Roll Forming - a study on machine deflection by means of experimental analysis and numerical developments

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“We do not know a truth without knowing its cause.”

Aristotle
Abstract

Process control of roll forming is of paramount importance in today’s highly competitive market. This is due to the high productivity of the process (sheet speeds are traditionally in the range of 5-150 m/min), which leads to high costs due to downtime or due to part defects. Process setup and control, especially in the case of tight tolerance profiles, is key to stable and trouble-free manufacturing. In this context, the deflection of the roll stands during the forming process was investigated, both experimentally and numerically. This phenomenon happens due to the forming forces occurring when the sheet is being bent. This can lead to higher springback angles, part distortions and accelerated tool wear if not accounted for.

A simulation model of the deflection of roll forming stands was developed, in order to investigate both the magnitude and the effects of the increase in roll gap due to deflection on the final profile and on the simulation data collected. Furthermore, an experimental roll forming line was designed and built, with which experiments were carried out to evaluate the machine behaviour during roll forming and subsequently validate the data from the numerical simulations of the process. Data collected from the experimental line included the forming load on the top roll and the displacement of different components during the process. An investigation was performed on the influence of the deflection modelling on the overall computational time and simulation stability. A simulation model is proposed which combines deflection calculation with position controlled fixed roll surfaces. The proposed model can, in certain cases, improve simulation stability and lower computation time while, at the same time, providing a better approximation to experimental conditions.

It was concluded that simulations with deformable tooling considerably improve the prediction of the forming loads. As the calculated force remains approximately constant with different machine stiffnesses, modelling deflection in a simulation even when the machine stiffness is not accurately known can still be beneficial. A good match with experimental data was achieved and the numerical model was proven to be stable, further proving that the developed model is a step forward in roll forming simulation. The data collected were not enough for a conclusive evaluation of the effect of roll deflection in profile accuracy, but theoretical investigations and roll forming experience indicate that modelling of deflection in industrial roll forming simulations can lead to a better prediction of defects, and thus appropriate actions can be taken to correct these defects.
Resumo

O controlo do processo de perfilagem é extremamente importante no contexto de elevada competitividade do mercado actual. Isto deve-se à elevada produtividade do processo (a chapa tem normalmente uma velocidade de entre 5 e 150 m/min) o que leva a que os custos associados com paragens de produção ou com peças defeituosas sejam extremamente elevados. A afinação e controlo do processo são cruciais para um fabrico estável e sem problemas, especialmente no caso de perfis com tolerâncias apertadas. Neste contexto, a deformação de stands de perfilagem foi investigada, tanto em termos experimentais como em termos de simulação numérica. Este fenómeno acontece devido às forças de reação que a chapa exerce nos rolos enquanto está a ser dobrada. Isto pode levar a ângulos de retorno elástico mais elevados do que o previsto, distorções no perfil e desgaste acelerado dos rolos.

Foi desenvolvido um modelo de simulação da deformação de stands de perfilagem de forma a investigar tanto a magnitude como os efeitos do aumento da distância entre rolos devido à deformação do stand no perfil final. Foi projectada e construída uma linha experimental de perfilagem com a qual foram realizados ensaios de perfilagem experimentais com a finalidade de avaliar o comportamento da máquina de perfilagem durante o processo, e posteriormente validar os dados da simulação numérica do processo. Os dados recolhidos na linha experimental foram a força exercida pela chapa no rolo superior e o deslocamento de vários componentes da máquina durante o processo. Foi feita uma investigação acerca da influência da modelação numérica da deformação da máquina no tempo total de computação e estabilidade da simulação. É proposto um modelo de simulação que combina o cálculo da deformação da máquina com superfícies dos rolos fixas, o que pode levar, em certos casos, a uma melhoria da estabilidade da simulação e a uma redução do tempo de simulação enquanto, ao mesmo tempo, proporciona uma melhor aproximação das condições reais do processo.

Concluiu-se que as simulações com movimento dos rolos melhoram consideravelmente o cálculo das forças na simulação e que a força calculada se mantém aproximadamente constante para diferentes valores de rigidez na simulação, o que torna a modelação de deflexão em simulação numérica de perfilagem interessante mesmo quando não há dados precisos disponíveis sobre a rigidez da máquina. Obteve-se boa concordância entre os resultados da simulação e os resultados experimentais. Adicionalmente, o modelo de simulação numérica revelou-se estável, comprovando que a inclusão de movimento dos rolos no modelo numérico é um passo em frente na simulação de perfilagem. Os resultados recolhidos não foram suficientes para efectuar uma avaliação conclusiva do efeito do movimento dos rolos na precisão dimensional do perfil, mas investigações teóricas e experiência de perfilagem indicam que a inclusão de deformação da máquina em simulações de perfilagem industriais pode levar a uma melhor previsão de defeitos de fabrico e das ações necessárias para os corrigir.
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1 Introduction

This dissertation project was developed in cooperation with data M Sheet Metal Solutions GmbH, as a part of a 6 month internship in this company. The following chapter sets out to introduce the company, the motivation for the project and the project goals, as well as the approach for the project and the structure for this dissertation.

1.1 data M Sheet Metal Solutions

The company, data M Sheet Metal Solutions, is located in Oberlaindern, about 40 km south of Munich, in Germany. It was founded in 1987, and has always focused on the development of design and simulation tools for sheet metal forming. Their first steps were taken towards developing design software based on HP-UX ME10 and AutoCAD® which simplified the design process of tools for roll forming lines (COPRA® RF, Figure 2). Later, the product range was expanded to providing simulation solutions, firstly using a finite difference method and later (from 2000) using the finite element method (COPRA® FEA RF, Figure 3). The finite element analysis was developed using the software MSC Marc/Mentat, which led to the establishment of an OEM partnership between data M and MSC (data M SMS 2012).

Figure 1 - Logo of data M Sheet Metal Solutions.
data M’s mission is nowadays to develop knowledge in all areas of roll forming technology, from process design and development, conception of roll forming lines, prototyping, to process control, defect analysis and troubleshooting. There is also a focus on user training and on the development of new roll forming applications, such as flexible roll forming (Abee, Sedlmaier, and Stephenson 2010).
1.2 Research and project context

The roll forming process, as will be established in subsequent chapters, is a high volume manufacturing process where downtime is very costly. As such, there has been a constant focus on improving the process variables in order to achieve low defect rates and predict possible problems before they appear; the final target is to have a robust process with good quality control. In this context, finite element analysis plays a major role, as it allows for the simulation of complete roll forming manufacturing lines where defects can be detected and eliminated even before the tooling is manufactured (Abee, Sedlmaier, and Stephenson 2010).

The problem tackled in this project was how to include deformable tooling (forming stands) in roll forming FEA (finite element analysis) models. This is an important innovation in roll forming simulation, and a direction that data M wants to explore in depth in the future. Analysing the forming forces during roll forming and their interaction with the various machine components is key to more precise simulations and to better detection of manufacturing defects during the design stage, leading to possible improvements which would be impossible in later development stages.

1.3 Project goals

The ultimate goal for the project was to develop an FEA model of the roll forming process which includes a representation of deformable tooling. Moreover, it was important to develop a method for the determination of the machine stiffness, so that users of the FEA model can model their own lines accurately resorting to a prescribed set of instructions and tools. In order to achieve these goals, an experimental roll forming line was developed during the course of this project, including several measurement devices in order to better understand the behaviour of its components during the forming. The strategy to achieve a final usable FEA model included the development of FEA models with different degrees of freedom for the rolls, in order to understand what the critical variables were. This way, a final model could be developed, striking a balance between precision and calculation time.

1.4 Experimental and numerical approach

In order to validate and improve the FEA model, an experimental line was designed. This line consisted of a single roll forming stand with entry and exit guiding for the metal strip. A roll set for forming a U-channel profile was used. A single roll stand was used in order to isolate and understand the interaction of the strip with the stand, eliminating the effects stemming from the interaction with the stands adjacent to it. A high strength steel material (DP800 cold rolled steel) was used in order to ensure high forming forces and because advanced high strength steels and ultra-high strength steels are being processed by roll forming with ever increasing success, for example for producing automotive chassis components (Lindgren 2007). Lundberg and Melander (2008) found that in terms of the finite element simulation of sheet metal forming processes, high strength steel shows better forming behaviour in roll
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forming than in air bending, with the roll forming process able to form tighter radii without failure. This was concluded to be due to the higher damage factor of air bending when compared to roll forming.

In terms of the numerical approach, the simulation model provided by COPRA® FEA RF was used as a base model, which was then improved using the Marc/Mentat interface. The deflection of the roll forming stand was modelled using springs. Additional structural simulations were developed in Inventor, as an additional way to understand the behaviour of the roll forming stand.

1.5 Dissertation structure

This dissertation is divided in four main parts:

- An introduction to the roll forming process and the state of the art
- A detailed analysis of the experimental setup
- An overview of the computational models used
- Results and discussion

Finally, the conclusions are presented, as well as an outlook of future development and research.
2 The roll forming process

This chapter provides an introduction to roll forming as a manufacturing process. This is a necessary step in understanding the importance of accurately calculating the roll stand deflection and in providing context for the applicability of this project. Special focus has been assigned to the design process of roll forming manufacturing lines, the mechanical construction of such lines and in service interaction between the sheet metal and the machine. Furthermore, a sub-chapter has been devoted to the new technology of flexible roll forming, to which the knowledge generated from this project has also been applied.

2.1 Overview

Roll forming is a high productivity manufacturing process which consists in “forming sheet metal strip along straight, longitudinal, parallel bend lines with multiple pairs of contoured rolls, without changing the thickness of the material, at room temperature” (Halmos 2006) (Figure 4). This continuous strip bending characteristic makes it an ideal candidate for manufacturing all kinds of straight profiles. It became a widespread manufacturing process after the Second World War, and nowadays 35% - 45% of all flat steel produced in North America is processed by roll forming (Halmos 2006). Figure 5 shows the variety and complexity of profiles manufactured by roll forming.

![Figure 4 – Schematic representation of the roll forming process (Abvabi 2014).](image)
Roll forming can be described as an extremely flexible process: around the basic premise of feeding sheet metal into rotating rolls which bend it in incremental steps, a number of operations can be selected, adapting it to the specifics of each part. Curving (or sweeping) stations can be added to produce curved profiles (Figure 6), welding can be applied in-line for tube manufacturing, punching or embossing stations are commonplace, making the profile coming out of the machine a completely finished product. Even the basic premises of the process can be changed: the forming lines do not need to be straight or parallel (flexible roll forming) and pre-heating or localized in-line heating can be used (titanium was roll formed this way in the 1960s) (Halmos 2006; Lindgren, Bexell, and Wikström 2009).
Owing to the high productivity of the process, where production speeds can range from 5-150 m/min, disruptions in production are costly and must be avoided. Thus, it is of paramount importance that potential part defects are detected as early as possible in the design process; simulation plays a crucial role in this. Decisions like the number of forming steps and the forming strategy, which used to be made purely based on experience and "feel", can be verified and analysed using FEA software before any production trials, reducing retooling and redesign costs, and also set up time (Halmos 2006; Abee, Sedlmaier, and Stephenson 2010).

Over the last decade, there has been an effort to enhance the precision of the simulation tools while also reducing calculation time. Software such as COPRA® FEA RF allows users to set up and run several simulations, including user customizable meshes, friction simulation between the rolls and the sheet metal or a pre-punched sheet (Figure 7). However, the modelling of roll stand deflection in the simulation, especially in the case of high strength steel forming, has not been studied in great detail.

2.2 Design of the process

Designing a roll forming process is, in essence, an optimization procedure: the goal is to use the smallest number of passes possible while still maintaining final product quality within acceptable levels. There is, however, a balance: too few passes and the stress introduced in the part in each forming step is too high, which leads to...
distortions; too many passes and the tooling costs become too high, and the process is uncompetitive.

Until the late 20th century, the “optimization procedure” for the number of passes consisted of the following: “The experienced roll designer doodles a bit, gazes at the ceiling and quite positively says, ‘I can do it in ten!’” (Gradous 1966). Indeed, roll forming design is still largely influenced by experience and “feel”. The knowledge generated throughout the years has remained undocumented, and few research papers are published which concern roll forming specifically (Halmos 2006).

The difficulty in developing consistent rules for roll forming design stems from the process geometry itself: the bending sequence as the sheet moves through the machine is composed of continuous movements, as opposed to discreet steps, as is the case in press-braking. This is an advantage as it is possible to reach higher equivalent plastic strains in roll forming due to the incremental characteristic, but it introduces a longitudinal strain in the strip edge due to the theoretical path of the sheet (Figure 8). Furthermore, the real path of the sheet is not the same as the theoretical one due to spring back, leading to the real path looking like a series of steps followed by sharp curves (Figure 9). Both these phenomena introduce strain in the leg of the profile; if this strain exceeds the elastic limit of the material, there will be permanent deformation leading to distortions in the finished product. Hosseini et al. (2010) developed a method for determining the residual strains in roll forming processes which can provide an insight into the stresses and strains built up in roll forming and how to design for them.

Figure 8 - Theoretical path of the strip edge during roll forming (Halmos 2006).
2.2.1 Number of steps

A number of factors influence the number of steps required to successfully form a profile. This means, as was already mentioned, that the decision on the number of passes to use is not straightforward and is, more often than not, an educated guess (Halmos 2006). However, the relative effect of each characteristic of the process and formed section can be established (Table 1).

For a long time there have been attempts at finding an equation which can provide a realistic estimate of the number of passes. Early attempts relied solely on the geometric characteristics of the forming process, attempting to establish the number of passes by comparing the length of the forming path on the base of the profile and on the strip edge. This can be a first approach for a new design, but it is a very simplistic one, which fails to take into account other very important factors (Halmos 2006).
Table 1 - Factors influencing the number of passes (Halmos 2006).

<table>
<thead>
<tr>
<th>Factors</th>
<th>Decrease</th>
<th>Increase</th>
</tr>
</thead>
<tbody>
<tr>
<td>Depth of the section</td>
<td>Shallow</td>
<td>Deep</td>
</tr>
<tr>
<td>Bend lines</td>
<td>Open</td>
<td>Hidden</td>
</tr>
<tr>
<td>Section tolerance</td>
<td>Loose</td>
<td>Tight</td>
</tr>
<tr>
<td><strong>Material</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Thickness</td>
<td>Thicker</td>
<td>Thinner</td>
</tr>
<tr>
<td>If mill is strong enough</td>
<td>Low yield strength</td>
<td>High yield strength</td>
</tr>
<tr>
<td>If mill is not strong</td>
<td>High elongation</td>
<td>Low elongation</td>
</tr>
<tr>
<td>Mechanical properties</td>
<td>Hot rolled steel</td>
<td>Pre-painted or luster surface</td>
</tr>
<tr>
<td>Surface</td>
<td>Tight tolerance material</td>
<td>Loose tolerance material</td>
</tr>
<tr>
<td>Uniformity (twist, waviness, camber, etc.)</td>
<td>Continuous strip</td>
<td>Precut</td>
</tr>
<tr>
<td>Continuity</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Other operations in the line</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Holes at the center</td>
<td>No influence</td>
<td></td>
</tr>
<tr>
<td>Holes close to edges</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Notches</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Welding</td>
<td>—</td>
<td></td>
</tr>
<tr>
<td>Curving</td>
<td>Same or less</td>
<td>—</td>
</tr>
<tr>
<td><strong>Mill characteristics</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Larger shaft diameter</td>
<td>May be less passes</td>
<td>—</td>
</tr>
<tr>
<td>For deep sections and panels</td>
<td>Less passes</td>
<td>—</td>
</tr>
<tr>
<td>Horizontal distance</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Large</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Small</td>
<td>—</td>
<td>—</td>
</tr>
</tbody>
</table>

Another approach was developed by Halmos (2006) and consists of an empirical equation (2.1) with variables accounting for the various factors described in Table 1:

\[
n = \left[ 0.237h^{0.8} + \frac{0.334}{t^{0.87}} + \frac{a}{90}\right]^{0.15} s(1 + 0.5z) + e + f + 5zs
\]  

(2.1)

where:

- \( n \) is the estimated number of passes
- \( h \) is the maximum height of the section
- \( t \) is the material thickness
- \( a \) is the sum of the formed angles on one side of the guide plane
- \( Y \) is the yield strength (MPa)
- \( U \) is the ultimate tensile strength (MPa)
- \( z \) is the pre-punched hole/notch and strip continuity factor
- \( s \) is the shape factor
- \( e \) is the number of extra passes for other operations
- \( f \) is the tolerance factor

The factors in equation 2.1 were determined by the author and can be found in the reference. Software packages use equations such as this one to offer a first estimate of the number of passes needed, often attempting to include as many factors as possible.
to increase the accuracy of the model. COPRA® RF has a Deformation Technology Module which uses analytical models to find the longitudinal strain at set intervals of the forming process. The strain levels at each point can be compared with the elastic limit of the material, thus providing a more complete picture of the forming behaviour of the material.

Equation 2.1 was used in section 3.1 in order to evaluate the forming strategy used on the experimental setup of the present work.

2.2.2 Bending strategy

The forming process design in roll forming is achieved by drawing the finished profile and then “unfolding” it in a series of steps corresponding to the projected number of passes. The overlaid drawings of all the passes are called the flower diagram (Figure 10); this is an important tool for visualizing the whole process in one drawing.

Developing a good bending strategy is very important in order to assure a good forming path of the material, minimize residual tensions in the part and allow access to the forming radii by the rolls. The first step is to choose a vertical guide plane (or more, in the case of complex sections), as can be seen in Figure 10. Usually, the points of the sheet located on this plane will either travel in a straight line or slightly upward, depending on the roll diameter selection. The selected vertical guide plane will influence the edge travel, and thus the edge stress, during the forming (Figure 11) (Halmos 2006).

![Figure 10 - Flower diagram of a roll formed part (Halmos 2006).](image-url)
The magnitude and location of each forming step angle is then selected, so that the forming is constant and smooth. Several bends can be formed in the same step, or one bend can be formed completely before the next. It is at this point that overbending is included in the forming in order to compensate for springback, the angle being dependent on the material properties and the forming characteristics. Furthermore, the designer has to pay attention to the accessibility of the bend lines; sometimes it is better to form a shallower angle in one step in order to ensure a better access to the bend radius in the next step (see Figure 12) (Halmos 2006).

An investigation by Abeyrathna et al. (2014) found that higher forming angle increments produce higher strains on the strip edge and that the processes where the highest forming angle increments were used produced higher bowing on the formed parts. It was also found that the permanent longitudinal strain on the strip edge may influence the springback behaviour.
Usually the last passes, which define the final angles of the profile, suppose a certain degree of roll adjustability, so that the machine operator can change the bent angles slightly and compensate for part defects (Halmos 2006).

2.2.3 Computer simulation

The introduction of finite element simulation to roll forming has been a crucial step in improving part quality and lowering costs in the industry. Every design decision, such as number of passes and bending strategy, can be checked before any manufacturing takes place and changed if necessary. Furthermore, the coupling of roll forming design software (such as COPRA® RF) with FEA software has led to an extreme reduction of the simulation preparation time: changes to the model are simple and fast to apply, and the simulation models are automatically generated, enabling roll forming design engineers without simulation experience to run several simulations in a short period of time (Abee, Sedlmaier, and Stephenson 2010).

The design process is upgraded by the use of numerical simulations through the introduction of a new iteration in the process. Whereas traditionally the tools were produced right after the initial design, initiating an iterative process of trial and error, nowadays FEA software allows the setup of a virtual roll forming line, where the iterative process can be carried out without the prohibitive retooling costs (Abee, Sedlmaier, and Stephenson 2010).

![Figure 13 - Design process of a roll forming line; in blue is the traditional process of trial and error; in orange, the upgraded process with virtual testing; dashed lines represent the connection between the cycles.](image)

It must be noted that the inclusion of FEA in the design process does not eliminate the need for roll forming experience and knowledge. Rather, it enables the experienced roll forming engineer to detect problems even before they appear in real life. Furthermore, FEA can be applied to existing processes when trying to find out the reasons for problems in the manufacturing process in terms of tolerances, surface quality or roll wear issues, for example. In fact it is possible, in the long run, to create a common base of knowledge founded on the FEA analyses the company has
run that can be accessed and shared by all employees, thus furthering the roll forming experience and capabilities (Abee, Sedlmaier, and Stephenson 2010).

2.3 Types of roll forming stands

An important point to consider when analysing tool deflection is how the rolls are supported in the machine. Thus, knowing the different types of stands and how the rolls are mounted on them is crucial to determining the machine stiffness. The roll stands mentioned in this chapter are: the standard mill, with the shaft supported on both sides; the cantilevered mill, where the shaft is only supported on one side; the double high mill, with two levels of rolls; and the side-by-side mill, with two sets of rolls mounted on the same shaft.

2.3.1 Standard mills

The standard roll forming stand has the shafts supported on both sides. On the drive-side, the shaft is coupled to a motor which drives the rolls; on the operator-side, the stand which supports the shafts is removable in order to enable tool changing. This is the most common type of stand to roll form metals (Halmos 2006), and as such (and due to its availability) it was chosen as the stand to be used in the experimental analysis of the present work. Figure 14 shows a drawing of a standard roll forming mill.

![Figure 14](image)

Figure 14 – Standard roll forming mill, with shafts supported on both sides; on the left is the drive-side, on the right is the operator side (Halmos 2006).

2.3.2 Cantilevered mills

Cantilevered mills have the shafts supported on only one side, the drive-side. These are simple and low-cost designs, which have been used to form simple and narrow sections (Halmos 2006). Figure 15 shows a cantilevered roll forming line in service.
The absence of an operator-side stand makes tool changes faster, thus enhancing the productivity of such machine designs.

However, the application of this constructive solution depends on the deflection analysis of the shaft during forming. As the shaft is only supported on the drive-side, it behaves as a cantilevered beam during forming, and the total deflection is four times higher than the deflection on the standard mill design (Halmos 2006). A possible solution for this problem can be the application of a connector between the operator-side ends of the shafts (Figure 16).
2.3.3 Double high mills

This kind of mill is similar in construction to the standard mill, but maximizes the available space by having two types of stands, one higher and one lower (Figure 17). This allows a factory with limited space to have two types of profile in production in the same machine. The profiles must be produced one at a time, not simultaneously. Thus, the changeover time between two profiles is short. However, installation of side rolls is hard due to lack of available space, and the inspection of forming conditions is not easy.

Figure 17 – Double-high mill; this setup uses the space between stations to install another set of rolls (Halmos 2006).

This kind of mill exhibits similar deflection characteristics to the standard roll forming mill.

2.3.4 Side-by-side mills

These consist in a standard mill with a longer shaft, which can accommodate two or even three sets of rolls (Figure 18). This reduces tool changeover time by enabling two profiles to be produced in the same machine, although not at the same time; this is in a way similar to the double-high mill. However, the longer shaft and the number of rolls on each shaft lead to problems in terms of machine deflection: the weight of the rolls is more significant, leading to an initial downward displacement, which makes setting the roll gap on all the roll sets harder; then, when forming is occurring, the forming forces lead to a higher displacement due to a lower assembly stiffness (Halmos 2006).
2.4 Machine components

In order to form the sheet metal into the desired profile, attention must be paid to the components surrounding the rolls, and to the rolls themselves. A correct manufacturing, setup and in line adjustment of the machine is key to obtaining final products within the desired tolerances. Figure 19 shows the main components of a roll forming line, as relevant for the present study.
2.4.1 Rolls
In practical terms, roll manufacturing must be extremely precise. As will be established further on, deviations of roundness or eccentricities of a very low order (0.1 mm) can have great implications in part quality. Indeed, one of the problems encountered in the simulation of roll forming processes is the influence of these imperfections, which is not easily implemented in the numerical models. The surface quality must be extremely high due to contact and lubrication issues. Often, rolls are split into several sections for ease of manufacturing (Figure 20); in fact, there are instances where this is the only possibility. This also allows for easier installation (Halmos 2006).

The roll gap must be set precisely and must be even throughout the roll profile. In fact, one of the drawbacks of roll forming is the very long set up time, as each pair of rolls must be adjusted individually.
2.4.2 Shafts

The shafts on a roll forming line, which support the rolls and connect them to the side stands, must be very stiff in order to ensure good dimensional stability of the roll gap. Usually, the thickness for these parts is around 50 mm; the shaft will have a keyway through its whole length which will fit with the rolls (Figure 21). The shaft rests on bearings on both stands. On the drive side it will have a section with a larger diameter which rests on the bearing. On the operator side, it is threaded and has a spacer which fits into the bearing and allows the operator to tighten the whole assembly (rolls and spacers) with the nut, independently of the stand and bearings. The threads are usually opposite (the top one is right thread and the bottom one is left thread) in order to avoid the unintentional untightening of the nuts during operation. On the drive side, the shaft has a standard shaft end for fitting a flange, cogwheel or chain drive (Halmos 2006).

![Figure 21 - Shafts mounted on a roll forming stand, without the rolls (Halmos 2006).](image)

2.4.3 Stands

Each side stand consists of two bearing housings, which are supported by connecting rods, in turn attached to cross bars. The stand used in the present work is shown in Figure 22. Each bearing housing usually has two conical bearings which support the shaft (Figure 23). The height of each shaft is adjusted by screws on the top and bottom crossbars (in the stand depicted in Figure 22 only the bottom roll is adjustable due to the inclusion of a load cell for force measurement). Aside from the adjustment screws, the housings slide freely on the connecting rods.

In order to change the roll set on a roll forming line, the operator side stand must be removed. This means that positioning on the machine table must be precise and repeatable. If the stands are not correctly mounted, there is the possibility that the shafts are not perpendicular to the direction of travel, which can lead to substantial problems during operation (Halmos 2006).
2.4.4 Driveshaft

Several driving methods can be used for connecting the electric motor to the shafts. Chain drives are usual. Nevertheless, the present study focuses on the universal joint driveshaft, as it was the chosen method for connecting the two parts. A telescopic universal joint driveshaft was chosen in order to have some flexibility for the positioning of the flanges on the gearbox and on the roll forming stand. Figure 24 shows an example of a roll forming line with this kind of setup. This setup allows for the independent driving of each roll, which can be useful due to the varying sheet speed through the roll forming line (Halmos 2006).
2.5 Roll load

Forming forces occur on the roll forming stand due to the bending of the sheet metal when it passes through the rolls. Knowing the value of this force is crucial to correctly design the stands, shafts and driving system, and to choose the correct motor and gearbox combination. However, due to the incremental nature of the process, it is hard to determine these forces analytically. An earlier effort to reach a mathematical formulation for the roll load (Bhattacharyya et al. 1987) equated the external amount of work to the internal dissipation of energy, while also resorting to the small deflection theory. Equation 2.2 is the result of that study:

\[ P = Y \left( \sqrt{\frac{2t^3 \Theta^3 a}{3 \sin^2 \Theta}} + \frac{3h_x E I}{(D-X)^3} \right) \]  

where:

- \( Y \), is the yield strength (MPa)
- \( t \), is the material thickness
- \( \Theta \), is the angle increase from the previous station to the current one
- \( a \), is the flange height
- \( h_x, D \) and \( X \) are machine parameters
- \( E \), is the Young’s Modulus of the profile
- \( I \), is the second moment of inertia of the profile at the current station

The first term of the equation corresponds to the bending and stretching of the profile as it is being formed, while the second term represents the downward force the profile exerts on the station due to it being curved downward in the previous station and then lifted up in the station to be calculated (Figure 25).
The roll load is responsible for the deflection of the roll forming stands, and thus is
-crucial for the present work. One challenge for the reliable simulation of roll forming
-processes is the inclusion of tool-sheet interaction. The usage of rigid body models for
-simplification foregoes the modelling of roll compliance, leading to force predictions
-which are not reliable (Groche et al. 2014). Experimental data verifying this
-phenomenon and possible improvements to the FEA model can be found in Chapter
-4.

Equation 2.2 was used in section 3.2 in order to get a first estimation of the forming
-forces in the case of the experimental line developed in the present work.

2.6 Roll stand deflection

Due to the forming loads described in section 2.5, a deflection of the shafts, stands
-and all other load bearing elements is to be expected during forming. Although this
-can have a negative influence on roll formed part quality, not much attention has
-been paid to this topic, and in fact few publications even mention it. On the Roll
-Forming Handbook (Halmos 2006), the part dedicated to roll stand deflection
-mentions the need for very stiff parts in order to avoid significant deflection, but
-offers no insight into the effects of this deflection or solutions to the issue.

The forming forces occurring in roll forming have increased lately due to the usage of
-high strength steels. In fact, applications of this material often coincide with the need
-for very strict tolerances, as is the case of the automotive industry. The forming
-forces act on the rolls, which in turn deflect the shaft and all other load bearing
-components. This results in an increase of the roll gap, which can lead to part
defects.
The deflection of the machine can be approximated by a spring model which contains the stiffness of each machine component. Figure 26 shows the spring model developed in an investigation of the forming forces (Groche et al. 2014) for a roll forming line with a similar setup. The roll stand shown is for the roll forming line used in the present work.

The stiffness of the machine can thus be approximated to a single spring, with stiffness given by:

\[
k_{\text{tot}} = \frac{1}{\frac{1}{k_{\text{Be}}} + \frac{1}{k_{\text{LC}}} + \frac{1}{k_{\text{CR}}} + \frac{1}{k_{\text{ST}}} + \frac{1}{k_{\text{SB}}} + \frac{1}{k_{\text{CB}}}} + 2 \times \frac{1}{k_{\text{Sh}}}
\]

(2.3)

where:

- \( k_{\text{tot}} \) is the total machine stiffness
- \( k_{\text{Be}} \) is the bearing stiffness
- \( k_{\text{LC}} \) is the load cell stiffness
- \( k_{\text{CR}} \) is the connecting rod stiffness
- \( k_{\text{ST}} \) is the top adjustment screw stiffness
- \( k_{\text{SB}} \) is the bottom adjustment screw stiffness
- \( k_{\text{CB}} \) is the crossbar stiffness
- \( k_{\text{Sh}} \) is the shaft stiffness

Mackel (1996) describes a process of calculating the stiffness and deformation of each machine component.

The asymmetry between drive side and operator side can also be an issue. Usually, the drive side is reinforced due to the torque of the driveshaft. Also, the bearings
used on the drive side are usually larger than on the operator side. This can mean the deflection is not symmetric and the rolls can end up not being parallel. Furthermore, many of the profiles produced by roll forming are asymmetric, which also contributes to uneven loads on each side of the stands.

Accurately knowing the expected tool deflection and its effects can have a very positive influence on part quality, and can lead to better optimization of the roll forming stand design. In a manufacturing environment where tolerances are ever tighter, roll forming can gain ground on other competing manufacturing processes through better and more consistent part quality.

2.6.1 Importance in terms of part quality

Tool deflection affects part quality through the increase of the roll gap. This increase can be very small, of the order of the tenth of a millimeter, but it can have an influence on the final product, leading to:

- A lower formed angle (the sheet is not bent as much as it should because both the web and the flange have room for movement)
- A higher bend radius, because the radius in the roll does not perfectly follow the desired contour
- Because the bend radius is higher, there is less plastic strain, distributed over a larger space; thus, springback will be higher, lowering the formed angle even more

Figure 27 shows the effect of an increase in gap on the forming behaviour of the sheet.

![Figure 27 - Effect of roll deflection in the forming process; in red dashed line is the designed profile; in blue is the material behaviour when the gap increases; the roll movement is magnified for clarity.](image-url)
An investigation by de Argandoña et al. (2012) found that roll gap influences the final part quality and that a higher roll gap leads to a higher springback. It was also concluded that the shaft deflection is an important parameter that must be studied in the context of profile quality investigations.

Abeyrathna et al. (2013) investigated the relationship between roll load and torque and the bow height of roll formed profiles. A link was found wherein higher roll loads produce higher bow heights. It could be the case that the bow height is affected by the roll deflection, which is itself a consequence of the roll loads. Further studies are needed to understand clearly the effect of roll load and roll deflection on the profile bowing.

### 2.7 Flexible roll forming

A new development in the roll forming field is flexible roll forming, which is roll forming of profiles with discontinuous cross sections, variable in width or depth. Examples of such profiles can be seen in Figure 28. The discontinuity is achieved through the use of rolls which are mounted on so called bipods (Figure 29). The rolls are moved by rotation of the two spindles; if both are rotated in the same direction, the stand moves forward or back, and if they are rotated in opposite direction, the stand rotates. This is referred to as parallel kinematics, and the motion control system (COPRA® AMC) for such drives has been developed over the past 12 years at data M (Albert Sedlmaier and Hennig 2011).

Figure 28 - Profiles obtainable by flexible roll forming.
The roll stand model shown is capable of flexibility in the width direction. Flexibility in depth is in an advanced stage but not yet implemented in the industry. The process is similar to roll forming, but the rolls are split symmetrically and they move and rotate according to the desired contour (Figure 30) (Albert Sedlmaier 2014).

This new implementation of roll forming has a high potential for application in the transport industry, especially in the case of commercial vehicles or passenger vehicles. In the case of commercial vehicles, there is the case of the long member of the truck.
chassis, a very long (14 m) beam with a varying cross section, which is currently produced by deep drawing. Flexible roll forming has the potential of reducing lead times while lowering the cost per part due to a higher automatization and higher process output. This is due to the ability to produce different families of parts, with different lengths and width profiles, with the same set of tools. This in turn lowers the manufacturing time per part by eliminating tool changeovers. Also, in the case of passenger vehicles, profiles for the chassis can be flexible roll formed, leading to a higher flexibility of design and project, while at the same time enabling the load optimization of each chassis member by adapting the cross section to the design loads. Load optimization is, in turn, key for lowering the overall weight of the chassis, which is a major objective of automotive companies worldwide. Furthermore, roll forming has proven to be a prime candidate for manufacturing of high strength steels; this kind of material has become a prime choice for passenger vehicle application due to the possibility of lower part thicknesses, further lowering the overall weight of the chassis (Ferreira et al. 2015).

2.7.1 Roll stand deflection in flexible roll forming

Flexible roll forming presents specific challenges in terms of deflection which can significantly influence the final parts. The stands are cantilevered, meaning the roll is only supported on one side; this leads to a higher shaft deflection. Moreover, the usage of spindles to move the stand means there is elasticity in the system, and the play in the nuts that thread on the spindles further enhances this effect. However, since the movement of the stands in flexible roll forming is controlled by a motion control system, these deflections of the moving components can be calculated and compensated for in the motion calculation (Albert Sedlmaier and Dietl 2015; Guan et al. 2013). This means the simulation of the deflection of all the components in flexible roll forming gains an added importance: not only is deflection of the rolls important in terms of overall part quality due to the effects discussed in section 2.6.1, but also the deflection of the stands themselves and their components is crucial to the geometric stability of the process. Accurately knowing these displacements in an essential component of the development of more advanced and more stable flexible roll forming lines, in width and depth (A Sedlmaier, Hennig, and Abee 2011).
3 Experimental analysis

The development of any FEA model must be done in parallel with an experimental analysis in order to validate the simulation data and understand what the model can accurately predict, and what its shortcomings are. An experimental setup was developed which could provide the necessary measurement data. The following chapter describes the process, machine and measurement systems developed, explains the design decisions that were taken and finally gives the results collected and their analysis.

3.1 Roll forming design

The experimental setup design started with the profile selection and bending strategy development. These were constrained due to the need to use tools already available at data M Sheet Metal Solutions. Nevertheless, the roll forming tool design process is described as it influenced the machine design in a later stage.

3.1.1 Profile

The profile used in the present work was a U-channel profile. This is a simple kind of profile with only two bend lines, which reduces the number of passes needed. The strip thickness used was 0.96 mm. It was decided that the profile length should be 2 m, so that the measurements could happen in a 1 m section of the sheet and the 0.5 m on the front and on the back of the profile could be discarded. This way, it was possible to have an approximation of a continuous process, discarding the effects of sheet entry and exit. The design process started by introducing the final profile shape to COPRA® RF (Figure 31).
The final bend angle of $80^\circ$ was used because a $90^\circ$ bend would imply an overbending pass. This kind of pass would then require a different kind of guiding to the one used for the previous forming steps, which would unnecessarily complicate the machine design. Furthermore, the forming forces during this step would be directed towards the machine stands and not upward, leading to a different deflection behaviour. This kind of forming pass should be studied in further investigations.

### 3.1.2 Bending strategy

Using COPRA® RF, the bending strategy was developed based on the final profile shape. A first approach to the selection of the number of passes was made using equation 2.1. The following assumptions were made:

- $h$ (maximum height of the section) = 0.0294 m
- $t$ (thickness) = 0.96 mm
- $\alpha$ (maximum bend angle on one side) = $80^\circ$
- $Y$ (yield strength) = 490 MPa
- $U$ (ultimate tensile strength) = 870 MPa
- $z$ (prepunched hole/notch and strip continuity factor) = 0.18 (due to pre-cut sheet)
- $s$ (shape factor) = 1
- $f$ (tolerance factor) = 0 (loose)

A recommendation of 13 passes results from that equation. However given the already existing tool inventory and the objective to minimize the number of passes and guarantee high forming forces in order to obtain a significant deflection of the roll forming line, a bending strategy with four passes was used, with a $20^\circ$ angle increase in each pass. The bending process used was constant arc length, which means the full length of the bend element is formed at each step, with the neutral layer radius decreasing with each pass (Halmos 2006). This bending method was found by Lindgren and Bexell (2014) to be the most appropriate bending method for
designing roll forming tools for high strength steel due to the material being less stretched in the bending radii. The flower diagram obtained from COPRA® RF is shown in Figure 32.

![Figure 32 – Flower diagram for the bending strategy developed using COPRA® RF.](image)

3.1.3 Tool design

Having developed the bending strategy, the next step was the design of the rolls for the roll forming process. Tool design is done in COPRA® RF using the flower diagram as a base. Each step is displayed individually, and the rolls are created automatically from the profile outline.

A shoulder was designed on the rolls (Figure 33), on each side of the profile. This had the purpose of guiding the sheet inside the rolls, keeping it aligned. These shoulders were rounded on each roll set in order to guide the sheet into the rolls. When the sheet first enters the roll, it still has the shape of the previous pass. The rounded shoulders guide the strip edge into the rolls for the next pass.

The rolls used in the present work were also designed with a slot on either side (Figure 33). This was done in order to enable the placement of the strain gauges on the sheet during forming, to evaluate the longitudinal strain applied on the sheet in each forming step. This investigation was not done in the present work but the possibility was left open for future investigations using the same equipment.

The final roll design is shown in Figure 34.
Figure 33 – Roll design, second forming step; the rolls are separated for clarity.
3.2 Roll load estimation

In order to have a value for the expected forming forces, and thus the load applied on the rolls, equation 2.2 was used. The second term of the equation was not considered as it is related to the influence of the previous station in the calculated station. As in this experimental procedure only one stand was used, this component of the force is not present. The following assumptions were made:

- Y (yield strength) = 490 MPa
- t (thickness) = 0.96 mm
- Θ (angle increase from the previous station to the current one) = 0.349 rad
- a (flange height) = 23.3 mm

Since the angle increase in each pass is constant, the forming force calculated using equation 2.2 is also constant. The estimation obtained was 1095 N for the forming force in each step.
3.3 Machine design

The experimental analysis of the tool deflection required the design of a dedicated roll forming line which enabled the measurement of the required parameters. The concept developed consisted of a single roll forming stand equipped with the measurement devices needed. The sheets would then be run through each forming pass individually, and the rolls would be changed after all the trials were run on each step. This isolated the measurements from the effects of previous or subsequent forming steps, focusing the investigation on what happened in each station individually.

3.3.1 Roll forming stand

The roll forming stand used in the present work was already available at data M Sheet Metal Solutions. It is a standard mill type stand (Figure 35), with the shafts supported at both sides. The shafts have a 50 mm diameter with a keyway through the length. The shafts are supported by DIN720 tapered roller bearings on the drive-side, and by DIN625 single row ball bearings on the operator side (Figure 36). These are housed inside the bearing housings. The rolls are mounted on the shaft with spacers, and fastened with an M48 nut (Figure 37). The drive-side of each shaft is connected to a flange, which connects the shafts with the powertrain (section 3.3.5).

Figure 35 – Roll stand design used in the experimental setup.
Each bearing housing slides freely on the connecting rods, is adjustable in height by screws. The top roll adjustment screw assembly includes a load cell for force measurement, and the connecting rods on the drive-side have strain gauges mounted on them for measuring the deflection of these components. The adjustment procedure is described in section 3.3.4. The measurement systems are discussed in section 3.4.

The bottom crossbars are fixed to the machine table with ISO10642 M14 countersunk flat head screws. These guarantee that the two stands are parallel to each other due to the conical shape of the recess.

3.3.2 Machine table

The machine table used in the present work was designed specifically for this project. The goal was to have a compact design which allowed for future expansion of the
experimental line. Furthermore, stand had to be developed to support the drivetrain. The final design was a 1080x300 mm steel plate (Figure 38). The 1080 mm dimension was chosen due to the standard shipping pallet size of 1100x800 mm, to allow the whole machine to be shipped in one standard pallet. The 300 mm dimension was chosen in order to enable the future expansion of the roll forming line by adding a second plate similar to the first one. If two plates are assembled together, the distance between roll forming stands will be 300 mm.

The powertrain stand is bolted to the machine table using ISO4762 M12 hex head screws. There are two positions for mounting the gearbox and the motor. The lower position was used in the present work in order to have the bottom roll driven. The powertrain stand design was, however, done in such a way that it is possible to mount two gearbox and motor assemblies, and have both rolls driven independently. This can be important in future studies of parameters such as roll torque.

The machine table is supported by three stands. These are manufactured using DIN59410 square tube with 50x50x2.9 mm. The horizontal tube (which contacts with the ground) has screws in order to adjust the height of the machine table and ensure that it is level with the ground (Figure 39).
3.3.3 Entry and exit guiding

In order to ensure the strip was correctly inserted in the rolls, an entry guiding system was developed. This consisted of an entry table with rollers to guide the strip in the vertical direction, and cylinders to guide it in the horizontal direction (Figure 40). Metal plates were added between each pair of rollers due to safety concerns: even though the sheet speed was low, if a finger or arm is placed between the sheet exiting the roll forming stand and the rollers, serious harm can occur.

Figure 39 – Machine stand.

Figure 40 – Front part of the entry guiding system.
The whole table was adjustable in height using its extendable legs and the attachment points with the machine table. The horizontal guiding was adjustable by untightening a screw beneath each cylinder and moving it.

Exit guiding was only done in the vertical direction by using single rollers on a stand (Figure 41). It was important to ensure that the profile was supported as it exited the stand in order to avoid interferences in the force measurement due to the profile moving downward under its own weight.

![Figure 41 – Vertical exit guiding system with single roller stands.](image)

### 3.3.4 Machine setup

Machine setup and adjustment was a crucial step in the experimental analysis, as even very small deviations could have a negative effect on the measurements. Furthermore, all the adjustments needed to be made before each forming step, after the rolls were installed. The setup of the experimental line in the present work consisted of aligning the entry guides with the rolls and setting the roll gap.

The adjustment of the roll gap was done using feeler gauges. Each stand had a screw which could independently adjust the height of each bearing housing. The bottom screws were directly connected with the bottom crossbar, but the top adjustment mechanism was substituted with the load cell. Because of this, an adjustment procedure was developed using a differential screw. The load cell threaded on a holder which, itself, threaded onto the cross bar (Figure 42). The load cell had a thread with a pitch of 2 mm, and the holder had a thread with a pitch of 1 mm. Thus, a 360° rotation of the holder produced a displacement of the load cell of 1 mm, allowing for precise adjustment.
The roll gap was set for 1.05 mm. It was decided to use a higher gap than the sheet thickness in order to rule out massive forming effects which elevate the forming forces considerably.

![Adjustment device using the load cell.](image)

Due to the unevenness of the spacers provided for the shaft, the rolls couldn’t be placed perfectly aligned with each other. Thus, it was decided to use a technique which is common in the roll forming industry that consists in leaving the top roll loose in the shaft, with approximately 0.4 mm of gap. The forming forces would then balance the roll into its aligned position. This meant that only the bottom shaft nut was tightened with the prescribed torque. The top shaft nut was only tightened up to the point which left the desired gap. This decision had an effect in the measurements which is discussed further in the Results section.

The entry guides were adjusted using two nylon wires which was taped to the rolls and then placed on the entry table. After ensuring the wires were parallel to each other and to the roll surfaces, the vertical entry guides were moved until they touched the wire, and then screwed in place. The distance between the guides was measured using a caliper and it was set to the strip width +0.5 mm.

### 3.3.5 Powertrain

The driving system used for the roll forming stand consisted of a servo motor (Schneider Electric BSH1004) assembled on a gearbox (SEW RZ57 AQH100/3/4, with a ratio of 48.28:1). The servo motor was available at data M Sheet Metal Solutions, and the gearbox was selected so that the forming speed could be as low as 1 m/min while still providing enough torque for the roll forming process. The RZ57
model was selected due to the future possibility of mounting a second gearbox and servo motor on the machine table and driving both rolls separately.

A driveshaft was used to connect the gearbox to the roll forming stand shaft. This was a telescopic shaft with two universal joints in order to compensate for the height difference between the gearbox and the shaft.

**Control system**

The control system for the servo motor consisted of a Lexium 05 controller (Figure 43) connected to a potentiometer for regulating the roll rotational speed, a switch for reversing the roll rotation direction, an emergency stop switch and a dead man’s switch (Figure 44).

![Figure 43 – Electrical cabinet for the machine control.](image)

![Figure 44 – Machine control panel.](image)
The roll rotation speed was regulated by the potentiometer, but since there was no display of the current speed, it had to be measured using a stopwatch. The desired linear sheet speed was 2 m/min. The diameter of the bottom rolls was 150 mm at the point where they contacted with the bottom of the profile, which meant the perimeter was 0.471 m. Therefore, a rotation speed of 4.25 rpm was selected.

Whenever the emergency stop switch was engaged, a reset button had to be pressed afterwards to re-activate the servo motor. Due to safety concerns, the dead man’s switch was used as the operator control for the machine. This switch has three positions: off, on and off. If the switch is not pressed, the servo motor is disabled. If the switch is pressed halfway, the servo drive is activated. If the switch is pressed too far, the servo motor is disabled again. This ensures the person operating the machine is deliberately activating the drive. The switch must be pressed through all the operating time. The goal was that there could never be the case where the operator is trapped by the rolls (for example, a hand getting drawn into the rolls) and cannot reach the emergency stop.

3.4 Measurement systems

The measurement system installed on the roll forming line in the present work had the goal of investigating the behaviour of the roll forming stand during forming. It was composed of load cells, to measure the forming force on the drive-side and operator-side stands, strain gauges mounted on the drive-side connecting rods, to measure the deflection of these components during forming, and dial indicators to measure the displacement of various machine components during forming (Figure 45). The following section describes the measurement setup and procedure.

3.4.1 Measurement setup

The three components of the measurement system were:

- Load cells
- Dial indicators
- Strain gauges

The load cells were placed on the connection between the top shaft bearing housings and the top crossbar. That way, it was guaranteed that the load applied on the top roll in the vertical direction would be in turn applied on the load cell; since the bearing housing slides freely on the connecting rods, the load cell is the only rigid connection between the bearing housing and the top crossbar.

The load cells chosen were the KM30z model from ME Systeme with a 20 kN rated load. This rated load was selected in order to have a safety margin during the forming and to enable future studies with higher sheet thicknesses or material grades.

The dial indicators were placed in four different positions:

- The first dial indicator was always placed on the top roll, measuring the vertical displacement (Figure 45)
• The second dial indicator was placed:
  
o On the bearing housing, measuring the displacement in the vertical direction (Figure 45)
  
o On the top crossbar, measuring the horizontal deflection of this component (Figure 46)
  
o On the top roll, measuring the side movement of the roll during forming (Figure 47)

These positions were chosen in order to observe the behaviour of different roll stand components. The dial indicator on the top roll measures the displacement of the shaft and the overall increase in roll gap. The dial indicator placed on the bearing housing measures the displacement of the bearing housing, and therefore how much the shaft displacement affects the bearing housing position. The dial indicator placed on the side of the top crossbar measures how much of the shaft bending is transferred to the connecting rods by the bearings and bearing housing, and how much it displaces the top crossbar. The dial indicator placed on the side of the top roll measures the lateral movement of the roll during forming due to the machine setup characteristics (the top roll is floating on the shaft with 0.4 mm of gap).
The strain gauges were mounted on the connecting rods, between the two bearing housings. Each connecting rod had two strain gauges in opposite positions, one on the

Figure 46 – Mounting position of the dial indicator on the top crossbar.

Figure 47 – Mounting position of the dial indicators on the top roll measuring the vertical displacement and the side displacement.
inside of the stand facing the rolls, and one on the outside facing the driveshaft (Figure 48). Each pair of strain gauges was connected to a Wheatstone bridge in a half bridge configuration, in adjacent arms, so that the sensitivity of the system was doubled and that the temperature effect was cancelled out (Figure 49). When strain gauges are mounted in a Wheatstone bridge in adjacent arms, the resistance variations are subtracted. This way, any variations in the same direction for both strain gauges are eliminated (temperature variations and traction of the connecting rod). The only strain which has symmetric values for both strain gauges is the bending strain of the connecting rod, so this is finally the only measured quantity.

Figure 48 – Strain gauge mounting position diagram (EETech Media 2015), adapted.

Figure 49 – Circuit diagram of the Wheatstone bridge to which the strain gauges were connected (EETech Media 2015).
However, in the present work, the strain gauges were found to be producing unstable readings and unreliable data such as showing a change in strain when no loading was applied or not registering any strain when a load was applied. It is believed this was due to a bad bonding between the strain gauges and the connecting rods or a welding problem on the wiring, possibly causing a short circuit. As such, the results section of the present work does not mention the strain gauge readings.

3.4.2 Data acquisition

The load cells were connected to a GSV-4USB M12 amplifier from ME Systeme. This amplifier had a USB interface for direct connection to a computer, and four inputs for sensors. The inputs for the two load cells were set in full bridge configuration, and the inputs for each strain gauge pair were set in half bridge configuration. The bridge completion resistances available in this amplifier were 120 Ω, 350 Ω and 1000 Ω, and the sampling rate could be set between 10 Hz and 125 Hz.

The data acquisition software was the one provided by ME Systeme, which allowed for the recording of the measurement data directly into an Excel file. Inside the software it was possible to also set the measurement parameters as needed. The sampling rate chosen was 12.5 Hz and the scaling value was taken from the calibration sheet of each load cell.

The measurements of the dial indicators were recorded using a video camera setup. Two cameras recorded the trials, one pointed straight at each dial indicator. Both cameras were set recording, and then the data acquisition was set to record the data. Simultaneously with the start of the data acquisition software, a loud noise was produced, which allowed the synchronization of the videos with the load measurement. This way, it was possible to know exactly which force measurements to keep (corresponding to the 1 m measurement section) based on the video time stamp, and to compare the force measurement data with the dial indicator readings.

3.5 Material

The material investigated was DP800 high strength steel supplied by TATA Steel with the following properties in the rolling direction:

- Young’s Modulus $E$ – 208 GPa
- Poisson’s Coefficient $v$ – 0.3
- Yield stress $R_{p0.2}$ – 490 MPa
- Tensile strength $R_m$ – 870 MPa
- Elongation at break $A_{80}$ – 17%

The complete material data, including the Swift Law parameters used in the simulation model, is discussed in section 4.5. High strength steel was used as it would provide higher forming forces than low carbon steel, and as it is a very used material in current roll forming processes. Sagström et al. (2008) found that ultra high strength steels are suited for roll forming, and that ultra high strength steel shows a
lower tendency for deformation hardening when compared to their ultimate tensile strength. Lindgren (2009) measured the forming forces with a similar setup, and DP800 steel presents forming forces 40% higher than those of DC01 carbon steel.

### 3.6 Experimental procedure

As only one roll forming stand was used, the strip was run through the first pass, then stored. After the eight trials, the roll set was changed to the next pass, and all the strips were run through the stand again. After each roll set was installed, the roll gap had to be adjusted. The test matrix for each forming step is shown in Table 2.

Table 2 – Test matrix for each forming step, with the forming conditions used and the position of the dial indicators.

<table>
<thead>
<tr>
<th>Trial</th>
<th>Forming speed</th>
<th>Roll gap</th>
<th>Dial indicator #1 position</th>
<th>Dial indicator #2 position</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Drive-side bearing housing, vertical</td>
</tr>
<tr>
<td>#2</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Drive-side bearing housing, vertical</td>
</tr>
<tr>
<td>#3</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Drive-side bearing housing, vertical</td>
</tr>
<tr>
<td>#4</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Drive-side bearing housing, vertical</td>
</tr>
<tr>
<td>#5</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Drive-side top crossbar, horizontal</td>
</tr>
<tr>
<td>#6</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Drive-side top crossbar, horizontal</td>
</tr>
<tr>
<td>#7</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Top roll, side</td>
</tr>
<tr>
<td>#8</td>
<td>2 m/min</td>
<td>1.05 mm</td>
<td>Top roll, vertical</td>
<td>Top roll, side</td>
</tr>
</tbody>
</table>

### 3.7 Experimental results

The measurements from the eight trials on each forming step were collected, averaged, and their standard deviation and standard error were calculated.

The force measurements are presented in Figure 50. In agreement with the data collected by Lindgren (2009), the forming forces in the first station are significantly lower than on the subsequent stations; however there was no explanation found as of
yet for this behaviour. There is also a noticeable difference between the measurements on the drive side and operator side load cells. This can be due to the torque of the motor having an additional effect on the stand, but also, and more likely, due to the fact that the rolls were not perfectly centred on the stand, having had an offset towards the drive side. This mounting position was due to the unavailability of spacers with the correct dimensions for such positioning.

The measured forces are significantly lower than the forces measured by Lindgren (2009) due to the differences in setup and material thickness (0.96 mm DP800 steel on the present study, 1.48 mm DP800 steel on the mentioned publication).

It should be noted that the standard error of the mean, represented by the error bars, is low in comparison with the overall values measured, giving a good confidence in the results.

![Force measurements](image)

Figure 50 - Force measurements for each station; error bars represent the standard error of the mean.

Figure 51 displays the force measurement of trial number 5 on station 2. The undulating behaviour of the force measurements was prevalent on all trials and stations, being especially pronounced on the first two stations. It is speculated that this was due to the eccentricity of the top roll, which was measured at 0.05-0.1 mm for all the rolls. This could lead to a higher force when the larger section of the roll is in contact with the sheet. This explanation is supported by the period of the wave, which is approximately 15 s. The base diameter of the roll is 150 mm, and the forming speed is 2000 mm/min, which gives a time per revolution of 14.13 s, thus confirming the pattern follows the roll revolution.

The oscillating force measurement is more pronounced on the operator side, which could be due to the top shaft not being parallel to the bottom shaft, or to the roll section closer to the operator side being misaligned with the shaft.
The deflection results presented are only for the dial indicator placed on the top roll, as the other dial indicator was moved between trials for getting a better picture of the deflection of the whole stand. Figure 52 shows the deflection results for each station. The standard error of the mean for each measurement is also shown in the error bars, and this value is high for all of the observations, which leads to a lack of confidence in the accuracy of these values. This is due to the measurement technique, which was based on the observation of the dial indicators (leading to poor accuracy) and to the fast movement of the indicator due to the lack of roundness of the top roll, which had to be factored in.

It is noticeable that, even though the first station had a significantly lower average force for all the trials, the deflection was higher. It is believed that this is due to the settling of the stand after assembly, which meant there were manufacturing clearances that weren’t stable. A possible solution would have been to run new trials with the first station, after the machine had settled.
Figure 52 - Deflection measurement obtained from the dial indicator on the top roll; error bars represent the standard error of the mean; the error is high due to the imprecision associated with observing a dial indicator.

It was also observed that the dial indicator placed on the bearing housing on the drive side registered a consistent deflection of 0.01 mm. The data collected is are presented in chart form due to the low number of collected samples.

The measurements of the side deflection of the cross bar (the dial indicator was placed horizontally with the probe touching the side of the crossbar) showed that there was significant movement of this part during the forming, which is consistent with the findings shown in Figure 63: the shaft was not moving due to play on the bearings, but was actually deflecting the whole stands as well. The magnitude of the displacement was higher than on the simulation; this is believed to be due to the effect of the torque of the electric motor (not considered in the simulation).

When the dial indicator was placed on the side of the top roll, it showed significant lateral movement consistent with the assembly gap provided (0.1-0.2 mm). It is believed that, due to the offset of the tools towards the drive side and due to the lack of roundness of the top roll, there was movement during the forming of the top roll towards both sides. This, in turn, contributed to a twist in the parts coming out of station 2. The movement was not so significant in stations 3 and 4 because those had lower forces in the shaft direction acting on the top roll.

The final parts after station 4 are shown in Figure 53 and Figure 54. The final profile quality was good in all the trials. The web showed a slight curvature in all the profiles due to the gap between the rolls being higher than the sheet thickness. There was a slight bowing of the profile through the length, which was predicted by the simulation.
Figure 53 - Final profile, side view; a slight bowing is noticeable through the profile.

Figure 54 - Final profile, front view; the final quality of the profile was acceptable, although the web had a slight curvature (due to the gap between the rolls being higher than the sheet thickness).
3.8 Conclusion

The test stand design used in the present work is thought to be appropriate for the analysis, even though there are weaknesses that must be addressed. The load cells introduce a weak link in the stiffness of the setup; instead of these components, strain gauges could be applied on the top part of the connecting rods, mounted in such a way that only normal stress would be measured. Knowing the geometry and material properties of the connecting rods, one could calculate the forming forces without changing any machine components. Furthermore, using simple adjustment screws instead of the load cells would allow for a better adjustment of the roll gap, and therefore better forming conditions and more consistent data. Another important change in the machine setup should be the manufacturing of new spacers for the shafts, so that both rolls could be mounted fixed and in the middle point between the two roll stands.

In order to better understand the machine behaviour, a different design could be used for the shafts, one which allowed the placement of strain gauges on the side of the shaft. This would give a much more precise estimation of the shaft deflection than using dial indicators. The strain gauges on the connecting rods could also provide important data; their placement on the connecting rods must be improved and the wiring must be redone in order to obtain this data.

On the other hand, a more detailed investigation could be made of the stand behaviour in numerical simulations. The inclusion of the motor torque on the drive-side stand in structural FEA of the roll stand behaviour could be a development, as could be the inclusion of assembly clearances (clearances between the rolls and the shaft, bearing play, clearances between the bearing housing and the connecting rod). Another interesting topic could be the accurate modelling of the forming forces as obtained from the simulation of the roll forming process, applying the loads to the rolls in the correct points and in the correct direction.
4 Numerical simulation

The development of the numerical models was based on the roll forming simulation experience at data M. Initial simulations were prepared and run using the software developed in house. This is the standard worldwide in terms of numerical simulation of roll forming processes, and it was deemed appropriate to use them as a basis for the present work as the wealth of experience collected proves their effectiveness. Further adapting of the models, such as the addition of a degree of freedom to the rolls and the variation of the stiffness of the connection between the rigid roll surfaces was carried on using the MSC Marc/Mentat software package, which is an implicit non-linear finite element analysis software. The following chapter presents the simulation models generated, the material model used for the simulations and the results, including a comparison with the experimental results.

4.1 Modelling concepts

In order to investigate the inclusion of deformable tooling in roll forming simulations, two basic modelling concepts were developed:

- Rigid simulations – the modelling of the tools (rolls) in the simulation was done using surfaces which were fixed in space
- Deformable tooling simulations – the modelling of the top rolls was done using surfaces which had a degree of freedom in the vertical direction, while the bottom rolls were still fixed in space. The degree of freedom was implemented using a spring connecting the surface with the world

The idea was to have a simulation similar to the ones used nowadays in roll forming, with rigid tools, and compare it to the same simulation, except with deformable tools, in terms of stability, computation time and accuracy (for example, in the load determination). The specifics of tool deflection modelling, as well as different concepts for including it in the roll forming simulation, are discussed in section 4.6.

4.2 Geometry

The simulation model used in the present work was developed using the COPRA® FEA RF package, which prepares the simulation automatically based on the roll design developed in COPRA® RF. It consists of a 1000 mm sheet with symmetry
(only one side of the profile is simulated), which is held in place using boundary and symmetry conditions. The rolls are modelled as rigid surfaces which are moved in the forming direction by a boundary condition. Thus the process is inverted in relation to reality: instead of the sheet moving forward, the rolls move back. For the simulations in this study, friction has not been taken into account. The station distance is 1100 mm in order to represent the process as in the experimental procedure, where the stations are isolated and the sheet passes through one whole station before entering the following one. Figure 55 shows the full simulation model, including springs on the upper rolls which were used to model the deformable tooling. This is described in detail in section 4.6.

![Simulation model of part of the roll forming process](image)

Figure 55 - Simulation model of part of the roll forming process; the position of each roll is prescribed by the boundary condition assigned to the respective node; the bottom nodes are fixed in all six degrees of freedom, while the top nodes can have linear displacement in the y direction; this displacement is restricted by a modelled spring between the node and the origin.

In all the simulations run, the standard coordinate system was:

- x for the horizontal direction perpendicular to the sheet travel
- y for the thickness direction in the flat sheet (vertical)
- z for the sheet travel direction (forming direction)

The boundary conditions on the sheet are represented in Figure 56 and Figure 57. On the back end of the sheet a y-lock boundary condition was used which fixes the displacement of the three last nodes in the y direction. On the front end of the sheet, a z-lock boundary condition was used which fixes the displacement of the front three nodes in the z direction. Throughout the middle of the profile there is a boundary condition that fixes displacement in the x direction as the symmetry condition.
The control nodes of the rolls are assigned the boundary conditions as seen in Figure 58. In the case of the rigid simulations (with both rolls fixed), both rolls have the boundary conditions:

- bc_spring_control_xz - locks the movement in x direction and controls the movement in the z direction by a table
- bc_spring_control_yz - locks the movement in the y direction

In the case of the deformable tooling simulations, the top roll doesn’t have the boundary condition bc_spring_control_yz, and instead has a spring with a set stiffness; this allows movement in y direction during forming due to the forming forces. In this case, the movement in the z direction by a table is still active.
4.3 Element type

The element type chosen for all the analyses was element type 7 of the Marc Element Library. This is an eight node hexahedral solid element (Figure 59), with trilinear interpolation. It was chosen as it is the standard element used by data M in the COPRA® FEA RF package and typically used in most of their simulations since their practical experience shows it provides a good representation of the roll forming behaviour of steel materials. It is used with the assumed strain formulation for a better representation of the bending behaviour. Shell elements were not considered for the analysis as previous research at data M showed that the material behaviour through the thickness is not constant and must be represented. Element type 7 is also recommended over higher order elements in contact analysis, which is the case of the present work. The element stiffness is determined using eight-point Gaussian integration (MSC Software 2013).

Using this element, it was possible to obtain the normal strains in the three principal directions and also the three shear strains.
4.4 Mesh definition

The definition of the element size and mesh parameters was based on the experience on roll forming simulations at data M. Over the length of the sheet a constant element size was used as there should be no difference in behaviour at different points of the sheet during forming (Figure 60). The element size in this direction was approximately 2.3 mm. Over the width of the sheet, a finer mesh was used in the bend zones (Figure 61), as this is the area of higher strain values, and on the strip edge in order to accurately model the interaction between the sheet and the roll, especially when the sheet enters a new station; the strip edge is also an area where longitudinal strain typically occurs.

![Figure 60 – Top view of the simulation mesh.](image)

![Figure 61 - Mesh used in the simulation models.](image)

Only one element was used over the thickness as the shear behaviour of the sheet itself was initially not considered relevant for the investigation. It is usual to use models with two or three elements over the thickness when it is important to study the springback behaviour and the end flaring of the parts. However, it is thought that the shear behaviour can have an influence on the forming forces, so further analyses of the tool deflection problem could include an observation of the influence of the number of thickness elements on the analysis.
4.5 Material model

The Swift law (Swift 1952) was used as a material model for the simulations. It is formulated as:

\[ \sigma_p = K(\varepsilon_0 + \varepsilon_p)^n \]  

(4.1)

where:
- \( \sigma_p \) is the equivalent stress
- \( \varepsilon_p \) is the equivalent strain
- \( K, \varepsilon_0, n \) are material constants determined from the uniaxial traction test

The parameters were calculated using the COPRA® FEA RF software. The modelled material was the same as the one used in the experimental tests, DP800 high strength steel. The data provided to the software was (Tata Steel 2013):

- Young’s Modulus \( E \) – 208 GPa
- Poisson’s Coefficient \( \nu \) – 0,3
- Yield Strength \( R_{p0.2} \) – 490 MPa
- Ultimate Tensile Strength \( R_m \) – 870 MPa
- Elongation at Ultimate Tensile Strength \( A_{80} \) – 17%

Using this input, the software provides the parameters described in Table 3.

<table>
<thead>
<tr>
<th>Swift hardening law parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>K (MPa)</td>
</tr>
<tr>
<td>--------</td>
</tr>
<tr>
<td>1361.29</td>
</tr>
</tbody>
</table>

The material was considered isotropic. The final material behaviour including both the linear elastic (Young’s Modulus) and plastic (Swift law) parts is shown in Figure 62.
4.6 Tool deflection modelling

The modelling of tool deflection in the simulation can be done in a number of ways. Several models were thought, including different degrees of freedom. These were:

- Modelling the bottom rolls fixed and the top rolls fixed in all degrees of freedom except displacement in y direction
- Modelling the bottom rolls fixed and the top rolls with freedom of displacement in the y direction and rotation in the z direction (forming direction)
- Modelling the tooling as a series of beam elements with the stiffness of each element corresponding to the stiffness of the tool section it represented

The first case offers the simplest analysis, and just supposes the top roll is displaced upwards by the forming force. It doesn’t take into account the rotation due to the shaft deflection and due to the clearances between the rolls and spacers and the shaft. This was the modelling approach used due to the low computational time required (the simulation is still stable) and due to the symmetry of the profile used.

The second case allows for the rotation of the top roll during forming in addition to the displacement. This modelling approach is interesting for asymmetric profiles and in situations where the forces acting on the top roll are not equal in both sides of the roll. In this case, it might be interesting to have springs also on the bottom roll to observe its behaviour.

The third case is the closest to the experimental behaviour of the roll forming line as it provides a realistic deflection of the shaft, but it also involves a higher computational effort due to the additional beam elements used. The extent of this
computational time increase must be investigated further. Furthermore, the machine stiffness cannot be approximated by a single spring but must be determined for each shaft section and then for the stands and bearings. This further increases the difficulty of obtaining accurate results.

The modelling approach selected, the one allowing only displacement in the y direction, was applied by creating a free node attached to each top roll model, to which the boundary conditions were assigned. The rolls were always fixed in the x direction. In the direction of sheet travel, as mentioned before, the rolls had a displacement corresponding to the sheet movement in the process. In the vertical direction, y, the node had no boundary condition. Instead, a linear spring link was modelled connecting the node with the ground, with a set stiffness. This stiffness had to be set as half of the real stiffness, as only half of the sheet was modelled due to symmetry.

The stiffness values used for the spring were obtained from literature (Groche et al. 2014) and from structural FEA analysis of the roll forming stand. Figure 63 shows this analysis for the first forming step. A load of 1000 N was applied to the rolls. Most contacts were considered as sliding without friction, in order to have a good depiction of the real deflection. The final stiffness values used were:

- 15750 N/mm [Weaker]
- 31500 N/mm [Standard]
- 83100 N/mm [Stronger]

Different stiffness values were used in order to observe the influence of the stiffness variation on the overall deflection behaviour. The values 15750 N/mm and 31500 N/mm were obtained from the literature, while the value of 83100 N/mm was obtained from the structural FEA analysis of the roll forming stand using the CAD/CAE software Inventor.
4.7 Simulation results

The results extracted from the simulation were the measured force in the y direction at the roll surface and the displacement of the roll surface contact body, in the case of the simulations with tool deflection. These were averaged and their standard deviation and standard error of the mean were calculated. The error bars represent the value of the standard error of the mean, and provide an insight into the variability of the force calculation during each pass.

Figure 64 shows the average calculated force in y direction on the top roll. The values for the rigid simulation are clearly higher than all the simulations with deformable tooling, and present a much greater variation. This is explained by the rolls being set at a gap of exactly the sheet thickness, and having a shoulder which restricts the sheet through the width; these constraints mean that massive forming of the sheet is taking place at each forming step, and the calculated forces reflect this behaviour. When relief is provided by way of a displacement of the top roll, the measured force becomes much more stable in the simulation, and the actual behaviour of the sheet is thought to be much more accurate. This behaviour can also be observed in Figure 65, which shows the force measurement through the simulation for the rigid model and for the model with deformable tooling with standard stiffness.

While the rigid model shows a wide variation of the calculated force, the deformable model shows a more constant value, much more consistent with the experimental behaviour of the forming process. The peak value on station 2 can be explained by the lack of a chamfer in the front of the sheet. This would have helped guide the sheet into the rolls and avoid such a peak in the force.
Figure 64 - Average calculated force in the y direction (top roll) during each forming step, for the rigid simulation and the simulations with deformable tooling; the error bars represent the standard error of the mean and illustrate the variation of the calculated force through the simulation.

Figure 65 - Force results in the y direction, through the simulation for all the stations, for the rigid rolls simulation and the simulation with standard spring stiffness (31500 N/mm).

The variation in the calculated force for the rigid simulation can also be explained by the coarse mesh used in the models. Groche et. al. (2014) performed simulations...
where the sheet had a zone with a very fine discretization in one section, while the rest had a coarser discretization, and found that the section with a finer mesh produced a constant force value, while the rest of the sheet presented a variation similar to the one observed in the present work. However, the same investigation found that even when the calculated force was stable, the value was higher than the experimentally measured forming forces. This difference could be eliminated by the use of a representation of the deflection in the top roll.

The results for the calculated deflection are presented in Figure 66. There is an expected pattern of lower stiffness producing higher deflections; however, it can be observed that the increase in deflection is not proportional to the decrease in stiffness. It is believed this effect is due to the non-linearity of the material behaviour in the plasticity region, and also due to geometric constraints on the rolls.

![Deflection results](image)

Figure 66 - Average deflection values calculated for each simulation with deformable tooling.

### 4.8 Calculation time analysis

After each simulation was run, the log files produced by Marc were analyzed. The values taken from these files were:

- Total CPU time
- CPU time distribution
- Total number of cycles and separations

These give an overall indicator of the simulation performance and shed light on possibilities to further develop the current models.
Figure 67 shows the total CPU time for each simulation model. Even though the rigid model is the fastest one, it is noticeable that there is not a significant time increase in the case of the standard model. This is significant as simulations with deflection of the rolls show better performance in terms of load determination (Groche et al. 2014).

Figure 68 presents the division of the calculation time on its components for each model. One can observe that the major factor driving up computation time is the time for matrix solution, which is the time spent in the equation solver by the program. This information is interesting as the total number of cycles, representing the number of iterations for each increment, is highest for the rigid simulation (Figure 69). Thus one can conclude that the springs add complexity to each model, increasing the solving time for each increment, while the rigid simulation is more unstable, leading to a higher number of iterations to achieve convergence in each increment. This is further confirmed by analyzing the number of separations, which is also highest for the rigid model. This means nodes are coming into contact and separating from surfaces more frequently, which increases the computation time per increment.

![Total CPU time](image)

Figure 67 - Analysis of the total computational time for the rigid simulation and for the deformable simulations; it is noticeable that the addition of springs does not greatly impact the total simulation time.
Figure 68 - Analysis of the components of calculation time for each model; the major factor driving the CPU time increase is the time for matrix solution.

Figure 69 - Analysis of the total number of cycles and separations for each model; the number of cycles represents the total number of iterations for achieving convergence in each increment; the number of separations is the number of times a node was separated (lost contact) from a surface.
4.9 Optimization of computation time

In parallel to the development of simulation models with deflection, there was an effort to develop a solution for the application of tool deflection modelling to commercial code. This consisted in a series of models with an increase in roll gap; this increase was taken as the average of the top roll displacement values from the deformable tooling model. In a standard simulation in COPRA® FEA RF, the roll gap is the sheet thickness per default, whereas in the new models the roll gap is the sheet thickness plus the calculated displacement of the top roll. The model geometry was the same as the one for the deflection simulation and the station distance was 400 mm, a normal distance for roll forming.

Six models were developed (Figure 70):

a) **No spring, no gap** – A fixed model with the roll gap equal to the sheet thickness, with no relief angles

b) **Spring** – A model with the roll gap equal to the sheet thickness but with springs (31500 N/mm stiffness) to model roll deflection

c) **Fixed gap, load controlled (with BC)** – A fixed model with the roll gap higher than the sheet thickness where the rolls were fixed by a boundary condition in y direction

d) **Fixed gap, position controlled** – A fixed model with the roll gap higher than the sheet thickness, where the roll surfaces were fixed in space, without any boundary conditions

e) **Spring with fixed gap (fixed y direction)** – A fixed model with the roll gap higher than the sheet thickness, with a spring in y direction and a boundary condition in y direction; this means the spring is modelled but is not loaded during the simulation

f) **Spring with preset gap from table** – A fixed model with the roll gap higher than the sheet thickness, with a spring in y direction and a fixed displacement boundary condition in y direction (controlled by a table) which displaces the rolls at the start of the simulation from zero to the preset gap; in this case the spring is modelled and is loaded during the simulation

All the models except of b) are described as “fixed”. This means the roll gap is constant through all the simulation.
The position control for contact bodies means that the surfaces are fixed by geometric conditions. On the other hand, load control uses a node which is assigned to the contact body, through which boundary conditions can be applied, such as forces, displacements or springs, and giving the contact body a maximum of 6 degrees of freedom. a) and d) use position controlled contact bodies, and b), c), e) and f) use load controlled contact bodies (Gülçeken et al. 2007).
In the case of simulation e), there is a spring linking the nodes on the top rolls to the ground, but these nodes are also fixed in all three directions, so the springs are unloaded during the simulation. In the case of simulation f), a table was created which moved the nodes on each top roll to the specified gap in the beginning of the simulation. This meant the springs were loaded throughout the simulation.

The outcome of these studies helps to choose the most cost-effective strategy for tool deflection modelling, after the gap is calculated. In future applications of tool deflection in roll forming simulations, it is believed that the best course of action is to develop a script which, after the first increments where the sheet is in contact with a new station, the position of the roll is fixed at the calculated gap in order to reduce the computational effort.

Figure 71 shows the total CPU time for each of the simulations. It is interesting to note that the rigid model without gap between sheet and rolls takes the longest time to run. Among the simulations with a fixed gap, it is noticeable that all of them are significantly faster than the rigid simulation without gap. It is also possible to observe that, all other things being equal, a simulation with a gap between the sheet and rolls runs faster than a simulation with roll gap equal to sheet thickness.

Extrapolating from the results in section 4.8, if a stiffer spring is used, the simulation time reduction from fixating the roll gap after the initial gap calculation will be higher, making this approach even more necessary.

![Figure 71 - Analysis of the total CPU time for each model developed for the time optimization of the deflection simulation.](image-url)
Comparing the results of the rigid model with the position controlled gap model, one can observe that all the components of the calculation time are faster (Figure 72). This could mean that generally the computation time could be lowered through the adoption of a set gap between the sheet and the rolls. Further proof of this is the total number of cycles and separations (Figure 73), which is once again higher in the case of the rigid model with no gap between the sheet and the rolls.

**Figure 72 - Analysis of the components of calculation time for each model; even though the actual time in the solver is highest for the model with springs, the rigid model with no gap takes a longer CPU time to run.**
Figure 73 - Analysis of the total number of cycles and separations for each model.
5 Conclusions and discussion

The objective of this work was to develop an FEA model of the roll forming simulation which included the deflection of the tooling (rolls). To that effect, the approach was deemed successful; the measured forces for the experimental setup match the data collected from the performed simulations, and the deflection values are similar (Figure 74 and Figure 75). In any case, the force values obtained on the simulations with deformable tooling closely mimic the experimental behaviour of the line during forming, which is an improvement over the rigid simulations which are used presently. This can lead to a better understanding of the machine behaviour during roll forming and ultimately allows for a more precise mechanical design, adapted to the load conditions expected from the simulation. Furthermore, the force values in simulations with different stiffness are not very different between themselves; thus, inclusion of machine stiffness data in the simulation is an advantage in terms of force calculation even when the precise machine stiffness is not known.

Figure 74 - Comparison between the simulation results and the experimental results obtained in the experimental roll forming line.
The deflection results for the simulations and for the experimental line show that the stronger stiffness used (83100 N/mm) is the closest to the experimental results (Figure 75). Considering that the stiffness value of 83100 N/mm was the one obtained from the structural FEA analysis of the roll forming stand, it is believed that this method of obtaining the stiffness data of the machine for use in the numerical model is viable and can be recommended.

![Deflection graph](image)

Figure 75 – Comparison between the simulation results and the experimental results obtained in the experimental roll forming line; the measured deflection in the experimental setup seems close to the simulation with a stiffness of 83100 N/mm.

The proposed numerical model for industrial application, considering also the computational expense of each simulation, is one which would use springs on the top roll for the first increments when the sheet is in contact with a new station, and a new loadcase after the gap has stabilized which would use the previously calculated deflection value for moving the top roll to its deflected position and fixing it. This would combine the advantage of the deflection model (stability, load calculation and accuracy of sheet behaviour) with the lower CPU time of the fixed gap model. However, only the implementation of the stiffness model allows for a precise determination of the loads and therefore the gap, which is crucial for then knowing what gap to implement on the fixed gap model.

Accurate setup of the roll forming line to be analyzed is also of importance. The preexisting clearances that allow relative movement of the rolls make the calculation effort futile: if the real world conditions are not the same as the ones modeled in the simulation, the results will almost always be found lacking.
The relevance of this model is directly related to the growing implementation of high strength steel as a roll forming material. Currently, rigid simulations are found to be accurate for most materials and roll forming designs, and the applicability of a simulation with deformable tooling must be assessed prior to developing the modelling effort needed.

It is believed this modelling approach to roll forming simulations, including deformable tooling, could be of importance in the near future in the case of flexible roll forming. This process employs complex roll forming stands which are moveable. This, together with the fact that this process is often considered for truck chassis members and other parts with high thickness and strength, contributes to a lower stiffness and higher propensity to unwanted deflection of the rolls, which in turn can lead to unforeseen defects. Considering that the development of Flexible Roll Forming applications is highly dependent on the accuracy of the underlying simulations, accurately modeling the deflection of these stands can be key to a better understanding of this rather new process. Measurements performed by the author on a prototype flexible roll forming line have shown a considerable deflection in the top roll during forming. Further study of this phenomenon may lead to the application of deformable tooling to flexible roll forming simulation as a standard procedure.

5.1 Future work

The work developed in this project gave an insight into the behaviour of the roll forming line during the process. In terms of the roll deflection, an analysis of the effect of leg height, material thickness and strength is needed to understand in which conditions is the application of deformable tooling important to the overall process simulation. Furthermore, a robust model should be developed which combines deformable rolls with a fixed gap setup for optimization of the computational time. It would be interesting to analyze the machine deflection in an industrial roll forming line, in order to get experimental stiffness values and in order to observe the interaction between stands, which was not considered in the present work.

The investigations about the reduction of computational time must be expanded in order to validate the findings of the present work. For example, the inclusion of a relief angle in the rolls was not considered; a relief angle reduces the number of separations, which in turn reduces the number of cycles. Additionally, different discretization strategies can be used (for example, using a finer mesh in a section of the sheet) to understand the impact of different mesh sizes on the simulation time. Also, more complex profiles, and a higher number of stations, can be used to verify if the results obtained using a simple U-channel are valid for more complex models.

Furthermore, simulations with different degrees of freedom of the top and bottom rolls should be developed, for example including angular displacement of the rolls in the z direction (forming direction) and using beam elements to simulate the real shaft stiffness. It is also considered important to study the deflection of roll forming stands...
in the case of asymmetric profiles and forming strategies, in order to understand the linear and angular displacement of the rolls in such situations.

Concerning the experimental setup for the measurement of the forming forces, it is thought that the load cells are not the most adequate method of determination of the forming forces in the context of machine deflection, as they introduce an extra element in the stiffness of the system that is not present in the industrial machinery. The next steps in terms of the investigation of machine deflection should be to adopt strain gauges as a standard measurement procedure, as they allow for the direct measurement of each component of the machine and its behaviour. A different experimental stand design must be investigated which allows for the introduction of strain gauges in each component, especially the ones which simulations show as more prone to deflection.

The application of deformable tooling simulations to flexible roll forming must be studied, paying attention to the specificities of this process, in order to understand the relevance of including roll displacement in the simulations.

The development of an experimental procedure for determination of the machine stiffness using available workshop tools and materials, as a complement or an alternative to the structural FEA analysis of the roll forming stands, is considered key to the success of this simulation model in an industry setting.
References


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