New Generation Nip Equipment

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Preface

The objective of this Master thesis is to replace a hydraulic cylinder with a servomotor solution. The need for this project has arisen due to the need for increased precision in future development of the laminator. I was offered this project as a master thesis project by Peter U Larsson and Lena Skytte at Tetra Pak®. The contact was made through my good friend Emanuel Karlsson who is also an employee at Tetra Pak®.

This project concludes my studies at Lund University at the Faculty of Engineering and my Master of Science in Mechanical Engineering. The thesis has been performed within the Packaging Material unit at Tetra Pak®, Lund. There are many people who contributed to this project, for their help and support I am very grateful.

I would like to thank Tetra Pak® for the opportunity to do my thesis here and for the invaluable experience it has given me. Working this close with the company has given me insight into working life and how my future hopefully will look like. Many thanks to Peter U Larsson, Lena Skytte and Per Engvall for entrusting me with this project and for the support/feedback they provided during my work.

I would also like to thank my mentor Håkan Leijon for all the help I have received during this time, our discussions and his feedback was vital for the making of this project. Many thanks to Emanuel Karlsson as well, for supplying me with the contact information for this thesis, providing technical/mechanical assistance and for being my moral support.

I am also very grateful for the help I have received from SKF, Exlar and Lesjöfors. Especially Jörgen Svensson and Per-Erik Andersson at SKF who have been very helpful and contributed in the solutions and technical support of this project, thank you.

Lund, May 2010

Joel Nordin

Abstract

In the laminator there are three laminator stations where protective layers are applied to the paperboard. In each station there are several rollers which help perform the lamination process, the most important rollers to be considered in this thesis are the nip roller and the chill roller. The nip roller is pushed towards the chill roller using hydraulic cylinders , thus creating a narrow contact surface between these rollers called "the nip". It is in the nip that the different additives are applied onto the paperboard (or rather the 'web' which is the printed paperboard when loaded into the laminator). Some additives are melted onto the paperboard while others are applied in a solid state (foil or film). When the additives are applied onto the web they are cooled while in contact with the chill roller and then fed to the next laminator section.

Even though the hydraulic cylinders satisfies the specifications for today they will probably not be able to handle the demands for future operation. With increased need for precision the hydraulics simply cannot reach the same level of precision as a servomotor operated solution. To help clarify the project mission the problem is decomposed into three sub problems: linear motion, elasticity and motion guiding. By searching for concept solutions to these problems both externally and internally a number of concepts are created, the existing motion guiding system is kept. From these the most potent concepts are chosen using target specifications and a rating system. After further evaluation the final concepts are selected; for the solution to the linear motion problem the electromechanical cylinder is chosen and the elasticity problem is best solved with a disc spring package. Performing the necessary analysis and calculations on these solutions concludes the concept selection phase.

Keywords:

Electromechanical cylinder

Servomotor

Disc spring package

Roller

Line load

Lamination

Sammanfattning

Laminatorn är en maskin vars syfte är att belägga kartongmaterial med skyddande barriärer för att tillgodose hållbarheten för förpackningens innehåll. Laminatorn är uppbyggd av laminatorstationer där barriärer påföres förpackningsmaterialet i olika steg. I varje station finns ett antal valsar som hjälper till med barriärernas påförande, de som är intressanta i denna uppsats är nypvalsen och kylvalsen, det är mellan dessa två som "nypet" skapas. I detta nyp beläggs kartongmaterialet med olika tillsatsmaterial i både smält och fast tillstånd enligt förpackningsmaterialets specifikation. Dessa tillsatser appliceras under högt tryck som byggs upp genom att trycka nypvalsen mot kylvalsen med hjälp av hydraulcylindrar.

I änden på varje hydraulcylinder ansluts en ackumulator, dess uppgift är att agera dämpare för systemet genom att ackumulera hydraulvätska i det fall då en överlast uppträder i systemet och tillåter då kolven i hydraulcylindern att backa. Nypvalsen är alltså rörlig och nypet kan öppnas och stängas genom att flytta nypvalsen bort från eller mot kylvalsen, nypvalsen har ett ytskikt av mjukare kvalité för att öka kontaktytans area. Kylvalsen är en motordriven vals vars uppgift är att kyla den heta smältan som appliceras på kartongmaterialet.

Målet med detta examensarbete är att hitta en servomotorlösning som ersätter befintliga hydraulcylindrar i laminatorn. En ersättare till hydraulcylindrarna är önskvärd eftersom framtida specifikationer kan bli svåra att uppnå med den existerande hydrauliken. Med servomotorer är det möjligt att uppnå bättre kontroll och precision i driften.

För att strukturera arbetet används en produktutvecklingsmetodik hämtad från boken *Product Design and Development* skriven av Ulrich och Eppinger. I detta projekt används de tre första stegen produktspecifikation, generering av koncept och val av koncept eftersom tidsbegränsningen på 20 veckor inte tillåter vidare utveckling av konceptförslaget.

För att förstå vad som krävs av enheten är det viktigt att göra analyser av det existerande systemet. Dessa analyser görs i huvudsak på ackumulatorn eftersom den utgör ett slutet system och därför kan diverse lagar och regler appliceras vid uträkningarna. I dessa analyser framkommer att hydraulsystemet riskerar att skada kylvalsen för vissa fel i processen.

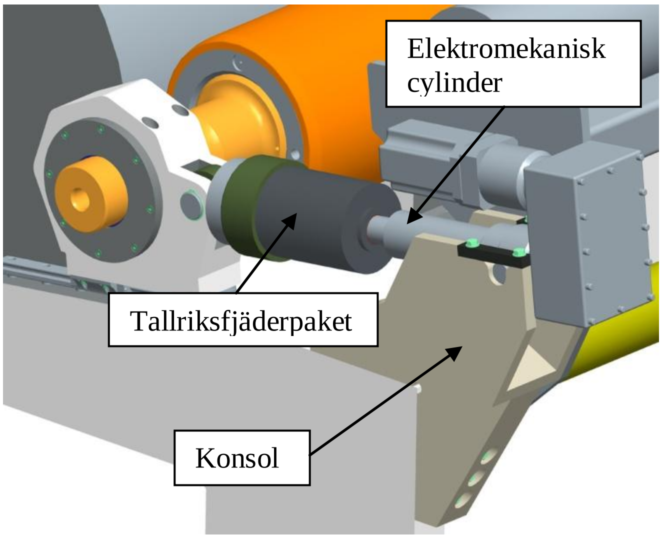
Det finns olika faktorer som påverkar nyptrycket och inducerar kraftökningar i systemet, däribland en överlappande skarv som ger dubbel materialtjocklek i nypet under en viss tid. Det förekommer också fel i processen som exempelvis då hårda klumpar stelnad polymer faller ner i nypet. Ett större fel är exempelvis ett banbrott som kan resultera i upprullning av förpackningsmaterial på kylvalsen, såkallat "*wrap-around*" som kan ge åtskilliga lagers tjocklek i nypet med påföljande kraftstegring. Sådan upprullning av förpackningsmaterial kan i värsta fall dessutom innebära att det ozonrör som hänger ovanför nypet dras ned i nypet av en lös ände förpackningsmaterial. Analyserna visar att mindre avvikelser ger små eller försumbara kraftökningar medan större fel såsom *"wrap-around"* kan ge laster större än vad som är tillåtet för kylvalsen. Det är viktigt att i en framtida lösning ta hänsyn till sådana händelser för att eliminera skador i systemet.

När analyserna är gjorda är det dags att ställa upp en lista över viktiga egenskaper, som bestämmer hur en kommande lösning ska se ut. Dessa egenskaper rankas efter hur viktiga de är för lösningen, genom diskussioner med tekniska experter och sakkunniga inom företaget och resultatet av denna rankning ligger till grund för målspecifikationerna som bestäms. I målspecifikationerna ställs egenskaperna upp i den ordning de blivit rankade, med den viktigaste överst. Sedan förses varje egenskap med ett idealt värde, det värde en önskvärd lösning ska uppfylla. I de flesta fall utgörs dessa värden av kraven som är satta på projektet men i de fall det inte går att sätta ett värde så benämns det som en konstruktionsvariabel och lämnas att tas omhand med hjälp av god konstruktionsförmåga (exempelvis då produkten ska skyddas mot väta eller ozongas). Målspecifikationerna ligger sedan till grund för det urval som görs efter att konceptförslagen tagits fram.

För att underlätta framtagningen av konceptförslag delas problemet upp i delproblem, i detta fall tre stycken: *linjär rörelse*, *elasticitet* och *styrning av rörelse*. För delproblemet *styrning av rörelse* bestäms att den existerande lösningen ska behållas i den mån det är möjligt. Framtagning av konceptförslag för delproblemen sker genom diskussion och sökning internt respektive externt. Intern sökning genomförs på företaget där olika personer konsulteras och existerande maskineri inspekteras, de externa sökningarna görs på internet och genom kontakt med olika tillverkare.

När ett antal konceptförslag tagits fram görs ett första urval för att eliminera de förslag som inte lever upp till specifikationerna, detta urval görs genom rankning av hur väl konceptet uppfyller en viss egenskap. Efter ett första urval återstår det två konceptförslag för *linjär rörelse*: en elektromekanisk cylinder (planetrullskruv som omvandlar ett vridmoment till en tryckande/dragande kraft genom en lösning liknande den för mutter och skruv) och ett linjärbord (en linjär skenstyrning med integrerad drift i form av en skruv). För delproblemet *elasticitet* går tre konceptförslag igenom första urvalet: servomotorstyrning (direkt kompensering i servomotorn för lastförändringar), tallriksfjäderpaket (tallriksfjädrar staplas mot varandra för att klara kraft/utböjning förhållandet) och brytpinne (en pinne som konstrueras för att brytas då en viss last överskrids).

Vid djupare undersökning visar det sig att linjärbordet och servomotorstyrning inte är möjliga lösningar på grund av lastkapacitet och reaktionstid. Dessutom väljs lösningen med brytpinne bort eftersom den inducerar svårkontrollerade variabler i systemet, såsom materialegenskaper och friktionsproblem. Kvar står den elektromekaniska cylindern och tallriksfjäderpaketet som alltså är de slutliga konceptförslagen. Genom kontakt med tillverkare tas specifikationer fram för dessa förslag och de kan nu ställas upp i den slutliga specifikationen. För att göra infästning möjlig konstrueras en konsol som tillgodoser de belastningskrav som finns på lösningen. Konsolen samt en axel från tallriksfjäderpaketet analyseras med hjälp av mjukvara som beräknar hållfastheten genom att applicera finita elementmetoden på de olika delarna.



Figur: Konceptlösning

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1 Introduction

Tetra Pak® Packaging Solution has a thesis on how to further develop the nip mechanism in the laminator station. The basic idea is to replace a hydraulic solution with an servomotor equivalence. The main reason for this change is to get better precision in the nip to be able to cope with future specifications.

1.1 Background

As one of the world's largest carton package manufacturer, Tetra Pak® handle all processes from the raw paperboard to the filled package found in the retailer's store. In the company's factories paperboard is processed into packaging material which is used in filling machines at the local food and/or beverage producers. The laminator is the machine that applies the protective layers onto the large rolls of printed and creased paperboard.

1.2 The company

Ruben Rausing founded Tetra Pak® in 1951. His idea was to produce carton packaging for food and beverages. This approach was very successful and today the company has expanded all over the world and is a well known brand for most people. Tetra Pak® has since then expanded their field of knowledge to include a wide variety of packaging products for consumables like cheese, fruit and wine. Apart from packaging, Tetra Pak® is also involved in treating and distributing the food in a healthy manner. Tetra Pak® is a member of the Tetra Laval® group, this group has two other members: De Laval and Sidel. De Laval, with their base in Sweden, produces equipment for milk production and is well known for their automatic milking system VMS. The French company Sidel develops and manufactures plastic bottles machinery. This thesis is performed at the department of Packaging Material in Lund, Sweden. Lund is the home of Tetra Pak® and the place where most research and development is done. Besides the development and engineering unit (D&E) there are also large facilities for packaging manufacturing and assembly of packaging machinery, in fact these divisions are the largest facilities throughout the whole company.

1.3 The nip unit

The laminator consists of several sections with different functions, this thesis focuses on the laminator station and especially the nip which is the contact surface between the nip roller and the chill roller. The nip roller is pushed against the chill roller with hydraulic cylinders and the lamination additives are applied in the nip that is created between these rollers. When all layers have been applied to the printed and creased paperboard it is once again stored on a roll. In the next step this roll is sent to the finishing section where it is slit into narrower rolls and possible defects are removed. After being loaded onto a pallet it is ready to be delivered to the filling machine at the food or beverage producer.

1.4 Objectives

The objective of this thesis is to replace the hydraulic cylinders in the nip unit with an electric solution. There are a number of disadvantages with a hydraulic system, low precision and leakage of hydraulic fluid being two of them. Though the hydraulic system works satisfactory in the present situation, future specifications will be difficult to reach with this system. Precision in the nip is the most important factor that has lead to the formulation of this thesis. Aside from keeping a constant load between the rollers, the system must also be able to handle splices as well as occurrence of solid laminate fragments, wrap-around that can occur in case of web break and other deviations from normal operation. With the hydraulic cylinders these deviations are easily handled with the assist of pressurized accumulators. To take care of these variations in a rigid servo application it is crucial to have some cushioning or release system. Thus, the objective must be expanded to include a mechanism to absorb variations in the lamination process that otherwise would damage expensive equipment.

1.5 Limitations

The thesis spans over a period of twenty weeks, since this include writing the report and prepare the presentation the development process is shorter than that. With regards to this time limit the main focus will be on dimensioning and modelling the unit while taking surrounding issues into consideration. The ambition is to have a complete unit specified by the end of the thesis, if this is not possible there will be trade-offs regarding drawings, automation, software and other parts that are not essential in the completion of this thesis.

2 Development methodology

The steps in a structured development project are well described by Ulrich and Eppinger[1] and these steps form the layout of this thesis. With their methodology the product development process is broken down into parts and becomes more understandable and easier to work with.

2.1 Product specifications

Basically the specification contains metrics (properties) that are derived from user demands. These metrics and corresponding values represents the performance of the unit and describes *what* the product has to do but not *how* to do it.

2.1.1 Target Specifications

In the ideal case, these values are the criteria for creating the concept solution. The target specifications must now be reworked so that every value is realistic, this is done through trade-off's where it is possible but also by consulting specialists and other resources.

2.1.2 Final specifications

All demands for the solution are now set and the concept generation can begin with a starting point in the specifications. The concepts that can't fulfill the specifications must either be improved, combined with others or, as the last option, be disregarded.

2.2 Concept generation

With help from the specifications it is now time to start looking at solutions for the problem. This step is the most important event in the product development process and also the most time consuming part. To address this rather wide area it is recommended to divide it into smaller constituents as follows.

2.2.1 Clarify the problem

To be able to handle a complex problem it is sometimes necessary to break down the entire problem into smaller parts, a sort of decomposition. By solving these smaller sub problems, the final result is often better and solutions that would otherwise be disregarded, or not even discussed, are lifted and taken into account.

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[1] Product Design and Development - Karl T Ulrich, Steven D Eppinger - Third edition

2.2.2 Search externally

In order to find already existing products and solutions an external search is necessary. This search provides input from lead users and technical information from suppliers. Studies of technical journals and consulting experts in relevant areas are also important steps to help further concept generation.

2.2.3 Search internally

A search within the company or project group to make sure that all ideas are considered, this step is often more open than the other because the knowledge is already present inside the group. A common platform for making an internal search is a brainstorming session where all ideas are pronounced in a non-analyzing environment where both impossible and realistic solutions are considered.

2.2.4 Explore systematically

After searching for solutions to the problem or any of the sub problems it is important to organize and synthesize these fragments so that complete solutions may be found. By creating a concept classification tree it becomes easier to see which solutions that can be combined and those that cannot.

2.3 Concept selection

Finding the most potential solutions is sometimes a difficult process but with the tools provided it can be done easier and with a better end result. Concept scoring is helpful when making the selections and will be used in this thesis. There is another similar method called concept screening which will not be used.

2.3.1 Concept scoring

By rating the concepts in relation to a reference solution (in most cases a competing product or a present solution) the selection process becomes less complex. The scoring also facilitates comparisons of combinations of different concepts but also improvements in general, to be able to present the best solution possible.

2.3.2 Concept presentation

Though the concept presentation is not standard procedure according to Ulrich & Eppinger, this thesis requires the involvement of the persons in charge of the development. So when the best concepts have been selected, these are presented to the board members. They are required to rate the concepts in a similar way that was done in the previous section.

2.3.3 Final concepts selection

With the input from the board members, it is time to select one or two concepts that will be further developed. With these final concepts in hand the time has come for the final design to begin.

3 Detailed description

The laminator is a machine that produces packaging material by adding different protective layers onto the printed paperboard or web as it is called when being processed. In every laminator there are laminator stations and it is in these stations that the various additives are applied. To ensure good packaging performance these layers are applied in a high pressure contact surface between two rollers, this is called the nip.

3.1 The nip unit

The nip is created between the nip roller and the chill roller where the melted polymer is distributed, the pressure is achieved by pushing the nip roller against the chill roller with the help of hydraulic cylinders, see figure 3-1. After being applied to the web the polymer is cooled while in contact with the chill roller and then delivered to the next laminator station or the rewinder unit.

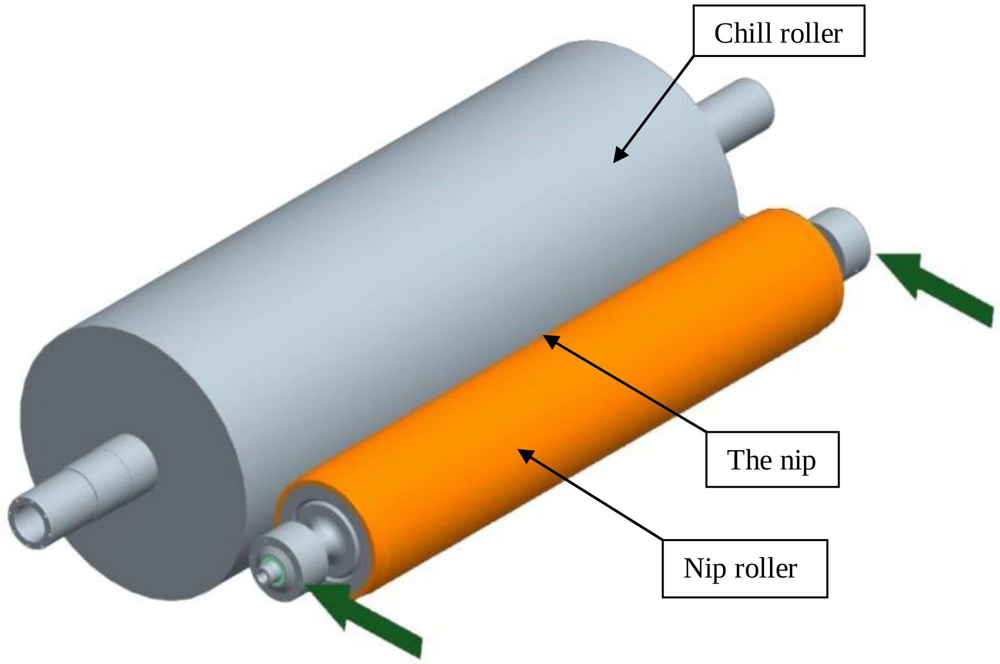

Figure 3-1: The nip unit

3.1.1 The nip roller

To ensure the quality of the package material it is necessary to control the way in which the protective layers are applied to the paperboard. The nip roller is a steel roller with a soft coating to increase contact surface area. It rests in bearing housings at both ends which are connected to hydraulic cylinders with attached, gas filled, accumulators

3.1.2 The chill roller

The chill roller has the main functions of cooling any hot additives applied, it is also driven and helps drive the paperboard or 'web' through the machine. This roller has a very fine surface along with internal cooling channels to exert the best cooling performance possible. The cooling channels weakens the structural integrity of the chill roller and since it is also a very expensive roller it will be a dimensioning factor when calculating load between rollers.

3.2 Nip pressure

To ensure that the additives are firmly in place the procedure of applying these layers is done under constant high pressure. To be able to reach the specified line load (load between the nip roller and the chill roller) the hydraulic cylinders must exert a force in both ends of the nip roller that corresponds to several tons of weight. This load magnitude requires high stability and rigid mounting which can be seen in the mechanical design of the whole nip unit.

3.3 Nip roller movement

The nip roller is mounted on profile rail guides and is operated by the hydraulic cylinders for closing and opening the nip. This motion is done mainly at start up and shutdown of the process but also for maintenance and if a process error occurs. In case of routine maintenance or process error the nip will be opened manually by the operator.

4 Problem formulation

To better understand the need of this project the advantages and disadvantages are listed for the hydraulic solution and the servomotor solution respectively.

4.1 Advantages and disadvantages - Hydraulics

The hydraulic cylinder has many advantages that must be compensated for when replaced. There are also disadvantages which describe the need for change. The most important factors:

\+ Compact:

Since the hydraulic cylinder is placed away from the hydraulic unit, the space required for this solution is minimal. However, this also means that the hydraulic unit must be placed elsewhere leading to additional space requirements in the facility.

\+ Accumulator

This gas damper makes load regulation easy at a low space impact. The disadvantage of the accumulator is the restrictions in the hydraulic system between the cylinder and the accumulator which renders the system rather stiff for fast load changes in comparison with a spring system.

\+ Pressure regulation

Nominal pressure is set directly on the hydraulic unit and kept for as long as the lamination process is active.

– Hydraulic fluid

Even though modern systems have less chance of leakage and the risk of hydraulic fluid ending up on the packaging material is negligible it can cause problems in other areas (slippery floors, fluid covered components etc.).

– Precision

To meet future demands the hydraulic cylinders will have problem reaching the precision needed. A servomotor solution can fine tune the process in a more detailed way than the hydraulics.

– Tubes and hosing

The space required for hydraulic pipes and hoses are generally larger than for the electric setup, the possibility of leakage must also be considered.

4.2 Advantages and disadvantages - Servo

The major advantages and disadvantages of the servomotor solution are as follows.

\+ Precision

The precision in a servo motor is superior to most alternatives and is one of the main reasons for choosing this solution. By choosing different gear ratio for the servo motor it is possible to further improve precision up to the point where the other components are not sufficiently precise. Changing gear ratio has, of course, a large impact on speeds and forces and therefore creates a number of possibilities to adapt the solution for the actual demands.

\+ Control system

The ability to control the motion with software programming is a big advantage and makes the process controllable in every detail. Since the system also gives feedback from the operation it can be used to detect certain errors and compensate for them. By utilizing these additional features it is possible to gather more exact data and to be able to further analyze the process.

\+ Stand-alone product

With the exception of some wiring and the controller unit the solution is mounted as one unit with no external components, which makes exchange and maintenance swift and easy.

\+ Clean system

By using an electrical unit the system will not be dependent on hydraulics and/or pneumatics and it also removes the sources of leakage.

– No internal elasticity

Since the system is rigid in its construction the system will not be able to handle vibrations and deviations in thickness of the material. The servomotor will not compensate feedback forces by moving in either direction because of its internal design. This, however, is a necessary disadvantage since it permits exact adjustment and good precision.

– Large, heavy unit

Since all parts of this solution will be assembled into one unit, it will be significantly larger and heavier than the hydraulic cylinder.

4.3 Paper splice, wrap-around and other errors

Overlap splice and solid polymer pieces are two examples of variations that induce small load changes in the nip. An example of a larger deviation is a web break that can cause wrap-around on the chill roller. This means that the chill roller will wind up several layers of material and the pressure between nip- and chill roller will increase dramatically before the emergency system is able to stop the operation.

The worst case scenario is when the free end of the web after a web break will drag an aluminium pipe, hanging above the nip, down into the nip.

5 Analysis of hydraulic system

These analysis are the foundation of the design work, the results determines how the solution must function. Since the geometry is symmetrical, calculations are done for one side of the laminator station only. In the first section the data used in the calculations is presented along with the sources of information. The different factors and properties are denominated to be more easily handled. In the second section the accumulator and its function are described along with some calculations of limitations. In section three there are analysis of how wrap-around and overlap splice affects the pressure between nip- and chill roller. The final section briefly presents the case when the ozone pipe enters the nip.

5.1 General data

Calculations made in the following sections are based on data from the hydraulic cylinder and the gas accumulator. Since this solution is handling the deviations in an acceptable way it is logical to use these properties as a reference when designing the new unit. The following data has been collected for the paperboard, accumulator, hydraulic cylinder and the laminator, see table 5-1. All values are upper extreme values and are safe to use as dimensioning factors and are gathered from either manufacturers' datasheet[2] or from project specification. The stroke equivalent for the hydraulic cylinder is based on the specifications for the new solution and is there to be able to perform a comparison between the existing hydraulic cylinder and the new servomotor solution.

[2] www.pmchytech.com 2010-01-18

Table 5-1: General data

Data	
Accumulator - SBO210-0.75E1/112U-210AK	
V_a = Volume (m^3) [1]	0,00075
L_p = Preloaded accumulator (N/mm) [2]	30
Hydraulic Cylinder - ISO 80/56-160MT1	
$Ø_p$ = Piston diameter (m) [1]	0,08
$Ø_r$ = Rod diameter (m) [1]	0,056
A_c = Cylinder - cross section area (m^2) [1]	0,00503
s_c = Cylinder, stroke (m) [1]	0,16
s_{ce} = Cylinder, stroke equivalent (m)	0,06
Laminator	
L_{op} = Operational line load (N/mm) [3]	50
$Ø_c$ = Chill roller diameter (m) [3]	0,9
L_s = Maximum permitted line load (N/mm) [3]	70
v_w = Web speed (m/min) [3]	800
t_s = Stop time, web break (s) [2]	10
h_p = Paperboard thickness (m) [2]	0,00055
w_m = Width (m) [3]	1,65

Data provided by:

[1] PMC Hytech www.pmchytech.com [2010-03-17]

[2] Tetra Pak® documentation

[3] Skytte, Lena, 2009, New generation nip equipment specification_Issue 3

5.2 Accumulator

In the hydraulic system there is an accumulator acting as a spring to add elasticity to an otherwise stiff system. The accumulator is placed in the back end of the hydraulic cylinder and will accumulate hydraulic fluid that is displaced when pressure increases in the nip. The nitrogen and the hydraulic fluid are separated by a diaphragm which also keeps the gas from escaping through the bottom connector while disconnected and when system pressure is below the preload pressure.

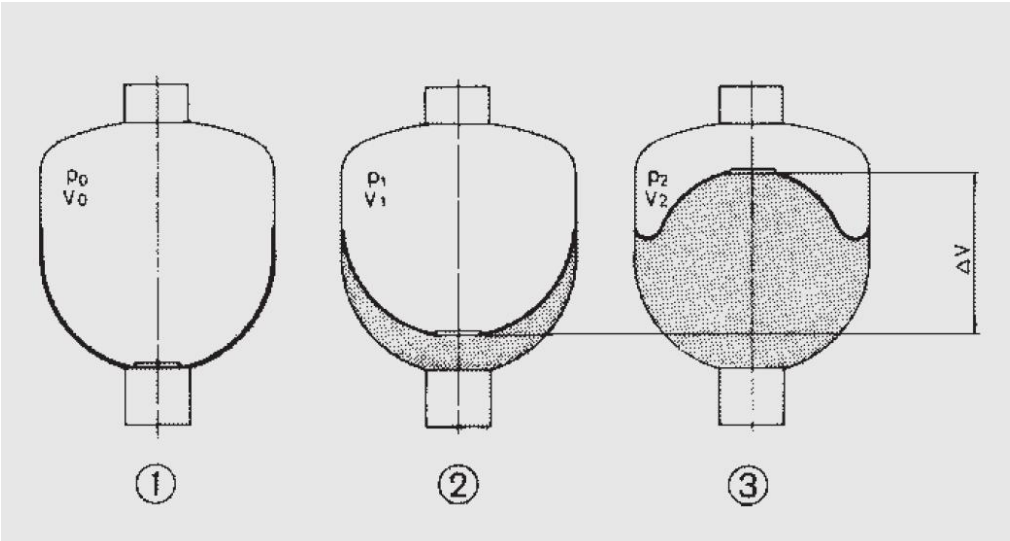

Figure 5-2: Accumulator

In figure 5-2 the first state corresponds to a line load below the equivalent preload pressure or a completely unloaded system, in the second state the load is just above preload pressure and hydraulic fluid has filled up some regions of the accumulator. Finally, in the third state, the pressure increased significantly and hydraulic fluid now constitutes the major volume in the accumulator.

Using data provided in table 5-1 some calculations can be made to determine the limitations of the accumulator, see table 5-3.

Table 5-3: Calculations I

Calculations I	
Accumulator - SBO210-0.75E1/112U-210AK	
F_p = Preload (N)	24750
P_p = Preload pressure (MPa)	4,92
F_{op} = Operational load (N)	41250
P_{op} = Operational pressure (MPa)	8,20

The preload F_p is the load that correspond to the specified line load equivalent L_p.

$$F_p - Preload\ (N) = \frac{L_p \cdot w_m}{2} = \frac{30 \cdot 10^3 \cdot 1{,}65}{2} = \mathbf{24750\ N}$$

$$P_p - Preload\ pressure\ (MPa) = \frac{F_p}{A_c \cdot 10^6} = \frac{24750}{5030} = \mathbf{4{,}92\ MPa}$$

$$F_{op} - Operational\ load\ (N) = L_{op} \cdot (w_m \cdot 10^3) = \mathbf{41250\ N}$$

$$P_{op} - Operational\ pressure\ (MPa) = \frac{F_{op}}{A_c \cdot 10^6} = \frac{41250}{5030} = \mathbf{8{,}20\ MPa}$$

To better understand what happens in terms of forces and pressures when chill roller wrap-around or overlap splice occurs the amount of material accumulated is described as an increase in thickness between nip roller and chill roller. This change in thickness can then be seen as a pressure increase in the hydraulic cylinder or a force increase in the piston rod.

By using Boyle's law[3] (polytrophic changes of state) that applies for systems where there are slow expansions/compressions and small or no energy loss it is possible to calculate every step of the process. This law states that the volume-pressure relation will be constant, thus $P1 \cdot V1 = P2 \cdot V2$. Since the hydraulic fluid can be considered incompressible the change in volume will occur in the accumulator which is filled with nitrogen but the pressure change is distributed throughout the system.

Table 5-4: Pressure-Volume I

Pressure - Volume I	
a. System at 30N/mm ► System at 50N/mm	
$P1_a$	4,920477137
$V1_a$	0,00075
$P2_a$	8,200795229
$V2_a$	0,00045
b. System at 50N/mm ► Maximum stroke	
$P1_b$	8,200795229
$V1_b$	0,00045
$P2_b$	24,90120009
$V2_b$	0,0001482
c. System at 50N/mm ► Maximum safe stroke	
$P1_c$	8,200795229
$V1_c$	0,00045
$P2_c$	11,48111332
$V2_c$	0,000321429
ΔV_c	**0,000128571**

[3] New World Encyclopedia www.newworldencyclopedia.org [2010-04-13]

a.

In the first load case the system will go from preloaded accumulator state to normal operation load state, see table 5-4. In the preloaded state the volume equals the specified volume of the accumulator $(V1_a = V_a)$ and the pressure as described in table 5-3 at preload $\left(P1_a = P_p\right)$. In normal operation state the pressure is raised to the equivalent to a line load of 50N/mm $\left(P2_a = P_{op}\right)$ whilst the compressed accumulator volume will be calculated through:

$$\mathrm{V2_a} = \mathrm{V_{op}} = \frac{\mathrm{P1_a} \cdot \mathrm{V1_a}}{\mathrm{P2_a}} = \frac{\mathrm{P_p} \cdot \mathrm{V_a}}{\mathrm{P_{op}}} = \frac{4{,}92 \cdot 0{,}00075}{8{,}20} = \mathbf{0,00045\ m^3}$$

The compressed volume:

$$\Delta V = V1_a - V2_a = V_a - V2_a = 0{,}00075 - 0{,}00045 = \boldsymbol{0,0003\ m^3}$$

$\mathrm{V2_a}$ is the normal operating volume for the accumulator since it is the gas volume at 50 N/mm, it is denominated $\mathrm{V_{op}}$ and is found in table 5-5 below.

b.

In case the full stroke of the cylinder is accumulated starting from normal operating mode $\left(P1_b = P_{op}\right)$ while the gas volume will be compressed from normal operating volume $\left(\mathrm{V1_b} = \mathrm{V_{op}}\right)$ to the remaining gas volume at full cylinder stroke ($\mathrm{V2_b} = \mathrm{V_{max}}$), see table 5-4. $\mathrm{V2_b}$ is calculated by taking the normal operating volume $\mathrm{V_{op}}$ and subtracting the cylinder cross section area $\mathrm{A_c}$ times the cylinder equivalent max stroke $\mathrm{s_{ce}}$:

$$V2_b = V_{max} = V_{op} - A_c \cdot s_{ce} = 0{,}00045 - 0{,}00503 \cdot 0{,}06 = \boldsymbol{0,0001482\ m^3}$$

Now it is possible to calculate $\mathrm{P2_b}$, the pressure at max stroke which will be referred to as $\mathrm{P_{max}}$ in table 5-5:

$$P2_b = P_{max} = \frac{P1_b \cdot V1_b}{V2_b} = \frac{P_{op} \cdot V_{op}}{V_{max}} = \frac{8{,}20 \cdot 0{,}00045}{0{,}0001482} = \boldsymbol{24,90\ MPa}$$

Converted into a cylinder load:

$$F_{max} = P_{max} \cdot A_c = 24{,}90 \cdot 0{,}00503 \cdot 10^6 = \boldsymbol{125253\ N}$$

c.

In case the stroke length for the specified maximum force allowed is derived from the highest line load that the chill roller can withstand. Given the largest permitted line load L_{max} the corresponding safe load becomes:

$$F_s = \frac{L_s \cdot 10^3 \cdot w_m}{2} = \frac{70 \cdot 10^3 \cdot 1{,}65}{2} = \mathbf{57750\ N}$$

consequently the pressure:

$$P_s = \frac{F_s}{A_c} = \frac{57750}{0{,}00503 \cdot 10^6} = \mathbf{11{,}48\ MPa}$$

As the initial state is normal operation, the gas volume and the pressure are unchanged $\left(P1_c = P_{op}\ \&\ V1_c = V_{op}\right)$ and the final pressure $P2_c$ equals the maximum safe pressure P_s (see table 5-4), thus the resulting volume $V2_c$ becomes:

$$V2_c = \frac{P1_c \cdot V1_c}{P2_c} = \frac{P_{op} \cdot V_{op}}{P_s} = \frac{8{,}20 \cdot 0{,}00045}{11{,}48} = \mathbf{0{,}0003214\ m^3}$$

This is the gas volume at the permitted load, to be able to calculate the corresponding cylinder stroke the difference in volume is required, this is given by taking the accumulator operating volume V_{op} and subtracting the actual gas volume $V2_c$:

$$\Delta V_c = V_{op} - V2_c = 0{,}00045 - 0{,}0003214 = \mathbf{0{,}0001286\ m^3}$$

By dividing with the cylinder cross section area A_c the safe stroke s_s is obtained:

$$s_s = \frac{\Delta V_c}{A_c} = \frac{0{,}0001286}{0{,}00503} = \mathbf{0{,}0256\ m}$$

The results are summarized in the following table.

Table 5-5: Results I

Results I	
Accumulator - SBO210-0.75E1/112U-210AK	
V_{op} = Operational volume (m^3)	0,00045
V_{max} = Acc. Volume, maximum stroke (m^3)	0,0001482
P_{max} = Pressure, maximum stroke (MPa)	24,90
F_{max} = Load, maximum stroke (N)	125253
F_s = Maximum safe load (N)	57750
P_s = Maximum safe pressure (Mpa)	11,48
s_s = Maximum safe stroke (m)	0,0256

5.3 Wrap-around and overlap splice

When a wrap-around occur the material accumulated on the chill roller causes an increased pressure between chill- and nip roller. This increase in pressure results in hydraulic fluid being pushed back in the cylinder by the cylinder rod. This excessive fluid flows into the accumulator reducing the gas volume, thus building up pressure in the gas. By calculating the increase in pressure it is possible to see which loads to expect in the system when wrap-around or overlap splice occurs. First the basic data for the process and material must be calculated as seen in table 5-6.

Table 5-6: Calculations II

Calculations II	
Laminator	
a_b = Acceleration (m/s^2)	-1,33
v_b = Average speed, deceleration (m/s)	6,67
r_b = Revolutions before standstill (no.)	23,6
h_w = Wrap-around thickness (m)	0,026
ΔV_s = Volume, overlap splice (m^3)	2,77E-06
ΔV_w = Volume, wrap-around (m^3)	1,30E-04

Assuming linear deceleration the process has the following properties:

$$a_b - Acceleration\ (m/s^2) = \frac{0 - \frac{v_w}{60}}{t_s} = \frac{0 - \frac{800}{60}}{10} = \mathbf{-1,33\ m/s^2}$$

$$v_b - Average\ speed, deceleration\ (m/s) = \frac{v_w}{2 \cdot 60} = \frac{800}{2 \cdot 60} = \mathbf{6,67\ m/s}$$

$$r_b - Revolutions\ during\ deceleration\ (no.) = \frac{v_b \cdot t_s}{\pi \cdot \emptyset_c} = \frac{6,67 * 10}{\pi \cdot 0,9} = \mathbf{23,6}$$

$$h_w - Wrap\text{-}around\ thickness\ (m) = r_b \cdot h_p \cdot 2 = 23,6 \cdot 0,00055 \cdot 2 = \mathbf{0,026\ m}$$

$$\Delta V_s - Volume, overlap\ splice = h_p \cdot A_c = 0,00055 \cdot 0,00503 = \mathbf{2,77 \cdot 10^{-6}\ m^3}$$

$$\Delta V_w - Volume, wrap\text{-}around = h_w \cdot A_c = 0,026 \cdot 0,00503 = \mathbf{0,00013\ m^3}$$

With this basic information it is possible to once again use Boyle's law (as seen in previous section) to compute the changes in pressure and volume in the accumulator.

Table 5-7: Pressure -Volume II

Pressure - Volume II	
d. System at 50N/mm ► System, overlap splice	
$P1_d$	8,200795229
$V1_d$	0,00045
$P2_d$	8,251523763
$V2_d$	0,000447234
e. System at 50N/mm ► System, wrap-around	
$P1_e$	8,200795229
$V1_e$	0,00045
$P2_e$	11,54896619
$V2_e$	0,00031954

d.

The system will be subjected to an overlap splice, which actually means that a length of double paperboard thickness will pass through the rollers. In this case the initial state is when the system is in an operational state $\left(P1_d = P_{op} \,\&\, V1_d = V_{op}\right)$ (see table 5-7) and the change in volume for an overlap splice (ΔV_s) as calculated previously, this gives the volume, pressure and pressure change through:

$$V2_d = V1_d - \Delta V_s = V_{op} - \Delta V_s = 0{,}00045 - 2{,}77 \cdot 10^{-6} = \mathbf{0{,}000447\ m^3}$$

$$P2_d = P_{os} = \frac{P1_d \cdot V1_d}{V2_d} = \frac{P_{op} \cdot V_{op}}{V2_d} = \frac{8{,}20 \cdot 0{,}00045}{0{,}000447} = \mathbf{8{,}25\ MPa}$$

The corresponding force in the cylinder:

$$F_{os} = P_{os} \cdot A_c = 8{,}25 \cdot 0{,}00503 \cdot 10^6 = \mathbf{41505\ N}$$

e.

The second load case determines the pressure and volume change after a 10 s wraparound with breaks applied. The initial state is normal operation $\left(P1_e = P_{op} \,\&\, V1_e = V_{op}\right)$ (see table 5-7) while the final volume is calculated by using the change in volume for wrap-around (ΔV_w) to calculate the new volume $V2_c$ and pressure $P2_c$ as:

$$V2_e = V1_e - \Delta V_w = V_{op} - \Delta V_w = 0{,}00045 - 0{,}00013 = \mathbf{0{,}00032\ m^3}$$

$$P2_e = P_{wa} = \frac{P1_e \cdot V1_e}{V2_e} = \frac{P_{op} \cdot V_{op}}{V2_e} = \frac{8{,}20 \cdot 0{,}00045}{0{,}00032} = \mathbf{11{,}55\ MPa}$$

The corresponding force in the cylinder:

$$F_{wa} = P_{wa} \cdot A_c = 11{,}55 \cdot 0{,}00503 \cdot 10^6 = \mathbf{58091\ N}$$

The results are summarized in the following table.

Table 5-8: Results II

Results II	
Overlapping splice	
V_{os} = Volume, overlap splice (m^3)	0,0004
P_{os} = Pressure, overlap splice (MPa)	8,25
F_{os} = Load, overlap splice (N)	41505
Wrap-around (10s)	
V_{wa} = Volume, wrap-around (m^3)	0,0003
P_{wa} = Pressure, wrap-around (MPa)	11,55
F_{os} = Load, wrap-around (N)	58091

Depending on what time the new system requires to react on a wrap-around the amount of material accumulated on the chill roller varies, therefore it is interesting to look at how this amount varies over time. Material thickness versus time respective force versus time is plotted in the following graph by calculating the following equations.

The deceleration is based on the ten seconds specified stop time t_s as calculated before:

$$a_b - Acceleration\ (m/s^2) = -\frac{v_w}{t_s}$$

$$v(t) - Speed\ (^m/_s) = v_w + a_b \cdot t = v_w \cdot \left(1 - \frac{t}{t_s}\right)$$

$$v_m(t) - Average\ speed\ (^m/_s) = \frac{v_w}{2} \cdot \left(1 - \frac{t}{t_s}\right)$$

$$r(t) - Revolutions\ (no.) = \frac{v_m \cdot t}{\pi \cdot \emptyset_c} = \frac{v_w \cdot t}{2 \cdot \pi \cdot \emptyset_c} \cdot \left(1 - \frac{t}{t_s}\right)$$

$$h(t) - Wrap\text{-}around\ thickness\ (m) = r(t) \cdot h_p \cdot 2 = \frac{v_w \cdot t}{\pi \cdot \emptyset_c} \cdot \left(1 - \frac{t}{t_s}\right) \cdot h_p$$

Using Boyles law again, the pressure is derived:

$$P(t) - Pressure\ (MPa) = \frac{P_{op} \cdot V_{op}}{V_{op} - h(t) \cdot A_c} = \frac{P_{op} \cdot V_{op}}{V_{op} - \left(\frac{v_w \cdot t}{\pi \cdot \emptyset_c} \cdot \left(1 - \frac{t}{t_s}\right) \cdot h_p\right) \cdot A_c}$$

$$F(t) - Load\ (N) = P(t) \cdot A_c = \frac{P_{op} \cdot V_{op}}{V_{op} - \left(\frac{v_w \cdot t}{\pi \cdot \emptyset_c} \cdot \left(1 - \frac{t}{t_s}\right) \cdot h_p\right) \cdot A_c} \cdot A_c$$

Plotting the wrap-around thickness and force over time:

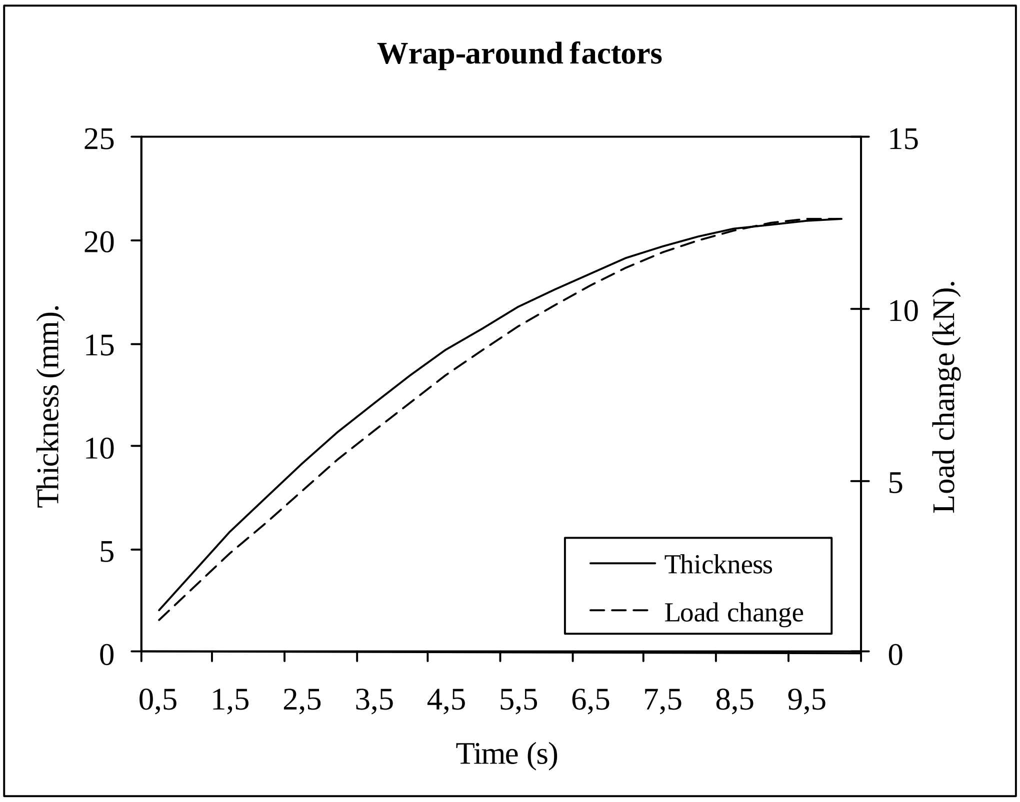

Figure 5-9: Wrap-around factors

5.4 Conclusions and reflections

From the point where the contact between nip- and chill roller is initialized the system line load increases up to 50 N/mm which is normal operation. The accumulator is preloaded to 30 N/mm which means that below this load it is unaffected and its volume constant.

Going from 30 N/mm to 50 N/mm the gas volume inside the accumulator will decrease from 0,75 l to 0,45 l leading to a pressure increase from 4,92 MPa to 8,20 MPa or the equivalent load increase from 24,75 kN to 41,25 kN. These are the figures for normal operation and only a control and a verification of the given data for operation.

The equivalent max stroke in the cylinder (60 mm) will result in a pressure of 24,90 MPa and a gas volume decrease in the accumulator down to 0,15 l. The force at this pressure will be 125,3 kN thus well beyond the 57,75 kN specified for the chill roller and therefore leading to severe damage to the chill roller. It is clear that the accumulator will never be completely filled (which means the full stroke of the cylinder can be utilized in an overload situation) but also that a full stroke probably leads to irreparable damage in the system.

To determine what stroke the accumulator can handle without having the pressure exceed the maximum force specified for the chill roller (57,75 kN) the force is converted into the corresponding pressure (11,48 MPa) and then reinserted into the Boyle’s equation. The pressure increase from 8,20 MPa to 11,48 MPa gives the volume decrease from 0,45 l to 0,32 l and the corresponding stroke 25,6 mm. Note that this stroke cannot be used instantly since the hydraulic fluid has a relatively high viscosity which in combination with restrictions on the way to the accumulator results in stiffer behaviour for very quick pressure changes while it acts as a damper when changes are slower.

The overlapping splice will result in very small pressure changes in the system, going from 8,20 MPa to 8,25 MPa does not induce any noticeable forces in the system. Looking at a web speed of 800 m/min or 13,3 m/s it is understandable that a splice will pass through the nip almost instantaneously, this means that the pressure increase caused by this change in thickness will look very much like an impact to the system.

When looking at wrap-around it is clear that a ten second wrap-around will cause a significant pressure increase in the system, after this time period the material thickness can be as thick as 25,9 mm. This scenario is based on the assumption that a long piece of web will be free at the point of the web break, causing it to be drawn into the nip together with the wrap-around. This means that the amount of material accumulated every turn will be twice the material thickness. A full ten second wrap-around results in a load change from 41,25 kN to 58,09 kN which is slightly above the permitted force of the chill roller. In the hydraulic system there is a potential chance of causing damage to the chill roller if a wrap-around of this type occurs.

6 Product development

The presentation of the product development procedure is presented here along with some discussions and reflections.

6.1 Methodology

Following Ulrich & Eppinger's methodology this thesis is divided into the corresponding sub-categories and decompositions. By following their methodology the work becomes more structured both in writing and execution. Though the methodology outline is clear it is necessary to adapt the different steps to the actual process, therefore some deviations from the theoretic plan may occur.

6.2 Product specifications

Most equipment and process data were presented at the start-up meeting for the thesis, these values specify the demands on the solution and ensures the products performance in future usage. The data[4] is converted into a list of metrics and additional items are added to get a complete functions description, see table 6-1.

With the list of metrics it is easier to see the result of every specification and every metric can now be ranked by setting their impact values respectively. All metrics are ranked using a numeric scale starting from one which is the most important metric. The ranking process is performed in collaboration with co-workers where their opinions are compiled into the final ranking through discussions.

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[4] Skytte, Lena, 2009, New generation nip equipment specification_Issue 3

Table 6-1: List of metrics

List of Metrics			
No.	**Metric**	**Impact**	**Unit**
1	Line load	2	N/mm
2	Linear motion	6	mm
3	Linear motion speed	7	mm/s
4	Detect and act, load change	5	s
5	Detect and act, paper splice	4	s
6	Adaptive mounting properties	9	Subjective
7	Width (across web direction)	15	mm
8	Length (web direction)	12	mm
9	Height (vertical)	17	mm
10	Interface compatibility	8	Protocol
11	Ozone resistant	11	Subjective
12	Heat resistant	16	ºC
13	Water resistant	18	Subjective
14	Melted PE resistant	19	Subjective
15	Maintenance friendly	10	Subjective
16	Price	13	SEK
17	Safety compliance	1	Protocol
18	Precision	3	mm
19	Weight	14	kg

A brief description of the most important metrics:

1. Line load (N/mm) - is specified directly in process data and is, together with the precision, the key parameter for the product to meet the demands. The line load is the force created in the contact between nip- and chill roller, in reality it is a pressure distributed on a narrow area.

2. Linear motion (mm) - the possibility to open the nip for start-up/shutdown and maintenance.

3. Linear speed (mm/s) - capacity to move the nip roller the specified length at a certain speed.

4. (and 5.) Detect and act, load change/paper splice - elasticity of the system measured in the ability to compensate for a small load changes (paper splice) and a larger load changes (wrap-around, ozone pipe conk out etc).

17. Safety compliance - it is vital that the product meet the company safety standards.

18. Precision (mm) - the ability to control the nip pressure in fine detail to reach future demands.

The subjective unit means that the metric can be achieved through good design, for example the solution can be water resistant if the design is well protected and not exposed to the environment.

Table 6-2: Target specifications

Target Specifications				
No.	**Metric No.**	**Metric**	**Unit**	**Value**
1	17	Safety compliance	Protocol	SIL-2, PFL-D
2	1	Line load	N/mm	50±2
3	18	Precision	mm	< ± 0,5
4	5	Detect and act, paper splice	s	0
5	4	Detect and act, load change	s	0
6	2	Linear motion	mm	60-100
7	3	Linear motion speed	mm/s	12-20
8	10	Interface compatibility	Protocol	Profibus/Profinet
9	6	Adaptive mounting properties	Subjective	Design Parameter
10	15	Maintenance friendly	Subjective	Design Parameter
11	11	Ozone resistant	Subjective	Design Parameter
12	8	Length (web direction)	mm	<800
13	16	Price	SEK	≤ Hydraulics
14	19	Weight	kg	<100
15	7	Width (across web direction)	mm	<280
16	12	Heat resistant	C	Design Parameter
17	9	Height (vertical)	mm	<400
18	13	Water resistant	Subjective	Design Parameter
19	14	Melted PE resistant	Subjective	Design Parameter

The target specifications, see table 6-2, are the properties from the list of metrics sorted by importance from ranking and inserted into a new list with the most important metric on top. To each metric a value is assigned and that is the ideal value, meaning what is wanted in the best possible solution. These values are in some cases data direct from project specification but can also be information from current system and otherwise desired properties. The values for line load, precision, linear motion and linear motion speed are gathered directly from project specification while the safety compliance and interface compatibility are standards for the company. Measurements are taken from the existing hydraulic solution, if the new solution can fit in the laminator station without making extensive change to the general design this can significantly lower costs and ease assembly in existing machinery.

The design parameters represent the possibility to design the solution in a way that it will resist the environments variables or so that it will be easy to mount on existing equipment.

6.3 Concept generation

The target product specifications will be the foundation for the concept generation. The suggestions presented in this chapter are the result of brainstorming sessions, searching the internet, looking at similar solutions and looking at product datasheets. To get the best result it is important to regard every suggestion and to keep a large portfolio of ideas to consider when making the concept selection.

To handle the complexity of the problem it is decomposed into three parts; guiding, elasticity and motion guiding. By doing this it is possible to focus on one part at a time and then combine these suggestions in the end. Making the concepts as attractive as possible in an investment point of view is important and therefore the work aims at replacing as few parts as possible along with making assembly and mounting as easy as possible.

6.3.1 Linear motion

The goal is to find the best way of moving the nip roller in a well controlled motion. The device has to be electrically manoeuvred, produce a linear motion and be able to comply with the listed specifications. The possible solutions presented here have been produced through discussions with colleagues and searches made on the internet.

- **Gear rack, stationary rail**

The nip unit is mounted together with a servomotor and will move along the gear rack that is mounted on the frame. Movement is done through a gear mounted on the engine that drives the nip roller unit along the stationary gear rack.

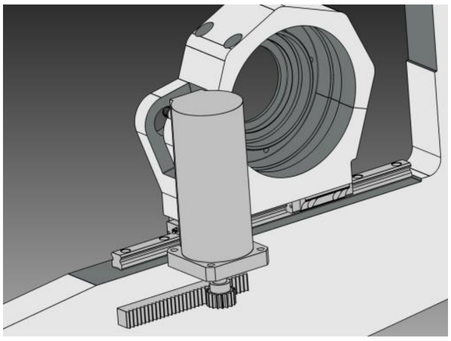

Figure 6-3: Gear rack, stationary rail

- **Gear rack, rail motion**

The nip unit is mounted on the gear rack which is manoeuvred by a servomotor with a mounted gear. This is the inverted version of the previous solution but the gear rack is now mobile while the servomotor is stationary.

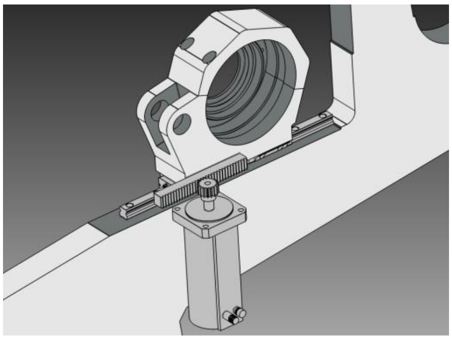

Figure 6-4: Gear rack, rail motion

- **Electromechanical cylinder**

With the servomotor connected on the back the linear motion is created by a planetary roller screw. Since the unit is compact and robust it can withstand large forces and fast movement without losing precision.

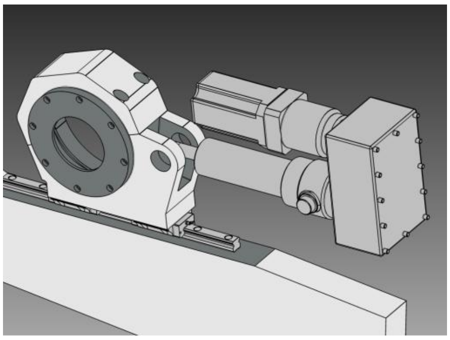

Figure 6-5: Electromechanical cylinder

- **Positioning table**

A system with an internal screw mechanism and a rail guide that handles both motion and guiding of the nip roller. It resembles the present profile rail guide solution but with an integrated drive mechanism.

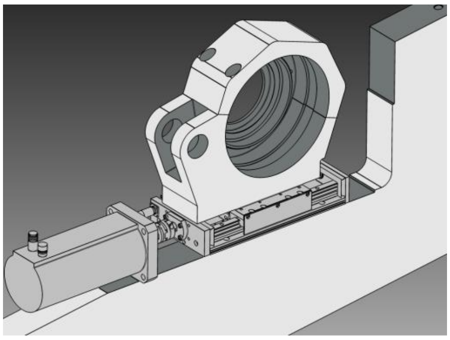

Figure 6-6: Positioning table

- **CAM system**

By using a CAM-mechanism with a push rod attached to the nip roller bearing housing the nip roller can be moved by applying torque exerted by a servomotor.

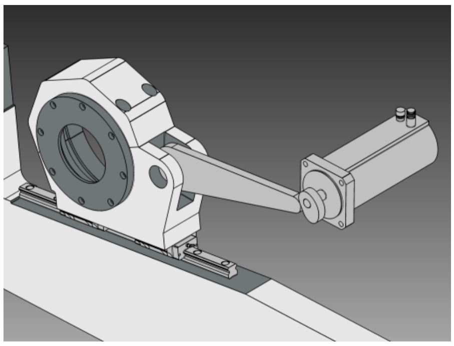

Figure 6-7: CAM-system

6.3.2 Elasticity

The solution must be able to handle sudden load changes and deviations from normal operation, there must be some kind of elasticity in the system. There are a few good ways of creating this elasticity in a system with these large forces. The possible solutions are:

- **Electromagnetic release mechanism**

By attaching the drive system to an electromagnetic coupling the connection can be released when the load exceeds a certain value. The released system will be restored to its initial position by springs inside the coupling. This solution will only handle push and pull forces and must be positioned between nip roller bearing housing and the linear solution.

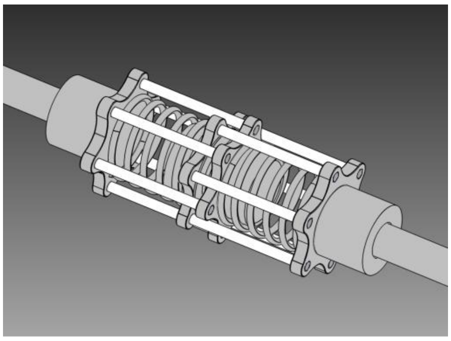

Figure 6-8: Electromagnetic release mechanism

- **Overload torque coupling**

By installing an overload torque coupling between the servomotor and the linear unit the driving shafts will be allowed to rotate independently to each other when an overload occurs. It depends on the linear unit's ability to translate the linear force to a rotational torque in both directions. This coupling is very similar to the coupling found in most cordless drills on the market.

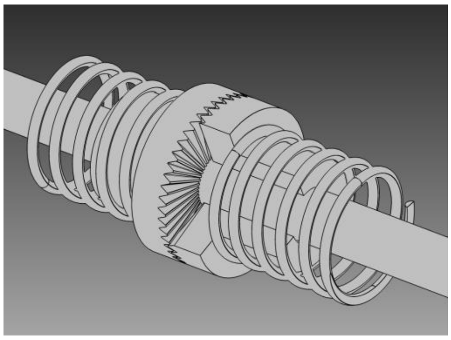

Figure 6-9: Overload torque coupling

- **Servomotor control**

Sensing pressure changes in the nip by load cell or similar equipment and then compensate directly by controlling the servomotor rotation means that no additional equipment is required.

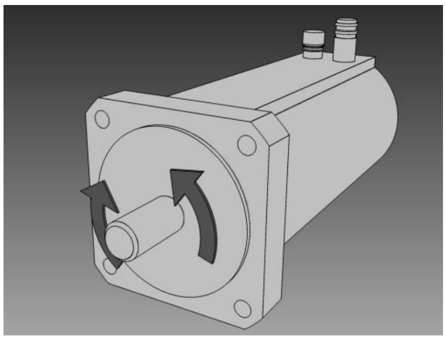

Figure 6-10: Servomotor control

- **Disc spring**

Disc springs have capacity to endure large loads but small deflections only, in a disc spring package it is possible to combine several disc springs to get the desired properties.

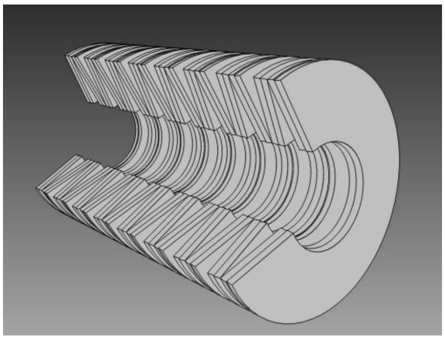

Figure 6-11: Disc spring

- **Breakpin**

This solution offers an overload failsafe mechanism and is basically a pin designed to be the weakest part in the unit and to break at a certain load. When the pin breaks it must be replaced with a new one manually.

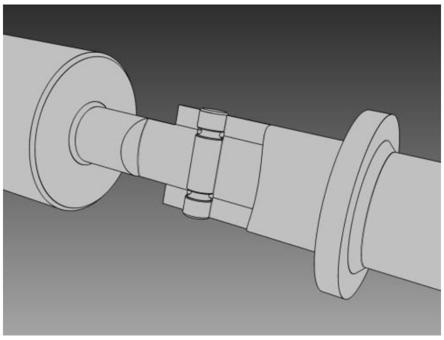

Figure 6-12: Break pin

6.3.3 Motion guiding

The nip roller must be guided in a low friction solution that keeps the housing parallel to the machine frame. Since there is already a working solution to this problem that works well there is no point in replacing it unless the design is altered in a radical way.

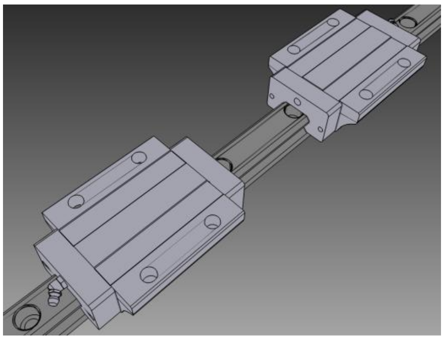

Figure 6-13: Motion guiding

6.4 Concept selection

Choosing the right concept solutions is important and therefore it is vital not to exclude concepts that seem bad at a first glance. By setting up a selection table it is possible to compare the concepts with each other and with an already existing solution. The criteria in the selection matrix are gathered from the target specification table and modified slightly to hold the most important properties. The criteria concerning profitability and usability are added to provide a more complete analysis. Since pricing and precise design of the concepts are not known in this stage the values are estimated from an engineering point of view and, when possible, gathered from the manufacturers. The concepts are rated in the categories "specification compliance" and "other" where the first section holds the criteria for how well the concept complies with the specifications and the latter handles aspects such as investment cost and usability to get a more complete view of the concepts. The concepts are rated using a numerical scale from one to five where a higher number equals a better solution.

6.4.1 Linear motion

A short description of the criteria:

- Safety - This criteria contains both the safety of usage and the process safety
- Load - The ability to generate and hold the specified load.
- Precision - Maintaining good parallelism between nip roller ends and keeping the load within the accurate tolerance limit.
- Speed - The ability to move the nip roller between opened and closed position within the specified time limits.
- Stroke - Moving the nip roller the specified distance.
- Cost - Estimating the cost for the concept with the hydraulic system as reference.
- Maintenance - The relative amount of maintenance required.
- Usability - Concept functionality and ability to be utilized.

The hydraulic system is rated as 'three' on all criteria because it is positioned as the reference system for the concept rating. A rating of four means that the concept performs well on these criteria and better than the hydraulic system while a rating of two means that it has a generally lower performance. The sum of every rating is presented as the key figure at the bottom and their individual average value at the top of every category. The total sum is presented at the bottom of the table along with the total average for all criteria.

Table 6-13: Concept selection - Linear motion

	Ref.	a	b	c	d	e
	Hydraulic system	Gear rack, staionary rail	Gear rack, rail motion	Electromechanical cylinder	Positioning table	CAM-system
Specification compliance	**3,0**	**2,8**	**2,8**	**4,2**	**3,8**	**2,8**
Safety	3	2	2	5	5	2
Load	3	2	2	4	2	3
Precision	3	2	2	5	5	4
Speed	3	4	4	4	4	3
Stroke	3	4	4	3	3	2
Other	**3,0**	**3,3**	**3,3**	**4,7**	**4,3**	**3,0**
Cost	3	4	4	4	4	3
Maintenance	3	2	2	5	5	3
Usability	3	4	4	5	4	3
Key figure	**6,0**	**6,1**	**6,1**	**8,9**	**8,1**	**5,8**
Average key figure	**3,0**	**3,1**	**3,1**	**4,4**	**4,1**	**2,9**
Total	**24,0**	**24,0**	**24,0**	**35,0**	**32,0**	**23,0**
Total average	**3,0**	**3,0**	**3,0**	**4,4**	**4,0**	**2,9**

- **Safety -** *The gear rack systems (a,b)* and the *CAM system (e)* gets a lower rating because of their exposed mechanisms that potentially can be hazardous. These solutions are in terms of control equally as safe as the others. Both *the electromechanical cylinder (c)* and *the positioning table (d)* are well protected and they are both superior to the hydraulic system because of their manoeuvrability.

- **Load -** While *the electromechanical cylinder (c)* has no problem reaching the specified load the others are questionable, especially *the positioning table (d)* which requires further investigation.

- **Precision -** *The gear rack systems (a,b)* and *the CAM system (e)* have potential back-lash problems along with some designing issues to get their power centred on the nip roller. *The electromechanical cylinder (c)* and *the positioning table (d)* are designed to produce an accurate movement and will outperform the hydraulic system every time.

- **Speed -** Since all solutions are using a servomotor they will be more or less faster than the hydraulics depending on what gear ratio is used.
- **Stroke -** The design of *the gear rack systems (a,b)* provide much freedom in length and they are easier to adapt for the right stroke than the others. It is difficult to get a long stroke with a cam and therefore *the CAM system (e)* gets a lower rating. *The electromechanical cylinder (c)* is similar to a hydraulic cylinder in its design while *the positioning table (d)* can be adjusted more easily.
- **Cost -** When calculating the cost for a hydraulic system the sum includes costs for cylinder, hoses and hydraulic unit which means that the concepts likely will be cheaper. *The positioning table (d)* will be significantly cheaper as it is a complete compact unit.
- **Maintenance -** *The electromechanical cylinder (c)* and *the positioning table (d)* are typically low maintenance systems while *the gear rack systems (a,b)* and *the CAM system (e)* require regular upkeep because of their exposed mechanisms.
- **Usability -** In general the servomotor is easily controlled and monitored which makes it superior to the hydraulic system.

The result of this evaluation leads to direct elimination of *the gear rack systems (a,b)* and the *CAM system (e)* since they have the lowest scoring and therefore a lower overall performance. *The electromechanical cylinder (c)* and *the positioning table (d)* are still potential concepts but *the positioning table (d)* must be further investigated to determine its load capacity which is still unknown at this point.

6.4.2 Elasticity

Criteria:

- Safety - This criteria contains both the safety of usage and the process safety
- Load - The ability to withstand the overloads that can occur in the system.
- Reaction time - Rating of possible delays in reaction for an overload.
- Reaction speed - The rate at which the concept compensates for an overload.
- Stroke - Meeting the demands for maximum deflection of wrap-around, ozone pipe conk out or another error of large motion amplitude.
- Cost - Estimating the cost for the concept with the accumulator as reference.
- Maintenance - The relative amount of maintenance required.
- Usability - Concept functionality and ability to be utilized.

In this case the accumulator has the rating 'three' for all criteria since it is the reference system. The same rating system applies to the elasticity problem as the previous linear motion section.

Table 6-14: Concept selection - Elasticity

	Ref.	f	g	h	i	j
	Accumulator	Electromagnetic release mechanism	Overload torque coupling	Servomotor control	Disc spring	Break pin
Specification compliance	**3,0**	**2,8**	**2,4**	**3,4**	**4,0**	**4,0**
Safety	3	1	3	5	3	2
Load	3	2	2	4	4	3
Reaction time	3	4	2	2	5	5
Reaction speed	3	4	2	2	5	5
Stroke	3	3	3	4	3	5
Other	**3,0**	**3,0**	**2,3**	**4,3**	**3,0**	**3,7**
Cost	3	3	3	5	2	5
Maintenance	3	3	2	4	3	3
Usability	3	3	2	4	4	3
Key figure	**6,0**	**5,8**	**4,7**	**7,7**	**7,0**	**7,7**
Average key figure	**3,0**	**2,9**	**2,4**	**3,9**	**3,5**	**3,8**
Total	**24,0**	**23,0**	**19,0**	**30,0**	**29,0**	**31,0**
Total average	**3,0**	**2,9**	**2,4**	**3,8**	**3,6**	**3,9**

The accumulator is the reference and all concepts are compared to it:

- **Safety -** *The electromagnetic release mechanism (f)* is rated low in safety because it will cause an uncontrolled motion when released, it is also depending on a reliable power supply and will release at power shortage which is not tolerated in this application. *The break pin (j)* will also cause an uncontrolled motion when breakage occurs, it will also bring uncertainty to the operation since it depends strongly on both material properties and design to work correctly. *The servomotor control (h)* is a very safe solution because it will add no external components to the system while both *the overload torque coupling (g)* and *the disc spring (i)* will provide sufficient safety.
- **Load -** *The electromagnetic release mechanism (f)* depends on strong electromagnets and *the break pin (j)* relies on good design and correct material properties. Preliminary research shows that finding an electromagnet strong enough can be hard. *The disc spring (i)* and *the servomotor control (h)*

handles the overload with precision, *the overload torque coupling (g)* shows potential load problems because of the dependence on force feedback passing through the linear motion unit which will probably absorb some load through inner resistance, gear ratio or similar.

- **Reaction time -** *The electromagnetic release mechanism (f)* and *the break pin (j)* has very fast reaction time as well as *the disc spring (i)* which has none or little internal inertia. *The overload torque coupling (g)* has some inertia caused by the sliding clutch while it seems hard reaching the required reaction time for *the servomotor control (h).*
- **Reaction speed -** The reasoning above applies on this criteria also. *The servomotor control (h)* will possibly be marginally better when the motion has started but in comparison with the other concepts it is still way behind.
- **Stroke -** *The electromagnetic release mechanism (f)* and *the break pin (j)* will have as much stroke as the system need. The other concepts can be designed so that they can manage the specification.
- **Cost -** The electromagnetic release mechanism (f), *the overload torque coupling (g)* and *the disc spring (i)* needs many additional components while *the servomotor control (h)* and *the break pin solution (j)* require few or none.
- **Maintenance -** *The servomotor control (h)* and *the disc spring (i)* are virtually maintenance free except for some minor inspection and/or calibration. *The electromagnetic release mechanism (f)*, *the overload torque coupling (g)* and *the break pin solution (j)* requires regular inspection and calibration.
- **Usability -** Except for some programming and design work *the servomotor control (h)*, *the disc spring (i)* and *the break pin (j)* are very easy to implement and control while *the electromagnetic release mechanism (f)* demands more from the user. *The overload torque coupling (g)* causes the servomotor to lose its position and it must be recalibrated after an overload.

As *the electromagnetic release mechanism (f)* and *the overload torque coupling (g)* have a generally lower performance than the other concepts and also show some more serious issues this results in the removal of these concepts. *The servomotor control (h)* has many advantages but it is unclear whether it is fast enough to react on load changes, this must be investigated further. *The break pin (j)* is a release mechanism and it will introduce a weakness in the system that may lead to breakage even in normal operation if designed in the wrong way or if the material does not perform as specified. To be able to handle all load changes it must be combined with some other solution. *The disc spring (i)* show promising results and will be explored further.

6.4.3 Motion Guiding

The current system fulfills every aspect of the specification and can be kept for the chosen concept as well. It is also of value for the project to keep the current solution since the replacement of the hydraulic system will then be easier and less costly.

6.5 Final concept selection

As the number of concepts are reduced it is possible to further investigate the remaining suggestions. Some concepts that made it through the first selection had some issues that needed exploration and that is of high priority before proceeding with the deeper analysis.

6.5.1 Linear motion

- **Positioning table**

  When in contact with experts from SKF it becomes clear that this concept is not able to produce the load that is required and is therefore removed from further analysis.

- **Electromechanical cylinder**

  Data for different products is gathered from a couple of manufacturers websites and put together in a table, seen in Appendix B. The manufacturers are contacted and a dialogue is established with SKF and Exlar which both can present a complete working unit for this application. Both suggestions live up to the specifications and the differences are minor. In both cases they offer a solution where the electromechanical cylinder is mounted parallel to the servomotor which results in a shorter unit, this is preferred since the vicinity of the laminator station is limited by walkways, pillar etc. A common property for most electromechanical cylinders is that they are very stiff and will act like a solid element when feedback loads travel through the system, therefore it is important to combine this solution with an elastic unit. If the unit length is critical it is possible to mount this unit beneath the frame using a lever arm, more investigation needs to be done if using this solution. An overview of the two offers:

  Exlar

  The offer (see figure 6-15) consists of a standard sized unit called FT-35 with a servomotor from Exlar or Siemens, since Tetra Pak® uses Siemens® as their standard supplier this will be the choice here as well. Even though the offered product has a full 152 mm stroke it is very compact and the total unit length (excluding rod end) is about 553 mm.

Figure 6-15: Exlar offer

SKF

The unit called SRSA3005-0060-SP0-T2-N-000-1FK7 (see figure 6-16) has a modified stroke to suit this application which contributes to keep the unit as short as possible. The modified stroke will not, according to SKF, render any increased costs of any significance which otherwise can be expected from custom made units. The total length is 523 mm and this solution includes a servomotor from Siemens as desired. Comparing it to the Exlar equivalent the length of the unit gives the SKF offer an advantage, if length is unimportant the Exlar model could also be considered. The SKF offer is specified in more detail which can be seen in appendix C.

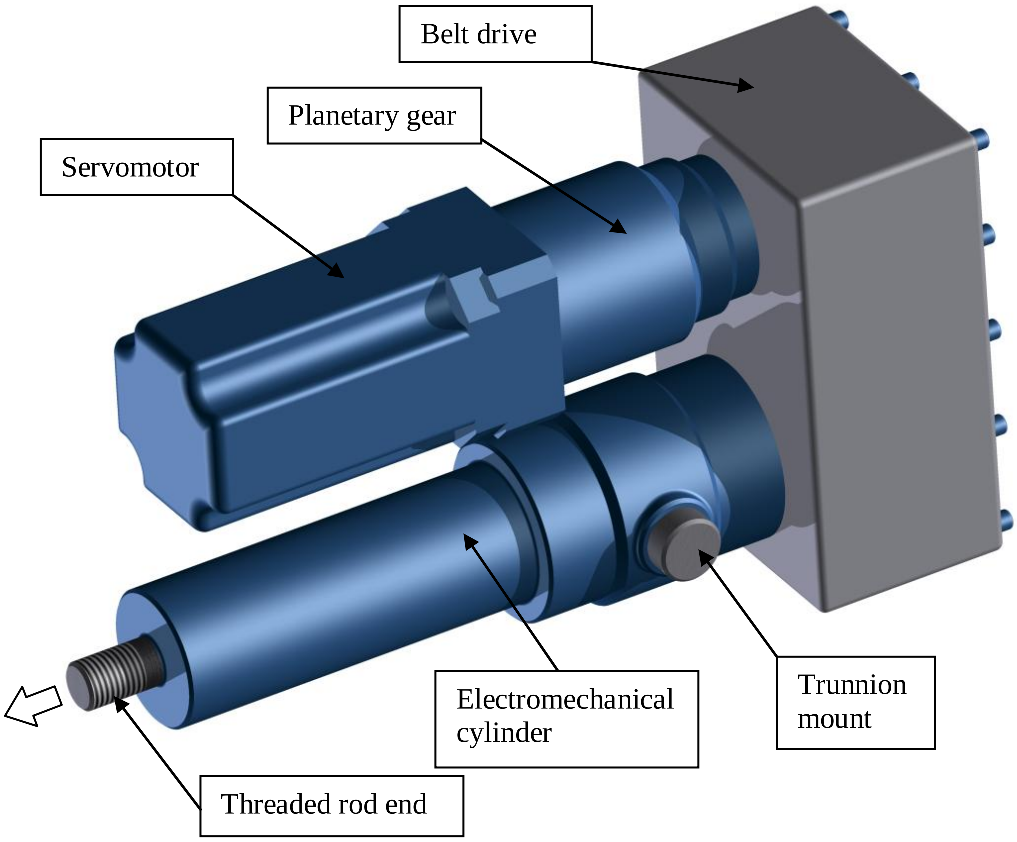

Figure 6-16: SKF offer

6.5.2 Elasticity

- **Servomotor control**

  By rough estimate the time for which a splice passes the nip is approximately (assuming the pressure increase is induced gradually for a length of 1 mm and the machine speed is 800 m/min):

  $$t - Time\ (s) = \frac{0{,}001 \cdot 60}{800} = \mathbf{7{,}5 \cdot 10^{-5}}\ \boldsymbol{s}$$

  By this time the motor or sensor should have sensed the load change and the servomotor should have started rotating to move the nip roller backwards, this performance is not possible to obtain from a standard servomotor. As a result this concept is disregarded from further analysis.

- **Break pin**

  Dimensioning the break pin in the right way is vital to receive the right break load. Since it is a disposable novelty it can only work together with another elasticity solution to handle more common load changes that does not require a halt in production. The break pin can be designed to fit several different positions in the elasticity unit, by integrating it in the shaft coupling between the electromechanical cylinder and the disc spring package the unit will stay connected even after a break pin failure. This also means the threaded connection between these shaft will have to be removed and the shaft will have to be oiled so that they can slide freely for a break pin failure.

  The reasons for not choosing the break pin as the final concept solution are the uncertainties added to the application and that it must be combined with another elasticity solution. The break pin strongly depends on correct material properties and accurate design and if these criteria are not satisfied the break pin can disengage the system in normal operating conditions or fail to disengage at the critical load. If the length of the unit is critical it is possible to use a break pin in combination with a disc spring package which then can be significantly reduced in size, if using a break pin solution the arguments stated here must be well regarded.

- **Disc spring**

  In collaboration with Lesjöfors, a major spring manufacturer, a disc spring set is dimensioned to handle all variations that are specified. The disc spring package must be able to handle a 20 mm stroke within the load criteria (41,25 kN - 57,75 kN) which corresponds to an ozone pipe going through the nip. The disc springs are configured in a certain way to fulfill the requirements as shown in this cross-section:

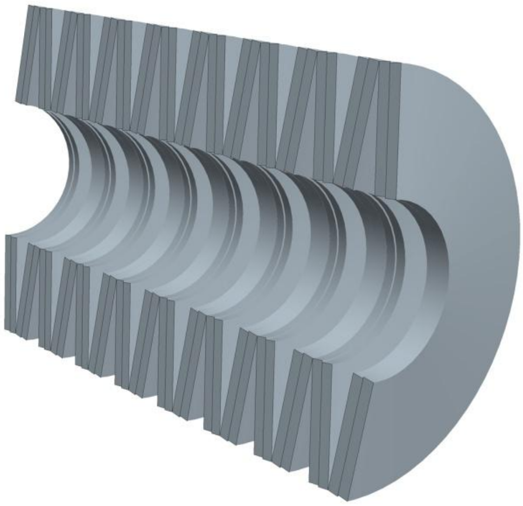

Figure 6-17: Disc spring configuration

By stacking the springs in pairs of two the force capacity doubles, to handle the stroke there are sixteen packets (see figure 6-17) which gives a total of 32 disc springs. By using the springs like this the stroke of every spring pair is added to the total stroke of the package with maintained force capacity[5].

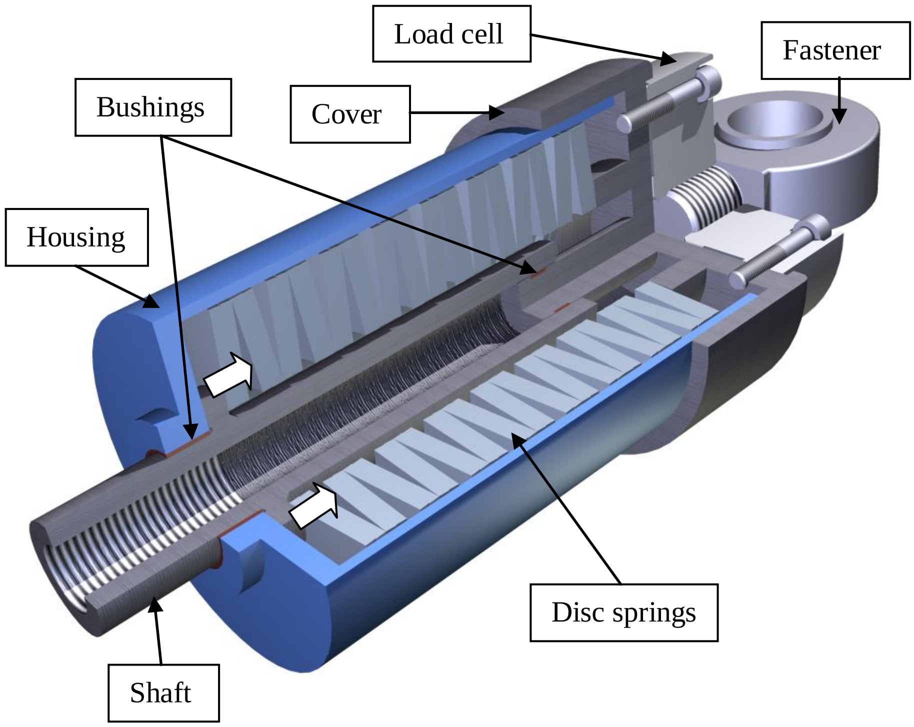

Figure 6-18: Disc spring unit

[5] Lesjöfors AB www.lesjoforsab.com [2010-01-25]

The housing permits a pre load to be applied to the springs, like the accumulator this means that the system is rigid at loads below this level to enhance precision. When assembled (see figure 6-18) the shaft is inserted into the housing and then the disc springs are inserted in a similar way, they are guided on their outer diameter and are not in contact with the shaft. The bushings enable the housing to slide freely on the shaft, efficiently minimizing friction between the parts. Finally the cover is mounted and by threads between housing and cover it is tightened to the specified pre load and locked into place. When an overload occur the housing will slide back on the shaft to compensate for the load change. If required it is possible to attach a load cell between the cover and the fastener to provide some additional load data, see Appendix D. Calculations made on disc spring package can be seen in the datasheet provided by Lesjöfors in Appendix E, to verify these values a manual control is made, using formulas[6] for disc spring characteristics, as seen below.

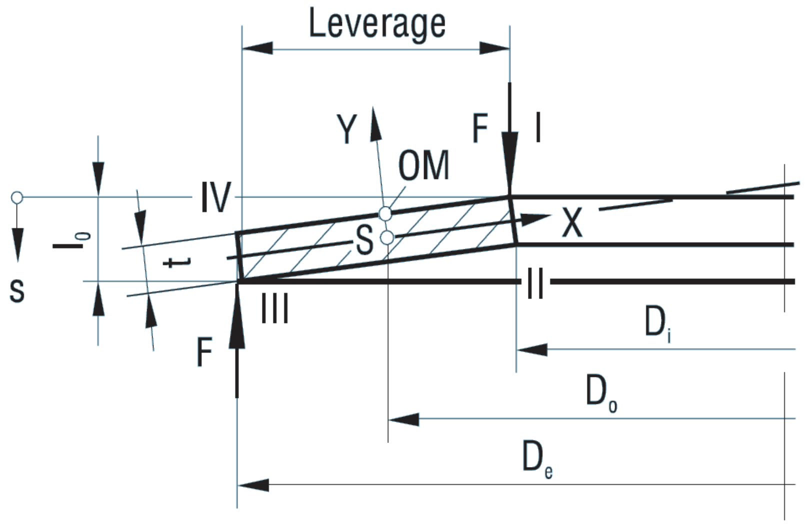

Figure 6-19: Disc spring geometry

$E = Young's\ modulus = 206 \cdot 10^3\ MPa$

$\mu = Poisson's\ Ratio = 0{,}3\ (spring\ steel)$

$t = Individual\ spring\ thickness = 5\ mm$

$D_e = Outside\ diameter = 150\ mm$

$D_i = Inside\ diameter = 61\ mm$

$s = Deflection\ of\ a\ single\ spring = 5{,}30\ mm$

$h_0 = Cone\ height\ of\ an\ unloaded\ single\ spring = 5{,}30\ mm$

[6] Schnorr www.schnorr.com [2010-01-26]

The ratio between outer and inner diameter δ:

$$\delta = \frac{D_e}{D_i} = \frac{150}{61} = \mathbf{2,46}$$

Constants for calculation K_1 and K_4:

$$K_1 = \frac{1}{\pi} \cdot \frac{\left(\frac{\delta - 1}{\delta}\right)^2}{\frac{\delta + 1}{\delta - 1} - \frac{2}{\ln \delta}} = \frac{1}{3,14} \cdot \frac{(0,59)^2}{2,37 - 2,22} = \mathbf{0,76}$$

$$K_4 = \mathbf{1}\ (for\ this\ type\ of\ spring)$$

Maximum spring force for one spring:

$$F_{max} = \frac{4E}{1 - \mu^2} \cdot \frac{t^4}{K_1 \cdot D_e^2} \cdot \frac{s}{t} \cdot \left[K_4^2 \cdot \left(\frac{h_0}{t} - \frac{s}{t}\right) \cdot \left(\frac{h_0}{t} - \frac{s}{2t}\right) + 1\right] = \mathbf{35207\ N}$$

Two parallel springs:

$$F_{max} = 2 \cdot 35207 = \mathbf{70414\ N}$$

Maximum deflection for package:

$$s_{max} = s * 16 = 5,30 * 16 = \mathbf{84,8\ mm}$$

Deflection for package at pre load (two parallel springs):

$$s_{p1} = \left[F = \frac{41250}{2} = 20625\ N\right] = 1,97mm \cdot 16 = \mathbf{31,6mm}$$

Deflection for package at pre load (two parallel springs):

$$s_{max} = \left[F = \frac{57750}{2} = 28875\ N\right] = 3,43mm \cdot 16 = \mathbf{54,8mm}$$

Deflection for package between load points:

$$s_{operation}(package) = 54,8 - 31,6 = \mathbf{23,2mm}$$

This package will deflect 23,2 mm between the specified load points, which fully complies with the demands set for this solution. The manual calculations confirm the accuracy in the datasheet delivered from Lesjöfors.

6.5.3 Console

To be able to mount the concept solution onto the existing frame, a console is needed. It is important to design a robust console since the weight it carries will be substantial. It is also important to have good accessibility and an easy assembly procedure.

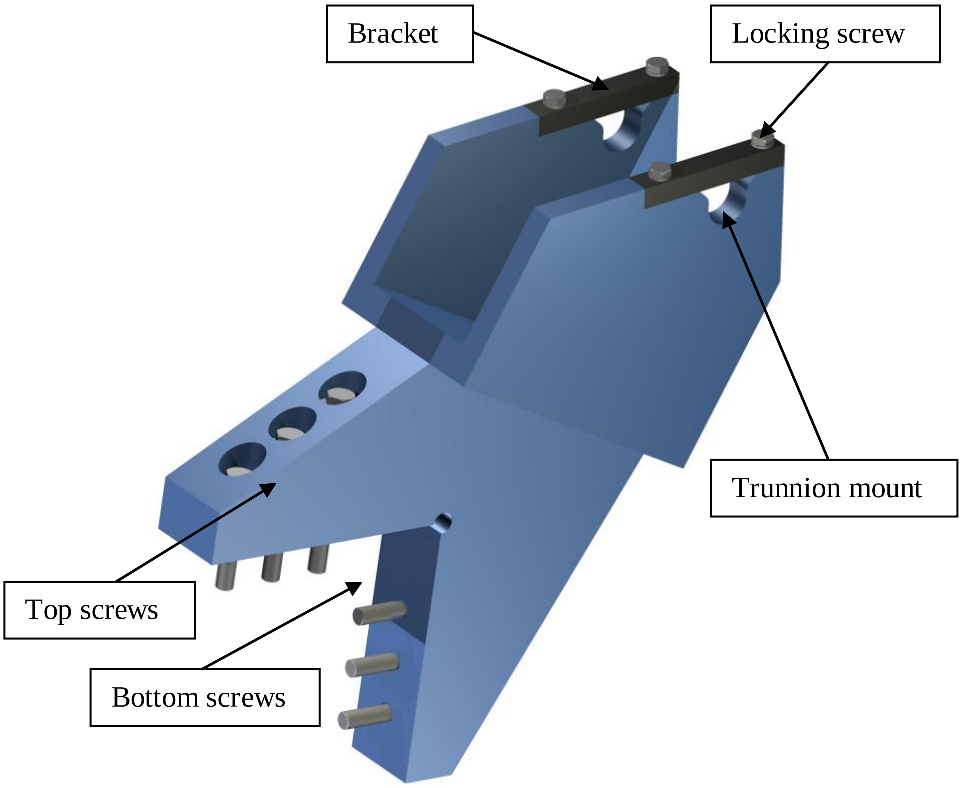

Figure 6-20: Console

When choosing to fasten the console with screws as showed here, the forces in the screws will be mainly of a push/pull type, though the screws on the top part can experience some shear stress. The design (see figure 6-20) allows this unit to be mounted on existing frame by drilling and threading the six holes for the screws. The trunnion mount makes assembly of the unit swift, easy and exact. The electromechanical cylinder is locked into place with the brackets and the locking screws. In addition to this console a plate can be attached on either side to support additional equipment such as conducting rollers and beams.

6.5.4 Final specification

As the final concept selection is made it is now possible to setup the final specifications (see table 6-21). To be able to compare with the target specifications a column is added on the right stating the target value for each metric.

Table 6-21: Final specifications

Final Specifications				
No.	**Metric**	**Unit**	**Value**	**Target values**
1	Safety compliance	Protocol	SIL-2, PFL-D	SIL-2, PFL-D
2	Line load	N/mm	67,7	50±2
3	Precision	mm	< ± 0,5	< ± 0,5
4	Detect and act, paper splice	s	0	0
5	Detect and act, load change	s	0	0
6	Linear motion	mm	60	60-100
7	Linear motion speed	mm/s	15,6	12-20
8	Interface compatibility	Protocol	Profibus/Profinet	Profibus/Profinet
9	Adaptive mounting properties	Subjective	Yes	Design Parameter
10	Maintenance friendly	Subjective	Yes	Design Parameter
11	Ozone resistant	Subjective	Yes	Design Parameter
12	Length (web direction)	mm	~950	<800
13	Price	SEK	<Unknown	≤ Hydraulics
14	Weight	kg	~90	<100
15	Width (across web direction)	mm	180	<280
16	Heat resistant	C	Yes	Design Parameter
17	Height (vertical)	mm	280	<400
18	Water resistant	Subjective	Yes	Design Parameter
19	Melted PE resistant	Subjective	Yes	Design Parameter

As can be seen when comparing the final values to the target values most of the metrics have sufficient performance or more. The electromechanical cylinder will have capacity to generate a constant line load of 67,7 N/mm which is well beyond the project specification of 50 N/mm. Though the exact accuracy is unknown at this point it is certain that it will meet the required tolerance level, the same goes for the precision metric where the limits are set by other components than the electromechanical cylinder (load cell, automation etc.). The unit can also move the nip roller the specified minimum stroke at the given time without any problem.

With a disc spring package the reaction time for load changes are close to zero seconds no matter what load change magnitude is induced to the system. Though the length of the solution will exceed the target value there should be no problem in most cases, when it does cause problems there are two possibilities: combining a smaller disc spring package with a break pin or mounting the electromechanical cylinder beneath the frame of the laminator station (this requires further modification of all components).

The cost of the unit is still unknown even though offers are being requested, a rough estimation shows that it will probably be somewhat cheaper than a hydraulic unit and most certainly significantly cheaper in operational cost.

An estimation of the weight shows that it will fulfill the target value of 100 kg (not taking the console in account). Since the frame of the laminator station is very stabile it will be possible to design a console that will carry the unit without any problems. An estimation of the unit weight:

$$\begin{aligned} Weight = 46{,}6\ kg\ EMC + 18{,}5\ kg\ disc\ springs + 6{,}3\ kg\ housing \\ + 6{,}4\ kg\ cover + 2{,}9\ kg\ shaft + 2{,}6\ kg\ connector \\ + 5\ kg\ load\ cell = \mathbf{88{,}3\ kg} \end{aligned}$$

Because of the robust design and well protected mechanisms the solution will endure ozone gas, water leakage and melted polymer without the risk of failure.

6.6 Mock-up design

With the selected concepts at hand it is possible to create some CAD-models to present the solution (see figure 6-22). The electromechanical cylinder is a draft model since no CAD files were received from SKF. The designs are preliminary but with the intention of being as close to the final solution as possible.

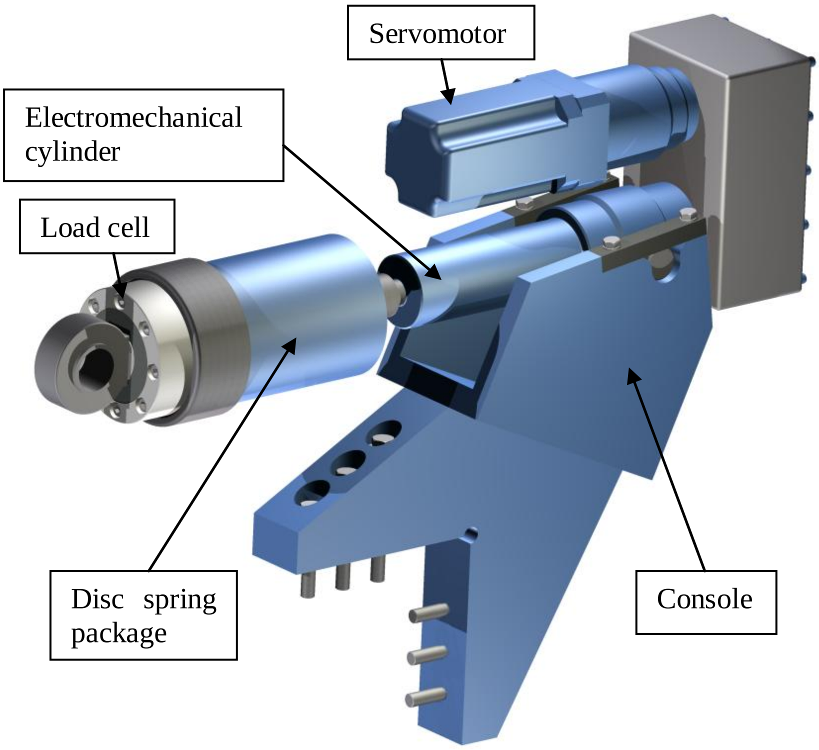

Figure 6-22: Mock-up design

6.7 Structural analysis

To verify that the mechanical strength of the design is sufficient it is necessary to perform some finite element method analysis on the weakest parts in the system.

6.7.1 Console

Starting with the console which will carry both the weight of the concept solution and the system load. The following constraints are applied to the model (see figure 6-23):

1 - The weight of the concept solution (90 kg)

2 - A maximum load from the system (57750 N / 2 = 28875 N)

3 - Frame connection (zero displacement in all six holes)

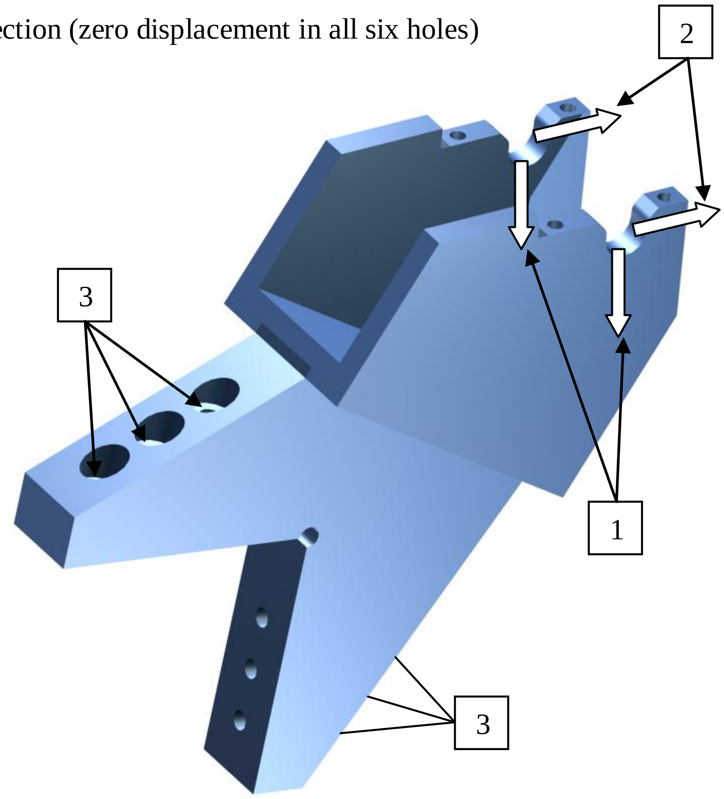

Figure 6-23: Console - Constraints

With the correct material assigned to the model (141650-01) the FE-analysis tool Mechanica built into ProEngineer can start the process. The result is shown below.

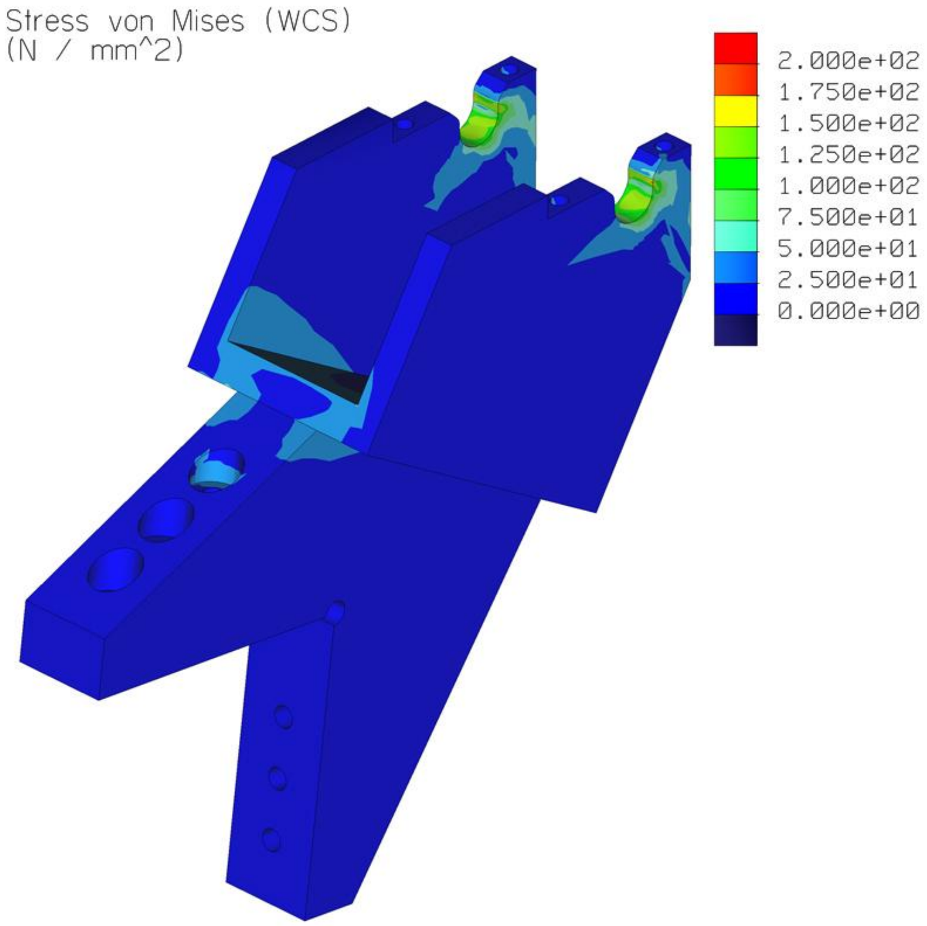

Figure 6-24: Console - Stress

The stress affecting the console reach a maximum at about 190 MPa, comparing this with the tensile strength in the material of approximately 640 MPa it indeed seems as a robust design, it will not suffer from any stress related problems (see figure 6-24).

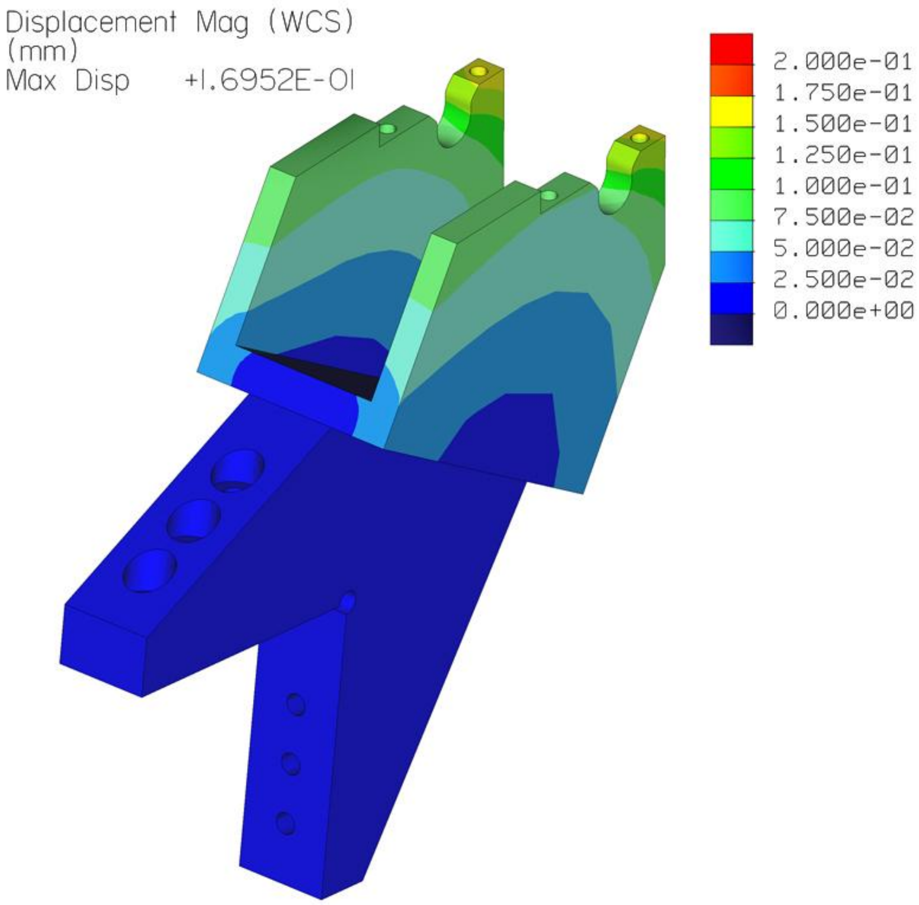

Figure 6-25: Console - Displacement

A quick look at the displacements once again confirms the structural integrity, with a maximum displacement of 0,17 mm it is obvious that this will not affect normal operation. The conclusion of this analysis is that the console is of robust design and maybe slightly over-sized since the brackets also will contribute to an increased stability (see figure 6-25).

6.7.2 Disc spring shaft

The shaft in the disc spring solution will hold the load of the entire unit when it exceeds the pre load force. Therefore it is important to analyze this part for potential weaknesses. The following constraints are applied (see figure 6-26):

1 - Contact surface (zero displacement around entire flange, contact with disc springs)

2 - A maximum load from the system (57750 N / 2 = 28875 N)

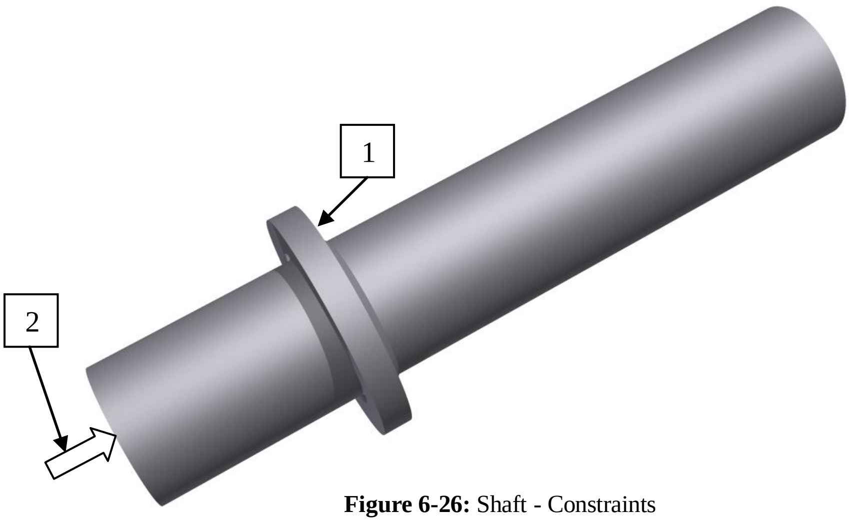

Figure 6-26: Shaft - Constraints

Once again the Mechanica module is used rendering the following stress analysis:

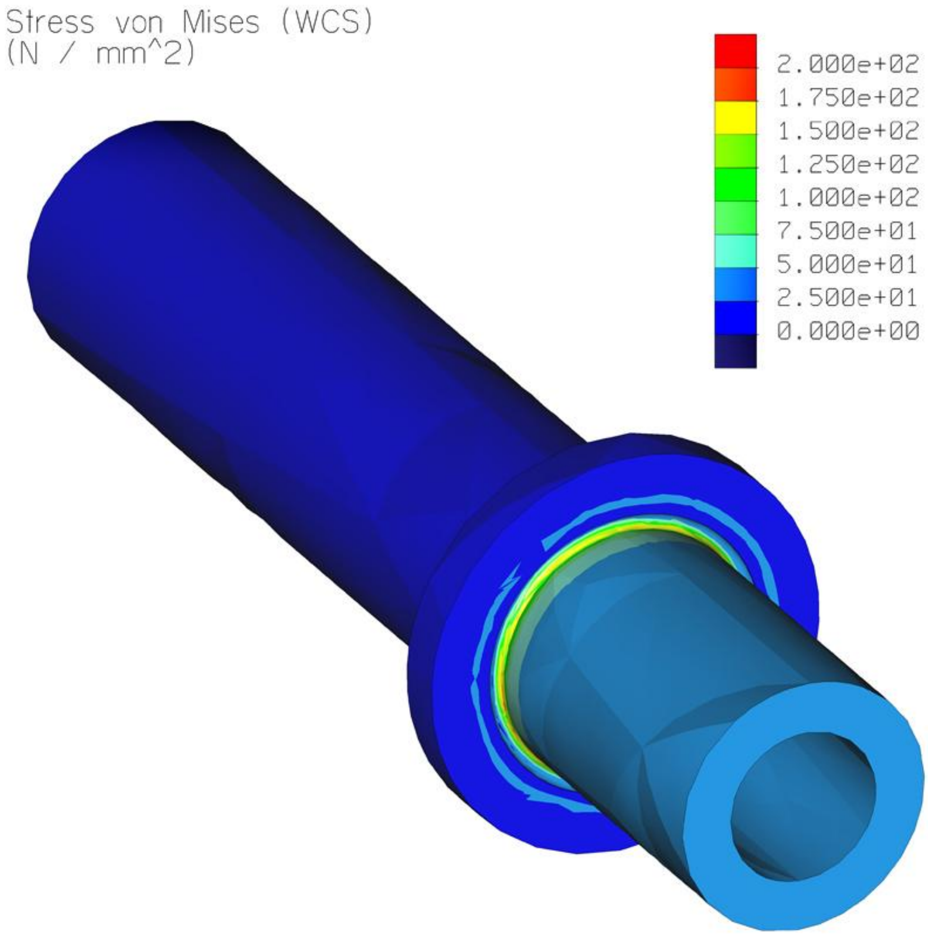

Figure 6-27: Shaft - Stress

With the maximum stress reaching approximately 180 MPa the comparison with the tensile strength reveals a structural integrity which can carry the load without problems (see figure 6-27).

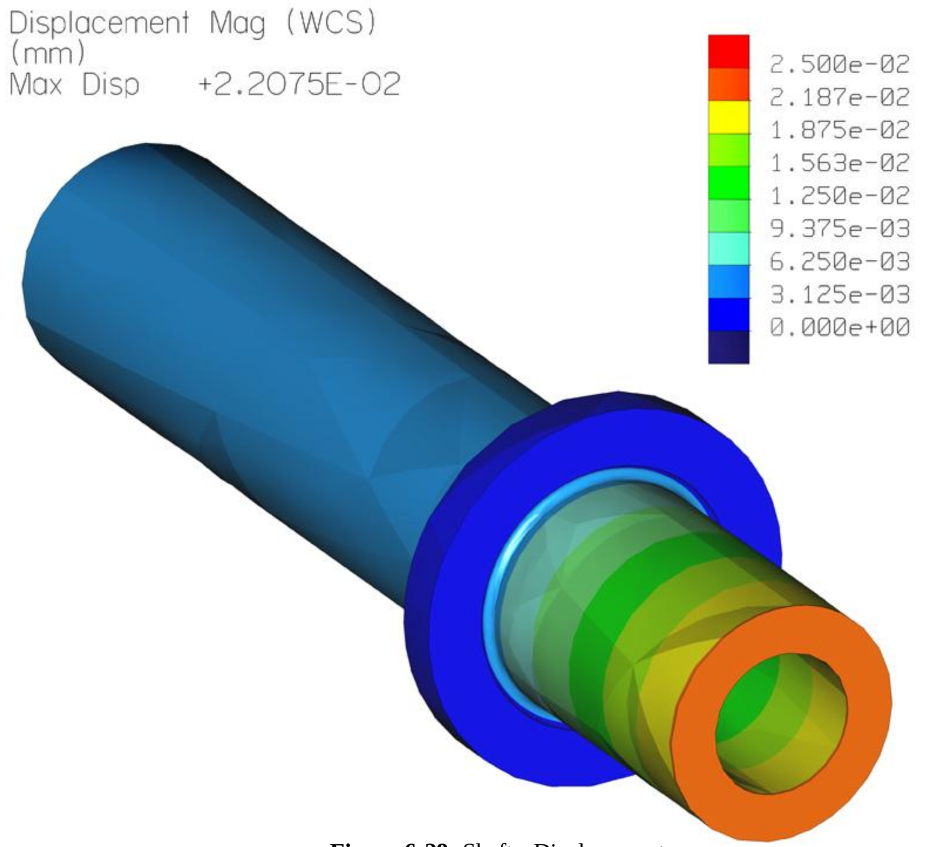

Figure 6-28: Shaft - Displacement

The displacements are as small as 0,022 mm for the maximum deflection, a good result showing that the part is robust and can handle the specified maximum loads (see figure 6-28).

6.8 Conclusions and reflections

By using the methodology provided by Ulrich and Eppinger the work is structured and the different steps makes the concept generation phase less complicated. By setting up the list of metrics and the target specifications in combination with a decomposition of the problem into three sub problems the search for concept solutions becomes more efficient. The concept selection uses a ranking system based on the target specifications which makes it easier to eliminate concepts that do not possess the right properties.

In this case there are two solutions left for the linear motion sub problem, one of these is eliminated as soon as some further investigation is made. In the ideal case it would be preferable to have more than one candidate when the final concept selection is made, but since the section owner and the technical specialists are updated continuously the decision to choose the electromechanical cylinder was well approved.

For the elasticity problem there are three concept solutions available after the first concept selection, the servomotor control concept is disregarded when it is clear that it will not work. It is established that the break pin concept cannot work by itself but only in combination with another elasticity solution and that it brings parameters to the system which are hard to control (material properties, difficult engineering problems etc.). With the base in these statements it is decided that the break pin concept is removed from further studies but also that it can be analyzed further if needed in future development.

This need for a break pin solution can arise if the length of the unit is a critical factor, in that case it is possible to redesign the disc spring solution to include a break pin that will handle the large overloads (full wrap-around, ozone pipe conk out etc.) while the disc spring system takes care of the small variations (paper splice, solid pieces of polymer etc.). The disc spring concept is analyzed further and finally it is approved to be the final concept selection.

Designing the console concludes the concept generation and selection process and together with the establishing of the final specifications and the FE-analysis the concept solution is verified and approved by the section owner and the technical specialists.

7 Results and conclusions

Based on the analysis and the disadvantages as described in chapter 4, the hydraulic system has many weaknesses and there are many benefits to be achieved by replacing it with a servomotor solution. The solution described here is able to meet all requirements and may also contribute with additional features specific for a servomotor solution for example increased controllability and the possibility of changing parameters during operation to fine-tune the process. The servomotor solution has the potential of replacing the hydraulic system completely and will help modernizing the nip unit to comply with future demands.

The electromechanical cylinder is a high precision unit made for large loads and tough environments. Connected with a servomotor in a parallel configuration it is both compact and robust, it is also well protected for usage in a tough environment. It is able to produce a nip pressure slightly above the required level and it will move the nip roller with great precision. The ambition is to control the unit by measuring the load between the unit and the nip roller bearing housing with the help of a load cell connected to the disc spring package. When thoroughly tested it may be possible to control the unit simply by measuring the torque inside the servomotor and removing the load cell, this has to be analyzed further.

The disc spring package can accumulate the load equivalent to an ozone pipe conk out as well as small load deviations. Since the package is preloaded the system will have full precision at normal operating pressure, it is also possible to adjust the preload on the disc spring housing if required. The design provides good protection for moving parts and keeps dirt and dust away from the springs, which otherwise could cause problems since the disc springs are depending on a low friction contact with the guide (which in this case is the housing). If different characteristics are desired there are many ways of configuring the disc springs to achieve various force/deflection ratios.

To be able to mount the new servomotor solution in a correct way the console is designed with the intention of creating a robust unit which is easy to assemble. Though maybe slightly over-sized the console provides a strong base for attaching the new unit, this is important to achieve good performance.

Overall the servomotor solution seems potent and is a good alternative for replacing the hydraulic cylinders. Requiring only small modifications it can easily be mounted onto existing equipment reducing costs for reconstruction significantly.

7.1 Suggestions and recommendations

In the case the unit length exceeds the space available there are alternatives to investigate further. The first alternative is to combine a smaller disc spring package with a break pin solution. By doing this the smaller disc spring package will handle the small loads such as overlap splices, solid pieces of laminate etc. while the break pin will release the system for extreme overload situations such as extensive wrap-around, ozone pipe conk out etc.

The other alternative is to place the electromechanical cylinder and servomotor beneath the nip roller system with the help of a lever arm, this alternative require extensive modification to the nip unit and needs to be analyzed thoroughly before implemented.

In the case of choosing between SKF and Exlar the choice to use the SKF unit in this thesis was made mainly because of the amount of assistance that they supplied during the project and because of the fact that their unit was slightly shorter than the Exlar offer. SKF also showed interest in understanding the process and all possible problems that can occur and therefore their offer seems more reliable.

Appendix A: **References**

Books/articles

Eppinger, Steven; Ulrich, Karl, 2003, Product Design and Development, McGraw-Hill, USA

Skytte, Lena, 2009, New generation nip equipment specification_Issue 3

Internet

PMC Hytech www.pmchytech.com [2010-03-17]

New World Encyclopedia www.newworldencyclopedia.org [2010-04-13]

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Schnorr www.schnorr.com [2010-01-26]

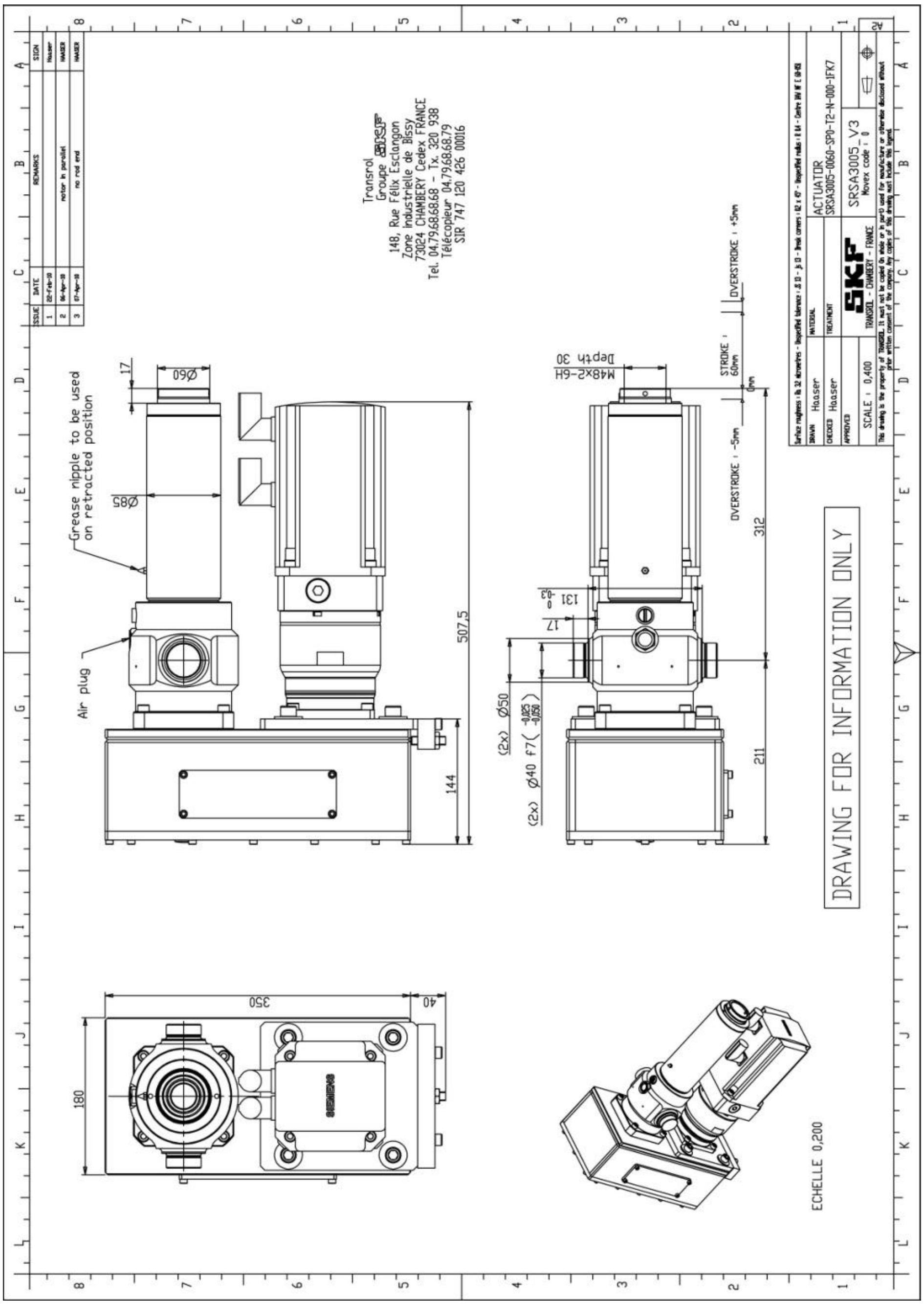
Appendix B: Linear actuators - Comparison Data

Linear Actuators - Comparison Data

Manufacturer	Type of unit	Product number	Stroke (mm)	Linear Speed (mm/s)	Nominal Force (kN)	Torque Input (Nm)	Rotational Speed (rpm)	Length[1] (mm)	Width[2] (mm)	Height[3] (mm)	Precision (mm/mm)	Screw Type	Screw Diameter (mm)	Weight (kg)
SKF	Modular Electromechanical Cylinder	SRSA 3005-xxx-SL1-X82R4B	500	81,3	52,9	n/a	n/a	1062	158	158		Planetary Roller Screw	30	47,2
	Modular Electromechanical Cylinder	SRSA 3005-xxx-SP1-X82R4B	500	81,3	52,9	n/a	n/a	583	158	280		Planetary Roller Screw	30	47,2
	Modular Electromechanical Cylinder	SRSA 3905-xxx-SL1-X82R5B	800	65	63,3	n/a	n/a	1154	185	185		Planetary Roller Screw	39	68,2
	Modular Electromechanical Cylinder	SRSA 3905-xxx-SP1-X82R5B	800	65	63,3	n/a	n/a	673	205	420		Planetary Roller Screw	39	68,2
	Modular Electromechanical Cylinder	SVSA 4001-xxx-SL1-X43P5B	1000	8,3	62,3	n/a	n/a	993	115	115		Recirculating Roller Screw	40	34,9
	Modular Electromechanical Cylinder	SVSA 4001-xxx-SP1-X43P5B	1000	8,3	62,3	n/a	n/a	583	135	280		Recirculating Roller Screw	40	34,9
CRD Drives		EC5 (Vac)	0-1810	11	50	(3 kW)	1400	960,5	200	272		Recirculating Roller Screw	105	
		EC5 VRS (ballscrew) (Vac)	0-1820	15	50	(1,1 kW)	1400	961,5	200	272		Recirculating Roller Screw	105	
Edrive	Eliminator HD Linear Actuator	HD516-06 †	152,4	355,6	71,68	152,1	1780	659,638	212,852	436,626		Ball Screw	50	79,4
	Eliminator HD Linear Actuator	HD618-06 †	152,4	248,92	80,64	170,7	1450	682,752	228,6	454,152		Ball Screw	63	108,9
	Eliminator HD Linear Actuator	HD625-06 †	152,4	365,76	112	396,1	1100	752,602	228,6	454,152		Ball Screw	80	112,0
	MT Precision Linear Actuator	MT512-06 †	152,4	142,24	53,76	60,3	1350	581,152	212,852	436,626		Ball screw	101,6	82,6
	MT Precision Linear Actuator	MT515-06 †	152,4	142,24	67,2	75,7	1350	631,952	212,852	436,626		Ball screw	101,6	88,0
Exlar	FT Series Linear Actuator	FT60-1206	305	201	90,8	104,0	2000	561,1	254	363,7	0,0004-0,001	Planetary Roller Screw	76,2	45,0
	FT Series Linear Actuator	FT60-1212	305	401	90,8	194,0	2000	561,1	254	363,7	0,0004-0,002	Planetary Roller Screw	76,2	45,0
	FT Series Linear Actuator	FT80-1206	305	175	178	220,0	1750	743,6	323,9	440,1	0,0004-0,001	Planetary Roller Screw	101,6	86,0
	FT Series Linear Actuator	FT80-1212	305	351	178	401,0	1750	743,6	323,9	440,1	0,0004-0,002	Planetary Roller Screw	101,6	86,0

1. Length = (Web Direction) with standard rod end, without stroke
2. Width = (Across Web Direction)
3. Height = (Vertical)
† - U-Parallel Offset Motor Configuration

Appendix C: **SKF Linear unit**



Appendix D: **Kistler load cell**

Force

KISTLER
measure. analyze. innovate.

Strain Gage Load Cell

Type 4576A...

for Tension and Compressive Forces, 0,5 ... 200 kN

This tension and compression sensor Type 4576A... with its compact construction is designed for heavy-duty use in rough environments as well as for laboratory and test purposes.

- Measuring ranges from 0 ... 0,5 kN up to 0 ... 200 kN
- Accuracy better than 0,25 %FSO
- Made of stainless steel
- Simple mounting
- Compact size

Description

Load cells Type 4576A... operate using strain gage technology. The measuring unit contains an applied strain gage full bridge which converts the affecting energy to an electrical signal.
A metric thread is cut in the middle axis through which the measurement force is fed either by means of a load button or an application-related screw part. To obtain best results, the load cell must be mounted on a plane flat surface.
Lateral forces within an angular range of ±2,5 ° to the horizontal can be neglected. In case of greater lateral forces, constructive methods must be taken to lead the lateral forces away from the sensor (for example by levers held by roller bearings, movable bearings). The use of the integral screw holes guarantees a simple mounting possibility for the sensor.

Application

Tension and compression load cell Type 4576A... is an all-round instrument for both static and dynamic measurements. Made of corrosion resistant steel, the sensor can be integrated easity into existing structures.

Applications include:

- Press-fit operations
- Draw-pull forces
- Spring power measurements
- Measurements of cutting forces
- Force measurements on mounting devices
- Functional tests

Technical Data

Direction of measurement (calibration in compression direction)		tension/ compression
Measuring ranges	kN	0 ... 0,5 up to 0 ... 200
Limiting force	%	150
Rupture force	%	>250
Dynamic load	%	70 (recommended) 100 (maximum)
Operating temperature range	°C	–30 ... 80
Rated temperature range	°C	15 ... 70
Accuracy (Combined value for non-linearity, hysteresis and repeatability)	%FSO	≤±0,25
Temperature influence		
on zero	%FSO/K	≤0,02
on span	%FSO/K	≤0,02
Weight	kg	≈0,25 ... 5,2
Material		stainless steel 1.4542
Degree of protection:		(IEC/EN 60529)
measuring ranges to 10 kN		IP52
measuring ranges of 20 kN		IP67
Cable port:		
measuring ranges to 50 kN		radial
measuring ranges of 100 kN		tangential
Mounting:		
measuring ranges to 2 kN		3 clearance holes with edges for three-point-support
measuring ranges of 5 kN		6 or 8 clearance holes

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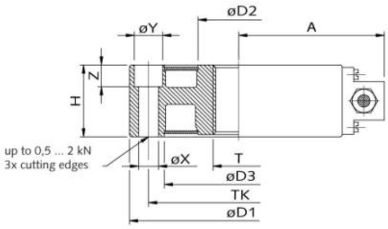
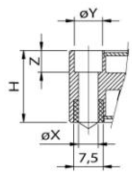
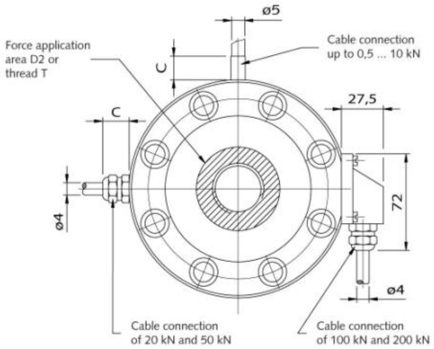
Dimensions

Measuring Range [kN]	Dimensions [mm]										Threaded Hole T	Holes T on TK	Weight [kg]	Natural frequency [kHz]
	øD1	øD2	øD3	H	A	C	TK	øX	øY	Z				
0 ... 0,5	54,5	15	35,5	16	–	10	45	4,5	8	4,6	M8x1,25	3	0,25	>2
0 ... 1	54,5	15	35,5	16	–	10	45	4,5	8	4,6	M8x1,25	3	0,25	>3
0 ... 2	54,5	15	35,5	16	–	10	45	4,5	8	4,6	M8x1,25	3	0,25	>5
0 ... 5	54,5	15	35,5	16	–	10	45	4,5	8	4,6	M8x1,25	6	0,25	>8
0 ... 10	54,5	15	35,5	16	–	10	45	4,5	8	4,6	M8x1,25	6	0,25	>12
0 ... 20	79	22	59	25	–	15	68	4,5	8	4,6	M12x1,5	8	0,65	>4
0 ... 50	119	44	94	35	–	15	105	6,6	11	6,8	M24x1,5	8	2	>3
0 ... 100	155	60	109	50	105	–	129	13,5	20	13	M36x3	8	5	>3
0 ... 200	155	60	109	50	105	–	129	13,5	20	13	M36x3	8	5	>5





Note:
Measuring ranges ≤2 kN are equipped with edges within the clearance holes, so they are 1,5 mm higher.



Assembly requirements for support area:

Height		≈ sensor height
Hardness	HRC	60
Evenness	µm	<20
Parallelism	µm	<50
Mechanical strength of screws		12.9

Electrical Specifications

Bridge resistance:		
Foil strain gage, full bridge circuit	Ω	350 nominal*
Supply voltage:		
recommended	VDC	5
maximum	VDC	10
Sensitivity	mV/V	1,5 ±0,25 % standardized
Optional sensitivity	mV/V	1,0 ±0,25 % standardized

* Deviations may occur.

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Electrical Connection

4-wire, shielded, flexible PVC-cable, length approx. 2 m, with 6 pin. connector.
Binder (round connector, 18 mm DIN 45326)
Degree of protection: IP40

Pin	Cable	Description
1	brown	supply voltage(–)
2	white	supply voltage (+)
3	(blank)	shield
4	yellow	signal output (+)
5	green	signal output (–)
6	–	not connected

Included Accessories

- None

Optional Accessories

- Connection cable, 5 m, 6 pin/6 pin KSM071860-5
- Connection cable, 5 m, 6 pin/open ends KSM103820-5

Ordering Key

Type 4576A □ □

Measuring Range [kN]	
0,5	**0,5**
1	**1**
2	**2**
5	**5**
10	**10**
20	**20**
50	**50**
100	**100**
200	**200**

Option*	
Sensitivity standardized 1 mV/V	**S**

Without option = sensitivity 1,5 mV/V

Ordering Example: **Type 4576A10S**

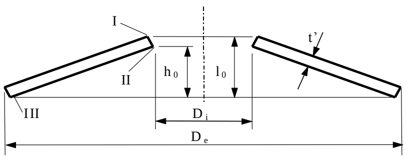
Load cell Type 4576A..., measuring range 0 ... 10 kN, with optional sensitivity standardized 1 mV/V.

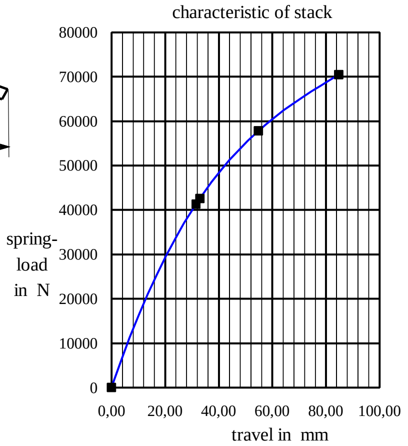
Ordering Example: **Type 4576A10**

Load cell Type 4576A..., measuring range 0 ... 10 kN, without option = sensitivity 1,5 mV/V.

Appendix E: **Lesjöfors disc spring package**

Mubea Version 19.7.98	**Disc Springs, Data Sheet** **group 2** part./drawing no.: **18 0126** project:
2010-05-28	Muhr und Bender, Tellerfedern und Spannelemente GmbH, Postfach 120, 57564 Daaden phone.: sales: 02743/806-184, -194, Fax.:-188; engineering: 02743/806-268, -236, -131, Fax.: -292





dimensions			
outer diam.:	D_e=	150,000	mm
inner diam.:	D_i=	61,000	mm
thickness:	t=	5,000	mm
red. thickness:	t'=	5,000	mm
spring height:	l_0=	10,300	mm
data	h_0=	5,300	mm
h_0/t= 1,060	h_0'=	5,300	mm
h_0'/t'= 1,060	D_e/D_i=	2,459	
stack:	32 springs as		
16 pacets	2 times stacked		

	load points of one spring				calculated stresses				load points of stack		
s/h_0'	load-point	height l mm	travel s mm	load F N	σ_I MPa	σ_{II} MPa	σ_{III} MPa	σ_{OM} MPa	height l mm	travel s mm	load F N
0%	0	10,300							244,800		
37%	1	8,328	1,972	20625	-1408	215	657	-500	213,243	31,557	41250
65%	2	6,875	3,425	28875	-2271	548	1045	-869	189,994	54,806	57750
39%	3	8,240	2,060	21257	-1464	231	683	-523	211,840	32,960	42514
	Flat	**5,000**	**5,300**	**35207**	**-3165**	**1196**	**1426**	**-1345**	**160,000**	**84,800**	**70414**

specification

material:	50 CrV 4	Youngs-modulus:	206000 MPa
surface finish:	shot peening	temperature:	20 °C
corrosion prot.:	phosphated and oiled		

fatigue life of Mubea springs **please contact Mubea!**

more than 15 pacets!

travel: 23,25 mm between l 1 : 213,24 mm and l 2 : 189,99 mm

remarks

Load tolerance: +10 / -5% at 75% of h0 of one spring

tolerance inner diam.:	61,000	mm	to	61,300	mm
tolerance outer diam.:	149,600	mm	to	150,000	mm