DEVELOPMENT OF DESIGN GUIDELINES
FOR PART, TOOLING AND PROCESS
IN THE TUBE HYDROFORMING TECHNOLOGY

DISSERTATION

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Today, an increasing competition and environmental regulations force the automotive industry to produce lightweight cars for savings in fuel consumption. On the other hand, the automotive industry desires to manage this in a cost-effective manner. There are, however, only two ways to achieve that from the manufacturing point of view (1) Development of new materials or composites that provide light weight parts while keeping the same required strength specifications, and (2) Development and improvement of new manufacturing techniques to produce parts with high strength-to-weight ratios.

Tube hydroforming (THF) process, which is a relatively new technology of the second category, utilizes mainly fluid pressure and hollow blank material to produce complex shaped parts such as exhaust system pipes, side rails, engine cradles, camshafts, crankshafts, differential casings, support panels, etc. THF process provides consolidation
of parts, high strength-to-weight ratios, tight tolerances, better rigidity, less post-process operations, easy assembly, cost effective parts and tooling.

As expected, the demand on the development of a knowledge base of this technology is also increasing so that mass production of new and more parts can become a reality. Since internal hydraulic pressure and axial compressive forces are simultaneously used to produce hollow THF parts, coordination of these external actions for different parts, material and geometry is one of the most crucial challenges. Effect of tubular material characteristics and geometrical parameters on formation of required features such as bulge and protrusions is another question that needs to be answered among many others.

A large variety of current and candidate THF parts were classified according to their common features to simplify the eventual analysis. Simple and useful analytical models were developed and improved using available techniques and approaches as a first step tool in the analysis of the deformation. Understanding of the tubular material characteristics was reached by development of a testing apparatus and methodology. Use of finite element analysis (FEA) with existing commercial codes was demonstrated to be an effective tool in design of part, process and tooling. Finally, an optimization concept via controlled FEA to determine the exact loading paths during hydroforming of any possible part was introduced along with acting criterion.
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FIELD OF STUDY

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\( \sigma_3 \)  
*Minimum Principal Stress* \[ \text{N/mm}^2 \]

\( \varepsilon_\theta \)  
*Tangential (Hoop) Strain*

\( \varepsilon_z \)  
*Longitudinal (Meridional) Strain*

\( \varepsilon_r \)  
*Radial (Thickness) Strain*

\( \varepsilon_1 \)  
*Maximum Principal Strain*

\( \varepsilon_2 \)  
*Second Principal Strain*

\( \varepsilon_3 \)  
*Minimum Principal Strain*

\( \alpha \)  
*Ratio of longitudinal stress to hoop stress, \( \frac{\sigma_z}{\sigma_\theta} \)*

\( \beta \)  
*Ratio of longitudinal strain to hoop strain, \( \frac{\varepsilon_z}{\varepsilon_\theta} \)*

\( m \)  
*Shear Friction Coefficient*

\( \mu \)  
*Coulomb Friction Coefficient*

\( H_p \)  
*Height of Feature* \[ \text{mm} \]

\( l_0 \)  
*Initial Length of Tube* \[ \text{mm} \]

\( l_1 \)  
*Instantaneous or Final Length of Tube* \[ \text{mm} \]

\( L_{pe} \)  
*Distance Between Feature and Edge* \[ \text{mm} \]

\( L_{pp} \)  
*Distance Between Features* \[ \text{mm} \]

\( d_0 \)  
*Initial Median Diameter of Tube* \[ \text{mm} \]

\( D_0 \)  
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\( D_i \)  
*Initial Inner Diameter of Tube* \[ \text{mm} \]

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CHAPTER 1

INTRODUCTION

The ever-increasing need to reduce weight of automotive products can be achieved either by a) using lighter materials, or b)-improved designs, which offer a high strength to weight ratio. The tube hydroforming process supports the second strategy; namely the reduction of weight by improved design and reduced assembly.

Components formed through tube hydroforming have many benefits, including lighter weight, greater strength, fewer part requirements, etc. Since the initial invention and development of the process, the hydraulic and computer control systems have improved to a point where the tube hydroforming process could be utilized as a practical forming method. With mass application of this technology now possible, the process is being applied in the automotive industry. Drastic improvements have been made in the hydraulic control systems, knowledge base, part designs, simulation packages, and forming machines.
THF is a metal forming process where a tubular raw material is converted into a complex shaped hollow part in a die cavity with the use of internal pressure, axial and counter forces, Figure 1.1. Internal pressure is supplied by filling and squeezing a fluid medium inside the tube after sealing the ends properly. Axial loading (or other forces such as counter force) is supplied by hydraulic cylinder and punch arrangements. Internal pressure values may vary between 1000-3000 bars (100-300 Mpa) whereas axial load ranges are from 100-500 tons (1000-5000 kN) depending on the part shape, material, and thickness, etc. THF is one of the liquid bulge forming methods as seen in Figure 1.2.

Figure 1.1 illustrates a typical part that is being formed by hydroforming technique. The ultimate aim in tube hydroforming is to form the tube into the given die cavity without any defects on the part. Furthermore, the minimum thickness on the part should be kept in a range determined by functional requirements. Figure 1.3 summarizes the parameters that should be considered in tube hydroforming process.

THF process has been in practical industrial use for more than a decade. With the advances in computer controls and high-pressure hydraulic systems, it has become a viable technique for mass production of hollows part used in automotive, aircraft and appliance. Main advantage of this technology is in weight savings while keeping the strength of the formed component at required levels determined by functional specifications.
Figure 1.1 Illustration of a typical part that is being formed by hydroforming technique
Figure 1.2: Classification of liquid bulge forming processes of which tube hydroforming is a member.
Figure 1.3: All parameters and limits effective in tube hydroforming process.
Limitations on this process are mainly (a) maximum pressure and axial loading that could be supplied by the system available, (b) maximum holding (clamping) capacity of the existing press, and (c) different failure modes in forming such as Bursting, Buckling, and Wrinkling. Figure 1.4 illustrates these common failure modes in tube hydroforming.

Bursting occurs as a result of excessive internal pressure ($P_i$). It starts with local necking of the tube wall at the expansion region. If the tube is not yet in contact with die wall, as expansion increases, necking transforms into fracture. Wrinkling is a consequence of excessive axial loading. Wrinkling can be eliminated by increasing internal pressure during final phase of the forming, i.e. calibration. Buckling is another form of failure due to excessive axial loading or insufficient internal pressure at the initial stages of the process. Buckling and wrinkling can be avoided by proper application of internal pressure and axial loading during the beginning and intermediate phases of the forming such that enough internal pressure is applied to support the tube wall under axial force.

Thus, the control and coordination of axial load versus time and internal pressure versus time are extremely important for successful tube hydroforming.
Figure 1.4: Common failure modes that limit the THF process [Dohmann, 1995]
THF process is the final operation in a series of other preparatory operations such as cutting of tubes, cleaning, pre-forming, annealing, etc. Figure 1.5 illustrates pre- and post-processes in THF technology. The implementation of tube hydroforming process in industrial mass production needs more investigation on different aspects of the technique. These areas, requiring additional research and development, can be summarized as follows:

(a) Formability of tubular material, and expected mechanical characteristics (flow stress) of tubular materials,

(b) Friction and lubricants appropriate for THF process, particularly under high pressure conditions,

(c) Pre-forming or bending of tubes for THF operations and their effect on the final part: pre-forming for only THF needs to be outlined and production rules should be established,

(d) Design guidelines for THF part, process and tooling,

(e) Mechanical and fatigue properties of final THF parts and their performance under real working environment,

(f) Cost analysis of THF process and its comparison with other methods such as stamping and welding.

Development of design guidelines is an urgent need for industrial applications of the technology since an expertise in this area has not been developed yet due the fact that the
technology is relatively new. Figure 1.6 shows steps involved in design of process, part and tooling for tube hydroforming.

Design of part indicates the determination of critical radius, transition radii, length, thickness, etc. for a part in consideration. Process design includes estimation of forces, pressure and feeding during operation. In order to reduce the trial-and-error procedure for the design of a new THF part, general design guidelines have to be established.

1.1. Advantages of THF process

A manufacturing process can be expected to be successful in today’s challenging market conditions only if it meets both technical and economical requirements. THF process would be useful in reducing weight and cost simultaneously by improving structural integrity, strength and rigidity. In addition, THF process satisfies these requirements with utilizing the common and available materials efficiently.

With THF process, significant benefits can be obtained when multi-piece-stamping assemblies are replaced by a single, hollow and strong part. Assemblies with fewer and simpler components would require fewer resources in design, manufacturing and assembly operations.
Figure 1.5: Illustration of operations involved in THF technology: cutting of tubes, pre-forming, annealing, etc. [Huber & Bauer, 1994].
Savings in tooling, material, design, production and assembly will altogether contribute reducing the overall cost of a THF part. Elimination or decrease of welds and welding operations is an additional factor in reduction of the overall cost.

A reduction in number of production steps and components in an assembly will be obtained with THF process. This would reduce dimensional variations, and facilitate assembly operations.

Following is a list of potential advantages gained with the use of THF technology:

(1) Reduction in weight

(2) Increase in stiffness and rigidity

(3) Economic material utilization

(4) Complex shaped and various part types

(5) Reduction in number of steps during manufacture and assembly (reduce welding and associated fixturing)

(6) Reduction in overall cost per part or cost of assembly

(7) Tight tolerances with good dimensional characteristics and less variation

(8) Good surface finish
Figure 1.6: Overall part, tooling and process design procedure for tube hydroforming process.
1.2 Motivation, research objectives and research plan

THF process has been already proven in production to manufacture hollows parts with
great structural integrity, smooth surface finish and less weight. However, the THF
technology is relatively new. A knowledge base does not exist, as it does in other
manufacturing processes such as stamping and forging. Consequently, for wider
implementation of the THF technology in mass production of different parts, detailed
investigations on various issues, including formability, friction, pre-forming, etc. are
required.

Particularly for practical purposes in industry, manufacturers want to know whether a
specific part, which is already manufactured by another method, can be produced using
THF process “as designed”. If not, what design changes should be done for
producibility. In addition, before going into detailed analysis of the design and forming
process, designers should be aware of the approximate process parameters, such as
internal pressure, axial, counter and clamping loads, required to form that part.

Furthermore, for a given part shape, dimensions and material specifications, initial tube
parameters, such as length, diameter, thickness, etc., need to be predicted in advance to
estimate cost and plan production steps. In addition, whether or not pre-forming and
annealing are necessary should be decided before investing considerable effort in process
and tool design.
Thus, establishment of design guidelines and development of techniques to predict process and tube parameters is a must for effective and profitable industrial implementation of THF technology.

1.2.1 Research objectives

Based on the observations made, the main objective of this study is to develop design guidelines that will help the designer during the initial part, process and tooling design to reduce product cost, process steps and development time. Specific objectives are, for a part with known material and final geometry, to determine:

(a) If that part can be produced as designed based on material formability and process limits

(b) If not, what are the necessary changes on the part geometry to ensure producibility

(c) Internal pressure, \( P_i \) versus time

(d) Axial load, \( F_a \) versus time

(e) Counter load, \( F_q \) versus time

(f) Clamping load, \( F_c \) versus time

(g) Initial tube parameters such as \( L_0 \), \( d_0 \), \( t_0 \), etc.
1.2.2 Research Plan

To achieve the objectives of this study, the following research phases are planned:

(1) State-of-the-art review of the technology, design rules, mechanistic models, etc.

(2) Determination of the scope for this study

(3) Classification of THF parts with respect to their common characteristics

(4) Development of analytical models for prediction of process parameters such as \( (p_i) \), \( (F_u) \), \( (F_q) \), \( (F_c) \), etc.

(5) Collection of design rules from other sources, improvement of existing and development of new design rules

(6) Development of a computer program (TH-DESIGN) for prediction of process parameters using analytical models for different part groups

(7) Construction of a database (TH-DBASE) that will contain example designs from available finite element analysis (FEA) and experimental data from processing different THF parts
(8) Integration of the computer program (TH-DESIGN) with the database (TH-DBASE) to form a useful and versatile computer software package (TH-CADS) that will help designers for part, process and tooling design in THF.

1.3 Outline of the dissertation

Chapter 1 presents an overall introduction to the topic. It summarizes the advantages of the technique, and outlines the limits of the process. Areas that need to be investigated are listed as well.

A state-of-the-art review of the technology is provided in Chapter 2. Investigations on common issues are discussed separately under various headings. Current and candidate THF parts are classified with respect to their spline axis geometry, common feature types, etc. in Chapter 3. Concept of modular design was mentioned in this chapter. Prediction of different process parameters, such as pressure and forces, with analytical models is presented in Chapter 4.

Major limitations and defect types in THF process are analyzed in Chapter 5. Buckling, wrinkling and bursting are presented as the common failure types. A simple computer program algorithm is also given in this chapter. Importance of characterization of material behavior is obvious from the experience in the art of metal forming. Determination of the flow stress properties for tubular materials is explained in Chapter 6.
briefly. Along with it, improvement in thickness predictions instead of tedious and lengthy measurements is presented with examples.

In order to analyze the THF process in depth, many FEA are performed. Case studies with different parts, and comparison of FEA results with experiments are provided in Chapter 7. A series of planned FEA is conducted. In Chapter 8, results of planned 2D and 3D FEA are provided in forms of guidelines. In the planning phase “design of experiment” techniques are employed.

Brief summary and conclusions of this study are briefed in Chapter 9 along with presentation of an overall methodology for design of THF parts, tooling and process. Recommended future work is also given at the end of this chapter.

There are five appendices giving details of some analyses and results of this study. In Appendix A, many part drawings according to their classification given in Chapter 3 are illustrated for readers’ reference. Fundamentals of plasticity and membrane theories are presented in Appendix B and C, respectively. Similarly, common knowledge on thick-walled cylinder approach is provided in Appendix D. Although limited, some design rules have been accumulated within the industrial community dealing with THF technology. Design rules gathered from literature and developed during this study are provided in Appendix E in a categorized manner for easy access of readers.
CHAPTER 2

STATE OF THE ART REVIEW

Tube Hydroforming (THF) has been called with other names depending on the time and country it was used and investigated. Bulge Forming of Tubes (BFT) and Liquid Bulge Forming (LBF) were the earlier ones. Hydraulic (or Hydrostatic) Pressure Forming (HPF) was another form of name used for a while by some investigators. Internal High Pressure Forming (IHPF) has been mostly used within German manufacturers and researchers. In some periods, it was even called as “Unconventional Tee Forming”. Throughout this document, THF will be used to describe the metal forming process whereby tubes are formed into complex shapes with or without a die cavity using internal pressure, which can be obtained by various means (hydraulic, viscous medium, elastomer, polyurethane, etc.). In many cases axial compressive forces are also applied.

Even though Tube Hydroforming (THF) process has been in practical industrial use only more than a decade, development of the techniques and establishment of the theoretical
background goes back to 1940s. Manufacturing of seamless copper fittings with T branches was investigated using internal pressure and axial load [Grey, et. al., 1939]. Davis tested tubes of medium carbon steel under internal pressure and tensile axial load in order to determine their yield and fracture characteristics [Davis, 1945]. Experimental and numerical studies were conducted to find the bursting pressure of thick-walled cylinders by Faupe [Faupe, 1956], Crossland, et al. [Crossland, et al., 1959], and Dietmann [Dietmann, 1967]. In 1960’s, experimental and theoretical investigations on instability of thin-walled cylinders were performed by many researchers at different countries [Mellor, 1960] [Weil, 1963] [Woo, 1964]. Fundamental investigations on thin- and thick-walled cylinders helped theoretical improvements in liquid bulge forming operations. Use of hydrostatic pressure in metal forming processes, in particular, for bulging of tubular parts was first reported by Fuchs [Fuchs, 1966]. In this paper, he reported experimental studies on expansion and flanging of copper tubes using hydraulic pressure.

Ogura and Ueada presented their experimental results on liquid bulge forming of Tee shapes from low and medium carbon steel [Ogura, et al., 1968]. Different configurations and number of Tee protrusions were formed using internal pressure and axial compressive loading. Safe zones of forming were defined for Tee protrusions using experimental results. Experimental results for forming of differential cases were also disclosed in this paper. In the same period, Al-Qureshi and his team performed bulging and piercing experiments of different materials including copper, steel and aluminum.
using polyurethane to provide internal pressure [Al-Qureshi, et. al., 1968]. Axial load was not used in these experiments.

In 1970s, research on different aspects of bulge forming continued under experimental and theoretical investigations by different authors. New shapes and materials, different tooling configurations and new machine concepts were used whereas the fundamentals remained the same. For instance, instead of polyurethane, rubber and elastomer were used to provide internal pressure [Al-Qureshi, 1970]. He presented that greater circumferential expansion of thin-walled tubes was obtained using rubber forming methods than using hydraulic forming technique. Effect of friction between rubber and inner side of the tubes was also mentioned. Limb, et al. performed bulge forming of tubes of different materials with changing wall thickness [Limb, et al., 1973]. They reported that increasing the internal pressure gradually during the application of axial load gave the best results on thinning and complete filling. Thickening of tube wall at feeding zone was also mentioned due to the friction between tube and die surface. In addition, experimentation of different lubricants such as PTFE film, colloidal graphite and Rocol R.T.D. spray were carried out. In case of insufficient lubrication, bulging of the Tee protrusion was found to be more pronounced. With proper lubrication, it was reported that a flatter bulging of the Tee protrusion was obtained.

Woo reported experimental and analytical results for tubes bulged under internal pressure and axial compressive loading [Woo, 1973]. He carried out a numerical study assuming
that the entire length of the bulged tube was in tension, and thus, free bulging took place. Comparison of experimental and theoretical results indicated good agreement when stress-strain properties of tubes obtained from bi-axial tests were used in calculations. Use of upper bound technique to calculate internal pressure as function of material properties and geometry was presented by Powel, et al. [Powel, et al., 1973]. They tried forming of 90° elbows with sharp radius from straight tubes using hydraulic pressure and a special tooling developed for this purpose.

Limb, et al. used oil as pressurizing medium in their experiments to investigate the forming of copper, aluminum, low carbon steel and brass Tee-shaped tubular parts [Limb, et al., 1976]. Results of lubricant and material evaluations were reported in terms of protrusion height attainable. Sauer et al. presented their theoretical and experimental work on necking criterion of bulged tubes [Sauer, et al., 1978]. Assuming a constant ratio of hoop and longitudinal stresses in tube wall during expansion, numerical and experimental results were found to be in agreement. Effective strain at necking was also explained in terms of pre-strain, strain-hardening exponent and stress ratio. Woo et al. described their experimental tooling for bulge forming of tubes, and presented a theoretical analysis of stresses and strains taking into account the anisotropy effect of the sheet metals in two separate papers [Woo, 1978] [Woo, et al., 1978]. They utilized Hill’s theory of plastic anisotropy in their work.
Starting from 1980’s, researchers in Japan concentrated on determining the material properties and their effects on tube bulging operations. Manabe, et al. investigated influence of the strain-hardening exponent and anisotropy on forming of tubes in hydraulic bulging and nosing processes [Manabe, et al., 1983]. They briefly presented the maximum internal pressure as a function of tube radius, thickness, strain-hardening exponent, and strength coefficient assuming that there was no axial loading. Manabe, et al. published their work on examination of deformation behavior and limits of forming for aluminum tubes under both internal pressure and axial force [Manabe, et al., 1984]. Axial cylinders and internal pressure were controlled by a computer-control-system to obtain pre-defined stress ratio during their experiments. They utilized fundamental analysis of thin-walled cylinders in their predictions for internal pressure and axial force.

Fuchizawa analyzed bulge forming of finite-length, thin-walled cylinders under internal pressure using incremental plasticity theory [Fuchizawa, 1984]. He presented the influence of strain-hardening exponent on limits of bulge height. Similar to Manabe, et al., he utilized the fundamental plasticity and membrane theories in his predictions. Internal pressure and maximum expansion radius were expressed in terms of length, diameter, strength coefficient and strain-hardening exponent. Later, Fuchizawa extended his studies to explore the influence of plastic anisotropy on deformation behavior of thin-walled tubes under only internal pressure [Fuchizawa, 1987]. He based his analysis on deformation theory and Hill’s theory of plastic anisotropy. Longitudinal anisotropy was found be effective on the critical expansion limit while anisotropy in hoop direction was
affecting the maximum internal pressure required. With increasing anisotropy in longitudinal axis, thinning is reduced while obtainable expansion gets larger with less internal pressure requirement. Experimental results were eventually compared with theoretical findings [Fuchizawa, 1990]. Different materials including aluminum, brass and copper were tested in their tooling, which only utilized internal pressure in a closed cavity. Assuming that the tube materials obey power law of strain hardening, experimental and calculated results were found to be in good agreement. Studies of Manabe and Fuchizawa on anisotropy effects were mostly found useful in THF applications involving aluminum products.

Hydraulic bulging of tubes was later used in determining the stress-strain characteristics of tubular materials by Fuchizawa, et al.[Fuchizawa, et al., 1993]. Annealed aluminum, copper, brass and titanium tubes were tested under only internal pressure. With the instrumentation and control systems available, tube thickness, radius of curvature in both longitudinal and hoop directions, and internal pressure measured and recorded during formation of the bulge. Using analytical methods by membrane and plasticity theories, stress-strain relations were derived. These findings were also compared with those obtained from tensile tests. Stress-strain relations for aluminum, copper and brass were found to be similar by two tests, whereas that for titanium were different. Since they did not use axial compressive load during bulging, stress-strain relation obtained was limited to low strain values up to 0.7.
Thiruvarudchelvan, et al. has worked on experimental and theoretical aspects of tube bulging process using both polyurethane and liquid as pressurizing medium [Thiruvarudchelvan, 1989]. They used computers to control the process parameters and for data acquisition in their experimental systems. Optimum values for axial forces were defined to obtain large bulge heights of tubes without any fracture. Hashimi, et al., investigated the bulge forming of axisymmetric and asymmetric components via experiments, analytical techniques and FEA [Hashimi, 1983] [Hashimi, et al., 1985] [Ahmed, et al., 1997]. Tonghai, et al. presented their experimental and analytical work for forming of Tee protrusions using polyurethane [Tonghai, et al., 1993]. Upper-bound technique was used to predict total forming load. As a major contribution, use of counter force and its effect on attainable Tee protrusion height were investigated and discussed. Use of upper-bound technique in calculation of maximum internal pressure and axial force was also presented in another work [Tirosh, et al., 1995]. Free bulging of aluminum tubes were conducted, and results were compared with theoretical solutions. Stress ratios between (-1) and (0) were found to be the optimum range for bi-axial forming of tubes. However, practical difficulty to maintain the stress ratio in this zone was also reported.

Finally, Dohmann and many of his students have been working on various issues related to THF technology since early 1980's [Klaas, 1987]. Their work was mostly based on the previous theoretical studies along with real and new industrial applications of the technique [Dohmann, et al., 1991]. They also utilized the capabilities of continuously
developing FEA and computer controls in their experimental and analytical works [Bohm, 1993] [Dohmann, et al., 1994]. They have established formability diagrams for different materials under certain circumstances in order to speed up the practical use of the technology [Dohmann, et al., 1996] [Dohmann, et al., 1998].

Controlling of process parameters and part types were investigated by Schmoeckel and his students in Darmstadt, Germany [Schmoeckel, et al., 1992]. Forming of crankshaft-like parts using thick-walled tubes were performed under experimental conditions [Engel, et al., 1995] [Engel, 1995]. They have also used principle theories for prediction of process parameters, and applied them into practical industrial use. Their work has been heavily based on experiments for different aspects of the technology including formability and producibility of certain automotive parts, and tribological issues like lubrication, die surface finish and wear [Schmoeckel, 1998]. THF parts were first classified by Engel [Engel, 1995] as part of his dissertation on development of fuzzy control systems for hydroforming process. He grouped parts with respect to (a) their variation along longitudinal axis, (b) variation of the feature position relative to the longitudinal axis, and (c) variation of the cross-section. Categorization of parts depending on their shape complexity was also conducted. However, this classification was limited with only parts in exhaust systems excluding structural frame parts, whereas Koç, et al. considered structural parts such as frame rails, axles, cradles, etc. in their classification along with exhaust components [Koç, et al., 1998]. Their classification was
based on (a) spline geometry, (b) common feature types, (c) cross-section of tubes, and (d) ratio of length to diameter (L/D).

Many other authors have presented their application and practical oriented studies in numerous occasions since the beginning of 1990’s. The following issues were common topics of these presentations [Bruggeman, et al., 1996] [Longhouse, 1997] [Mason, M., 1996] [Leitloff, 1996] [Morphy, 1997]:

- Industrial applications of THF technology,
- Production of structural frame components,
- Product development and design procedures,
- Evaluation of hydroformed parts in comparison to stampings,
- Incorporation of piercing into hydroforming tooling,
- Assembly and welding issues,
- Weight savings, etc.

In the next sections, technical background on different issues related to THF process will be presented in detail.

2.1 THF parts, technology, presses, hydraulic and control systems

Various parts for automotive, appliance and plumbing are produced by THF technology, as discussed below:
• Exhaust System Parts; Usually made of Stainless Steel for obtaining required structural, thermal and corrosion properties:

  Exhaust parts, Engine tubes, Catalytic converters, Pressure tubes, Tail pipes, Connectors, Manifolds

• Chassis Parts; Common material is low to medium carbon steels and aluminum for structural and cost related reasons:

  Frame rails, Engine sub-frames (cradles), Roof rails, and bows, Instrument panels, Rear axle frames, Radiator frames

• Engine and Power train components:

  Suspension cross members, Hollow camshafts, Drive shafts, Gear shafts

• Body and Safety parts:

  Windshield headers, A/B/C pillars, Space frame components, Seat frames, Shock absorber housings,

Figures 2.1 through 2.3 illustrate examples of THF parts for automotive applications. There are also a number parts that are targeted to be produced by THF process. These are camshaft, crankshafts and casings such as differential cases. Design of the tube hydroforming machine is of special importance since high hydraulic pressures and complex shaped parts involved. The equipment needed for tube hydroforming consists of the followings [Viehweger, 1996]:

• Press; for closing the dies

• Tooling (or dies)

• Pressure system; intensifier

• Hydraulic cylinders and punches; for sealing the tube and move the material
• Process control systems; computers, data acquisition, transducers, etc.

**Presses**

In contrast to other forming operations, in tube hydroforming processes, the press is used in general only to keep the tooling closed. The closing force is dependent on the internal pressure of the process. Large components with big wall thickness, i.e. chassis components, and intricate regions, i.e. small corner radii, need high closing forces up to 7000-8000 tons [Vielweger 1996]. At the present, presses with 6000-ton capacity are in operation at many production facilities. Existing hydraulic presses with appropriate closing forces and bed sizes are being used for THF processes [Lietloff, 1997] [Cherek, 1997]. New machines manufactured particularly for THF technology are also in use at some plants [Bieling, 1997]. The purpose of developing special machines is to increase capabilities on process control, obtain better dimensional accuracy via high clamping load, access larger bed size, reduce cycle time, increase flexibility for different parts, and reduce investments, etc. A tube hydroforming press must have the following features:

• Appropriate die closing force
• Appropriate bed size to hold the dies
• Adjustable/movable axial punches with computer controlled positioning
• Adjustable/movable rams for counter forces with free and position control
• Optional: automatic workpiece handling
• High pressure (2000 to 5000 bar) and fluid pumping capability with tight control
Figure 2. 1: Examples of structural frame parts for automobile applications. In (a), Roof headers (A), Instrument panels (B), Radiator frame (C), Engine cradle and rear axle (D), Roof rails (E), and Lower rail frames (F) can be manufactured by THF. (b) Rear axle of a car produced by THF process, (tubular components were formed then welded together) [Vari-form, 1995] [BMW, 1995].
Figure 2. 2: Examples of (a) and (b) exhaust system parts such as connectors and manifolds, (c) pressure tube [Schaefer, 1996].
Figure 2. 3: (a) Differential case and its forming sequence, (b) part of a aluminum rear axle, (c) Frame rail and drive shaft [Tube Forming Inc., 1996] [BMW, 1995] [Schaefer, 1996].
Tooling

Dies in tube hydroforming are divided in either lengthwise or crosswise (Figure 2.4) direction. Crosswise split dies are cheaper to produce compared to lengthwise split dies. They also have the advantage that predefined diameter dimensions can be better controlled. Moreover, closing forces are also smaller. However, lengthwise split dies must be used, if the workpiece has bends or subsidiary shapes perpendicular to the main axis of the tubular pre-form. This is the case, for example, in parts having multiple junctions at varying angles. [Dohmann, et al., 1991].

There are two different models of the crosswise die [Dohmann, et al., 1993]. In the first case “closed die” (Figure 2.5), the forming die, which is divided transversely, is closed prior to the start of the shaping process. There are guides for holding the tubular blank at each end. The blank is pushed by means of punches throughout the forming process. Friction occurs between the tubular wall and the inside surface of the die. By using the closed die, the free length of the tube can be limited to a minimum, ensuring good tube guidance. Hence, the danger of buckling is reduced.

In the second case model “open die” (Figure 2.5), the die is open at the beginning of the forming process. The distance of the opening corresponds to the difference in the length of the tubular blank and the final workpiece. As the process progresses, the die walls are closed and the expanded tube comes into contact with the die. Using the open die, there will be no or very little friction between the tube wall and the die. The drawback is the
greater danger of buckling compared to forming with a closed tool. Buckling basically limits the forming potential of this method [Dohmann, et al., 1996].

In general, the followings are main requirements for THF tooling:

- High strength against stresses due to large internal pressure and axial loading,
- Good surface finish to minimize friction and increase formability,
- Flexibility by interchangeable inserts,
- Good guiding systems,
Figure 2.4: (a) Crosswise and (b) lengthwise split dies. Lengthwise split dies are placed in a press so that the die closure is achieved by the press ram [After Dohmann, et al., 1991].
Figure 2. 5: Closed (a) and open (b) crosswise die [After Dohmann, et al., 1996]
**Pressure System**

The pressure system (pump, intensifier and control valves) should be designed and selected so as to provide the required pressure levels for a wide range of parts to obtain flexibility in the system invested. The applied pressure should have a range from 2000 bars (30 ksi) up to 10000 bars (150 ksi) depending on the parts in consideration [Klaas, 1994]. In many current industrial applications use pressures up to 3000 bars (45 ksi) are sufficient. The flow rate can reach up to 50 liter per minute in order to allow short cycle times.

**Hydraulic Cylinders and Punches**

The axial punches should feed the material into the deformation zone in a controlled way, and in synchronization with internal pressure, i.e. pressure versus time and axial force versus time should be controlled and coordinated. Counter punches are sometimes used on bulged or protrusion sections to avoid premature fracture by providing a controlled material flow. Axial cylinders are expected to generate forces of up to 7000 kN (700 tons) while counter cylinder limits extend up to 2000 kN (200 tons). The smaller size also allows close control of the punch position.

### 2.2 Materials and formability in THF

Followings are the required characteristics of tubular materials for quality THF applications:
• High and uniform elongation
• High strain hardening exponent
• Low anisotropy
• Close mechanical and surface properties of weld line to the base material
• Good surface quality, free of scratches
• Close dimensional tolerances (thickness, diameter and shape)
• Burr free ends; should be brushed
• Tube edges perpendicular to the longitudinal axis

According to the requirements above, all alloys that are used in deep drawing or extrusion are suitable for tube hydroforming [Cudini, et al., 1988] [Schmoeckel, et al. 1995]. Table 2.1 tabulates some of the tubular materials used in THF process.

In addition, available tube types can be listed as follows:
• Seamless drawn circular tubes
• Seamless drawn tubular profiles
• Longitudinally seam welded circular tubes
• Longitudinally seam welded tubular profiles
• Tailored tubes; round seam welded or longitudinally seam welded
Different testing methods have been used to determine the quality of tubing for purposes other than THF process [Sokolowski, et al., 1998]. These tests can be listed as follows:

(a) Tensile test, (b) Expansion test, and (c) Cone test

Investigation of formability limits, failure or necking criterion and flow stress characteristics of tubular materials started with establishment of instability points in sheet metal forming processes [Mellor, 1960] [Weil, 1963].

Fuchizawa et al. conducted experimental and theoretical studies to determine the stress-strain relations of tubular materials [Fuchizawa, et al., 1993]. They developed a bi-axial testing method for tubular materials. This test uses internal hydraulic pressure to bulge tubing which is supported between two dies. Figure 2.6 shows the tooling set developed for this test. The ends of the tube are restrained by a set of dies, which are separated by a predetermined length of tubing. One of the supporting dies is restricted in movement, while the other is free to move in the axial direction, thus reducing axial stretching during the test. The internal pressure, thickness, diameter, and meridional curvatures are measured continuously, as the test is executed. From the recorded data, a stress – strain relationship is analytically determined.
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Table 2.1: Common materials for tube hydroforming
This test was used for analysis of aluminum, copper, brass, and titanium alloys. The results were compared with those of tensile testing. While the values for the aluminum, copper, and brass showed little distinction between the two testing methods, the titanium showed great difference between the properties determined by the tensile and bulge testing methods. For this reason, it was concluded that a bi-axial hydrostatic testing method should be used for testing materials to be used with tube hydroforming processes.

The effects of the strain hardening exponent and plastic anisotropy were thoroughly discussed through theoretical analyses in his other presentations [Fuchizawa, 1984] and [Fuchizawa, 1987] respectively.

The strain hardening exponent (n-value) study showed that as the n-value increased, the internal pressure required to form a certain bulge height is decreased, thickness distribution became more uniform, and greater expansion was realized. This result was also found by Manabe, et al. [Manabe, et al., 1983]. Results of the plastic anisotropy study showed that the r-value in circumferencial direction affected in the internal pressure, while r-value in longitudinal direction affected the maximum expansion of the tube. Sokolowski, et al., utilized an approach similar to that of Fuchizawa in order to determine the flow stress curves of low carbon and stainless steel tubes, Figure 2.7. They introduced the use of FEA as additional tool to the analytical and experimental techniques [Sokolowski, et al., 1998]. Both studies were limited with bulging with only internal pressure. Thus, working strains were in the range of 0.1 to 0.7
Figure 2.6: The bi-axial formability test instrumentation and tooling. This test uses internal pressure to bulge the tubing, which is supported between two dies [Fuchizawa, et al., 1993].

![Flow stress for 304 Stainless Steel](image)

Figure 2.7: Flow stress for 304 Stainless steel, determined with analytical and simulation means [Sokolowski, et al., 1998]
2.3 Friction in THF process and evaluation of lubricants

Until recent years, there was not any reported testing methods or equipment development to measure or evaluate friction in tube hydroforming process. However effect of friction and different lubricants on formability and extend of protrusion height was mentioned at many occasions starting 1970's [Limb, et al., 1973]. Thickening of tube wall at feeding zone was also reported due to the friction between tube and die surface. In addition, experimentation of different lubricant such as PTFE film, colloidal graphite and Rocol R.T.D. spray were carried out. In case of insufficient lubrication, bulging of the dome of the Tee protrusion was found to be more pronounced. With proper lubrication, it was reported that a flatter bulging of the Tee protrusion was obtained. As reported by Ahmed, et al., Hutchinson carried out experimental studies to investigate the effect of different lubricants on bulging of tubes [Ahmed, et al., 1996]. The influence of the following parameters on friction and forming should be examined:

- Lubricants
- Surface pressure (the most important influence)
- Sliding velocity
- Work piece and die materials and their surface conditions
- Coating on the die, if any
- Effects of the parting-line (in transversal and longitudinal direction) on the forming process.
Schmoeckel, et al. identified different friction zones on a typical THF process depending on the effects of axial force, feeding and geometrical aspects [Schmoeckel, et al. 1997]. The surface pressure, sliding velocity, and state of stress and strain were identified to be different in these zones as follows (Figure 2.8):

- Guide zone,
- Transition zone,
- Expansion zone

In these three zones, the following conditions prevail:

- **Guide Zone:**
  - Medium surface pressure
  - High sliding velocity
  - High axial pressure
  - Little expansion of the surface;

- **Transition Zone:**
  - Surface expansion or reduction
  - Sliding velocity smaller than that of the guide zone, but still appreciable
  - Stresses are somewhere between axial pressure and tensile hoop stress
  - Tensile stresses in the tube are in hoop direction;

43
• Expansion Zone:
  • Tensile stresses are prevalent (axial and hoop direction)
  • Sliding velocity is small
  • Surface enlargement is large.

In order to investigate the influence of the above parameters in different zones of friction, Schmoeckel et al. used an experimental setup where a straight tube is expanded under internal pressure and pushed to investigate the friction conditions in only guide zone [Schmoeckel, et al., 1997]. During the same period, Dohmann, et al. developed another tooling, which would permit investigation of friction in all zones [Dohmann, et al., 1997]. Similar experimental tooling were developed in the US at ERC/NSM of the Ohio State University [Lorenz, et al., 1998].
Figure 2. 8: Schematic of a basic tooling design for friction testing, and various friction zones during a typical hydroforming process.
2.4 Pre-forming of tubes for hydroforming process

Many tube hydroforming operations require a pre-formed tube in order to reach the desired shape at the end of the process. Pre-forming of tubes usually includes bending and crushing operations. Furthermore, annealing may be necessary after bending or crushing to remove residual stresses.

In order to take the effects of pre-forming into account and to design parts, tooling and process parameters properly, an investigation of bending, crushing and annealing of tubes is required. In literature, investigations in this area have been dated to very recent years, and in a limited manner [Schmoeckel, et al., 1997] [Dohmann, et. al., 1998]. These investigations are mainly experimental [Longhouse, 1997] or based on FEA [Hurton, 1997] of complex shaped parts. In addition, there is enough background and experience in tube bending for other purposes [Zhang, et al., 1996] [Granelli, 1997].

In order to analyze the entire THF process, it is a good idea to use FEA for both pre-forming and THF. This way, strain history gained during pre-forming will be directly carried into the THF stage just as in actual forming of complex parts. Along with FEA, theoretical analyses can be also performed for simple cases [Shr, et al., 1998] or two-dimensional conditions like cross-section of a part.
CHAPTER 3

CLASSIFICATION OF PARTS PRODUCED BY TUBE HYDROFORMING PROCESS

Parts that are produced by tube hydroforming process vary over a wide range of shapes. This variety goes from a simple bulged tube to a complex exhaust pipe with multiple features (like protrusions and bends), or to a frame rail with long longitudinal axis along which different cross-sections blend each other. There are also various candidate parts with different shapes and characteristics that are suitable for THF process. Figure 3.1 depicts some of THF parts.

It is necessary to classify the THF parts into different categories with respect to the common characteristics they have in order to handle the design process with ease and without trial-and-error. Development of design guidelines would be also facilitated. Mainly, THF parts have the following common features on them (see Appendix A for detail):
(a) **Protrusion**: Tee and Y protrusions are manufactured to provide connections particularly in exhaust parts. Figure 3.2 illustrates a Y protrusion.

(b) **Bulging**: It is local expansion of a tube either freely or into a die cavity. A typical simple bulged part is shown in Figure 3.3.

(c) **Bend**: Figure 3.2 illustrates a bend section encountered on many THF parts

(d) **Crushing**: Initial shape of crushing is given in pre-forming (crushing) stage. They are a common group of feature encountered frequently especially on structural parts. However, hydroforming gives them their final shape. Figure 3.2 presents a typical crushed section on a part.

This task would help designers to approach the design of a new part systematically, and reduce time, effort and cost of product development by shortening trial-error procedure. Moreover, important geometrical parameters of various parts would be identified for eventual use during prediction of process parameters.
Figure 3. 1: Examples of automobile parts produced by tube hydroforming process [Schaefer, 1995] [Huber & Bauer, 1995].
Figure 3.2: Bulge, protrusion, bend, and crushing are common features that can be found on THF parts.
In addition, parts can be divided into groups in terms of their spline geometry along which they are formed. Spline shape of parts is important since it determines the position of axial cylinders during production.

Another criterion that can be used to distinguish THF parts is their overall length to diameter ratio (L/D). For instance, exhaust parts have smaller (L/D) ratios when compared with structural frame parts whose (L/D) ratio may vary between 20-50. Furthermore, structural parts with large (L/D) ratios usually extend along their one- or two-dimensional longitudinal spline with varying cross-sections. Their cross-section vary from round to rectangular shapes to provide flatness where other parts of the car are assembled.

3.1 Classification of part

In accordance to the above discussion, it is possible to group THF parts according to their spline shape as follows (detailed drawings of them are presented in Appendix A):

(a) One-dimensional (1D) parts (the spline is a straight line)
(b) Two-dimensional (2D) parts (the spline is a curved line on a plane)
(c) Three-dimensional (3D) parts (the spline is space curve)
(d) Structural frame parts (the spline is a curve in a plane or space, and the cross-section varies considerably along the spline)
**One-dimensional (1D) parts:**

- Figure 3.3 illustrates this type of parts,
- They may have multiple features,
- Feature type varies (i.e. protrusion, crushing or bulge)
- They do not have a “bend” feature,
- They lie in a plane,
- Their ends have to be on “one line”.

**Two-dimensional (2D) parts:**

- This type of parts has a spline lying in a plane (Figure 3.4)
- They may have single or multiple features along their spline including “bends”,
- Their ends may lie on
  - “one line”,
  - “two-parallel lines”
  - “two lines at an angle to each other”,

**Three-dimensional (3D) parts:**

- Figure 3.5 depicts examples of this category.
- Spline of this type goes through different planes, their shape can not be described in one plane.
• They are rather complex parts with necessary pre-forming operations like 3-D bending,
• They may have single or multiple various features (including bends, protrusions, etc.).
• Their ends may go through
  
  “one line”
  “on two-parallel lines”
  “two lines at an angle to each other”,

Structural frame parts:
• These are the frame and chassis parts with large (L/D) ratios,
• Their (L/D) ratio varies between 10-50 depending on the application and material,
• They usually have large wall thickness when compared to exhaust parts,
• Their ends may go through
  
  “one line”
  “on two-parallel lines”
  “two lines at an angle to each other”
• Figure 3.6 and Figure 3.7 illustrates this type of automotive parts
<table>
<thead>
<tr>
<th>Diagram</th>
<th>Description</th>
</tr>
</thead>
</table>
| ![Diagram 1](image1.png) | - Part with 1D spline  
  - Ends have to be on "one line"  
  - Carries single protrusion |
| ![Diagram 2](image2.png) | - Part with 1D spline  
  - Ends have to be on "one line"  
  - Carries double protrusions on the same side |
| ![Diagram 3](image3.png) | - Part with 1D spline  
  - Ends have to be on "one line"  
  - Carries double protrusions on the opposite sides |
| ![Diagram 4](image4.png) | - Part with 1D spline  
  - Ends have to be on "one line"  
  - Carries double protrusions at angle to each other |
| ![Diagram 5](image5.png) | - Part with 1D spline  
  - Ends have to be on "one line"  
  - Carries double protrusions at angle to each other  
  - Protrusions are at a distance from each other |

Figure 3.3: One-dimensional (1D) THF parts with various features.
<table>
<thead>
<tr>
<th>Diagram</th>
<th>Description</th>
</tr>
</thead>
</table>
| ![Diagram](image1.png) | Part with 2D spline  
Ends are on “one line”  
Carries multiple bends and a single protrusion |
| ![Diagram](image2.png) | Part with 2D spline  
Ends are on “two parallel lines”  
Carries multiple bends and a single protrusion |
| ![Diagram](image3.png) | Part with 2D spline  
Ends are on “two lines at an angle to each other”  
Carries a single bend and several protrusions |
| ![Diagram](image4.png) | Part with 2D spline  
Ends are on “two lines at an angle to each other”  
Carries single bend and several protrusions  
Protrusions are also at angle to each other |

Figure 3.4: Parts with two-dimensional spline. 2D parts lay down on one plane. These parts may have different geometric features along the spline.
<table>
<thead>
<tr>
<th>Part with 3D spline</th>
<th>Ends are on “one line”</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carries multiple “bends” and single “bulge”</td>
<td>With round-like cross-section</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Part with 3D spline</th>
<th>Ends are on “two parallel lines”</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carries multiple “bends” and single “protrusion”</td>
<td>With round-like cross-section</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Part with 3D spline</th>
<th>Ends are on “two lines at an angle to each other”</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carries multiple “bends” and “protrusions”</td>
<td>With round-like cross-section</td>
</tr>
</tbody>
</table>

Figure 3. 5: Three-dimensional (3D) THF part types with various feature and end positioning.
Figure 3. 6: Examples of structural frame parts for automobile applications. In (b), Roof headers (A), Instrument panels (B), Radiator frame (C), Engine cradle and rear axle (D), Roof rails (E), and Lower rail frames (F) can be manufactured by THF. (c) Engine cradle of a car produced by THF process, then welded together. [Shah, 1997], [Vari-form, 1995], [BMW, 1995].
Figure 3.7: Variety of structural parts manufactured by THF process for automotive applications.
THF parts can also be categorized according to their initial cross-sectional shapes. Even though the cross-section may change over the tube length, major cross-section that is of importance to the designer may be utilized for classification purposes. Initial tube cross-section shapes can be divided into four groups as follows:

(a) Round-like cross-section
(b) Elliptical-like cross-section
(c) Rectangular-like cross-section
(d) Irregular shaped cross-section

Figure 3.8 shows these cross-section types. Please note that this is an approximate classification to be used as tools for the design purposes.

The premise of classification of THF parts is that design of parts with similar features can be generalized by a modular design concept. This way, any experience or information gained during design and production of a particular part can be utilized during design of other similar parts.

Effect of geometrical parameters (features, splines, cross-sections) on producibility and required process parameters can be analyzed in detail for each group under particular process conditions. Then, the results of this detailed analysis can be extended for other
conditions of similar parts under certain circumstances. For example, common features may have the following crucial geometrical parameters (Figure 3.9):

(a) Corner radii, \( (R_c) \)
(b) Entry (fillet) radii, \( (R_e) \)
(c) Distance between features, \( (L_{pp}) \)
(d) Distance between feature and edges, \( (L_{pe}) \)
(e) Height of feature, \( (H_p) \)
(f) Diameter of feature, \( (D_p) \)
(g) Angle of feature with respect to the spline of the part, \( (A_p) \)

The process conditions under which parts are formed can be listed as follows:

(a) Internal pressure and axial loading from both ends
(b) Internal pressure and axial loading from one end
(c) Internal pressure without any axial loading
(d) Internal pressure, axial loading from both ends, and counter loading
(e) Internal pressure, axial loading from one end, and counter loading
(f) Internal pressure and counter loading
Figure 3.8: Common and possible cross-sections of initial tubes used in THF process.
Figure 3.9: One-dimensional (1D) part with two Y protrusions. Important geometrical parameters are shown. \( L_{pe1} \) and \( L_{pe2} \) are distance between feature (protrusion) and edges. \( L_{pp1} \) is the distance between features. \( H_p \) is height of feature. \( D_p \) is diameter of feature. \( D_i \) is diameter of base tube. \( A_p \) is the angle that feature makes with base. \( R_{e1} \), \( R_{e2} \), \( R_{e1} \), and \( R_{e2} \) are corner and fillet (entry) radii on the feature, respectively.
3.2 Collection of available experimental and FEA data into a database (TH-DBASE)

Design of part, tooling and process for tube hydroforming process is an “art”. Hence, in order to have continuity, improvement and effectiveness in design process, information, design rules and experience should be preserved and transferred to others. As mentioned before, classification of parts was an initial step towards achieving this objective.

In order to gather the information and experience available, a systematic and permanent method would be to use the current computer technology and to construct an expandable database of available experimental and FEA results.

A database, TH-DBASE, was initiated in order to start this effort. It is hoped that along with other tools of analysis and design, this database will help designer during the initial phases of part, tooling and process design for tube hydroforming technology.

This database simply consists of two modules. The first one is the module where the experimental and FEA data are stored in terms of geometry, picture, material, and dimensions. The second module is an interface, which facilitates the use of stored data. It helps searching of the stored data. In addition through this interface, new and additional information can be stored easily.
Storage of the data is based on, first, the part classification outline in the previous sections. Second, material of the work-piece, and third dimensions of the part such as diameter and thickness.

Hence, a designer can search for or store in a part knowing its overall classification, material, and dimensions. TH-DBASE will, simply, bring the information closest to the “desired” conditions to the attention of user. Using this current information, designer may design his/her own part, or have changes on different parameters by simple comparisons.

Eventually, when this database becomes large enough for every part type and material, linear interpolations and extrapolations can be added to further assist the designer. However, as it will be explained in the last chapter of this study, author hopes that a new concept of “optimization of loading paths via finite element analysis” may eliminate or reduce the importance of this need.
CHAPTER 4

PREDICTION OF PROCESS PARAMETERS

Simultaneous and coordinated application of internal pressure and compressive axial force provides complete forming of tubular parts into a given die shape without any defects. Due to the desired final geometry and initial conditions of the tubes, usually very large tensile tangential (hoop) strains are required in tube hydroforming. In addition, these large plastic strains must be obtained without causing any instability (necking, wrinkling and buckling) on the part being formed.

Considering the fact that most of the expansion takes place while tube is not in contact with the die surface yet, having large plastic deformation without any instability requires tight controlling and synchronization of the process parameters. In order to reach this condition, usually compressive axial force is applied to provide a stress field such that large tensile hoop strains can be achieved. However, this introduces another variable that must be controlled carefully. Otherwise, excessive compressive axial force may cause buckling or wrinkles prior to desired expansion. Therefore, proper controlling of internal
pressure and axial force is a must to achieve defect free parts with required expansion ratios. As a consequence, prediction of these process parameters becomes a necessary task in design of tube hydroforming process.

In this section, prediction of process parameters, such as Internal Pressure ($P_i$), Axial Force ($F_a$), and Counter Force ($F_c$), will be explained using the fundamentals described in the Appendices B, C, and D. In addition to these parameters, prediction of wall thinning and calculation of clamping force will be presented. Figure 4.1 illustrates the process parameters involved in a typical tube hydroforming operation.

Analytical methods presented in this report may not predict the exact values of the process parameters in question since process parameters in tube hydroforming will vary depending on the part shape, loading conditions, and material type. However, these methods will provide a guideline for further analysis of the process via sophisticated tools such as FEA. In addition, these methods will help the designer to understand the fundamentals of the process.
Figure 4.1: Process parameters during forming of a crankshaft-like tubular part. In this figure, \( (P_i) \) is Internal Pressure, \( (F_a) \) is Axial Force, \( (F_q) \) is Counter Force, and \( (F_t) \) is Transverse Force.
4.1 Calculation of internal pressure ($P_i$)

In tube hydroforming process, Internal Pressure ($P_i$) is applied to the internal walls of the tubular part via pressurizing of fluid medium filled into the tube prior to the forming. In order to manufacture a part without any defects and within required dimensional tolerances, internal pressure versus time needs to be calculated and designed properly. ($P_i$) varies depending on the material, part shape, thickness, intricate section sizes (such as corner radius), and other process parameters involved in the process (i.e. axial and counter forces). Thus, Internal Pressure ($P_i$) in function of time should be:

- High enough to prevent buckling due to excessive compressive axial force at the beginning of forming. This value, ($P_i$)$_{\text{min}}$, is determined by the minimum ($P_i$) required for preventing buckling or wrinkling at very beginning of the forming.

- High enough to start deformation of the tube walls at the initial stages of deformation. This value, ($P_i$)$_{\text{yp}}$, is usually dictated by the yield strength of the tubular material.

- High enough to form the tubular material into intricate die cavities (such as corners). This limit, ($P_i$)$_{\text{max}}$, is dependent on the ultimate tensile strength, corner radius and thickness.
• High enough not to allow any wrinkling during the intermediate stages of the forming due to excessive compressive axial force.

• Low enough not to cause instability by necking (i.e. bursting pressure). This value, \( (P_i)_n \), can be estimated by the instability criterion. It depends on the ultimate tensile strength of the tubular material, strain hardening exponent \( (n) \), thickness \( (t) \), tube radius \( (r) \) and loading conditions \( (\alpha) \).

Note that introduction of other process parameters, such as axial and counter forces, would change the required \( (P_i) \) at different stages of forming.

As a result, in this section, estimations \( (P_i)_{\text{min}} \), \( (P_i)_{\text{sp}} \), \( (P_i)_b \) and \( (P_i)_{\text{max}} \) will be presented briefly using the fundamentals mentioned in Appendices B, C, and D. Minimum internal pressure at the beginning of the deformation should be high enough to prevent buckling or wrinkling of the tube walls due to the compressive axial force applied to seal the ends of the tube. This can be found by equating the forces acting on the tube wall as seen in Figure 4.2.

\[
(P_i)(\text{Area}) = (F_a)(\tan \beta) \quad (4-1)
\]

\[
(P_i)(\ell)(2\pi r) = (F_a) \left( \frac{x}{\ell} \right) \quad (4-2)
\]
Assuming the value of thickness instead of buckling height ($x$), we obtain

\[
(P) = \left( F_u \right) \left( \frac{t}{\pi r \ell^2} \right)
\]

If axial force is assumed to be the initial sealing force on the tube walls:

\[
F_s = 2\pi rt\sigma_{yp}
\]

Minimum pressure will be found as:

\[
(P) = 2\left( \sigma_{yp} \right) \left( \frac{t}{\ell} \right)^2
\]

Where minimum internal pressure, $(P)_{min}$, is given as a function of yield strength, wall thickness ($t$), and tube length ($\ell$).
Figure 4.2: Force equilibrium on a tubular part being forced axially and pressurized internally at beginning of the process [After Hibbeler, et. al., 1993].
Using Equation (C-8) obtained as a result of derivations in Appendix C, internal pressure in a thin-walled tube can be related to the hoop and longitudinal stresses as follows:

\[
\frac{P_i}{t} = \left[ \frac{\sigma_\theta + \sigma_z}{\rho_1 \rho_2} \right]
\]

For a cylindrical tube

\[\rho_1 = r \quad \rho_2 = \infty\]

Then, Equation (C-8) can be reduced to the following:

\[P_i = \sigma_\theta \frac{t_1}{r_1} \quad (4-4)\]

Similarly, using hoop and radial strain equations, see Equation (B-22) in Appendix B, it can be written that:

\[r_1 = r_o e^{e_\theta} \quad t_1 = t_o e^{e_r} = t_o \left( \frac{r_1}{r_o} \right)^{\frac{1+\alpha}{\alpha-2}}\]

Where

\[\alpha = \frac{\sigma_z}{\sigma_\theta}\]

Substituting into Equation (4-4):

\[P_i = \frac{\sigma_f t_o e^{e_r-e_\theta}}{\sqrt{1-\alpha + \alpha^2} r_o} \quad (4-5)\]

Here, flow stress can be expressed in the following form:

\[\sigma_f = K(e_o + \bar{e})^n\]

Proceeding further with substitution of flow stress and final thickness, we obtain:
\[ P_i = \frac{K(\varepsilon_0 + \bar{\varepsilon})^\alpha}{\sqrt{1 - \alpha + \alpha^2}} \frac{t_o}{r_i} \left(\frac{r_i}{r_o}\right)^{1+\alpha} \]  

(4-6)

Or, it can be written as:

\[ P_i = \frac{K(\varepsilon_0 + \bar{\varepsilon})^\alpha}{\sqrt{1 - \alpha + \alpha^2}} \frac{t_o}{r_o} e^{-\varepsilon_\theta (2+\alpha)} \]  

(4-7)

Equations (4-5) through (4-7) indicate that internal pressure required for bulging a thin-walled tube into any final radius \( r_i \) and thickness \( t_i \) can be estimated as a function of flow stress \( \sigma_f \), initial median radius \( r_o \) and thickness \( t_o \), stress ratio \( \alpha \), degree of expansion, \( \varepsilon_\theta \) and thinning \( \varepsilon_r \).

In order to predict the internal pressure that is enough to cause starting of deformation, Yield Strength \( \sigma_{yp} \) of the tubular material can be substituted instead of flow stress value in Equation (4-7):

\[ (P_i)_{yp} = \frac{\sigma_{yp}}{\sqrt{1 - \alpha + \alpha^2}} \frac{t_o}{r_o} e^{-\varepsilon_\theta (2+\alpha)} \]  

(4-8)

Equation (4-8), for yield pressure estimation, can be simplified as follows for practical calculation purposes:

\[ (P_i)_{yp} = \sigma_{yp} \frac{2t_o}{(2R_o - t_o)} = \sigma_{yp} \frac{2t_o}{(D_o - t_o)} \]
Similarly, Ultimate Tensile Strength ($\sigma_{UTS}$) can be substituted into the flow stress value to estimate the bursting pressure:

$$ (P_B) = \frac{\sigma_{UTS}}{\sqrt{1-\alpha + \alpha^2}} \frac{t_o}{e^{(-\alpha^2)}} $$  \hspace{1cm} (4-9a)

However, for practical and quick calculations, bursting pressure can be approximated as:

$$ (P_B) = \sigma_{uts} \frac{2t_o}{(2R_o - t_o)} = \sigma_{uts} \frac{2t_o}{(D_o - t_o)} $$  \hspace{1cm} (4-9b)

Figure 4.3 shows the comparison of bursting pressures obtained from experiments and predictions by Equation (4.9b) for axisymmetric bulge parts of different material, diameter and thickness. Some of the experimental information is obtained from Klaas [Klaas, 1987], the rest is obtained from formability tests at ERC (see Chapter 6) [Sokolowski, 1998] [Arnold, 1999] [Aueulan, 1999]. Predictions agree with the experimental findings within a reasonable range.

Internal Pressure at some special conditions can be described as follows using Equation (4-5), stress ratio ($\alpha$), and strain ratio ($\beta$):

$$ (P_I)_{\alpha=-1} = \frac{\sigma_f}{\sqrt{3}} \left( \frac{t_o}{r_i} \right) \quad @ \quad \alpha = -1 \quad \beta = -1 $$  \hspace{1cm} (4-10)

Similarly, Internal Pressure at ($\alpha = 0$, where $\beta = -0.5$) is as follows:

$$ (P_I)_{\alpha=0} = \sigma_f \left( \frac{t_o}{r_o} \right)^{\frac{1}{2}} e^{\left( \frac{r_o}{t_o} \right)^2} $$
Figure 4.3: Comparison of maximum expansion (burst) pressure obtained from experiments and predictions for different materials, diameter and thickness values.
\[(P_i)_{\alpha=0} = \sigma_f \left( \frac{t_o}{r_o} \right)^{3/2} \left( \frac{r_o}{r_i} \right)^{3/2} \]  \hspace{1cm} @ \hspace{1cm} \alpha = 0 \hspace{1cm} \beta = -1/2 \hspace{1cm} (4-11)\]

Combining Equations (4-10) and (4-11), internal pressure estimation can be approximated as follows:

\[(P_i) = \sigma_f \left( t_o \right) \left( \left( \frac{r_o}{r_i^{3/2}} - \frac{1}{r_i^{3}} \right) \alpha + \frac{r_o}{r_i^{3}} \right) \]  \hspace{1cm} (4-12)\]

Equation (4-12) implies that internal pressure is dependent on the flow stress \( \sigma_f \) of the tubular material, initial median radius \( r_o \) and wall thickness \( t_o \), stress ratio \( \alpha \), and final radius \( r_i \). Figure 4.4 illustrates the variation of Internal Pressure with stress ratio \( \alpha \) and tube radius \( rl \). It simply represents the Equations (4-5) through (4-12).

In order to estimate the maximum internal pressure required for forming of intricate die cavities (i.e. corner radii), it is found quite useful to model the corner radius regions as thick-walled cylinders as shown in Figure 4.5. This is because radius to thickness ratio at this region is less then 10. Then, maximum pressure required could be calculated using the Equation (D-8) as derived in Appendix D.

\[(P_i)_{max} = \sigma_f \left[ k \ln \frac{r_c}{r_o} \right] \]
Figure 4.4: Variation of Internal Pressure with stress ratio ($\alpha = \sigma_z / \sigma_\theta$) and tube radius ($r_1$). For each case, there is a critical point where pressure starts decreasing even bulge radius increases. ($\sigma_f = K(\varepsilon_o + \bar{\varepsilon})^n$)
Substituting the followings into the equation above:

\[ k = \frac{2}{\sqrt{3}} \quad \text{for Von Mises criterion} \]

\[ r_a = r_c - t \]

We obtain the following:

\[ (P_i)_{\max} = \frac{2}{\sqrt{3}} \sigma_f \left[ \ln \frac{r_c}{r_c - t} \right] \quad (4-13a) \]

As it is obvious, maximum internal pressure, \((P_i)_{\max}\), is mainly dictated by flow stress of the material (usually Ultimate Tensile Strength is used instead), corner radius \((r_b)\), and thickness \((t)\).

### 4.1.1 Comparison of analytical predictions with experiments for corner radius case

As reported by Mason [Mason, 1996], SAE 1010 tubular material at three different thickness values was pressurized into an intricate region (i.e. corner). Corner radius at different pressure levels for each initial thickness values were recorded as illustrated in Figure 4.6 [Mason, 1996]. It depicts the variation of \((R_c / t_o)\) ratio with internal pressure. Figure 4.7 shows the comparison of experimental and predicted maximum pressure values. Faupel presented a similar equation that takes the yield and ultimate tensile strength of the thick-walled tubes into account simultaneously [Faupel, J.H., 1956] as follows:

\[ (P_i)_{\max} = \frac{2}{\sqrt{3}} \sigma_{yp} \ln \left( \frac{r_c}{r_c - t} \right) \left[ 2 - \frac{\sigma_{yp}}{\sigma_{ult}} \right] \quad (4-13b) \]
Figure 4.5: Corner radius regions in tube hydroforming processes can be well modeled and approximated by thick-walled cylinder approach. Wall thickness is \( t = r_c - r_a \)
Figure 4.6: Measured data; variations of the ratio of corner radius to initial thickness $(R_c / t_o)$ with internal pressure for different thickness values of SAE 1010 material [after Mason, 1996]
Figure 4.7: Variation of maximum internal pressure with tube radius to thickness ratio \( \left( \frac{r_c}{t} \right) \) for thick-walled cylinder approach.
4.2 Calculation of axial force, \((F_a)\)

In order to compensate the thinning in the expansion regions due the pressurizing, compressive axial force is required for moving the material into those regions. This way, not only thinning and eventual fracture will be prevented or postponed but also, introducing an additional stress component in those expansion regions will extend forming boundaries of the tubular material. In industrial applications, hydraulic cylinders are used for application of axial forces into the tube walls. Axial cylinders (punches) in tube hydroforming operations should:

- Provide enough sealing force to counteract the pressure at the beginning of the process. This value is equivalent to the force on the punch area due to internal pressure: \((F_p)\) and \((F_{so})\)

- Give enough force to overcome resistance due the friction between tube and die walls. Shear friction stress acting along the friction region area will be the resulting value for this component: \((F_f)\)

- Deliver enough force to cause the tube wall deforming. Usually, longitudinal stress acting on the tube cross-section area will be equal to this component: \((F_n)\). For practical purposes, flow stress of the tube material is used in calculations.
• Be coordinated with the application of \( \bar{P}_i \) and other forces so that desired part could be formed without any defects and within given specifications.

Figure 4.8 depicts a typical end condition of a tube being bulged by compressive axial forces from both ends and internal pressure. As explained above, common and major force components acting in this region can be expressed as follows:

Initial sealing force,

\[ F_{so} = \pi (R_o t_o \sigma_{yp}) \]  
(4-13a)

Sealing force during forming (i.e. reaction force),

\[ F_r = \pi (R_i t_i \bar{P}_r) \]
\[ F_p = \pi (R_i - t_i \bar{P}_r) \]  
(4-13b)

Friction force

\[ F_f = 2\pi (R_i \ell \mu \bar{P}_r) \]  
(4-14b)

Forming force,

\[ F_u = 2\pi (R_i t_i \sigma_z) \]
\[ F_u = 2\pi \left( R_i - \frac{t_i}{2} \right) (t_i \sigma_z) \]  
(4-15)

From the force equilibrium in Figure 4.8, it can be written that:

\[ F_u = F_{so} + F_r + F_f + F_u \]
\[ F_u = \pi R_o t_o \sigma_{yp} + \pi (R_i - t_i \bar{P}_r) + 2\pi (R_i \ell \mu \bar{P}_r) + 2\pi \left( R_i - \frac{t_i}{2} \right) (t_i \sigma_z) \]  
(4-16)
Figure 4.8: Axial forces involved in a typical tube hydroforming process. $(F_f)$ is the friction force due to friction between tube and die walls. $(F_p)$ is the force applied on the punches due to internal pressure. $(F_u)$ is the force that tube walls take for deformation. $(F_a)$ is the total minimum axial force that needs to be applied by the axial cylinders in order to form the tube material into the desired die cavity. (1) Axial punch, (2) Tube wall, (3) Die.
Hence, total axial force is dependent on the outer tube radius, wall thickness, internal pressure, axial feeding, friction, and axial stress on the tube walls. After substituting $(\sigma_z)$ and $(P_i)$ from Equations (B-27) in Appendix B, and (4-5), respectively, we obtain:

$$
F_a = \left[\pi R_o t_o \sigma_y \right] + \left[ \pi (R_i - t_i)^2 \frac{\sigma_f}{\sqrt{1 - \alpha - \alpha^2}} \frac{t_o}{r_i} \left( R_i \left( \frac{R_i}{R_o} \right)^{\frac{1+\alpha}{\alpha-2}} \right) \right] \\
+ \left[ 2\pi (R_i) \ell \mu \frac{\sigma_f}{\sqrt{1 - \alpha - \alpha^2}} \frac{t_o}{r_i} \left( R_i \left( \frac{R_i}{R_o} \right)^{\frac{1+\alpha}{\alpha-2}} \right) \right] \\
+ \left[ \pi t_o \left( \frac{R_i}{R_o} \right)^{\frac{1+\alpha}{\alpha-2}} \left( 2R_i - t_o \left( \frac{R_i}{R_o} \right)^{\frac{1+\alpha}{\alpha-2}} \right) \frac{\alpha \sigma_f}{\sqrt{1 - \alpha - \alpha^2}} \right]
$$

Equation (4-17) states that total compressive axial load is a function of initial and final outer tube radius, $(R_o)$ and $(R_i)$; initial and final tube wall thickness, $(t_o)$ and $(t_i)$; internal pressure $(P_i)$; axial feeding length $(\ell)$; shear friction coefficient $(\mu)$; and flow stress of the tubular material $(\sigma_f)$ as well as stress ratio $(\alpha)$. Figure 4.9 depicts a comparison of measured and predicted axial force curves for an axisymmetric bulge part. Pressure versus force profile used in the experiments $(a)$ can be predicted within a reasonable accuracy $(b)$. The deviations may be due to the material information used in the calculations since they are properties of sheet material not tubular material. Furthermore, it may be due to the modeling of the friction in the analytical models since only a single coulomb friction coefficient is used whereas in the actual process friction coefficient changes with time and location.
Figure 4. 9: Comparison of experimental and predicted axial force profiles used in deformation of an axisymmetric bulge part with the dimensions shown above.

Material St 30 Al (Low carbon steel), UTS = 420 MPa
$L_0 = 300 \text{ mm}, D_0 = 100 \text{ mm}, t_0 = 6 \text{ mm}$
4.3 Calculation of counter force, \( (F_y) \)

In tube hydroforming, expansion of tubes takes place either freely at the beginning stages or against a static die wall during the rest of the forming. Thus, until the tube contacts the die wall, there is a bi-axial state of stress at the expansion regions. If the required expansion is more than what tube material can sustain, necking and consequently fracture occur. Obviously, this is a major limitation on formability of the tubes in hydroforming process.

In order to deal with this problem, following steps (one by one or a combination of them) should be carried out

- Expansion is limited to some known safe degrees,

- Axial load is applied to the tube walls to move some material into the expansion region. This would compensate the thinning of the walls,

A load that counteracts the expansion in a controlled manner is applied at that region (Figure 4.10). This would increase the degree of formation by changing the state of stress from bi-axial to tri-axial case where obtainable effective stress and strain are higher than on the former state of stress.
Figure 4. 10: Counter force \( (F_q) \) is used in a typical protrusion forming (Tee protrusion) along with axial load \( (F_a) \) and \( (P_i) \). In a close view, all acting forces at protrusion region are illustrated. (1) Counter punch, (2) Tube wall, (3) Die.
Use of counter force becomes particularly necessary for cases where the amount of expansion is large, and there is a less possibility of feeding material into that section by axial loads. Formation of protrusions, such as T- and Y-shapes and large circumferential bulges, requires the use of counter forces. It was reported that use of counter force in elastomer forming process increased the obtainable protrusion height to 1.5 times of the original diameter for low carbon steel tubes whereas expansion ratio without counter force was found to be 1.2 [Tonghai, et. al, 1995].

However, improper use of counter forces may not guarantee the desired formability. Translation of the counter cylinder and variation of force on the cylinders need to be adjusted very carefully considering the other forces in the system (i.e. coordination of axial, counter loads, and pressure). Moreover, the addition of counter force cylinders and their controllers to the tube hydroforming system would increase the production costs by both reducing the production rate and adding some extra capital expenses. Figure 4.10 presents the acting forces on a tube hydroforming operation where a protrusion (T-shape) is being produced by the use of \((P_i)\), \((F_u)\) and \((F_q)\). The force equilibrium along the direction where counter force is acting gives the following:

\[
F_q = F_p - F_f - F_u \quad (4-19)
\]

Where

\[
F_p = \pi \left( R_{p1} - t_{p1} \right)^2 P_i \quad (4-20)
\]

\[
F_f = 2\pi R_{p1} (\ell) \mu P_i \quad (4-21)
\]
\[ F_u = 2\pi \left( R_{p1} - \frac{t_{p1}}{2} \right) t_{p1} (\sigma_z) \] (4-22)

Then, counter force would result in the following:

\[ F_q = \left[ \pi \left( R_{p1} - t_{p1} \right) P_t \right] - \left[ 2\pi \left( R_{p1} \right) \left( \mu \right) \mu \left( P_t \right) \right] - \left[ \pi \left( 2R_{p1} - t_{p1} \right) P_t \sigma_z \right] \] (4-23a)

\[ F_q = \left[ \pi \left( R_{p1} - t_{p1} \right)^2 \left( \frac{(\sigma_f)}{\sqrt{1-\alpha - \alpha^2}} \right) \left( \frac{t_{po}}{R_{po}} \left( \frac{R_{p1}}{R_{po}} \right)^{\frac{1+\alpha}{\alpha-2}} \right) \right] - \left[ 2\pi \left( R_{p1} \right) \left( \mu \right) \mu \left( P_t \right) \right] \]

\[ - \left[ \pi \left( t_{po} \left( \frac{R_{p1}}{R_{po}} \right)^{\frac{1+\alpha}{\alpha-2}} \right) \left( 2R_{p1} - t_{po} \left( \frac{R_{p1}}{R_{po}} \right)^{\frac{1+\alpha}{\alpha-2}} \right) \left( \frac{\alpha(\sigma_f)}{\sqrt{1-\alpha - \alpha^2}} \right) \right] \] (4-23b)

Equation (4-23) indicates that counter force necessary to maintain a tri-axial state of stress at the bulge region is a function of stress ratio, internal pressure, flow stress, protrusion radius and thickness.

Notice that flow stress \((\sigma_f)\) in reality of the tube material varies with the location on the tube and with time as forming takes place. However, since it is impossible to obtain this information, the flow stress \((\sigma_f)\) of the tubular material at an estimated strain could be used in predictions.
As tube wall contacts the die wall at the expansion (or protrusion) region, counter cylinders’ position should be leveled with the die wall. At this moment, counter load should be kept at a value, which is an equivalent of the reaction force (i.e. first term in Equations (4-19)). Eventually, it should be decreased gradually by an amount of the forming force (third term). This way as internal pressure expands the tube walls into the protrusion, counter cylinder will go up as applying an enough force to keep the desired state of stress.

Counter cylinder needs to be retracted (i.e. pulled in a direction of protrusion) as protrusion forms into the cavity just made. While it is retracted, the force on the cylinder should be adjusted so as to maintain the tri-axial state of stress at the protrusion walls. Even though this value changes according to the forming conditions (i.e. material, size of protrusion, etc.), it should be equal to summation of the reaction, friction and forming forces. Notice that reduction in the counter force at this stage have to be performed gradually since friction force will increase as protrusion forms. As protrusion reaches its final height, some additional counter force may be needed to permit the expansion to be completed into the corners. Figure 4.11 presents variation of counter force during forming of a Tee-shaped part as shown. The maximum counter force can be also predicted by Equation (4-23a) approximately.
Figure 4.11: Variation counter force, \( F_q \), during forming of a typical T-protrusion. This curve is obtained via FEA in PAM-STAMP. The maximum counter force (70 kN = 7 tons) can be also approximately calculated by Equation (4-23a).
4.4 Calculation of transverse force, \( (F_t) \)

An additional cylinder acting at an angle to the axial cylinder direction becomes necessary in order to manufacture parts with bends in tube hydroforming dies. Crankshaft-like parts as shown in Figure 4.1 and Figure 4.12 are examples of this type. Then, prediction and design of another force component, called Transverse Force \( (F_t) \) is required.

This force basically needs to overcome:

- Reaction force acting on transverse punch area,

- Friction force generated due to the friction between tube and die walls along the length transverse punch moves, and

- Resistance force due to the deformation of material throughout the tube wall.

Application of other parameters and their synchronization during the forming period will obviously affect the selection of transverse load.
Figure 4.12: Forces involved in forming a crankshaft-like part. $F_t$ is the required transverse load, $F_p$ is the force on the transverse punch due to internal pressure, $F_a$ is the force required to start deformation the tube walls, and $F_r$ is the force due to friction between tube and die walls. (1) Transverse punch, (2) Tube, (3) Die and (4) Counter punch.
Force equilibrium along the line that transverse load acts will result in the following:

\[ F_i = F_p + F_f + F_u \]  \hspace{1cm} (4-24)

\[ F_i \geq \pi(R_1 - t_1)^2 P_i + \sum_j 2\pi R_i \ell (\mu)(P_j) + \sum_j \pi_1 (2R_1 - t_1) (\sigma_j) \]  \hspace{1cm} (4-25)

Variable \((j)\) indicates that flow stress \((\sigma_f)\) of the tubular material will vary with the location on the tube and during the forming. However, since it is not an easy task to determine these flow stress values, it is a common practice and necessary assumption to use the flow stress at an estimated strain.

### 4.5 Calculation of clamping force, \((F_c)\)

Due to the high pressures \((2000-4000 \text{ bars})\) involved in tube hydroforming, large presses are required to keep the die halves together during deformation. Press selection, which is the fundamental member of a tube hydroforming system, is therefore dependent on estimation of clamping force requirements.

Clamping force is required to be high enough to keep the die halves together with tight tolerances. Hence, elastic deflection of dies is also mainly affected by clamping force selection. This would, in turn, affect the tolerances on the final part. Clamping force is
mainly dependent on internal pressure, material properties, thickness and overall size of the final part. Since internal pressure raises with decreasing recess areas, corner radii, for instance, can be directly related to clamping force. Expression below summarizes all of it in one:

\[ F_c = f(P_i, A_p) \]  \hspace{1cm} (4-26a)

While

\[ P_i = f(\sigma_f, t, R_c) \]

Analytical calculations of the clamping force can be approximately calculated by using
the internal pressure \( P_i \) and the projection area where internal pressure acts \( A_p \).
(Figure 4.13):

\[ F_c = A_p P_i \]  \hspace{1cm} (4-26b)

\[ A_p = (\ell)(D) \]

For complex parts, \( A_p \) can be expressed as follows:

\[ A_p = \sum_j (\ell_j)(D_j) \]  \hspace{1cm} (4-26c)

Where, \( j \) is the number of different sections that their projection area can simply be characterized by Equation (4-26c). Equation (4-26b) overestimates the clamping force since it does not consider the effect of tube walls. Even though overestimation can be
allowed for safety margin considerations, more precise calculations that take into account the force that is carried by the tube wall can be performed as follows:

\[ F_c = F_{\text{total}} - F_u \]  \hspace{1cm} (4-27)

- \( F_c \): Clamping force
- \( F_{\text{total}} \): Total force generated by the internal pressure, Equation (4-26)
- \( F_u \): The amount of force carried by the tube wall

The amount of force, carried by the tube wall can approximately be calculated as:

\[ F_u = \sigma_f \times A_t \]

Where:

- \( A_t \): Tube wall cross-sectional area
Figure 4.13: A simple tubular part (2) being expanded by internal pressure and fed simultaneously with compressive axial cylinder (1). In order to keep the die halves (3) together during expansion, clamping force ($F_c$) is required. This load is usually delivered by a press (4).
4.6 Prediction of final wall thickness, \( (t_f) \), i.e. thinning

As internal pressure acts on the tube walls to expand them freely or into a die cavity, tube walls become thinner and thinner. Finally, when the expansion reaches the stretchability limit of the material, first necking and eventually fracture occurs. This is known as bursting in tube hydroforming, and it is one of the common defects that must be avoided. Thus, for a given geometry, designers need to know how to predict thinning (or final thickness) of the tube wall in function of internal pressure.

Thinning will not be uniform all around the final part and during the forming. Thickness will change around the cross-section and along the longitudinal axis depending on friction, tube position and distance from ends and feature (i.e. protrusions) regions. In addition, thinning is not only affected by internal pressure but also axial and counter forces when these are used. Material properties such as ductility or elongation will also have a great influence on thinning. Therefore, prediction of thinning at the highest expansion region (most critical section) is the most important.

Thickness strain is expressed as in Equation (B-22), Appendix B:

\[
\varepsilon_t = \ln\left(\frac{t_f}{t_o}\right) = \varepsilon_t
\]

Then final thickness can be written as:

\[
t_f = t_o e^{\varepsilon_t}
\]
From Equation (B-21):

$$\frac{d\varepsilon_z}{2\sigma_z - \sigma_\theta} = \frac{d\varepsilon_\theta}{2\sigma_\theta - \sigma_z} = -\frac{d\varepsilon_r}{2\sigma_z + \sigma_\theta} = d\lambda$$

Thickness strain will be:

$$\varepsilon_r = -\varepsilon_\theta \cdot \frac{\sigma_z + \sigma}{2\sigma_\theta - \sigma_z} \quad (4-29)$$

Substituting into (4-28) and proceeding further, we will obtain:

$$t_1 = t_o \cdot \exp \left[ -\ln \left( \frac{r_1}{r_o} \right) \cdot \frac{\sigma_z + \sigma}{2\sigma_\theta - \sigma_z} \right] \quad (4-30)$$

Since:

$$\alpha = \frac{\sigma_z}{\sigma_\theta}$$

Final thickness, $t_1$, will be:

$$t_1 = t_o \cdot \left( \frac{r_1}{r_o} \right)^{\frac{1+\alpha}{\alpha-2}} \quad (4-31)$$

Equation (4-31) will give:

$$t_1 = t_o \sqrt[\alpha}{\frac{r_o}{r_1}} \quad @ \quad \alpha = 0$$

$$t_1 = t_o \quad @ \quad \alpha = -1$$
Then, \( t_i \) can be simply expressed as:

\[
t_i = t_o \left[ \left( \frac{r_o}{r_i} - 1 \right) \alpha + \frac{r_o}{r_i} \right]
\]  

(4-32)

Equation 4-32 simply implies that final thickness on a bulged tube will be dependent on the initial and final median radius, initial thickness and stress ratio, i.e. loading path. Figure 4.14 and Figure 4.15 illustrate the meaning of above equations graphically. Simply, they illustrate the variation of thickness with tube radius \((r_i)\) and stress ratio \((\alpha)\).
Figure 4.14: Variation of thickness ($t_1$) with tube radius ($r_1$) and stress ratio ($\alpha$) for a simple bulged part. Obtained by using Equation (4-32).
Figure 4. 15: Variation of final wall thickness (t_f) with stress ratio (\( \alpha \)) for a simple bulged part. Obtained by using Equation (4-32).
CHAPTER 5

LIMITS OF TUBE HYDROFORMING PROCESS (FAILURE TYPES, INSTABILITY MODES)

As with any metal forming process, there are certain limitations on the degree of deformation achievable in tube hydroforming (THF) process such that parts with desired specifications (like expansions) can not be formed without any defects.

Depending on the defect type observed, limits of THF process (failure types or instability modes) can be grouped as:

- Buckling
- Wrinkling
- Bursting (Necking, Fracture)

Figure 5.1 presents buckling, wrinkling and bursting failures in THF. Instability modes, which limit the extent of formability in THF process, occur when stress and strain state in
Figure 5.1: Common failure modes that limit the THF process [Dohmann, 1995]
a part reach a critical level that equilibrium can not be sustained any longer between external forces applied and the internal resistance of the material (i.e. strength).

Basically, buckling occurs when axial compressive stress on an element of a part exceeds the strength of the material. Buckling in THF process is reported to take place during the initial stages of deformation when strain level is very small.

Wrinkling, on the other hand, is observed during both initial and intermediate stages of forming in the form of symmetric corrugations. Even though the cause of wrinkling is also excessive compressive loading, the way buckling and wrinkling take place are quite different depending on the geometrical configuration of the tubular component. Buckling is observed usually on long tubes with relatively thick walls whereas wrinkling is found on both long and short tubes with relatively thin walls.

Bursting is a consequence of necking, which is a condition of local instability under the influence of large tensile forces. Once necking starts, the deformation becomes non-uniform throughout the part as it is under buckling and wrinkling cases. Deformation is, then, concentrated in a local area causing necking proceed very rapidly towards bursting (or fracture).
In this chapter, first, buckling and wrinkling are studied in detail under the same section heading since they are closely interrelated to each other. Bursting will be topic of another sub-section.

5.1 Buckling and wrinkling

Before going into the details of analysis for buckling and wrinkling types of failure in THF process, it is necessary to indicate the following points:

- There are still many unknowns in understanding of the behavior of even simple mechanical components (such as columns) under buckling conditions [Southwell, etc]. Particularly, the deviations between results of theoretical and experimental investigations have raised numerous questions about the physics of buckling and wrinkling phenomena. Currently, design of mechanical component for buckling conditions is performed with relatively high safety factors based on empirical findings and assumptions such as effect of initial conditions, imperfections and experimental errors [Popov, 1990] [Triotiskiy, 1997].

- There are not any buckling or wrinkling studies that investigated the tubes under both axial compressive load and internal pressure in the literature, as far as author could reach. Most of the studies are about tubes under only axial compressive loading or under external pressure.
• Buckling analysis of tubular components was considered in terms of structural stability as design of structures with tubes was their common interest [Popov, 1990]. On the other hand, in tube hydroforming case, plastic buckling and wrinkling analysis is our subject.

• Furthermore, most of the available studies involved tubes with very large diameter-to-thickness (D/t) ratios since, assumingly, this kind of structures were used vastly in aircraft and oil refinery industry (i.e. during some stages of firing, the thin casings of rockets are heavily loaded in compression) [Popov, 1990].

Buckling of tubes can be analyzed in two ways as they occur in the physical world. They are namely:

• Buckling of tubes as a column (i.e. global buckling)
• Wrinkling of tube walls (i.e. local buckling)

Figure 5.2 illustrates buckling and wrinkling cases. After on in this report, buckling will be used to describe the global buckling, and wrinkling will be used to describe the local buckling. Buckling of tubes as a column is observed when the tube is long and with relatively thick walls. Wrinkling, however, occurs when the tube is either short or long but with thin walls. However, there is no definite boundary between buckling and wrinkling conditions since they are both dependent on combination of many factors such as material, edge conditions, geometry, imperfections and loading types [Almroth, 1975].
Figure 5.2: (a) Wrinkling (local buckling) is formation of symmetric corrugations of tube walls under axial compressive forces whereas (b) buckling (global buckling) is the instability situation of the tube as whole.
5.5.1 Analysis of buckling (global buckling)

As shown in Figure 5.2, buckling (global buckling) of tubes can be described as column buckling since a thin-walled tube under compressive loading fails by the instability of the tube as a whole. Hence, many authors suggested to apply column buckling analysis to that of tubes particularly for tubes with relatively small (D/t) ratios [Troitsky, 1982] [Thomsen, 1965]. Column buckling was studied first by Euler, and developed by others [Lorenz, 1908] [Timoshenko, 1910] [Southwell, 1913] [Von Karman, 1940] [Almroth, 1975] [Popov, 1990]. Analysis of buckling for column-like members is conducted under the following assumptions:

- Material behavior is linearly elastic,
- Member has the same cross-section throughout the length, and
- Loading is not eccentric, and it is uniform

The critical load at which elastic instability of a column occurs is defined as Euler load, and it is expressed as follows:

\[ F_{cr} = \frac{\pi^2 EI}{L^2} \quad (5-1) \]

Equation (5-1) indicates that the critical load \( F_{cr} \) that causes buckling of a column is function of elastic modulus of material \( E \), end conditions \( C \), moment inertia \( I \) and length \( L \) of the component. Longer columns are susceptible to fail under buckling easily. Note that for a column with both-ends-fixed condition, \( (C) \) is four \( (C = 4) \) [Popov, 1990].
Similarly, the critical buckling stress can be written as:

$$\sigma_{cr} = \frac{F_{cr}}{A} = C \frac{\pi^2 E}{(L/r_g)^2}$$  \hspace{1cm} (5-2)$$

According to the expression in Equation (5-2) the critical buckling dependent on elastic modulus of material ($E$), ends conditions ($C$), radius of gyration ($r_g$), cross-sectional area ($A$) and length ($L$) of the component.

Based on the assumption that a long tube with relatively thick walls can be analyzed as a column under buckling conditions, critical buckling stress, which is the lowest possible bifurcation stress causing the tube lose its straight configuration, for a tubular material can be expressed as follows:

$$\sigma_{cr} = C \frac{\pi^2}{2} \left( \frac{r^2}{L} \right)$$  \hspace{1cm} (5-3)$$

$$F_{cr} = C \pi^3 E \frac{r^3 t}{L^2}$$  \hspace{1cm} (5-4)$$

Note that for a tubular material:

$$r_g^2 = \frac{r^2}{2} \hspace{1cm} \text{Radius of gyration}$$

$$I = r^3 t \pi \hspace{1cm} \text{Moment of inertia}$$

$$A = 2rt \pi \hspace{1cm} \text{Cross-sectional area}$$

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As a result, Equations (5-3) and (5-4) define the critical buckling stress and loading magnitudes for a tubular material in the elastic range, respectively. Accordingly, critical buckling and loading values for a tube are a function of elastic modulus of material \((E)\), end conditions \((C)\), radius \((r)\), wall thickness \((t)\), and length \((L)\) of the tube. Similarly, longer tubes are susceptible to fail under buckling easily. Note that for a tube with both-ends-fixed condition, \((C)\) is four.

As expressed previously, however, under the circumstances of tube hydroforming process, our purpose is to investigate the variation of critical buckling stresses or forces in the plastic range. In order to facilitate this investigation, tangent modulus \((E_t)\) of a material is substituted instead of elastic modulus \((E)\) in Equations (5-3) and (5-4) as suggested by Shanley [Shanley, 1947], Thomsen [Thømsen, 1965] and others.

Tangent modulus of a material is defined as the slope of the flow stress curve in the plastic range:

\[
E_t = \frac{d\bar{\sigma}}{d\bar{\varepsilon}} = nK\bar{\varepsilon}^{n-1}
\]  

(5-5)

Assuming the flow stress is in the form of

\[
\bar{\sigma} = K\bar{\varepsilon}^n
\]
As a result, critical buckling stress and force values of a tubular material under plastic deformation can be expressed as:

$$\sigma_{cr} = \frac{2\pi^2 E_i r^2}{L^2}$$  \hspace{1cm} (5-6)

$$F_{cr} = 4\pi^3 E_i \frac{r^3}{L^2} t$$  \hspace{1cm} (5-7)

5.5.2 Analysis of wrinkling (local buckling)

Wrinkling (local buckling) is observed as symmetric corrugations on a tube wall under axial compressive loading as illustrated in Figure 5.2.

Timoshenko and others [Timoshenko, 1963] derived theoretical equations to predict the critical axial compressive stress causing a tubular member to undergo instability in form of wrinkles. In their analysis, perfect conditions were assumed. These are (a) homogenous and isotropic tubular material, (b) concentric and uniform loading and (c) flat, burr-free tube edges. Details of their analysis in elastic region will not be repeated here. As a result, the critical axial compressive stress that initiates wrinkles on a tube can be defined as follows:

$$\sigma_{cr} = \frac{1}{\sqrt{3(1-\nu^2)}} E \frac{t}{r}$$  \hspace{1cm} (5-8)
This equation indicates that critical axial compressive stress value for wrinkling case is dependent on elastic modulus of the tubular material \(E\), poisson’s ratio \(v\), radius \(r\) and thickness \(t\) of the tube. As it can be noticed, the model does not show any effect of tube length but wall thickness and radius on the wrinkling instability, which is as expected in terms of simple physical consideration. For steel tubular materials, the substitution of a poissons’ ratio of 0.3 would simplify the equation into the following:

\[
\sigma_{cr} = \frac{2}{3} E \frac{t}{r}
\]  

(5-8)

However, as mentioned previously, there are deviations between theoretical and experimental findings as reported by many authors [Triotsky, 1997] [Von Karman, 1941]. Experimental results were found to be around one to two thirds of the theoretical values. The deviations have been explained to be due to the imperfections in the real life such as inhomogeneous and anisotropic materials, non-uniform and eccentric loading, and imperfect edge conditions of tubes used in the tests.

Similar to the buckling case, in order to take the plastic deformation into account, tangent modulus \((E_t)\) of a tubular material can be used instead of elastic modulus \((E)\) in Equations (5-8) and (5-9). Then, critical axial compressive stress and force for wrinkling case will be:
\[ \sigma_{cr} = \frac{2}{3}(E_r)^t_r \]  
\[ F_{cr} = \frac{4\pi}{3}(E_r)^2 \]  
\[ (5-10) \]  
\[ (5-11) \]

where the tangent modulus is as follows:

\[ E_t = \frac{d\sigma}{d\varepsilon} = nK\varepsilon^{n-1} \]

Assuming the flow stress is in the form of

\[ \bar{\sigma} = K\varepsilon^n \]

5.2 Bursting (necking, fracture)

Necking in sheet metal forming is a result of tensile forces. It is an instability situation that causes fracture eventually. Thus, prediction of start of necking is an important issue in metal forming processes before designing the details of processes. Even though there are many different criteria to determine the fracture in metal forming processes, their extensive validation with experiments and real life has not been proved solidly yet. Furthermore, many of these criteria require tedious calculations and work. Therefore, within the industrial community simple but useful criterion like thinning are in use. Nevertheless, in this section one of the criterion, i.e. effective strain at necking point, for fracture (necking) instability is presented. In case of tubular parts being deformed under internal pressure \( P_i \) and compressive axial load \( F_a \), state of stress and strain needs to
be defined for achieving the required expansion before necking begins. This section concentrates on defining such stress or strain range where required expansion is obtained while keeping the tubular part away from buckling and wrinkling due to compressive axial loads when the tube is not in contact with the die wall yet.

Necking in the tangential direction can be defined by relating it with the tangential force per unit length of the tube thickness. The tensile instability in the hoop direction is given as:

\[
\frac{dF_\theta}{dt} = 0 \quad (5-12)
\]

Tangential force \((F_\theta)\) can be expressed as follows

\[
F_\theta = t\sigma_\theta \quad (5-13)
\]

Proceeding with Equation (5-12) after substitution of Equation (5-13):

\[
\frac{d(t\sigma_\theta)}{dt} = 0
\]

\[
t \frac{d\sigma_\theta}{dt} + \sigma_\theta = 0
\]

\[
\frac{dt}{t} = -\frac{d\sigma_\theta}{\sigma_\theta} \quad (5-14)
\]

where

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\[
\frac{dt}{t} = \varepsilon_r = \varepsilon_i = \ln \frac{t}{t_o}
\]

(5-15)

Thickness strain can be expressed as:

\[
d\varepsilon_r = \frac{d\bar{\varepsilon}}{2\bar{\sigma}} \left[ -\frac{1}{2} (\sigma_r + \sigma_\theta) \right] = -\frac{d\bar{\varepsilon}}{2\bar{\sigma}} (1 + \alpha)\sigma_\theta
\]

Then, from Levy-Mises equations, thickness strain will be:

\[
d\varepsilon_r = -\frac{(1 + \alpha)}{2\sqrt{1-\alpha + \alpha^2}} d\bar{\varepsilon}
\]

(5-16)

Substituting Equation (5-16) into Equation (5-14), we will obtain:

\[
\frac{d\sigma_\theta}{\sigma_\theta} = \frac{(1 + \alpha)}{2\sqrt{1-\alpha + \alpha^2}} d\bar{\varepsilon}
\]

(5-17)

Since, hoop stress can be expressed as:

\[
\sigma_\theta = \frac{\bar{\sigma}}{\sqrt{1-\alpha + \alpha^2}}
\]

By manipulating, and introducing other known relations, it will become:

\[
\frac{d\bar{\sigma}}{d\bar{\varepsilon}} = \frac{(1 + \alpha)}{2\sqrt{1-\alpha + \alpha^2}} \bar{\sigma}
\]

(5-18)

Flow stress of material can be modeled by either

\[
\bar{\sigma} = K\bar{\varepsilon}^n
\]

(5-19)
or

\[ \bar{\sigma} = K (\varepsilon_o + \bar{\varepsilon})^n \]  \hspace{1cm} (5-20)

Then, derivative of Equation (5-19) would give:

\[ \frac{d\bar{\sigma}}{d\bar{\varepsilon}} = \frac{d(K\bar{\varepsilon}^n)}{d\bar{\varepsilon}} \]

\[ \frac{d\bar{\sigma}}{d\bar{\varepsilon}} = \frac{nK\bar{\varepsilon}^{n-1}}{\bar{\varepsilon}} \]  \hspace{1cm} (5-21)

Substituting Equations (5-21) and (5-19) into Equation (5-18):

\[ \frac{nK\bar{\varepsilon}^n}{\bar{\varepsilon}} = \frac{(1 + \alpha)}{2\sqrt{1 - \alpha + \alpha^2}} K\bar{\varepsilon}^n \]

After cancellations, effective strain at instability condition can be defined as follows:

\[ \bar{\varepsilon} = \frac{2n\sqrt{1 - \alpha + \alpha^2}}{(1 + \alpha)} \]  \hspace{1cm} (5-22)

In case, Equation (5-20) is used instead of Equation (5-19), we would obtain:

\[ \bar{\varepsilon} = \frac{2n\sqrt{1 - \alpha + \alpha^2}}{(1 + \alpha)} - \varepsilon_o \]  \hspace{1cm} (5-23)

As a result, the effective strain that a tabular part could sustain before necking is found to be dependent on (a) pre-straining in the material due to previous cold working, (b) strain-
hardening exponent of that material, and (c) stress ratio ($\alpha$) at necking instant. Figure 5.3 shows variation of effective strain at instability with stress ratio.

When pre-straining is zero due to annealing, for instance, infinitely large effective strain will be obtained without instability (i.e. necking) when stress ratio ($\alpha$) is ($-1$). This conclusion agrees very well with the fact that thinning in the wall thickness of a tube will be zero for a stress ratio ($\alpha$) of ($-1$) as presented below:
Figure 5.3: Variation of effective stress at instability point with stress ratio. This curve is obtained using Equation (5.23) for a simple bulged part. The flow stress is assumed to be in the form of $\sigma_f = K(\epsilon_o + \overline{\epsilon})^n$. 

**Stainless Steel 304 Tubular Material**

\[ K = 1400 \text{ Mpa} \quad n = 0.606 \quad \epsilon_o = 0.06 \]

\[ R_o = 28.575 \text{ mm} \quad r_o = 27.75 \text{ mm} \quad t_o = 5.65 \text{ mm} \]
When stress ratio \((\alpha)\),

\[
\alpha = \frac{\sigma_2}{\sigma_\theta} = \frac{\sigma_2}{\sigma_1} = -1
\]

Strain ratio will be:

\[
\beta = \frac{\varepsilon_2}{\varepsilon_\theta} = \frac{\varepsilon_2}{\varepsilon_1} = -1
\]

Since

\[
\beta = \frac{2\alpha - 1}{2 - \alpha}
\]

In order to utilize necking criterion (i.e. Equations 5-22 and 5-23) in practical applications, let us try to explain it in terms of normal strains \((\varepsilon_z)\), \((\varepsilon_\theta)\), and \((\varepsilon_r)\):

Substituting Equation (5-22) into Equation (5-16), assuming that pre-strain is zero, we obtain that maximum thickness strain (i.e. at necking condition) will be equal to strain-hardening exponent of the material:

\[
\varepsilon_r = -n \quad \text{(5-24)}
\]

Since,

\[
\varepsilon_z + \varepsilon_\theta + \varepsilon_r = 0
\]

It is found that sum of axial and hoop strains will be \((n)\) at necking point:

\[
\varepsilon_z + \varepsilon_\theta = n \quad \text{(5-25)}
\]
As a result of these findings, it can be concluded that large tensile hoop strain can be achieved as long as sufficient compressive axial strain is supplied to the expansion region while keeping the total strain below \((n)\) value. When the sum of \((\varepsilon_\varphi)\) and \((\varepsilon_z)\) reaches \((n)\) at any area on the tube, thickness strain will be \((-n)\) which causes necking at that local area.

5.3 Development of TH-DESIGN based on analytical models and limiting criteria in THF process

A simple computer program (TH-DESIGN) is developed using the analytical models in the previous chapters and limiting criteria just presented. Given the final (desired) expansion diameter, approximate thinning, final length, and initial diameter, thickness and material information, the program first calculates the initial length from the volume constancy approximately. Then, assuming a linear expansion and linear thinning profile, pressure and axial force variations are calculated. Minimum and maximum pressure versus force diagrams are drawn to obtain a window of operation for such a simple axisymmetric part. Figure 5.4 illustrates the algorithm used in the program.

Even though this computer program would not give exact results for every part available in the market, it may be used to analyze the crucial sections of complex parts under certain approximations for quick and initial estimations. Results can be used for further and detailed analysis via FEM. Hence, trial-and-error during FEA could be reduced in order to tune in the correct loading profile for a given part shape, dimensions and material.
Given the Final expansion diameter (D1), Thickness (t1), Thinning expected (% TH), Length (Lpe), Corner radius (Rc), Initial diameter (Do), Ultimate tensile (Su) and Yield strength (Sy), Strength coefficient (K), Strain hardening exponent (n)

Calculate the Initial Diameter (Do), Thickness (t0) and Length (L0) from volume constancy, approximately

Assuming a linear expansion and linear thinning (or thickening), divide the process into A steps

For each step, calculate the following: Expansion diameter (Di), Thickness (ti), Strains, Effective stress and strains, Tangent modulus

Then, for each step calculate:
- Internal pressure (Pi)
- Buckling force limit (Fb)
- Wrinkling force limit (Fw)
- Sealing force limit (Fp)

Draw the limits onto a Pressure versus Force Diagram, indicating expansion and calibration phases

Figure 5.4: Simple algorithm used in the computer program (TH-DESIGN) to determine the limits of process for an axisymmetric part
Figures 5.5 through 5.8 present a case study where experimental information found in the literature [Boehm, 1997] are compared with the results predicted by the program (TH-DESIGN). As it is seen from these figures, the followings can be concluded:

- Pressure versus force profile used in the experiments (a) can be predicted within a reasonable accuracy (b), Figure 5.5. The deviations may be due to the material information used in the calculations since they are properties of sheet material not tubular material. Furthermore, it may be due to the modeling of the friction in the analytical models since only a single coulomb friction coefficient is used whereas in the actual process friction coefficient changes with time and location,

- In the experiments, an axisymmetric bulging is obtained using the pressure versus force profile shown as (a) in Figure 5.6. Prior to the end of the bulging and formation of corners, authors of this work indicated that wrinkles occurred as illustrated (m). When compared to the limiting “buckling” and “wringling” curves (d and e), it is obvious that with the pressure versus force curve such wrinkles would be predicted, Figure 5.6.

- Another pressure versus force curve (c), which is lower than the previous one, was used in their work, Figure 5.7. With this one, they obtained wrinkles of smaller height (p) as it is closer to the limits drawn by the program (d and e).

- A suggested pressure versus force curve (g) is shown in Figure 5.8 to avoid buckling or wrinkling

- Pressure values at the end of expansion and calibration phases could be predicted within a reasonable accuracy when compared to the experiments.
Figure 5.5: Comparison of experimental and predicted axial force profiles used in deformation of an axisymmetric bulge part with the dimensions shown above.
Figure 5.6: Comparison of experimental and predicted force curves with the predicted limiting curves. As seen from this figure, since the applied force in the experiments is higher than the limits wrinkling is obtained.
Figure 5.7: Comparison of experimental and predicted force curves with the predicted limiting curves. Since the applied force is higher than the limiting force curves wrinkling occurs even though it is with small corrugations.
Figure 5.8: Illustration of the suggested pressure versus force curve to avoid wrinkling during deformation of an axisymmetric bulge part.
CHAPTER 6

DETERMINATION OF FLOW STRESS CURVE FOR TUBULAR MATERIALS

In metal forming processes, selection of material, determination of process boundaries, and proper design of process parameters require a comprehensive knowledge about the characteristics of material. For tube hydroforming, the initial raw material is in tubular form. Some process designers use tensile material properties of the flat sheet from which the tube is rolled. These properties are primarily constructed from tensile testing of the sheet material, prior to rolling. However, properties of sheet material from which tubes are rolled will not reflect the actual characteristics of the tubular material. The reason is that sheet material goes through a series of roll forming and sizing operations, which will alter the mechanical properties until it becomes into a tubular shape. These operations are mainly rolling, welding and sometimes drawing processes.

Another problem with the current material definition for tubes is that the testing methods do not accurately reflect the state of deformation encountered in tube hydroforming. Since internal pressure is used to form the tubes, the material is stretched in a bi-axial
fashion. The standard test methods are generally based on uni-axial stretching. In addition, to reduce process development time, it is often necessary to conduct computer simulations. In order to ensure accurate simulation results, the developer must know the specific material properties of the tube such as yield strength, ultimate tensile strength, flow stress, percent elongation, etc. Currently, almost no mechanical data exists for the tubular materials commonly used in tube hydroforming.

Hence, a testing method, which would define the stress-strain behavior of tubular materials, based on bi-axial stretching, is necessary for improved accuracy of process simulation and control. In this section, a simple analytical method to determine the flow stress curve of tubes, both under internal pressure only case, and internal pressure plus axial force case, will be explained briefly.

6.1 Tooling

In order to use this method, a basic tooling that can apply bi-axial stress to tubes is required. Such a tooling and its descriptions were already discussed in previous ERC reports [Sokołowski, et al., 1998]. Figure 6.1 illustrates this tooling. Figure 6.2 presents tubes deformed under (a) pressure only, and (b) pressure and axial force cases. With an accurate testing method, designers will be able to better develop and select materials for tube hydroformed components, process designers will be able to more accurately (and timely) create forming sequences for the manufacture of these parts, and the manufacturing facilities could reduce the number of defective parts.
Figure 6.1: Tooling used for hydraulic bulge testing of tubular materials. Tooling is shown without hydraulic system.
Figure 6.2: Tube being bulged by (a) internal pressure and axial load, (b) only internal pressure. (c) is the nomenclature and state of stress at the tip of bulge region. Notice that height of bulge in case (a) is greater than in case (b), \( (h_a) > (h_b) \).
During the component development phase, designers must select materials, which will optimally fit the component requirements. This is not an easy task, since there are many different materials available with no published data on their formability characteristics. Unless the designer can rely on a knowledge base of how these materials will react under hydroforming conditions, the selection of the material is tough and often an iterative procedure is required to determine the best material for a certain application. A concise database of material formability measures would simplify the selection of the proper material. These measures may include expansion characteristics, consistency of strength characteristics of the tubing, bursting pressures, etc.

Due to the nature of the hydroforming process, quality control of the tubular raw material is essential. The formability of the tubular material will depend on the material composition, rolling process, state of heat treatment, etc. Very small differences in the tube consistency may cause a run of defective parts, due to bursting or buckling. This can occur due to inconsistent eccentricity of the tubing, thin spots, poor weld quality, poor material processing, etc.

A simple and easy to use method for testing the quality of incoming tubing would reduce the number of scrap parts due to these problems. A testing rig as shown in Figure 6.1, on the plant floor, would allow random samples from each batch of tubing to be tested for consistency of quality and strength characteristics.
6.2 Methods

Here, three different methods are presented to obtain flow stress characteristics of tubular materials. The followings (some or all) need to be measured in these methods successively at different stages (j steps) during bulging of tubes in order to use them in calculations:

(a) Radius of curvature in hoop direction \((r_\theta)\),

(b) Radius of curvature in longitudinal direction \((r_z)\),

(c) Internal pressure \((P_i)\),

(d) Axial load \((F_a)\), and

(e) Thickness \((t)\).

6.2.1 Method A: All parameters are measured on line

This kind of a method requires sophisticated measurement equipment and techniques such as laser or ultrasonic, non-touching sensor, etc. Even though it is tedious and expensive to build such a tooling, measurement, control and data acquisition system, this method will be presented to reflect the perfect circumstances to obtain flow stress characteristics of tubular materials. Once the parameters listed above are measured for different stages of bulging for the same tubular material, finite strains \((\varepsilon_\theta)\), \((\varepsilon_z)\), and \((\varepsilon_t)\) can be calculated with the following equations:

\[
(\varepsilon_\theta)_j = \ln \left( \frac{r_\theta}{r_o} \right)_j
\]  

\[(6-1a)\]
\[
(e_z)_j = \ln \left( \frac{t_1}{t_o} \right) 
\]

\[
(e_r)_j = \ln \left( \frac{r_1}{t_o} \right) 
\]

(6-1b)

(6-1c)

Then, effective strain can be obtained easily using hoop, longitudinal and radial strains.

Using membrane Equations (C-8) and (C-9) in Appendix C, stresses in hoop and longitudinal directions can be calculated as follows:

\[
(\sigma_\theta)_j = \left[ \frac{(P_i)_j (r_\theta)_j}{(t_1)_j} - \frac{(\sigma_z)_j (r_\theta)_j}{(r_z)_j} \right] 
\]

(6-2a)

or simply as

\[
(\sigma_\theta)_j = \frac{(P_i)_j (r_\theta)_j}{(t_1)_j} 
\]

(6-2b)

\[
(\sigma_z)_j = \frac{(P_i)_j (r_\theta)_j}{2(t_1)_j} - \frac{(F_a)_j}{2\pi(t_1)_j (r_\theta)_j} 
\]

(6-3)

Once directional stresses are calculated, effective stress values for successive stages can be found from Equation (B-26), as:

\[
(\bar{\sigma})_j = \sqrt{(\sigma_\theta)_j^2 - (\sigma_\theta)_j (\sigma_z)_j + (\sigma_z)_j^2} 
\]

(6-4)
Eventually, successive effective stress ($\bar{\sigma}$) and effective strain ($\bar{\varepsilon}$) values can be fitted into the general flow stress equation as follows using the least square method:

$$\bar{\sigma} = K (\varepsilon_o + \bar{\varepsilon})^n$$

Where ($\varepsilon_o$) is found by equating ($KE_o^n$) to the Yield Strength of the material, ($\sigma_{yp}$). In the Equation (6-3), force component can be simply dropped when ends of the tubes are fixed as it is in the experiments conducted at ERC/NSM of the Ohio State University [Sokolowski, 1998], [Arnold, 1999] and [Aueulan, 1999].

**6.2.2 Method B: All parameters are measured off-line and ($r_z$) is calculated**

Alternatively, a number of tubes of the same material and initial conditions can be bulged to certain stages when “on-line” measurement of thickness ($t_1$), is ($r_o$) is not possible. Thus, these parameters can be measured after the tubes are bulged to known diameters or pressures, and taken out of the apparatus. In addition, longitudinal radius ($r_z$) is difficult to measure even in “off-line” case. Therefore, it can be calculated as shown in Figure 6.3 and as follows:

$$r_o = r_o + h$$

$$r_z = \frac{\left(\frac{w}{2}\right)^2 + h^2}{2h}$$

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This is the method currently applied at ERC/NSM's experiments where tubes are bulged to burst to determine the maximum pressure, first. Then, different pressure levels are applied to expand the tubes. At each expansion level, tubes are taken out of the tooling. Expansion diameter and thickness are measured. Eventually, as presented through Equations (6-1) through (6-4), diameter and thickness information is converted into stress and strain values which are later fit into a flow stress equation, or simply plotted [Aueulan, 1999]. Figure 6.4 illustrates flow stress curve of SS 304 tubular material obtained as explained above. Figure 6.5 shows the deformed SS 304 tubes at various pressure levels.

6.2.3 Method C: All parameters are measured except \((r_z)\) and \((t)\) which are calculated

As another alternative technique, \((t_i)\) can be derived analytically as well as \((r_z)\) without any lengthy measurement efforts. It is as follows:

From Equation (B-21), it is derived that:

\[
\frac{\sigma_z}{\sigma_\theta} = \frac{\varepsilon_\theta + 2\varepsilon_r}{\varepsilon_r - \varepsilon_\theta} \tag{6-5}
\]

Then,

\[
\frac{(\varepsilon_\theta)_j + 2(\varepsilon_r)_j}{(\varepsilon_r)_j - (\varepsilon_\theta)_j} = \frac{(\sigma_z)_j}{(\sigma_\theta)_j} = \frac{\pi (P_i)_j (r_\theta)_j^2 - (F_a)_j}{2\pi (r_\theta)_j^2 (P_i)_j} \tag{6-6}
\]
Figure 6.3: Configuration of a bulging tube under internal pressure

Figure 6.4: Flow stress curve of SS 304 tubular material obtained in hydraulic bulge test

\[ \bar{\sigma} = K (\varepsilon_0 + \bar{\varepsilon})^n \]

K = 1400 MPa
n = 0.688
\varepsilon_0 = 0.06
Figure 6.5: Deformed SS 304 tubes at various pressure levels

<table>
<thead>
<tr>
<th>Dimension of the initial tube and die geometry (Figure 6.3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OD</td>
</tr>
<tr>
<td>2.25 in (57.15 mm)</td>
</tr>
</tbody>
</table>
Hence, knowing \((P_i)\), \((r_o)\), and \((F_a)\), \((\varepsilon_r)\) can be calculated from the equation above. In turn, \((t_1)\) will be obtained easily using Equation (6-1) as follows:

\[ t_1 = e^{At_0} \]

where

\[
A = \ln \left[ \frac{r_0}{r_o} \right] \left[ \frac{3}{2} - \frac{r_0}{2r_z} \right] \]

\[ \frac{3}{2} - \frac{r_0}{2r_z} \]

4.3 Verification of method C

Thickness predictions for bulging of different materials were performed using the method C outlined above. These materials include Low Carbon Steel 1008, SS 304 and aluminum alloy 6260-T6. These predictions are compared with measurements of thickness from the experiments in Figures 6.6, 6.7, and 6.8, respectively.

Under various internal pressures, tubes with various materials and bulge lengths were formed. Prediction of thickness is within 5% of the measurement in all cases. Figure 6.9 illustrates the comparison of the flow stress curves obtained using measured and predicted thickness values for SS 304 tubular material. As it seen, flow stress curves are quite close to each other suggesting that predicted thickness values are reasonably accurate to use in determining flow stress curves.
### Low Carbon Steel 1008

<table>
<thead>
<tr>
<th>Pressure (Mpa / psi)</th>
<th>Bulge Height (mm / in)</th>
<th>Thickness (Experiment) (mm / in)</th>
<th>Thickness (Prediction) (mm / in)</th>
<th>Thickness Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>23.7 / 3436.5</td>
<td>1.32 / 0.052</td>
<td>1.55 / 0.061</td>
<td>1.57 / 0.062</td>
<td>-1.552</td>
</tr>
<tr>
<td>30.0 / 4350.0</td>
<td>2.39 / 0.094</td>
<td>1.50 / 0.059</td>
<td>1.51 / 0.059</td>
<td>-0.7905</td>
</tr>
<tr>
<td>31.6 / 4582.0</td>
<td>5.46 / 0.215</td>
<td>1.35 / 0.053</td>
<td>1.31 / 0.052</td>
<td>2.4469</td>
</tr>
</tbody>
</table>

---

**Graph:**

- **Experimental results**
- **Prediction results**

---

### Dimension of the initial tube and die geometry (Figure 6.3)

<table>
<thead>
<tr>
<th>OD</th>
<th>$L_0$</th>
<th>$t_0$</th>
<th>$w$</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.25 in (57.15 mm)</td>
<td>8 in (203.2 mm)</td>
<td>0.065 in (1.65 mm)</td>
<td>2 in (50.4 mm)</td>
</tr>
</tbody>
</table>

---

Figure 6.6: Comparison of thickness measurements in the middle of the bulge with predictions for LCS 1008 tubular material.
### Stainless Steel 304 tubes

<table>
<thead>
<tr>
<th>Pressure (MPa)</th>
<th>Expansion radius (mm)</th>
<th>Thickness (Experiment) (mm)</th>
<th>Thickness (Prediction) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>11.93 / 1700</td>
<td>6.73 / 0.186</td>
<td>0.486 / 0.019</td>
<td>0.50</td>
</tr>
<tr>
<td>13.13 / 1850</td>
<td>6.16 / 0.243</td>
<td>0.448 / 0.018</td>
<td>0.474</td>
</tr>
<tr>
<td>16.27 / 2100</td>
<td>7.83 / 0.308</td>
<td>0.427 / 0.017</td>
<td>0.441</td>
</tr>
<tr>
<td>16.55 / 2400</td>
<td>10.97 / 0.432</td>
<td>0.379 / 0.015</td>
<td>0.383</td>
</tr>
<tr>
<td>18.30 / 2650</td>
<td>15.31 / 0.603</td>
<td>0.320 / 0.013</td>
<td>0.314</td>
</tr>
</tbody>
</table>

![Graph showing thickness measurements and predictions](image)

**Dimension of the initial tube and die geometry (Figure 6.3)**

<table>
<thead>
<tr>
<th>OD</th>
<th>L₀</th>
<th>t₀</th>
<th>w</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.25 in (57.15 mm)</td>
<td>8 in (203.2 mm)</td>
<td>0.024 in (0.6 mm)</td>
<td>2 in (50.4 mm)</td>
</tr>
</tbody>
</table>

Figure 6.7: Comparison of thickness measurements in the middle of the bulge with predictions for SS 304 tubular material.
Aluminum Alloy 6260-T4

<table>
<thead>
<tr>
<th>Pressure (Mpa / psi)</th>
<th>Bulge Height (mm / in)</th>
<th>Thickness (Experiment) (mm / in)</th>
<th>Thickness (Prediction) (mm / in)</th>
<th>Thickness Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>13.74 / 1992.3</td>
<td>2.22 / 0.087</td>
<td>1.96 / 0.077</td>
<td>1.91 / 0.075</td>
<td>2.35</td>
</tr>
<tr>
<td>16.25 / 2066.2</td>
<td>2.98 / 0.117</td>
<td>1.92 / 0.076</td>
<td>1.86 / 0.073</td>
<td>3.33</td>
</tr>
<tr>
<td>15.04 / 2181.0</td>
<td>3.69 / 0.145</td>
<td>1.83 / 0.072</td>
<td>1.81 / 0.071</td>
<td>1.09</td>
</tr>
</tbody>
</table>

![Graph](image)

Experimental results = ○ ○ ○ = Prediction results

<table>
<thead>
<tr>
<th>Dimension of the initial tube and die geometry (Figure 6.3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OD</td>
</tr>
<tr>
<td>-----</td>
</tr>
<tr>
<td>2.375 in (60.325 mm)</td>
</tr>
</tbody>
</table>

Figure 6. 8: Comparison of thickness measurements in the middle of the bulge with predictions for Al6260-T4 tubular material.
Figure 6.9: Comparison of flow stress curves for SS 304 tubular material obtained using measured and predicted thickness values.
On the other hand, this way may cause deviations in the final result due to the assumptions involved in calculations. Ends of the tubes can be fixed instead of applying an axial load \( (F_a) \). This would drop a parameter to measure from experimental task. Rest of the procedure will be the same except dropping \( (F_a) \) term from necessary equations. However, bulging without axial loading will limit the size of the bulge that can be obtained. Thus, only a small range of effective strains will be obtained. For large strain ranges, axial force needs to be used.

In order to increase the accuracy of the determination, tubes of the same material with different initial conditions can be bulged and measured. Averaging the results would give more versatile flow stress curves. Also, radius of curvature and thickness measurements should be performed on various but consistent points distributed longitudinally and tangentially, for each stage of deformation and for each tube. This procedure would reduce the variation in measurements, and will reflect the overall condition of the bulged tubes. Furthermore, measurements on consistent and various locations will prepare a basis to investigate the effect of welding on tubular parts. This can be subject of another study. As a drawback of this technique to determine the flow stress characteristics of tubular materials, it is worth to mention that friction losses are neglected during calculation of longitudinal stress. Proper and consistent use of lubricants during experiments, and well maintenance of die surfaces would partially compensate this drawback.
CHAPTER 7

APPLICATION OF FINITE ELEMENT ANALYSIS (FEA) IN TUBE HYDROFORMING (THF) - CASE STUDIES

7.1 Introduction

As for other metal forming processes, in tube hydroforming technology, the development of a new part, tooling and process design requires expensive try-outs to establish the process parameters to be used, such as die and initial tube dimensions, clamping, axial, and counter forces, internal pressure, lubrication, etc. This development effort traditionally relies greatly on intuition, experience, and rules of thumb before try-outs in conventional metal forming processes such as stamping and forging. As the part complexity increases, experience and design rules may help only partially. As for tube hydroforming case, even this experience is not available. Hence, in order to reduce the development time, effort and cost, other tools such as Computer Aided Engineering (CAE) have to be used.
Finite Element Analysis (FEA) has been proven to be a useful CAE tool in the past for conventional metal forming processes to obtain more and precise information about the process, geometry and material parameters [Thomas, et al., 1999]. Feasibility of forming for a specific part can be predicted by analyzing thinning, thickening, strain-stress distribution on a deformed tube. Effect of different parameters can be investigated by varying important dimensions or loading conditions on common part types. Hence generic rules can be established for future analysis. Optimized loading conditions for any part can be deduced by using controlled analysis techniques.

In order to have accurate results from FEA, modeling and input of the following factors should be performed correctly and carefully:

- Material: flow stress curve and anisotropy
- Contact descriptions: lubricant, coating and friction coefficient
- Actions of process elements: internal pressure, axial cylinders, and their coordination
- Failure criteria: buckling, wrinkling and bursting
- Prior history of the material: bending, crushing and annealing
- Behavior of material after forming: residual stresses and spring-back

The following results should be available and feasible with the FEA code used:

- Thickness distribution
- Strain and stress distribution
- Deflections and movements, etc.
• Final geometry of the part, tooling, etc.

• History of the loading conditions

In this chapter, various case studies of FEA for different parts will be presented. The aim of these case studies was to (a) verify the accuracy and use of different FEA codes by comparing with available experimental data, (b) develop a consistent methodology for conducting FEA in tube hydroforming case, and (c) provide sponsors of this research with comprehensive guidelines for use of FEA.

As it will be seen in the next chapter, upon completion of the objectives of this chapter, generic guidelines for THF are developed via use of FEA and improved “design of simulation methods”.

7.2 Implicit and explicit codes

Even though many FEA software programs were tried, two of them were heavily used in the case studies and planned simulations. An implicit, two-dimensional, and static code, DEFORM 2D was used for simple, axisymmetric and plane strain analysis. For three-dimensional, complex parts, PAM-STAMP, an explicit and dynamic code, was used. Before going into the details of the case studies, author would like give some brief but useful background information about differences, advantages, disadvantages, and uses of “implicit” and “explicit” codes.
Theoretically, both techniques can be used for any metal forming application whether they are two- or three-dimensional. However, their effectiveness should be taken into account for practical use. Effectiveness can be combination different considerations such as:

- Accuracy and reliability,
- Computational time and cost, and
- Robustness

The mechanics of the metal forming process are very complex and involve large plastic deformations, and large relative movement between work-piece and die surface. This introduces non-linearity to the model. Along with this, there are other factors that bring non-linearity such as geometrical, material, and contact non-linearities. Due to this non-linear nature of metal forming processes, an iteration procedure is used to reach equilibrium in each state of quasistatic forming. Depending on the iteration procedure, iteration requires formation and solution of linear equation systems. When the size of the models increases, the system of equation can become very large and the solution may take long CPU times. Sometimes, due to the large stiffness differences between membrane and bending terms (i.e. particularly in sheet metal forming simulations), the resulting stiffness matrix of shell elements can become “ill-conditioned”, which introduces another complication upon solution of large equation systems.
In order to solve such large system of equations, iterative solution techniques are implied at each time incrementation. Whatever the incrementation and iteration techniques are a successful solution demands satisfaction of convergence criteria at each time increment.

Generally, the speed of convergence is one of the important problems in implicit technique, and it is usually case dependent. Many FEM codes are protected against non-convergence cases by ending the solution after a certain number of iterations.

Implicit procedure uses either an automatic incrementation technique based on a procedure monitoring the speed of convergence or can be defined by the user before beginning the analysis. Appropriate time increments are very important since they affect the convergence of the problem. Implicit method solves the following incremental equation for the equilibrium at the time \((t + \Delta t)\):

\[
K^T (u^{t+1}) \delta u^i = F - I(u^{t+1})
\]  

(7-1)

\[
\Delta u^i = \Delta u^{i-1} + \delta u^i
\]

where

\[
K^T = \text{Current tangent stiffness matrix of the deformation system}
\]

\[
\Delta u = \text{Incremental displacement vector}
\]

\[
F = \text{Applied external force vector}
\]

\[
I = \text{Internal force vector}
\]
The implicit solution techniques have a number of inherent disadvantages. Due to the nature of this technique, disadvantages become more pronounced in three-dimensional forming cases. These disadvantages can be listed as follows:

- Generally, takes longer solution times for three-dimensional forming cases
- Even it is difficult to predict how long a forming problem would take to be completed
- In addition, it can not be guaranteed that the solution would always converge
- Additional time is required to handle small time increments when large number of contact points are in the model: Stable contact algorithms to assure the “no-penetration” condition between work-piece and tool surface requires an additional iteration loop. Consequently, time increments smaller than what mechanics of the problem would allow becomes necessary. Thus, for large model with many contact points additional time is required for the extra iteration loop mentioned.
- Hence, an optimization between (a) small time increment requirement due to contact iteration, (b) overall time to complete the simulation, and (c) optimum time increment to achieve convergence has to be satisfied
- Particularly in sheet metal forming cases, local instabilities occur on the workpiece close to the tooling surface when compressive stresses are very high. Although this kind of stresses produces wrinkles in the reality, with an implicit and direct solver based FEA can cause failure in the analysis (i.e. non-convergence)
As mentioned before, PAM-STAMP was used for three-dimensional problems throughout this study. It is an explicit dynamic analysis program, mainly developed for the analysis of sheet metal forming processes. The most common explicit time integration operator used in nonlinear dynamic analysis is probably the central difference operator. As in linear analysis, the equilibrium of the finite element system is considered at time \( t \) in order to calculate the displacements at time \( t + \Delta t \). If the effect of a damping matrix is neglected, each discrete time step solution can be obtained with the equation.

\[
M \cdot \ddot{U} = I - F \tag{7-2}
\]

Where

\[
M = \text{Mass matrix;}
\]
\[
\ddot{U} = \text{Acceleration matrix;}
\]
\[
I = \text{Internal force vector;}
\]
\[
F = \text{External force vector;}
\]

The solution for the nodal point displacements at time \( t + \Delta t \) is obtained using the central difference approximation for the accelerations to substitute for \( \ddot{U} \). The solution, therefore, simply corresponds to a forward marching in time.

The main advantage of the method is that with \( M \) a diagonal matrix the solution of
$(^{t} + \Delta U)$ does not involve a triangular factorization of a coefficient matrix. The shortcoming in the use of the central difference method lies in the severe time increment restriction: for stability, the time increment size ($\Delta t$) must be smaller than a critical time step ($\Delta t_{cr}$) which equals to the smallest period in the finite element system.

The simulation time ($t_s$) consist of the number of increment ($I_i$) and the time increment ($\Delta t$):

$$t_s = \sum I_i \times \Delta t_i$$  \hspace{1cm} (7-3)

The simulation time ($t_s$) depends on the velocity curve and the maximum velocity. The time increment ($\Delta t$) depends on the ($\Delta t_{cr}$) as mentioned above and on the speed of the available workstation. The number of increment depends essentially on the knowledge of the programming structure. An increment, ($I_i$) signifies one step in the solution process and is determined by the total simulation time ($t_s$) and the size of the time step, ($T$):

$$I_i = \frac{t_s}{T}$$  \hspace{1cm} (7-4)

The time step is the temporal length of an increment.
\[ T = L^2 \times \sqrt{\frac{\rho}{E}} \]  \hspace{1cm} (7-5)

As shown in the above formula, the time step is a function of the following:

\[ \begin{align*}
L &= \text{Size of the smallest element} \\
E &= \text{Young's modulus} \\
\rho &= \text{Mass density}
\end{align*} \]

This time step restriction was derived considering a linear system, but the result is also applicable to non-linear analysis, since for each time step the nonlinear response calculation may be thought of, in an approximate way, as a linear analysis. However, whereas in a linear analysis the stiffness properties remain constant, in a non-linear analysis these properties change during the response calculations. These changes in the material and/or geometric conditions enter in the evaluation of the force vector \((\mathbf{F})\). Therefore, the value of \((T)\) is not constant during the response calculation, the time increment \((\Delta t)\) needs to be decreased if the system stiffens, and this time step adjustment must be performed in a conservative manner, so that with certainty the condition \((\Delta t < T)\) is satisfied at any time. The proper choice of the time step \((\Delta t)\) is therefore a most important ingredient of the explicit time integration.
The size of the element greatly affects the time step. In order to save CPU time, the element size should be as large as possible, but should be small enough in relation to the disintegration of the investigation area. The Young’s module is fixed for a given material. Experience shows that mass density can be increased by a factor of 7.0E+04. However, this may not be the case in some simulations.

Since PAM-STAMP is a dynamic FEM program, oscillations have to be avoided. The total simulation time should not be too small. If a very small simulation time is used, there will be a significant dynamic effect on the output curve, such as the internal pressure curve. For large problems of explicit procedures, the natural time can be reduced by artificially increasing the speed of moving elements. Artificial increasing of mass density is another way of achieving shorter run times. However, as mentioned before such attempts at improving the analysis efficiency may result in an increase of inertia effects which distorts the accuracy of the solution. The main factors limiting the velocity scaling are the size of the sheet metal, thickness, and extent of the deformation (i.e. height of bulging for instance). A thin sheet of large dimensions is likely to display inertial effects at much smaller accelerations compared to a thick sheet metal with small size.

For explicit codes the run time is directly dependent on the size of the finite element model (i.e. number of elements and nodes). This is a major advantage over implicit method, where the run time, consequently cost, is proportional to the square of the matrix
bandwidth (wave-front) of the mesh times the number of degrees of freedom for large models.

Another advantage of the explicit codes is that simpler algorithms are used to treat the contact conditions with small time increments. As it may be remembered that with implicit codes, treatment of contact brings another complication to the problem where a compromise has to be satisfied between convergence and accuracy of contact modeling.

However, for two-dimensional and small finite element models, implicit analyses appeared to be much more efficient than the explicit dynamic methods since due to the nature of the problem, first of all, there are few numbers of contact points to be handled. Second, the wave-front remains relatively small and easy to handle with implicit analyses.
7.3 Two dimensional FEA case studies

Many tube hydroforming applications are axisymmetric. Hence, design of process, part and tooling for such parts would be simpler and cost effective using two-dimensional (2D) FEA. This would also speed up the design procedure since many of the 2D FEA codes, such as DEFORM 2D, are easier to learn, use and apply different loading and boundary conditions when compared with three-dimensional codes, such as PAM-STAMP or LS-DYNA.

As discussed in previous ERC reports [Koç, 1998], and in the previous chapters through the understanding and application of modular design concept, most of the tube hydroformed parts can be analyzed by partitioning into simple 2D sections. Many common features like bulging and protrusion (Figure 7.1) can be simply analyzed approximately in 2D.

Limiting parameters such as maximum pressure, critical expansion, and the smallest corner radius determine the overall design procedure in tube hydroforming. Many of these limiting parameters can be estimated by a simple 2D evaluation. Even though the results of 2D analysis may be approximate for some cases, they would be useful enough at the initial phases of designing a new part for THF process.
Figure 7.1: Common features in tube hydroforming.
This chapter summarizes the results of four different FEA and comparisons with experimental findings. In addition, based on the feasibility of the 2D FEA, a series of simulations were performed to investigate the effect of various parameters in tube hydroforming. Results of this study are included in the form of simple guidelines for axisymmetric and Tee-shaped tube hydroformed parts in the next chapter.

7.3.1 Axisymmetric bulging of a short tube from SS 304 via use of viscous pressure medium

*Experimental conditions and results*

Stainless steel 304 tubes with 7.53 mm of thickness ($t_0$), 177 mm of length ($L_0$) and 64 mm of diameter ($D_0$) were formed into an axisymmetric bulge shape using a simple laboratory tooling as shown in Figure 7.2. Internal pressure was generated by the displacement of the volume inside the tube using a viscous medium that was loaded into the tube before starting the experiments. By use of this viscous medium, sealing problems were eliminated greatly. Cushion movement of a 160-ton hydraulic press available at ERC/NSM lab was utilized to displace the internal volume via a pin connected to the bolster of the press bed. Axial feeding was obtained using the ram motion of the press.
Figure 7.2: Isometric view of the vertical bulge tooling (VBT) that uses viscous medium to generate internal pressure by displacing volume via a “pressure pin”.

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Axial feeding and pressure could be applied simultaneously when cushion force is controlled properly with respect to its position. However, due to practical difficulties to repeat the experimental conditions, only a couple tubes were tested under simultaneous loading conditions. Figure 7.3 and Table 7.1 give the conditions of two cases tested (Case A and B). Basically, in Case A, internal pressure and axial feeding is more like a simultaneous application whereas in Case B, the loading is sequential, that is, axial feeding is applied first then pressure is increased.

Using a data acquisition system connected to the tooling, (a) internal pressure, (b) ram position, and (c) cushion position were measured as a function of time. Figure 7.4 illustrates the loading conditions measured during these experiments. The same loading conditions and tube dimensions were used in 2D FEA with DEFORM. Figure 7.3 depicts the FEA model and related dimensions tabulated in Table 7.7. As it is seen from this figure, the length at both sides of the bulge is not the same, and only feeding from one end (top) is applied while the other end is free. Table 7.2 presents the flow stress characteristics of SS 304 tubular material [Sokolowski, 1998].

**Results of FEA for SS 304 tubes**

One of the aims of performing these simulations was to investigate the validity of using of FEA software DEFORM for tube hydroforming technology. Another aim was to verify the 2D FEA for certain cases. Finally, verification with experimental measurements would be a strong base for future studies.
Figure 7.3: Geometry of the tooling, initial and deformed tube (axisymmetric tooling)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Parameters</th>
<th>Case A</th>
<th>Case B</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_{o} )</td>
<td>Initial length of the tube</td>
<td>178.3</td>
<td>177.4</td>
<td>mm</td>
</tr>
<tr>
<td>( L_{d} )</td>
<td>Final length of the tube</td>
<td>168.6</td>
<td>164.4</td>
<td>mm</td>
</tr>
<tr>
<td>( A_{f} )</td>
<td>Total axial feeding</td>
<td>9.7</td>
<td>13</td>
<td>mm</td>
</tr>
<tr>
<td>( R_{o} )</td>
<td>Initial radius of the tube</td>
<td>32</td>
<td>32</td>
<td>mm</td>
</tr>
<tr>
<td>( t_{o} )</td>
<td>Initial thickness of the tube</td>
<td>7.534</td>
<td>7.534</td>
<td>mm</td>
</tr>
<tr>
<td>( w )</td>
<td>Bulge width</td>
<td>65.3</td>
<td>65.3</td>
<td>mm</td>
</tr>
<tr>
<td>( A_{e} )</td>
<td>Fillet (entry) angle</td>
<td>30</td>
<td>30</td>
<td>degree</td>
</tr>
<tr>
<td>( R_{e} )</td>
<td>Fillet (entry) radius</td>
<td>12.7</td>
<td>12.7</td>
<td>mm</td>
</tr>
<tr>
<td>( R_{c} )</td>
<td>Corner radius</td>
<td>25.4</td>
<td>25.4</td>
<td>mm</td>
</tr>
<tr>
<td>( R_{1} )</td>
<td>Final radius of the bulge</td>
<td>47.55</td>
<td>47.55</td>
<td>mm</td>
</tr>
<tr>
<td>( \mu )</td>
<td>Coulomb friction coefficient</td>
<td>0.06</td>
<td>0.06</td>
<td></td>
</tr>
</tbody>
</table>

Table 7.1: Initial and final dimensions, and process conditions used in experiments and FEM simulations

<table>
<thead>
<tr>
<th>SS 304 tubular material</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \varepsilon )</td>
</tr>
<tr>
<td>( \bar{\varepsilon} )</td>
</tr>
<tr>
<td>( \sigma ) (MPa)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( \varepsilon )</th>
<th>0.002</th>
<th>0.179</th>
<th>0.26</th>
<th>0.363</th>
<th>0.445</th>
<th>0.542</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \bar{\sigma} )</td>
<td>304</td>
<td>588</td>
<td>715</td>
<td>831</td>
<td>968</td>
<td>1061</td>
</tr>
</tbody>
</table>

Table 7.2: True strain-stress values for SS 304 tubular material [Sokołowski, 1998]
Figure 7.4: Variation of process parameters (internal pressure and axial feeding with time) during experiments and FEM simulations for both cases. Other dimensions are provided in Table 7.1.
Figure 7.5 presents the deformation sequence of both cases A and B in FEM simulations. Notice the wrinkling of tube at the initial stages for Case B (i.e. time = 7.2 seconds). This is because of the excessive axial feeding. From Figure 7.4, the axial feeding is around 2.9 mm for Case B at the time of 7.2 seconds. The same amount of axial feeding is reached at around 2.3 seconds for Case A, and yet no wrinkling can be observed. For the same time frame, internal pressure values for both cases are close to each other, even pressure value of Case B is slightly higher than that of Case A.

Figures 7.6 and 7.7 illustrate the wall thinning variation along the longitudinal contour of the formed parts for both cases. Both figures present a comparison of FEA and experimental results. In Figure 7.6, the effect of friction coefficient on thinning can be observed to be small. However, it is observed from these figures that prediction of thickness at the edge regions is not as good as at the expansion region. This deviation might be due to the friction coefficient used in FEM simulations or might be due to the measurement errors during the evaluation of experimental samples.

Since forming took place into a closed die cavity, the thinning variation for both cases is very similar at the bulge region even though different loading paths were followed. However, it is obvious, from loading curves in Figure 7.3 and deformation sequences in Figure 7.5, that the same degree of forming can be obtained by less amount of energy resources (i.e. pressure and axial feeding) with simultaneous loading (i.e. Case A) when compared to sequential loading (Case B). This can be because of the strain hardening of
the tube walls after being deformed (wrinkling) under compressive axial feeding in Case B. That strain hardening would require more internal pressure to be formed into the required expansion at later phases of deformation.

Finally, Figure 7.8 illustrates the variation of thinning at middle and edge regions of tubes during deformation period for both cases. This figure indicates the wrinkling at the middle region as thickening during the initial phases of Case B (i.e. at around time = 7.2 seconds).

As a result, it can be concluded that 2D FEA with DEFORM can be used to predict deformation defects, and to verify process conditions provided that realistic material properties are given.
<table>
<thead>
<tr>
<th>$L_o$ (Case A/B)</th>
<th>$L_1$ (Case A/B)</th>
<th>$R_o$ (Case A/B)</th>
<th>$R_1$ (Case A/B)</th>
<th>$t_o$ (Case A/B)</th>
</tr>
</thead>
<tbody>
<tr>
<td>178/177 mm</td>
<td>168/164 mm</td>
<td>32 mm</td>
<td>47.5 mm</td>
<td>7.53 mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Case A</th>
<th>Case B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time =0 sec, Stroke = 0 mm</td>
<td>Time =0 sec, Stroke = 0 mm</td>
</tr>
<tr>
<td>Time =2.34 sec, Stroke=2.78 mm</td>
<td>Time = 7.2 sec, Stroke = 2.94 mm</td>
</tr>
<tr>
<td>Time =4.34 sec, Stroke=5.20 mm</td>
<td>Time = 2 sec, Stroke = 4.97 mm</td>
</tr>
<tr>
<td>Time =6.34 sec, Stroke=7.62 mm</td>
<td>Time = 4.8 sec, Stroke = 12 mm</td>
</tr>
<tr>
<td>Time =8 sec, Stroke=9.56 m</td>
<td>Time = 8 sec, Stroke = 13 mm</td>
</tr>
</tbody>
</table>

Figure 7.5: Comparison of deformation sequences for both cases. Notice the wrinkling for Case B at time = 7.2 second whereas for Case A wrinkling is avoided. Pressure and axial feeding profiles are given in Figure 7.4. Other dimensions are provided in Table 7.1.
Figure 7.6: Comparison of thinning for experiments and FEM simulations for the Case A (i.e. simultaneous loading). Other dimensions are provided in Table 7.1.
Figure 7.7: Comparison of thinning for experiments and FEM simulations for the Case B (sequential loading). Other dimensions are provided in Table 7.1.
Figure 7.8: Variation of thinning at the top edge and at the middle regions of the bulge part in function of time for both cases. Dimensions are provided in Table 7.1.
7.3.2 Finite element analysis (FEA) of a long axisymmetric bulge part—Comparison with published experimental data

Process conditions

Process conditions and geometry for this analysis are taken from a technical paper written about experiments and simulations conducted using DQSK steel tubes with 7.89 mm of thickness ($t_o$), 1150 mm of initial length ($L_o$), and 50.8 mm of diameter ($D_o$) [Srinivasan, et al., 1998]. In their simulations, LS-DYNA, an explicit 3D code, was used. Welding line was also modeled using different material properties.

Material properties of a similar steel grade are assumed as given in Table 7.3 in the 2D simulations performed for comparison purposes. A Coulomb friction coefficient of 0.05 is used to model the interface conditions between tube and die and punch.

An important characteristic of this part is that it is a simple axisymmetric bulging on a very long tube. Hence, axial feeding from the edges of the tube is not expected to be as effective as it would be if the tube were short.

The geometry of the die, initial and final tube dimensions are illustrated in Figure 7.9. The dimensions of the deformed tube are given in Table 7.3 and Figure 7.10. Due to the symmetry, only a quarter of the part was modeled in implicit, 2D DEFORM code. Related dimensions and material properties are presented in Table 7.3. Figure 7.10 shows the loading conditions as a function of time.
Figure 7.9: Geometry of the tooling, initial and deformed parts for long axisymmetric bulged part

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Symbols</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial length of tube</td>
<td>(L_o = 2L_{pe})</td>
<td>1150</td>
<td>mm</td>
</tr>
<tr>
<td>Initial radius of tube</td>
<td>(R_o)</td>
<td>25.4</td>
<td>mm</td>
</tr>
<tr>
<td>Initial thickness of tube</td>
<td>(t_o)</td>
<td>7.89</td>
<td>mm</td>
</tr>
<tr>
<td>Bulge width</td>
<td>(w)</td>
<td>107.6</td>
<td>mm</td>
</tr>
<tr>
<td>Fillet (entry) radius</td>
<td>(R_e)</td>
<td>1</td>
<td>mm</td>
</tr>
<tr>
<td>Final radius at edges</td>
<td>(R_1)</td>
<td>37.75</td>
<td>mm</td>
</tr>
<tr>
<td>Bulge radius at middle</td>
<td>(R_2)</td>
<td>27</td>
<td>mm</td>
</tr>
<tr>
<td>Axial feeding</td>
<td>(\lambda_f)</td>
<td>17</td>
<td>mm</td>
</tr>
</tbody>
</table>

DQSK (Draw Quality Specially Killed) Steel

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strength coefficient</td>
<td>(K)</td>
<td>567</td>
</tr>
<tr>
<td>Strain hardening exponent</td>
<td>(n)</td>
<td>0.145</td>
</tr>
<tr>
<td>Yield strength</td>
<td>(\sigma_y)</td>
<td>317</td>
</tr>
<tr>
<td>Ultimate tensile strength</td>
<td>(\sigma_u)</td>
<td>372</td>
</tr>
<tr>
<td>Elongation</td>
<td>Elongation</td>
<td>32.7</td>
</tr>
<tr>
<td>Friction coefficient</td>
<td>(\mu)</td>
<td>0.05</td>
</tr>
</tbody>
</table>

Table 7.3: Geometrical parameters process conditions and material properties used in experiments and FEM simulations
Figure 7.10: Process conditions (pressure and axial feeding profiles) used in experiments and FEM simulations for the long axisymmetric bulged part from DQSK steel. Dimensions of the deformed tube are also indicated [Srinivasan, 1998]
Results of FEA and comparison with published experimental data

Figure 7.11 presents the deformation sequence of simple bulging under the conditions depicted in Figure 7.10. Thinning distribution along the longitudinal contour length is shown in Figure 7.12 for experiments and both simulations.

As it can be seen, even though the prediction of thinning with DEFORM 2D is not as good as that of with 3D LS-DYNA, it is within an acceptable range of accuracy particularly when the ease and quickness of DEFORM 2D is considered. The deviation from experimental results can be due to the errors in (a) modeling of the material characteristics, (b) modeling of friction coefficient, and (c) extraction of thinning data from the published data [Srinivasan, 1998]. When the predictions made by both codes are compared with experiment, it becomes apparent that there may be errors in modeling of the material and friction.

Another interesting information related to this analysis was revealed when the same pressure profile was applied to the same tube without any axial feeding. The final length of the part was the same with that of the part formed under both pressure and axial feeding conditions. In other words, the axial pulling in was the same as axial feeding. This indicates that axial feeding was used only to keep the end of the tube sealed while pressurizing since under expansion the tube ends would be pulled in due to the volume constancy. This is a rather important point that may be difficult and tricky to achieve during real production. This kind of information can be only detected by proper FEA.
Figure 7.11: Deformation sequence of the long axisymmetric bulge part
Figure 7.12: Variation of thinning along longitudinal contour of the long axisymmetric part for both experiments and simulations. Dimensions of deformed tube are also indicated.
7.3.3 Investigation of the effect of corner radius on internal pressure- Comparison of experimental DATA with FEA

In tube hydroforming (THF) process, one of the factors that dictates the maximum internal pressure requirement is the smallest intricate feature on the desired part and its size. Usually, the corner radius is encountered as the smallest feature on a typical tube hydroformed part. Hence, in order to estimate the maximum pressure and develop guidelines for practical use, experimental data, available in the literature, was utilized and improved with analytical and finite element methods. This section gives details of the experimental results, analytical predictions, FEA and their comparison.

Published experimental results with SAE 1010

As reported by Mason [Mason, 1996], SAE 1010 tubular material at three different thickness values was pressurized into an intricate region (i.e. corner). Corner radius at different pressure levels for each initial thickness values were recorded as illustrated in Figure 7.13 [Mason, 1996]. Same information is shown in another form in Figure 7.14. It depicts the variation of $(R_c/t_o)$ ratio with internal pressure.
Figure 7.13: Measured data; variation of corner radii ($R_c$) with internal pressure ($P_i$) at different thickness values ($t$) [Mason, 1996]. Material is SAE 1010.
Figure 7.14: Measured data; variations of the ratio of corner radius to initial thickness \((R_c / t_o)\) with internal pressure for different thickness values of SAE 1010 material [after Mason, 1996]
Comparison of FEA results with published experimental data

Two-dimensional FEA was performed to be compare with the experimental results. DEFORM 2D was used for the simulations. Basically, for the FE model shown in Figure 7.15, internal pressure was increased linearly up to 300 MPa. Variation of radius was recorded, and compared with experimental results as shown in Figure 7.16 for SAE 1010 material case. A Coulomb friction coefficient of 0.96 was used to model the interface between tube and die. As it is seen from this figure, FEA results match very well with experimental findings. Similar FEAs on corner radius formation were performed for two more materials, A6061 and SS 304. Results of FEA for A6061 and SS304 are presented in Figure 7.17 and Figure 7.18, respectively. Material properties used in these FEA are tabulated in Table 7.4 for AKDQ 1008, A6061 and SS304.

| AKDQ 1008 (SAE 1010) (\(\sigma = K\bar{\varepsilon}^n\)) |
|-----------------|-------|-------|-------|
| K               | 0.227 | \(\sigma_y\) | \(\sigma_u\) |
| 537 MPa         |       | 240 MPa | 350 MPa |

<table>
<thead>
<tr>
<th>A6016 (AlMgSi0.5)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\bar{\varepsilon})</td>
</tr>
<tr>
<td>(\bar{\sigma}) (MPa)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SS 304 tubular material</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\bar{\varepsilon})</td>
</tr>
<tr>
<td>(\bar{\sigma}) (MPa)</td>
</tr>
</tbody>
</table>

Table 7.4 Properties of AKDQ 1008 (SAE 1010) sheet material used in calculations [Dana Corp., 1997], True strain-stress values for A6061 sheet material [Boehm, 1995], True strain-stress values for SS 304 tubular material [Sokołowski, 1998]
Figure 7.15: FE model and deformation sequence in FE analysis of corner radius formation
Figure 7.16: Comparison of experimental and FEA results for the formation of corner radius. FEA results agree with experimental findings well. FEA is performed using the material properties of low carbon steel 1010 (~AKDQ 1008) given in Table 7.4.
Figure 7.17: FEA findings for corner radius-to-initial thickness ratio versus internal pressure for Aluminum alloys (A6061-T4 is used in simulations. Properties are given in Table 7.4)
Figure 7.18: FEA findings for corner radius-to-initial thickness ratio versus internal pressure for stainless steel SS 304 is used in simulations. Properties are given in Table 7.4.
7.4 Three-dimensional FEA of Tee-shaped part

The objective was to verify the application of PAM-STAMP software in tube-hydroforming process simulations. T-shape hydroformed part was the case handled in this work. The results of tube hydroforming experiments on T-shape part done by H. J. Kim [Kim; 1993] were used to compare the simulation results from PAM-STAMP.

Geometrical parameters

In tube hydroforming process, pressurizing liquid is used to plastically form the tube into the T-shape die cavity. To postpone fracture in the bulge region, the tube material was pushed axially from both sides (see Table 7.5). In order make the simulation results comparable with Kim’s experimental results, all the parameters (i.e., geometry, process, and material) used in the simulations were obtained from the Kim’s experiments (see Table 7.5).

Material properties

The tube material used in the experiment was reported as JIS G 3452 SGP [Kim; 1993], which is equivalent to ASTM A53 or A120 [Kenkyujo, 1985]. However, exact flow stress characteristics (i.e., K and n values) of the tube material were not available in the literature. Since it was a low carbon steel, K and n values of LCS 1008 steel (K = 460 MPa and n = 0.19) were used in the simulation (see Table 7.5), assuming that it would give reasonably comparable results.
Figure 7. 19: FEA of tube hydroforming process of T-shape
Table 7.5  Geometrical parameters, material properties, and process parameters applied in the simulations
**Process parameters**

The applied internal pressure reaches the maximum at 32 MPa as used in the experiment (see Figure 7.20). It was indicated in the paper that axial feeding at both ends of the tube was applied for the distance of 53.8 mm. The total forming operation took about 7 sec. To reduce the simulation time, the forming time used in the simulation was speed up by 1000 times (i.e., the forming time in the simulation was 7 mili-sec). Figure 7.20 shows the loading curves applied in the simulations.

**Results, comparisons and discussions**

All the results obtained from the FE simulation were compared with the experimental results (see Figure 7.21, 7.22, and 7.23). Figure 7.24 shows comparisons of the bulge heights at the final forming step. Simulation (FEM) results showed good agreements with those of the experiment (EXP). The final protrusion height (Hp) was predicted to be 63 mm which was nearly the same as that of the experiment, 65 mm.

The overall thinning variations, in both longitudinal and hoop directions, from the PAM-STAMP simulation showed the same trends as of the experimental results (see Figure 7.22 and 7.23). In the protrusion zone, the thinning occurred as expected, and so did the thickening in the straight areas. Furthermore, see Figure 7.23, the thinning variations along hoop direction at the middle of the protrusion showed good agreements with the experimental results. The thinning variations along the longitudinal direction, however, were under predicted, especially in the straight regions.
Figure 7.20: Pressure curve applied in the experiment and simulation (top), punch velocity curve applied in the simulation (bottom)
Figure 7. 21:  Comparison of protrusion height at the final step

Figure 7. 22:  Comparison of thinning variation along the longitudinal contour length
Figure 7.23: Comparison of thinning variation along the hoop contour length
Axial feeding seems to have significant effects on the thickness variations especially in the areas near the edges of the tube. This can be noticed from Figure 7.22 showing large discrepancies near the edge of the tube and at the die corner. The discrepancies between the experiment and the simulation can be due to the followings:

- The difference in material properties used in the simulation (since the exact material properties for JIS G 3452 SGP were not available and sheet metal properties of LCS 1008 was assumed instead)
- Unrealistic coulomb-friction coefficient applied in the simulation
- Measurement errors in the experiments and during extracting data from the paper

In conclusion, PAM-STAMP has been proved viable for 3D process simulations of tube hydroforming processes, in this work, such as T-shaped geometry. The protrusion heights could be predicted quite accurately. However, the predicted thinning values were not quite accurate at the thickening zones; but the trends of the thinning variations were in good agreements. Material properties (K and n values) seemed to have significant effects on the thinning variations (see Figure 7.22 and 7.23).
7.5 Three-dimensional FEA of structural part

This part is courtesy from one of the member companies of Tube Hydroforming Consortium and has a 3-dimensional profile with variable cross-sections (see Figure 7.24). The forming process of the structural part consists of 1) bending process of the initial tube blank, 2) crushing the preformed bent tube in the die cavity and, 3) hydroforming the tube. In this report, only the crushing and hydroforming process are discussed. The bending process was also simulated in ERC/NSM and the results has been summarized in a separate ERC report [Shr, 1998].

*Finite element model*

The FEM model for simulating the 3D structural part is shown in Figure 7.25. In the simulation process, the upper die moved down first to crush the tube. The right and left pistons were used to apply axial feeding at both ends. After the lower die and upper die were fully closed, the axial feeding and pressure were applied simultaneously. It should be noticed that the tube used in the simulation was not the bent tube. Geometry of the tube used in the simulation has uniform thickness distribution as the initial tubular blank and is designed according to the bending process parameters, such as bending angle, rotation angle and feeding length.
Figure 7.24: Final shape of three-dimensional structural part
Figure 7.25: Finite element model for the analysis of structural part. Table 7.7 gives details of the model.
Process parameters

The material parameters were provided by a member company of the THF consortium at ERC/NSM, and are listed in Table 7.6. Table 7.7 gives the information about the structural part simulation process. The strength coefficient (K), strain hardening exponent (n), and pre-strain ($\varepsilon_o$) were presented in Table 7.6.

Flow rate was used to build up the forming pressure inside the tube during the simulation process. In this simulation, the flow rate curve was obtained from the real pressure curve provided (Figure 7.26).

According to the requirement of the real forming process, axial feeding was applied at both ends of the tube and was controlled by given piston velocity curves as used in the experiments (see Figure 7.27). As shown in Figures 7.24 and 7.25, this part is a complex 3D structure without any symmetry. Therefore, the axial feeding at the two ends of the part is different. The left piston has larger axial feeding. The total amount of axial feeding from left side was 122 mm, and 70 mm from right end. Initial length of the tube was 2574 mm with a diameter of 127 mm.

The crushing process and hydroforming processes were conducted separately due to the repositioning of the two pistons during the different process. After the crushing simulation, analysis of hydroforming process was performed using restart function in
PAM-STAMP. In this way, the deformation information generated in the crushing process can be used as the initial deformation history for the tube hydroforming process.

<table>
<thead>
<tr>
<th>Material</th>
<th>1008 AKDQ</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Unit</td>
</tr>
<tr>
<td>Young’s modulus</td>
<td>$E$</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>$\nu$</td>
</tr>
<tr>
<td>Initial thickness</td>
<td>$t_0$</td>
</tr>
<tr>
<td>Normal anisotropy</td>
<td>$r$</td>
</tr>
<tr>
<td>Strength coefficient</td>
<td>$K$</td>
</tr>
<tr>
<td>Offset strain</td>
<td>$\varepsilon_0$</td>
</tr>
<tr>
<td>Hardening exponent</td>
<td>$n$</td>
</tr>
</tbody>
</table>

Table 7.6: Material parameters of the tubular blank, $\sigma = K(\varepsilon_0 + \varepsilon)^n$

<table>
<thead>
<tr>
<th>Mesh type</th>
<th>Shell</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of nodes</td>
<td>8850</td>
</tr>
<tr>
<td>Number of elements</td>
<td>8746</td>
</tr>
<tr>
<td>Simulation time</td>
<td>12 (ms)</td>
</tr>
<tr>
<td>Type of contact surface</td>
<td>Node-to-segment Lagrangian contact</td>
</tr>
</tbody>
</table>

Table 7.7: FE model parameters
Figure 7.26: Variation of internal pressure and flow rate used in analysis of the structural part
Figure 7.27: Axial feeding and velocity curves used in the analysis of the structural part.
Comparison of FEA results with experimental data

The hydroformed tube and the crushed tube are shown in Figure 7.28 and Figure 7.29 respectively. Figure 7.28a illustrates the thinning distribution on the hydroformed tube. The marked area presents the most severe thinning spot. The maximum value of thinning in this area is about 37.55%. Usually, the acceptable maximum thinning in industry is about 25-30%. Therefore, a fracture is anticipated if the thinning reaches around 30%. The actual hydroformed part provided is shown in Figure 7.28b. As indicated in this figure, accurate prediction of the fracture spot was possible with the FEA when compared to the fracture spot on the actual part. Figure 7.29b shows the actual and predicted crushed part.

The actual hydroformed tube was cut at different sections for the measurement of thickness distribution along section contours. The positions of these sections are shown in Figure 7.30. The FEA results of thickness in the corresponding sections were compared with the measurements.

Figures 7.31 through 7.36 illustrate the thickness distribution along the contours of different sections for both simulation results and real part measurements. It can be seen from these figures that the thickness distribution trend in all sections is the same. The maximum difference in thickness between FEM results and measurements is not more than 10%. Overall, the main reasons for this difference may be due to the followings:
(1) A designed preformed tube with uniform thickness was used in the simulations. However, in reality, the preformed tube would have a non-uniform thickness (thickness inside and outside of local bends)

(2) The amount of feeding of actual part might be less than that in simulations

(3) There might be some deviations in initial thickness of the actual tube along both sections contour and longitudinal axis

(4) Positioning and orientation of the tube into the die might be different for simulations and experiments.

Figure 3.33 shows the section where the fracture occurred in experiments. As observed from these figures, the simulation results match the measurement of the actual part reasonably well.
Figure 7.28: Predicted and actual hydroformed part indicating the fracture spot.
Figure 7.29: Predicted and actual crushed tube
Figure 7.30: Location of the cross-sections, where thickness measurements are performed on the experimental part. Section 3 corresponds to the fracture location.
Figure 7.31: Thickness distribution along the contour of section 1. The maximum difference between simulation results and measurements is around 10%.
Figure 7.32: Thickness distribution along the contour of section 2. The maximum difference between simulation results and measurements is around 10%. 

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Figure 7.33: Thickness distribution along the contour of section 3. The maximum difference between simulation results and measurements is around 10%.
Figure 7.34: Thickness distribution along the contour of section 4. The maximum difference between simulation results and measurements is around 10%.
Figure 7.35: Thickness distribution along the contour of section 5. The maximum difference between simulation results and measurements is around 10%.
Figure 7.36: Thickness distribution along the contour of section 6. The maximum difference between simulation results and measurements is around 10%.
CHAPTER 8

CONSTRUCTION OF GUIDELINES FOR TUBE HYDROFORMING VIA
FINITE ELEMENT ANALYSIS

Since application of tube hydroforming technology in the mass production is relatively new compared to the other metal forming processes such stamping and forging, there are hardly available knowledge base, design rules, rules of thumb and past experience. Hence, application of this technology to new parts requires extensive trial and error work. Consequently, this leads to high development cost, which decreases the chance of tube hydroforming process’ competitiveness with others.

As mentioned before at the beginning of this study, main objective was to develop design guidelines to help designer at the beginning phases of part, tool and process design. One way to achieve this was to conduct experiments on different part types and process conditions, then compile them in a form of database. However, as it can be expected, this was not possible and cost effective.
Hence, use of FEA was considered to develop some guidelines for common features and cases. This investigation is part of a study to develop design guidelines for THF technology. Based on the verification of FEA with experimental data through many case studies presented in the previous chapter, additional and planned FEM simulations are conducted. Some pre-determined parameters are varied over a practical range to form a specific part while keeping the rest of the parameters at the same level in order to understand the effect of different parameters.

The factors that effect formability in tube hydroforming process can be categorized into three groups:

1. Geometrical parameters: Length, radius, angle, etc.
2. Material parameters: (a) Different material groups (steel, aluminum, etc), (b) Strain hardening exponent (n) values
3. Process parameters: Pressure, and axial feeding profiles

It is crucial to understand all the effects and interactions of all these parameters on the formability in order to successfully produce a tube hydroformed part. With this knowledge, the designer is empowered to design hydroformed parts as they can be manufactured with a minimum number of modifications.
Geometrical parameters of interest can be length between feature (protrusion or bulge) and edge \( L_{pe} \), protrusion diameter \( D_p \) or bulge width \( w \), and fillet radius \( R_e \) (see Figure 8.1 and 8.2). It has been observed that the length between feature (bulge or protrusion) and edge is the most influential factor effecting the forming (i.e. protrusion or bulge height without fracture). It is also known that friction is in fact the driving factor that makes this effect become prominent. Longer tube distance from the edge to feature is influenced more by friction, thus less metal flowing into expansion zone.

An axisymmetric simple bulge and non-axisymmetric Tee-shaped parts are used in this study since these kinds of features are very common on the tube hydroformed parts as evaluated in the previous chapters. The axisymmetric part geometry used in the study is shown in Figure 8.1. Nomenclature used throughout this report is also presented in the same figure. Geometry and nomenclature used for Tee-shaped part are shown in Figure 8.2.
Figure 8.1: Geometry, dimensions and nomenclature for the axisymmetric bulge part and tooling.
Figure 8.2: Geometry, dimensions and nomenclature for the Tee-shaped part and tooling.
8.1 Planned FEA for axisymmetric bulge part

8.1.1 Investigation of the effect of geometrical parameters for axisymmetric bulge part

Table 8.1 tabulates the factors (geometrical parameters), their selected ranges and levels used in FEM simulations. Predicted values of the bulge radius ($r_1$) are used to compare the effect of geometrical parameters. In these FEM simulations, internal pressure versus time and axial feeding versus time profiles are kept the same in order to compare the results.

<table>
<thead>
<tr>
<th>Factors (Geometrical parameters)</th>
<th>Levels (Dimensions)</th>
</tr>
</thead>
<tbody>
<tr>
<td>#</td>
<td>Name</td>
</tr>
<tr>
<td>1</td>
<td>Length</td>
</tr>
<tr>
<td>2</td>
<td>Fillet radius</td>
</tr>
<tr>
<td>3</td>
<td>Bulge width</td>
</tr>
<tr>
<td>4</td>
<td>Entry angle</td>
</tr>
</tbody>
</table>

Responses

1 Bulge radius $r_1$ ($H_b$) mm

Geometrical parameters that are constant

| #      | Diameter | $D_o$ (= $2r_o$) | 45 mm |
| 2      | Thickness | $t_o$ | 2 mm |

Table 8.1: Factors, responses and ranges (levels) used in the FEM simulations to investigate the effect of geometrical parameters
**Simulation model and process conditions**

Table 8.2 presents the process condition profiles (i.e. pressure and axial feeding) used in the simulations. Pressure and axial feeding profiles are determined through analytical calculations and improved via FEA. Table 8.3 includes the properties of the LCS 1008 material used in the FEM simulations.

<table>
<thead>
<tr>
<th>Time</th>
<th>0 sec</th>
<th>1 sec</th>
<th>10 sec</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Pressure</strong></td>
<td>0 MPa</td>
<td>25 MPa</td>
<td>33 MPa</td>
</tr>
<tr>
<td><strong>Punch velocity</strong></td>
<td>1 mm/sec</td>
<td></td>
<td>1 mm/sec</td>
</tr>
<tr>
<td><strong>Axial feeding</strong></td>
<td>10 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Friction coefficient</strong></td>
<td>Coulomb, 0.06 ($\tau_f = \mu \sigma_n$)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(100 MPa = 18.5 ksi = 1000 bar)

Table 8.2: Process conditions

<table>
<thead>
<tr>
<th>LCS tubular material</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\bar{e}$</td>
</tr>
<tr>
<td>$\bar{T}$ (MPa)</td>
</tr>
</tbody>
</table>

(100 MPa = 18.5 ksi)

Table 8.3: True strain-stress values for LCS 1008 tubular material [Arnold, et al. 1998]

**Results of FEA on geometrical factors on axisymmetric bulge part**

Final bulge height ($r_1 = H_b$) is accepted as the response to variations in the geometrical factors since it would be a good indication of formability for this case. Results of simulations are given in Table 8.4.
Figure 8.3 illustrates the effect of different geometrical parameters on the final bulge radius. It is seen from this figure that the distance between the feature (i.e. such as a bulge, protrusion, etc.) and the edge where axial feeding is applied has the greatest effect on the bulge radius. Effects of fillet radius ($R_e$), bulge width ($w$), and fillet angle ($A_e$) are very small compared to that of length ($L_{pe}$). In order to justify the effect of $L_{pe}$, an additional simulation was performed with a shorter value ($L_{pe} = 100$ mm). Result of this additional simulation is also included in Figure 8.3.

<table>
<thead>
<tr>
<th>Number</th>
<th>$L_{pe1} - L_{pe2}$</th>
<th>$R_e$</th>
<th>$w$</th>
<th>$A_e$</th>
<th>Expansion ratio, ($r_1 / r_o$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>150 - 150</td>
<td>9</td>
<td>60</td>
<td>0</td>
<td>1.55</td>
</tr>
<tr>
<td>2</td>
<td>200 - 200</td>
<td>9</td>
<td>60</td>
<td>0</td>
<td>1.49</td>
</tr>
<tr>
<td>3</td>
<td>250 - 250</td>
<td>9</td>
<td>60</td>
<td>0</td>
<td>1.45</td>
</tr>
<tr>
<td>4</td>
<td>200 - 200</td>
<td>6</td>
<td>60</td>
<td>0</td>
<td>1.48</td>
</tr>
<tr>
<td>5</td>
<td>200 - 200</td>
<td>12</td>
<td>60</td>
<td>0</td>
<td>1.54</td>
</tr>
<tr>
<td>6</td>
<td>200 - 200</td>
<td>9</td>
<td>45</td>
<td>0</td>
<td>1.46</td>
</tr>
<tr>
<td>7</td>
<td>200 - 200</td>
<td>9</td>
<td>75</td>
<td>0</td>
<td>1.66</td>
</tr>
<tr>
<td>8</td>
<td>200 - 200</td>
<td>9</td>
<td>60</td>
<td>30</td>
<td>1.484</td>
</tr>
<tr>
<td>9</td>
<td>200 - 200</td>
<td>9</td>
<td>60</td>
<td>45</td>
<td>1.478</td>
</tr>
</tbody>
</table>

Table 8.4: Results of FEM simulations for different levels of geometrical factors
Figure 8.3: Effect of geometrical factors on the bulge height of an axisymmetric part, \( r_1 \). (a) \( \frac{L_{pc}}{D_0} \), (b) \( w \), (c) \( \frac{R_c}{t_0} \), (d) \( A_e \) \( (r_0 = 22.5 \text{ mm}, t_0 = 2 \text{ mm}) \)
8.1.2 Investigation of the effect of material parameters

In order to investigate the effect of material parameters on the bulge radius of a simple axisymmetric part, first, three different material groups are simulated. Second, strain-hardening exponent of one material group is varied to observe its effect on the bulge radius.

Table 8.5 gives the flow stress properties of low carbon steel 1008, stainless steel 304, and aluminum A6061, which are used in the simulations. In the first phase, internal pressure profiles for each material group are calculated using their corresponding yield and ultimate tensile strength values. All other geometrical parameters and axial feeding profile are kept the same during this phase to obtain the pure effects of material parameters. Variation of internal pressure requirement for each material to obtain similar bulge radius and thinning is investigated.

Figure 8.4 shows the difference in required internal pressure for three material groups to reach the same level of expansion. It is apparent that stainless steel requires the highest pressure to obtain a certain level of bulging when compared to low carbon and aluminum cases. Note that axial feeding is the same for all cases.
<table>
<thead>
<tr>
<th>LCS tubular material</th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$\bar{\varepsilon}$</td>
<td>0.002</td>
<td>0.129</td>
<td>0.265</td>
<td>0.339</td>
<td>0.401</td>
<td>0.450</td>
</tr>
<tr>
<td>$\bar{\sigma}$ (MPa)</td>
<td>260</td>
<td>359</td>
<td>426</td>
<td>460</td>
<td>503</td>
<td>528</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SS 304 tubular material</th>
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<th></th>
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<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$\bar{\varepsilon}$</td>
<td>0.002</td>
<td>0.179</td>
<td>0.26</td>
<td>0.363</td>
<td>0.445</td>
<td>0.542</td>
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<tr>
<td>$\bar{\sigma}$ (MPa)</td>
<td>304</td>
<td>588</td>
<td>715</td>
<td>831</td>
<td>968</td>
<td>1061</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>A6061 (AlMgSi0.5)- air quenched</th>
<th></th>
<th></th>
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</tr>
</thead>
<tbody>
<tr>
<td>$\bar{\varepsilon}$</td>
<td>0</td>
<td>0.04</td>
<td>0.08</td>
<td>0.12</td>
<td>0.16</td>
<td>0.2</td>
</tr>
<tr>
<td>$\bar{\sigma}$ (MPa)</td>
<td>80</td>
<td>100</td>
<td>115</td>
<td>125</td>
<td>135</td>
<td>145</td>
</tr>
</tbody>
</table>

Table 8.5: True strain-stress values for materials used in FEA

In the second phase, strain hardening exponent (n) value (in flow stress equation $\bar{\sigma} = K\bar{\varepsilon}^n$) of low carbon 1008 material is varied to investigate its effects on formation of the bulge.

Table 8.6 gives the hypothetical strain hardening exponent (n) values that are used for LCS material group. In these simulations, all parameters including geometry, internal pressure and axial feeding profiles are kept the same.
<table>
<thead>
<tr>
<th>LCS material group</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
</tr>
<tr>
<td>K (MPa)</td>
</tr>
<tr>
<td>n</td>
</tr>
</tbody>
</table>

Table 8.6: Ranges of hypothetical strain hardening exponent (n) for LCS material group (in $\bar{\sigma} = K\bar{\varepsilon}^n$)

Figure 8.5 illustrates the effect of strain hardening exponent value (n) on the expansion ratio. As strain hardening (n) value increases, flow stress of the material decreases hence expansion increases, too. In another word, materials with the same strength coefficient (K) but high (n) values can be deformed easily. They require less pressure compared to materials with low (n) value.

Effect of strength coefficient (K) on expansion is obvious that as K increases flow stress increases hence it becomes difficult to deform (bulge) the part. For high K values, the flow stress becomes so large that for the given pressure profile, tube can not be expanded, and under the effect of axial feeding the tube wrinkles.

8.1.3 Investigation of the effect of process parameters, i.e. internal pressure

In this set of simulations, internal pressure ($P_i$) profile is varied within a range determined by the maximum expansion pressure ($P_b$). All the other material and geometrical parameters are kept the same. The maximum expansion pressure is given as:
Figure 8.4:  (a) Effect of material on the required internal pressure for the same expansion ratio and thinning (b)
Figure 8.5: Effect of strain hardening exponent (n) on the expansion ratio in simple axisymmetric bulging of a tube
\[ P_b = \frac{2\sigma_U t_o}{D_o - t_o} \] \hspace{1cm} (8-1)

Where maximum expansion pressure is a function of ultimate tensile strength \((\sigma_U)\) of the material, initial thickness \((t_o)\) and initial diameter \((D_o)\) of the tube. Similarly, pressure to start deformation is given as (yielding pressure):

\[ P_y = \frac{2\sigma_y t_o}{D_o - t_o} \] \hspace{1cm} (8-2)

Where yielding pressure is dependent on the yield strength \((\sigma_y)\) of the material, initial thickness and diameter of the tube.

In these simulations, axial feeding profile is kept constant at 10 mm, and pressure profile is varied such that timing of yielding pressure is changed as seen in Figure 8.6a.

Maximum expansion ratio for different pressure profiles is illustrated in Figure 8.6b. From this figure, it can be observed that the earlier the application of yield pressure the higher the expansion is obtained. Late application of the yield pressure is observed to allow wrinkles to occur before the tube expands into the bulge area. Formation of wrinkles can be described as strain hardening of the wrinkled zones. Hence, wrinkles would prevent reaching high expansion. As a result, in THF applications, a pressure level high enough to start deformation of the material should be applied as early as possible to achieve high expansion ratios without any wrinkling.
Figure 8.6: Effect of pressure profile (a) on the expansion ratio for a simple axisymmetric bulging (b)
8.2 Planned FEA for non-axisymmetric Tee-shaped part

In this section, the effects of geometrical parameters on the protrusion height were investigated via the use of FEA. In order to quantify the effect within acceptable accuracy by running a small number of simulations, “design of experiment” DOE methods were applied in this study.

This chapter contains the design of the sensitivity analysis plan where the response and factors were specified. The array of the simulations, in which the factors were varied, was designed using low cost response surface method [Allen, 1999]. The effects of the geometrical parameter will be shown and discussed at the end.

8.2.1 Design for sensitivity analysis of geometrical parameters

Response and Factors

The most desirable feature of a tube hydroformed part, especially for a T-shaped part, is the highest protrusion height. Thus, in this study, the protrusion height \( H_p \) was selected to be the response. Table 8.7 summarizes the response and factors that were used in this study. The geometrical parameters that were anticipated having significant effects on the response were selected as follows: distance between protrusion and edge \( L_{pe1} \) and \( L_{pe2} \), fillet radius \( R_e \), and protrusion diameter \( D_p \).
<table>
<thead>
<tr>
<th>Factors</th>
<th>Levels</th>
<th>Ranges (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>-1</td>
<td>-0.5</td>
</tr>
<tr>
<td>Lpe1</td>
<td>100</td>
<td>200</td>
</tr>
<tr>
<td>Lpe2</td>
<td>100</td>
<td>200</td>
</tr>
<tr>
<td>Re</td>
<td>6</td>
<td>9</td>
</tr>
<tr>
<td>Dp</td>
<td>25</td>
<td>30</td>
</tr>
</tbody>
</table>

Table 8.7: Response, factor and factor levels used in the simulations of Tee-shaped part
Five levels of each factor were selected within a practically wide range of typical dimensions of T-shaped parts. With the use of five level factors, non-linear effects of the geometrical parameters could be revealed. To obtain the effect of the all-possible combinations of levels of all factors, full factorial array have to be applied. However, in this study, the number of the simulation runs was minimized by the application of a fractional factorial design. The simulation array is discussed in the following section.

Simulation Array

As discussed earlier, the concept of fractional factorial designs was applied to reduce the investment on the simulation time. Fractional factorial designs are based upon the assumptions that some interactions between factors are insignificant. While the full factorial designs consider all combinations of the factor levels, the fractional factorial designs consider only the combinations that would reveal presumably significant effects. Furthermore, in most engineering experiments, the fundamental knowledge in the problems would help safely reducing insignificant factor combinations. Low cost response surface method was introduced by Allen [Allen, 1999]. This method was developed to replace the central composite design (CCD), which is commonly used in industry. In present work, the array for the simulations was designed to employ 4 factors with 5 levels, 1 response, it requires 18 runs. In the array, there were 3 repeated runs. Theses repeated runs were used to minimize the random error of the finite element analysis itself. Table 8.8 contains the array used in the study. In order to achieve the same information, at least 27 simulations would have been required if CCD were used.
<table>
<thead>
<tr>
<th>Run</th>
<th>Lpe1</th>
<th>Lpe2</th>
<th>Re</th>
<th>Dp</th>
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<tbody>
<tr>
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<td>0</td>
</tr>
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<td>-0.5</td>
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<td>18</td>
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<td>-1</td>
<td>-1</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 8.8: Simulation array; levels for each factor and simulation run order
**Geometrical Parameters**

According to the factor ranges shown in Table 8.7, the four factors of the geometrical parameters were varied in the array of simulations. To make all these simulations comparable, the original tube dimensions were kept constant in all the simulations. A summary of the fixed geometrical parameters is shown in Table 8.9.

**Material Properties**

Low carbon steel (LCS 1008) was selected to be the tube material used in the simulations. The material properties are summarized in Table 8.8. In this study, the material properties were fixed parameters. However, the effects of material properties on formability in THF will be conducted as the next step of this study.

**Process Parameters**

Like the material properties, a set of loading paths (i.e., pressure versus time and axial feeding versus time) was applied in all the simulations. In order to determine the loading curves necessary to hydroform a part, simple analytical models were used to obtain some of the initial values of the curves [Koç; 1998].

For example, the analytical models would yield the necessary pressure to initiate deformation in the tube, the maximum pressure that would form the tube without bursting, and the amount of axial feeding material. However, these values are not obtained within the time domain. Hence, couple FE simulations are usually run to refine
the loading paths. A combination of the geometrical factor ranges was selected for a preliminary simulation to determine the loading curves, which would be applied for all of the simulations. The combination of $L_{pe1}$ and $L_{pe2}$ 300 mm, $R_e$ 12 mm and $D_p$ 45 mm was selected as a representative of the entire combinations. Figure 8.7 shows the loading paths applied for all of the simulations.

<table>
<thead>
<tr>
<th>Geometrical Parameters</th>
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<tbody>
<tr>
<td>to</td>
</tr>
<tr>
<td>Do</td>
</tr>
<tr>
<td>Dp</td>
</tr>
<tr>
<td>Re</td>
</tr>
<tr>
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<tr>
<td>Lpe2</td>
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</table>

<table>
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<tr>
<td>E</td>
</tr>
<tr>
<td>K</td>
</tr>
<tr>
<td>n</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Process Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. Pressure</td>
</tr>
<tr>
<td>Axial Feeding</td>
</tr>
<tr>
<td>Friction Coef.</td>
</tr>
</tbody>
</table>

Table 8.9: Geometrical, material, and process conditions used in the simulations
Figure 8.7: Pressure curve (top) and punch velocity curve (bottom) applied in the simulations
Discussion of results

The response of interest, the protrusion height \( H_p \), was measured from the simulation results at the final step. The thinning in the protrusion areas was checked to assure that the formed protrusion was usable (i.e., thinning up to 20-22\% would be unacceptable). The protrusion heights were plotted against the corresponding levels of the factors (see Figure 8.8).

To reveal the effects of the generalized geometrical parameters, the ratios the protrusion height to the original outside diameter \( D_o \) of the tube and the corresponding ratios of the response to \( D_o \) were also tabulated (see Table 8.10).

It can be concluded that, as expected, the most influential geometrical parameter was the distance between protrusion and edge \( L_{pel1} \) and \( L_{pel2} \). In addition, the effect of the protrusion diameter \( D_p \) was found equally significant. The length of the tube, however, has an opposite effect from that of the protrusion diameter \( D_p \). Forming of shorter tubes yields higher protrusions than that of longer tubes with less thinning in the protrusions. The forming of tubes with larger protrusion diameter yield higher protrusions than that of smaller protrusion diameter with less thinning in the protrusions. The fillet radius \( R_o \), on the other hand, has the smallest effect on the protrusion height. With larger fillet radius, the higher protrusion was obtained.
Figure 8.8: Main effect chart of the geometrical parameters
Table 8.10: Normalized main effects of the geometrical parameters on the protrusion height
Based on the knowledge on the effects of friction in metal forming, the observed effects of the tube length ($L_{pe1}$ and $L_{pe2}$) can be easily explained. The shortest tube length results in the highest protrusion height, but the longest tube length yields the shortest protrusion height. From Figure 8.8, it can be noticed that the response is very sensitive to the short lengths of the tubes. However, the response almost does not change when the tube is very long. In other words, there exists a critical tube length after which the formability (i.e., the protrusion height) is almost not effected. Furthermore, thickening edges of the tube tend severe when the tube becomes longer.

The main effect of the protrusion diameter is almost linear. The protrusion height increases as the protrusion diameter increases. This may be explained by constraining force induced by the cross sectional area of expansion zone. The effect of the fillet radius (Re) is almost insignificant. The protrusion height slightly increases, as the fillet radius becomes bigger. This can be explained by the bending stresses. Smaller fillet radius will induce more bending stress while forming, thus more difficult to form.

*Interactions between the geometrical parameters*

The $L_{pe1}$ and $L_{pe2}$ were expected to give the similar effect on the response. However, possible interactions between these two factors were of interest. It was because the information on these interactions might be applied in other tube geometries where different lengths between protrusions are inevitable, such as double T-shaped protrusions.
Figure 8.9 shows the interactions between the $L_{pe1}$ and $L_{pe2}$. Clearly, when both of them ($L_{pe1}$ and $L_{pe2}$) are short the forming would yield the highest protrusion. This can be observed from the main effect curves of the Lpe1 and Lpe2 in Figure 8.8 (also see Figure 8.9, the contour plot, $L_{pe1}$ 100 mm and $L_{pe2}$ 100 mm). However, it can be seen that when one side of the tube is very long while the other side is short, the protrusion height obtained is still satisfactorily high (see the contour plot, $L_{pe1}$ 100 mm and $L_{pe2}$ 300 mm). This is an indication of a fact in THF that a feature that requires large expansion should be located near the edge where axial feeding is provided.
Figure 8.9  Response surface (up) and response contours (bottom)
CHAPTER 9

METHODOLOGY, CONCLUSIONS AND RECOMMENDED WORK

In the preceding chapters, various aspects of design for THF process are presented in detail. However, in the real world of engineering, usually a group of engineers has to consider all the aspects simultaneously and interactively to find optimum solutions and results depending on the case they are interested in.

In this section, an overall methodology for design of THF part, tooling and process is given. Whenever necessity is raised, references to related chapters and appendices are indicated. The design methodology for THF covers all the steps beginning with the concept phase to the production of final part. Figure 9.1 illustrates this methodology in block diagrams.

It is the understanding of the author that the followings should be pointed out before getting into details of the methodology. First of all, full advantage of THF technology can only be gained when the whole assembly or the surrounding parts of a compartment of an
automotive (or aircraft), for instance, are intended for manufacturing with THF as THF offers consolidation of many parts into fewer and reduction in assembly steps.

Second, there is not a single and simple way out for any design case in THF technology. Altering the restrictions of the design within the permitted boundaries, different and better solutions can be accomplished. Therefore, allowances on the restrictions should be known thoroughly before starting a design procedure for THF process.

Third, a combination of factors such as process parameters, (i.e. internal pressure and axial feeding, their coordination and control), material parameters, geometry parameters are effective in achieving a sound and desired final product. Thus, extent of the above factors where they can be stretched up to should be considered.

9.1 Methodology

Having put the general considerations on the table, it can be, now, explained how to proceed for design of part, tooling and process in THF technology given a specific part, its usage, functional, assembly and manufacturability restrictions as follows:

1. Conduct a brief feasibility analysis knowing the desired shape of the part. This short initial feasibility should include overall technical and economical aspects of production
- Functional requirements,
- Current part geometry, material, and specifications
- Experience

Conceptual (initial) part geometry, material, and specifications

Preliminary feasibility analysis based on visual observations on critical features, expansions, etc.

Does the part seem producible with THF?

Feasibility with cross-sectional analysis based on circumferential expansion, thinning, required pressure, etc?

Does the part seem producible with THF?

Predict approximate process parameters such as pressure, force, feeding, thinning, etc.

Check the predicted process parameters with limitations

Necessary Changes

Necessary Changes

Necessary Changes

3

1

2

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Perform two-dimensional (2D) finite element analysis (FEA) of critical cross-sections

Compare the thinning and cavity filling with desired part

Construct three-dimensional (3D) model of the part and tooling for 3D FEA with necessary changes, improvements, and adjustments.

If the part is complex and requires pre-forming, estimate a bent tube geometry with uniform circular cross-section by blending critical cross-sections linearly to each other

Then, perform bending FEA before going for the hydroforming analysis

If the part is relatively simple and requires no pre-forming, simply assume a straight tube with uniform circular cross-section and thickness for hydroforming analysis without any pre-forming

Check the predicted process parameters, thinning, and cavity filling with limitations and desired shape

- Finalize the design of process parameters, tooling and part after necessary iterations,
- Store the results into TH-CADS for future use

Figure 9.1: Block diagram of the proposed methodology for design of THF process
2. Evaluate the geometrical data of the part in detail. Most of the time CAD data would be advantageous to do this:

- Inspect the critical regions on the part
- Identify smallest and largest cross-sections
- Determine the change of spline geometry in the plane or space to decide about any prior bending and pre-forming operation
- Check for sharp transition regions between different cross-sections

Such an evaluation would give designer an overall and initial idea about the part he/she is dealing with.

3. By comparing different cross-sections and their size, and by simple calculations, determine an approximate initial tube diameter and wall thickness if not indicated by the requirements.

4. Calculate circumferential elongations (i.e. expansion ratio) for all cross-sections on the part.

5. Considering the material data, thinning limitations and possible pre-forming operations, try to assess the feasibility of each cross-section.

6. Upon detection of unfeasible cross-sections, work on them varying their size, location and relation with others within the allowed boundaries.
7. After deciding that the part seems roughly feasible, calculate critical values of various process parameters such as internal pressure, axial force and feeding, clamping force, counter force, etc. using the information and guidelines presented in other chapters and appendices.

8. Check each process parameter against available limitations. If there is any conflict, go back to the related step, and try to adjust different parameters to the allowed degrees with required compensations and compromises.

If the part seems still feasible for THF process, and offers the desired advantages, proceed with the next step. Otherwise iterate among the prior steps until reaching a decision.

9. Using full knowledge of material and part geometry parameters, analyze critical cross-sections with 2D FEA. Optimize cross-sections with results of simple and quick 2D FEA. 2D FEA of longitudinal cross-sections can also be performed if necessary after some assumptions such plane strain or symmetry sections through which no metal flows.

By investing only very little, producibility of a part can be obtained through the steps until here. Notice that, through steps 1 to 9, analytical models, classification of parts, database and TH-CADS can be used extensively.
10. Now, it is time to construct a full 3D CAD model of the part if not done yet. Necessary adjustment may arise during detailing of the shape and dimensions

11. Design of final (finishing) dies and parting lines can be easily constructed using 3D model of the part

12. In order to analyze the deformation of the part in the dies, a pre-form geometry may be needed depending on the case being handled. If the part is relatively simple, lies on a straight spline axis or lies on a plane with uniform cross-sections, an assumption of uniform initial tube diameter along the part’s axis should be good enough.

13. If the part geometry is complex with many undercuts, bends and crushings, each cross-section a different circular starting tube can be assumed. By blending these circular but different sized initial tube cross-sections along the part’s spline axis, an intermediate (i.e. blocker) part and die geometry can be obtained. This geometry should determine the crushing stage.

14. Then, by simply deciding on the initial tube’s diameter, this shape can be converted into the bent tube shape. This should define the necessary bending operations. Later, initial length of the tube can be calculated by using the bending angles and radii on the bent tube.
15. Hence, with the initial tube geometry and already decided, if not given, material information, first 3D FEA can be conducted. Note that process parameters are already calculated approximately during steps 7 – 9.

16. First FEA would reveal the whole information required for design of a part and process: stress and strain distributions, thinning and defects, etc.

17. With the use of approximate calculations methods and TH-CADS, few 3D FEA should be enough to finalize the design before sending the results for manufacturing of the try-outs dies.

18. If bending or pre-forming operations are required, their 3D FEA should be performed before step 15.

19. Upon finding of any defects or realization of a last minute unfeasibility during try-outs FEA can be repeated with necessary changes until problems are eliminated. Notice that at this point having couple more FEA would not be difficult since the CAD and FE models are already done and set for further use.
9.2 Conclusions

The scope of this was to develop analytical models, guidelines, and methodology to help the designer during the initial phases of a design sequence, after helping him understand the THF process, as shown in Figure 9.2.

Along this line, first, tube hydroformed parts were classified into four major groups depending on their spline geometry. Protrusions, bulges, bends and crushing were also identified as common features on THF parts. Important geometrical parameters on classified parts were specified for future use in calculations.

Analytical methods were studied through previous publications and other information sources. Analytical models for prediction of process parameters including internal pressure, axial, counter and clamping loads were developed and improved. These models presented in previous chapters will help designers in the following ways:

(a) Provide an approximate prediction for process parameters,
(b) Provide the designer an overall understanding of the process and formation of a part in question
(c) Let the designer realize the interactions between process, material, tooling and system parameters

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Figure 9.2: Design procedure for THF process including the guidelines and TH-CADS to help designers at the initial stages of part, tooling and process design.
Moreover, analytical models are a basis for the development of design guidelines for THF process. FEA of particular parts under certain process conditions has to be performed as the next step after mechanistic analysis in order to determine the process parameters to be used in try-out production.

Limitations of the THF process, such as wrinkling and bursting, are explained with experimental and analytical examples.

Determination of flow stress characteristics of tubular materials is also presented along with simple analytical techniques. Principle methodologies for determining flow stress characteristics of tubular materials were outlined for possible use. Improvement of current testing methods becomes possible with calculation of wall thickness during online measurement.

2D and 3D FEA were performed to verify the codes modeling with experimental data through many case studies. Furthermore, FEA and “design of experiments (DOE)” techniques were combined to develop guidelines for common part types in THF.

During this study, it was expressed that for the ultimate success of THF process for any given material and geometry conditions application of process parameters (i.e. internal pressure, axial force and feeding), and their coordination (i.e. loading path) with each other and time are very important. Even though some guidelines were presented at the
end of analytical studies and FEA results, overall success of designing loading paths in terms of real time can only be achieved via use of controlled FEA. This concept will be presented as a recommended work at the end of this chapter.

Following items are, briefly, contributions as a result of this study:

1. Principle guidelines to determine the producibility of a part with THF
2. A methodology for part, tooling and process design for THF technique
3. Analytical models for predicting forces, internal pressure, thinning, etc.
4. An initial version of a computer aided design system (TH-CADS) as an integration of design program (TH-DESIGN) with a database (TH-DBASE) to compute process parameters for various parts efficiently for given final part and material information, Figure 9.3.
Figure 9. 3: Concept of the computer program (computer aided design system for tube hydroforming; TH-CADS) package to help designers at the initial stages of part, process and tooling design for THF. Program consists of two main parts as analytical models and database.
9.3 Recommended future work

Based on the experience gained throughout this study on tube hydroforming technology, which is relatively new and has many unknowns, author would like to express his ideas on the recommended future work that he anticipates and wishes:

(1) Design of loading path (i.e. prediction of pressure versus time and feeding versus time) is the key to a successful part production. Since it is not practical and expensive to obtain optimum loading paths by trial-and-error in real production, FEA can be used to do this. Even in FEA trial-and-error is too lengthy and tedious as experienced. Hence, a concept of controlled FEA to find out optimized loading paths is introduced at ERC. Figure 9.4 illustrates the concept schematically.

First, couple important limiting criterion, such as (a) direction of the velocity of the nodal points, (b) wrinkling and (c) thinning, are determined as explained in previous chapters. Second, a scanning and control loop connected to the FEA code, such PAM-STAMP or DEFORM, is used to detect and vary. Third, an initial starting loading path for both pressure and feeding is predicted using analytical models and guidelines. Then, the FEA is started for any specific part, material and geometry conditions. At every step increment, the scanning module scans all the elements and nodes of the FE model, and check for their velocity direction, for instance as shown in Figure 9.4. If the velocity of any node is inwards, it is detected as a beginning of wrinkling defect, and
Figure 9.4: Flow chart of the methodology to predict the loading path versus time (it is assumed that total forming time is divided into n = 10 steps)
consequently to prevent wrinkling internal pressure is increased while feeding is kept at its previous value. If excessive thinning of an element is detected during scanning at any step, then feeding will be increased while pressure is kept at the prior value. This way, complete forming of the part will be accomplished, at the same time an optimized loading path will be stored for experimental use. This concept would eliminate many trial-and-error steps in FEA and real try-outs.

(2) In order to assess the process parameters and loading paths with reasonable accuracy, even with the optimized concept explained above, correct and dependable characterization of tubular materials should be provided to the FEA persons. Use of tensile test results would not reflect the actual deformation conditions in THF process. Hence, their use in FEA for THF will distort the results that are sought from the reality. To obtain suitable flow stress characteristics of tubes, hydraulic bulge tests have been developed at ERC as briefly explained before. However, this testing tooling and method is not suitable for plant floor purposes although principally it works. Thus, improvements on the tooling and method should be performed along the lines described in the relevant chapter.

(3) For the similar reasons, measurement of friction coefficient for different lubricant and material combinations should be carried out using appropriate methods and tooling.
(4) As many tube hydroformed parts require pre-processes such as bending and crushing, detailed analysis of these pre-forming processes should be conducted in conjunction with hydroforming analyses. Moreover, to obtain accurate results from FEA of hydroforming, FEA of bending and crushing should precede it since they will change thickness and strain distribution of the tube.

(5) An investigation of structural properties of hydroformed parts and their evaluation under actual working conditions should be conducted. This would enhance the use of THF technology as a replacement of stamping and forging processes.

(6) Elastic analysis of tooling in THF should be done to reveal the effect of high internal pressures on the die deflections and part tolerances.

(7) Design and analysis of clamping devices or presses for THF is another area of possible investigation along with development of more appropriate THF hydraulic and control systems.
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APPENDIX A

CLASSIFIED PARTS THAT ARE/CAN BE PRODUCED BY THF PROCESS
Common features:

Figure A. 1: Two of the common features in THF parts: Tee and Y protrusions. Geometrical parameters, which are of importance to the designer, are shown.
Figure A. 2: Crushing and bends are other common features encountered frequently on THF parts. Their important geometrical parameters are indicated.
Figure A. 3: One-dimensional (1D) THF part with double features (Tee protrusions in this case). Geometrical parameters, which are of importance to the designer, are shown.
Figure A. 4: One-dimensional (1D) THF part with double features (Y protrusions in this case). Geometrical parameters, which are of importance to the designer, are shown.
THF parts with 2D spline:

Figure A. 5: Two-dimensional THF part with “ends at an angle to each other”. Detailed views show dimensions on the features.
Figure A. 6: Cross-sections of 2D THF parts with "ends at angle to each other". They carry double feature. Necessary geometrical parameters are shown.
Figure A. 7: Another 2D part with “ends at angle to each other” case. However, features on this part also at an angle to each other. Detailed figures illustrate the side view and the geometrical parameters on the features.
Figure A. 8: Two-dimensional (2D) THF part with "ends on two parallel lines". Detailed views show geometrical parameters on the feature.
Figure A. 9: Two-dimensional (2D) THF parts with "ends on one line". Detailed view illustrates dimensions on the feature.
THF parts with 3D spline:

Figure A. 10: Three-dimensional (3D) part with “ends at angle to each other”. Detailed views show dimensions on features.
Figure A. 11: Three-dimensional (3D) part with “ends on two parallel lines”. Detailed views show dimensions on features.
Figure A. 12: Three-dimensional (3D) part with “ends normal to each other”. Detailed views show dimensions on features.
Figure A. 13: Three-dimensional (3D) part with “ends on one line”. Detailed views show dimensions on features.
Structural THF parts:

Figure A. 14: An example of structural THF part group. Along a long longitudinal axis, varying cross-sections blend each other. Large (L/D) ratio is their main characteristic. In addition, structural parts have a thick wall. Initial tube may not necessarily be round.
APPENDIX B

APPLICATION OF PLASTICITY THEORY IN TUBE HYDROFORMING PROCESS

Theory of plasticity deals with the methods of calculating stresses and strains in a deformed body after all or part of the body has yielded [Johnson, et al., 1973]. For cases where plastic strains are large compared with the elastic strains, it is usually permissible to neglect elastic strains to simplify the problem. The general solution between stress and strain must contain the followings:

- Elastic stress-strain relations, \((\sigma^e - \varepsilon^e)\)
- Yield criterion indicating the onset of plastic flow, \((\sigma : \sigma_f)\)
- Plastic stress-strain relations or stress-strain increment relations, \((\sigma^p - \varepsilon^p)\).
Fundamental Stress-Strain Relations

Mathematical theories of plasticity can be divided into two types: (a) Deformation theories, and (b) Flow theories. Deformation theory states the relations between stress and strain. It utilizes an averaging process over the entire deformation history and relates the total plastic strain to the final stress. However, it should be known that this case is valid only under proportional loading.

Flow theory defines the stress-strain rate relations. It considers a succession of infinitesimal increments of distortion in which the instantaneous stress is related to the increment of the strain rate. Flow theory describes large plastic deformations in a better way than Deformation theory does.

General assumptions for plasticity theory can be presented as follows:

- Material is continuous and isotropic
- Principal axes of plastic stress and strain are assumed to coincide at all times
- Time (dynamic) effects are neglected
- Volume remains constant

\[ \varepsilon_1 + \varepsilon_2 + \varepsilon_3 = 0 \]  \hspace{2cm} (B-1)

- Strain increment is proportional to the total strain
\[ \frac{d\varepsilon_i}{\varepsilon_i} = \frac{d\varepsilon_2}{\varepsilon_2} = \frac{d\varepsilon_3}{\varepsilon_3} \]  \hspace{1cm} (B-2)

Stress-Strain \((\sigma - \varepsilon)\) relations or flow rules) during plastic deformation has been described as follows [Hosford. et al., 1983]:

\[ \frac{d\varepsilon_i}{\sigma_i} = \frac{d\varepsilon_2}{\sigma_2} = \frac{d\varepsilon_3}{\sigma_3} = d\lambda \]  \hspace{1cm} (B-3)

where \((d\lambda)\) is a constant called plastic compliance. Its magnitude depends on the amount of deformation. \(\sigma_i\) is called as deviatoric stress. It can be stated as follows:

\[ \sigma_i' = \sigma_i - \sigma_m \]  \hspace{1cm} (B-4)

where mean (hydrostatic) stress is given as

\[ \sigma_m = \frac{(\sigma_1 + \sigma_2 + \sigma_3)}{3} \]  \hspace{1cm} (B-5)

Equation (B-3) is known as Levy-Mises Flow Rule. In order this equation to hold, (a) elastic deformation is neglected as stated before. Therefore, these relations can not be used to obtain information about springback or residual stresses, (b) there is no change in volume of the component. In turn, this means that the condition of incompressibility is valid. Incompressibility can be stated as follows:

In terms of incremental principal strains
\[ d\varepsilon_1 + d\varepsilon_2 + d\varepsilon_3 = 0 \]  \hspace{1cm} (B-6a)

In terms of total principal strains

\[ \varepsilon_1 + \varepsilon_2 + \varepsilon_3 = 0 \]  \hspace{1cm} (B-6b)

**Yield Criterion and Flow Rule**

A yield criterion is a given expression of the states of stress that would result in the onset of plastic deformation or yielding [Hosford, et al., 1983]. It is necessary to determine the stress at which yielding will occur in a material in metalworking processes. This stress value would indicate the point where (a) plastic or permanent deformation begins in a formed material, and (b) tooling or dies fail.

Von Mises criterion indicates that yielding will occur when some value of the root-mean shear stress reaches a constant. For tri-axial state of stress, Von Mises yield criterion can be expressed as:

In terms of normal and shear stresses

\[ \bar{\sigma} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)} \]  \hspace{1cm} (B-7)
And, in terms of principal stresses

\[ \bar{\sigma} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \quad (B-8) \]

Note that,

\[ \bar{\sigma} = \sigma_{yp} = \sigma_f = 2K \quad (B-9) \]

Where

\( \bar{\sigma} \) = Effective stress as a result of applied state of stress

\( \sigma_f \) = Flow stress of the material

\( \sigma_{yp} \) = Yield strength of the material

\( K \) = Shear yield strength of the material

Combining Equations (B-3), (B-4) and (B-5), Levy-Mises flow rule can be described in another form as follows:

\[ d\varepsilon_1 = \frac{2}{3} d\lambda \left[ \sigma_1 - \frac{1}{2} (\sigma_2 + \sigma_3) \right] \quad (B-10a) \]

\[ d\varepsilon_2 = \frac{2}{3} d\lambda \left[ \sigma_2 - \frac{1}{2} (\sigma_1 + \sigma_3) \right] \quad (B-10b) \]

\[ d\varepsilon_3 = \frac{2}{3} d\lambda \left[ \sigma_3 - \frac{1}{2} (\sigma_1 + \sigma_2) \right] \quad (B-10c) \]
Or, it can be also stated as another form of Levy-Mises equation as follows:

\[
\frac{d\varepsilon_1 - d\varepsilon_2}{\sigma_1 - \sigma_2} = \frac{d\varepsilon_2 - d\varepsilon_1}{\sigma_2 - \sigma_3} = \frac{d\varepsilon_3 - d\varepsilon_1}{\sigma_3 - \sigma_1} = d\lambda
\]  

(B-11)

It is useful to define effective stress (\(\bar{\sigma}\)) and effective strain (\(\bar{\varepsilon}\)) terms in order to extend the material behavior determined for simple cases to complex loading situations. This would also facilitate the understanding of yield criterion. Both, \(\bar{\sigma}\) and \(\bar{\varepsilon}\), are functions of the applied state of stress on the component. For instance, if the magnitude of the \(\bar{\sigma}\) reaches a critical value, the applied state of stress will cause the onset of yielding.

Effective stress can be defined in terms of the yield locus, which is the locus of all possible combinations of states of stress that will initiate yielding or plastic flow in a component. According to Von Mises yield criterion, effective stress (\(\bar{\sigma}\)) will be as follows [Hosford, et al., 1983]:

\[
\bar{\sigma} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}
\]  

(B-12)

or, in terms of deviatoric stress components, it will be as follows:

\[
\bar{\sigma} = \sqrt{\frac{3}{2} \left[ \left( \sigma_1' \right)^2 + \left( \sigma_2' \right)^2 + \left( \sigma_3' \right)^2 \right]}
\]  

(B-13)
after combining Equations (B-4), (B-5), and (B-12).

Similarly, effective strain would be:

$$\bar{\varepsilon} = \sqrt{\frac{2}{3}} \left( \varepsilon_1^2 + \varepsilon_2^2 + \varepsilon_3^2 \right)$$  \hspace{1cm} (B-14)

Or, in terms of incremental strain:

$$d\bar{\varepsilon} = \sqrt{\frac{2}{3}} \left( d\varepsilon_1^2 + d\varepsilon_2^2 + d\varepsilon_3^2 \right)$$  \hspace{1cm} (B-15)

From Equation (B-6b),

$$\varepsilon_3 = -\varepsilon_1 - \varepsilon_2$$

Substituting into Equation (B-14),

$$\bar{\varepsilon} = \sqrt{\frac{2}{3}} \left[ (\varepsilon_1)^2 + (\varepsilon_2)^2 + (-\varepsilon_1 - \varepsilon_2)^2 \right]$$

Then, finally:

$$\bar{\varepsilon} = \frac{2}{\sqrt{3}} \sqrt{\varepsilon_1^2 + \varepsilon_1 \varepsilon_2 + \varepsilon_2^2}$$  \hspace{1cm} (B-16)

Letting,

$$\beta = \frac{\varepsilon_2}{\varepsilon_1}$$

$$\bar{\varepsilon} = \frac{2\varepsilon_1}{\sqrt{3}} \sqrt{1 + \beta + \beta^2}$$  \hspace{1cm} (B-17)
Cylindrical Shell (Axisymmetric Loading) Case

In case of axisymmetric loading, a cylindrical shell (e.g. tube) is loaded by axial forces from both ends, and by an internal pressure to expand the cylinder as shown in Figure B.1. If the value of plastic deformation and the flow curve of the material are known, stresses and strains in the component can be calculated using the plasticity equations described above.

Since there is no shear stresses, axial \( (\sigma_z & \varepsilon_z) \), tangential \( (\sigma_\theta & \varepsilon_\theta) \), and radial \( (\sigma_r & \varepsilon_r) \) stresses and strains are treated as principal stresses and strains. For thin walled members, bending and shear stresses across the wall thickness is negligible since thickness is so small when compared with radius [Mellor, P.B. 1960]:

\[
\sigma_r = 0 \tag{B-18}
\]

Then, stresses and strains for such components and loading conditions can be derived as follows:

The mean stress \( \sigma_m \) reduces into:

\[
\sigma_m = \frac{\sigma_z + \sigma_\theta}{3} \tag{B-19}
\]

Deviatoric stresses, then, can be written as follows:

\[
\sigma_\theta' = \frac{1}{3}(2\sigma_\theta - \sigma_z) = (\sigma_\theta - \sigma_m) \tag{B-20a}
\]
Figure B.1: Axisymmetric loading case. A cylindrical shell loaded by axial forces and internal pressure.
\[
\sigma' = \frac{1}{3}(2\sigma_z - \sigma_\theta) = (\sigma_z - \sigma_m) 
\]  
(B-20b)

According to Levy-Mises rule, i.e. Equation (B-3)

\[
\frac{d\varepsilon_1}{\sigma_1} = \frac{d\varepsilon_2}{\sigma_2} = \frac{d\varepsilon_3}{\sigma_3} = d\lambda
\]

Then, combining Equation (B-3) and (B-20) we obtain:

\[
\frac{d\varepsilon_z}{2\sigma_z - \sigma_\theta} = \frac{d\varepsilon_\theta}{2\sigma_\theta - \sigma_z} = \frac{-d\varepsilon_r}{\sigma_z + \sigma_\theta} = d\lambda 
\]  
(B-21)

where

\[
\varepsilon_z = \ln \left( \frac{l_z}{l_0} \right) 
\]  
(B-22a)

\[
\varepsilon_\theta = \ln \left( \frac{d_\theta}{d_\theta_0} \right) 
\]  
(B-22b)

\[
\varepsilon_r = \ln \left( \frac{t_r}{t_0} \right) = \varepsilon_z
\]  
(B-22c)

Effective strain for a cylindrical shell can be expressed as follows:

\[
\bar{\varepsilon} = \sqrt[3]{\frac{2}{3} \left( \varepsilon_\theta^2 + \varepsilon_z^2 + \varepsilon_r^2 \right)} 
\]  
(B-23)

Manipulating (B-23), we will obtain:

\[
\bar{\varepsilon} = \frac{2}{\sqrt{3}} (\varepsilon_z) \sqrt{1 + \left( \frac{\varepsilon_\theta}{\varepsilon_z} \right) + \left( \frac{\varepsilon_\theta}{\varepsilon_z} \right)^2} 
\]

Or, in another form:
\[ \bar{\varepsilon} = \frac{2}{\sqrt{3}} (\varepsilon_{\theta}) \sqrt{1 + \left( \frac{\varepsilon_z}{\varepsilon_{\theta}} \right)^2 + \left( \frac{\varepsilon_\phi}{\varepsilon_{\theta}} \right)^2} \]  

(B-24)

Letting

\[ \beta = \frac{\varepsilon_z}{\varepsilon_{\theta}} \]

Proceeding with simplification, effective strain on a cylindrical shell will be explained by either:

\[ \bar{\varepsilon} = \frac{2\varepsilon_z}{\sqrt{3}} \sqrt{1 + \frac{1}{\beta} + \frac{1}{\beta^2}} \]  

(B-25a)

Or,

\[ \bar{\varepsilon} = \frac{2\varepsilon_{\theta}}{\sqrt{3}} \sqrt{1 + \beta + \beta^2} \]  

(B-25b)

Similarly, since Equation (B-12) and (B-18) state the followings:

\[ \bar{\sigma} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \]

\[ \sigma_r = 0 \]

Effective stress on a cylindrical shell will be

\[ \bar{\sigma} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_z - \sigma_\theta)^2 + \sigma_\theta^2 + \sigma_z^2} \]

Proceeding, we will obtain:
\[ \bar{\sigma} = \sqrt{\sigma^2 - \sigma_\theta \sigma_z + \sigma_z^2} \]  \hspace{1cm} (B-26)

Letting

\[ \alpha = \frac{\sigma_z}{\sigma_\theta} \]

And, substituting into (B-26), Effective stress will be described as either:

\[ \bar{\sigma} = \sigma_\theta \sqrt{\alpha^2 - \alpha + 1} \]  \hspace{1cm} (B-27a)

Or,

\[ \bar{\sigma} = \sigma_z \sqrt{\frac{1}{\alpha} + \frac{1}{\alpha^2}} \]  \hspace{1cm} (B-27b)

From Equation (B-21), the following can be obtained:

\[ \frac{d\varepsilon_z}{d\varepsilon_\theta} = \frac{2\sigma_z - \sigma_\theta}{2\sigma_\theta - \sigma_z} \frac{\varepsilon_z}{\varepsilon_\theta} = \beta \]  \hspace{1cm} (B-28)

Substituting (B-28) into (B-25), we will obtain:

\[ \bar{\varepsilon} = \frac{2\varepsilon_\theta}{(2\sigma_\theta - \sigma_z)\sqrt{\sigma^2 - \sigma_\theta \sigma_z + \sigma_z^2}} \sqrt{\sigma^2 - \sigma_\theta \sigma_z + \sigma_z^2} \]

\[ \bar{\varepsilon} = \frac{2\varepsilon_z}{(2\sigma_z - \sigma_\theta)\sqrt{\sigma^2 - \sigma_\theta \sigma_z + \sigma_z^2}} \sqrt{\sigma^2 - \sigma_\theta \sigma_z + \sigma_z^2} \]

Combining with Equation (B-27), strains on a cylindrical shell would be as follows:
\[ \varepsilon_\theta = \frac{\bar{\varepsilon}(2\sigma_\theta - \sigma_z)}{2\bar{\sigma}} \]  \hspace{1cm} (B-29a)

\[ \varepsilon_z = \frac{\bar{\varepsilon}(2\sigma_z - \sigma_\theta)}{2\bar{\sigma}} \]  \hspace{1cm} (B-29b)

Similarly,

\[ \sigma_\theta = \frac{2\bar{\sigma}}{3\bar{\varepsilon}} \cdot (2\varepsilon_\theta + \varepsilon_z) \]  \hspace{1cm} (B-30a)

\[ \sigma_z = \frac{2\bar{\sigma}}{3\bar{\varepsilon}} \cdot (2\varepsilon_z + \varepsilon_\theta) \]  \hspace{1cm} (B-30b)

Make a note of that Effective Stress (\(\bar{\sigma}\)) and Yield Strength (\(\sigma_{yp}\)) or Flow Stress (\(\sigma_f\)) of a material are assumed to be equal in calculations to determine stresses on a component.

\[ \bar{\sigma} = \sigma_{yp} = \sigma_f \]  \hspace{1cm} (B-31)

Equation (B-29) and (B-30) give the basis for the calculations of stresses and strains according to the plasticity theory. For example, if \((\varepsilon_z)\), \((\varepsilon_\theta)\) and \((\bar{\sigma})\) are known, as a result of an experiment for instance, beginning of deformation on a tube being formed can be calculated easily.
According to the derived equations above, Von Mises yield curve can be drawn as in Figure B.2. for a bi-axial loading case, i.e. \((\sigma_r = 0)\). This condition would be obtained by loading a tubular part by axial forces and internally pressurizing it as indicated in Figure B.2. In order to reach maximum stretching (i.e. maximum bulge height) on the tube, wall thickness at the stretching region should be kept constant. This means, in turn, thickness strain ought to be zero. Defining stress and strain ratios as follows:

\[
\beta = \frac{\varepsilon_r}{\varepsilon_{\theta}} \quad \text{(B-32)}
\]

\[
\alpha = \frac{\sigma_r}{\sigma_{\theta}} \quad \text{(B-33)}
\]

It can be obtained that:

\[
\beta = \frac{2\alpha - 1}{2 - \alpha} \quad \text{(B-34)}
\]

Figure B.2 is constructed to present the variation of \((\alpha)\) according to Von Mises yield curve.
Figure B.2: Yielding curve of a component under biaxial loading. In the first quadrant, part is under bi-axial stretching. In the second quadrant, it is compressive in axial direction while tensile in tangential direction.
<table>
<thead>
<tr>
<th>Case #</th>
<th>( \alpha = \frac{\sigma_z}{\sigma_\theta} )</th>
<th>( \beta = \frac{\varepsilon_z}{\varepsilon_\theta} )</th>
<th>( \varepsilon_r = \varepsilon_t )</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-1</td>
<td>-1</td>
<td>0</td>
<td>No thinning</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-(\frac{(1/2)}{2})</td>
<td>(1 - \frac{\varepsilon_\theta}{2})</td>
<td>Buckling or wrinkling</td>
</tr>
<tr>
<td>3</td>
<td>(\frac{(1/2)}{2})</td>
<td>0</td>
<td>-(\varepsilon_\theta)</td>
<td>Thinning</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>1</td>
<td>-(\varepsilon_z + \varepsilon_\theta)</td>
<td>Thinning</td>
</tr>
</tbody>
</table>

Table B. 1: Bi-axial loading conditions. Case 1 indicates the perfect bulging operation situation where there is no thinning. However, this is an impossible and unrealistic case.

As presented in Figure B.2 and Table B.1, at point 1 in second quadrant, stress ratio \(\alpha\) and strains ratio \(\beta\) becomes (-1). That means that axial and tangential components of stress and strain are equal in opposite signs. Consequently, strain in thickness direction \(\varepsilon_r\) becomes zero theoretically as shown below: At Point 1 in second quadrant:

\[
\beta = \frac{\varepsilon_z}{\varepsilon_\theta} = -1
\]

Then,

\[
\varepsilon_\theta = -\varepsilon_z
\]
Since,
\[ \varepsilon_\theta + \varepsilon_z + \varepsilon_r = 0 \]

We obtain:
\[ \varepsilon_r = 0 = \varepsilon_t \]

As a result:
\[ t_1 = t_o \]

Theoretically, there is a bi-axial loading at this point such that thinning is reduced to zero by preventing the local necking. This is an ideal case for sheet metal working processes, where maximum stretching is obtained without fracture. In other words, level of forming, (i.e. stretching in this particular case), can be increased by delaying the onset of plastic instability.

However, it is practically impossible to keep the stress and strain states at this critical point (i.e. point 1 in Figure B.2) throughout the workpiece and during the entire forming period. Stress and strain states will vary locally and during forming process. Thus, practical aim should be to have such state of stress and strain that minimizes the thinning by postponing the necking. In second quadrant of a bi-axial yielding curve, region between point 1 and 2 is suitable to reach this goal.
APPENDIX C

APPLICATION OF MEMBRANE THEORY IN TUBE HYdroFORMING

PROCESS

There are two different theories to analyze the stresses on a shell structure. Membrane theory, which is limited to moment-free structures, can be used to analyze forming processes containing tubular parts with thin walls. Bending theory, on the other hand, includes the effects of bending stresses. It contains a membrane solution with improvements in discontinuity effects. However, it is not as simple as membrane theory in application. Therefore, this section will concentrate on membrane solutions of tubular components. Membrane theory gives in relatively accurate results under the following assumptions:

The ratio of the wall thickness to the radius of curvature is small when compared to unity:

\[
\left( \frac{t}{R} \geq 10 \right)
\]  

(C-1a)
Two-dimensional state of stress prevails in the tube wall. These stresses are in axial and tangential (hoop) directions. As a consequence, radial stress across the tube wall is neglected:

\[
\begin{align*}
(\sigma_z \neq 0) & \quad (\sigma_\theta \neq 0) & \quad (\sigma_r = 0) \quad \text{(C-1b)}
\end{align*}
\]

There is no bending stress in the tube wall.

A surface of revolution, such as in Figure C.1, can be formed by the rotation of a meridian curve \((op)\) about the \(Z\)-axis. As it is indicated, any point on the shell may be located by coordinates \((\theta, z, \text{ and } r)\). An elemental surface \((abef)\) is cut by two meridional sections, \((ab \text{ and } ef)\) and two normal sections, \((ae \text{ and } bf)\). A condition of symmetry exists and only normal stresses act on the sides of the element \((abef)\).

\[\sigma_\theta = \text{Tangential (hoop) stress (stress along a parallel circle)}\]

\[\sigma_z = \text{Longitudinal (meridional) stress (stress acting in the meridional direction)}\]

\[t = \text{Thickness of the shell structure (i.e. tubular part)}\]

\[dv = \text{Element dimension in the longitudinal direction (face } ab \text{ and } ef)\]

\[ds = \text{Element dimension in the tangential (hoop) direction (face } ae \text{ and } bf)\]

\[\rho_1 = \text{Tangential radius of curvature}\]

\[\rho_2 = \text{Longitudinal radius of curvature}\]

\[P_i = \text{Internal pressure}\]
Figure C. 1: Diagram for analysis of a tubular part. An axisymmetric element (abef) taken for detailed analysis.
Figure C. 2: Stresses on an element (abef) taken from a cylindrical shell as in Figure C.1. Axial forces and internal pressure load this tubular part.
Referring to Figure C.2 and Figure C.3, force acting on the sides of the element \((abef)\) can be written as, (i.e. tangential force):

\[ \sigma_\phi (\rho_2 d\phi)(t) \quad \text{(C-2)} \]

Forces on top and bottom of the element \((abef)\) is expressed as follows, (i.e. meridional forces):

\[(\sigma_z + d\sigma_z) (\rho_i d\theta)(t + dt) \quad \text{(C-3)}\]

and

\[\sigma_z (\rho_i d\theta)(t) \quad \text{(C-4)}\]

respectively.

Force acting on the plane of the element \((abef)\) is also found as:

\[ P_i (\rho_i d\theta)(\rho_2 d\phi) \quad \text{(C-5)} \]

From Figure C.2 and Figure C.3, it is observed that

\[ dv = \rho_2 d\phi \quad \text{(C-6a)} \]

\[ ds = \rho_i d\theta \quad \text{(C-6b)} \]

\[ dr = \rho_2 d\phi \cos \phi \quad \text{(C-6c)} \]

\[ r = \rho_i \sin \phi \quad \text{(C-6d)} \]
Figure C. 3: Two different views of element (abef). (a) Side view, (b) top view.
Figure C.4: Forces on element (abef). (a) and (b) show all the forces acting the element.
Force equilibrium along the surface normal (along the direction where internal pressure acts) would result in the following after substitutions and simplifications, see Figure C.4 for details:

\[ p_1(\rho_2 d\phi)(rd\phi) - [\sigma_z(\rho_2 d\phi)(d\phi)] - [\sigma_\theta(\rho_2 d\phi)(d\phi)(\sin \phi)] = 0 \]  

(C-7)

After cancellations and simplifications, it reduces to

\[ \frac{p_1}{t} = \left[ \sigma_\theta \left( \frac{\sin \phi}{r} \right) + \frac{\sigma_z}{\rho_2} \right] \]

Finally, after substitution of (C-6), we can obtain the following

\[ \frac{p_1}{t} = \left[ \frac{\sigma_\theta + \sigma_z}{\rho_1 \rho_2} \right] \]  

(C-8)

Where,

- \( P_1 \): Internal pressure applied to the walls of the tubular part, MPa
- \( \sigma_\theta \): Tangential (hoop) stress developed as a result of loading on the part, MPa
- \( \sigma_z \): Longitudinal (meridional) stress, MPa
- \( \rho_1 \): Radius of curvature in tangential direction, mm
- \( \rho_2 \): Radius of curvature in longitudinal direction, mm

For a cylindrical tube, radius of curvature in longitudinal direction is infinite, and radius in tangential direction equals to the outer radius of the tube:
\[ \rho_1 = r \quad \rho_2 = \infty \]

Then, internal pressure can be calculated as follows:

\[ p_i = \frac{\sigma_\phi(t)}{r} \]  \hspace{1cm} (C-9)

Force equilibrium in Meridional (longitudinal) direction can be written as follows after necessary simplifications and substitution, see Figure C.4 for further details:

\[ [(\sigma_z + d\sigma_z)(r + dr)(t + dt)(d\theta)] - [\sigma_z(r)(d\theta)(t)] - [\sigma_\phi(\rho_2 d\phi)(d\theta)(\cos \phi)(t)] = 0 \]

\[ (C-10) \]

By proceeding further, and cancellation of some terms each other, we obtain:

\[ (d\sigma_z)r + (\sigma_z)dr - (\sigma_\phi)dr = 0 \]

Then, it reduces to the followings

\[ \frac{d\sigma_z}{dr} r = \sigma_\phi - \sigma_z \]

\[ \frac{d\sigma_z}{dr} - \frac{\sigma_\phi - \sigma_z}{r} = 0 \] \hspace{1cm} (C-11)

With Equation (C-11) and given boundary conditions, stresses on an axisymmetric component can be obtained easily.

Equilibrium of forces in longitudinal direction would give the following:

\[ 2\pi rt \sigma_z = \pi Pr^2 + F_\phi \] \hspace{1cm} (C-12)
APPENDIX D

APPLICATION OF THICK-WALLED CYLINDER APPROACH IN TUBE HYDROFORMING PROCESS

Another simplification in the analysis of metal forming processes involving tubular parts would be thick-walled shells or cylinders approach. In this case, any cylindrical part whose radius to thickness ratio is less than 10, \( \left( \frac{r}{t} \geq 10 \right) \), can be treated as a thick-walled cylinder.

In case of a thick-walled cylinder subjected to uniform internal pressure, the deformation is symmetrical about the axial (z) axis as illustrated in Figure D.1. The static equilibrium equations in cylindrical coordinates can be written as follows:

\[
\frac{\partial \sigma_r}{\partial r} + \frac{1}{r} \frac{\partial \tau_{r\theta}}{\partial \theta} + \frac{\partial \Sigma_z}{\partial z} + \frac{\sigma_r - \sigma_\theta}{r} = 0
\]

\[
\frac{\partial \tau_{r\theta}}{\partial r} + \frac{1}{r} \frac{\partial \tau_{\theta\theta}}{\partial \theta} + \frac{\partial \tau_{r\theta}}{\partial z} + \frac{\tau_{r\theta}}{r} = 0
\]  

(D-1a)  

(D-1b)
Figure D. 1: A thick-walled cylinder subjected to internal pressure.
\[
\frac{\partial \tau_\theta}{\partial r} + \frac{1}{r} \frac{\partial \tau_\theta}{\partial \theta} + \frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_\zeta}{\partial z} = 0 \quad \text{(D-1c)}
\]

For the axial symmetry case,

\[
\tau_\theta = \tau_r = 0
\]

Then, Equation (D-1) reduces to the following:

\[
\frac{\partial \sigma_r}{\partial r} + \frac{\tau_\zeta}{r} + \frac{\sigma_r - \sigma_\theta}{r} = 0 \quad \text{(D-2a)}
\]

\[
\frac{\partial \tau_\zeta}{\partial r} + \frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_\zeta}{\partial z} = 0 \quad \text{(D-2b)}
\]

For an axisymmetrically loaded tubular part, the equilibrium of forces in radial direction can be expressed as:

\[
\frac{d\sigma_r}{dr} + \frac{\sigma_r - \sigma_\theta}{r} = 0 \quad \text{(D-3)}
\]

For plain strain situation where ends of the cylinder are constraint:

\[
(\varepsilon_z = 0)
\]

\[
(d\varepsilon_z = 0)
\]

As a result:

\[
\sigma_z = \frac{\sigma_r + \sigma_\theta}{2} \quad \text{(D-4)}
\]

Also, Yield criterion gives that:

\[
\sigma_\theta - \sigma_r = k\sigma_{yp} \quad \text{(D-5)}
\]
where

\[ k = \frac{1}{2} \] for Tresca criterion

\[ k = \frac{2}{\sqrt{3}} \] for Von Mises criterion

Then, substituting into (D-3),

\[
\frac{d\sigma_r}{dr} + \frac{k\sigma_{yp}}{r} = 0
\]

\[
d\sigma_r = \frac{k}{r} \sigma_{yp} dr
\]

After integration, it will be

\[
\sigma_r = (k\sigma_{yp} \ln r) + c \quad \text{(D-6)}
\]

Using the boundary condition at the inner surface of the cylinder, when

\[ r = r_a \]

\[ \sigma_r = -P_i \]

Then, integration constant will be

\[ c = -P_i - (k\sigma_{yp} \ln r_a) \]

Substitution of c into (D-6) would result in the following:

\[
\sigma_r = -P_i + \left[ k\sigma_{yp} \ln \frac{r}{r_a} \right] \quad \text{(D-7)}
\]

Boundary condition at

\[ r = r_b \]

\[ \sigma_r = 0 \]
Results in

\[ P_i = \sigma_{yp} \left[ k \ln \frac{r_h}{r_w} \right] \]  

(D-8)

After substitution of (D-8) into (D-7), we obtain:

\[ \sigma_r = k\sigma_{yp} \left[ -\ln \left( \frac{r_o}{r} \right) \right] \]  

(D-9)

Using Equations (D-5) and (D-6), and proceeding with (D-8) and (D-9), we obtain:

\[ \sigma_\theta = k\sigma_{yp} \left[ 1 - \ln \left( \frac{r_h}{r} \right) \right] \]  

(D-10)

\[ \sigma_z = k\sigma_{yp} \left[ \frac{1}{2} - \ln \left( \frac{r_h}{r} \right) \right] \]  

(D-11)

As a conclusion, with the Equations (D-9), (D-10) and (D-11), state of stress in thick-walled tube can be obtained. In addition, Equation (D-8) gives the necessary internal pressure required to start deformation.
APPENDIX E

DESIGN RULES FOR TUBE HYDROFORMING TECHNOLOGY

Most of the rules gathered in this section are collected from different sources available in the literature. Some of the rules are developed via planned FEA by the author. Some of them are improved after FEA. More convenience of the readers, rules are grouped into relevant categories.

E.1 Overall rules and observations

1. The degree of difficulty of the hydroforming is defined by the following overall factors:
   - Differences in the cross-sections of the part; the size of the largest part cross-section should not be higher than double the size of the smallest part cross-section.
2. Influence of the initial cross-section of the tube on the subsequent process steps can be briefly outlined as follows [Dohmann, 1997]:

- The bent tube must be able to be easily inserted into the tool, i.e. the tube outer diameter must be smaller by the insertion clearance than the smallest die width. In this case, the maximum ratio of expansion, based on the largest component circumference is about 2.5. The tube wall thickness must be selected, so that a minimum wall thickness is achieved in the area of the greatest expansion. This results in a maximum wall thickness difference in component longitudinal direction; intermediate annealing may also be necessary.
- The tube cross-section corresponds to the smallest component cross-section. This enables forming of the smallest component cross-section without wrinkles; the degree of expansion referenced to the largest circumference reduces to the value 2.0. The wall thickness difference, viewed in component longitudinal direction is less than in case 1. However, depending on the cross-section geometry, the use of an insertion aid may be necessary, i.e. in the form of an insertion slant on the tool.

3. The respective differences in circumference of the cross section in the cone, in the pipe, as well as in the die have to be chosen low enough to guarantee that the permanent elongation limit remains clearly below the ultimate elongation of the material. The heavy deformation of the contour, has to take into account the resulting extra longitudinal elongation. By modifying the cross-sections, the differences in the circumference could be reduced [Schaefer, 1996]
4. Avoid tiny, intricate regions and features on the part to be hydroformed. Such features, for instance corner radius, cause high-pressure requirements.

5. Establish smooth transitions between regions with different cross-sections. Sharp transitions may prevent metal flow and lead to fracture [Schaefer, 1996]
6. For complex and large parts, first, a conduct two-dimensional cross-sectional analysis. Assume plane-strain conditions. In such analysis, overall cross-expansions up to 40-50% can be allowed.

7. For features (such as protrusions and bulges) far away from the edges of the part, plane-strain assumptions can be applied since axial feeding from the ends of the tube would not be effective.

8. For cases such as above, if high expansion ratios are required, additional process parameters (like forces, feeding and restrictions) can be used to increase the degree of deformation. For instance, counter punches are usually used to achieve high protrusions on Tee-shaped parts on a long axis or where no axial feeding is available.

9. Large differences between diameter or height of different regions on the final part (i.e. more than 80-90% difference) will indicate a necessity of a pre-forming operation such as crushing.

10. Part geometry on a plane or in space definitely indicates the necessity of a pre-forming operation such as bending [Koç, 1998]
11. In order to avoid possibility of severe buckling or wrinkling, unsupported length of the tube (i.e. where tube walls are not in contact with die surface) should be less than or around two to three times of the tube diameter [Schäfer, 1996] [Dohmann, 1997]

\[ \textit{Unsupported tube length} < 2-3 \, D_o \]

12. Tubular materials can be easily moved into the distant zones, to form features like protrusions, through bent regions in THF process if the bent radius is at least five times of the initial tube diameter [Schäfer, 1996]
13. Limits of attainable protrusion and bulge heights are case-dependent. Effect of axial feeding is pronounced on large feature expansions. However, some overall guidelines can be established for approximate estimations. They are as follows [Boehm, 1997] [Klaas, 1987] [Schaefer, 1996]:

For a simple axisymmetric bulge part:

Expansion diameter, $D_1 = (0.5 \text{ to } 1) \ D_o$

For a Tee-shaped part:

Protrusion height, $H_p = (0.4 \text{ to } 0.75) \ D_o$
For a protrusion on a bend axis

Protrusion height, \( H_p = (0.3 \text{ to } 0.7) \ D_o \)

For a protrusion on a bend axis form both sides:

Protrusion height, \( H_p = (0.1 \text{ to } 0.2) \ D_o \)

14. Limits and extent of overall geometrical relations on simple features can be described as follows [Boehm, 1997] [Klaas, 1987] [Schaefer, 1996]:

\[
\begin{align*}
  d_o &= \text{Initial tube diameter} \\
  t_o &= \text{Initial tube thickness} = 1 \text{ mm} < t_o < 0.18 \ d_o \\
  w &= \text{Bulge width} = \text{Maximum } 2d_o \\
  d_f &= \text{Final diameter of bulge} = \text{Maximum } 1.9 \ d_o
\end{align*}
\]
15. Required internal pressure is a function of smallest feature size (such as corner radius, \( R_c \)), flow stress of the material, the wall thickness and diameter of the tube:

\[
P_i = f \left( \bar{\sigma}, t_o, \frac{1}{D_o}, \frac{1}{R_c} \right)
\]

16. Pressure required to start deformation of the tubular material is a function of Yield strength of the material (\( \sigma_y \)), wall thickness (\( t_0 \)), tube diameter (\( D_0 \)), and it can be approximately calculated as follows:

\[
P_{yield} = \frac{2\sigma_y t_o}{D_0 - t_o}
\]

17. Pressure required to achieve maximum expansion before bursting is a function of Ultimate tensile strength of the material (\( \sigma_{ult} \)), wall thickness (\( t_0 \)), tube diameter (\( D_0 \)), and it can be approximately calculated as follows;
\[ P_{\text{burst}} = \frac{2\sigma_{\text{UTS}} t_0}{D_0 - t_0} \]

18. Maximum internal pressure required to form a intricate feature on a tubular part (such as corner radius) is dictated mainly by the size of the feature (\(R_c\)), and flow stress of the material. It can be approximately calculated as follows:

\[ (P_i)_{\text{max}} = \frac{2}{\sqrt{3}} \sigma \ln \left[ \frac{R_c}{R_c - t} \right] \quad R_c \geq 3t_0 \]

19. Corner radii with three times of the initial tube thickness can be formed easily with internal pressure values around 2000 bars (200 MPa). As the corner radius gets smaller, internal pressure required to form will increase.

20. Basic guidelines regarding smallest corner radius and required maximum internal pressure can be illustrated by charts obtained through experiments [Mason, 1997] [Boehm, 97], analytical calculations, FEA results [Koç, 1998]:

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For low carbon steels such as SAE 1010, AKDQ 1008, St 37, St 44, St 52, etc. whose yield strength ranges between 230-300 MPa:

\[ R_c > (3.5 \text{ to } 4.5) \, t_o \]
- For stainless steels such as SS 304, SS 409, X5CrNi1810, etc. whose Yield strength ranges between 360-350 MPa:

\[ R_c > (3 \text{ to } 4) t_0 \]
• For aluminum alloys such as A6061, Al99.5, A99.8, AlMgSi, etc. whose Yield strength ranges between 130-165 MPa:

\[ R_c > (1 \text{ to } 1.5) t_o \]

![Graph showing corner radius to initial thickness ratio against internal pressure]

• For copper alloys such as CuZn05, CuZn37, etc. whose Yield strength ranges between 130-180 MPa:

\[ R_c > (1 \text{ to } 1.5) t_o \]

21. Total axial force required to be delivered by punch in a typical THF process can be expressed in terms of flow stress of the tubular material, internal pressure, size of the tube, friction conditions and amount of axial feeding as follows:
\[ F_a = f(\bar{\sigma}, P_i, d_o, t_o, \mu, A_f) \]

- Total axial force can be expressed in terms of separate components as follows:

\[ F_a = F_{so} + F_u + F_p + F_f \]

- \( F_{so} \) : Initial sealing force
- \( F_u \) : Deformation Force
- \( F_p \) : Reaction Punch Force
- \( F_f \) : Friction Force

- Initial sealing force can be approximately calculated as:

\[ F_{so} = (0.3 - 0.6)\left[\pi(d_o - t_o)\sigma_y t_o\right] \]

\( t_o \): initial tube thickness, \( D_o \): initial tube outside diameter

- Reaction punch force is the force required to be applied by punch to counter act the \( (P_i) \) inside the tube and it can be calculated as follows:

\[ F_p = \frac{\pi(D_o - 2t_o)^2}{4} P_i \]

\( t_o \): initial tube thickness, \( D_o \): initial tube outside diameter, \( P_i \): Internal pressure
- Friction force is the force required to overcome the friction between tube walls and die and it can be calculated as follows:

\[ F_f = \pi D_0 l \mu P_i \]

\( \mu \): friction coefficient \( P_i \): Internal Pressure, \( D_0 \): Tube diameter, \( l \): Axial feeding

- Deformation force is function of \( t_0 \), initial tube thickness; \( D_0 \), initial tube outside diameter; \( \sigma \), flow stress and can be calculated as follows;

\[ F_u = \pi t_0 (D_0 - t_0) \sigma \]

\( t_0 \): initial tube thickness, \( D_0 \): initial tube outside diameter, \( \sigma \): Flow Stress
E.3 Guidelines about geometrical parameters

22. Effect of distance between bulge region and tube edge on the attainable bulge height (i.e. radius) for a simple axisymmetric bulge part can be illustrated as follows:

Attainable bulge height (radius) decreases as the distance between bulge region and edges increases. This is due to the fact that axial feeding can not compensate the thinning at the expansion region due to friction effects. \( L_{pe} \) is dominant factor affecting bulge height compared to other geometrical parameters.
23. Effect of bulge width on the attainable bulge height (i.e. radius) for a simple axisymmetric bulge part can be illustrated as follows:

![Diagram of bulge width and radius](image)

![Graph showing expansion ratio vs. (w / Do)](image)

Attainable bulge height (radius) increases with larger bulge width. Rate of increase is not as much as in case of distance between bulge and edges.
24. Effect of fillet radius on the attainable bulge height (i.e. radius) for a simple axisymmetric bulge part can be illustrated as follows:

![Diagram showing effect of fillet radius on attainable bulge height]

Attainable bulge height (radius) increases with larger fillet radius. Rate of increase is not as much as in case of distance between bulge and edges.
25. Effect of fillet angle on the attainable bulge height (i.e. radius) for a simple axisymmetric bulge part can be illustrated as follows:

Attainable bulge height does not vary much with variation of fillet angle.
26. Effect of geometrical parameters such as distance between protrusion region and tube edge \( (L_{pe}) \), fillet radius \( (R_e) \), and protrusion diameter \( (D_p) \) on the attainable protrusion height for a Tee-shaped part can be illustrated as follows:
E.4 Guidelines regarding hole punching

27. The various types of holes can be produced using the tube hydroforming method. The number of holes is only restricted by the mechanical possibilities within the die. For the transition radii, the rules of the cutting geometry are applied. [Schaefer, 1996]

<table>
<thead>
<tr>
<th>Type of Holes</th>
<th>With sharp edges</th>
<th>inserts</th>
</tr>
</thead>
<tbody>
<tr>
<td>Round</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Rectangular</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Oblong</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Multiple corner</td>
<td>X</td>
<td></td>
</tr>
</tbody>
</table>
28. In order to achieve a larger bearing surface for attachments, **hole embossing** can be performed. The height of the embossing (h) should not exceed 2.5 x the plate thickness (t) [Schaefer, 1996]

![Diagram of embossing types](image)

29. The optical appearance of a hole is determined by various factors.

- Internal Pressure
- Plate Thickness
- Material Tensile Strength
- Type of Hole
- Geometry of the Cutting Edge
- The decisive factors are the internal pressure and the plate thickness. Given the fact that for cutting, the high pressure fluid is utilized instead of a cutting tip, the intensity of the pressure is decisive for the optical appearance of the cut edge. The height of the collapsed area at the cut edge can be regarded as a quality feature.
- The ratio of hole tolerance to punch diameter (width, length) are normally in the range of between 0 and 0.2 mm. [Schaefer, 1996]

<table>
<thead>
<tr>
<th>Plate thickness (t)</th>
<th>Collapsing (h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>0.3</td>
</tr>
<tr>
<td>2.4</td>
<td>0.5</td>
</tr>
</tbody>
</table>

D: Hole diameter, 6 mm,
Dₖ: Diameter of collapsed area,
h: Height of inset,
Pᵢ: Internal pressure, 1500 bar
r: Radius of collapsed area,
t: Plate thickness