh. It is thus important that the FEM mesh is homogeneous (without a few very small elements, which drive the time step size down)
- When the goal with the simulation is to achieve the quasi-static response one has to be very careful that the dynamics does not cause “unphysical” straining of the specimen.
- The solid elements have only one integration point and this may cause hourglass problems.
The main advantages using implicit time integration are:

- The 2D elements can be chosen to have four integration points. This increases the accuracy and decreases the tendency to hour glassing of the elements.
- Any unwanted dynamic effects are completely avoided if the simulation is carried out as static.
- The simulation can be carried out in “real time”, which makes it easy to carry out a thermal mechanical coupled analysis (there is no need for load or mass scaling and no need for scaling of the thermal properties).
- The smallest element does not control the time step size, which means that areas in the FEM model, e.g. a corner on a forming tool, can be very finely meshed without having an effect on the time step size (as long as the fine mesh does not create convergence problems).

The main disadvantages using implicit time integration are:

- With implicit time integration it can sometimes be impossible to achieve a converged solution and the simulation stops.
- Simulations with many degrees of freedom require large storage capacity.

When simulating complex 3D forming processes with many degrees of freedom, the storage capacity requirement becomes so large that the simulation cannot be carried out on a standard PC. When carrying out a 2D metal forming simulation, the number of elements is usually limited, typically less than 20,000 elements, and if implicit time integration is used, it is convergence or rather the lack of convergence, which gives rise to most of the problems. Unfortunately it is not always possible to foresee if convergence problems will be encountered in a given simulation. The author’s experience is that sometimes convergence problems occur in a simulation of a seemingly simple process whereas the simulation of a fairly complex process sometimes runs through without any convergence problems.

With LS-Dyna it is very easy to switch a simulation from implicit time integration to explicit time integration and visa versa. This can be very useful if a simulation is carried out using explicit time integration and using load and/or mass scaling. By carrying out the same simulation using explicit and implicit time integration and comparing the results it is very easy to check if the load and/or mass scaling applied when using explicit time integration has a significant influence on the results obtained. If this is the case the load and/or mass scaling must be reduced (provided of course that the goal of the simulation is to obtain the quasi static response and not the dynamic response).

The author has proposed to John Hallquist, LSTC [23], to make it possible in LS-Dyna to combine implicit and explicit time integration in a simulation in such a way that the elasto-plastic deformation of the specimen and the contact between specimen and tool parts is determined using explicit time integration whereas the stresses and strains in the tool parts are determined using implicit time integration. Such a combination could be very efficient because explicit time integration is very well suited to handle the strong non linearity associated with elasto-plastic deformation and with contact whereas implicit time integration is very well suited to handle the elastic deformation of the tools (the elements in the tools can be made very small without affected the time step size and assuming that the tools only deform elastically, the stiffness matrix for the tool parts has only to be established once in a simulation). John Hallquist was of the opinion that it might be possible to implement such an option in LS-Dyna.
3 Residual stresses: a brief introduction

It is well known that residual stresses present in a component may have a significant influence on e.g. the static and dynamic strength and on the corrosion resistance. In some cases the residual stresses can improve the material properties. This is e.g. utilized in the shot peening process, a process which introduces compressive stresses in the surface of the shot peened component and hereby increases the fatigue strength. In other processes the residual stresses are regarded to have a negative effect. This is e.g. the case in welding, where the residual stresses can cause cracking and where it may be necessary to heat treat the part after welding to remove the residual stresses. It is also the residual stresses, which with welding causes warping, a problem that may have a significant influence on the total welding cost; when welding a ship the problems with warping are claimed to account for 30% of the total cost of welding the ship. In sheet metal forming it is the residual stresses, which gives rise to many of the production problems: the stresses causes spring back after stamping, and this spring back must be accounted for when designing the stamping tools. Another important issue when forming sheet metal parts is that variations in the material properties or in the sheet thickness give rise to variations in the spring back. Residual stresses can also give rise to stress corrosion cracking, which according to [24] amounts to about 25% of all cases of corrosion failure in the chemical industry.

When manufacturing a part the manufacturing process induces residual stresses in the finished formed part, but despite the importance of the residual stresses it is very rare that the residual stresses in the finished part are taken into account. An oversimplified picture of the situation at the French automaker Renault in the year 1992 is shown in Figure 2; a wall is separating the materials laboratory and the product engineering department and hardly any information regarding residual stresses passes the wall. In the materials laboratory there is lot of know how about residual stresses, which influence the residual stresses have, a number of methods are available for the experimental determination of residual stresses e.g. X-ray and neutron diffraction, hole drilling ect.. On the other side of the wall is the product engineering department where residual stresses are known but are rarely taken into account. The main reasons why residual stresses are rarely taken into account when planning a production (besides from when it is known that residual stresses may cause premature fracture if they are not removed by heat treatment) are according to [25]:

- It requires a lot of experimental work in order to specify the residual stresses.
- It is very difficult and time consuming to verify that a part fulfils the specified requirements regarding residual stresses.
- Simple software for predicting the residual stresses and for predicting the behaviour of the component taking into account the residual stresses does not exist.

Figure 2 is more than ten years old but the situation with regard to the first two reasons has not changed very much during the last ten years. With regard to the last reason things have changed, mainly due to the development of advanced FEM codes and to the increase in computer power. Although the determination of residual stresses requires the use of an advanced non-linear FEM code, the code may be simple to use and FEM simulations are today used routinely in the automobile industry to predict spring back in stamped parts when designing stamping tools and the automobile industry is also beginning to take into account the residual stresses in the formed parts when carrying out crash simulations.
Residual stresses in a formed sheet metal part are caused by inhomogeneous deformation. The residual stresses are in equilibrium over a certain volume of material and can be classified according to the extension of this volume as follows:

- **Microscopic residual stresses**: the extension of the volume is of the size of the grain size of the material or even smaller.
- **Macroscopic residual stresses**: the extension of the volume is large compared to the grain size of the material.

The minimum element size in a FEM model used to simulate a sheet metal forming process is many times larger than the grain size of the material and the FEM simulation can thus only be used to predict the macroscopic residual stresses. From a practical point of view this is not so severe a limitation because according to [26] it is the macroscopic residual stresses that are the most important with regard to spring back and/or failure.

The problem, which inspired the author to work with residual stresses, is the cup shown in Figure 3. In the upper part of the cup there are a number of cracks caused by residual stresses. Right after the deep drawing the cup did not have any cracks, but after a couple of minutes the first crack snapped open and after 24 hours the cup looked as shown in the Figure. If the cup had been annealed right after the deep drawing the cracks could have been avoided. However, the company the author was collaborating with would like to avoid an annealing process; annealing would lead to increased production cost, an inexpedient production flow and could give rise to problems with maintaining close tolerances on the part due to distortion during the annealing process.

Figure 2 An oversimplified view of how residual stresses are treated in car manufacturing company in the year 1992 [25].
The kind of cracking shown in Figure 3 is often denoted delayed cracking. It is also sometimes denoted corrosion cracking. The best term is in the opinion of the author delayed cracking because the cracks develop without any corroding media, e.g. chloride, present. It is known that with meta-stable austenitic stainless steel the tendency to delayed cracking is heavily influenced by the hydrogen content in the stainless steel blank [27].

It is well known that residual stresses in a formed sheet metal part are caused by inhomogeneous deformation. It is also well known that residual stresses can be removed or at least reduced significantly by a homogeneous deformation. This approach is e.g. used to remove residual stresses from an extruded aluminium profile by subjecting the profile to a small uniform elongation. The author has investigated the possibility of reducing the residual stresses in a deep drawn cup by ensuring that the last deformation mode, which a material element in the cup wall becomes subjected to, is nearly homogeneous. Two approaches have been investigated; the first approach is to iron the cup wall (summarized in Chapter 4) and second approach to modify the profile of the deep drawing die (summarized in Chapter 5 and 6). Both approaches were analyzed using FEM; to the knowledge of author, the author was among the first to try to use FEM to analyze and predict residual stresses in formed sheet metal parts.

Figure 3 Deep drawn cup made from meta-stable austenitic stainless steel. A number of cracks are after the deep drawing formed in the upper part of the cup due to residual stresses in the cup wall.
4 The use of ironing to reduce the residual stresses in the cup wall

The work summarized in this Chapter is mainly based on [28**]. Some of the results have been published in [29*,30,31]. The purpose of the investigation was to analyze the residual stress distribution in a deep drawn cup made in a two stage deep drawing process and to analyze to what degree subsequent ironing of the cup wall affected the residual stress distribution in the ironed cup wall. It is well known that ironing can cause a reduction of the residual stresses, see. e.g. [32,33], but the analysis carried out had hitherto mainly been experimentally based. To the knowledge of the author the work summarized in this Chapter is among the first attempts to used FEM to analyze the residual stresses in a deep drawn cup and in an ironed cup wall.

4.1 The FEM model

Figure 4 shows the FEM model used and in Table 2 are listed the main tool dimensions and material parameters used in the simulations.

![Figure 4 The FEM model used in the simulation of the two stage deep drawing process followed by ironing.](image-url)
Main tool dimensions

<table>
<thead>
<tr>
<th></th>
<th>first draw</th>
<th>second draw</th>
<th>ironing</th>
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<tbody>
<tr>
<td>punch diameter</td>
<td>ø99.61 mm</td>
<td>ø83.23 mm</td>
<td>ø83.23 mm</td>
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<td>punch nose radius</td>
<td>7 mm</td>
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<td>2.41 mm</td>
</tr>
<tr>
<td>die profile radius</td>
<td>8.5 mm</td>
<td>7 mm</td>
<td></td>
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<tr>
<td>die clearance</td>
<td>1.9 mm</td>
<td>1.9 mm</td>
<td>1.595 mm</td>
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Main blank parameters

<p>| |</p>
<table>
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<tbody>
<tr>
<td>blank material</td>
</tr>
<tr>
<td>blank diameter</td>
</tr>
<tr>
<td>blank thickness</td>
</tr>
<tr>
<td>Young’s modulus</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
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<tr>
<td>true stress – strain curve</td>
</tr>
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</table>

Friction coefficient $\mu$ (Coulomb friction)

<p>| | |</p>
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<tr>
<td>between blank and punches</td>
<td>0.1</td>
</tr>
<tr>
<td>between blank and draw dies</td>
<td>0.08</td>
</tr>
<tr>
<td>between blank and blank holder</td>
<td>0.08</td>
</tr>
<tr>
<td>between blank and ironing die</td>
<td>0.08</td>
</tr>
</tbody>
</table>

Table 2 Main data used in the FEM-simulations

The results described in [28**] were obtained from 2D simulations assuming rotational symmetry and with the blank modelled with 5*125 elements. The FEM code used was LS-Dyna2D using explicit time integration. The results presented in this Chapter were obtained from similar simulations carried out in 2004 using LS-Dyna (version 970 double precision) with the blank modelled with 10*400 elements and using implicit time integration. In 1993 when the original simulations were made, it was not practically possible with the computers available at that time to carry out the simulations implicitly.

4.2 FEM-results

Figure 5 shows the vertical stress distribution and the hoop stress distribution in the cup wall after redrawing draw and after ironing. After redrawing there are, in the lower $\frac{3}{4}$ of the cup wall, large compressive vertical stresses (in the range from -1600 to -500 MPa) towards the inside of the cup wall and large tensile vertical stresses (in the range from 600 to 800 MPa) towards the outside. In the same region the hoop stresses are also compressive towards the inside (in the range from -900 to -300 MPa) and tensile (in the range from 300 to 700 MPa) towards the outside. The ironing has as can be seen caused a drastic change in the residual stresses; both the vertical stress distribution and the hoop stress distribution is much more even, the though the thickness variation is small and the maximum values of the residual stresses are significantly lower than in the cup wall after redrawing.

Figure 6 shows the hydrostatic pressure distribution and the v.Mises stress distribution in the cup wall after redrawing and after ironing. After redrawing the hydrostatic pressure is in the lower $\frac{3}{4}$ of the cup wall tensile towards the inside (in the range from 400 to 800
MPa) and compressive towards the outside (in the range from -600 to -400 MPa). In middle height the \( v.\text{Mises} \) stress is very large towards the inside (in the range from 1200 to 1400 MPa) whereas the \( v.\text{Mises} \) stress is lower towards the outside (in the range from 500 to 600 MPa). It can be seen that the ironing causes a drastic change in the hydrostatic pressure distribution and the \( v.\text{Mises} \) stress distribution; after ironing the stress distributions are much more even, the variation though the thickness is small and the maximum values of the hydrostatic pressure and the \( v.\text{Mises} \) stress are significantly lower than after the redrawing.

Figure 7 shows the vertical stress and the hoop stress in ten element through the thickness, laying in 20%, 50% and 80% cup height (0% = cup bottom, 100% = cup rim) after redrawing and after ironing. It is apparent from the plots that according to the FEM results, the ironing has caused a significant reduction both in the vertical stresses and in the hoop stresses.

The residual hoop stresses in the cup wall after redrawing and after ironing were also determined experimentally by Dirks and Gradman [34] using the machining method [35]. The procedure using the machining method is to cut a ring from the drawn cup and then slit the ring open. By measuring the curvature of the ring before and after slitting the maximum residual hoop stress can be estimated from the Equation

\[
\sigma_{\text{hoop,max}} = E \frac{t}{2} (\rho_{\text{before}} - \rho_{\text{after}})
\]

where \( E \) is Young’s modulus of the blank material, \( t \) the cup wall thickness (thickness of the ring), and \( \rho_{\text{before}} \) and \( \rho_{\text{after}} \) is the curvature of the ring before and after slitting, respectively. The larger the opening of the ring is the smaller is the curvature \( \rho_{\text{after}} \). Equation (1) is based on the assumptions that the hoop stress distribution through the thickness is linear and symmetric around the centre of the cup wall. In Figure 7 are also plotted the experimentally determined hoop stress distributions. Despite the fact that the hoop distribution through the thickness after redrawing is not linear Equation (1) gives a fairly good estimation of the maximum hoop stress, especially in 50% and 80% cup height. After ironing the maximum hoop stress is, both determined experimentally and predicted by FEM, small.

In Figure 8 are plotted corresponding values of the mean stress and the \( v.\text{Mises} \) stress in all the elements in the cup wall after redrawing and ironing. It is obvious that the ironing has caused a significant reduction in both the \( v.\text{Mises} \) stress and in the mean stress. The lower the \( v.\text{Mises} \) stress is and the lower the mean stress is, the higher is the fatigue life or fatigue strength. According to the FEM results, the ironed cup will thus have significantly higher fatigue life or fatigue strength compared to the redrawn cup.

The residual stress distributions in the redrawn cup wall shown in this Chapter are in good agreement with the distributions shown in [28**,29*], but the residual stress distributions in the ironed cup wall are somewhat different although they all show a significant reduction of the residual stresses. The difference in number of elements used (5*125 in the blank in original simulations versus 10*400 used in the simulations presented in this Chapter) will of course give rise to slightly different results, but the main reason for the difference is believed to be due to the modelling of the contact between punch and blank and between blank and ironing die. The contact algorithm used in both the simulations carried out with LS-Dyna2d and with LS-Dyna is penalty based. With a penalty based contact formulation any node that penetrates through a contacting surface causes a linear spring to be inserted into the stiffness matrix that couples the penetrating node to two adjacent nodes on the contact surface. How much a node penetrates into a contact surface can be controlled by adjusting a penalty factor; the larger the penalty factor the smaller the node penetration. The best choice of penalty factor is a compromise;
if the penalty factor is made too small the node penetration may become too large and if the penalty factor is too large convergence problems may be encountered. Simulations carried out show that the residual stress distribution after redrawing is only marginally affected by the choice of the penalty factor whereas the residual stress distribution in the ironed cup wall is heavily influenced by the choice of the penalty factor. One should thus be very careful when evaluating residual stress distributions in processes/situations where even a slight node penetration may have a significant influence on the predicted residual stresses.

The author presented some of the results at a Danish metal conference back in 1994. Inspired by this presentation a Danish company very successfully applied ironing to reduce the residual stresses in a component, which had given problems with regard to the fatigue life.

4.3 Conclusions

The FEM simulations discussed here show that after redrawing there are very high compressive residual stresses towards the inside of the cup wall and very high tensile residual stresses towards the outside.

The maximum hoop stress determined experimentally using the machining method is in fairly good agreement with the maximum hoop stress determined from the FEM simulations despite the fact that the hoop stress distribution according to the FEM results is not linear.

The FEM simulations and the experimental results show that the ironing causes a significant reduction of the residual stresses in the cup wall.

According to the FEM results, the ironed cup will have significantly higher fatigue life or fatigue strength compared to the redrawn cup.

The residual stress distribution in the ironed cup wall determined from a FEM simulation is heavily influenced by the choice of the penalty factor when a penalty based contact algorithm is used to model the contact between cup wall and punch and cup wall and die. In processes/situations where even a minor node penetration may have a significant influence on the residual stress distribution one should be very careful in the evaluation of the FEM results.
Figure 5 The vertical stress distribution and the hoop stress distribution in the cup wall after redrawing (left) and after ironing (right). The cup wall is scaled *16 in horizontal direction.
Figure 6 The hydrostatic pressure distribution and the v.Mises stress distribution in the cup wall after redrawing (left) and after ironing (right). The cup wall is scaled *16 in horizontal direction.
Figure 7 The residual vertical stress and hoop stress in 10 elements lying through the thickness in 20%, 50% and 80% cup height (100% = cup rim). Also shown is the experimentally determined hoop stress distribution using Equation 1.
Analysis of deep drawing, ironing and backward can extrusion

Figure 8 Corresponding values of the mean stress $\sigma_m$ and the v.Mises stress $\sigma_{v,Mises}$ in all the elements in the cup wall after redrawing and after ironing.
5 Reducing the residual stresses by modifying the draw die profile

The residual stresses in the finished drawn cup can be so large that they can cause cracks to be formed in the cup wall as shown in Figure 3. In order to understand why and where the residual stresses develop during the deep drawing, the deep drawing of a cylindrical cup was analyzed using 2D rotational symmetric FEM simulations. The work summarized in this Chapter is mainly based on [36**]. Some of the results obtained have been published in [37*,38*,39*,40].

Figure 9 Sketch of the tools used for the deep drawing of cylindrical cups. The draw die is commonly made with a circular profile.

5.1 Analysis of what causes the residual stresses

Figure 9 shows a sketch of the tools used in conventional deep drawing of a cylindrical cup. The die profile is commonly made with a circular profile as shown in the sketch. The deep drawing of a cylindrical cup was analysed using 2D rotational symmetric FEM simulations. The main process and materials parameters used in the simulations are listed in Table 3.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>Punch diameter</td>
<td>ø46 mm</td>
</tr>
<tr>
<td>Punch nose radius</td>
<td>R6 mm</td>
</tr>
<tr>
<td>Die profile radius</td>
<td>R7 mm</td>
</tr>
<tr>
<td>Inside diameter of die</td>
<td>ø50 mm</td>
</tr>
<tr>
<td>Inside diameter of blank holder</td>
<td>ø50 mm</td>
</tr>
<tr>
<td>Blank diameter</td>
<td>ø92 mm</td>
</tr>
<tr>
<td>Blank thickness</td>
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<tr>
<td>True stress-strain curve for blank material</td>
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<tr>
<td>Blank holder pressure</td>
<td>0.825 MPa</td>
</tr>
<tr>
<td>Friction coefficient in all interfaces</td>
<td>$\mu = 0.1$</td>
</tr>
</tbody>
</table>

Table 3 The main process and material parameters used in the FEM simulations of the deep drawing of a cylindrical cup.
The Figures in the following are not from the original simulations, which were carried out using the FEM code LS-Dyna2d and using explicit time integration, but from similar simulations carried in 2004 out using LS-Dyna (version 970, double precision), using implicit time integration and an increased number of elements in the blank (300*10 elements versus 125*5 elements in the original simulations). Figure 10 shows the FEM model used.

![Figure 10 FEM model of the cylindrical deep drawing with a conventional circular profiled die.](image)

Figure 11 shows the radial stresses in the cup in the plane of the sheet during the deep drawing. When a material element reaches the entrance to the draw die profile, the element is bent. This bending causes an inhomogeneous through the thickness stress distribution with tensile radial stresses towards the outside and compressive radial stresses towards the inside (towards the draw die). When the element is drawn further towards the punch (slides down the draw die profile) the element is being deformed nearly homogeneously and the material “forgets” the inhomogeneous deformation caused by the bending at the die profile entrance. When the material element reaches the exit of the die profile the element is instantaneously unbent and this unbending causes a very inhomogeneous through the thickness stress distribution with large compressive radial stresses towards the inside of the cup wall and large radial tensile stresses towards the outside of the cup wall.

Figure 12 shows the vertical stresses in the cup wall of the finished drawn cup. In the middle height there are very large compressive residual stresses towards the inside of the cup wall and very large tensile residual stresses towards the outside. In other words the inhomogeneous stress distribution, which is cause by the unbending when the material reaches the exit of the die profile, remains in the finished drawn cup.

The 2D FEM simulations thus show that the residual stresses in the finished drawn cup are mainly due to the unbending, which the blank becomes subjected to when passing the exit of the die profile. This is in agreement with results obtained by Saito and Shimahashi [41] who determined the residual stresses in a cylindrical deep drawn cup experimentally and using classical theory of plasticity.
Figure 11 The radial stress (in the plane of the sheet) in the cup during deep drawing.

Figure 12 The vertical stresses in the cup wall of the finished drawn cup. The cup wall has been scaled 9 times in horizontal direction.
5.2 Modification of the draw die profile to reduce the residual stresses

If it is correct that the primary reason for the residual stresses in the wall of a deep drawn cup is due to the unbending, to which the material becomes subjected when reaching the exit of the draw die profile, an obvious way to reduce the residual stresses is to avoid the unbending. This is of course not possible; the material must be unbend in someway or another, but it is possible to make the unbending take place gradually instead of instantaneously. If the die profile has the shape of a tractrix as shown in Figure 13 the unbending will take place gradually and at the same time the cup wall is drawn towards the punch. The possibility of reducing the residual stresses by making the draw die with a tractrix shaped profile was analyzed using FEM-simulations. The FEM model employed is shown in Figure 13. Figure 14 shows the radial stresses in the plane of the sheet in the cup during the drawing when the drawing is carried out with the tractrix shaped die profile. When a material element reaches the entrance to the die profile the element is bent instantaneously causing an inhomogeneous through the thickness stress distribution with compressive radial stresses towards the die profile and tensile radial stresses on the other side. When the material element starts to slide down the die profile the material element becomes unbent causing an inhomogeneous through the thickness stress distribution with radial compressive stresses towards the inside of the cup wall and tensile radial stresses towards the outside. After unbending the material element is drawn towards the punch as it slides down the tractrix shaped profile and the material element more or less “forgets” the inhomogeneous stress distribution caused by the unbending.

![Figure 13 FEM model of the cylindrical deep drawing carried out with a die with a tractrix shaped die profile.](image-url)
5.3 Comparison of the stress distributions in the cup wall

Figure 15 shows the vertical stresses and the hoop stresses and Figure 16 the pressure and the v. Mises stresses in the wall of the finished drawn cup drawn with the die having a circular shaped die profile (to the left) and having a tractrix shaped die profile (to the right). It is obvious from these Figures that the shape of the die profile has a significant influence on both the shape of the cup wall and on the residual stresses. With the circular profiled die, the vertical stresses and the hoop stresses, Figure 15, are compressive towards the inside and tensile towards the outside and the magnitude of the stresses are large. With the tractrix shaped die profile the vertical stresses are also compressive towards the inside and tensile towards the outside, but the magnitude of the stresses is significantly smaller than in the cup made with the circular profiled die and there is hardly any residual hoop stresses in the cup wall. In the cup made with the circular profiled die, Figure 16, there is a high positive hydrostatic pressure towards the inside and high negative pressure towards the inside. With the tractrix shaped die profile there is also a positive hydrostatic pressure towards the inside and a negative towards the outside but the magnitude of the pressures is significantly lower. The v. Mises stress in the cup made with the circular profiled die is very high towards the inside in the middle height of the cup wall. With the tractrix shaped die profile the magnitude of the v. Mises stresses in the cup wall is reduced significantly and the variation is much smoother compared to when the drawing is carried out with the circular profiled die.
Figure 15 The vertical stress distribution and the hoop stress distribution in the cup wall (scaled 9* in horizontal direction). The cup to the left is made with the die having the circular die profile and the cup to the right with the die having a tractrix shaped die profile.
Figure 16 The pressure distribution and the v.Mises stress distribution in the cup wall (scaled 9* in horizontal direction). The cup to the left is made with die having the circular die profile and the cup to the right with the die having the tractrix shaped die profile.
Figure 17 The radial stress (x-stress) and the hoop stress (z-stress) in ten elements lying initially 36.5 mm from the centre as function of simulation time when the drawing is carried out with circular profiled die. Element A (2919) is facing the blank holder and element J (219) the draw die.
Figure 17 shows the development of the radial stress (in the plane of the sheet) and of the hoop stress as function of the simulation time in ten elements laying initially 36.5 mm from the centre of the blank, when the drawing is carried out with the circular profiled die. In the finished drawn cup these ten elements are located approximately in middle height of the cup wall. From Figure 17 it can be seen that a very inhomogeneous stress distribution develops when the elements reach the entrance to the die profile and are bent. As the elements are drawn down the circular die profile towards the punch the stress distribution becomes nearly homogeneous but becomes again very inhomogeneous when the elements reach the exit of the die profile and start to become unbent. The inhomogeneous stress distribution remains in the cup wall also after the cup has been drawn completely trough the draw die at $t \approx 93$.

In Figure 18 are plotted the residual radial stress and the residual hoop stress in these ten elements in the finished drawn cup and the significant variation though the thickness is apparent. There are large tensile residual stresses towards the outside and large compressive stresses towards the inside. The maximum difference in the residual radial stresses is approximately 1800 MPa and approximately 1200 MPa in the residual hoop stresses. The through thickness residual stress distributions are in good qualitative agreement with the distributions shown in Figure 19 determined theoretically and experimentally by Saito and Shimahashi [41].

Figure 20 shows the development of the radial stress (in the plane of the sheet) and the hoop stress as function of the simulation time in the same ten elements when the drawing is carried out with the tractrix shaped die profile. When the elements reach the inlet to the tractrix shaped die profile at $t \approx 23$ they are bent and the bending introduces an inhomogeneous through the thickness stress distribution. Right after the bending the elements start to become unbent again and the unbending (at $t \approx 33$) completely reverses the inhomogeneous stress distribution caused by the bending. As the elements slide down the die profile and at the same time are drawn towards the punch an inhomogeneous
through the thickness distribution develops. The inhomogeneous stress distribution remains fairly constant until the elements reach the draw die exit at $t \approx 80$ at which time the elastic spring back starts to cause a significant change in the residual stress distributions up till the cup is drawn completely through the draw die at $t \approx 95$. When the cup is drawn completely through the die there is a slight change in the stresses, especially in the hoop stresses, due to elastic spring back. In Figure 21 are plotted the residual radial stress distribution and the residual hoop stress distribution in the ten elements in the finished drawn cup; also plotted are the residual stresses when the drawing is carried out with the circular profiled die. As with the circular profiled die the residual stresses are tensile towards the outside and compressive towards the inside, but the magnitude of the stresses is significantly lower; the maximum difference in the residual radial stresses is approximately 700 MPa (1800 MPa with the circular profiled die) and the maximum difference in the hoop stresses approximately 140 MPa (1200 MPa with the circular profiled die). It can also be seen that the variation through the thickness, contrary to when the circular profiled die is employed, is nearly linear.

\[\sigma_r^*\] and \[\sigma_t^*\] determined theoretically and experimentally by Saito and Shimahashi [41].

Figure 19 The residual radial stresses ($\sigma_r^*$) and hoop stresses ($\sigma_t^*$) determined theoretically and experimentally by Saito and Shimahashi [41].
Figure 20 The radial stress (x-stress) and the hoop stress (z-stress) in ten elements lying initially 36.5 mm from the centre of the blank as function of simulation time when the drawing is carried out with the tractrix profiled die. Element A (2919) is facing the blank holder and element J (219) the draw die.
In Figure 23 are plotted corresponding values of the mean stress $\sigma_{\text{mean}}$ and the v. Mises stress $\sigma_{\text{v,Mises}}$ in all the elements in the finished drawn cup. From the plot it is apparent that the die profile has a significant influence on both the mean stress ($= -$ pressure) and the v. Mises stress (this is of course also apparent from Figure 16; the data used in the plot are the same as used in Figure 16); the maximum v. Mises stresses are significantly lower in the cup made with the tractrix shaped die profile than in the cup made with the circular profiled die and so is both the maximum and minimum mean stress. In service when a part becomes subjected to an external load, the stresses (assuming that the external load only gives rise to elastic deformation of the part) caused by the external load are added to the residual stresses. The equivalent strain and thus the yield stress in the cup wall as function of distance from the bottom is nearly the same in the two cups independently of the draw die profile used, and it is thus apparent from Figure 23 that the cup made with the tractrix shaped die profile can sustain a higher external load before the cup becomes deformed plastically. With regard to fatigue strength the position of the mean stress and the v. Mises stress is important; the lower the v. Mises stress is and the lower the mean stress is the higher is the fatigue strength and it is thus also apparent that the cup made with the tractrix shaped die profile can sustain a higher fatigue load and/or has a higher fatigue life than the cup made with the circular profiled die.

### 5.4 Shape of the cup wall

The shape of the cup wall of the cup drawn with the circular profiled die is significantly different from the shape when the cup is drawn with the tractrix profiled die. This is apparent from Figure 15 and Figure 22, the latter Figure showing the cup wall scaled in horizontal direction and overlayed a grid. With the circular profiled die the outside cup wall is barrel shaped (the barrel shape is typical when a circular profiled die is employed and the drawing carried out with a fairly large gap between the punch and the die). With the tractrix shaped die profile the outside cup wall is nearly straight and has
thus a significantly better cylindricity compared to the cup drawn with the circular profiled die.

Another interesting thing, which can be observed, is that the outside radius of the cup drawn with the circular profiled die is, besides from the very rim of the cup, smaller than the inside radius of the draw die (= 25 mm) and that the outside radius of the cup drawn

![Graph showing stress values](image)

**Figure 23** Corresponding values of the mean stress $\sigma_m$ and v. Mises stress $\sigma_{vMises}$ in all the elements in the finished drawn cup, drawn with the circular profiled die and drawn with the tractrix profiled die.

![Cup wall shapes](image)

**Figure 22** Shape of the cup wall. To the left the cup drawn with the circular profiled die and to the right with the tractrix profiled die. The cup wall has been scaled in horizontal direction.
with the tractrix shaped die profile is larger than the inside radius of the draw die (= 25 mm).

5.5 Experiments

The deep drawings simulated were also analyzed experimentally. The tool and the blank dimensions used in the experiments were the same as those used in the simulations and the material data used in the simulations were determined from uniaxial tension test carried out with test specimens cut from the material used in the experiments.

The dimensions of the drawn cups were measured using a 3D coordinate measuring machine. The roundness of the outside cup wall was measured in 23 mm cup height and the inside and outside radius of the cup wall was measured in 0°, 45° and 90° in relation to the rolling direction of the blank.

![Figure 24 Shape of the cup wall measured in 0°, 45° and 90° to the rolling direction and the shape determined from the FEM-simulations. Upper Figure is for circular die profile and lower Figure is for the tractrix shaped die profile.](image)
Figure 24 shows the measured inside and outside radius of the cup wall as function of the distance from the cup bottom and with the angle to the rolling direction as parameter. Also shown are the radii determined from the FEM simulations. It can be seen that there is fairly good agreement between the experimental results and the FEM results for both the circular die profile and the tractrix shaped die profile.

Figure 25 shows the inside and outside radius of the cup wall measured in 23 mm cup height as function of the angle to the rolling direction. It can be seen that the cup produced with the tractrix shaped die profile has a significantly better roundness than the cup produced with the circular profiled die.

The residual stresses were not explicitly measured experimentally, but to get an idea of the magnitude of the residual hoop stresses rings were cut from the drawn cups 18 mm and 31 mm from the cup bottom. After cutting the rings were slit open and it was measured how much each ring opened up due to residual hoop stresses. The average results for how much the rings opened are given in Table 4.

<table>
<thead>
<tr>
<th></th>
<th>Circular profiled die</th>
<th>Tractrix shaped die</th>
</tr>
</thead>
<tbody>
<tr>
<td>18 mm cup height</td>
<td>17.37 mm</td>
<td>1.65 mm</td>
</tr>
<tr>
<td>31 mm cup height</td>
<td>13.70 mm</td>
<td>4.97 mm</td>
</tr>
</tbody>
</table>

*Table 4 Opening of the rings cut from the cups.*

The experimental results given in Table 4 show that the opening up of the rings (and thus the maximum hoop stress) is significantly larger in the cup drawn with the circular profiled die than in the cup drawn with the tractrix profiled die. This is in agreement with
the residual hoop stresses distributions determined from the FEM simulations, see Figure 21.

5.6 Conclusions

Both the FEM simulations and the experimental results show that the shape of the die profile has a significant influence on both the shape of the cup wall and the residual stresses in the cup wall.

With the circular profiled die the final plastic deformation mode is unbending, which causes a very inhomogeneous stress distribution through the thickness, and this inhomogeneous stress distribution remains to a large extent in the finished drawn cup wall. With the tractrix shaped die profile, the final plastic deformation mode is mainly a reduction of the radius of the cup wall. This deformation mode is nearly homogeneous. Both the FEM simulations and the experimental results show that the residual stresses are significantly lower in the cup produced with the tractrix shaped die profile than in the cup produced with the circular profiled die. The results thus support the hypothesis that a significant reduction of the residual stresses can be achieved if the final deformation mode in the drawing is nearly homogeneous.

Both the FEM simulations and the experimental results show that a significant improvement in the cup quality can be achieved with the tractrix shaped die: the outside cup wall is nearly straight compared to barrel shaped when using the circular profiled die and the roundness of the outside cup wall is also significantly improved.

The main disadvantage of using a tractrix shaped die profile is that the punch stroke is increased.
6 The application of a tractrix shaped die profile in a two stage deep drawing process

A two stage deep drawing process using different combinations of die profiles were investigated using FEM simulations (using LS-DYNA2d and explicit time integration) and experimentally. Some of the results summarized in this Chapter have been published in [39*,42*]

6.1 Geometry of the tools used

In Figure 26 are shown the deep drawing tools used.

![Figure 26 The tools used in the first (left) and second stage (right). The draw die on the left half is the conventional die with a circular die profile and on the right half with a tractrix shaped die profile.](image)

The material used was 1 mm thick meta-stable austenitic stainless steel (AISI 304) with a blank diameter of ø84.96 mm giving a drawing ratio of 1.8 in the first stage. The drawing ratio in the second stage was 1.25 and the total drawing ratio 2.25.

The following combinations of draw die profiles were investigated:

<table>
<thead>
<tr>
<th>First stage</th>
<th>Second stage</th>
</tr>
</thead>
<tbody>
<tr>
<td>circular profiled die</td>
<td>circular profiled die</td>
</tr>
<tr>
<td>tractrix shaped profiled die</td>
<td>tractrix shaped profiled die</td>
</tr>
<tr>
<td>circular profiled die</td>
<td>tractrix shaped profiled die</td>
</tr>
</tbody>
</table>

The blank material was in the FEM simulations modelled as elasto-plastic with the true stress-true plastic strain curve given by $\sigma = 1393(0.0002 + \varepsilon_{\text{plastic}})^{0.43} \text{ MPa}$ and with 125 elements in radial direction and 5 elements through the thickness. The tool parts were modelled as elastic with Young’s modulus $E = 210.000 \text{ MPa}$. Coulomb friction was assumed in all interfaces with the friction coefficient $\mu = 0.1$. 

The cups drawn experimentally were measured on a 3D coordinate measuring machine.

**Figure 27** The inside and outside radius of the cup wall after the first stage measured in $0^\circ$, $45^\circ$ and $90^\circ$ to the rolling direction and the results obtained from FEM simulations, when the draw was carried out with the circular profiled die and with the tractrix profiled die.

### 6.2 First stage

Figure 27 shows the inside and outside radius of the cup wall after the first stage (as function of the cup height) measured in $0^\circ$, $45^\circ$ and $90^\circ$ to the rolling direction and the results obtained from FEM simulations, when the first stage was carried out with the circular profiled die.

From Figure 27 it can be seen that with the circular profiled die the outside cup wall is barrel shaped whereas with the tractrix profiled die the outside cup wall is nearly straight. There is reasonably agreement between the FEM results and the experimental results for the outside cup wall whereas the agreement for the inside cup wall, especially for the cup drawn with the circular profiled die, is not so good. With both die profiles FEM predicts a too small inside cup radius.

The same effect with regard to residual stresses as discussed in Chapter 5 was also seen in these simulations and in the experimental results; the residual stresses in the cup drawn with the tractrix profiled die were significantly smaller than the residual stresses in the cup drawn with the circular profiled die.

Rings were cut from the drawn cups in 20 mm cup height and slit open, and the change in diameter of the rings due to the slitting was measured. In the ring cut from the cup drawn with the circular profiled die the diameter change was 7.76 mm whereas the diameter change in the ring cut from the cup drawn with the tractrix profiled die was 1.02 mm.
6.3 Second stage

Figure 28 shows the inside and outside radius of the cup wall after the second stage as function of the cup height measured in 0°, 45° and 90° to the rolling direction when the drawing was carried with a circular profiled die in both stages and carried out with a tractrix profiled die in both stages.

![Figure 28 The inside and outside radius of the cup wall after the second stage measured in 0°, 45° and 90° to the rolling direction for the cup drawn with circular profiled dies in both stages and drawn with tractrix profiled dies in both stages. The results obtained from the FEM simulations are also shown.](image)

From Figure 28 it can be seen that the agreement between the experimental results and the FEM results, besides from the outside cup wall of the cup drawn with the tractrix profiled dies, is not very good. The agreement is especially bad for the inside radius, where FEM predicts a significantly smaller radius in the height range from 15 to 35 mm. Despite the not very good agreement, both the experimental results and the FEM simulations show that using a tractrix shaped die profiles in both stages gives a significant improvement in the straightness of the outside cup wall. It can also be seen that the outside radius of the cup wall is larger than the smallest inside radius of the draw die (= 20.28 mm) when the cup is drawn with the tractrix profiled dies and smaller when drawn with the circular profiled dies. That the FEM results predict a too small inside radius has of course the effect that FEM also predicts a too small cup height. In Table 5 are listed the predicted cup height and the average measured cup height. It can be seen that after the first stage the cup made with the circular profiled die is higher than the cup made with the tractrix profiled die, and that this difference increases after the second stage. The agreement between the measured height and the height predicted from the FEM simulations is not very good, especially for the cup height after the second stage when the drawing is carried out with circular profiled dies in both stages.

In Figure 29 are shown the vertical stress distribution and the hoop stress distribution in the cup wall after the second stage; to the left in the cup drawn with dies having a circular die profile in both stages and to the right with dies having a tractrix shaped profile in both stages. Independently of the die profiles used there are large tensile residual stresses towards the outside of the cup wall and large compressive residual stresses towards the inside.
In Figure 30 are shown the pressure stress distribution and the v. Mises stress distribution in the cup wall after the second stage; to the left when the cup is drawn with circular die profiles in both stages and to the right with tractrix shaped profiles in both stages. Contrary to the residual stresses in the cup wall after the first stage, the residual stresses are of the same magnitude in the cup drawn with the tractrix shaped die profiles and with the circular die profiles. The main difference in stress distributions as function of the die profiles used is that the distributions are somewhat smoother in the cup drawn with tractrix shaped profiles in both stages compared to when the cup is drawn with a circular die profiles in both stages.

In Figure 31 are shown the v. Mises stress distribution in the cup drawn with a tractrix shaped profile in both stages (left Figure) and with a circular profiled die in the first stage and a tractrix profiled die in the second stage. The v. Mises stress distribution and the shape of the cup wall is nearly identical in the two cups. According to the FEM simulations it is thus the shape of the draw die profile in the second stage, which to a large extend determines the shape of the cup wall and the residual stress distribution in the cup wall.

<table>
<thead>
<tr>
<th>First stage</th>
<th>Average measured cup height (mm)</th>
<th>FEM results (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Circular profiled die</td>
<td>29.41</td>
<td>27.44</td>
</tr>
<tr>
<td>Tractrix shaped profile</td>
<td>27.96</td>
<td>27.27</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Second stage</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Circular profiled die in both stages</td>
<td>43.29</td>
<td>38.81</td>
</tr>
<tr>
<td>Tractrix shaped profile in both stages</td>
<td>38.86</td>
<td>36.76</td>
</tr>
</tbody>
</table>

*Table 5 The average cup height measured experimentally and predicted from the FEM simulations.*
Figure 29 The vertical stress distribution and the hoop stress distribution in the cup wall after the second stage. To the left with circular die profiles in both stages and to the right with tractrix shaped profiles in both stages.
Figure 30  The pressure distribution and the v. Mises stress distribution in the cup wall after the second stage. To the left with circular die profiles in both stages and to the right with tractrix shaped profiles in both stages.
To get a qualitative idea of the magnitude of the residual stresses in the cup wall in the cups drawn in the experiments, rings were cut from the drawn cups in 15 mm and 30 mm cup height. After cutting each ring was slit open in 0° to the rolling direction and it was measured how much the diameter of the ring changed due to the slitting. In 15 mm cup height the measured change in diameter was 4.83 mm for the cup drawn with dies having a circular die profile in both stages and 4.61 mm for the cup drawn with dies having a tractrix shaped profile in both stages. In 30 mm cup height the measured change in diameter was 5.82 mm for the cup drawn with circular profiled dies in both stages and 3.21 mm for the cup drawn with tractrix profiled dies in both stages.

6.4 Industrial application of modifying the draw die profile

The manufacturing of the cup shown in Figure 32 was the subject in a student project [43]. The cup has as can be seen in Figure 32 cracks due to residual stresses. The outside diameter of the cup is ø87 mm and the cup was drawn in two drawing stages from 1.5 mm thick stainless steel (AISI 304). The total drawing ratio was 2.20. The company producing the cup used two separate drawing tools to make the cup. From time to time there were problems with delayed cracking. The purpose of the student project was to investigate if the draw profile in the second stage could be modified in such a way that the

![Figure 31 The v. Mises stress distribution in the cup drawn with a tractrix shaped die profile in both stages (left) and with a circular die profile in the first stage and a tractrix shaped die profile in the second stage (right).](image-url)
residual stresses in the cup after the second draw could be reduced and if so to make a new tool design in which the two deep drawings were combined. Figure 33 shows the combined tool. The main optimization effort was concentrated on the shape of the draw die profile in the second draw, part 18 in Figure 33. FEM simulations of the drawing process were carried out both as 2D rotational symmetric using LS-Dyna2d and as 3D using LS-Dyna3D. The 2D rotational symmetric simulations were used to determine the residual stresses in the finished drawn cup and the 3D simulations were used to determine the tendency to the formation of wrinkles in the cup wall. Different draw die profiles in the second draw were investigated ranging from profiles having a tractrix shaped profile to profiles having an elliptic shape. Based on the FEM simulations an elliptic profile having a minor axis of 7 mm and a major axis of 50 mm were chosen because this profile was found to give the best compromise between the magnitude of the residual stresses and tendency to the formation of wrinkles.

In Figure 34 is shown the residual v. Mises stress distribution in the cup drawn conventionally (with a die having a circular die profile in both stages) and drawn with a die having the elliptically shaped profile in the second draw. In the cup drawn with the die having an elliptic shape the v. Mises stress is, besides in a small region close to the rim, below 400 MPa whereas it is in the range from 600 to 1000 MPa in the cup drawn conventionally. It can also be seen that the cup drawn with the elliptically shaped die profile is higher than the cup drawn conventionally. This agrees with the experiments, which showed that with the same blank diameter the cup drawn with the elliptically shaped profile in the second draw was approximately 5% higher than the cup drawn conventionally.

In the cups drawn conventionally there could be seen a slight tendency to formation of wrinkles on the inside cup wall in the upper part of the cup. In the cups drawn with the
elliptically shaped profile in the second draw, there were no signs of wrinkles in the cup wall.

To experimentally verify the difference in the residual stress distribution depending on the draw dies used, rings were cut from cups in three different cup heights and slit open, and it was measured how much the rings opened up due to the slitting. The results are listed in Table 6. It can be seen that the opening of the rings cut from the cups drawn with the elliptically shaped profile in the second draw is much smaller than the opening of the rings cut from the cups drawn conventionally. The experimental results are thus in quantitative agreement with the results from the FEM simulations. Figure 35 show a cup (with cracks) drawn conventionally and a cup drawn using the combined drawing tools shown in Figure 33. To knowledge of the author the company making the cups has not had any problems with delayed cracking after the company started producing the cups with the combined deep drawing tool shown in Figure 33.

<table>
<thead>
<tr>
<th>Cup height</th>
<th>Opening of the ring Circular draw profiles in both stages</th>
<th>Opening of the ring Elliptically shaped profile in the last stage</th>
</tr>
</thead>
<tbody>
<tr>
<td>25 mm</td>
<td>48.5 mm</td>
<td>10.3 mm</td>
</tr>
<tr>
<td>50 mm</td>
<td>56.0 mm</td>
<td>9.2 mm</td>
</tr>
<tr>
<td>75 mm</td>
<td>31.5 mm</td>
<td>3.4 mm</td>
</tr>
</tbody>
</table>

*Table 6 Opening of the rings cut from cups drawn with circular die profiles in both stages and from cups drawn with an elliptically shaped die profile in the second stage.*
Figure 33 Combined two stage drawing tool for the manufacturing of a ø 87 mm cup made from 1.5 mm stainless steel [43].
Figure 34 The residual v. Mises stress distribution in the cup drawn with a die having a circular die profile in both stages (left) and in the cup drawn with the die having the elliptic shape in the second stage (right).

Figure 35 Cup drawn conventionally (the left) and using the elliptically shaped profile in the second stage (right).
6.5 Conclusions

The conclusions regarding the residual stresses in the cup wall and the shape of the cup wall after the first draw as function draw die profile are the same as those given in Chapter 5; that is the use of a tractrix shaped die profile compared a conventional circular profiled die yields

- A substantial reduction of the residual stresses in the cup wall
- A substantial increase in the straightness of the outside cup wall
- A cup with larger outside diameter than the smallest diameter of the draw die (with the circular profiled die the largest diameter of the cup (besides from the very rim) is smaller than the inside diameter of the draw die profile).

The FEM simulations and the experimental results show that the residual stresses in the finished drawn cup are of the same magnitude when the cup is drawn with the dies having a circular profile in both stages and with the dies having a tractrix shaped profile in both stages. The main difference in the stress distributions is that the distributions are smoother when the tractrix shaped die profiles are used.

When a tractrix shaped die profile is used in both stages, the outside cup wall is nearly straight whereas it is barrel shaped when a circular die profile is used in both stages. The FEM simulations show that when the second stage is carried out with a tractrix shaped die profile, the shape of cup wall and the residual stresses in the cup wall are nearly independent of the shape of the die profile used in the first stage.

The shape of the draw dies has also an influence on the final cup height. The cups made with a die having a circular die profile in both stages were higher than the cups made with a die having a tractrix shaped profile in both stages. In the industrial application where the die in the second stage was made with an elliptic shape it was found that the cup was approximately 5% higher than the cup made with a circular profiled die in both stages. It can thus be concluded that the final cup height can both decrease and increase depending on die design. In the industrial application it was also found that the use of the elliptically shaped die profile in the second stage increased the straightness of the outside cup wall of the finished drawn cup, decreased the tendency to the formation of wrinkles in the upper part of the cup and lead to a substantial decrease in the residual stresses.
7 Wear

The content of this Chapter is based on [44**]. To the knowledge of the author [44**] is one of the first publications describing how FEM can be used for the estimation of tool wear in a sheet metal forming process. Some of the results regarding FEM used to determine wear have been published in [45-49].

7.1 Estimation of wear using FEM simulations

The two main wear mechanisms governing the wear of a draw die are adhesive and abrasive wear. Many factors influence the wear volume, but it is generally accepted that as a first approximation the wear volume is proportional to the normal pressure between the contacting bodies and to the sliding length and inversely proportional to the hardness of the tool. A simple model for the wear volume can thus be formulated as

$$Z = K \frac{q_s}{H}$$

where $Z$ is the wear volume, $q$ the normal surface pressure, $s$ the sliding length, $H$ the hardness of the tool and $K$ a constant, which depends on surface topography, lubrication, the chemical composition of the two materials in contact etc..

If $A$ is a small area on the draw die, a rough estimate of the wear volume $Z_A$ on area $A$ can be obtained by integrating the pressure $q_A$ between the blank and the draw die in this area times the sliding velocity $v_A$, that is

$$Z_A = K^* \cdot \int_0^t q_A(t)v_A(t)dt \approx K^* \sum \frac{q_{A,n}}{v_{A,n}} \Delta t$$

where $K^*$ is a constant and $t$ the process time. The integration can, as indicated in Equation (3), be carried out by dividing the simulation time into $n$ intervals of equal length $\Delta t$. In each time interval, the average of the surface pressure $q_{A,n}$ on the element $A$ and the average sliding velocity $v_{A,n}$ over the element during the time interval $\Delta t$ can be determined from the FEM simulation and the summation is then carried out over the number of intervals.

The above approach was utilized to estimate the wear on a draw die. Two different die geometries were investigated, a conventional circular die profile and a tractrix shaped die profile. The tool geometries used are shown in Figure 36 and Figure 37 shows the FEM models. The length of the elements on the die profile was made same for all the elements on the profile. For this reason there are more elements on the tractrix shaped profile than on the circular die profile.

The blank used was ø180.78 mm with a thickness of 1.5 mm. The blank was modelled as an elasto-plastic material with the true stress – true plastic strain curve given by $\sigma = 1367 \varepsilon_{\text{plastic}}^{0.41}$ MPa. Coulomb friction was assumed in all interfaces with the friction coefficient $\mu = 0.1$. The FEM simulations were carried out using LD-Dyna2D and using explicit time integration.
During the FEM simulation data about the pressure on the die profile were dumped 100 times at constant time intervals. To make it easy to obtain a rough estimate of the wear profile on the die it was assumed that the sliding velocity $v_{A,n}$ between the blank and an element on the draw die profile was constant and the same for all the elements along the die profile. With this assumption Equation (3) reduces to

$$Z_{A} = K' \cdot \int_{0}^{1} q_{A}(t)v_{A}(t)dt \approx K' \sum_{1}^{n} q_{A,n}v_{A,n} \Delta t \approx K' \sum_{1}^{100} q_{A,n}$$  (4)

where $K'$ is a constant.
7.2 Wear profile on the circular die

In Figure 38 is shown the pressure on the nodes of the draw die for the 100 states dumped during the FEM simulation. Node 1 is the node where the blank material enters the circular die profile and node 27 is where the blank material leaves the profile (see Figure 39).

During most of the deep drawing the areas with high contact pressure are located in two regions. The first region is from node 4 to 7 and the second region is further down the draw die profile from node 16 to 22. The pressure in these regions is lying in the interval from 200 to 400 MPa. During the last part of the deep drawing process, the pressure on the lower part of the die, from node 20 to 27 increases and reaches pressures in the range from 400 to 600 MPa. During the very last part of the drawing process, the pressure on the nodes close to the die outlet increases very rapidly and becomes very high (in the range from 1400 to 1600 MPa). The reason for the drastic increase in pressure toward the end of the drawing is discussed in more details in Chapter 7.5.

Figure 38 The pressure on the nodes on the circular draw die profile at the different states during the FEM simulation.
Figure 40 shows the wear volume $Z_A$ at each node determined using Equation (4).

From Figure 40 it can be seen that the “wear” is located in two regions on the draw die profile; a region from node 4 to 7 near the die inlet and a region from node 16 to 22 closer to the die outlet. The locations of these regions are shown Figure 39. The location of the regions with high wear determined from the FEM simulation is in good agreement with the wear profiles measured experimentally [52,53].
7.3 Tractrix shaped die profile

In Figure 41 is shown the pressure on the nodes of the tractrix shaped draw die for the 100 states during the deep drawing. Node 1 is at the die inlet and node 106 at the die outlet, see Figure 42.

The nodes near the die inlet are during the first part of the deep drawing subjected to a very high pressure (> 600 MPa). The high pressure is caused by the bending of the blank material near the die inlet. The effective bending radius is with the tractrix shaped die profile so small that due to the bending the blank is, as can be seen from Figure 44, only in contact with the first couple of the nodes on the die profile. The small area of contact causes a very high pressure on these nodes.

Toward the end of the deep drawing the pressure becomes high on the nodes near the die outlet. The reason for the high pressure on these nodes is discussed in Chapter 7.5.

In Figure 43 is shown the wear volume $Z_A$ on each node determined using Equation (4). The wear is very high near the die inlet (from node 1 to 3) due to the high pressure here during the first part of the deep drawing. The wear is also high near the die outlet (from node 102 to 106). There is hardly any wear from node 3 to node 13 because during most of the deep drawing, there is, as shown in Figure 44, no contact between blank and die here. The wear on the remaining part of the die is low and fairly evenly distributed.

*Figure 41 The pressure on the nodes on the tractrix shaped die profile during a the different states the deep drawing*

Toward the end of the deep drawing the pressure becomes high on the nodes near the die outlet. The reason for the high pressure on these nodes is discussed in Chapter 7.5.
Figure 42 The node point numbering on the tractrix shaped die profile.

Figure 43 The wear volume $Z_A$ on each node point on the tractrix shaped die profile. Node 1 is at the die inlet and node 106 at the die outlet.
An obvious way to reduce the wear near the inlet of the tractrix shaped die is to modify the die geometry in such a way that the contact pressure between blank and die is reduced in this region. This can be accomplished by increasing the die radius near the die inlet. In Figure 45 is shown a modified tractrix shaped die profile, where the tractrix near the die inlet is replaced with a circular profile. FEM simulations were carried out with the modified die, and the results showed that the modification only had very minor influence on both the shape of the drawn cup and on the residual stresses in the cup wall.

In Figure 46 is shown the pressure on the nodes of the modified tractrix shaped die profile at the 100 states during the deep drawing and in Figure 47 is shown the wear $Z_A$ on each node on the die profile. The change in die profile has reduced the pressure near the die inlet from larger than 600 MPa (see Figure 41) to approximately 400 MPa. The reduced pressure lowers the maximum wear volume $Z_A$ at the die inlet from approximately 19000 (see Figure 43) to 12000. The wear on the remaining part of the
modified die is fairly even (besides from near the die outlet) and on the same level as with the original tractrix shaped profile.

Figure 46 The pressure on the nodes on the modified tractrix shaped die profile during the different states the deep drawing

Figure 47 The wear volume $Z_A$ on each node point on the modified tractrix shaped die profile. Node 1 is at the die inlet and node 106 at the die outlet

7.5 The high pressure towards the end of the deep drawing process

The pressure between the between the blank rim and the draw die becomes, as mentioned above, very high towards the end of the drawing process. This high pressure is caused by the unbending of the blank after the blank rim has passed the die outlet. Figure
48 shows the horizontal stresses and the vertical stresses in the cup wall and the circular profiled draw die toward the end of the drawing process. The unbending causes vertical tensile stresses towards the outside of the cup wall and vertical compressive stresses towards the inside (near B in Figure 48). The bending moment is caused by the contact pressure near A and B. Toward the end the vertical distance between A and B decreases, and as the unbending moment remains nearly constant, the decrease in distance A-B causes the contact pressure to increase near A and B. The contact pressure between punch and blank near B is lower than the contact pressure between blank rim and die near A because the contact area near B is much larger than the contact area near A. In practice the pressure in area A can become so high, that cup rim is flattened as shown in Figure 49.

**Figure 48** The horizontal (left) and the vertical stresses (right) in the cup wall toward the end of the deep drawing process.

**Figure 49** A deep drawn cup. The rim of the cup is shiny and bright due to the high contact pressure, which the rim has been subjected to during the very last part of the deep drawing.
7.6 Conclusions

The wear analysis described in [44**] and summarized in this Chapter shows that FEM simulations can be used effectively to estimate the wear profile on sheet metal forming tools. The predicted wear profile on a circular die profile is in good qualitative agreement with wear profiles measured experimentally [52,53].

The analysis of the wear on the tractrix shaped die profile shows that it is possible by making minor adjustments to the die profile to reduce the wear.

The work on wear determination using FEM simulations was continued in the Sheet Metal Forming Group, Department of Production by M.Sc., Ph.D. Morten Rikard Jensen, who refined the approach outlined in this Chapter and showed how FEM-simulations and optimization can be combined to determine the optimal shape of the die profile with regard to wear [48,50,51].

The work on determination of wear, summarized in this Chapter, was carried out by post processing data generated during the FEM simulations. However it is fairly simple to build into the FEM code the possibility of determining the wear profile directly during a FEM simulation and such an option has been implemented in some of the commercial FEM codes to day. In LS-Dyna such an option could be implemented by making a user defined friction subroutine where in each surface element of the tool the integral of surface pressure*sliding length is stored in a history variable.
8 The extrusion processes analyzed

Tools used in extrusion processes are usually designed in such a way that the tool section where the material leaves the deformation zone is parallel to the direction of the punch movement. In Figure 50 are shown sketches of the direct extrusion process, the ironing process and the backward can extrusion process and the tool parts parallel to the direction of the punch movement are indicated.

In direct extrusion, Figure 50(a), the parallel part of the extrusion die is commonly denoted bearing land or bearing area; in ironing, Figure 50(b), the part is commonly denoted die land and in backward can extrusion, Figure 50(c), the part is commonly denoted punch land. The extrusion processes are widely used in industry, but despite the industrial importance, which these processes have, surprisingly little has been published regarding which influence the bearing land, the die land and the punch land have in the respective extrusion processes. Surprising because as shown by Ackeret et al [54], Lof [55,56], Ras el al [57] and the author the bearing land, the die land and the punch land may have a significant influence on the stability of the extrusion processes.

Lof [55,56] has among other things investigated which effect the length and the angle of the bearing land have in direct extrusion of thin walled sections. Lof carried out his investigations using detailed 2D FEM simulations in which an ALE approach was used in order to avoid the problems with severe mesh distortion which would have occurred had the simulations been carried out with an Lagrangian description without remeshing. Lof showed that changes in the length of or small changes in the angle of the bearing land can cause abrupt changes in the friction conditions in the bearing channel; e.g. changing the angle of the bearing land from $-0.01^0$ to $0.01^0$ gave in some of the cases investigated rise to a nearly doubling of the extrusion pressure. Lof concludes that the use of parallel bearings in direct extrusion can be considered to be one of the most important factors contribution to the unpredictability of the direct extrusion process; unpredictability which can give rise to severe problems when running in a new extrusion die. Lof suggests that

Figure 50 Sketches of direct extrusion (a), ironing (b) and backward can extrusion (c). The bearing land (a), the die land (b) and the punch land (c) are indicated with circles in the sketches.
the bearing land in direct extrusion should be made either choked or relived because this will significantly reduce the effect, which small changes in the angle of the bearing land otherwise have.

Ras et al [57] analyzed the ironing of thin walled cans using the slab method and showed that due to the die land that there exists a reduction ratio (in this thesis denoted the critical reduction ratio) at which the radial force (per unit width in circumferential direction) has a maximum and pointed out that the ironing process is, if carried out close to the critical reduction ratio, in a state of unstable equilibrium and may become unstable leading to an uneven wall thickness in the ironed cup wall. Some years ago, the ironing of stainless steel cans was analyzed theoretically and experimentally in a M.Sc. project [34], which the author supervised. In this project severe variations in the can wall thickness were found in the upper part of the ironed cup wall, but the students could not (and neither could the author) at the time when the project was carried, explain why the thickness variations occurred. It was first when the author became aware of Ras et al’s paper, that a sensible explanation for the thickness variation could be given. The paper by Ras et al inspired the author to carry out a more detailed analysis of the influence from the die land and from material and process parameters than given in [57]. This more detailed analysis is summarized in Chapter 9.

The analysis of how the critical reduction ratio in the ironing process is influenced by process and material parameters gave the inspiration to analyze the effect, which a slight tilt of the die in relation to the punch has in the ironing process (this work is summarized in Chapter 10 and 12). The analyses carried out show that if the die land is cylindrical then a slight tilt of the die has a significant influence because due to the tilt the ironing process becomes unstable leading to variation in the cup height and variation in the cup wall thickness, primarily in the upper part of the cup. The 3D FEM simulations carried out also strongly suggest that a slight tilt of the die may cause fracture in the cup wall toward the end of the ironing process. The analyses also show that if the cylindrical die land is replaced with a die land having a circular profile, the ironing process becomes significantly more robust with regard to a slight tilt; if the die land is circular profiled a slight tilt of the die has nearly no effect on quality of the cups produced.

Eccentricity in a backward extruded can is according to Avitzur [58] one of the most common defects in backward can extrusion. The most obvious cause of eccentricity is according to Avitzur misalignment of the punch face in relation to the punch shaft as sketched in Figure 51; if the punch face is not orthogonal to the punch shaft the material flow beneath the punch can flow more freely to the left and the horizontal force component on the punch face will swing the punch to the right.

The author has analyzed the effect which a slight tilt of the punch land in relation to the direction of the punch movement has in backward can extrusion (this work is summarized in Chapter 13-19). The slab analyses, FEM simulations and experiments show that if the punch land is cylindrical, as recommend by the ICFG [59], a slight tilt of the punch land will cause the backward can extrusion process to become unstable leading to eccentricity in the produced can. FEM simulations and experiments also show that if the punch land is made profiled, e.g. circular profiled, the backward can extrusion process becomes significantly more robust with regard to a slight tilt of the punch land; the can produced is nearly unaffected by a slight tilt. Punches used in backward can extrusion is commonly made with very tight tolerances, and in the opinion of the author a slight tilt of the punch land is a much more likely explanation for the eccentricity problems in backward can extrusion than the explanation given by Avitzur [58].

It is in the opinion of the author thought provoking that the commonly used die design (with a cylindrical die land) used in ironing and the commonly used punch design (with a
cylindrical punch land) used in backward can extrusion are probably the worst possible designs with regard to process robustness; with the commonly used designs the processes are, in the opinion of the author, in a state of unstable equilibrium, and any disturbances will cause the processes to become unstable.

*Figure 51 Cause of eccentricity in backward can extrusion according to Avitzur [58].*
9 Analysis of the ironing process

The analyses of the ironing process summarized in this Chapter and in Chapter 10 and 11 are mainly based on the research report [60** - 64**]; some of the results have been published in [65*-69].

9.1 Introduction

Ironing is a commonly used process to reduce the wall thickness of a cup (or can), and is widely used for the manufacturing of beverage cans, pressure vessels for fire extinguishers, photosensitive drums etc.. Due to the importance of the ironing process, the process has been the subject of numerous experimental and theoretical investigations. Some of these investigations are referenced in [60**].

In Figure 52 is shown a sketch of the ironing process. The initial wall thickness $t_0$ is reduced to $t_1$ by forcing the cup though the die.

![Figure 52 Sketch of the ironing process.](image)

The reduction ratio, which the cup wall becomes subjected to during the ironing, is defined as

$$\text{reduction ratio (\%)} = \frac{t_0 - t_1}{t_0} \ast 100$$ (5)

In Figure 53 is shown a sketch of the commonly used die design, where the die has a conical die inlet with a semi-die angle $a$ and a die land with length $l$. Most of the ironing dies described in the literature have a die land which is cylindrical; that is the angle $a_1$ in Figure 53 is zero. The semi-die angle $a$ is usually somewhere between $5^\circ$ and $15^\circ$. In many of the investigations carried out focus has been on the determination of the optimal semi-die angle yielding the lowest drawing stress in the drawn cup wall and thus the highest reduction ratio. In all the papers describing the ironing process, besides the paper by Ras et al [57], the influence from the die land is not dealt with; in most of the papers the length of the die land is not specified and in those were the length is specified the length varies greatly; from $l = 0.2 \ast t_0$ to $l = 8 \ast t_0$. That the influence from the die land has not been dealt with in greater detail is somewhat surprising because as shown in this Chapter the length of the die land has a significant influence on the both the contact forces...
and on the stability of the ironing process. Especially the stability issue is in the opinion of the author very important. Many of the problems, which are seen in industry e.g. uneven cup height, uneven wall thickness, wrinkles in the cup wall, can in the opinion of the author be referred back to instability problems. Despite the large amount of papers dealing with the ironing process, the author has only found one paper, the paper by Ras et al [57] dealing with instability.

The ironing process is, at least in some cases, an extremely sensitive process. The author has been collaborating with a company manufacturing thin walled cups. These cups were made in a number of stages in a transfer press. In the first stage a round blank was punched from a sheet strip. This blank was in the second stage formed to a cup in a combined deep drawing and ironing process. The subsequent stages were pure ironing stages in which the thickness of the cup wall was reduced from approximately 0.8 mm to 0.1 mm. From time to time production problems occurred (uneven cup height, uneven cup wall thickness, fracture in the cup wall). The company was of the opinion that the main reason for the production problems was variations in the material properties. A lot of effort was put into how the material should be specified in order to avoid the problems but without a great deal of success. Despite a very accurate specification of the blank material with regard to thickness tolerances, chemical composition, grain size, hardness etc., there were productions problems from time to time. The production problems were solved on the shop floor on a trial and error basis; the dies were dismounted from the press, polished and mounted again, and this procedure was repeated until a satisfactory cup could be produced. The fact that a die producing cups with an unacceptable quality (uneven cup height, uneven cup wall thickness) can be made to produce cups with the required quality demands just by polishing shows that the ironing process, at least in some cases, is extremely sensitive; very minor changes in the die geometry and/or surface topography can have a significant influence on the stability of the ironing process.

Generally the ironing process is regarded as a self-centring process. This is however not always the case. Ras et al [57] showed that the ironing process may not be self-centring and that there for a given design of the ironing die exists a reduction ratio, in the following denoted the critical reduction ratio, at which the ironing process may become unstable. Ras et al carried out the analysis of the ironing process using the slab method and determined the radial force on the die and the tensile stress in the ironed cup wall as function of the reduction ratio. In Figure 54 are shown some of the results obtained by Ras et al. The radial force on the die has a maximum for a reduction ratio around 34%; the reduction ratio corresponding to this maximum is the critical reduction ratio. If the ironing
is carried out close to the critical reduction ratio the ironing process is in a state of unstable equilibrium; a slight disturbance can cause the reduction ratio to increase on one side and decrease on the opposite. The increase respectively decrease of the reduction ratio takes place under falling radial force on the die and at the same time radial force equilibrium is maintained. Ras et al discuss only very briefly how the critical reduction ratio is influenced by the die design and by process and material parameters, and the main purpose of the work carried out by the author and summarized it this Chapter has been to obtain a better understanding of the ironing process and to determine how the critical reduction ratio is influenced by the most important process and material parameters. A better understanding of why problems may occur will hopefully make it possible to take an engineering approach to the trouble solving and not base the problem solving on a trial and error approach on the shop floor.

The author analyzed the ironing process using the slab method and FEM-simulations. Two different die designs were investigated, the first one was the conventional die design with a cylindrical die land as shown in Figure 52 and in the other die design the die was

![Figure 54](image.png)

*Figure 54 Radial force on the ironing die (ring expansion force) and the tensile stress (tensile strength) in the ironed cup wall as function of the reduction ratio [57].*

The circular profile has the radius $R$. 

![Figure 55](image.png)

*Figure 55 Sketch of ironing process carried out with circular profiled ironing die. The circular profile has the radius $R$.  

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made with a circular profile as shown Figure 55.

The FEM-simulations were carried out as plane strain and 2D rotational symmetric using the implicit FEM code LLNL-Nike2d [1]. The slab analyses were carried out assuming plane strain conditions in the cup wall; an assumption which is reasonable if the thickness of the cup wall is small compared to the radius of the punch. To carry out the slab analyses two finite difference programs were developed, one used for the analysis when using the conventional die design and one when using the circular profiled die. The programs are described in detail in [60**] and [64**], respectively. The approach used is very similar to the approach described in [57] and [70].

9.2 The critical reduction ratio as function of the die design and of process and material parameters

9.2.1 Ironing carried out with the conventional die design with cylindrical die land

Initially it was the believed that it would be very simple to make an accurate and robust FEM-simulation of the ironing process; the geometry of the tools and the cup are simple and the boundary conditions well defined. However this turned out not to be the case and the problem was related to the die land. The contact algorithm used in Nike2d was penalty based and simulations carried out with different values of the penalty factor showed that the radial force on the die land was so heavily influenced by the value of the penalty factor that the results obtained for the radial force on the die land was judged to be unreliable. To circumvent this problem the die land was made slightly tapered by making the angle $a_1 = 0.3^\circ$ and increasing the die corner radius $r_{die}$ to 5 mm (see Figure 53). These changes made the predicted radial and vertical force on the die land nearly independent of the value of the penalty factor. In the 2D rotational symmetric simulations the radius $R$ of the punch was in all the simulations 19.99 mm, the inside radius of the cup before ironing 20 mm and the initial wall thickness $t_0$ 1 mm. In some of the simulations the die was prescribed a small radial displacement toward the centre (equivalent to a gradual decrease of the inside diameter of the ironing die) at the same time as the punch was prescribed to move vertically down. By doing so it was possible to obtain the forces on the die as function of reduction ratio (for a given set of process and material parameters) in one simulation.

9.2.1.1 Results obtained using the slab method

Figure 56 shows an example of the radial force on the conical part of the die, the radial force on the die land and the total radial force on the die as function of the reduction ratio. It can be seen that the radial force (per unit width) on the conical part increases with increasing reduction whereas the radial force on the die land decreases. The total radial force on the die, which is the sum of the two former mentioned forces, has a maximum around a reduction ratio of 26% (the critical reduction ratio). If the reduction ratio in the ironing process is close to 26%, a small disturbance may cause the reduction ratio to increase on one side and decrease correspondingly on the opposite side and at the same time horizontal force balance is maintained.
Influence from the length of the die land

Figure 57 shows the total radial force on the die and the drawing stress in the ironed cup wall as function of the reduction ratio with the length of the die land $l$ as parameter for the ironing carried out with a die having a semi-die angle $a = 10^\circ$, initial cup wall thickness $t_0 = 1$ mm, ideal plastic material with a yield stress $= 200$ MPa, and the friction coefficients $\mu_{\text{die}} = 0.1$ in respectively the cup-die interface and the cup-punch interface. From Figure 57 it is clear that the length of the die land $l$ has a significant influence on both the total radial force (per unit width) on the die and on the critical reduction ratio. The longer the die land $l$ is the larger is the total radial force and the smaller is the critical reduction ratio; with $l = 1$ mm ($l = t_0$) the critical reduction ratio is 48% whereas with $l = 10$ mm ($l = 10t_0$) the critical reduction ratio is less than 5%. It can also be seen that there is very good agreement between the slab results and the results...
obtained from FEM-simulations. From Figure 57 it can be seen that the drawing stress in the ironed cup wall increases with increasing reduction ratio, but that the value of the drawing stress for a given reduction ratio is independent of the length of the die land.

**Influence from the semi-die angle**

![Figure 58](image)

*Figure 58 The total radial force on the die (per unit width) and the drawing stress in the ironed cup wall as function of the reduction ratio with the semi-die angle $a$ as parameter.*

Figure 58 shows the total radial force on the die (per unit width) and the drawing stress in the ironed cup wall as function of the reduction ratio with the semi-die angle $a$ as parameter. The other main parameters are: length of the die land $l = 3$ mm ($= 3 \times t_0$), initial cup wall thickness $t_0 = 1$ mm, $\mu_{\text{die}} = \mu_{\text{punch}} = 0.1$, ideal plastic material with yield stress $= 200$ MPa. It can be seen that the semi-die angle $a$ has a significant influence on both the total radial force on the die and on the critical reduction ratio, whereas the drawing stress in the ironed cup wall is independent of the semi-die angle $a$. The smaller the semi-die angle $a$ is, the larger is the total radial force on the die and the larger is the critical reduction ratio. With a semi-die angle $a = 5^\circ$ the critical reduction ratio is 43% whereas the critical reduction ratio is less than 5% with a semi-die angle $a$ larger than 15$^\circ$.

**Influence from friction**

Figure 59 shows the total radial force on the die (per unit width) and the drawing stress in the ironed cup wall as function of the reduction ratio and with different combinations of the friction coefficient $\mu_{\text{die}}$ in the cup-die interface and $\mu_{\text{punch}}$ in the cup-punch interface. The other main parameters are: semi-die angle $a = 10^\circ$, length of die land $l = 3$ mm ($= 3 \times t_0$), initial cup wall thickness $t_0 = 1$ mm and ideal plastic material with yield stress $= 200$ MPa. It can be seen that the friction conditions in the interfaces have a significant influence on the radial stress on the draw die, on the drawing stress in the ironed cup wall and on the critical reduction ratio. It can also be seen that it is the difference between friction coefficients $\mu_{\text{die}}$ and $\mu_{\text{punch}}$ which is important and not the absolute values of the friction coefficients. The larger $\mu_{\text{punch}} - \mu_{\text{die}}$ is, the larger is the radial force on the die and the critical reduction ratio and the smaller is the drawing stress in the ironed cup wall. For $\mu_{\text{punch}} - \mu_{\text{die}} = 0.05$ the critical reduction ratio is 36% whereas with $\mu_{\text{punch}} - \mu_{\text{die}} = -0.05$ the critical reduction ratio is 18%.
Analysis of deep drawing, ironing and backward can extrusion

Influence from strain hardening

Figure 60 shows the radial force on the ironing die (per unit width) and the drawing stress in the ironed cup wall as function of the reduction ratio for three different values of the tangent modulus $E_{\text{tan}}$ in the expression $\sigma = \sigma_0 + E_{\text{tan}} \cdot \varepsilon_{\text{plastic}} = 200 + E_{\text{tan}} \cdot \varepsilon_{\text{plastic}} \text{ MPa}$ for the true stress–true plastic strain curve of the cup material. The other main parameters are: semi die angle $a = 10^\circ$, length of the die land $l = 4 \text{ mm} (= 4 \cdot t_0)$, initial cup wall thickness $t_0 = 1 \text{ mm}$ and $\mu_{\text{punch}} = \mu_{\text{die}} = 0.1$. It can be seen that the larger $E_{\text{tan}}$ is the larger is the total radial force on the die, the drawing stress and the critical reduction ratio. Increasing $E_{\text{tan}}$ from 0 MPa to 200 MPa increases the critical reduction ratio from 16% to 49%.

Figure 59 The total radial force on the die (per unit width) and the drawing stress in the ironed cup wall as function of the reduction ratio for different combinations of the friction coefficients $\mu_{\text{punch}}$ in the cup-punch interface and $\mu_{\text{die}}$ in the cup-die interface.

Figure 60 The total radial force (per unit width) and the drawing stress in the ironed cup wall as function of the reduction ratio for three different values of the tangent modulus $E_{\text{tan}}$. 
If the true stress-true plastic strain curve is given by $\sigma = C\varepsilon^n_{\text{plastic}}$, results presented in [60**] show that an increase in the deformation hardening exponent $n$ has a similar effect as an increase in the tangent modulus $E_{\text{tan}}$.

9.2.1.2 2D rotational symmetric FEM simulations

In the 2D rotational symmetric FEM simulations the die was prescribed a small radial displacement toward the centre at the same time as the punch was prescribed to move vertically down. By doing so it was possible to obtain the forces on the die as function of reduction ratio (for a given set of process and material parameters) in one simulation. Figure 61 shows the total radial force (per radian) on the die as function of the reduction ratio for three different values of the length of the die land $l$ ($l = 2$, $3$ and $4$ mm) determined using the slab method (the curves with the symbols) and using FEM (the thick solid curves).

![Figure 61 The total radial force on the die (per radian) as function of the reduction ratio for three different values of the length of the die land $l$ ($l = 2$, $3$ and $4$ mm) determined using the slab method (the curves with the symbols) and using FEM (the thick solid curves).](image)

At low reduction ratio the total radial force is primarily determined by the radial force on the die land (see Figure 56), and the reason for the difference between slab method results and FEM results is believed to be that the slab method overestimates the radial force on the die land. Figure 62 shows the radial stresses in the cup wall (determined from a FEM simulation) during the steady state phase for a reduction ratio of $9\%$ and it can be seen that there is a drop in the radial stresses right below the entrance to the die land region. In [60**] it is shown the radial stress distribution shown in Figure 62 is nearly independent of the penalty factor in the die-cup wall interface, of how many elements are used to model the cup wall and of the elastic deformation of the die (nearly the same distributions are obtained independently of whether the die is modelled as elastic with Young’s modulus $= 200000$ MPa or as rigid). It is thus believed that the drop in radial stress is primarily caused by the entry to the die land.
stress just below the entrance to the die land region is not due to some specific FEM-settings (choice of penalty factor, mesh size etc.) but is “physical”; that is the drop will also occur in the real ironing process. The slab method cannot capture a drop in the radial stress; with \( \mu_{\text{die}} = \mu_{\text{punch}} \) the slab method predicts a constant radial stress on the die land (around 210 MPa) and the slab method will thus predict a too high radial force on the die. According to the slab method the radial force on the die is 715 N (per mm width) and this force is made up of 93 N (per mm width) on the conical part of the die and 632 N (per mm width) on the die land.

Figure 62 Radial stresses in the cup during the steady state phase of the ironing process (reduction ratio = 9\%, semi-die angle \( \alpha = 10^\circ \), \( \mu_{\text{die}} = \mu_{\text{punch}} = 0.1 \), length of the die land \( l = 3 \text{ mm} \), initial cup wall thickness \( t_0 = 1 \text{ mm} \), elasto-plastic plastic material with constant yield stress = 200 MPa). The circle indicates the region below the entrance to the die land region where the radial stress drops.

Figure 63 Radial stresses in the cup during the steady state phase of the ironing process (reduction ratio = 49\%, semi-die angle \( \alpha = 10^\circ \), \( \mu_{\text{die}} = \mu_{\text{punch}} = 0.1 \), length of the die land \( l = 3 \text{ mm} \), initial cup wall thickness \( t_0 = 1 \text{ mm} \), elasto-plastic plastic material with constant yield stress = 200 MPa). The circle indicates the region below the entrance to the die land region where the radial stress drops.
Figure 63 shows the radial stress distribution (determined from a FEM simulation) when the reduction ratio is 49%. With this reduction ratio the drop in radial stress below the entrance to the die land region is not so pronounced and the stress distribution is more even compared to when the reduction ratio is 9%. With the high reduction ratio of 49% the radial stress in the die land region is also much lower; the slab method predicts the pressure on the die land to be around 76 MPa, the radial force on the conical part to be 439 N (per mm width) and 229 N on the die land.

In [60**] it is shown that the elastic deformation of the die and the punch is much more sensitive to variations in the friction conditions when the die land is long compared to when it is short (l/t₀ = 10 mm versus l/t₀ = 1). The die land is sometimes denoted “calibration length”. In the opinion of the author “calibration length” is thus a misleading term because according to the results given in [60**] the longer the calibration length is the larger is the variation in the elastic deformation of the punch and die (and thus larger variation in the thickness of the ironed cup wall) due to variation in the friction conditions.

9.2.1.3 Plane strain FEM-simulations

The influence from the length l of the die land on the stability of the ironing process was investigated using a plane strain FEM-model. A plane strain model can of course not show what is going to happen in the real 3D process. However, it is believed that there will be qualitative agreement between the results obtained with the plane strain model and a corresponding 3D model. Figure 64 shows the FEM model used. The cup wall was modelled as elasto-plastic with constant yield stress = 200 MPa. The punch and die were modelled as rigid. The friction coefficient in all the interfaces was 0.1 and the semi-die angle α was 10°. The initial cup wall thickness was constant = 1 mm on the left side, but had on the right side an increasing thickness from 1 mm at the punch nose to 1.5 mm at the cup rim. The initial reduction ratio was 20%. With a fixed die and the punch constrained not to move horizontally the reduction ratio would on the left cup wall remain at the initial reduction ratio of 20%, but increase on the right cup wall due to the increasing wall thickness. The die was free to move in horizontal direction and the punch was prescribed a vertical displacement and constrained not to move in horizontal direction. Figure 65 shows the equivalent strain distribution in the cup wall at different stages during the ironing for the length of the die land l = 1mm (the left Figures) and for l = 10 mm (the right Figures) (the Figures have been scaled in horizontal direction). With the length of the die land l = 1mm, the reduction ratio increases on both sides (the die is moved horizontally to the right) until the reduction ratio on the left side becomes so large that the cup wall fractures. With the length of the die land l =
10 mm the reduction ratio decreases on the left side (the die is move horizontally to the left) and increases on the right side until the reduction ratio on this side becomes so large that the cup wall fractures. If there is qualitative agreement between the plane strain FEM results and results obtained with the corresponding 3D FEM model, the plane strain FEM results indicate a significant influence from the length of the die land on the material flow; with a short die land, a small variation in the cup wall thickness in circumferential direction will not cause the ironing process to become unstable, whereas with a long die land the ironing process may become unstable.

Figure 65 The equivalent strain distribution in the cup wall at different stages during the ironing. The Figures to the left are for a die land length $l = 1 \text{ mm}$ and to the right for a die land length $l = 10 \text{ mm}$. 
9.2.2 Ironing carried out with a circular profiled die

The ironing process when using a circular profiled die as shown in Figure 55 was analyzed using the slab method and FEM. Figure 66 shows the radial force on the die (per unit width) as function of the reduction ratio when the ironing is carried out with a die having a circular profile with a radius of 20 mm. The other main parameters are: initial cup wall thickness $t_0 = 1$ mm, ideal plastic cup material with yield stress $= 200$ MPa, $\mu_{\text{die}} = \mu_{\text{punch}} = 0.1$. It can be seen that there is good agreement between the slab method results and the FEM results and the agreement is, contrary to when a die with a cylindrical die land is used, also good for low reduction ratios.

Due to the good agreement between the slab method results and the FEM results only the slab method results will be shown in the following.

![Figure 66 The total radial force on the die (per unit width) as function of the reduction ratio determined using the slab method and using FEM.](image)

Influence from the radius of the die profile

Figure 67 shows the radial force on the die (per unit width) and the drawing stress in

![Figure 67 The radial force on the die (per unit width) and the drawing stress in the ironed cup wall, both as function of the reduction ratio and with the radius $R$ of the circular die profile as parameter.](image)
the ironed cup wall both as function of the reduction ratio and with the radius R of the ironing die as parameter, when the ironing is carried out with ideal plastic cup material with a yield stress = 200 MPa, initial cup wall thickness $t_0 = 1$ mm and with $\mu_{\text{die}} = \mu_{\text{punch}} = 0.1$. From Figure 67 it can be seen that the drawing stress in the ironed cup wall is independent of the radius R of the ironing die and that the total radial force on the die increases with increasing radius of the ironing die. The critical reduction ratio is approximately 36% independently of the radius R of the circular die profile. This is quite different from when a die with a cylindrical die land is used; with a cylindrical die land, the length of the die land has a significant influence on the critical reduction ratio.

### Influence from friction

![Figure 68](image)

Figure 68 The total radial force on the die and the drawing stress in the ironed cup wall as function of the reduction ratio with five different combinations of friction coefficients in the cup-die and cup-punch interface.

Figure 68 shows the total radial force (per unit width) on the die and the drawing stress in the ironed cup wall as function of the reduction ratio with five different combinations of the friction conditions in the cup-die and cup-punch interface. The other main parameters are: radius R of the draw die profile = 20 mm, initial cup wall thickness $t_0 = 1$ mm, ideal plastic cup wall material with yield stress = 200 MPa. It can be seen that the friction conditions have a significant influence on both the total radial force and on the drawing stress and that it is not the absolute values of the friction coefficients, which are important, but the difference between $\mu_{\text{die}}$ and $\mu_{\text{punch}}$ which is important; the larger $\mu_{\text{punch}} - \mu_{\text{die}}$ is the larger is the total radial force on the die and the smaller is the drawing stress in the cup wall. It can also be seen that critical reduction ratio increases with increasing value of $\mu_{\text{punch}} - \mu_{\text{die}}$; for $\mu_{\text{punch}} - \mu_{\text{die}} = -0.05$ the critical reduction ratio is approximately 32%, for $\mu_{\text{punch}} - \mu_{\text{die}} = 0.00$ the critical reduction ratio is approximately 36% and for $\mu_{\text{punch}} - \mu_{\text{die}} = 0.05$ the critical reduction ratio is approximately 41%. With $\mu_{\text{punch}} - \mu_{\text{die}} = 0.05$ the drawing stress in the ironed cup wall is less than 0 for a reduction ratio less than 10%. This means that the ironing can be carried out without any drawing stress in the cup wall; the force to carry out the ironing is transmitted to the cup wall by the friction between the punch and the cup wall and at the outlet of the ironing die the cup wall is moving faster than the punch.
Influence from strain hardening

Figure 69 shows the total radial force (per unit width) on the die and the drawing stress in the ironed cup wall as function of the reduction ratio for three different values of the tangent modulus $E_{\text{tan}}$ in the true stress – true plastic strain curve $\sigma = 200 + E_{\text{tan}}\varepsilon_{\text{plastic}}$ MPa. The other main parameters are: radius of the die profile $R = 20$ mm, $\mu_{\text{punch}} = \mu_{\text{die}} = 0.1$. It can be seen that for a given reduction ratio an increase in $E_{\text{tan}}$ increases the total radial force on the die as well as the drawing stress in the ironed cup wall. It can also be seen that an increase in $E_{\text{tan}}$ increases the critical reduction ratio. With $E_{\text{tan}} = 0$ the critical reduction ratio is approximately 36%, with $E_{\text{tan}} = 100$ MPa the critical reduction ratio increases to 47% and with $E_{\text{tan}} = 200$ MPa the critical reduction ratio is larger than 50%.

If the true stress – true plastic strain curve is given by $\sigma = C_n\varepsilon_{\text{plastic}}^n$, the deformation hardening exponent $n$ has a similar effect as $E_{\text{tan}}$. With the radius of the die profile $R = 20$ and $\mu_{\text{punch}} = \mu_{\text{die}} = 0.1$, the critical reduction ratio is 39% for $n = 0.05$, 42% for $n = 0.1$ and 47% for $n = 0.2$.

9.3 Conclusions

The ironing process carried out with a conventional die design with a cylindrical die land and carried out with a die having a circular profile has been analyzed using the slab method and FEM. The main focus has been on analysis of the total radial force on the die as function of material and process parameters. The results show that there exists a reduction ratio at which the total radial force on the die has a maximum. The reduction ratio corresponding to this maximum is denoted the critical reduction ratio. If the ironing process is carried out close to the critical reduction ratio, the ironing process is in a state of unstable equilibrium and a small disturbance may cause the ironing process to become unstable (or not self centring) leading to an increase of the reduction ratio on one side of the die and a corresponding decrease on the opposite side.

With the conventional die with a cylindrical die land there is very good agreement between radial force on the die determined using the slab method and using FEM for
reduction ratios larger than 25%; for reduction ratios smaller than 25%, the slab method overestimates the radial force on the die. With the die having a circular profile there is very good agreement between the results obtained with the slab method and with FEM for all reduction ratios. The results show that the radial force on the die is heavily influenced by the die design and by the material and friction parameters.

If the ironing is carried out with a conventional die having a cylindrical die land the total radial force on the die is increased

- when the semi-die angle is decreased
- when the length of the die land is increased
- when the value of $\mu_{\text{punch}} - \mu_{\text{die}}$ is increased
- when the strain hardening of the cup material is increased

and that the critical reduction ratio is increased when

- when the semi-die angle is decreased
- when the length of the die land is decreased
- when the value of $\mu_{\text{punch}} - \mu_{\text{die}}$ is increased
- when the strain hardening of the cup material is increased

If the ironing is carried out with a die having a circular profile the results show that the total radial force on the die is increased if the radius $R$ of the die is increased and that the critical reduction ratio is approximately 36% and nearly independent of the radius $R$. The influence from friction and strain hardening is similar to when a conventional die with cylindrical die land is used.
10 Instability in the ironing process due to a slight tilt of the die land

10.1 Introduction

When the ironing process is carried out with a conventional die with a cylindrical die land as shown in Figure 52 the ironing process can also become unstable if the die land for some reason or another is slightly tilted. Such a tilt may be caused by e.g. inaccurate machining, inaccurate mounting/alignment of the tools in the press and/or elastic deformation of the tools and press during the ironing.

In Figure 70 is shown a sketch of a slightly tilted ironing die. Due to the tilt contact between the cup wall and die land is completely lost on side A, whereas a slight tilt only will have a marginal effect on the opposite side, side B. To the right in Figure 70 is shown the radial force (per unit width) on the conical part of the die, the radial force on the die land and the total radial force as function of the reduction ratio; the forces have been determined using the slab method. On side A the total radial force on the die is equal to the radial force on the conical part of the die, because due to the loss of contact between cup wall and die land, the radial force on the die land is zero. On the opposite side, the total radial force on the die is the sum of the radial force on the conical part and the radial force on the die land. On the right Figure is indicated what happens if the die is slightly tilted during the ironing. Before the tilt of the die, the reduction ratio is 20% and the radial force on the die is approximately 400 MPa. Due to the tilt, the total radial force on side A, where contact between cup wall and die land is lost, drops to approximately 225 MPa, whereas the total radial force on side B is nearly unaffected by tilt. This difference in total radial force on side A and side B creates a net radial force on the die which will try to deflect the die to the left (or the punch to the right) and the reduction ratio will thus increase on side A and decrease correspondingly on side B. If the die is floating, radial force equilibrium will first be reached when the reduction ratio has increased to 30% on side A and decreased to 10% on side B. As the tilt angle required to cause a complete loss of contact between the cup wall and die land on side A is infinitely small (at least if the
elastic strains are neglected) even a “perfect” ironing process carried out way below the critical reduction ratio is in the opinion of the author in a state of unstable equilibrium; any disturbances will cause the ironing process to become unstable.

If the ironing die is made with a circular profile as shown in Figure 55, a slight tilt of the die will only give rise to minor changes in the contact conditions between cup wall and die and it is thus expected that the effect of a slight tilt will be significantly smaller compared to when a conventional die with a cylindrical die land is used.

The influence, which a slight tilt of die has when using a conventional die with a cylindrical die land and when using a circular profiled die, was analyzed using 2D FEM simulations (plane strain and rotational symmetric) using the codes Nike2d and LS-Dyna and 3D simulations using the code LS-Dyna. All the 2D simulations were carried out using implicit time integration whereas the 3D simulations were carried out using explicit time integration.

10.2 2D plane strain FEM-simulations

Plane strain simulations can of course not show the effect of a tilt of the die in the real 3D process, but it is believed that there will be qualitative agreement between the 2D plane strain results and the corresponding 3D results. The plane strain FEM model used to simulate the effect of a slight tilt of the die when using a conventional die with a cylindrical die land is very similar to the FEM model shown in Figure 64; the main differences compared to the FEM model shown in Figure 64 were that the cup wall thickness was constant and equal to 1 mm on both the left and the right side of the punch and that the die was tilted 0.2° in relation to the punch. The FEM model when using the circular profiled die was besides from the geometry of the die identical to this FEM model. The cup wall material was modelled as elasto-plastic with constant yield stress = 100 MPa. The friction coefficient in all interfaces was set to 0.1. The die was with both die designs tilted 0.2° in counter clockwise direction in relation to the punch and the die was free to move in horizontal direction (floating). The nominal reduction ratio was in both cases 20%.

In Figure 71 to the left is shown the equivalent strain distribution in the ironed cup wall when the ironing is carried out with a die with a cylindrical die land (semi-die angle α = 10°, length of die land l = 10 mm, initial cup wall thickness t₀ = 1mm) and to the right when carried out with the circular profiled die (radius of die profile R = 10, initial cup wall thickness t₀ = 1 mm). From Figure 71 it is clear that when the die has a cylindrical die land the influence from a slight tilt of the die is significant; the die is shifted to the left to such a degree that the reduction ratio on the left side decreases to zero. The significant difference in reduction ratio on the left and the right side gives of course rise to a significant difference in the cup height. When the circular profiled die is used the ironing process stabilizes very rapidly; the die is shifted slightly to the left but stops drifting when horizontal force balance has been reached. The slightly larger reduction ratio on the right side gives of course rise to a slightly higher cup height on this side, but the difference in the cup height on the left and right side is significantly smaller compared to when the die with the cylindrical die land is employed. The 2D FEM simulations thus suggest that the ironing process becomes significantly more robust (or less sensitive) with regard to a slight tilt of the die, when the ironing die is carried out with a circular profile die in place of the conventional die with a cylindrical die land.
Two 3D simulations were carried out of the ironing of a thin walled cup, one with a conventional die with a cylindrical die land and one with a circular profiled die. In both simulations the die was tilted 0.4° in relation to the punch. Figure 72 shows the FEM model of the ironing process with the die with the cylindrical die land. Besides from the geometry of the ironing die, the FEM model of the ironing carried out with the circular profiled die was identical to this model. Due to symmetry the model was made as a half model. The main parameters used in the simulations are listed below:

- Initial cup wall thickness $t_0 = 0.2$ mm
- Punch radius: 10 mm
- Reduction ratio: 25%
- Cup material: elasto-plastic material with constant yield stress = 100 MPa
- Friction coefficient in all interfaces: 0.1
- Tilt angle of die in relation to the punch: 0.4°

Conventional die with cylindrical die land:
- Semi-die angle $a$: 13°
- Length of die land $l$: 1 mm

Circular profiled die:
- Radius $R$ of the circular profile: 20 mm

The cup wall was modelled using 8 node solid elements with 5 elements through the wall thickness. The total number of elements in the cup wall was 264000. The die and the
punch were modelled using rigid shell elements; the total number of shell elements was 16380. The punch was free to move in horizontal directions but had all other degrees of freedom constrained. The die was prescribed a vertical displacement and had all other degrees of freedom constrained.

Figure 73 shows a close up of the die with the cylindrical punch land tilted 0.4°. To capture the effect of the tilt, the node penetration in the cup wall-die interface must be significantly less than 0.007 mm. In LS-Dyna there are a number of different contact algorithms, most of them are penalty based. Simulations were carried out using different penalty based contact algorithms and different penalty factors, but it was not possible to obtain a sufficiently small node penetration using a penalty based contact algorithm. Instead of a penalty based contact algorithm a constrained based contact algorithm was used (the contact algorithm *CONTACT_CONSTRAINT_SURFACE_TO_SURFACE in LS-Dyna). With this contact algorithm the maximum node penetration was less than 0.4 µm.

Figure 74 shows the equivalent strain distribution in the cup made with the conventional die with the cylindrical die land (left Figure) and with the circular profiled die (right Figure). From these strain distributions it is clear that the tilt of the die has a significant influence on the cup produced with the conventional die; the strain distribution, the cup height and the cup wall thickness is very uneven. The cup produced with the circular profiled die is nearly unaffected by the tilt; the strain distribution, the cup height and the cup wall thickness is nearly even. The FEM simulations show that a slight tilt of the die will cause the ironing process to become unstable when a conventional die with a cylindrical die land is used and support the hypothesis that the ironing process becomes significantly more robust with regard to the effect of a slight tilt of the die when a circular profiled die is used.
From Figure 74 it can be seen that the inhomogeneous strain distribution in the cup produced with the conventional die with the cylindrical die land is constrained to the upper part of the cup wall. The reason for this is believed to be as follows: if the reduction ratio is to increase on one side and decrease on the opposite side, the cup wall above the ironing die must move faster in vertical direction where the reduction ratio increases than on the opposite side where the reduction ratio decreases (provided that cup wall does not fracture or wrinkles). However the cup wall above the ironing die will resist such a velocity difference and only when there is a “limited” amount of cup wall above the die (and thus a limited amount of material to deform to facilitate a velocity difference) can the reduction ratio increase on one side and decrease on the opposite side. A bulge is formed above the die land as shown in Figure 75 on the side where the reduction ratio tends to increase (the left side). This bulge increases the radial force on the die and balances the radial forces such the ironing process remains fairly stable. The bulge also gives rise to vertical compressive stresses in the cup wall above the ironing die, and towards the end of the ironing process, these vertical compressive stresses can plastically deform the cup wall above the ironing die with the effect that the reduction ratio increases on one side and decreases on the opposite side as shown in Figure 74.

Figure 74 The equivalent strain distribution in the cup produced with the die with the cylindrical die land (left) and with the circular profiled die (right). The die is in both cases tilted 0.4°.
Some years ago an experimental investigation of the ironing of thin walled cups made from austenitic stainless steel was carried out at the Department of Production [34]. In these experiments conventional dies with a cylindrical die land were used. In some of the ironed cups large variations in the cup wall thickness were observed in the upper part of the cup wall. Figure 76 shows the cup wall thickness in the upper part of an ironed cup, measured on two opposite sides. It can be seen that the wall thickness begins to vary greatly from around 30 mm below the cup rim. The ironing die, tool alignment, the press and the guiding of the punch in relation to the ironing die were carefully checked but the thickness variation could not be attributed to inaccurate machining of the ironing die, inaccurate guiding of the punch, inaccurate tool mounting etc.; at the time when the experiments were conducted it was concluded that the thickness variation was due to some kind of process instability but no rational explanation could be given for the instability. To day the author is of the opinion that the thickness variation has been caused by a slight tilt of the die.

When carrying out the ironing experiments reported in [34] fracture occurred from time to time in the upper part of the cup wall and Figure 77 shows a cup with such a fracture. Figure 78 shows the equivalent strain distribution in the cup wall at the end of...
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the ironing process and very good qualitative agreement between the fractured cup, Figure 77, and the FEM-simulation, Figure 78, can be seen. The FEM-simulation was carried out without a fracture criteria in the constitutive material model; had the simulation been carried out with an equivalent strain based fracture criteria causing fracture if the equivalent strain exceeded 1.0, the FEM-simulation would predict the place of fracture initiation and the shape of the fractured upper part in very close agreement with the experimentally observed fracture pattern.

Figure 76 The cup wall thickness in the upper part of an ironed stainless steel cup measured on two opposite sides.

Figure 77 Ironed stainless steel cup with fracture in the upper part of the cup wall [34].
10.4 Conclusions

If the ironing is carried out with a conventional die having a cylindrical die land a slight tilt of the die (a slight tilt, which may be cause by inaccurate machining of the die, inaccurate mounting of the die and/or elastic deformation of the tools and/or press during the ironing) will cause a complete lost of contact between the cup wall and the die land on one side, whereas the tilt will only give rise to minor changes in the contact conditions on the opposite side.

The slab analysis of the ironing carried out with a conventional having a cylindrical die land shows than the loss of contact between die land and cup wall due to a slight tilt of the die will give rise to a net horizontal force, which will try to deflect the punch of centre leading to an increase in the reduction ratio, where contact between cup wall and die land is lost and a corresponding increase in the reduction ratio on the opposite side.

If the ironing is carried out with a die having a circular profile in place of the cylindrical die land, it is expected than a slight tilt of the die will only give rise to minor changes in the contact conditions between the cup wall and the ironing die, and the hypothesis is that the ironing process thus becomes significantly more robust with regard to a slight tilt of the die. This hypothesis was tested using 2D plane strain simulations and 3D simulations.

The 2D plane strain simulations indicate that if a die having a cylindrical die land is used, then a slight tilt of the die will give rise to an increase in the reduction ratio on the side, where the tilt causes a complete loss of contact between cup wall and die land, and a corresponding increase in the reduction ratio on the opposite side. The 2D plane simulations also indicate that the ironing process becomes significantly more robust with regard to a slight tilt of the die if the die is circular profiled.

The 3D simulations show that when a conventional die with a cylindrical die land is used, a slight tilt of the die gives rise to a very uneven cup wall thickness and cup height. The 3D FEM simulations of the ironing process carried out with a circular profiled die
show that a slight tilt of the die only gives rise to very minor changes in the strain distribution in the ironed cup wall and that the height of the ironed cup is nearly unaffected by a slight tilt of the die. The 3D simulations thus strongly support the hypothesis that the ironing process becomes significantly more robust with regard to a slight tilt of the die when the ironing die is made circular profiled in place of the conventional die design having a cylindrical die land.

When the conventional die with the cylindrical die land is used, the simulations show that the increase in reduction ratio, on the side where contact is lost due to the tilt of the die, and the corresponding decrease on the opposite side, is constrained to the upper part of the cup wall. When ironing the lower part of the cup, the tilt of the die causes a bulge to be formed just above the die land on the side where contact is lost due to the tilt. The bulge gives rise to vertical compressive stresses in the cup wall above the bulge, but at the beginning of the ironing process there is so much cup material above the die land that the vertical stresses can not plastically deformed the cup wall above the die; instead the bulge grows to such a size that horizontal force balance is maintained. Towards the end of the process, when there is a limited amount of cup wall above the die land, the cup wall above the die land becomes plastically deformed allowing a significant increase in the reduction ratio on the side where contact between cup wall and die land is lost due to the tilt and a corresponding decrease in reduction ratio on the opposite side. The increase in reduction ratio where contact is loss may become so large and thus the cup wall so thin that the cup wall may fracture here.

Experimentally when ironing stainless steel cups with a conventional die with a cylindrical die land [34] it was observed that from time to time the upper part of the cup broke of and that the place of fracture initiation was where the cup had become very thin. The experimentally observed fracture pattern is in good agreement with the 3D FEM simulation results.
11 The formation of an “elephant foot”

Sometimes a bulge is formed near the bottom of the ironed cup. Figure 79 shows such a bulge.

The author is not aware of a common name for this defect. In a Danish company the defect is denoted “elephant foot” due to the resemblance. The author has carried out FEM-simulations of a number of ironing processes and in one of these simulations an “elephant foot” was formed. Figure 80 shows the FEM model of the three stage ironing process simulated. There was no tendency to the formation of an elephant foot in stage one and two, but the elephant foot showed up in stage three as shown in Figure 81. From Figure 81 it can be seen that the elephant foot is caused by material being drawn out from below the punch nose; that is an elephant foot will be formed if the drawing stress in the ironed cup wall exceeds the stress required to plastically deform the material beneath the punch nose. As the drawing stress in the ironed cup wall depends on die geometry, material parameters and friction conditions, the tendency to the formation of an elephant foot will also depend on these parameters. How this dependency is to the knowledge of the author unknown, but as the elephant foot can be made to show up in an FEM-simulation, FEM will also be a very effective tool to determine the process window within which the ironing process should be carried out in order to avoid the formation of an elephant foot.
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Figure 80 FEM model of a three stage ironing process.

Figure 81 The cup in the third ironing stage right before the formation of the elephant foot (left) and after the formation of the elephant foot (right).
12 Analysis of a combined deep drawing and ironing stage

The content of this Chapter is primarily based on the research reports [72**,73**]. Some of the results have been published in [66*,67*,74*,75*]

12.1 Introduction

Very long thin walled cups are commonly produced in several stages in a transfer press. In Figure 82 is shown an example of the stages in a transfer press.

![Figure 82 Production of a long and thin walled cup in a transfer press.](image)

The first stage is a blanking process in which a circular blank is cut from a sheet strip. In the second stage the blank is formed into a cup and in the subsequent stages the wall thickness of the cup is reduced by ironing. The second stage is sometimes a pure deep drawing process and sometimes a combined deep drawing and ironing process, and it is an analysis of the combined deep drawing and ironing process, which is dealt with in this Chapter.

From time to time a company producing long thin walled cups had severe problems making the cups; the cup wall thickness was uneven and/or fracture occurred in the cup wall. The company was of the opinion that the problems were primarily related to variation in the material properties of the sheet material, but despite very tight tolerances on the thickness of the sheet material, on the chemical composition, the mechanical properties and on the surface roughness the problems occurred from time to time. The purpose of the investigation carried was to identify likely explanations for the production problems and to come up with possible solutions to the identified problems.

12.2 Observations on the shop floor

According to the craftsmen running the production on the shop floor it is extremely important that the deep drawn and ironed cup after the second stage has an even cup height. If this is not the case there will inevitably be severe problems in the subsequent ironing stages (uneven wall thickness, uneven cup height, fracture in the cup wall, the cup wall is not straight (has the form like a banana), and it is not possible to produce a long thin walled cup with the required quality.

If the cup produced in the second stage did not have the required quality with regard to evenness of the cup height, the deep drawing and ironing die was dismounted from the
press, polished and remounted again and this procedure was repeated until the cup had a satisfactory quality. It can thus be concluded that the deep drawing and ironing stage is extremely sensitive to very minor changes in the tool geometry or surface topography because the die could be made to produce cups with an acceptable quality just by polishing the die.

12.3 Measurement carried out on cups taken from the production

Two cups were picked at random from the actual production while the production was running satisfactorily; the cups picked thus fulfilled the normal quality requirements. The height of these two cups (the cups are in the following denoted cup 1-5 and cup 2-5) and the wall thickness were measured using a 3D coordinate measuring machine. Wall thickness measurements were carried out for every $45^\circ$, with $0^\circ$ taken as the direction in which the cup was highest.

![Figure 83 The measured cup height](image)

Figure 83 shows the measured cup height and it can be seen that the maximum difference in cup height is approximately 1.2 mm for both cups. It can also be seen the two cups are nearly identical.

Figure 84 shows the cup wall thickness, measure for every $45^\circ$, as function of the cup height. The Figures to the left are for cup 1-5 and the Figures to the right for cup 2-5. Figure 85 shows the cup wall thickness as function of the angle and with the cup height as parameter (the data in Figure 85 are the same as in Figure 84, just plotted differently).
Figure 84 Cup wall thickness (measured in different angles) as function of cup height. The Figures to the left are for cup 1-5 and the Figures to the right for cup 2-5.
By comparing the thickness distributions for cup 1-5 and cup 2-5 it can be seen that the distributions are nearly identical.

From Figure 85 it can be seen that the thickness variation in circumferential direction in 2 mm cup height is very different from the variation in 4 mm cup height; in 2 mm cup
height the smallest wall thickness is in approximately $180^0$, whereas from 4 mm to 12 mm
cup height the smallest wall thickness is in approximately $0^0$. The wall thickness variation
in 2 mm cup height could indicate that the punch initially was positioned slightly off
centre. In 14 mm cup height the variation changes again, so the smallest wall thickness is
in $180^0$ and the largest in $0^0$.

Figure 86 shows the average of the cup wall thickness in $0^0$ and $180^0$ and in $90^0$ and
$270^0$. It can be seen that the average wall thickness increases as function of cup height;
this increase is due to elastic deformation of the tools. It can also be seen than the average
of the wall thickness in $0^0$ and $180^0$ is slightly smaller than the average of the wall
thickness in $90^0$ and $270^0$.

![Graph showing average cup wall thickness](image)

**Figure 86 The average of the cup wall thickness in $0^0$ and $180^0$ and in $90^0$ and $270^0$ as
function of the cup height; to the left for cup 1-5 and to the right for cup 2-5.**

### 12.4 Possible explanations for the uneven cup height and
variations in cup wall thickness

Figure 87 shows a sketch of the combined deep drawing and ironing process. In the
press the die and the punch are fixed. This is in contrast to the die in the subsequent
ironing stages, where the die is floating allowing the die to be displaced radial to the
direction of the punch movement during the ironing process. The combined deep drawing
and ironing has a cylindrical die land with a length of 2.75 mm (= 3.44 times the initial
blank thickness). The deep drawing ratio is 1.67 and the nominal thickness reduction in
the ironing process is 8.75%.

As described in Chapter 9 the ironing process may become unstable if the reduction
ratio is close to the critical reduction ratio. Despite the low nominal reduction ratio, the
real reduction ratio is significantly larger due to the increase in wall thickness caused by
the deep drawing. To determine the real reduction ratio the deep drawing and ironing
process was simulated taking into account the elastic deformation of the tools. From these
simulations the real reduction ratio towards the end was found to be approximately 20%.
This reduction ratio may be close to the critical reduction ratio (see e.g. Figure 57).

However, the variation in wall thickness is not believed to be due to instability caused by
the reduction ratio being close to the critical reduction ratio because the variation in cup
wall thickness in circumferential direction is not confined to the upper part of the cup
wall, where the reduction ratio has been close to 20%; there is also variation in the wall
thickness in the lower part of the cup, where the reduction ratio has been low.
As described in Chapter 9 the ironing process, when carried out with a die having a cylindrical die land, becomes unstable if the die for some reason or another is slightly tilted and that the ironing process becomes significantly more robust with regard to a slight tilt if the die land is made circular profiled. To investigate if a slight tilt of the die can be the reason for the thickness variations in the cup, the deep drawing and ironing was analyzed using FEM simulations. A small tilt may also explain why the average of the wall thickness in 0° and 180° is slightly smaller than the average of the wall thickness in 90° and 270° (see Figure 86).

12.5 2D plane strain simulation of ironing carried out with a constrained die

In the 2D plane strain simulations only the ironing was simulated. In Figure 89 is shown the FEM model used. The die is constrained not to move in vertical direction and is attached to a spring, which constrains the horizontal displacements (a force is required to displace the die horizontally). The punch is constrained not to move horizontally and is prescribed to move vertically down with constant velocity. The nominal reduction ratio is 8.75% and the length of the cylindrical die land is equal to 2.75 times the initial cup wall thickness. The die is tilted 0.2° in clockwise direction in relation to the punch and displaced slightly to the left in order to have initially a larger reduction ratio on the right half. The reason for this slight displacement to the left was to simulate the off-centre punch setting, which may have been used when producing the two measured cups. In Figure 88 is shown equivalent strain distribution at different stages during the ironing (the Figures have been scaled in horizontal direction in order to make it easier to see what happens). To start with (the left Figure) the reduction ratio is larger on the right side than on the left side due to the off-centre placement of the die. Due to the tilt contact between cup wall and die land is lost on the left half of the die and this loss of contact gives rise to a net horizontal force, which will try to move the die to the right decreasing the reduction ratio on the right side and increasing it on the left. It can be seen from the middle Figure that the spring has not been stiff enough to avoid the die to move horizontally towards the right. The initial off-centre displacement of the die has reduced the difference in cup
height on the left and right side, but there is as can be seen a significant variation in the cup wall thickness.

Figure 89 2D plane strain FEM model of the ironing carried out with a die having a cylindrical die land. The movement of the die in horizontal direction is constrained by a spring and the die has been tilted 0.2°.

Figure 88 The equivalent strain distribution in the cup wall at different stages during the ironing with the die tilted 0.2°. The horizontal displacement of the die was constrained by a spring. The die was initially placed slightly off-center.

When interviewing the craftsmen running the production on the shop floor they told the author that they sometimes tried to obtain a cup with a sufficiently even cup height by intentionally offsetting the die in relation to the punch. The FEM simulation indicates that it is possible to obtain a cup with nearly no variation in cup height by displacing the die off centre in relation to the punch. However, offsetting the die will inevitably produce a cup with variations in the cup wall thickness in circumferential direction. The FEM simulation thus suggest that measuring the evenness of the cup rim should not be used as
a quality measure; an even cup rim does not guarantee that there is no variation in the wall thickness in the circumferential direction.

At the very end of the ironing process, Figure 88 right, the radial force between the cup wall and die decreases on the right side and the spring force is large enough to squeeze the upper part of the right cup wall; the thickness of the upper left cup wall increases correspondingly. If the wall thickness variation to the left and right is compared to the wall thickness variation measured in $0^\circ$ respectively $180^\circ$, Figure 85, there is qualitatively good agreement between the FEM results and the measured thickness distributions: near the cup bottom the wall thickness is largest on the left side ($0^\circ$ in Figure 85), higher up on the cup wall the largest wall thickness is found on the right side ($180^\circ$ in Figure 85) and towards the very end the largest wall thickness is again found on the left side ($0^\circ$ in Figure 85).

### 12.6 3D simulations

3D simulations were carried out of the deep drawing and ironing with the conventional die with the cylindrical die land and with a die having a circular profiled die land. Figure 90 shows a sketch of the die having a circular profiled die land and Figure 91 shows the conventional die drawn together with the die having the circular profiled die land.

![Figure 90 Die having a circular profile with radius R.](image)

From Figure 91 it can be seen that the difference between the geometry of the conventional die with the cylindrical die land and the geometry of the die having the circular profiled die land is very small.

Figure 92 shows the FEM-model. The FEM-model was due to symmetry made as a half model. The blank was modelled with solid elements (4 elements in the through

![Figure 91 The geometry of the conventional die with a cylindrical die land (dotted line) drawn together with the die with a circular profiled die land.](image)
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thickness direction, total number of elements 30656) and the tools were modelled using rigid shell elements (total number of shell elements: 36606). The blank material was modelled as an elasto-plastic von Mises material hardening according to Hollomon’s hardening law $\sigma = 786\varepsilon^{0.46}$ MPa, where the parameters were determined from uniaxial tension tests. Coulomb friction was assumed in all interfaces with the friction coefficient $\mu = 0.1$. The die and the blank holder had all degrees of freedom locked whereas the punch was free to move in radial direction, had all rotational degrees of freedom locked and was prescribed a vertical displacement. To capture the effect of a slight tilt it was essential to have very little node penetration in the die – cup interface and to achieve this the contact was modelled using a constrained based contact algorithm (*CONTACT_CONSTRAINT_SURFACE_TO_SURFACE in LS-Dyna); using this contact algorithm the maximum node penetration could be kept less than 0.4 µm.

The FEM simulation carried out with the die having a circular profiled die land was besides from the geometry of the die identical to the FEM simulation carried out with the conventional die having a cylindrical die land. Both simulations were carried out using explicit time integration. To analyze the effect of a slight tilt of the punch in relation to the die simulations were carried out with the punch tilted 0.4°.

Figure 93 shows the equivalent strain distribution in the deep drawn and ironed cup made with the die with the cylindrical die (upper Figure) and made with the die having a circular profiled die land (lower Figure); in both cases the punch was tilted 0.4° in relation to the die. With the cylindrical die land the tilt causes an uneven cup height and uneven strain distribution, especially in the upper part of the cup wall. With the circular profiled die land the tilt has very little effect; the cup height is nearly even and so is the strain distribution.
It is not possible to compare qualitatively the experimental results and the results.

Figure 93 The equivalent strain distribution in the cup drawn with the die having a cylindrical die land (upper Figure) and in the cup drawn with the die having a circular profiled die land. In both cases the die was tilted 0.4° in relation to the punch.
obtained from the FEM-simulations. The tools in the FEM simulations were modelled as rigid, but as can be seen from Figure 86, the elastic deformation of the tools has a significant influence on the cup wall thickness, and in the FEM simulations the die was modelled as floating which it was not in the real process. However, qualitatively there is good agreement between the variations in cup height measured experimentally, Figure 83, and the results obtained from the FEM simulation, Figure 93 (upper Figure).

### 12.7 Conclusions

The measurements of the variation in cup wall thickness suggest that the punch in the real process has been aligned off-centre in relation to the die. The 2D plane strain simulations indicate that it is possible by offsetting the punch to obtain a nearly even cup height even if the die is slightly tilted in relation to the punch. The 2D plane strain simulation results thus indicate that evenness of the cup height should not be used as a quality measure; it may be possible to produce a cup with even cup height and still have variation in the wall thickness in circumferential direction, variation which may cause severe problems in the subsequent ironing stages.

The 3D FEM simulations show that a slight tilt of the die in relation to the punch has a significant influence on the quality of the cup produced when the die is made with a cylindrical die land (which is common practice); a slight tilt causes an uneven cup height and variation in the wall thickness in circumferential direction. The variation in cup height, caused by a slight tilt of the punch, predicted from the FEM simulation, is in good agreement with the height variation measured experimentally, and the FEM simulation thus indicate that the cup height variation in the real cups may have been caused by a slight tilt of the die in relation to the punch.

It is suggested to make the die with the die land circular profiled, because the FEM simulations show that with a circular profiled die land, a slight tilt of the die in relation to the punch will only cause minor changes in the contact conditions between die land and cup wall and that the deep drawing and ironing process thus becomes significantly more robust; the quality of the cup produced is nearly insensitive to a slight tilt of the die in relation to the punch.

The increased robustness of the deep drawing and ironing process when using a circular profiled die land in place of a cylindrical die land may have several important practical implications:

- Easier mounting of the tools in the press.
- Fewer problems when running in a new tool. In the opinion of the author, the reason why the craftsmen on the shop floor can make a conventional die work just by polishing the die is that they unintentionally by polishing the die land makes this slightly curved and thus make the die more robust with a regard to a slight tilt of the die.
- The stroke rate of the transfer press can be changed without affecting the quality of the cup produced. When the stroke rate is changed so is the elastic deformation of the tools and press, and a change in the elastic deformation may cause the deep drawing and ironing process to become unstable. If the die land is made circular profiled, a small tilt of the die in relation to the punch, caused by elastic deformation, will only have minor influence on the quality of the cups produced.

In the opinion of the author, the analysis carried out demonstrates that observations on the shop floor (how production problems are solved), experimental results (measurement of cup height variation and measurements of variation in cup wall thickness) combined with FEM simulations can lead to a substantial better understanding of why production problems occur and that it based on this better understanding in this case has been
possible to, by very simple means, to significantly improve the robustness of the deep drawing and ironing process.

The idea of using a die having a circular profiled die land in place of the cylindrical die land is to the knowledge of the author new, and as the use of a die having a circular profiled may improve the robustness of the deep drawing and ironing process significantly, Aalborg University has applied for a patent, which encompasses the use of a die having a non-cylindrical die land employed in a combined deep drawing and ironing process [75*].
13 Backward can extrusion

The work dealing with backward can extrusion summarized in Chapter 13 to Chapter 19 is mainly based on the research report [76**]. Some of the results have been published in [77*-80*]

13.1 Common punch design in backward can extrusion

The punch used in backward can extrusion is commonly made according to the recommendations of the International Cold Forging Group (ICFG) [59]. In Figure 94 is shown the geometry of the punch nose as recommended by the ICFG for punches used for backward can extrusion of steel components. It can be seen from Figure 94 that the ICFG recommends a cylindrical punch land and that the height $h$ of the punch land should be from $0.3\sqrt{D_p}$ to $0.7\sqrt{D_p}$, where $D_p$ is the diameter of the punch land. The design of the punch face has a significant influence on the material flow and thus a significant influence on the tendency to break down of the lubricant and on pick up. Bay et al [81] carried out an experimental investigation of the backward can extrusion of steel at low reduction ratios and concluded that ICFG’s recommendations are inappropriate for low reduction ratios.

![Figure 94](image.png)

Figure 94 The punch geometry recommended by the ICFG. $h$ is the height of the cylindrical punch land [59].

The author has not been able to find specific recommendations for the design of punches used for backward can extrusion of aluminium. To the knowledge of the author punches used for backward can extrusion of aluminium are commonly made with a cylindrical punch land as recommended by the ICFG for the backward can extrusion of steel.

13.2 Slab analysis of the radial pressure on the punch land

The radial pressure on the punch land was determined using the slab method assuming that the deformation in the region near the punch land takes place under plane strain conditions; an assumption, which is reasonable if the can wall thickness is small compared to the radius of the punch. Assuming that the stresses in the can wall above the
punch land are zero, the radial force $P$ on the punch land per unit width in circumferential direction can easily be obtained as

$$P \approx 2kh \left[ 1 + \frac{\mu h}{t} \right]$$  \hspace{1cm} (6)

$$P \approx 2kh \left[ 1 + \frac{mh}{2t} \right]$$  \hspace{1cm} (7)

where Equation (6) is for Coulomb friction with friction coefficient $\mu$. Equation (7) is obtained if constant friction with the friction factor $m$ is assumed. $k$ is the shear yield stress of the can material, $h$ the height of the punch land (see Figure 94) and $t$ the thickness of the can wall. From Equation (6) and (7) it can be seen that the radial force
- is proportional to the shear yield stress of the can material
- increases with increasing friction and increasing value of $h/t$

13.3 The Influence from a slight tilt of the punch land

A slight tilt of the punch land may be caused by e.g. inaccurate machining, inaccurate mounting of the tools in the press and/or elastic deformation of the tools and/or press during the extrusion. Figure 95 shows a sketch of the backward can extrusion process where the punch land is slightly tilted (the tilt angle is highly exaggerated in the Figure). Due to the tilt contact between punch land and can wall is completely lost on one side (side A) whereas a slight tilt will only have a marginal effect on the opposite side (side B). The loss of contact on side A reduces the radial force $P$ (per unit width) on the punch land to zero whereas on side B the radial force remains nearly unaffected and is given by Equation (6) or (7). This difference in radial force on the punch land will try to deflect the punch to the right causing a decrease in the extruded wall thickness on the right side (side A) and a corresponding increase in the wall thickness on the opposite side (side B).

If the punch land is made conical (either choked as shown in Figure 96 a or relieved as shown in Figure 96 b) or profiled, e.g. circular profiled as shown in Figure 96 c, a slight tilt of the punch land is expected to give rise to only minor changes in the radial forces on the punch land.
Figure 96 Sketch of a punch with a choked punch land (a), with a relieved punch land (b) and with a circular profiled punch land (c).
14 2D FEM analysis of the effect of a slight tilt of the punch

14.1 The backward can extrusion processes modelled

The backward can extrusion carried out with a cylindrical punch land and with a circular profiled punch land was simulated in 2D (as plane strain and rotational symmetric). With both punch geometries simulations were carried out with a slightly tilted punch in order to investigate the effect, which a slight tilt of the punch has on the radial and vertical forces on the punch.

![Figure 97 Sketch of the backward can extrusion process carried out with the punch with a cylindrical punch land.](image)

Figure 97 shows the backward can extrusion process carried out with the punch with the cylindrical punch land. The height of the punch land is 2 mm and the nominal thickness of the can wall 1 mm. Simulations were carried out with the punch rotated 0.4° and -0.4° respectively around the rotation point indicated in Figure 97.

In Figure 98 is shown the backward can extrusion carried out with the punch with a circular profiled punch land. The radius of the circular profiled punch land is 10 mm and the centre of the profiled punch land is placed such that the vertical length of the circular profiled punch land, which will come into contact with the can wall, is 2 mm (if the punch is not tilted). Besides from the geometry of the punch everything else is the same as in the backward can extrusion process shown in Figure 97. With the circular profiled punch land, Figure 98, simulations were carried out with the punch rotated 0.4° and -0.4°, respectively, around the rotation point shown in Figure 98.
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Figure 98 Sketch of the backward can extrusion carried out with the punch with a circular profiled punch land.

14.2 FEM model

All the simulations have been carried out with the FEM code LS-Dyna version 971, revision 1477 (double precision).

Figure 99 shows the FEM-model of the backward can extrusion carried out with the punch with a cylindrical punch land. Due to symmetry, the FEM-model was made as a half model. Besides from the geometry of the punch, the FEM model of the backward can extrusion process carried out with the circular profiled punch land was identical to this FEM model.

All tool parts were made as rigid bodies with 4 node elements. The specimen was modelled as elastic perfectly plastic material (E-modulus = 200000 MPa, Poisson's ratio = 0.3, yield stress = 100 MPa). When carrying out the simulation explicitly, a 4 node element with one integration point was used and when carrying out the simulations implicitly, a 4 node element with 4 integration points was used.
14.2.1 Constraints

The container wall and the container bottom had all degrees of freedom locked. The punch nose and the punch land were prescribed a vertical displacement; all other degrees of freedom were locked. The nodes on the axis of symmetry were constrained not to move in radial direction.

14.2.2 Friction conditions

The friction in the contact surfaces was modelled as a combination of Coulomb friction with the friction coefficient $\mu = 0.15$ and constant friction with the friction factor $m = 0.15$ as follows:

$$
\tau = \mu \cdot p = 0.15 \cdot p, \quad p < 57.73 \text{ MPa}
$$

$$
\tau = m \cdot k = 8.86 \text{ MPa}, \quad p \geq 57.73 \text{MPa}
$$

where $p$ is the specific contact pressure and $k$ the shear yield stress of the can material.

The friction factor $m = 0.15$ was determined experimentally by Hansen [82] when using the commercial lubricant "Glisapal" from the company Chemetall; "Glisapal" was used in the experimental study of the backward can extrusion of a thin walled aluminium can discussed in Chapter 16.

In LS-Dyna there are a number of contact algorithms and the one used in all the simulations was *CONTACT_AUTOMATIC_2D_SURFACE_TO_SURFACE.

14.3 Implicit versus explicit time integration

The simulation of the backward can extrusion process with the cylindrical punch land and with the punch tilted $-0.4^0$ (positive rotation = anti clockwise direction) was carried
out using both explicit and implicit time integration. The simulations were carried assuming plane strain conditions.

With explicit time integration the simulation time was 0.01 sec, and mass scaling was used by specifying a minimum time step size of 1.0e-7 sec. With implicit time integration, the simulation time was 100 sec. and an automatic time stepping scheme was employed with a maximum time step size of 0.2 sec; that is the simulation was carried out in 500 time steps if no convergence problems were encountered.

To overcome problems with mesh distortion remeshing was employed. During each simulation 100 remeshings were done. The number of elements in the specimen was of the order 7500. The size of the elements in the rigid tool parts was significantly smaller than the elements in the specimen.

In Figure 100 and Figure 101 are shown examples of the equivalent strain distribution in the specimen obtained using explicit and implicit time integration respectively. As it can seen from these Figures the equivalent strain distributions obtained are nearly identical. The CPU time for carrying out the simulations was also nearly identical; approximately 1½ hour on a 2.6 MHz Linux PC.

Figure 100  Equivalent strain distribution in the specimen when using explicit time integration.
Figure 101  Equivalent strain distribution in the specimen when using implicit time integration.

Figure 102  Radial force on the punch nose and on the punch land determined using explicit and implicit time integration. Punch tilt: -0.4°.
In Figure 102 are shown the radial forces on the punch nose and on the punch land as function of the punch displacement obtained using explicit and implicit time integration and it can be seen that there is significantly more noise in the forces obtained using explicit time integration than when using implicit time integration. In this investigation focus was on the contact forces, especially on the punch land, and it was thus decided to carry out all simulations using implicit time integration, provided that the simulations could be made to converge. Unfortunately it is not always possible to foresee if a given forming problem will cause convergence problems when using implicit time integration. In LS-Dyna it is possible to automatically switch from implicit to explicit time integration if severe convergence problems are encountered (and back again to implicit when the convergence problems have been overcome). However it was not necessary to use this facility in the simulations.

14.3.1 Choice of penalty factor

The penalty factor in the contact, especially the contact in the punch nose - can interface and in the punch land - can interface, should be chosen such that on the one hand to have as little node penetration as possible and on the other hand to avoid convergence problems. To find a suitable choice of penalty factor the simulation of the can extrusion, carried out with the punch with a circular profiled punch land (see Figure 98) tilted 0.4°, was carried out with different penalty factors (changing the penalty factor was accomplished by changing the parameters sfact in the contact definition in the input file to LS-Dyna). Simulations were carried out with the following values of sfact in the punch nose - can interface and in the punch land - can interface: 0.1, 1, 3, 8, 15.

With sfact = 0.1 there was severe node penetration as can be seen in Figure 103.

![Figure 103: Node penetration in the punch nose - can interface when sfact = 0.1.](image)

In Figure 104 are shown the radial force on the punch land as function of the punch displacement for the different values of sfact (sfact = 1, 3, 8, 15). There is a lot of noise in all the curves independently of the value of sfact; the noise level is for all values of sfact...
significantly larger than the noise level for the force on the cylindrical punch land (obtained using implicit integration) shown in Figure 102. Due to the noise it is very difficult to distinguish the curves from each other in Figure 104.

Circular profiled punch land (H2R10), plane strain, tilt: 0.4

**Figure 104** Radial force on the punch land as function of the punch displacement with different values of sfact as parameter.

After smoothing (average 8)

**Figure 105** Radial force on the punch land (after smoothing) as function of the punch displacement with different values of sfact as parameter.
In Figure 105 are shown the same curves as in Figure 104, but smoothed. After smoothing there is only a slight difference in the curves for the different values of sfact. To begin with, at a punch displacement of approximately 1.5 mm, the radial force is somewhat larger for sfact = 1 than for the other values; this difference can also, despite the noise, be seen in Figure 104. There are also some differences between the curves for a punch displacement larger than 3 mm. For some reason or another, the simulation with sfact = 1 took approximately 3 times as long CPU time as with sfact = 3, 8 and 15, which all took approximately 1 hour 30 minutes CPU time. It was thus decided to carry out all the plane strain simulations with sfact = 3 in all the contact interfaces.
15 2D FEM results

15.1 Simulations carried out assuming plane strain

15.1.1 Cylindrical punch land (2D plane strain simulations)

In Figure 106 are shown the radial forces on the punch nose and on the punch land as function of the punch displacement for the punch untilted and tilted 0.4° and -0.4°. There is some noise in the curves for the radial force on the punch land, especially for tilt angle 0.0° and 0.4° and in Figure 107 are shown the curves for the radial forces on the punch land after smoothing. With a tilt angle of +0.4° there should be no contact between the punch land and can wall and the radial force should therefore be very close to zero. The reason why the FEM simulation predicts a radial force, which is not completely zero, is believed to be due to a slight node penetration in the region where the punch nose and punch land meet. Changing the tilt angle of the punch to 0.0° and -0.4° increases the radial force on the punch land significantly (the radial force for 0.0° may not be reliably predicted; as mentioned in Chapter 9.2.1 it may be difficult to determine the contact forces correctly when the material slides parallel to a straight surface). With a tilt angle of -0.4°, the radial force on the punch land increases to approximately 275 N (per mm width) (for a punch displacement = 2 mm). The radial force on the punch land should according to Equation (6) be 300 N, and according to Equation (7) 265 N. The FEM simulation predicts a radial pressure on the punch land which is greater than 75 MPa. The predicted pressure is thus larger than 1.3 times the shear yield stress of the cup material k and with such a high pressure, Coulomb's friction law will overestimate the radial force and Equation (7) should be used to determine the radial force using the slab method. The radial force on the punch land is in good agreement with the radial force predicted using the law of constant friction, Equation (7). From Figure 106 it can be seen that changing the tilt angle from 0.4° to 0.0° and -0.4° increases the radial force on the punch nose from approximately 380 N to 450 N (per unit width).

In Figure 108 and Figure 109 are shown the vertical forces on the punch land and on the punch nose as function of the punch displacement with the punch tilted 0.0°, 0.4° and -0.4°, respectively. Changing the tilt angle from 0.4° to 0.0° or -0.4° increases the vertical force on the punch nose from approximately 765 N to approximately 840 N. The vertical force on the punch land is very little compared to the vertical force on the punch nose, independently of the tilt angle; changing the tilt angle from 0.4° to -0.4° increases the vertical force on the punch land from approximately 0 N to approximately 18 N. The change of tilt angle has a much larger effect on the vertical force on the punch nose, which as mentioned above, changes from approximately 765 N to approximately 840 N.
Figure 106 Cylindrical punch land (plane strain). The radial force on the punch land and punch nose as function of punch displacement with the punch untilted and tilted 0.4° and -0.4°.

Figure 107 Cylindrical punch land (plane strain). The radial force on the punch land as function of the punch displacement with the punch untilted and tilted 0.4° and -0.4°. The curves have been smoothed.
15.1.2 Circular profiled punch land (2D plane strain simulations)

In Figure 110 and Figure 111 are shown the radial forces on the punch nose and on the circular profiled punch land as function of punch displacement with the punch tilted 0.4° and -0.4°. From Figure 110 it can be seen that there is significant noise in the radial force on the punch land for both tilt angles. To be able to better distinguish the radial force on the punch land as function of tilt angle, the curves for the radial force on the punch land were smoothed, and the smoothed curves are shown in Figure 111. The noise in the
curves for the radial force on the punch land may be due to the fact that the punch land becomes nearly vertical where the can wall loses contact with the punch land. Another thing which may have contributed to the noise is the fact that the can wall became curved as shown in Figure 112 and from the simulations it could be seen that the can wall above the punch land moved slightly back and forth in radial direction and this movement back and forth may have contributed to the noise. With the cylindrical punch land the can wall was nearly straight independently of the tilt angle of the punch.

From Figure 110 it can be seen that changing the tilt angle of the punch from $0.4^\circ$ to $-0.4^\circ$ only has little effect on the radial force on the punch nose and from Figure 111 also limited effect on the radial force on the punch land; the radial force on the punch land increases from approximately 200 N for the tilt angle $= 0.4^\circ$ (at a punch displacement $= 2$ mm) to approximately 225 N for the tilt angle $= -0.4^\circ$.

In Figure 113 and Figure 114 are shown the vertical force on the punch nose and on the circular profiled punch land as function of punch displacement with the punch tilted $0.4^\circ$ and $-0.4^\circ$. From Figure 113 it can be seen that changing the tilt angle from $0.4^\circ$ to $-0.4^\circ$ only has a very small effect on the vertical force on the punch nose. It can also be seen that the vertical force on the punch land is very little compared to the vertical force on the punch nose, independently of the tilt angle of the punch. There is, as can be seen in Figure 114, significant noise in the vertical force on the punch land. To make it easier to distinguish the curves, the curves were smoothed and the smoothed curves are in Figure 114 shown as the thick curves; it can be seen that changing the tilt angle from $0.4^\circ$ to $-0.4^\circ$ increases the vertical force on the punch land approximately 6 N.

![Circular profiled punch land (H2R10), plane strain](image)

**Figure 110** Circular profiled punch land, (plane strain). The radial force on the punch nose and on the punch land as function of punch displacement with the punch tilted $0.4^\circ$ and $-0.4^\circ$. 
Analysis of deep drawing, ironing and backward can extrusion

Figure 111  Circular profiled punch land, (plane strain). The radial force on the punch land (after smoothing) as function of punch displacement with the punch tilted 0.4° and -0.4°.

Figure 112 Plane strain extrusion with the circular profiled punch land. The can wall above the punch curls.
Figure 113  Circular profiled punch land (plane strain). The vertical force on the punch nose and on the punch land as function of punch displacement with the punch tilted 0.4° and -0.4°.

Figure 114  Circular profiled punch land (plane strain). The vertical force on the punch land as function of punch displacement with the punch tilted 0.4° and -0.4°. The thick curves are smoothed curves.

15.2 Simulations carried out assuming rotational symmetry

The simulations of the backward can extrusion with the cylindrical punch land and with the circular profiled punch land were also carried out as 2D rotational symmetric. The main differences in the 2D rotational symmetric simulations compared to the simulations carried out assuming plane strain were:

- the punch displacement was increased from 3.75 mm to 4.5 mm
- the penalty factor (sfact) in the punch land - specimen interface and in the punch nose - specimen interface was increased from 3 to 8.
15.2.1 Cylindrical punch land (2D rotational symmetric simulations)

In Figure 115 and Figure 116 are shown the radial and vertical forces on the punch nose and on the punch land as function of the punch displacement for the punch tilted 0.0°, +0.4° and -0.4° respectively for the backward can extrusion carried out with the punch with the cylindrical punch land simulated as rotational symmetric.

The radial force on the punch land for a tilt angle of 0.4°, Figure 115, is quite noisy; it is believed that the noise is mainly due to a slight node penetration, especially in the region where the punch nose and punch land meets. Changing the tilt angle from 0.4° to -0.4° changes the radial force on the punch land from close to 0 N to approximately 800 N (for a punch displacement = 3.5 mm) and changes the radial force on the punch nose from approximately 850 to 975 N. The curve for the radial force on the punch land for a tilt angle of 0.0° is also rather noisy and the main reason for the noise is also believed to be slight node penetration.

From Figure 116 it can be seen that the vertical force on the punch land is small compared to the vertical force on the punch nose, independently of the tilt angle. Changing the tilt angle from +0.4° to 0.0° or -0.4° increases the vertical force on the punch nose (for a punch displacement = 3.5 mm) from approximately 925 N/radian to approximately 1010 N/radian.

![Figure 115 Cylindrical punch land (rotational symmetric). The radial force on the punch nose and on the punch land as function of the punch displacement for the punch tilted 0.0°, +0.4° and -0.4°, respectively.](image-url)
15.2.2 Circular profiled punch land (2D rotational symmetric simulations)

In Figure 117 are shown the radial forces on the punch land and on the punch nose as function of the punch displacement obtained from the simulations of the backward can extrusion carried out with the circular profiled punch land assuming rotational symmetry. The curves for the forces on the punch land are somewhat noisy; the main reason for the noise is believed to be slight node penetration. Contrary to the simulations carried out assuming plane strain no tendency to curling of the can wall was observed in these simulations. To better see the effect of the tilt of the punch the curves for the force on the punch land were smoothed and the smoothed curves are shown in Figure 118. From Figure 118 it can be seen that changing the tilt angle of the punch from $+0.4^0$ to $-0.4^0$ increases the radial force on the punch land approximately 50 N/radian; with the cylindrical punch land, Figure 116, the same change in tilt angle gives rise to a change in the radial force on the punch land of approximately 800 N/radian, (see Figure 115).

In Figure 119 are shown the vertical forces on the punch nose and on the punch land as function of the punch displacement. It can be seen from Figure 119 that changing the tilt angle form $+0.4^0$ to $-0.4^0$ only gives rise to a very minor change in the vertical force on the punch nose. With the cylindrical punch land, Figure 116, the same change in tilt angle gives rise to change in the vertical force on the punch nose of approximately 85 N/radian (for a punch displacement of 3.5 mm).
Figure 117 Circular profiled punch land (rotational symmetric). The radial force on the punch nose and on the punch land as function of punch displacement for the punch tilted +0.4° and -0.4°, respectively.

Figure 118 Circular profiled punch land (rotational symmetric). The radial force on the punch land as function of punch displacement for the punch tilted, +0.4° and -0.4°, respectively. The curves have been smoothed.
15.3 Residual stress distribution in the can wall

To get an impression of which effect a slight tilt of the punch land may have on the residual stress distribution in the can wall the stress distribution in the can wall, obtained from the 2D rotational symmetric simulations, were analyzed.

In Figure 120 are shown the vertical stress (y-stress) distribution and the hoop stress distribution in can wall for the punch tilted +0.4° and -0.4° respectively with the extrusion carried out with the punch with the cylindrical punch land and in Figure 121 are shown the v.Mises stress distribution and the pressure distribution. In Figure 122 and Figure 123 are shown the corresponding distributions obtained with the punch with the circular profiled punch land.

It can be seen from Figure 120 and Figure 121 that in case of the punch with the cylindrical punch land a change of the tilt angle of the punch from +0.4° to -0.4° gives rise to changes in the residual stress distributions, whereas in case of the punch with the circular profiled punch land, Figure 122 and Figure 123, the residual stress distributions are nearly unaffected by the tilt of the punch.

The 2D rotational symmetric simulations thus indicate that the residual stress distribution will be more affected due to a slight tilt of the punch land if the punch is made with a cylindrical punch land compared to if the punch is made with a circular profiled punch land.

The 2D rotational symmetric simulations cannot be used to say anything quantitatively about the residual stress distribution in the can wall in a real backward can extrusion process. In the real process a slight tilt of the punch land will, especially if the punch land is cylindrical, give rise to an uneven pressure distribution on the punch land, which will deflect the punch off-centre leading to an unsymmetrical material flow, and which effect an unsymmetrical material flow may have on the residual stress distribution cannot be analyzed (and taken into account) in the 2D – rotational symmetric simulations. In order to be able to get a reliable prediction of the residual stress distribution in the can wall, the FEM simulations must be carried out as 3D.

Figure 119 Circular profiled punch land (rotational symmetric). The vertical force on the punch nose and on the punch land as function of punch displacement for the punch tilted +0.4° and -0.4°, respectively.
Figure 120 The vertical stress distribution (y-stress) and the hoop stress distribution in the can extruded with the punch with the cylindrical punch land. To the left with the punch tilted $+0.4^0$ and to the right tilted $-0.4^0$. 

Vertical stress  
Tilt: $+0.4^0$  

Vertical stress  
Tilt: $-0.4^0$  

Hoop stress  
Tilt: $+0.4^0$  

Hoop stress  
Tilt: $-0.4^0$
Figure 121 The v.Mises stress distribution and the pressure distribution in the can extruded with the punch with the cylindrical punch land. To the left with the punch tilted +0.4° and to the right tilted -0.4°.
Figure 122 The vertical stress distribution ($y$-stress) and the hoop stress distribution in the can extruded with the punch with the circular profiled punch land. To the left with the punch tilted $+0.4^0$ and to the right tilted $-0.4^0$. 
Figure 123 The v.Mises stress distribution and the pressure distribution in the can extruded with the punch with the circular profiled punch land. To the left with the punch tilted +0.4° and to the right tilted -0.4°.
15.4 Main results from the 2D simulations

The main results from the 2D FEM simulations are summarized in Table 7, Table 8, Figure 124 and Figure 125.

<table>
<thead>
<tr>
<th></th>
<th>Radial force on punch nose (N/mm width)</th>
<th>Radial force on punch land (N/mm width)</th>
<th>Vertical force on punch nose (N/mm width)</th>
<th>Vertical force on punch land (N/mm width)</th>
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<td><strong>Cylindrical punch land</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
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<td>~275</td>
<td>~840</td>
<td>~18</td>
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<tr>
<td>difference</td>
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<td>~255</td>
<td>~75</td>
<td>~18</td>
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<td></td>
<td></td>
</tr>
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<td>~225</td>
<td>~740</td>
<td>~43</td>
</tr>
<tr>
<td>tilt: +0.4°</td>
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<td>~200</td>
<td>~740</td>
<td>~38</td>
</tr>
<tr>
<td>difference</td>
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<td>~25</td>
<td>~0</td>
<td>~5</td>
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Table 7 The radial and vertical forces on the punch nose and on the punch land obtained from the plane strain simulations. The forces are for a punch displacement = 2 mm.

<table>
<thead>
<tr>
<th></th>
<th>Radial force on punch nose (N/radian)</th>
<th>Radial force on punch land (N/radian)</th>
<th>Vertical force on punch nose (N/radian)</th>
<th>Vertical force on punch land (N/radian)</th>
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<td></td>
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<td>~780</td>
<td>~85</td>
<td>~50</td>
</tr>
<tr>
<td><strong>Circular profiled punch land</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>tilt: -0.4°</td>
<td>~740</td>
<td>~840</td>
<td>~860</td>
<td>~130</td>
</tr>
<tr>
<td>tilt: +0.4°</td>
<td>~750</td>
<td>~780</td>
<td>~870</td>
<td>~115</td>
</tr>
<tr>
<td>difference</td>
<td>~10</td>
<td>~60</td>
<td>~10</td>
<td>~15</td>
</tr>
</tbody>
</table>

Table 8 The radial and vertical forces on the punch nose and on the punch land obtained from the rotational symmetric simulations. The forces are for a punch displacement = 3.5 mm.
Figure 124 Plane strain simulations. The radial force on the punch land as function of the punch displacement with the shape of the punch land as parameter.

Figure 125 Rotational symmetric simulations. The radial force on the punch land as function of the punch displacement with the shape of the punch land as parameter.
In Table 7 are listed the radial and vertical forces on the punch nose and on the punch land obtained from the plane strain simulations and in Table 8 are listed the corresponding results obtained from the rotational symmetric simulations.

From the results listed in Table 7 and Table 8 it is clear that the tilt of the punch with the cylindrical punch land has a significant influence on the radial force on the punch land. In the plane strain simulations the change in tilt angle from -0.4° to +0.4° gives rise to a change in the radial force on the punch land of approximately 255 N/mm width (from ~275 N/mm width to ~20 N/mm width). In the rotational simulations the corresponding change is approximately 780 N/radian (from ~800 N/radian to ~20 N/radian).

With the circular profiled punch land the change in radial force is significantly smaller. In the plane strain simulations, the change in tilt angle from -0.4° to 0.4° gives rise to a change in the radial force on the punch land of approximately 25 N/mm width (from ~225 N/mm width to ~200 N/mm width). In the rotational symmetric simulations the corresponding change is approximately 60 N/radian (from ~840 N/radian to ~780 N/radian).

It can also be seen from the results in Table 7 and Table 8 that the changes in the radial forces on the punch nose and the vertical forces on the punch land and punch nose, due to the tilt, are much smaller than the changes in the radial forces on the punch land.

Figure 124 shows the radial force on the punch land (after smoothing) as function of punch displacement and with the shape of the punch land as parameter for the simulations carried out assuming plane strain and Figure 125 shows the corresponding force curves (also after smoothing) for the simulations carried out assuming rotational symmetry. The significant change in the radial force on the punch land due to the change in tilt angle, when the punch land is cylindrical compared to when the punch land is circular profiled, is apparent from Figure 124 and Figure 125.

Both the plane strain simulations and the rotational symmetric simulations thus indicate that

• if the punch is made with a cylindrical punch land, a slight tilt of the punch may give rise to a significant change in the radial force on the punch land

In the opinion of the author the backward can extrusion process is, when carried out with a punch with a cylindrical punch land, in a state of unstable equilibrium; if the punch land for some reason or another is slightly tilted, e.g. due to inaccurate machining, inaccurate alignment of the tools in the press and/or elastic deformation of the tools and press, the punch will be subjected to net radial force, which will try to deflect the punch off-centre leading to an uneven can wall thickness in the extruded can wall.

The simulations also indicate that

• if the punch is made with a circular profiled punch land, a slight tilt of the punch land will only give rise to a minor change in the radial force on the punch land; the resulting net radial force on the punch is thus significantly smaller with a circular profiled punch land compared to when a punch with a cylindrical punch land is employed.

and the simulations thus also indicate that

• the backward can extrusion process is significantly more robust when a punch with a circular profiled punch land is employed compared to when a punch with a cylindrical punch land is employed.
16 Experimental investigation

The backward extrusion of the can shown in Figure 126 was analyzed experimentally.

Figure 126 Backward extruded aluminium can. To the left is shown the slug.

The can is made from pure aluminium, has an outside diameter of 73 mm, a nominal can wall thickness of 0.915 mm and a height of approximately 250 mm. The nominal reduction of area is 0.950. The can is made from a cylindrical slug with a diameter of 72.8 mm and a height of 15.1 mm. The slug is lubricated with the lubricant “Glisapal” from Chemetall. According to Hansen [82] this lubricant yields a friction factor \( m \) of 0.15.
Figure 127 Tool setup used for the manufacturing of the aluminium cans.

16.1 Tool setup

The can is produced by a Danish company on a 500 t hydraulic press. The tool used for the manufacturing of the aluminium can is sketched in Figure 127.
The punch is screwed onto the punch shaft and accurate alignment of the punch in relation to the punch shaft is ensured by a guide bushing. Accurate guiding and centring of the punch nose and punch shaft in relation to the container is ensured by the punch guide, which also controls the maximum punch displacement. The container is made in two, the upper container and the lower container. The lower container is shrink fit in a shrink ring, whereas the upper container is a mono container. The design of the container may not be the most brilliant and this will be discussed in more detail in Chapter 17.

Aluminium cans were produced with two different punches. The one punch having a cylindrical punch land (this punch design has been used hitherto in the industrial production of the cans) and the other punch having a circular profiled punch land. The two punch designs are shown in Figure 128. In Figure 129 the punch land of the two punches are drawn together, and as it can be seen, the difference is very small.

![Figure 128 The geometry of the punches used in the experiments. To the left the punch with the circular profiled punch land and to the right with the cylindrical punch land.](image)

The diameters of the punches and the punch heights as well as roundness, flatness and cylindricity were measured on a coordinate measuring machine. The main measured results with regard to accuracy are listed below:

**Punch with the cylindrical punch land:**

- Flatness of punch face: 5.8 μm
- Parallelism of punch face in relation to punch top: 8.2 μm
- Diameter of punch land: ø71.17 mm
- Coaxiality of punch land in relation to punch face: 7.9 μm

**Punch with the circular profiled punch land:**

- Flatness of punch face: 1.3 μm
- Parallelism of punch face in relation to punch top: 2.1 μm
Max diameter of punch land: ø71.15 mm

The punch noses were also measured with a very accurate profile measuring machine (Surfascan) and the dimensions besides from the diameters and the height of the punch shown in Figure 128 were obtained from the profile measurements. The punch noses were polished and the roughness Ra was measured to be less than 0.05 µm.

16.2 Experimental procedure

The cans were produced with the tool setup shown in Figure 127. The tool was mounted in a 500 t hydraulic press. The aluminium slugs were lubricated with the lubricant “Glisapal” from Chemetall; no other kind of lubrication was employed.

The actual production of the aluminium cans was carried out using a punch with a cylindrical punch land (see Figure 128). 10 cans were picked at random from the actual production. After the actual production run was finalized, the punch with the cylindrical punch land was unscrewed from the punch shaft and the punch with the circular profiled punch land was screwed onto the punch shaft. 12 cans were then produced with the punch with the circular profiled punch land.

The cans produced with the conventional punch with the cylindrical punch land and the cans produced with the punch with the circular profiled punch land were produced with

- The same slug material
- The same slug dimensions
- The same lubrication and same lubrication procedure
- The same tool (besides from the punch) and the same press
- The same pressing speed
- The same tool setup. The tool was mounted in the press when the punch with the circular profiled punch land was replaced with the punch with the circular profiled punch land
Main emphasis was placed on keeping everything the same (besides from the punch) when producing the cans with the punch with the cylindrical punch land and the cans with the punch with the circular profiled punch land. It is thus believed that any systematic difference between the cans produced with the punch with the cylindrical punch land and the cans produced with the punch with the circular profiled punch land can be attributed to the difference in the geometry of the punch land.

### 16.3 Measuring procedure

Figure 130 shows the 10 cans picked from the actual production produced with the punch with the cylindrical punch land and Figure 131 the 12 cans produced with the punch with the circular profiled punch land. As it can be seen from the pictures all the cans look pretty much the same; the height of the cans are nearly the same and the can rim is fairly even. The largest difference between the highest and lowest point on the can rim on each can was less than 2 mm for all the cans. By just visual inspection no systematic difference between the cans produced with the cylindrical punch land and the cans produced with the punch with the circular profiled punch land could be observed.

All the cans were measured using a coordinate measuring machine. On the measuring machine the can was fixated in vertical position in a fixture. Two circles were scanned; one approximately 100 mm below the can rim and the other approximately 200 mm below the can rim and the can was aligned using the centres of theses two circles with z = 0 at the lowest point on the can rim. The can wall was probed for every 2 mm from z = 0 to z = -209 mm on the inside and outside in 0°, 90°, 180° and 270° (0° = the direction with the lowest can height). The can wall was also probed by scanning circles on the inside and outside of the can wall from 5 mm below the can rim and for every 25 mm down to z = -200 (200 mm below the can rim).

Vickers hardness measurements were also carried out on one can. The Vickers hardness were measured (on can B3 made with the circular profiled punch land) just below the can rim and for every 20 mm in vertical direction down to 240 mm below the can rim.

![Figure 130 The 10 cans produced with the conventional punch with the cylindrical punch land.](image-url)


16.4 Results

16.4.1 Vickers hardness measurements.

Figure 132 shows the measured Vickers hardness in the can wall of a can (made with the punch with the circular profiled punch land) as function of the distance from the can rim. The mean hardness is HV\textsubscript{5} = 48.16 (kp/mm\textsuperscript{2}) with the standard deviation = 1.15 kp/mm\textsuperscript{2}. No systematic change in hardness as function of the distance from the can rim can be seen. According to Tekkaya [83] a hardness of 48.16 kp/mm\textsuperscript{2} corresponds to a yield stress $\sigma_y$ of

$$\sigma_y = 9.81 \frac{48.16}{2.475} = 190.89 \approx 191 \text{ MPa} \quad (8)$$

In the above determination of the yield stress from the Vickers hardness, the strain induced by the hardness indentation (= 0.112 according to [83]) has been neglected as it was assumed that the true stress strain curve is very flat at the high equivalent strain level, which the material in the can wall was strained to due to the backward can extrusion.

![Image of Vickers hardness measurements](image_url)

*Figure 132 The Vickers hardness in the can wall as function of the distance from the can rim. The can was made with the punch with the circular profiled punch land (can B3).*
16.4.2 Results obtained by measuring the cans on a 3D coordinate measuring machine.

Measurements carried out in $0^0$, $90^0$, $180^0$ and $270^0$ in vertical direction.

Figure 133 and Figure 134 show the measured outside radius of the cans produced with the punch with the cylindrical punch land and Figure 135 and Figure 136 the outside radius of the cans produced with the circular profiled punch land. For all the cans there is a significant variation in the outside radius in the upper part of the cans; from 0 mm to approximately 100 mm below the can rim. The significant larger variation from 0 to 100 mm below the can rim than from 100 mm below and to the bottom is partly due to the alignment procedure; on the coordinate measuring machine the can was aligned by measuring two circles, one 100 mm below the can rim and the other 200 mm below the can rim. However, the results show that the cans are not cylindrical. It is difficult to identify a systematic difference in the cans produced with the two punches from the results shown in Figure 133 to Figure 136.

Figure 137 and Figure 138 show the can wall thickness in the four different directions on the cans made with the cylindrical punch land and Figure 139 and Figure 140 show the can wall thickness in the four different directions in the cans made with the punch with the circular profiled punch land. The average of the can wall thickness in $0^0$ and $180^0$ and in $90^0$ and $270^0$ as function of the distance from the can rim are also shown in the Figures. The difference between the average wall thickness in $0^0$ and $180^0$ and in $90^0$ and $270^0$ is for all the cans less than approximately 0.01 mm.

On some of the curves there is a shift, e.g. in Figure 139 can B2 in $0^0$ around 100 mm below the can rim. This shift is believed to have been caused by pick up of lubricant on the measuring probe. The cans were not degreased prior to measurement on the coordinate measuring machine.

The difference in can wall thickness in the different directions is for all the cans small near the can rim. The variation in can wall thickness as function of distance from the can rim and as function of the different directions has thus not been caused by an initial offset of the punch in relation to the container(s).

The typical thickness curves for the cans made with punch with the cylindrical punch land, e.g. Figure 137, can A1, show that the punch, when extruding the first 100 mm can wall, has been displaced radially (in Figure 137, can A1 mainly in the $90^0 – 270^0$ direction) and that the radial displacement reaches a maximum approximately 100 mm below the can rim. From approximately 100 mm to 210 mm below the can rim, the radial displacement is slightly diminished.

It can also be seen that the average wall thickness, which is in the range form 0.83 to 0.86 mm, is much smaller than the nominal wall thickness of 0.915 mm. This difference cannot be explained by elastic deformation due to residual stresses in the can wall, and the main reason is believed to be elastic deformation of the punch during the backward can extrusion process.

The thickness curves for the cans made with the circular profiled punch land, Figure 139 and Figure 140, are quite different from the thickness curves for the cans made with the cylindrical punch land, Figure 137 and Figure 138, both with regard to the shape of the curves and with regard to magnitude of the thickness variation. All the curves show a distinct kink around 100 mm below the can rim. The curves for the average thicknesses in $0^0$ and $180^0$ and $90^0$ and $270^0$ also show a distinct kink, which is much larger than the kink
in the average curves for the cans produced with the cylindrical punch land. By comparing Figure 138 and Figure 139 with Figure 139 and Figure 140, it is obvious that the wall thickness variation in the cans produced with the punch with the circular profiled punch land is significantly smaller than the variation in the cans produced with the cylindrical punch land. The average can wall thickness, which is in the range from 0.85 to 0.89 mm, is smaller than the nominal wall thickness of 0.925 mm. As with the punch with cylindrical punch land, the main reason for the smaller can wall thickness compared to the nominal can wall thickness is believed to be elastic deformation of the punch during the backward can extrusion process.
Figure 133 Can A1 to A 6 made with the punch with the cylindrical punch land. The outside radius as function of the distance from the can rim in the four different directions \(0^\circ, 90^\circ, 180^\circ\) and \(270^\circ\).
Figure 134 Can A7 to A10 made with the punch with the cylindrical punch land. The outside radius as function of the distance from the can rim in the four different directions (0°, 90°, 180° and 270°).
Figure 135 Can B1 to B6 made with the punch with the cylindrical punch land. The outside radius as function of the distance from the can rim in the four different directions (0°, 90°, 180° and 270°).
Figure 136 Can B7 to B12 made with the punch with the circular profiled punch land. The outside radius as function of the distance from the can rim in the four different directions (0°, 90°, 180° and 270°).
Figure 137 Can A1 to A6 made with the punch with the cylindrical punch land. The can wall thickness as function of the distance from the can rim in the four different directions (0°, 90°, 180° and 270°).
Figure 138 Can A7 to A10 made with the punch with the cylindrical punch land. The can wall thickness as function of the distance from the can rim in the four different directions (0°, 90°, 180° and 270°).
Figure 139 Can B1 to B6 made with the punch with the circular profiled punch land. The can wall thickness as function of the distance from the can rim in the four different directions (0°, 90°, 180° and 270°).
Figure 140 Can B7 to B12 made with the punch with the circular profiled punch land. The can wall thickness as function of the distance from the can rim in the four different directions ($0^\circ$, $90^\circ$, $180^\circ$ and $270^\circ$).
Can wall thicknesses determined from circles scanned on the inside and outside of the can wall.

Figure 141 and Figure 142 show the can wall thickness as function of the angle and with the distance from the can rim as parameter for the cans produced with the punch with the cylindrical punch land. The wall thicknesses were determined from circles scanned on the inside and outside of the can wall at different distances from the can rim. Figure 143 and Figure 144 show the corresponding curves for the cans produced with the circular profiled punch land. By comparing the thickness distributions shown in Figure 141 and Figure 142 with the distributions shown in Figure 143 and Figure 144 it is obvious that the cans produced with the punch with the cylindrical punch land have a significantly larger variation in the wall thickness than the cans produced with the circular profiled punch land. According to Figure 141 and Figure 142, the can produced with the punch with the cylindrical punch land showing the largest variation in wall thickness is can A10, where \( t_{\text{max}} - t_{\text{min}} \approx 0.260 \) mm and the can showing the smallest variation is can A4 where \( t_{\text{max}} - t_{\text{min}} \approx 0.218 \) mm \( (t_{\text{max}} \) and \( t_{\text{min}} \) are the maximum and minimum wall thickness). The can produced with the punch with the circular profiled punch land showing the largest thickness variation is B5 where \( t_{\text{max}} - t_{\text{min}} \approx 0.117 \) mm and B10 is the one showing the smallest variation, where \( t_{\text{max}} - t_{\text{min}} \approx 0.067 \) mm \( \) (the thicknesses measured 5 mm below the can rim have been left out in these evaluations, because it looks as if the stripping of the can from the punch in some cases has caused deformation of the can very close to the rim).

The average thickness variation \( \Delta t_{\text{ave}} \) for all the cans in each series is for the punch with the cylindrical punch land:

\[
\Delta t_{\text{ave}} = 0.238 \text{ mm (standard deviation = 0.016 mm)}
\]

and for the punch with the circular profiled punch:

\[
\Delta t_{\text{ave}} = 0.092 \text{ mm (standard deviation = 0.018 mm)}
\]

where

\[
\Delta t_{\text{ave}} = \frac{\sum_{i=1}^{n} (t_{max,i} - t_{min,i})}{n} \quad (9)
\]

\( n \): number of cans in the series
\( t_{\text{max,i}}, t_{\text{min,i}} \): the maximum, respectively the minimum wall thickness in can number i.
Figure 141 Can A1 to A6 produced with the punch with the cylindrical punch land. The can wall thickness as function of the angle with the distance from the can rim as parameter.
Figure 142 Can A7 to A10 produced with the punch with the cylindrical punch land. The can wall thickness as function of the angle with the distance from the can rim as parameter.
Figure 143 Can B1 to B6 produced with the punch with the circular profiled punch land. The can wall thickness as function of the angle with the distance from the can rim as parameter.
Figure 144 Can B7 to B12 produced with the punch with the circular profiled punch land. The can wall thickness as function of the angle with the distance from the can rim as parameter.
The can rim is on all the cans (see Figure 130 and Figure 131) fairly even and the can wall thickness near the can rim is also fairly even (besides from a few cans, where the can rim seems to have been distorted due to the stripping from the punch). It can thus be concluded that the quality of the can (how large or small the thickness variation is) can not be evaluated by looking at the variation in can height in circumferential direction or from measurements of the can wall thickness near the can rim. Even with a fairly even can rim and small variations in the can wall thickness near the can rim there can be significant variation in the can wall thickness, and this is especially true for the cans produced with the punch with the cylindrical punch land. This observation may be of importance for the implementation of the quality control in an actual industrial production of cans.
16.5 **Lubrication breakdown**

In both the cans produced with the punch with the cylindrical punch land and the cans produced with the circular profiled punch land, there is a distinct change in surface appearance of the outside can wall approximately 140 mm below the can rim. This change in surface appearance can be seen on Figure 145.

The change in surface appearance may be due to a change in lubrication conditions and or to a break down of the lubricant film. To investigate from which part of the slug the lubricant came from, cans were produced where the slug was marked with a black and a blue ink pen as shown in Figure 146.
Figure 147 shows the cans made with the punch with the cylindrical punch land and Figure 148 the cans made with the punch with the circular profiled punch land. The areas where smeared out black and blue ink can be seen are indicated on the cans. The areas with the smeared out black ink is in the transition area between the upper shiny part (with no galling) and the lower part with severe galling. The experimental results thus show that the upper shiny part has been lubricated with lubricant coming from the side of the slug, whereas the rough lower part has been lubricated with lubricant coming from the bottom of the slug; apparently the lubricant coming from the bottom has not been able to give a sufficient lubrication resulting in galling of the surface.

Figure 150 shows two cans; the can to the left produced with the punch with the cylindrical punch land (can A3) and the can to the right with the punch with the circular profiled punch land (can B10). On the cans are drawn ink lines indicating the transition between the area with no galling and the area with galling. On the can produced with the punch with the circular profiled punch land the ink line is nearly horizontal whereas the ink line is inclined on the can produced with the punch with the cylindrical punch land. This difference in inclination is due to the wall thickness variations. With the punch with the cylindrical punch land, the punch has been shifted to the right causing a decrease in the wall thickness here (in the 90 degree direction, see Figure 137 can A3), and a corresponding increase in the thickness on the opposite side. The difference in wall thickness can only be accomplished by an asymmetric material flow beneath the punch face; more of the material beneath the punch face moves to the left and the neutral axis moves to the right as indicated in Figure 149; this is believed to be the reason why the distance from the can rim to the ink line is smaller on the left side, where the can wall is thick than on the right side, where the can wall is thin. The variation in the can wall thickness in the can produced with the punch with the circular profiled punch land is much smaller (see Figure 140, can B10) and the material flow beneath the punch face has thus been nearly symmetrical.
Figure 147 Smeared out ink marks on cans produced with the punch with the cylindrical punch land.

Figure 148 Smeared out ink marks on cans produced with the punch with the circular profiled punch land.
Figure 149 A shift of the punch to the right gives rise to an asymmetric material flow beneath the punch. The neutral axis shifts to the right.

Figure 150 Can (A3) produced with the punch with the circular profiled punch land (left) and produced with the punch with the circular profiled punch land (right) (B10). The black ink line indicates the transition between the area with no galling and with galling.
17 Tool design

All the cans have a kink in the average wall thickness around 100 mm below the can rim and for most of the cans produced with the punch with the cylindrical punch land the largest difference in wall thickness is found around 100 mm below the can rim (see Figure 137 and Figure 138). In the cans produced with the punch with the cylindrical punch land (see Figure 139 and Figure 140) the kink in the average thickness distributions is for some reason or another significantly larger than in the cans produced with the cylindrical punch land.

The main reason for the distinct change in the wall thickness approximately 100 mm below the can rim is believed to be due to an inappropriate design of the containers. The parting line between the lower and upper container, see Figure 151, is placed such that the punch land during the backward can extrusion process passes the parting line; to begin with the radial pressure near the punch land acts on the upper container, whereas towards the end of the extrusion process the pressure acts on the lower container. The variation in pressure on the upper and lower container as function of punch displacement may displace the upper container in relation to the lower container is such a way that the wall thickness will change when the punch land passes the parting line. To investigate if this can explain the variation in the can wall thickness observed experimentally a FEM simulation was carried out. Figure 152 shows the FEM model. The simulation was carried out as 2D rotational symmetric. The slug was modelled only as a thin slice; if the whole slug had to be included in the FEM model this would have, with the element size used, resulted in a tremendous amount of elements. In radial direction the thin slice was constrained between a rigid wall and the container.

The slug material was modelled as an elastic - linear hardening material with yield

![Figure 151 The parting line between upper and lower container.](image-url)
Analysis of deep drawing, ironing and backward can extrusion

stress 191.64 MPa (as determined from hardness measurements on an aluminium can) and the tangent modulus = 10 MPa (the modelled material was thus very close to an elastic – ideal plastic material with yield stress = 191.64 MPa), Poisson’s ratio = 0.3 and Young’s modulus = 70,000 MPa.

All the tool parts were modelled as elastic with Young’s modulus = 200,000 MPa. The lower nodes on the lower container were constrained not to move in vertical direction and the lower right node on the upper container was constrained not to move in vertical direction. The punch was prescribed to move vertically down with constant velocity. The contact conditions in the punch – slug interface and in the slug – container interfaces were modelled as a combination of Coulomb friction and constant friction given by

\[ \mu = 0.04 \]
\[ m = 0.15 \]

These friction values have been determined experimentally by Hansen [82] when using the commercial lubricant “Glisapal” to lubricate aluminium.

The FEM simulation was carried out using implicit time integration with automatic remeshing of the slug for every 0.1275 mm displacement of the punch.

Figure 153 shows a close up of the can wall at the beginning of the backward can extrusion process, where the punch land is above the parting line between the upper and lower container. It can be seen that the upper container is displaced in radial direction in relation to the lower container.

Figure 154 shows a close up of the can wall and upper and lower container when the punch land has just passed the parting line. It can be seen that the lower container is displaced in radial direction in relation to the upper container.

A situation as shown in Figure 154 where the can wall nodes have to pass a sharp corner between upper and lower container is very unhealthy seen from a FE point of view and the FE simulation also did crash due to severe mesh distortion of the slug material.
near node 739, the bottom left node of the upper container. If the deformation of the containers determined from the FEM simulation is in qualitative agreement with reality, a container displacement as shown in Figure 154 may also be harmful in reality; the sharp corner may damage the lubrication film. It could be that this has attributed to the galling of the lower part of the outside can wall, which was observed experimentally (see Figure 145).

Figure 155 shows the radial position (x-coordinate) of the lower left node of the upper container (node 739) and upper left node of the lower container (node 707, see Figure 154) as function of time, where the time is proportional to the punch displacement. It can be see that up to a time $t \approx 30$ (this is shortly before the situation shown in Figure 153) there is nearly no radial displacement of node 707, whereas the radial displacement of node 739 increases steadily and reaches a maximum value of approximately 0.021 mm at $t \approx 35$. When the punch land starts passing the parting line after $t \approx 35$, the radial displacement of node 739 decreases whereas the radial displacement of node 707 increases steadily until $t \approx 52$ (at which time the FE simulation crashed), where the radial displacement is approximately 0.025 mm. The maximum node displacement in radial direction is thus of the same magnitude as the increase in average wall thickness near the kink in the cans produced with the punch with the circular profiled punch land (see Figure 139 and Figure 140).

Figure 153 Close up of the “can wall”. The punch land is above the parting line between the upper and lower container. The upper container has been displaced in radial direction in relation to the lower container.
Analysis of deep drawing, ironing and backward can extrusion

Figure 154 Close up of the can wall and upper and lower container. The punch land has just passed the parting line between upper and lower container. The lower container has been displaced in radial direction in relation to the upper container.

Figure 155 Radial position (x-coordinate) of the lower left node of the upper container (node 739) and upper left node of the lower container (node 707) as function of time.
18 3D-FEM simulations

It has been tried to carry out 3D-FEM simulations of a backward can extrusion process. A 3D simulation has to be carried out with remeshing, due to the severe mesh distortion which otherwise will occur. However the experience with LS-Dyna, both version 970 and version 971, is that the currently used remesher in LS-Dyna is not sufficiently robust; from time to time the mesh after remeshing is wrong. An example of a faulty mesh after remeshing is shown in Figure 156.

![3D simulation with automatic remeshing of backward can extrusion. Due to a bug in LS-Dyna the mesh after remeshing is wrong.](image)

18.1 FEM model

Instead it has been tried to make a 3D simulations, where the slug material is modelled as a thin slice in such a way that the simulations can be carried out without remeshing. Figure 157 shows the FEM model. Due to symmetry the FEM model was made as a half model. The tool parts were made from solid shell elements and the slug (the thin slice) from 4 node fully integrated solid elements. FEM model contained 43440 solid elements and 52320 shell elements. The material parameters and the friction conditions were the same as those used in the FEM-simulations discussed in Chapter 17.

The slug (the thin slice) was constrained not to move in radial direction by an inner and an outer rigid container wall. The lower nodes of the slug were constrained not to move in vertical direction. The punch was prescribed a vertical displacement, but free to move in radial direction (see Figure 157).

Two simulations were carried out, one with a punch with a cylindrical punch land and one with a punch with a circular profiled punch land. A close up of the two punches used
are shown in Figure 158. Besides from the punch geometry everything was the same in the two simulations. In both simulations the punch was tilted 0.2° around the centre of the punch land. Both the simulations were carried out using implicit time integration. With implicit time integration and the fairly large number of solid element the simulation time becomes large (3-4 weeks), and the simulations could probably have been carried out more efficiently using explicit time integration and appropriate mass scaling. However, implicit time integration was chosen to avoid any dynamic effects.

Figure 157 3D FEM model of backward can extrusion carried out with a punch with a cylindrical punch land. (The inner container is not shown in the Figure).

Figure 158 Close up of the FEM models. To the left with punch with the cylindrical punch land and to the right with the punch with the circular profiled punch land.
18.2 3D FEM results

Figure 159 shows the radial displacement of the punch; to the left for the punch with the cylindrical punch land and to the right with the circular profiled punch land. From Figure 159 it can be seen that with the cylindrical punch land the tilt of the punch causes the punch to keep on drifting until \( t = 45 \), at which time the simulation was stopped. For the punch with the circular profiled punch land, the tilt causes to start with a slight displacement of the punch and that the punch then moves back towards its initial position; the punch is nearly self centring.

![Figure 159 The radial displacement of the punch as function of simulation time. To the left for the punch with the cylindrical punch land and to the right for the punch with the circular profiled punch land.](image)

Figure 159 shows the radial displacement of the punch; to the left for the punch with the cylindrical punch land and to the right with the circular profiled punch land. From Figure 159 it can be seen that with the cylindrical punch land the tilt of the punch causes the punch to keep on drifting until \( t = 45 \), at which time the simulation was stopped. For the punch with the circular profiled punch land, the tilt causes to start with a slight displacement of the punch and that the punch then moves back towards its initial position; the punch is nearly self centring.

Figure 160 shows the equivalent strain distribution (for the same punch displacement) in the can wall, to the left when the extrusion is carried out with the punch with the cylindrical punch land and to the right when carried out with the punch with the circular profiled punch land. It is clear that the slight tilt of the punch causes a very uneven strain distribution in the can wall when the cylindrical punch land is employed whereas the tilt has only very little effect on the strain distribution (it is nearly homogeneous) when the circular profiled punch land is employed.

The backward can extrusion modelled does not reflect the material flow in the real backward can extrusion process. In the real process a shift of the punch in radial direction causes, as discussed in Chapter 16.5, an asymmetric material flow beneath the punch face. In the FEM model there is no material beneath the punch face and a shift of the punch in radial direction can due to volume constancy only be accomplished by material flow in circumferential direction and/or uneven material velocity in the extruded can wall.

Uneven velocity in the extruded can wall gives rise to an uneven can rim. Figure 161 shows the effective stress distribution in the can wall, to the left for the extrusion carried out with the cylindrical punch land and to the right with the circular profiled punch land. With the cylindrical punch land the effective stress is in a large part of the can wall above the punch land equal to the yield stress and from the simulation it could be seen that plastic deformation did occur in the can wall above the punch land. With the circular profiled punch land the effective stress in the can wall above the punch land is below the yield stress and after the material has passed the punch land the material is not plastically deformed anymore.
Although the material flow in the FEM simulations is very different from the material flow, which occurs in the real backward can extrusion process, the FEM results do show that when the punch land is cylindrical a slight tilt of the punch causes a significant change in the material flow and will cause the punch to be forced off centre. The FEM simulations also show that if the punch land is made with a circular profiled punch land then a slight tilt of the punch only causes very minor changes in the material flow and that the tendency of the punch to be forced off centre is reduced significantly.

Figure 160 The equivalent strain distribution in the can wall. To the left when the extrusion is carried out with the cylindrical punch land and to the right when the extrusion is carried out with the circular profiled punch land. The punch displacement is the same in both cases and the punch is tilted 0.2° in both cases.

Figure 161 The effective stress distribution in the can wall. To the left when the extrusion is carried out with the cylindrical punch land and to the right when the extrusion is carried out with the circular profiled punch land. The punch displacement is the same in both cases and the punch is tilted 0.2° in both cases.
19 Main conclusions regarding backward can extrusion

The punch used in backward can extrusion is commonly made with a cylindrical punch land as also recommend by the ICFG [59]. The influence of the punch land on the stability of the backward can extrusion process has hitherto not been investigated and this is somewhat surprising because as shown in this thesis the influence can be very significant.

The hypotheses that have been investigated are:

- A slight tilt of the punch land e.g. due to inaccurate machining, inaccurate alignment of the tools in the press and/or elastic deformation of the tools and press, will cause an uneven force distribution on the punch land (in circumferential direction), which will deflect the punch off centre leading to variations in the can wall thickness.

- By making the punch land non-cylindrical, e.g. circular profiled, the influence from a slight tilt of the punch land will be reduced.

The radial force on the punch land has been determined using the slab method and using FEM. Good agreement between the slab method results and the FEM results is obtained if constant friction is assumed in the punch land – can wall interface.

The effect of a slight tilt of the punch has been investigated by carrying out 2D simulations of the backward can extrusion process both as plane strain and as rotational symmetric. The results from the simulations show that with a punch with a cylindrical punch land a slight tilt of the punch causes the radial force on the punch land to drop to zero where contact between punch land and can wall is lost due to the tilt whereas the the radial force on the punch land on the opposite side is nearly unaffected by the tilt. This difference in radial force on the punch land causes a net radial force on the punch, which will deflect the punch off centre.

If the punch land is made circular profiled the simulations show that a slight tilt of the punch will only give rise to minor changes in the radial force on the punch land and that the net radial force on the punch due to the tilt is significantly smaller compared to when a punch with a cylindrical punch land is employed.

The backward can extrusion of a thin walled aluminium can has been analyzed experimentally. Cans were made both with a conventional punch with a cylindrical punch land and with a punch with a circular profiled punch land. The can wall thickness was measured on a coordinate measuring machine and the results show that the cans produced with the conventional punch have a significantly larger variation in the wall thickness than the cans produced with the punch with the circular profiled punch land; in the cans produced with the conventional punch the average can wall thickness variation is approximately 2.5 times as large as the average can wall thickness variation in the cans produced with the punch with the circular profiled punch land.

The slab analysis results, the FEM results and the experimental results support the hypotheses above and in the opinion of the author there is no doubt about

- That punches used in backward can extrusion should not be made with a cylindrical punch land, which currently is common practice and also recommended by the ICFG.

- That variations in the can wall thickness can be reduced significantly by using a punch with a circular profiled punch land in place of a conventional punch.
• That the robustness of the backward can extrusion process can be improved significantly by using a punch with a circular profiled punch land in place of a conventional punch.

The cans produced with the punch with the circular profiled punch land has a distinct kink/change in the can wall thickness around 100 mm below the can rim. The reason for this kink is believed to be an inappropriate design of the container used in the experiments. The container is made up of two containers and FEM-simulations carried out show that the parting line between the two containers is placed in an inappropriate height and that the elastic deformation of the containers during the backward can extrusion process is a likely explanation for the kink in the can wall thickness observed experimentally.

The upper part of the outside can wall on all the cans was shiny and bright whereas the lower part was dull due to severe galling. Some cans were produced with a slug, which had the lower corner marked with an ink pen. On the extruded cans the ink mark could be seen in the transition area between the upper part of the can without galling and the lower part with galling. The experiments thus show that the upper part without galling has been lubricated with lubricant coming from the side of the slug whereas the lower part has been lubricated with lubricant coming from the bottom of the slug.

Although there are many other things than the shape of the punch land, which influences the robustness of the backward can extrusion process and the quality of the cans produced, employing a punch with a circular profiled punch land instead of a conventional punch is in the opinion of the author a very easy and cheap way to increase the robustness of the backward extrusion process and to increase the quality of the cans produced. A lot of research work is still needed in this area in order to be able to prescribe the optimal geometry of the punch land with regard to can material, punch diameter, can wall thickness, tool life, residual stresses in the can wall etc..

Besides from increasing the robustness of the can extrusion process and improving the quality of the cans produced, the new punch design (a punch with a non-cylindrical punch land e.g. circular profiled punch land) may have other advantages compared to the conventional punch design (punch with a cylindrical punch land):

• It is difficult to manufacture a conventional punch with the very stringent tolerance requirements on the punch land which is required. With the new punch design the tolerance requirements may not be so stringent due to the increase in the robustness of the backward can extrusion process.
• When a conventional punch is employed very accurate tool alignment in the press is of outmost importance. With the new punch design very accurate tool alignment may not be so important, and this may mean that the tool can be mounted faster and easier in the press.
• With a conventional punch a slight tilt of the punch causes the backward can extrusion process to become unstable. This puts requirements on the quality and stiffness of the press used. With the new punch design the backward can extrusion process is not so sensitive to a slight tilt of the punch, and this may as consequence have that cans can be produced successfully with a lower capacity and/or quality press.

Billions of cans are made by backward can extrusion (e.g. aerosol cans, aluminium bottles, cases used for ink pens etc.). Any improvement in the backward can extrusion process may thus be of high industrial relevance. This is the reason why Aalborg University has claimed a patent covering the design of punches having a non-cylindrical punch land and products produced with such a punch [80*]. Currently the new punch
design has only been used for the production of two industrially products. The first one is the production of the aluminium can discussed in this thesis. The other product is an aluminium can used for production of a part, which is used by most European car makers. The later aluminium can is 175 mm heigh with a can wall thickness of 5 mm. Hitherto this can has been made with a conventional punch and there has been problems to obtain the desired quality regarding thickness variations. The company producing this part has made a trial production using the new punch design and the main conclusions from this trial production are:

- The average wall thickness variation is reduced from 0.32 mm to 0.14 mm
- The maximum thickness variation is reduced from 0.5 mm to 0.27 mm
- With the new punch design it is faster and easier to mount the tools in the press
- With the conventional punch production stops occurred quite frequently due to pick up (galling). With the new punch design not a single production stop was encountered during the entire trial production.
- With the conventional punch the production was carried out on a stiff high quality 1000 t press. The trial production was carried out on a 500 t press with much lower stiffness. Despite this difference in press quality the cans produced with the new punch design were significantly better than the cans produced with the conventional punch.
20 Danish summary (dansk resume)

Denne afhandling beskriver noget af det forskningsarbejde, forfatteren har gennemført i perioden 1991 – 2005 ved Institut for Produktion, Aalborg Universitet. Afhandlingen er primært baseret på nogle af forfatterens forskningsrapporter; disse rapporter udgør ikke en del af afhandlingen, men er af hensyn til de, som måtte være interesseret i en mere detaljeret beskrivelse af de gennemførte analyser, indeholdt på den CD, som er vedlagt afhandlingen. En del af de opnåede forskningsresultater er publiceret i artikler og patentansøgninger, hvoraf 14 er indeholdt i afhandlingen. Når der refereres til en af de artikler, der er indeholdt i afhandlingen, er referencen markeret med en *, og når der refereres til en forskningsrapport, som er indeholdt på den vedlagte CD, er referencen markeret med **.

De formgivningsprocesser, som behandles i afhandlingen, er dybtrækning, strækningsreduktion og baglæns koldflydepresning. Fælles for disse processer er, at værktøjerne sædvanligvis laves på basis af retningslinier, der er udformet på basis af mange års praktisk erfaring.

Udviklingen indenfor ikke-lineær FEM og indenfor computere har i løbet af de sidste 10 år markant forbedret mulighederne for at gennemføre detaljerede analyser af formgivningsprocesserne, men de markant forbedrede muligheder for at analysere og dermed forstå de grundlæggende procesmekanismer har kun haft meget begrænset indflydelse på de retningslinier, som anvendes, når værktøjerne skal konstrueres; retningslinierne har stort set ikke ændret sig de sidste 25 år.

Den ”røde tråd” i afhandlingen er at vise, at det, på basis af en detaljeret forståelse af de grundlæggende procesmekanismer, er muligt at forbedre processerne, fx med hensyn til ennekvalitet, værktøjslevetid eller robusthed af processen, samt at vise at forbedringerne i visse tilfælde kan opnås ved meget små ændringer i værktøjsgeometrien. Udgangspunktet for de gennemførte analyser har været industrielle problemer. Formålet med at gennemføre analyserne har været at opnå et detaljeret forståelse af de grundlæggende procesmekanismer, er muligt at forbedre processerne, fx med hensyn til ennekvalitet, værktøjslevetid eller robusthed af processen, samt at vise at forbedringerne i visse tilfælde kan opnås ved meget små ændringer i værktøjsgeometrien. Formålet med denne forståelse er at fremkomme med forslag til løsninger.


Kapitel 4-6 omhandler en analyse af restspændinger i kupvæggen af dybtrukne pladeemner. Restspændingerne kan i visse tilfælde blive så store, at de medfører revnedannelse. Hovedårsagen til dannelse af restspændinger er inhomogen deformation, og det er derfor blevet analyseret, i hvilket omfang restspændingerne kan reduceres ved at tilstræbe, at den afsluttende plastiske deformation er så homogen som mulig.

Kapitel 4 omhandler en eksperimentel og teoretisk analyse af en to trins dybtrækning efterfulgt af strækningsreduktion. Kapitlet er baseret på [28**], og nogle af de opnåede resultater er publiceret i [29*,30,31]. De gennemførte FEM-simuleringer viser, at der i den dybtrukne kop er meget store kompressive restspændinger mod indersiden af kupvæggen og meget store trækspændinger mod ydersiden. Der er god overensstemmelse
mellem den maksimale omkredsspænding bestemt eksperimentelt og bestemt ved hjælp af FEM; dette på trods af, at FEM simuleringerne viser, at spændingsfordelingen gennem kopvæggen ikke er linear, hvilket forudsættes i den eksperimentelle bestemmelse af den maksimale omkredsspænding. Såvel de eksperimentelle resultater og resultaterne opnået ved hjælp af FEM viser, at strækningsreduktionen medfører en markant reduktion af restspændingerne. Ifølge resultaterne opnået ved hjælp af FEM vil den strækningsreducerede kop have markant forbedrede egenskaber mht. udmattelsesstyrke og –levetid sammenlignet med den ikke strækningsreducerede kop.

I FEM simuleringerne er der anvendt en penaltybaseret kontaktalgoritme, og simuleringer af strækningsreduktionen, gennemført med forskellige værdier af penaltyfaktoren, viser, at den bestemte restspændingsfordeling afhænger meget af, hvilken værdi penaltyfaktoren har. Det konkluderes derfor, at i FEM simuleringer af formgivningsprocesser, hvor selv en lille knudpenetration kan have en markant indflydelse på restspændingsfordelingen, skal man være meget forsigtig, hvis restspændingerne ønskes bestemt kvantitativt på basis af en FEM-simulering.


Kapitel 6 beskriver en eksperimentel og teoretisk analyse af en to-trinsdybtrækning gennemført dels med konventionelt udformede trækninger dels med traktriceudformede trækninger. Nogle af de opnåede resultater er publiceret i [39*,42*]. De eksperimentelle resultater og FEM-simuleringerne viser, at i den analyserede to-trinstrykning medfører anvendelsen af traktriceformet trækning ikke en markant reduktion i størrelsen af restspændingerne, men at anvendelsen af traktriceformede trækninger giver en mere jævn restspændingsfordeling samt en væsentlig forbedring af retheden af koppens yderside. FEM-simuleringerne viser også, at restspændingsfordelingen og kopgeometrien stor set er uafhængig af trækringens udforming i første trin; resultaterne fra FEM-simuleringerne af trækningen, hvor der er anvendt en traktriceformet trækning i andet trin, er stort set identiske, hvad enten der er anvendt en konventionelt udformet eller traktriceudformet trækning i første trin. Både de eksperimentelle resultater og FEM-simuleringerne viser, at emnet trukket med konventionelt udformede trækninger i begge trin er lidt højere end emnet trukket med traktriceformede trækninger i begge trin.
To-trinstrækningen af et industrielt emne var temaet i et studenterprojekt [43], og formålet med dette projekt var bl.a. at optimere udformningen af trækringsgeometrien i andet træk med henblik på minimere restspændingerne. På basis af gennemførte FEM-simuleringer blev det valgt at udforme trækringsrundingen i andet trin som en del af en ellipse. Såvel de eksperimentelle undersøgelser som FEM-simuleringerne viste, at anvendelsen af den ellipseformede trækning i andet trin medførte en markant reduktion i restspændingerne, en væsentlig forbedring af kopretheden samt at kophøjden blev forøget med ca. 5%.

De gennemførte analyser af to-trinstrækning viser således, at trækningens udformning i andet trin kan have en markant indflydelse på kvaliteten af den trukne kop. Med en hensigtsmæssig udformning, fx ellipseformet som anvendt i [43], kan der opnås en væsentlig reduktion af restspændingerne, en forbedring af retheden af kopvæggens yderside og en forøgelse af kophøjden (det sidste er ikke nødvendigvis en kvalitetsforbedring, men medførte for det pågældende emne en ikke uvæsentlig reduktion i materialeforbruget).

Kapitel 9 til 12 beskriver analyser af strækningsreduktionsprocessen. Kapitlerne er primært baseret på [60** - 64**, 72**, 73**] og de væsentligste resultater er publiceret i [65*, 69, 74*, 75*]. Analyserne, der er gennemført ved hjælp af snitelementmetoden og FEM, er gennemført med fokus på stabilitetsproblemer.


- hvis vinklen af det koniske indløb formindskes
- hvis længden af kalibreringsstykket formindskes
- hvis værdien af $\mu_{punch} - \mu_{die}$ øges
- hvis materialets deformationshærdning øges

Specielt den store indflydelse, som længden af kalibreringsstykket har på den kritiske reduktionsgrad, er tilsyneladende en ny erkendelse. Der er skrevet mange artikler om strækningsreduktion, men det er kun fundet en artikel, artiklen af Ras et al [57], i hvilken det er omtalt, at længden af kalibreringsstykket kan have en indflydelse på processtabiliteten. I de fleste artikler om strækningsreduktion er længden af
kalibreringsstykket ikke specificeret, og hvor den er specificeret, er der specificeret meget forskellige længder (fra \( l = 0.2t_0 \) til \( l = 8t_0 \)).

Hvis der anvendes en matrice med cirkulær profil, viser analyserne at den kritiske reduktionsgrad er næsten konstant 36\% uafhængigt af radius \( R \) på det anvendte profil. Med hensyn til indflydelsen fra friktion og deformationshærdning er indflydelsen den samme som nævnt ovenfor for det konventionelle matricedesign.


Strækningsreduktion af cylindriske kopper blev analyseret i et studenterprojekt [34]. I visse tilfælde blev der observeret en meget uensartet godstykkelsetordeling i den øverste tredjedel af koppen, ligesom der også af og til opstod brud i den øverste del af kopvæggen. Der er meget fin kvalitativ overensstemmelse mellem resultaterne beskrevet i [34] og 3D FEM simuleringerne af strækningsreduktionen gennemført med en svag kæntring af reduktionsmatricen; bl.a. viser 3D simuleringerne, at en svag kæntring vil medføre godstykkelsevariationer primært i den øverste del af koppen. Simuleringerne indikerer derfor kraftigt, at en mulig årsag til den uensartede godstykkelsetordeling og brud i koppen beskrevet i [34] har været forårsaget af en svag kæntring af matricen.

værtøjsgeometrien, idet det var muligt at bringe en matrice til at fungere blot ved at polere denne.

To kopper blev udtaget tilfældigt fra den løbende produktion. Kopperne blev opmålt med hensyn til kophøjde og kopvægstykkelse med en 3-koordinatmålemaskine. Opmålingerne viste, at de to kopper med hensyn til variation i kophøjde og med hensyn til godstykkelsevariation stort set var identiske. Det blev på basis heraf konkluderet, at variationen i kophøjde og godstykkelse ikke skyldtes stokastiske variationer i proces- og/eller materialeparametre, men at årsagen formentlig skulle findes i det anvendte værktøj. Dybtræk- og strækningsreduktionsprocessen, i hvilken der anvendes en konventionel designet matrice med cylindrisk kalibreringsstykke, blev analyseret ved hjælp af 2D og 3D FEM simuleringerne. 2D simuleringerne indikerer, at det er muligt, ved i pressen at forskyde stemplet lidt i forhold til matricen, at producere en kop med ensartet kophøjde, på trods af at godstykkelsen varierer i omkredsetningen. Simuleringerne indikerer således, at måling af kophøjden alene ikke kan bruges som kvalitetskontrol.

3D simuleringerne viser, at en svag kæntring af matricen i forhold til stemplet medfører, at den producerede kop har en meget varierende kophøjde og meget varierende godstykkelse i omkredsetningen. Simuleringerne indikerer således, at produktionsproblemerne kan skyldes en svag kæntring af kalibreringsstykket; denne kæntring kan skyldes uøjagtig bearbejdning, uøjagtig opretning af værktøjerne i pressen og/eller elastisk deformation af værktøjer og presse.

Ved at lave kalibreringsstykket med en cirkulær profil frem for cylindrisk, vil en lille kæntring, i modsætning til hvis der anvendes et cylindrisk kalibreringsstykke, kun medføre marginale ændringer i kontaktforholdene mellem kalibreringsstykke og kopvæg. De gennemførte 3D FEM simuleringer viser, at hvis matricen laves med et kalibreringsstykke, som har en cirkulær profil, er kvaliteten af de producerede kopper stor set uafhængig af en svag kæntring af matricen. Simuleringerne indikerer derfor, at dybtræk-strækningsreduktionsprocessen bliver væsentligt mere robust, hvis der anvendes en matrice med et cirkulært formet kalibreringsstykke frem for, som konventionelt, et cylindrisk kalibreringsstykke. Ideen med at lave kalibreringsstykket profileret frem for cylindrisk, er så vidt forfatteren ved, ny; Aalborg Universitet har derfor søgt ideen patenteret [75*].

Kapitel 13 – 19 beskriver en eksperimental og teoretisk analyse af baglæns koldflydepresning. Kapitlerne er primært baseret på [76**] og nogle af de opnåede resultater er publiceret i [77*-80*].

Stemplet, der anvendes i baglæns koldflydepresning, designes sædvanligvis i overensstemmelse med ICFG’s anbefalinger [59] (ICFG: International Cold Forging Group). ICFG anbefaler, at stemplet laves med et cylindrisk kalibreringsstykke (på engelsk benævnt punch land). Ideen bag de gennemførte analyser var en forventning om, at det cylindriske kalibreringsstykke med hensyn til en svag kæntring, vil have samme effekt som ved strækningsreduktion, samt at processen kan gøres mere robust ved at lave kalibreringsstykket profileret, fx som en del af en cirkelprofil.

De teoretiske analyser er gennemført primært ved hjælp af FEM (beskrevet i kapitel 14 og 15). 2D FEM simuleringerne viser, 1) at hvis der anvendes et stempel med et cylindrisk kalibreringsstykke, som anbefalet af ICFG, vil en svag kæntring af kalibreringsstykket medføre en stor radiel kraft på stemplet; en radiel kraft, der vil forskyde stemplet i radiel retning, og derfor medføre uensartet godstykkelse i den færdige kop, og 2) at hvis der anvendes et stempel, hvor kalibreringsstykket har en cirkulær profil, vil en svag kæntring medføre en markant mindre radiel kraft på stemplet. De
gennemførte simuleringer indikerer således, at der kan opnås en væsentlig forøgelse af robustheden ved at anvende et stempel med et cirkulært profileret kalibreringsstykke frem for at anvende et stempel med et cyldriske kalibreringsstykke, som anbefalet af ICFG.

Kapitel 16 beskriver en eksperimentel analyse af baglæns koldflydepresning af et aluminiumskop. Koppen, der har en udvendig diameter på ø73 mm, en kophøjde på ca. 250 mm og en nominel godstykkelse i kopvæggen på 0.915 mm, er et industrielt emne, som i den pågældende virksomhed fremstilles i en 500 t hydraulisk presse. I den industrielle produktion blev der anvendt et konventionelt udformet stempel med et cyldriske kalibreringsstykke, som anbefalet af ICFG. 10 kopper blev fremstillet med det konventionelt udformede stempel, og 12 kopper blev fremstillet med et stempel, hvor kalibreringsstykket havde en cirkulær profil. Ved skift af stempeleffektiviteten blev der lagt stor vægt på, at det eneste, der blev ændret, var stempeleffektiviteten; alle kopperne blev fremstillet ud fra samme udgangsmateriale og –dimensioner, med samme smøring, med samme presseehastighed og med samme værktoj (bortset fra stemplet). Alle forøgsemlerne, både de, der var fremstillet med den konventionelle stempeleffektivitet, og de, der var fremstillet med stempel med det cirkulært profilerede kalibreringsstykke, havde næsten samme højde, og der var en meget begrænset variation i kophøjden på den enkelte cop, men den maksimale højderetning på den enkelte kop var for alle kopperne mindre end 2 mm. Alle kopperne blev opmålt ved hjælp af en 3-coordinatmålemaskine; dels blev kopygdeprofyllet målt i koppens højderetning (for hver 2 mm) langs fire frembringere, og dels blev kopygdeprofyllet målt ved at scanne cirkler på indvendig og udvendig side af koppen for hver 5 mm i højderetningen. Tykkelsesmålingerne viste, at koppeproduceretet med det konventionelle stempel med cyldriske kalibreringsstykke havde en væsentlig større tykkelsesvariation end koppeproduceret med stemplet med det cirkulært profilerede kalibreringsstykke; den gennemsnitlige maksimale godstykkelsesvariation i kopperne produceret med det konventionelt udformede stempel var 0.238 mm, hvorimod den i kopperne produceret med stemplet med det cirkulært profilerede kalibreringsstykke var 0.092 mm. Ændringerne i koldflydepresningen eksperimentelt, har anvendelsen af stemplet med det cirkulært profilerede kalibreringsstykke således medført en markant forbedring med hensyn til godstykkelsesvariation. Ideen med at lave kalibreringsstykket profileret, fx som en del af en cirkel, er, så vidt forfatteren er orienteret, ny. Aalborg Universitet har derfor ønsket at udtage patent på en stempeleffektivitet med ikke-cyldriske kalibreringsstykke [80*].

På alle de analyserede koppe (10 fremstillet med det konventionelt udformede stempel og 12 fremstillet med stempel med det cirkulært profilerede kalibreringsstykke), var der en markant forskel i overfladeruheden på koppernes yderside; de øverste ca. 140 mm af koppen havde en glat og skinnende overflade, hvorimod de nederste ca. 110 mm af koppen var ru. Denne forskel i ruhed kunne indikere en væsentlig ændring i smørebetingelserne under koldflydepresningen. For at analyserere, hvorfra smøremidlet til smøring af henholdsvis den øverste og nederste del af koppen kom fra, blev flydepresningerne udført med udgangsmateriale mærket med blå og sorte tuschstreger. Tuschstregerne placer en på de pressede koppe viste, at smøremidlet til smøring af den øverste del af kopvæggen stammer fra udgangsemmens sider, hvorimod smøremidlet til smøring af den nederste del af kopvæggen stammer fra udgangsemmens underside.

Målning af vægtyskelsesvariationerne viste, at der for alle de analyserede koppe er en markant ændring i vægtyskelsesvariation ca. 100 mm under kopranden. Ændringerne er mest markante i kopperne fremstillet med stempel med det cirkulært profilerede kalibreringsstykke. I den anvendte værktoj er containeren delt i en øvre og en nedre container. For at undersøge, om en uheldig placering af delelinien mellem øvre og nedre container kan forklare den målte tykkelsesvariation, blev koldflydepresningen søgt
simuleret. I disse simuleringer var værktøjerne modelleret identiske med de virkelig anvendte værktøjer. Simuleringsresultaterne indikerer, at delelinien mellem øvre og nedre container er placeret uhensigtsmæssigt, samt at den elastiske deformation af øvre container i forhold til nedre container er en mulig årsag til, at godstykkelsereductionen udviser et markant spring omkring 100 mm under koppens kant.

De gennemførte undersøgelser af såvel strækningsreduction som baglæns koldflydepresning viser, at anvendes der et konventionelt design, hvor kalibreringsstykket et cylindrisk, medfører en svag kæntring af kalibreringsstykket (denne kæntring kan være forårsaget af unøjagtig bearbejdning, unøjagtig opretning af værktøjerne i pressen og/eller elastisk deformation af værktøjer og presse), at processen bliver ustabil, resulterende i varierende vægttykkelse i den producerede kop. Undersøgelserne viser også, at laves kalibreringsstykket profileret, fx som en del af en cirkel, bliver processen væsentlig mere robust med hensyn til en svag kæntring af kalibreringsstykket; en svag kæntring giver kun anledning til små variationer i vægttykkelsen.

Det er forfatterens opfattelse, at anvendelse af et cylindrisk kalibreringsstykke gør, at processen er i en tilstand af ustabil ligevægt, og at enhver forstyrrelse, fx forårsaget af uensartet smøring, vil medføre at processen bliver ustabil. Det er tankevækkende, at det konventionelle design med et cylindrisk kalibreringsstykke, er noget nær det værste tænkelige design med hensyn til procesrobusthed, og også tankevækkende at meget små ændringer i værktøjsdesign, ændringer, der er lavet på basis af på et detaljeret kendskab til procesmekanismerne, kan medføre markante forbedringer i procesrobustheden og på kvaliteten af de producerede emner.
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