Design of an Active Wire Rope Tensioner

Extending wire rope lifetime in offshore cranes

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Huisman
Equipped for impact

Design of an Active Wire Rope Tensioner

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by

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Preface

Before you lies the thesis that represents the culmination of my journey toward earning a master's degree in mechanical engineering. The process of writing this thesis has required countless hours and faced me with numerous challenges. It has been an invaluable experience, teaching me lessons that lectures and books never could.

My lifelong fascination with large machines and their mechanisms inspired me to pursue a degree in mechanical engineering after high school. This decision proved to be the right one, as I continued with a master's degree in mechanical engineering to gain a deeper understanding of these machines, enabling me to design them myself. This project became the crown jewel of my academic journey, preparing me to become a skilled engineer.

I am deeply grateful to Huisman for granting me the opportunity to complete my graduation project at their company. Working alongside industry leaders in the field of large-scale equipment has been an extraordinary experience. I would especially like to extend my gratitude to Joost Sweep for his daily supervision and invaluable insights, which were instrumental in shaping this project. My sincere thanks also go to Wouter van den Bos for his guidance and for bridging the academic and professional aspects of this work.

Lastly, I would like to thank my family and friends for their incredible support over the past three years. Your encouragement has been a vital aspect of my journey to obtaining this master's degree. A special thanks to my parents for their unwavering support over the past 23 years, which has enabled me to reach this milestone.

B. van Empel
Delft, February 2025

Summary

This thesis presents the design process of an active wire rope tensioner, a device designed for cranes, that serves two primary functions. Firstly, it increases the tension in the wire rope during spooling, ensuring neat and tight winding on the drum to reduce damage to the wire rope and cutting-in problems. Secondly, it reduces the required lower block weight, thereby improving the crane's load curve.

The research has commenced with a study of relevant background information and working principles. A thorough literature and patent review has followed, revealing a gap in the state-of-the-art of tensioning devices that both prevent cutting-in and reduce the required lower block weight. In the conceptual design phase, seven innovative concepts have been generated based on the literature review and current principles. These concepts have been assessed for their capability to meet the requirements and their performance against the key performance indicators (KPIs). Consequently, five concepts remain, with the clamping track concept emerging as the most promising.

The clamping track concept utilizes a chain drive with clamps attached to it that press on the wire rope. By applying force to the chain, the tension in the wire rope can be manipulated. Detailed development of the clamping track concept has yielded a conceptual design with three potential deployment scenarios, each requiring a slightly different version of the active wire rope tensioner and offering distinct advantages and disadvantages. Scenario 1 offers the greatest possible lower block weight reduction but comes with the cost of a more complex and riskier system. Scenario 3 does not allow for any lower block weight reduction but is simpler and safer. Scenario 2 strikes a balance between the two.

This conceptual design aims to bolster confidence in the active wire rope tensioner principle and represents a significant step towards further research and development.

Samenvatting

De scriptie die voor u ligt, presenteert het ontwerpproces van een actieve kabelspanner. Een apparaat, ontworpen voor kranen, dat twee primaire functies vervult. Ten eerste het verhogen van de spanning in de kabel tijdens het spoelen, wat zorgt voor een nette en strakke winding op de trommel om schade aan de kabel en problemen met insnijden te reduceren. Ten tweede het verminderen van het vereiste gewicht van het onderblok, waardoor de hijscapaciteit van de kraan wordt verbeterd.

Het onderzoek begint met een studie van relevante achtergrondinformatie en werkingsprincipes. Vervolgens wordt een grondige literatuur- en octrooistudie uitgevoerd. Hierbij is een gat in de 'state-of-the art' van kabelspanners ontdekt met betrekking tot kabelspanners die zowel insnijden voorkomen als het vereiste gewicht van het onderblok verminderen. In de conceptuele ontwerpfase zijn zeven innovatieve concepten gegenereerd op basis van de literatuurstudie en huidige principes. Deze concepten zijn beoordeeld op hun vermogen om aan de eisen te voldoen en hun prestaties ten opzichte van belangrijke prestatie-indicatoren (KPIs). Na deze selectie zijn er nog vijf concepten over, waarbij het 'clamping track' concept als het meest veelbelovend naar voren komt.

Het 'clamping track' concept maakt gebruik van een kettingaandrijving met klemmen die op de kabel drukken. Door kracht op de ketting uit te oefenen, kan de spanning in de kabel worden gemanipuleerd. Detaillering van het 'clamping track' concept leverde een conceptueel ontwerp op met drie mogelijke inzetscenario's, die elk een iets andere versie van de actieve kabelspanner vereisen en verschillende voor- en nadelen bieden. Scenario 1 biedt de grootst mogelijke verlaging van het blokgewicht, maar gaat gepaard met een complexer en risicovoller systeem. Scenario 3 staat geen verlaging van het onderblok toe, maar is eenvoudiger en veiliger. Scenario 2 vormt een balans tussen de twee.

Dit conceptuele ontwerp van de actieve kabelspanner is bedoeld om het vertrouwen in het principe van de actieve kabelspanner te vergroten en vormt een belangrijke stap naar verder onderzoek en ontwikkeling.

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Nomenclature

Abbreviations

Abbreviation	Definition
COF	Coefficient Of Friction
COG	Centre of Gravity
ISO	International Organization for Standardization
KPI	Key Performance Indicator
LB	Lower Block
LEC	Leg Encircling Crane
LMP	Load Measuring Pin
MBL	Minimum Break Load
PU	Polyurethane
RHLL	Right-Handed Lang's Lay
RHOL	Right-Handed Ordinary Lay
SWL	Safe Working Load
SWOT	Strengths, Weaknesses, Opportunities and Threats
UB	Upper Block
UCU	Undercut U-groove

1
Introduction

The offshore industry has been at the forefront of technological advancements, particularly driven by the global shift towards renewable energy. As the demand for sustainable energy sources continues to rise, the offshore market, especially in wind energy, has seen remarkable growth. The increased demand for wind energy and the technological advancements in that field have increased the size of wind turbines significantly over the years. These enormous wind turbines require larger cranes with more lifting capacity and larger outreach to install them.

Huisman, a leader in the design and manufacturing of heavy construction equipment, has adapted to these needs. Huisman designs and builds various types and sizes of cranes tailored to meet specific applications. In recent years, the demand for larger cranes has surged, driven by the increasing dimensions of offshore wind turbines [1]. Huisman has responded to this demand by innovating and scaling up their crane designs to meet the evolving requirements of the offshore renewable energy sector.

1.1. Background

Despite significant advancements, Huisman and other crane manufacturers continue to face challenges in the design and operation of their cranes. Notably, issues such as wire rope damage resulting from low tension and the substantial weight of the lower block have a considerable impact on crane operations. The following subsections provide a detailed examination of these challenges.

1.1.1. Wire rope damage

Wire ropes are critical components of cranes as they are the driving mechanisms behind the luffing of the boom and the hoisting systems of a crane. In a crane system, the wire rope is attached to a load on one end and reeved through a series of sheaves before being connected to the drum on the opposite end. The drum serves a dual purpose: it stores the wire rope and facilitates its spooling in and out to manoeuvre the load. This is often done with multi-layer winding systems. In these systems the wire rope is wound around a single drum multiple times. This allows for compact storage of a large amount of wire rope. Therefore it is often used in hoisting systems.

Wire ropes are versatile yet complex components, exhibiting behaviour markedly different from conventional solid materials. One extensively studied aspect of wire ropes is their wear, a highly complex process with many contributing factors still not fully understood [2]. Weiskopf [3] identifies 18 distinct factors influencing the lifetime of wire ropes in multi-layer winding systems. These factors can be broadly categorized into three groups: drum design, wire rope type, and the forces acting on the wire rope. The dominant wear mechanisms that determine a wire rope's lifetime are highly dependent on the crane's geometry and usage.

One critical factor is the tension applied to the wire rope during winding onto the drum. Uneven tension can result in irregular winding, leading to high localised pressures that damage the wire rope [4][5]. Properly arranged winding can significantly enhance the wire rope's lifespan. Weiskopf [3] found that increasing the tension during the winding process ensures a neat arrangement, resulting in a 20-30%

improvement of the lifetime. Subsequent studies have shown even greater improvements. Wehking [6] reported up to a 40% increase, and Hofmann and Schmidt [7] found an 85% increase in wire rope lifespan due to increased tension during winding.

Next to wire damage due to uneven winding an even more damaging problem can occur. Wire rope cutting-in is a significant issue in multi-layer winding systems. Cutting-in occurs when the wire rope forces itself between the lower layers of the winding, as illustrated in Figure 1.1. This phenomenon typically arises during heavy lifting operations, where large tensional forces are exerted on the wire rope. The resultant pressure and friction from cutting-in can cause severe damage to the wire rope, leading to premature failure of the wire rope.

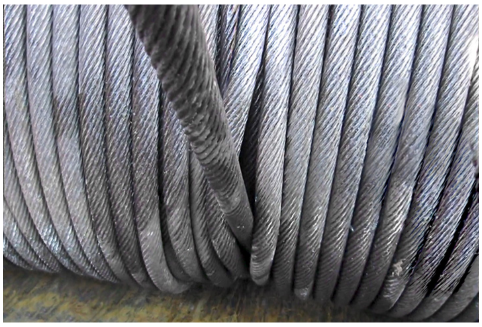


Figure 1.1: Photo of a drum where cutting-in occurred [8]

Cutting-in issues have been a longstanding challenge. Crane manufacturers such as Liebherr [8] and Huisman have proposed various procedures to mitigate this problem. These procedures involve using a heavy external load or external tuggers to regularly spool the drum under high tension. Additionally, the crane's design can be modified to reduce or prevent cutting-in. Two effective aspects to achieve this are the weight of the lower block and the size of the drum. The reasons for not selecting this approach in this study are elaborated upon in section 1.3.

The tension, also referred to as line pull, is critical in preventing wire rope cutting-in. Inadequate or uneven tension during the spooling process can lead to cutting-in problems. Ensuring that the winch is wound with sufficient pre-tension can effectively mitigate this issue. During spooling, there exists a critical tension threshold. If the tension falls below this threshold, cutting-in can occur [7]. This critical tension value is influenced by factors such as the drum type, the wire rope type, and the number of layers on the drum [9].

Wire rope, sometimes referred to as cables, consists of strands, which in turn are composed of individual wires. Each wire rope can be tailored to meet specific requirements. There are two primary manufacturing methods for wire ropes: Lang's lay and ordinary (or regular) lay. The distinction between these methods lies in the lay direction of the wires within the strands, as depicted in Figure 1.2. Lang's lay cables offer greater flexibility for bending but are more susceptible to damage. Both manufacturing techniques can produce either left-handed or right-handed cables.

In conventional wires, a twisting effect known as spinning occurs when tension is applied. Non-spin wires mitigate this effect by designing the wire such that the inner and outer layers twist in opposite directions, resulting in a non-spin wire. These wires are used in single-fall applications to prevent the lower block from rotating under load. However, the disadvantages of non-spin wire ropes are their lower fatigue life due to bending and their reduced resistance to external torque [11].

The type of wire significantly influences its resistance to cutting-in. Figure 1.3 presents the results of tests conducted by Huisman. Lang's lay wire rope exhibits the poorest resistance to cutting-in, whereas ordinary lay non-spin wire demonstrates significantly better resistance. This difference is due to the lay direction of the strands in the wire rope. In ordinary lay wire ropes, the strands lie parallel to the wire rope, causing adjacent windings to interlock like teeth.

Figure 1.3 illustrates the resistance to cutting-in as a function of the applied load on the wire rope, relative to the maximum permissible number of layers. Cutting-in can occur after a few layers, dependant upon the tension with which the lower layers are spooled. The graph indicates the existence of a critical

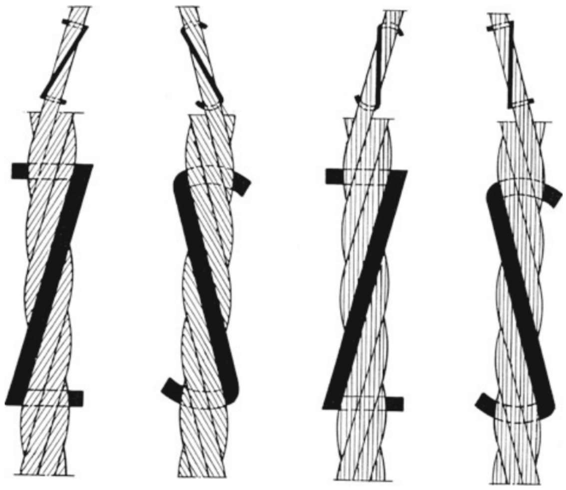


Figure 1.2: Schematic drawings of different types of wire rope constructions [10]

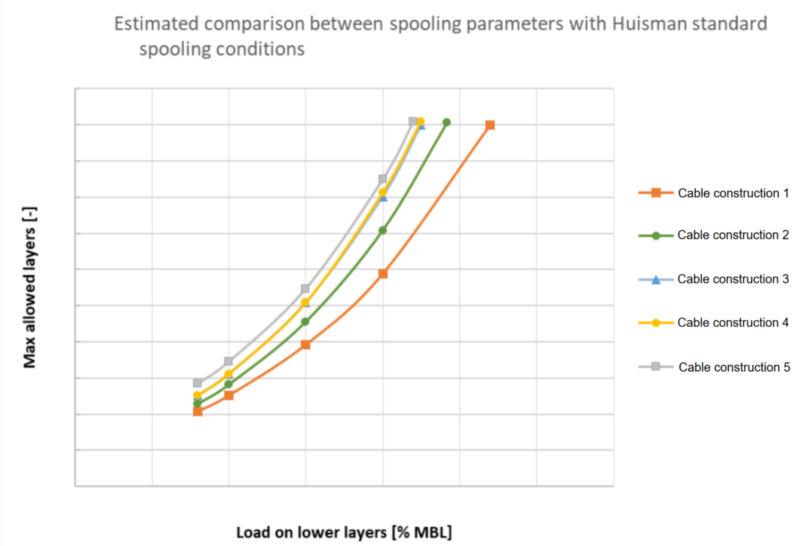


Figure 1.3: Effect of wire rope type on cutting-in based on tests performed by Huisman[12]

tension value, beyond which cutting-in does not occur for a specified number of layers. This critical tension value is system-specific.

Determining the critical tension value for which cutting in does not occur has proven to be a challenge. Not only is cutting-in influenced by a number of factors, but also measuring or determining whether cutting in has occurred is is equally complex. Multiple experiments in order to determine the critical tension value have been performed by Huisman [9] [12] [13]. However, due to the aforementioned difficulties, these tests did not yield a definitive critical tension value.

After thoroughly reviewing literature on wire rope damage in multi-layer systems, analyzing multiple cutting-in test results conducted by Huisman, and consulting with wire rope experts within the company, it was concluded that a line pull of *Confidential*% of the minimum break load (MBL) is adequate to prevent cutting-in. Ensuring his critical tension value is always met during hoisting will also reduce other types of wire rope damage caused by insufficient tension.

1.1.2. Lower block weight

The lower block is a crucial component of the crane, connecting the wire rope to the hook. The wire rope is reeved multiple times between the lower block and the upper block, creating several vertical segments known as falls. By decreasing the torque on the drum, the weight of the lower block will cause it to descend. This will only happen if the lower block is heavy enough to overcome the own weight of the wire rope and the sheave inefficiencies. This causes a need for heavy lower blocks. The substantial weight of the lower block directly impacts two aspects of crane operation.

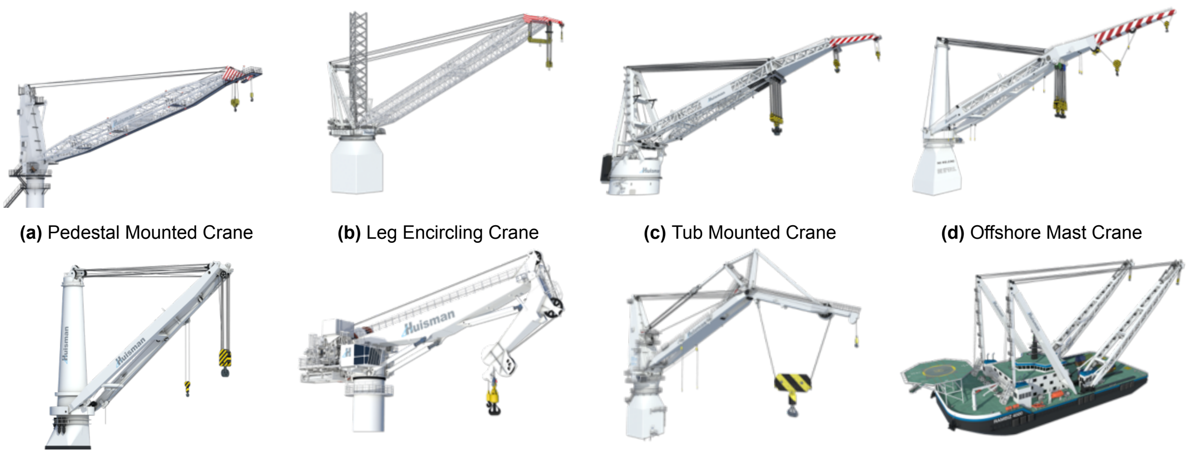
The most apparent impact is that any additional mass added to the lower block directly reduces the crane's lifting capacity. Therefore, reducing the mass of the lower block would directly increase the crane's safe working load (SWL).

As cranes have increased in size over the years, a new challenge has emerged: fatigue damage caused by the weight of the lower block. This issue arises in cranes where the boom is longer than the vessel on which it is mounted, a common scenario for leg encircling cranes (LEC) on jack-up vessels. On these vessels, the boom rest is positioned along the boom rather than at its end. The significant distance between the boom rest and the lower block creates a bending moment. Due to the vessel's motion during transport, this results in fatigue damage in the boom, significantly impacting the crane's service life [14]. An effective way to mitigate fatigue damage is to reduce the weight of the lower block.

Lower blocks are currently designed with a specified minimum target weight, which is necessary for proper lowering of the lower block, as detailed in subsection 2.1.1. To achieve this minimum weight, the lower block is either over-dimensioned, additional dead weight is incorporated, or a combination of both methods is employed. Consequently, lower blocks can be made lighter if the minimum target weight is reduced. However, a minimum weight will always be required due to structural constraints.

1.1.3. Crane and rigging configurations

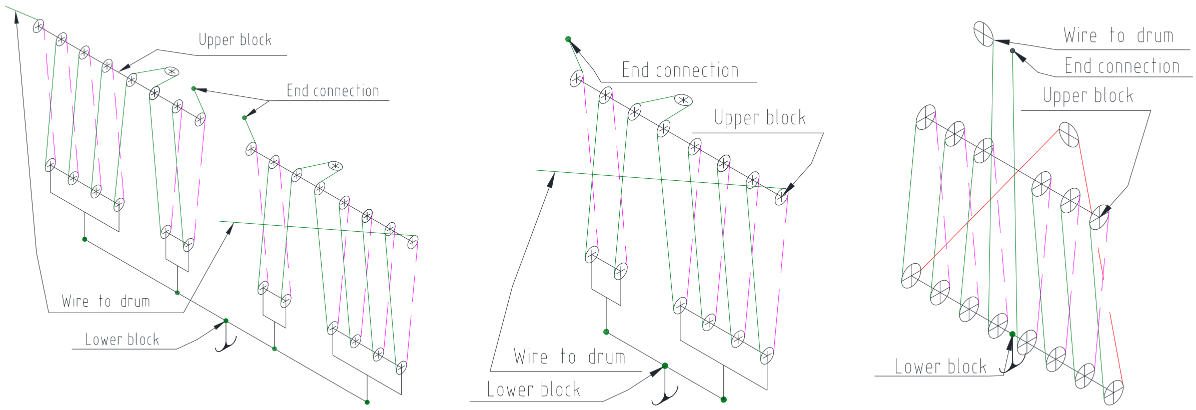
Cranes are large pieces of equipment engineered for lifting, moving, and lowering heavy objects. Each crane is designed to fulfil a specific function. Based on their intended use and operational constraints, cranes can be classified into various categories, as illustrated in Figure 1.4. These categories are differentiated by their operational roles, the type of base they are mounted on, and their dimensions. While cranes within a category share a similar design philosophy, the detailed specifications of each crane are customized to meet the specific requirements of the user. Consequently, nearly every crane is unique.



(a) Pedestal Mounted Crane **(b)** Leg Encircling Crane **(c)** Tub Mounted Crane **(d)** Offshore Mast Crane
(e) Heavy Lift Mast Crane **(f)** Knuckle Boom Crane **(g)** Hybrid Boom Crane **(h)** Sheer Leg Crane

Figure 1.4: Overview of ship mounted crane types Huisman produces

One of the characteristics of a crane that is relevant to the design of an active wire rope tensioner is the rigging configuration. The rigging configuration of a crane refers to the selection of wire rope diameter and length, as well as the number of falls utilized between the lower block and the upper block. But also the type of rigging.



(a) Double tackle **(b)** Single tackle - left to right **(c)** Single tackle - Inside to outside

Figure 1.5: Overview of different rigging types applied in cranes

Designing the rigging configuration of a crane involves making multiple trade-offs. The wire rope diameter, wire rope length, and the number of falls are all interdependent. Choosing a larger wire rope diameter reduces the number of falls required, but significantly increases the size of the sheaves and the line pull. Conversely, opting for a smaller wire rope diameter results in smaller sheaves and less line pull, but requires more falls, necessitating a drum with more layers and reducing rigging efficiency.

Additionally, the type of rigging must be considered. The first decision is whether to use single or double tackle. Double tackle rigging uses 2 separate sets of falls connected to the lower block as shown in Figure 1.5a. Single tackle is when only 1 set of falls is used and only 1 wire runs back towards the drum. The first two riggings show a rigging configuration where the falls are continuously rigged from one side to the other. This is the simplest manner, but in single tackle this could cause uneven tension resulting in undesired tilting of the lower block. To prevent this, the rigging shown in Figure 1.5c can be used. In this configuration, the wire rope starts in the middle and moves outward. Upon reaching the end, it is connected to the opposite end with a turn sheave, then rigged back inward until it reaches the middle again, where the end connection is placed.

Although some types of cranes have a common rigging configuration, knuckle boom cranes mostly have only 1 fall for example, there is no standard rigging configuration within Huisman. The trade-offs mentioned before should be made for each crane, which result in a lot of possible rigging configurations.

1.2. Objectives

These challenges can be solved with a single device: an active wire rope tensioner. This device can increase the tension in the wire rope during spooling, thereby improving the winding quality on the drum, reducing wire rope damage, and preventing cutting-in. Additionally, the device can be used to drive the wire rope, allowing for the use of a lighter lower block. This research aims to design an innovative wire rope tensioning system. The following research objective and research questions are formulated to achieve this goal and provide guidance throughout the research.

The following research objective is defined:
Design an active wire rope tensioner system to reduce wire rope damage and enable the usage of lighter lower blocks.

This research objective is reached by answering the following research questions:

1. What are the requirements and KPIs for a successful system?
2. How can the tension in the wire rope be manipulated?
3. Which concept has the most potential based on the requirements and criteria?

1.3. Scope

The research described in this thesis limits itself to the design process of the conceptual design of an active wire rope tensioner. A design process is an iterative process with steps that dive deeper into the specifics at each step forward. Because of the limited time available the goal of the research is to generate a conceptual design. Areas where details could not explored due to the time limit should be clearly reported in the recommendation.

As described in subsection 1.1.1 cutting-in problems could be reduced by making use of certain procedures. These procedures are outside the scope the this research. The tension generation should be able to be performed with a mechanism or technique which only uses parts of the crane. Procedures using an external weight of external tuggers are therefore excluded.

Reducing wire rope damage and preventing cutting-in could be achieved by altering the design of the drum. A larger drum can accommodate the same wire storage capacity with fewer layers of wire rope. Damage to the wire rope decreases with fewer layers due to a reduced amount of layer changes and lower pressure on the underlying layers. Additionally, cutting-in can be prevented by using fewer layers on the drum. Research conducted by Huisman [12] indicates that using a maximum of *Confidential* layers with *Confidential* windings prevents cutting-in in cranes without any measures to increase tension. To achieve the same wire rope storage, drums would need to approximately double in diameter. This significant increase in size poses challenges for application in cranes, not only because of the increased size, but also because of the increased inertia. While acknowledging the positive effects of reducing the number of layers on wire rope damage, this study focuses on mitigating wire rope damage by increasing and controlling tension during hoisting.

Further detailing of the researches scope is described in the following subsections. Firstly the description of the functions in subsection 1.3.1 details on which functions the research focusses. The requirements of the active wire rope tensioner are described in subsection 1.3.2. These requirements, along with the KPIs described in subsection 1.3.3, establish the parameters of the system to ensure it fulfils its functions.

1.3.1. Functions

A successful active wire rope tensioner has to fulfil three functions. It has to increase the tension in the wire rope during hoisting, decrease the required lower block weight and adapt to high speed operations. These functions are detailed below.

1. Increase the tension in the wire rope during hoisting
The main objective of the active wire rope tensioner is to increase the tension in the wire rope during hoisting to reduce wire rope damage and increase the lifetime of the wire rope. How this tension is generated is one of the main research questions and will be therefore at the centre of the research. The required tension increase varies for each crane, depending on the properties of the wire rope used and the crane's geometry.

2. Decrease the required lower block weight
The secondary goal of the active wire rope tensioner is to allow a reduction of the lower block weight. As described before the lower block weight is currently needed to overcome the wire ropes own weight and sheave efficiency. A device can be used to negate the need to overcome the wire ropes own weight.

It is possible that the solution space of functions 1 and 2 overlap significantly. But they are considered separately to allow more freedom in finding possible solutions.

3. Adapt system to allow high speed operations
The operational efficiency of cranes, and the vessels they operate on, is of paramount importance to their users. Therefore, it is crucial to consider how an active wire rope tensioner might impact the operational speeds of the crane. Given the significance of this factor, it is treated as a function. By defining it as a function, it will be central to the concept design phase later in the process.

1.3.2. Requirements

In order to determine if a concept is viable option for the active wire rope tensioner a number of requirements have been established. Each viable concept should adhere to all of the requirements below. The requirements ensure the device is suitable for its intended application.

1. The system must achieve a line pull of *Confidential*% MBL at the drum during hoisting, without significantly increasing the lower block weight

The active wire rope tensioner achieves its main goal, reducing wire rope damage, by ensuring *Confidential*% MBL of line pull is reached at the drum during hoisting. It should be noted that this cannot be done by simply making the lower block heavier. This would increase the line pull, but it would significantly impact the cranes capabilities. Therefore the lower block weight can be increased with a maximum of 5%.

2. The system must have a net positive impact on the wire ropes lifetime

It is possible that the added active wire rope tensioner itself will damage the wire rope slightly in order to generate the required tension. This is determined to be acceptable, as long as there is a net positive effect on the wire rope lifetime.

3. The system shall not create an unacceptable risk

Safety is very important in the design and operations of cranes. Therefore the addition of the system cannot create an unacceptable safety risk. Determining whether a risk is acceptable or not will be done using a risk analysis in consultation with experts.
Next to safety risk, operational risks also need to be acceptable. Malfunctioning of the system resulting in an inoperative system is considered an operational risk.

4. The system must work for different types of crane and rigging configurations

As described in subsection 1.1.3 almost each crane, and its rigging, is unique. The type and extend of wire rope damage is determined by the usages of a crane and is not type specific. The intend of this device is to reduce wire rope damage in cranes, regardless of their type. Therefore it is important that the working principle of the system can be applied to different types of cranes and rigging configurations.

1.3.3. Key Performance Indicators

The requirements discussed in subsection 1.3.2 determine which concepts are suitable solutions for the active wire rope tensioner. In order to determine which of these concepts has the most potential, the Key Performance Indicators (KPIs) described in this subsection have been established. The order they are presented in is the order of importance for a successful system.

1. Impact on operating speed

The impact on operating speed caused by the active wire rope tensioner is considered the most important KPI. If the active wire rope tensioner significantly affects operational speed, it could substantially impact the operational efficiency of the crane and the ship, rendering the system less attractive.

The impact on operating speed can be subdivided into two aspects. The first aspect is the speed at which the system operates when it is in use. The second aspect is the impact of the system when it is inactive. An ideal system does not affect the nominal operating speeds when in use and has no impact on the speed of the crane when inactive.

2. Added procedures

An ideal system does not introduce any additional procedures during crane operation. This involves multiple aspects. Firstly, it considers whether the crane operator needs to actively decide to use the system and manually select its activation. Secondly, it is important to assess whether any additional operational steps are added. Ideally, no additional steps should be required for the crane's operation.

3. Decrease required lower block weight

Decreasing the required lower block weight is deemed the least important KPI, despite the possible benefits mentioned in subsection 1.1.2. This is because it is viewed as a beneficial possible side effect of an active wire rope tensioner rather than a primary goal. Additionally, decreasing the lower block weight directly opposes the objective of achieving a *Confidential*% MBL line pull, as the wire rope tension relies on the lower block's weight. An active wire rope tensioner has the potential to achieve both goals. However, if prioritisation is necessary, increasing the tension in the wire rope should take precedence.

1.4. Methodology

As mentioned in section 1.2, the goal of this study is to design an active wire rope tensioner that can both increase the tension in the wire rope during hoisting to reduce wire rope damage and prevent cutting-in, and allow the use of a lighter lower block. It is important to emphasise that these two goals inherently counteract each other. Specifically, reducing the weight of the lower block decreases the tension in the wire rope. While the principle of an active wire rope tensioner that achieves both goals is theoretically possible, determining the feasibility of designing such a system is the core focus of this research.

To guide the design process in the right direction, a solid design methodology is required. The process is divided into three phases, each with two steps.

The first phase is the research phase. During this phase, a literature study is conducted to gather the necessary background information and understand the working principles. This helps in determining the functions, requirements, and KPIs. Additionally, a literature review and patent research are performed to identify and study relevant devices.

Next is the concept phase. In this phase, new solutions to the functions established during the research phase are generated. This includes expanding on solutions identified in the literature and patent studies, as well as focusing on innovative, 'out of the box' ideas. All potential solutions are compiled into a morphological overview, which is then used to select different concepts. Using a parametric model, risk assessment, and analysis of the concepts, they are evaluated against the requirements and rated using the KPIs, resulting in the identification of the most promising concept.

The final phase focuses on detailing the most promising concept. Using the parametric model from the concept phase, the exact requirements of the chosen tensioner are specified. This enables the detailed design of the tensioner track. Subsequently, a description of the control system is provided, along with an assessment of the improvement in the crane's load curve.

By following this structured methodology, the project aims to develop a feasible concept of the active wire rope tensioner

1.5. Report structure

Chapter 1 has covered the first half of the research phase, discussing the background information and working principles related to wire rope damage, the effects of the lower block weight, and crane configurations. It also outlined the objective, scope, and methodology.

The second part of the research phase is detailed in chapter 2. This chapter begins with an examination of the relevant working principles encountered in the design of an active wire rope tensioner. Following this, the state of the art section provides a literature review of research and patents pertinent to the conceptual design of an active wire rope tensioner.

The concept phase is described in chapter 3. This section starts with the morphological chart used to generate the seven concepts, which are explained in the subsequent sections. A discussion of the performance in relation to the requirements and KPIs follows, concluding with the selection of the most promising concept for an active wire rope tensioner.

The detailing of the most promising concept is described in chapter 4. This starts with determining the optimal installation location of the tensioner in the crane, after which the requirements of the systems are detailed. This allows the detailing of the mechanical design. The effective improvement of the load curve of the crane is also assessed. The chapter concludes with the control system description and a discussion of the detailed design.

2

Working principles & State of the art

The previous chapter covered the first part of the research phase. This chapter discusses the second half, by exploring relevant working principles encountered in the design of the active wire rope tensioner. These are elaborated upon in section 2.1. Existing research and patents that are relevant to the design process described in this thesis are presented in section 2.2.

2.1. Hoisting and lowering operations

This section delves into the working principles of hoisting and lowering operations in cranes. Understanding these principles is crucial for comprehending the background of certain requirements and KPIs. They are also essential for designing an optimal active wire rope tensioner.

2.1.1. Minimal lower block weight

As described in the introduction the lower block of a crane has a certain minimal weight. The weight of the lower block causes the tension in the wire rope. A certain amount of tension is needed to ensure the lower block lowers properly, and to ensure proper spooling on the drum during hoisting. How to calculate the line pull at the drum for the crane configuration displayed in Figure 2.1 is shown below. These calculations can be scaled to each unique crane configuration.

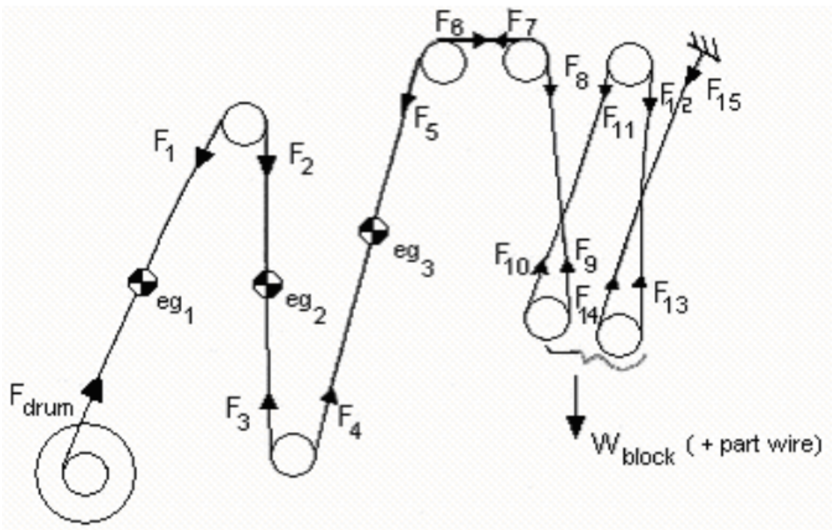


Figure 2.1: Schematic drawing of the reeving of a crane used for lower block calculations [15]

In Figure 2.1 and the equations below the following variables can be found.

- F_n [kN] is the line pull or tension in the wire rope right before or after a sheave.
- F_{drum} [kN] is the line pull or tension in the wire rope at the drum.
- eg [kN] is the own weight of a part of the wire rope.
- η [-] is the efficiency of the sheaves.
- W_{block} [kN] is the weight of the lower block.
- d_{wire} [mm] is the diameter of the wire.
- MBL [kN] is the minimum break load, the maximum tension a wire rope can handle.

$$F_8 = \frac{W_{block}}{1 + \frac{1}{\eta} + \frac{1}{\eta^2} + \frac{1}{\eta^3}} \tag{2.1}$$

$$F_{drum} = ((F_8\eta^2 - eg_3)\eta + eg_2)\eta - eg_1 \tag{2.2}$$

If Equation 2.1 and Equation 2.2 are combined and rewritten the following equation for W_{block} can be found.

$$W_{block} = \frac{(F_{drum} + eg_1 - eg_2\eta + eg_3\eta^2)(1 + \eta + \eta^2 + \eta^3)}{\eta^7} \tag{2.3}$$

Following from Equation 2.3 it is evident that the efficiency of the sheaves have a big impact on the required lower block of the crane. Important to note is that the efficiency value dependent on whether the cranes is hoisting or lowering. During lowering the efficiency value should be used as mentioned above with $\eta_{lowering} \in [0, 1]$. During hoisting the inverse should be taken: $\eta_{hoisting} = 1/\eta_{lowering}$.

The required line pull at the drum needed for proper lowering and spooling is empirically determined by Huisman. This results in the following rules of thumb shown below. The minimum lower block weight must be high enough so it fulfils both the rules of thumb.

$$F_{drum} \geq \textit{Confidential} \quad \text{for proper lowering of the lower block.} \tag{2.4}$$

$$F_{drum} \geq \textit{Confidential} \quad \text{for proper spooling of the drum during hoisting.} \tag{2.5}$$

To visualize the effect of the wire rope's own weight and sheave efficiency, Figure 2.2 was created using Equation 2.1 and 2.2. This schematic illustrates the tension throughout the crane's rigging. The decrease in tension due to the wire rope's own weight is clearly visible. The jumps in tension at the sheaves are caused by the sheave efficiency.

2.1.2. Sheave efficiency

As discussed in subsection 2.1.1, the efficiency of the sheaves in a crane significantly impacts the wire rope tension. Therefore, it is crucial to study sheave efficiency. Sheave efficiency is defined as the ratio of the tension before and after a sheave, expressed mathematically in Equation 2.6.

$$\eta = \frac{F - \Delta F}{F} = 1 - \frac{\Delta F}{F} \tag{2.6}$$

In his study, Kyllingstad [16] concluded that the primary cause of efficiency losses is bending-induced friction within the wire rope, with losses in the bearings being negligibly small. Sheave efficiency can be determined based on two factors. The first factor is the D/d ratio, which is the ratio between the sheave diameter (D) and the wire rope diameter (d). A larger D/d ratio results in higher sheave efficiencies. The second factor is the tension in the wire rope at pulling side of the wire rope. Higher tension in the wire rope improves the efficiency. Based on past research by Rubin [17] and Hecker [18] the following equation to calculate the sheave efficiency was determined by Feyrer [10].

$$\Delta F = (\frac{D}{d})^{-1.33} \cdot (c_0 + c_1 \cdot \frac{F}{d^2}) \cdot d^2 \tag{2.7}$$

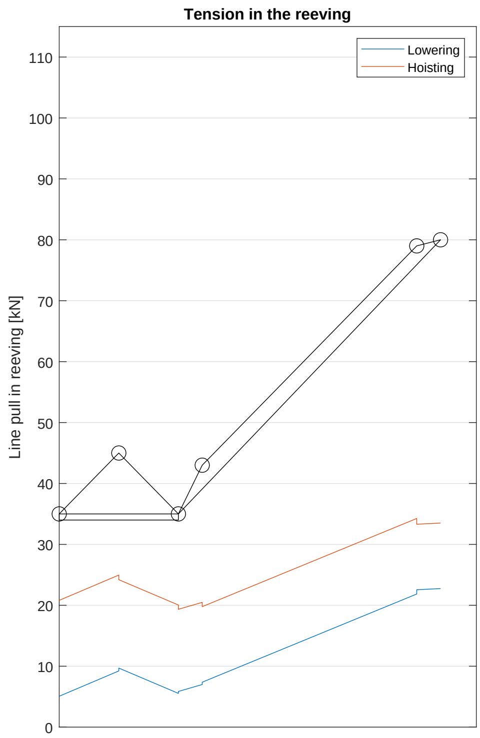


Figure 2.2: Schematic drawing of the tension in the wire rope

The constants c_0 and c_1 need to be empirically determined for each type of wire rope. In his book Feyrer gives an example of the efficiency for different combinations of D/d ratios and tension values, for a certain type of wire rope. Tests conducted by DNV also demonstrated similar efficiency values as Feyer. The results, shown in Figure 2.3, indicate lower efficiencies for wire ropes under low tension. Increasing the tension improves efficiency. These tests were performed for a D/d ratio of 20 and extrapolated for different D/d ratios. Measurements taken with a Huisman crane on a vessel in 2019 showed similar efficiencies as those depicted in Figure 2.3.

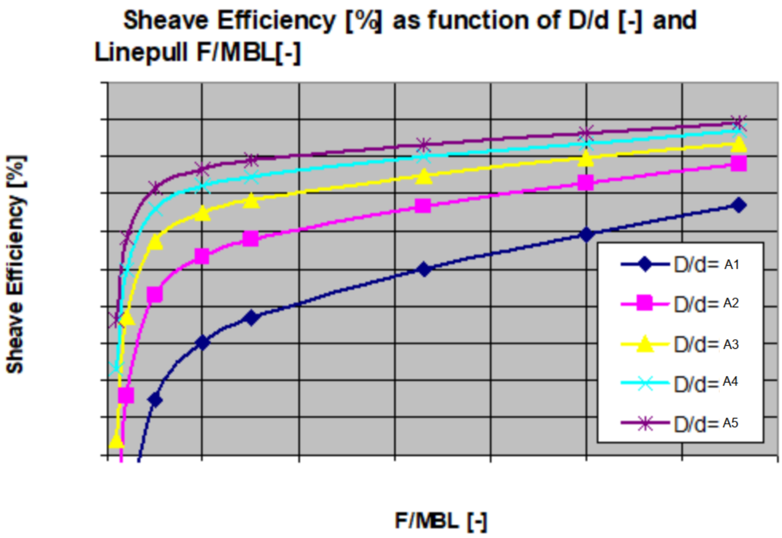


Figure 2.3: Sheave efficiency curves based on test results by DNV
[19]

Concluding from Equation 2.7 and Figure 2.3 it is apparent that using a constant value for the efficiency in the lower block calculations, is a significant simplifications of the real world. The changes in the tension in the wire rope cause each sheave to operate with a different efficiency. To illustrate this the minimum lower block weight to ensure proper lowering and tension at the drum during empty hoisting were calculated for a 1600mt Leg Encircling Crane (LEC) by using the equations shown in subsection 2.1.1. This was done with constant efficiency values ranging between 0.95 and 1, and by linearly interpolating between the efficiency values from Figure 2.3 for each sheave. Figure 2.4 shows the results. The graph shows that increasing the sheave efficiency reduces the required lower block weight, but also reduces the the tension during hoisting. The dots on the graph show the results from the calculation using the linearly interpolated efficiencies. It is apparent that using a constant efficiency value of 0.98, the current Huisman standard, gives inaccurate results.

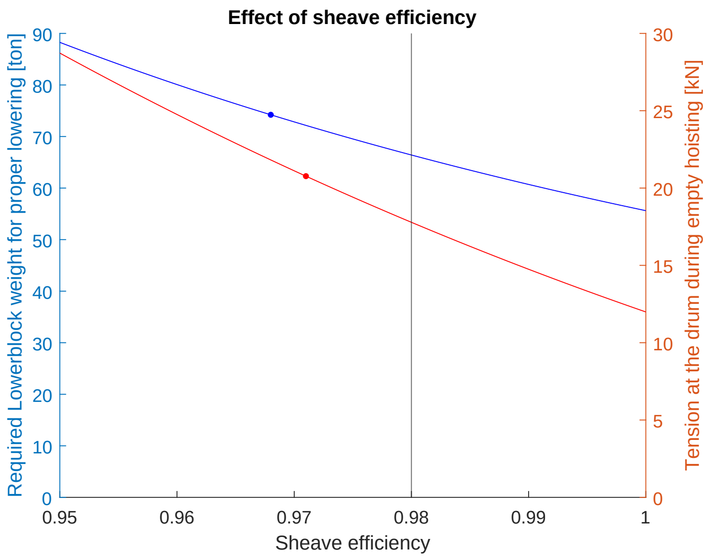


Figure 2.4: The effect of sheave efficiency on the required lower block weight and tension during empty hoisting in a 1600mt LEC.

2.1.3. Friction between wire rope and sheaves

During crane operations the friction between the wire rope and the sheaves is an important property. Mainly if some kind of traction systems is used the coefficient of friction (COF) is crucial. The design of a such a system depends on the the COF as it directly influences the frictional forces. The Huisman standard is to use a COF of 10% between wire ropes and sheaves [20]. This 10% also suggested as a minimum value by Hrabovský, Učeň, Kudrna, *et al.* [21].

In order to substantiate this 10% standard, Huisman performed multiple tests to measure the COF [20] [22] . Table 2.1 shows the results of these tests.

Table 2.1: Overview of results of COF tests performed by Huisman

Type of rope	Type of sheave	D/d	Minum COF	Maximum COF
24mm Diepa 1315CZ non-spin	480 mm 1.4418			
24mm Diepa 1315CZ non-spin	480 mm StE690		*Confidential*	
18mm QS816V(G) right handed ordinary lay	436mm GS20Mn5			
72mm Redaeli Flexpack left handed langs lay	1368mm GS20Mn5			

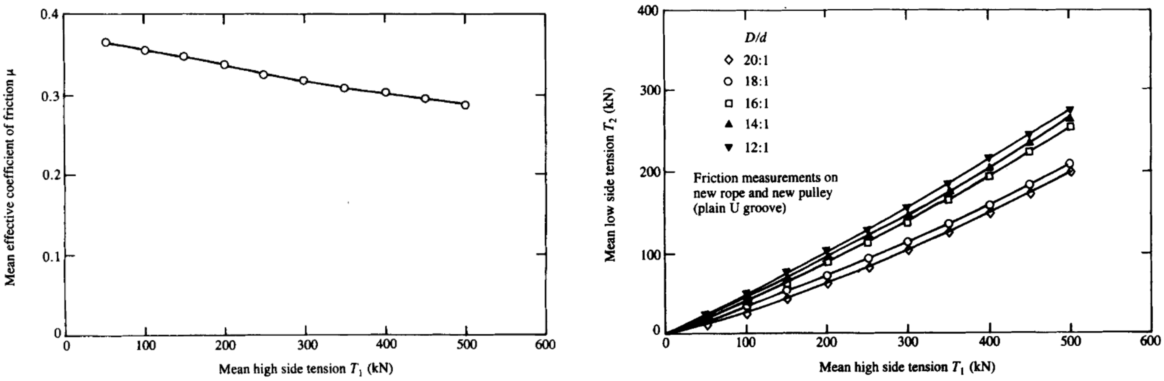
The tests performed by Huisman show that the 10% minimum is a suitable choice, but they offer little insight in what parameters influence the friction between sheaves and wire rope. Research done by Hrabovský, Učeň, Kudrna, *et al.* [21] and Nabijou and Hobbs [23] studies the effects of different parameters on the COF.

Wire tension

Both studies show a decrease in the effective COF under an increase in the tension in the wire rope. Figure 2.5a clearly shows a downwards trend of the effective COF as function of the tension.

Sheave and wire rope diameter
The diameter of the sheave used also has influence on the COF. In sheave design the D/d ratio is common way to address the size of a sheave. The D is the diameter of the sheave and d is the diameter of the wire rope. Figure 2.5b shows that an increase in the D/d ratio improves the frictional hold, meaning an improved COF. This also partly explains the high COF values found by Huisman with the 24.2 D/d test.

It was also found that wire ropes with a smaller diameter have a higher COF. Nabijou and Hobbs [23] argue this is because of the lower hardness and larger bending stiffness of larger wire ropes.



(a) Effective COF as function of high side tension

(b) Effect of sheave D/d ratio on friction

Figure 2.5: Results of friction tests performed by Nabijou and Hobbs [23]

Sheave and wire rope type
The tests performed by Huisman shows a difference in the COF for different types of wires and sheaves. This could, among other things, be caused by the material choice, how much the sheave was hardened and galvanization of the wire rope. The effect of galvanization was studied by Nabijou and Hobbs [23] and showed a clear decrease in COF of galvanized wire ropes.

Contamination
If the wire rope sheave contact gets contaminated this can have significant impact on the COF. Contamination with sand or mud can slightly improve the effective COF while water contamination slightly decreases the COF [23].

Contamination with oil can have a detrimental impact on the friction. Hrabovský, Učeň, Kudrna, *et al.* [21] show a reduction in tension from 56% to 11%. This large reduction of the COF can be detrimental if not taken into account.

Dynamic friction
The discussion regarding wire rope friction above assumes static friction. However, in reality dynamic friction may also occur. Huisman's tests investigated dynamic friction as well. The results of these experiments are presented in Table 2.2. It is often observed that the dynamic COF is lower than the static COF. However, the tests conducted by Huisman indicate that the minimum COF is higher for dynamic friction. However, the maximum COF is significantly lower, and the range between the maximum and minimum COF is quite narrow.

Nabijou and Hobbs [23] also concluded that repeated slipping of the wire rope over the sheave increases the mean COF. However, an increase in irregular slipping resulted in a wide scatter of results. The speed at which the wire rope slips on the sheave also influences the dynamic COF. Research on this topic is limited and contradictory. Shirong [24] found that the COF increased in their friction test between a PVC lining and a wire rope under increasing speed. Conversely, the study by Ma, Zhu, Xu, *et al.* [25] found a decreasing COF between the wire rope and the GM-3 lining with increasing sliding speed. At first glance this contradiction could be caused by the difference in lining material. Investigating all causes behind these contradictory results could be a study in itself and is therefore considered beyond the scope of this research.

Table 2.2: Overview of results of dynamic COF tests performed by Huisman [22]

Type of rope	Type of sheave	D/d	Minum COF	Maximum COF
24mm Diepa 1315CZ non-spin	480 mm 1.4418		*Confidential*	
24mm Diepa 1315CZ non-spin	480 mm StE690			

2.2. State of the art

The idea of increasing the tension in the wire rope of a hoisting system is not new. Increasing the tension in the wire rope can serve multiple purposes. A primary design objective in numerous patents and research efforts is to improve spooling quality. Extending this objective is the prevention of the cutting-in phenomenon. Other design goals include reducing the load on the sheaves and the drum within the system. Or allowing high tension in the wire rope while it is spooled under low tension on a drum.

In order to increase the tension in the wire rope a force needs to be exerted on it. Exerting a force on the wire rope somewhere along the wire ropes path is often done with a friction based device. In order to calculate the frictional forces between a tensioning system and the wire rope the Coulomb friction model can be used. Equation 2.8 shows the Coulomb model where F_f is the frictional force, μ the coefficient of friction and F_n the normal force. The equations shows that the maximum friction that can be created is a function of the normal force and the friction between the wire rope and the tensioning system.

$$F_f \leq \mu F_n \tag{2.8}$$

In lifting applications like cranes and elevators it is common to use sheaves to exert a frictional force on a wire rope. In order to calculate the frictional force that can be transferred to a wire rope by a sheave, an altered version of Coulombs model can be used. This equation is known as the Capstan equation or the Euler–Eytelwein formula. This equation relates the tension in the wire rope on both sides of the sheave to the friction and wrapping angle. The Capstan equation is shown in Equation 2.9.

$$T_1 = T_2 e^{\mu\varphi} \tag{2.9}$$

Where

- $\boldsymbol{T_1}$ is the high tension side of the wire rope.
- $\boldsymbol{T_2}$ is the low tension side of the wire rope.
- $\boldsymbol{\mu}$ is the coefficient of friction between the wire rope and the sheave.
- φ is the wrapping angle.

Following from the equations discussed above, a wire rope tensioning device could increase its potential in multiple ways. Therefore the research into the current state of art is divided into the three categories shown below.

- Increasing the normal force
- Increasing the coefficient of friction
- Increasing the wrap angle

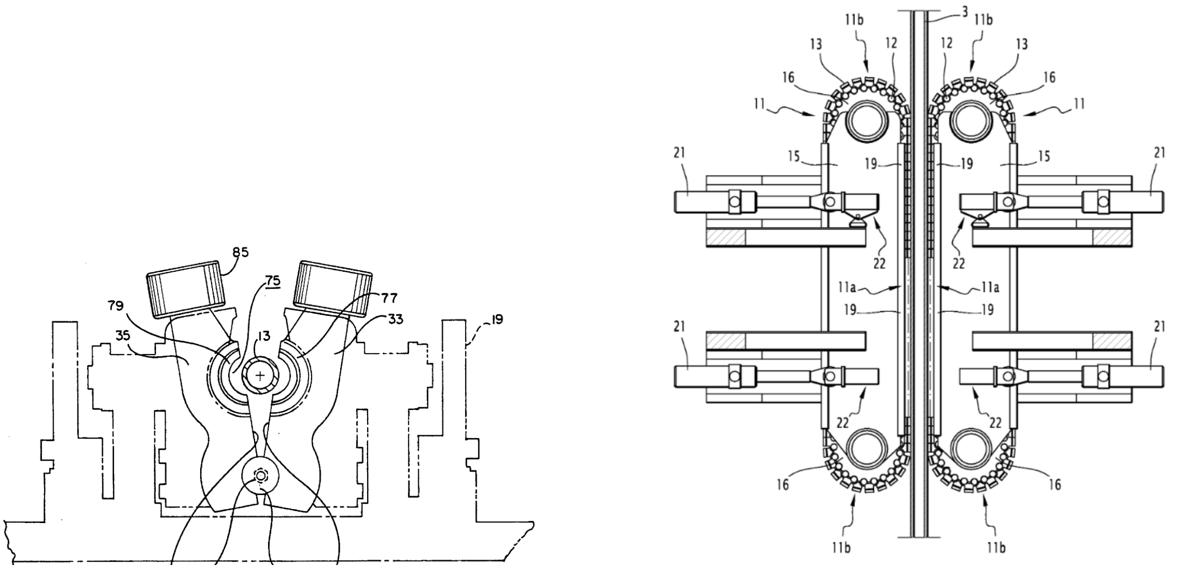
2.2.1. Increasing the normal force

Increasing the normal force has a large impact on the frictional capabilities of a tensioning device. This is apparent from Equation 2.8, but when looking at the derivation of the Capstan equation the normal force also has a significant influence on the friction. This subsection discusses current techniques to increase the normal force in wire rope tensioner systems.

Clamping the wire rope with tracks

An approach to a wire rope tensioning device is the use of clamping tracks. Clamping tracks are straight pieces of track which clamp a straight part of the wire rope. Each track consists of a chain of links which can be driven by multiple types of drives. Each chain link has a pad on it which contacts the wire rope. The normal force on the wire rope is increased by pressing on the wire rope from multiple sides. The magnitude of the normal force is determined by the clamping force of the clamping mechanism. The length of the track can be adjusted to reach a desired pressure on the wire rope. Clamping tracks can have different configurations.

A configuration with a single track with opening and closing chain links is shown in Figure 2.6a. In this configuration only 1 track is used. The links can open and close to clamp the wire rope along its length, the following patents make use of this principle: [26][27][28][29]. The advantages of a single clamping track is the relative compactness of the system. The clamping action of the tracks can also be achieved with the usages of multiple tracks as shown in Figure 2.6b. This approach is very common in controlling the tension in pipe-laying operations on ships [30][31][32]. The same concept can also be applied to wire ropes. Having multiple tracks allows the system to adapt to different diameters.



(a) Schematic of a single tensioner track [26]

(b) Schematic of a double tensioner track [30]

Figure 2.6: Schematic of two clamping track configurations

There are also innovation in the pads which contact the wire rope. This can be seen in the development of adaptive and adaptable tracks. Adaptive pads automatically change their shape when they get into contact with a wire rope, or pipe, of a different diameter [33][34][35]. If the shape of the pad and the wire rope match the normal force gets distributed across the wire ropes surfaces more evenly, resulting in lower pressure. A decrease in peak pressure values is beneficial to the wire rope life. The adaptable pad patented by Mackinnon [36] reaches the same goal, but it does not automatically change shape. Adapting the pad shape needs to done by hand or with a different external device.

In addition to the development of adaptive and adaptable pads capable of altering their shape, advancements have been made in the ability of chain links to modify their shape. Specifically, these chain links can undergo longitudinal elongation when the shear stress between the wire rope and pad exceeds a certain threshold [28][37]. Due to the increased tension along the length of the clamping track, strain is induced in the wire rope. Chain links that can elongate longitudinally are able to adapt to this increased tension, thereby preventing slippage caused by the strain in the wire rope. Preventing slippage reduces wire rope damage and ensures static friction.

There are countless more patents exploring various aspects of clamping tracks. These aspects include the type of chain support, the mechanisms for moving the tracks, the type of drive, and the number of tracks utilised. While these details are highly relevant for the intricate design of clamping tracks, they are of lesser importance for the scope of this study.

Pressing the wire rope on a sheave
Increasing the normal force acting on a wire rope running on a sheave can be achieved by clamping it to the sheave. The simplest approach to this is to apply a load on the wire rope with a number of passive rollers. A schematic drawing of this is shown in Figure 2.7a. Each passive roller locally increases the normal force on the wire rope. When multiple passive rollers are used around the sheaves circumference the Capstan equation can be extend to the following equation.

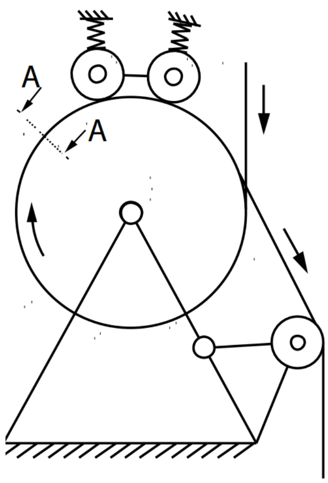
$$T_1 = T_2 e^{\mu\varphi} + \frac{NP}{\varphi}(e^{\mu\varphi} - 1) \tag{2.10}$$

In Equation 2.10, N is the amount of passive rollers and P the normal force added by each roller. It can be seen that the tension on the high tension side is now partly independent of the tension on the low tension side.

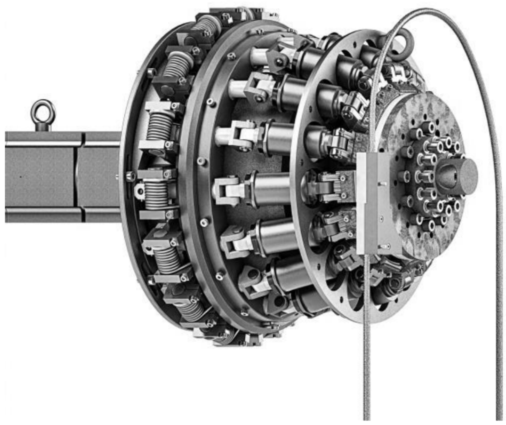
An advanced version of this approach was designed by Schumann, Schmidt, and Leonhardt [38][39]. They designed and prototyped a system that clamps the wire rope to the sheave with clamping jaws. These clamping jaws rotate with the sheave. Because the jaws rotate with the sheave an extra frictional component is added as can be seen in Equation 2.11. ξ is the angle under which the jaw presses on the wire rope and μ_c is the friction between the jaws and the wire rope. This extra frictional components increases the traction capabilities compared to the passive roller approach. A similar device is patented by Peng and Zhou [40].

$$T_1 = T_2 e^{\mu\varphi} + \frac{NPcos(\xi)}{\varphi}(e^{\mu\varphi} - 1) + NP\mu_c + NP\mu sin(\xi) \tag{2.11}$$

Figure 2.7b shows the system. The movement of the jaws is facilitated by a combination of a cam disk and spring-loaded arms. The cam disk engages the jaws to close when the wire rope contacts the sheave and releases the claws at the end of the wrap angle.



(a) Passive rollers [38]



(b) Image of the rotating claw system [38]

Figure 2.7: Illustration of two forms of pressing the wire rope on a sheave

Belt wrap capstan devices are a category of traction devices in which the wire rope is pressed onto the sheave by a belt[41][42][43]. The tension in the belt adds an evenly distributed pressure along the wire ropes surface to increase the normal force. The belt can either be driven or passive. In case of a passive belt the only function of the belt is to increase the pressure on the wire rope. If the belt is driven the power transfer capabilities of the system increase because of the friction between the wire rope and the belt, similar to the clamping jaws concept. These belt wrap capstan devices are often proposed to be used in combination with fibre ropes, instead of wire ropes. Similar to the belt wrap capstan is the track wrapped capstan patented by Schumann, Schmidt, and Leonhardt [44]. The working principle is the same as a belt wrap capstan, but the belt is replaced by a chain with small links and pads. The advantage of this is that the pads on the links are stiff with a complementary form with respect to the wire rope. Whereas a belt is flat and flexible.

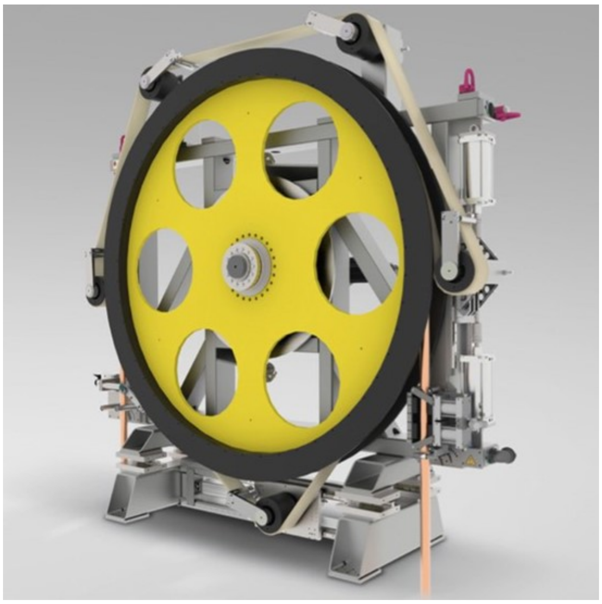


Figure 2.8: Illustration of a belt wrap capstan [45]

Pushing sheave halves together

An alternative method to increase the normal force involves utilizing a sheave composed of two separate halves. These halves can be pressed together, forming a sheave with an undercut groove or V shape (see subsection 2.2.2), thereby clamping the wire rope in between[46][47][48][49]. The two halves are pressed together using springs or compliant clamping devices. To facilitate the entry and exit of the wire rope, rollers are positioned between the sheave halves at the sections where there is no contact with the wire rope. Because these rollers are wider than the cable, the sheave halves are not perfectly collinear. This causes the disadvantages of an uneven distribution of the clamping force around the wrapping angle and limiting the the angle under which the clamping forces are exerted on the wire rope. The concept is illustrated in Figure 2.9. This technique can achieve high clamping forces and make the traction capabilities of the traction devices independent of the tension on the low tension side. Equation 2.12, where N is the force pressing the sheave halves together, shows this.

$$T_1 = T_2 e^{\mu\varphi} + N\mu \tag{2.12}$$

Magnetic sheave

Magnets can also be used to increase the normal force between the wire rope and the sheave. In this traction sheave magnets are placed insides the sheave that attracts the wire rope. The magnetic force adds an element to the capstan equation that is independent of the tension in the wire rope. The altered capstan equation is shown in Equation 2.13. Where q is the linear magnetic load and D is the sheave diameter. As shown this form of traction sheave can also create tension independent of the low tension sides tension. This is however strongly dependant on the strength of the magnets. In their study Schmidt, Leonhardt, and Anders [50] showed that for tensions upwards of 2 kN the effect of the

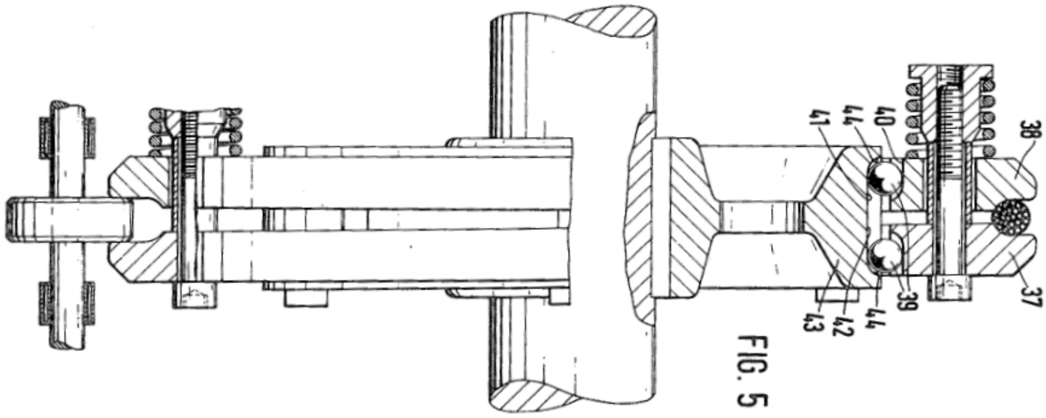


Figure 2.9: Illustration of the sheave consisting of two halves [46]

magnets is almost negligible. They do not mention the exact magnets used. So determining if there is more potential to this concepts is difficult.

$$T_1 = T_2 e^{\mu\varphi} + \frac{qD}{2}(e^{\mu\varphi} - 1) \tag{2.13}$$

2.2.2. Increasing the coefficient of friction

Increasing the coefficient of friction (COF) between the wire rope and the tensioning system would be very effective in increasing the potential of such a system. The COF is a material property which is hard to manipulate. There are options to increase or optimize the COF, these are explained in this subsection.

Generally a few small things can be done to optimize friction. As subsection 2.1.3 shows this is a large D/d ratio, sheave and wire rope combinations that create high COF and avoiding contamination with water and oil. Eliminating oil and water contamination is difficult. As wire ropes are lubricated and sometimes operate in wet conditions. Techniques to improve the COF are discussed below.

Sheave material

As mentioned in subsection 2.1.3 the type of sheave can have influence on the COF. This has let to research into optimal sheave materials and coatings. Krešák, Peterka, Ambriško, *et al.* [51] studied the COF of different types of rubber sheave lining. These rubber types had different chemical properties. These rubber linings were compared against a conventional sheave lining. The results show that all rubber types have significantly better frictional performance than the standard lining(K25), 0.6 for the best rubber lining compared to 0.27 for K25. But the COF of the rubber types were significantly impacted by the pressure on the lining, thus by the tension in the wire rope. The standard lining was not negatively impacted by the increase in pressure, its performance even increased slightly. Important to note is that these frictions materials are tested and used under low contact pressure, below 3 MPa. This is low compared to contact pressures encountered in cranes, up to 90 MPa.

The use of a PVC lining could improve the coefficient of friction to 0.2 according to Shirong [52]. He finds that this value is dependant on the speed and tension in the wire rope. Increasing with higher speed and higher tension. The wear of the PVC lining is not discussed. A different study confirms this value for the COF between a PVC lining and wire rope [53]. This study simulates shortcomings of common friction tests by simulating the conditions a hoisting device is in. These PVC linings are also only tested and used up to 3 MPa.

Applying a coating to the sheave could also prove effective. Hoffman and Schmidt [4] discusses the use of a tungsten carbide coating which is said to have a COF of 0.43 with the used wire rope. This is 4 times larger than a COF of 0.1, which is the current standard. There is however very little study into the use of such a coating for the application of increasing the friction with wire ropes. A study by Cai, Wang, Xu, *et al.* [54] does find similar values for the COF. But these tests measured the COF between a flat piece of iron coated with tungsten carbide and a metal ball. Whether the COF between a wire rope and a sheave coated with tungsten carbide would be similar is not known.

Groove shape

In addition to altering the actual coefficient of friction (COF) through the use of different material combinations, another technique to enhance the apparent or effective COF involves modifying the groove shape in the sheave. In Figure 2.10, the groove shape on the left is identified as the standard groove shape, also referred to as a semi-circular or plain groove. This groove shape mirrors the circular shape of the wire rope. In the field of elevator systems these grooves are found in older systems. To increase the traction capabilities of the elevators a shift is seen towards two different groove shapes[55].

The groove shape in the middle is referred to as the undercut U-groove or undercut circular groove, while the groove on the right is termed the V-groove. Implementing these groove shapes significantly enhances the pressure exerted on the sheave, thereby increasing the apparent coefficient of friction (COF). This augmentation in the apparent COF is highly advantageous for the design of traction devices. These groove designs have been and continue to be employed in the development of traction devices, particularly in traction winches for elevators [56]. In their study, Hrabovský, Učeň, Kudrna, *et al.* [21] demonstrated a substantial improvement in the COF for the V-groove compared to the standard groove. This improvement is most pronounced in scenarios where the system is contaminated with oil, showing an increase in COF from 0.11 to 0.36.

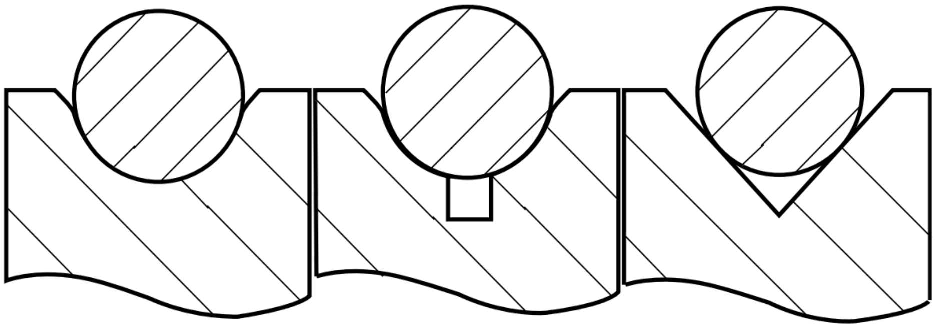


Figure 2.10: Illustration of three common groove shapes [38]

These groove configurations present a notable drawback. The increased pressure and specific shape of the groove can significantly accelerate the wear of both the sheave and the wire rope [38] [56] [57]. Feyrer [10] states that depending of the design of the undercut U-groove or V-groove the lifetime reduction can be upwards of 90%. In their research, Nabijou and Hobbs [23] observed the following during tests with the Undercut U-groove (UCU):

"The slipping of the rope over the UCU groove resulted in shredding of the metal from the sharp edges of the undercut, even at relatively low loads."

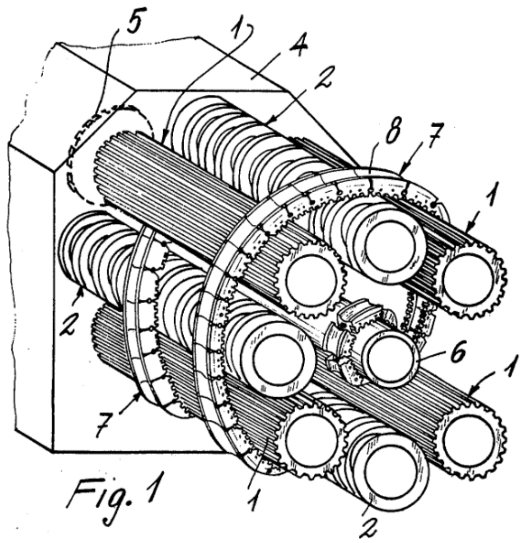
2.2.3. Increasing the wrap angle

Traction devices making use of sheaves can benefit of increasing the wrap angle. This can improve the capabilities of the system as seen in the Capstan equation. In most common traction sheaves the wire rope is wrapped around the wire rope up to 270°. A higher wrapping angle on a singular sheave needs a method to axially displace the wire rope, otherwise the different wraps start rubbing against each other.

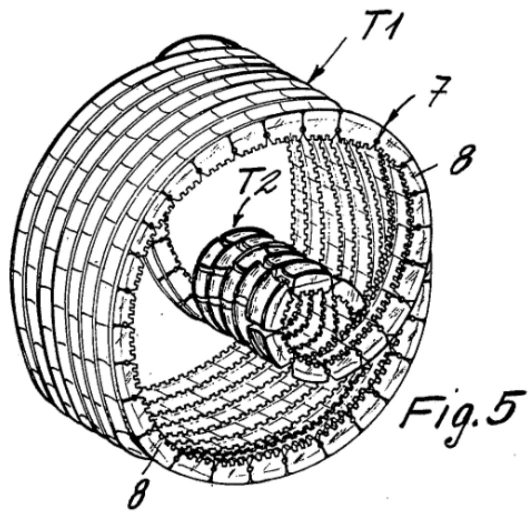
Multiple wraps on a singular sheave

Increasing the wrap angle beyond 270° on a single sheave can be achieved in numerous ways. An approach employed in many patents is the use the use of chain links with pads[58][59][60][61]. This principle will from now on be referred to as the tracked sheave. The tracked sheave allows for multiple rotations on the same drum because of the helical shape of the track. The chain links have pads on them, just like in the clamping tracks. These pads are formed to complement the form of the wire rope. This evenly distributes the pressure across the wire ropes surface.

The similarities between tracked sheaves and clamping tracks extend beyond the utilisation of chain links with pads. In tracked sheaves there are also patents that allow for longitudinal elongation to accommodate the strain in the wire rope [62]. Also like the clamping tracks there are a number of different approaches to the design of tracked sheaves. These differences are primarily observed in the drive system, the guidance of the chain links, and the spacing of the wraps.



(a) Schematic of drive system



(b) Schematic of chain links with pad

Figure 2.11: Schematic of an approach to the tracked sheave principle [58]

A different approach to create a drum over which the wire advances in the axial direction is the use of 2 interlocking drums which are slightly offset with respect to each other [63][64][65]. These interlocking drum consist of multiple 'fingers'. Because of the offset the wire rope contacts only one drum the first half of the circumference and only the second drum at the other part of the circumference. This causes the wire rope to laterally displace. This prevents the wire rope wraps coming into contact, improving the life time of the wire rope. This system has the benefit of having significantly less moving parts compared to the tracked sheave. A downside of this system is the shape of the drum. The two interlocking drums form a flat surface, a cylinder. Wrapping a wire rope around a cylinder flattens the wire rope because it is not properly supported. This flattening causes damage to the wire rope. These kinds of systems are therefore often intended to be used with fibre ropes instead of wire ropes.
Figure 2.12a shows a picture of a prototype of the system build by the UK based company Parkburn. A schematic drawing of the path of the wire rope across such a system is shown in Figure 2.12b,

(a) System overview [64]

(b) Path of the wire rope on the sheave [63]

Figure 2.12: Schematics of the offset drum system

Soil Machine Dynamics Ltd., a UK based company has developed a product aiming to solve the problem of wire rope wraps touching when there are multiple windings on the same drum. The so called self-fleeting cable drum engine is a large drum around which a cable can be wound multiple times [66]. The drum is made up out of fingers across its entire surface. These fingers can translate in the axial direction. By axially translating the fingers during rotation the wire rope wraps will be distanced from each other. Figure 2.13 shows the lateral displacing fingers and the distance between the wire rope wraps. In this system the same problems with flattening the wire rope can be expected. This system is therefore also intended to be used with fibre ropes.

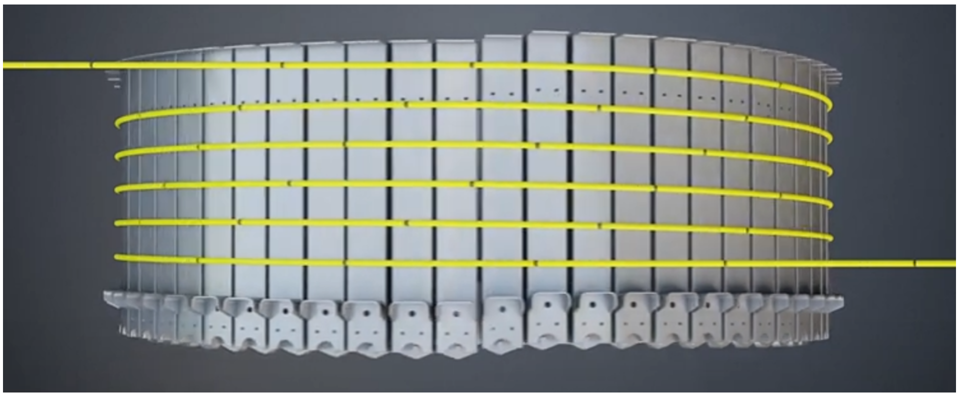


Figure 2.13: Lateral displacement of the wire rope on the self-fleeting cable drum engine [67]

Multiple wraps on two drums

Simplifying the concepts above lead to the design of a mechanism commonly known as traction winches or double-head capstan winches, shown in Figure 2.14. In these devices the wire rope is wrapped around two drums multiple times. This can create large tension in the wire rope because of the large total wrap angle. Traction winches are already commonly employed in cranes, most often in deep sea hoisting operations where the tension can variate a lot.

Wrapping the wire rope around two drums multiple times can have several negative consequences. The most evident is the increase in bending cycles of the wire rope, which leads to a significant reduction in its lifespan [7]. Additionally, the tension difference between the wire rope segments results in uneven strain on the wire rope. Since the drums rotate at a constant speed, this causes the wire rope to slip on the drum, potentially damaging the wire rope. This effect is amplified by the uneven wear of the drums due to the tension difference [68], which further increases the slippage of the wire rope.

Figure 2.14: Photo of a traction winch currently installed in a pipelay system [69]

2.3. Literature gap

The current state of the art in wire rope handling systems, as described in section 2.2, primarily focuses on mechanisms for controlling and increasing the tension in the wire rope. These systems, including various patented devices, are designed to ensure the effective control of wire rope tension, which is crucial for improving the winding quality to increase the wire rope life and prevent cutting in.

A wide variety of concepts has been proposed to enhance wire rope tension. Among these, clamping tracks and tracked sheave designs have been extensively explored, resulting in numerous patents over the years. Recent research and developments seems to focus on sheave based traction devices. Notable developments in this area include the traction device with interlocking drums, the self-fleeting cable drum, and the sheave equipped with clamping jaws. These sheave based devices are often designed to increase and control the tension at the drum.

Research and innovation in the area of frictional coefficients are relatively limited, which is surprising given the potential benefits of improving COF. Traditional traction devices, such as those used in elevators, enhance COF through groove shapes. However, recent innovations are moving away from this approach to extend the lifespan of wire ropes [38]. These newer traction devices employ coatings to achieve high traction capabilities. Unfortunately, research into the effectiveness of these coatings is sparse, making it difficult to assess their performance under varying conditions. A thorough analysis of these coatings is needed to asses the feasibility of applying them in the real world.

A notable gap in the current literature and patents is the application of these systems to reduce the required weight of the lower block. While the concept of increasing wire rope tension has been described and studied in many patents and researches, there has been no significant focus on decreasing the lower block's weight. Although this reduction might not drastically impact a crane's capabilities, it is a small optimization step that leads towards achieving the best possible crane design. In addition to the insufficient focus on reducing the lower block weight, there is a notable lack of research on cutting-in prevention. While increasing the tension in the wire rope to ensure proper winding is frequently explored, studies specifically addressing cutting-in prevention are lacking.

This study aims to close this gap by designing an active wire rope tensioner focussed on preventing the cutting-in problem and allowing the usage of a lighter lower block.

3

Conceptual design

The previous chapter covered a literature and patent research into existing devices and principles which can be used to design the active wire rope tensioner. This chapter covers the concept phase of the research. Section 3.1 shows the morphological chart made to create an overview of all working principles that could be used. The generated concepts are discussed in section 3.3, after which they are compared using the KPIs in section 3.4. The chapter concludes with section 3.5 discussing the decision on the most promising concept.

3.1. Morphological chart

The utilisation of a systematic approach for the concept design process is crucial for the exploration of all potential solutions. A morphological chart was selected to facilitate this process. The morphological chart, as illustrated in Figure 3.1, presents possible solutions to achieve the system functions. These solutions are derived from the research detailed in 2.2 or generated through brainstorming sessions.

The three main functions of the system are shown in the left column, with in the rows next to them the possible solutions at fulfilling that function. Three subfunctions can also be found in the first column. These subfunctions are important aspects for optimizing friction based tensioning devices. Because of their importance and the extensive variety of solutions available, these subfunctions were added to the morphological chart.

In the morphological chart methodology, theoretically, every possible solution can be combined with any other to create a viable concept. However, in practice, certain combinations may counteract each other or may not be practical to implement. Such combinations are therefore excluded.

Using this morphological chart, seven concepts were generated. The combinations utilised in these concepts are depicted in Figure 3.1. The seven concepts are numbered and have a name for referencing. They are enumerated below:

- Concept 1 - Split tackle
- Concept 2 - Driven sheaves in upper block
- Concept 3 - Double drum
- Concept 4 - Clamping track
- Concept 5 - Sheave with roller chain
- Concept 6 - Tracked sheave
- Concept 7 - Sheave with clamping jaws

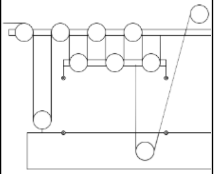
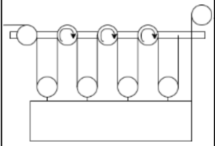
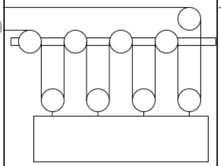
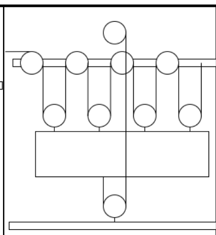
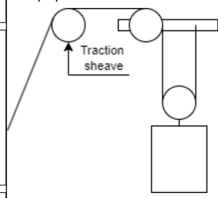
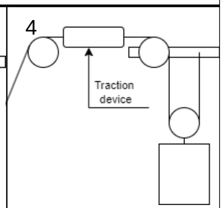
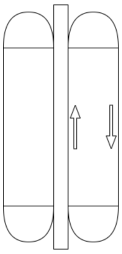
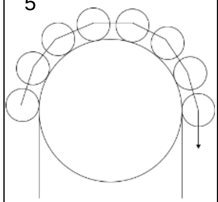
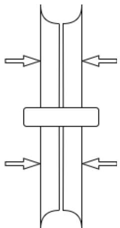
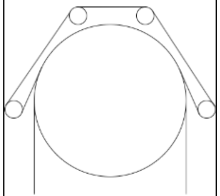
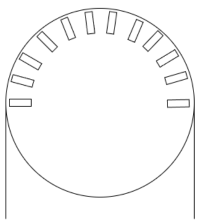
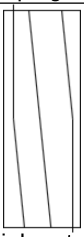
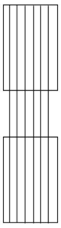
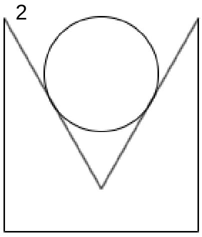
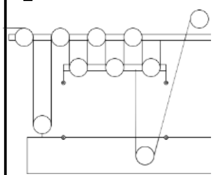
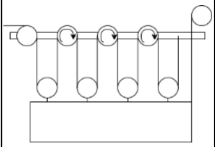
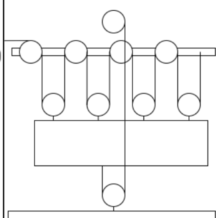
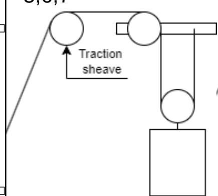
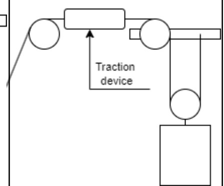
Function		Possible solutions					
Increase tension during hoisting		1  Split tackle attached to deck	2  Driven/braked sheaves in UB	3  Double drum	 Wire between LB and deck	5,6,7  Driven/braked traction sheave	4  Driven/braked linear traction device
Subfunctions friction-based devices	Increase normal force	4  Clamping tracks	5  Rollers on sheave	7  Clamping jaws on sheave	 Clamping sheave halves	 (Driven) belt on sheave	 Magnetic sheave
	Increase wrap angle	6  Multiple rotations on 1 drum	 Multiple rotations on 2 drums				
	Increase COF	4,7 Optimal lining or pad material	Coating	2  Groove shape			
Decrease required lower block weight		1  Split tackle	2  Driving sheaves in UP	 Wire between LB and deck	5,6,7  Driven traction sheave	4  Driven linear traction device	
Adapt to allow high speed operation		4 Remove system from rigging	2 Disconnect drive	5,6,7 Let drive freewheel	1,3 Select drive with sufficient speed		

Figure 3.1: Morphological chart

3.2. Parametric model

The two primary functions of the active wire rope tensioner are to manipulate the wire rope tension during hoisting and to reduce the required lower block weight. Consequently, it is imperative to evaluate a concept's capabilities in fulfilling these functions. To facilitate this evaluation, a parametric model was developed to determine the tension in the wire rope throughout the rigging. This model is based on the equations discussed in subsection 2.1.1. To enhance the model's accuracy, the efficiency of each sheave is determined by linearly interpolating between the efficiency values presented in Figure 2.3. This approach yields more precise tension calculations compared to the utilisation of a constant efficiency, which is the current standard.

The base model, which simulates a crane without an active wire rope tensioner, produces four outcomes. Firstly, it calculates the tension at the drum during lowering, based on the specified crane properties. Secondly, it determines the required lower block weight to achieve a set target tension at the drum during lowering. Additionally, it provides both of these outputs in the case of hoisting. This base model serves as the foundation for developing concept-specific models that integrate the active wire rope tensioner. For concepts 4 to 7 the location of the tensioning device is relevant to its performance. In these models the tensioning devices are located between the first two sheaves after the falls. This is the part of wire rope between F_6 & F_7 in Figure 2.1. Choosing the optimal location for the system is considered in the detailed design phase.

Given the necessity for the active wire rope tensioner to be compatible with various types of cranes and rigging configurations, the model is parametric. This is achieved by the use of a separate file which contains all properties of a crane, thereby enabling straightforward assessment of a concept's applicability across different crane configurations.

3.3. Concept design

This section describes the seven concepts developed with the help of the morphological chart. These seven concepts do not cover all possible approaches to an active wire rope tensioner system. Using all possible solutions from the morphological chart would require a large number of concepts, some of which are expected to be unfeasible or unsatisfactory upfront. They do, however, cover a wide variety of the possible solutions to an active wire rope tensioner.

Each subsection highlights a concept. The working principles to achieve the functions of increasing tension during hoisting, reducing the lower block weight, and adapting to high-speed operations are described. Also a SWOT analysis is applied to each concept to clearly set apart the concept's characteristics. A SWOT analysis is a technique that identifies the strengths, weaknesses, opportunities, and threats [70]. The strengths are characteristics of a concept that provide an advantage over other concepts. Weaknesses, conversely, highlight the disadvantages. Opportunities describe potential secondary benefits of a concept. Threats identify possible problems or limitations that may arise during further development of the concept.

3.3.1. Concept 1 - Split tackle

The first concept utilises the split tackle approach to increase tension during hoisting and reduce the required lower block weight. Split tackle is a technique already employed in cranes for a different purpose. Hoists with split tackle can disconnect several sheaves from the lower block, resulting in fewer falls between the lower and upper block. This leads to higher operating speeds but lower hoisting capacity. This principle can also be applied to increase tension during hoisting and decrease the required lower block weight.

If a significant number of falls can be split off, the existing lower block weight will induce tensions in the wire rope exceeding *Confidential*% MBL at the drum. Additionally, splitting falls reduces the required lower block weight during lowering. The procedures associated with this concept are as follows:

To increase the tension during hoisting, the lower block must be lowered onto the deck and securely strapped down. The procedure begins by splitting off the sheaves. The drum can then commence winding, causing the split tackle sheaves to move upwards. During this movement, the external drum, as shown in Figure 3.2, applies back tension to ensure *Confidential*% MBL at the drum. When the split sheaves reach the block catcher at the top, the lower block can be disconnected from the deck and will start moving upwards. During this hoisting, the external wire should be maintained at a constant low tension to prevent slack. The tension in this second phase of hoisting is determined by the ratio between the number of falls still in use and the lower block weight.

The third step in the procedure is to reconnect the split sheaves. This can be achieved by moving the lower block all the way to the top, where the split sheaves are located. Alternatively, if the hook is connected to a heavy object, the external wire could be used to pull the split sheaves back down to the lower block. This requires robust control to ensure the lower block itself does not lower during this step.

Lowering a lower block with low weight uses the same procedure, but in reverse. The amount of weight that can be reduced in the lower block is proportional to the ratio of splitted falls and falls still connected to the lower block.

In order for this concept to work at high operating speeds, the drive of the external drum should be dimensioned correctly. This drum should be able to reel the external wire in and out with the force and speed required to follow the speed of the splitted falls.

The Strengths, Weaknesses, Opportunities and Threats analysis for this concept is displayed in Table 3.1.

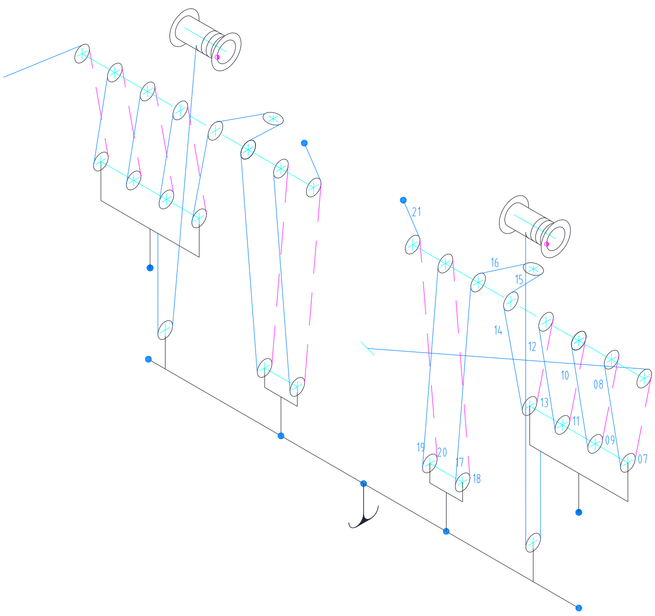


Figure 3.2: Schematic of concept 1 - Split tackle

Table 3.1: SWOT analysis for concept 1

Strengths • Split tackle known solution • No complicated device needed	**Weaknesses** • Redesign of lower block needed • Complicated and time consuming procedure • First layer on drum cannot be tensioned
Opportunities • Allows very fast movement of lower block	**Threats** • Tension limited by lower block weight • Strapping the lower block to deck needs to be possible

3.3.2. Concept 2 - Driven sheaves in upper block

This concept works by driving the existing sheaves in the upper block. This will increase the tension in the wire rope after each driven sheave as illustrated in Figure 3.3. Because there are a significant amount of sheaves in the upper block this can build significant tension. This tension build up is limited by the weight of the lower block.

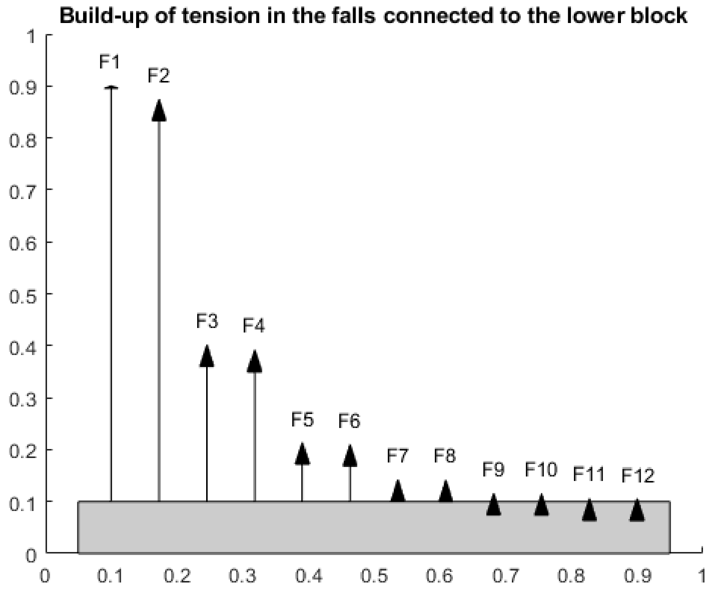


Figure 3.3: Illustration of tension build-up in concept 2

The biggest advantage of this system is that it would not require the addition of extra sheaves in the rigging. Since each driven sheave would need to be operated independently multiple smaller drives can be used, instead of a single large drive, which could be advantageous. By braking the sheaves during hoisting the tension at the drum can be increased. During lowering the sheaves can be driven to allow a lighter lower block.

To enhance the traction capability of each sheave, V-groove sheaves were selected. These significantly increase the COF, thereby significantly improving traction capability, as demonstrated in subsection 2.2.2. Each sheave drive should be dimensioned to operate with the required torque and speed. When increased tension is not necessary, the drives can be disconnected from the sheaves, thereby negating their impact on performance.

The Strengths, Weaknesses, Opportunities and Threats analysis for this concept is displayed in Table 3.2.

Table 3.2: SWOT analysis for concept 2

Strengths • No extra sheaves needed • Multiple small drives	**Weaknesses** • *Confidential*% MBL of tension cannot be reached • Lower block will tilt because of uneven forces in falls
Opportunities • Increased total rigging efficiency	**Threats** • Tension generation limited by lower block weight • V-groove damages wire rope significantly

3.3.3. Concept 3 - Double drum

The third concept is the double drum concept. This concept also uses a technique that is already being used in cranes, similar to concept 1. This concept uses 2 drums per tackle with a wire rope that runs trough the entire rigging from drum 1 to drum 2, as can bee seen in Figure 3.4. The usages of two drum per tackle is currently being used in cranes to double to speed of hoisting. But this could also be used to wind the drum with high tension.

To achieve high tension during winding of the drum, the following procedure can be used: Start by emptying drum 1 onto drum 2. This results in one empty drum and one full drum under low tension. After this the wire rope can be transferred from drum 2 onto drum 1. By braking drum 2 the desired tension of *Confidential*% MBL can be achieved at drum 1. During this process the lower block needs to be either all the way at the top resting in the block catcher, or it needs to be secured to an object heavier than 15% of the SWL. As long as the lower block is secured properly any tension value can be reached.

This concept does not permit the lower block to become lighter, as it lacks a mechanism to reduce the required lower block weight. Since this concept employs two full-sized drums and drives, it can operate at all crane speeds. It could even increase the speed by hoisting with both drums simultaneously. However, this would not enhance the tension during winding.

The Strengths, Weaknesses, Opportunities and Threats analysis for this concept is displayed in Table 3.3.

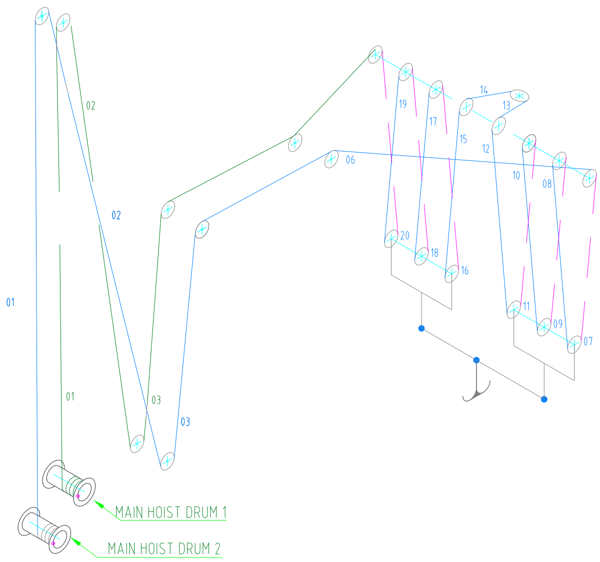


Figure 3.4: Schematic of concept 3 - Double drum

Table 3.3: SWOT analysis for concept 3

Strengths • Double drum cranes already exist • No tensioning device needed	**Weaknesses** • Extra procedure required • Cannot reduce required lower block weight
Opportunities • Spooling can happen during transport • Can also be used to increase operational speed	**Threats** • Extra bending cycles, not ideal for wire rope fatigue

3.3.4. Concept 4 - Clamping track

Concept 4 is the first of the friction-based tensioner concepts. It is inspired by the single clamping track configurations discussed in section 2.2. This linear tensioner device can be positioned anywhere along a straight section of the crane's rigging, providing great flexibility in its application. The decision was made to use a single clamping track with links that can open and close to grip the wire, instead of a double track system commonly applied in pipe laying systems. A single track configuration significantly reduces the system's footprint while also minimising the number of moving parts required.

Tension generation in this concept is achieved by clamping the wire rope with the track links. Figure 3.5 shows a schematic of this, the numbers in the schematic are used to explain the workings. Clamping is accomplished by pressing the links(1) together using rollers(4), rolling on two CAM guides(5). By regulating the position and forces on the CAM guides, the normal force on the wire rope(3) can be controlled. To enhance the traction capabilities of the tensioner, a lining can be used in the links to improve the COF.

During hoisting, the tracks will be braked to increase tension on the drum side. To allow a light lower block to be lowered, the track can be driven, thereby reducing or eliminating the need to overcome the wire rope's own weight. If the system is not required, the CAM guides can be moved, which will open all links, effectively removing the system from the rigging entirely.

The Strengths, Weaknesses, Opportunities and Threats analysis for this concept is displayed in Table 3.4.

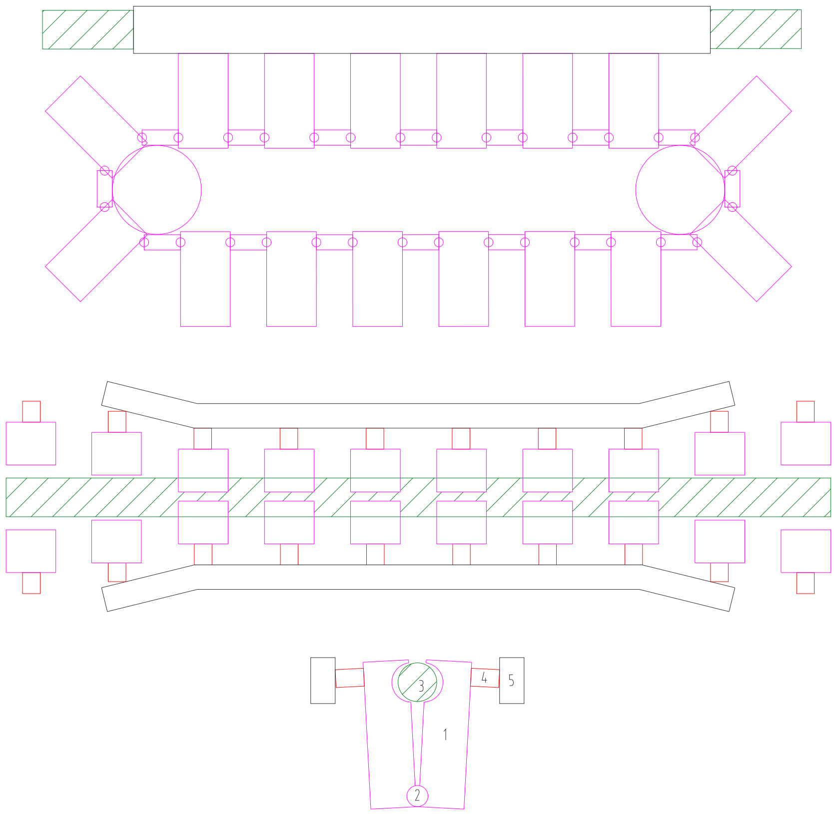


Figure 3.5: Schematic of concept 4 - Clamping track

Table 3.4: SWOT analysis for concept 4

Strengths	**Weaknesses**
• Completely removable from rigging • Very high traction capability • No required lower block weight	• Complex device • Large device for the clamping length
Opportunities	**Threats**
• No extra bending, optimal for wire rope fatigue • Applied force evenly distributed, low peak pressure • Easy to scale	• Reliability • Sensitive to slack rope • Possible wire rope damage during slippage

3.3.5. Concept 5 - Sheave with roller chain

The choice for a sheave based traction device was evident after the numerous working principles uncovered in section 2.2. This concept makes use of a driven sheave that uses rollers to increase the normal force. These rollers are connected to each other, forming a chain of rollers. This roller chain is wrapped around the wire rope and sheave. If the roller chain is tensioned the rollers will exert a force on the wire rope, pressing it on the sheave and increasing the traction capability of the driven sheave. A schematic drawing of the concept is shown in Figure 3.6.

This concept increases tension during hoisting by braking the sheave. Driving the sheave during lowering reduces the required lower block weight. If the system is not required, the tension can be removed from the roller chain reducing the pressure on the wire rope. The drive of the sheave can then either freewheel or be controlled to rotate with the wire rope without increasing tension in the wire.

The Strengths, Weaknesses, Opportunities and Threats analysis for this concept is displayed in Table 3.5.

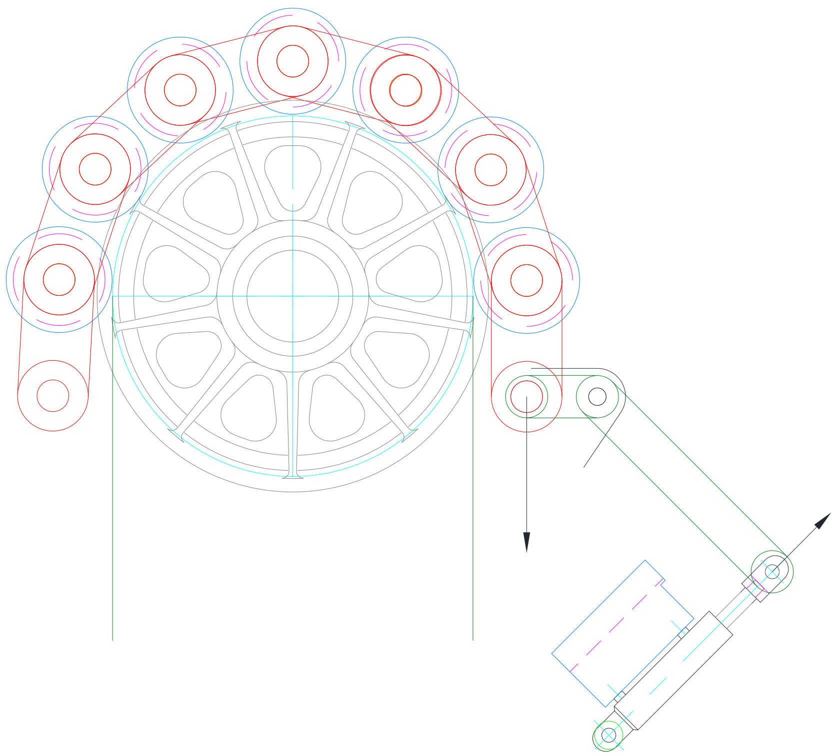


Figure 3.6: Schematic of concept 5 - Sheave with roller chain

Table 3.5: SWOT analysis for concept 5

Strengths • Simple device • No required lower block weight	**Weaknesses** • Tension generation limited, 2 in series devices needed
Opportunities • If slippage occurs, rollers do not damage wire rope	**Threats** • High peak stress at roller contact

3.3.6. Concept 6 - Tracked sheave

A sheave-based tensioning device with substantial traction capabilities can also be achieved without increasing the normal force. This is accomplished by increasing the wrap angle around the sheave. In this concept, this is realised through the tracked sheave principle. This concept employs a track with pads shaped complementary to the wire rope. The tracked sheave allows for multiple rotations on the same drum by axially displacing the wire rope to prevent contact between the wire rope wraps. A schematic of the axial displacement of the wire rope across the sheave is shown in Figure 3.7

The tracked sheave approach was selected over other working principles because of the complementary form of the pads. The pad shape enables an even distribution of contact forces, which is more beneficial for wire rope lifetime compared to the offset drum or self-fleeting drum. The operational principles of this concept to achieve the design goals are the same as those for concept 5.

The Strengths, Weaknesses, Opportunities and Threats analysis for this concept is displayed in Table 3.6.

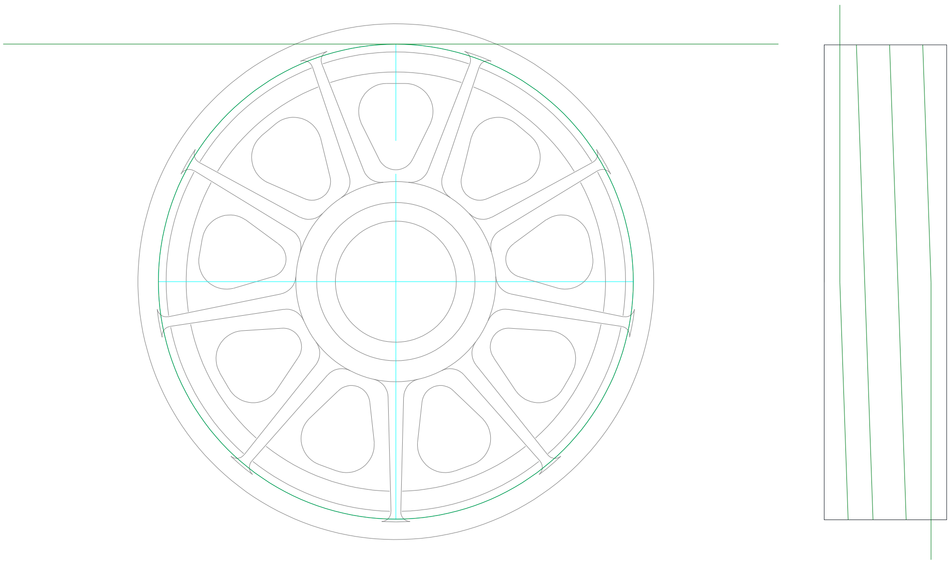


Figure 3.7: Schematic of concept 6 - Tracked sheave

Table 3.6: SWOT analysis for concept 6

Strengths • High traction capability • Can significantly reduce required lower block weight	**Weaknesses** • Complex device
Opportunities • No external pressure on wire, optimal for wire rope lifetime	**Threats** • Reliability

3.3.7. Concept 7 - Sheave with clamping jaws

The final concept is also a sheave-based tensioning device, inspired by the design of Schumann, Schmidt, and Leonhardt [38]. Figure 3.8 presents a schematic of the concept. To enhance traction capabilities, this concept clamps the wire rope (2) to the sheave (1) using clamping jaws (3). These clamping jaws rotate with the wire rope around the sheave. This results in not only a frictional force between the wire rope and sheave, but also a frictional force between the wire rope and the clamping jaws. This gives it higher traction capabilities compared to concept 5.

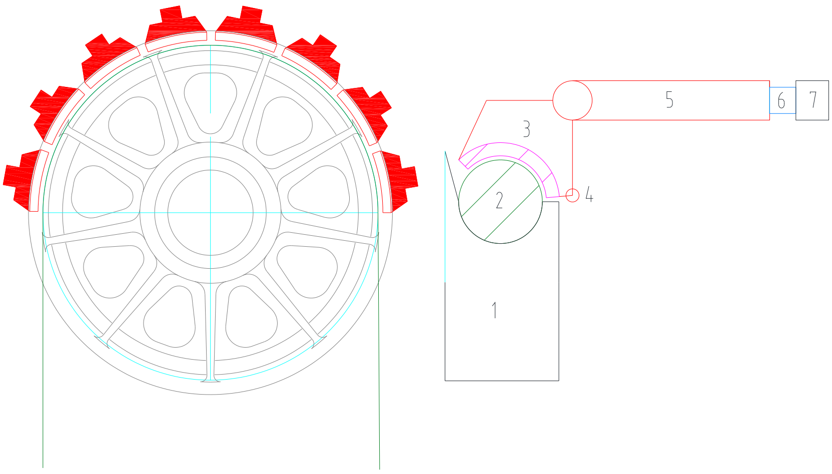


Figure 3.8: Schematic of concept 7 - Sheave with clamping jaws

The clamping of the wire rope is achieved by the compression of a sprung cylinder (5), which clamps the jaw to the wire rope by rotating around a hinge point (4). The compression of the cylinder is induced by a roller (6) moving along a CAM disk (7), which determines the cylinder's compression. This concept enhances the functionality of the design by Schumann, Schmidt, and Leonhardt [38] by making the CAM disk movable. By adjusting the CAM disks position, the force on the wire rope can be controlled, and the clamping jaws can be completely disengaged from the wire rope.

By moving the CAM disk the clamping jaws get removed from the wire rope, removing the increased traction capability. The drive can now freewheel or be set to follow the ropes speed to allow high speed operation. Increasing the tension during hoisting and lowering the required lower block weight is achieved with the same approach as the other sheave based concepts.

The Strengths, Weaknesses, Opportunities and Threats analysis for this concept is displayed in Table 3.7.

Table 3.7: SWOT analysis for concept 7

Strengths • Very high traction capability • No required lower block weight	**Weaknesses** • Complex device
Opportunities • Applied force evenly distributed, low peak pressure	**Threats** • Reliability • Possible wire rope damage during slippage

This concept is very similar to Concept 5, as both utilise an external device to increase the normal force between the wire rope and the sheave. The clamping jaw concept can be viewed as a more complex version of Concept 5, offering improved traction performance. Therefore, it is interesting to compare the tension generation capabilities of these concepts. The graph displayed in Figure 3.9 illustrates the tension generation capabilities of both concepts. It is evident that the clamping jaw concept has greater potential. This graph can be used to determine whether the increased complexity of Concept 7, compared to Concept 5, is justified for a given use case.

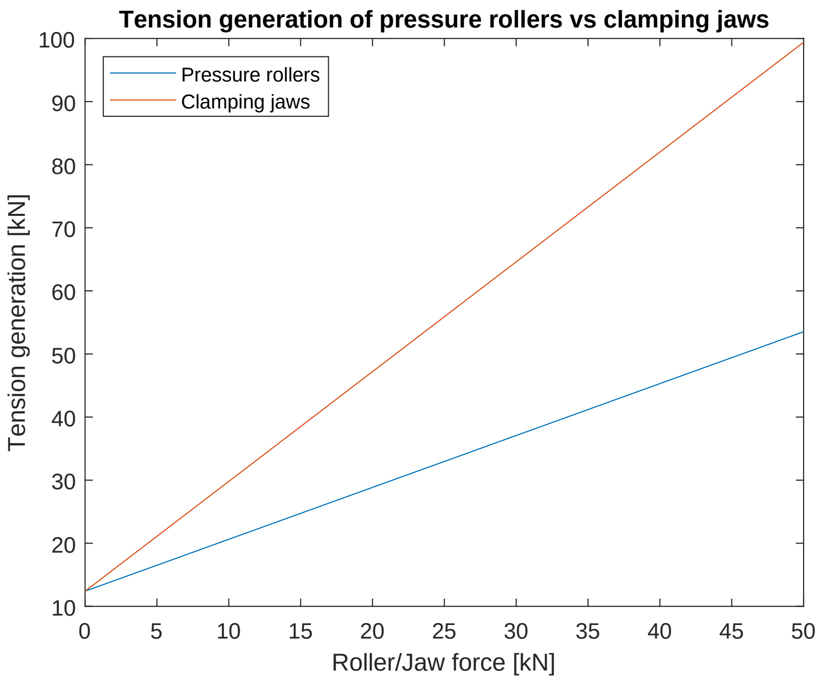


Figure 3.9: Comparison of the tension generation capabilities of concepts 5 & 7

3.4. Requirement and KPI rating

The previous section described the seven concepts and highlighted their positive and negative sides using a SWOT analysis. This section will answer the research question: *"Which concept has the most potential based on the requirements and criteria?"* Firstly the requirements are analysed trough calculations, research and discussion. This analysis will determine which concepts do not meet all the requirements. For all remaining concepts, their performance with respect to the KPIs is analysed. These KPIs, together with the SWOT analysis, will be used to determine which concept has the most potential.

3.4.1. Requirements

The main design goal and therefore most important requirement is the capability of a concept to achieve a line pull of *Confidential*% MBL during hoisting, without significantly increasing the lower block weight. Whether a concept achieves this goal is determined with the altered versions of the parametric model discussed in section 3.2. When designing a tensioning device there are numerous design variables that can be tweaked to gain an optimal tensioner. In order to simplify the assessment a crane previously designed and build by Huisman was chosen as the basis for the calculations. The chosen crane is a 1600mt LEC. It is possible that a concept will work for this crane, but not on others. This is discussed later under the requirement *"The system must work for different types of crane and rigging configurations"*. Table 3.8 shows an overview of the capabilities of the concepts regarding tension generation.

Table 3.8: Assessment of requirement 1 - The system must achieve a line pull of *Confidential*% MBL during hoisting, without significantly increasing the lower block weight

Concept	Performance	Result
1 - Split tackle	During the first step the tension is determined by the tension in the external wire. As long as the external wire an drum have sufficient capabilities, the desired tension can be reached. During the second stage the tension is determined by the lower block weight and the amount of remaining falls. For the 1600mt LEC the tension at the drum is 7.5% MBL.	Pass
2 - Driven sheaves in upper block	The tension generation of this concept is limited by the weight of the lower block and the friction between the sheaves and the wire. for the 1600mt LEC a maximum tension of 4.5% MBL can de reached at the drum.	Fail
3 - Double drum	In this concept any tension, up to the wire rope's safety limit, can be reached at the drum. The only restriction is that the lower block needs to be in the block catcher or connected to a sufficiently heavy load.	Pass
4 - Clamping track	The performance of this concept heavily relies on the chosen values for parameters like friction and clamping force. Even with conservative values, the model shows that this concept can easily be scaled to achieve the desired tension.	Pass
5 - Sheave with roller chain	This concept's performance is mainly influenced by friction and clamping forces as well. Using values deemed appropriate by research and discussion shows the need of 2 devices in series if this concept is chosen.	Pass
6 - Tracked sheave	In this concept the amount of wraps on the sheave is the main design parameter. For the 1600mt LEC, two full wraps are sufficient. Other cranes might require a larger wrapping angle.	Pass
7 - Sheave with clamping jaws	The friction and clamping force determine the friction capabilities of this concept. Using appropriate values results in the model showing this concept can generate sufficient tension.	Pass

The second requirement states that the concept should have a net positive impact on the wire ropes lifetime. As subsection 1.1.1 described, wire rope damage is very complex. This makes it hard to exactly quantify the effect of a concept on the wire rope lifetime. The assessment whether there is a net positive is done to the best of the authors ability and shown in Table 3.9

Table 3.9: Assessment of requirement 2 - The system must have a net positive impact on the wire ropes lifetime

Concept	Performance	Result
1 - Split tackle	While this concept can ensure sufficient tension during hoisting, which significantly reduces wire rope damage, there is a design limitation in this concept. The first layer on the drum cannot be tensioned, as this is still on the drum when the lower block is on the deck.	Fail
2 - Driven sheaves in upper block	The use of V-groove pulleys can reduce the wire rope lifetime upwards of 90%, as discussed in subsection 2.2.2. This is more than the gain in lifetime, resulting in a net negative effect on wire rope lifetime.	Fail
3 - Double drum	The double drum concept will require more bending cycles than current cranes experience, up to 50% more cycles. Tests by Wehking [6] showed a lifetime improvement of 40% with a tripling in bending cycles. As the double drum concept has a lower increase in bending cycles it will probably have an even bigger positive effect on wire rope lifetime.	Pass
4 - Clamping track	This concept does not add any extra bending cycles and grips the wire with relatively low pressure compared to the sheave based tensioners. It is therefore expected to have a net positive effect. Risks of wire rope damage during slippage between the tensioner and wire rope should be taken into account in the risk assessment.	Pass
5 - Sheave with roller chain	This concept introduces two additional bends per cycle. With similar reasoning as Concept 3, it is anticipated that this will result in a net positive effect on the wire rope's lifespan. It is important to note that the rollers will induce high peak stresses at the contact points with the wire rope. However, these peak pressures are lower than those encountered elsewhere in the rigging and are therefore considered acceptable by experts within Huisman.	Pass
6 - Tracked sheave	Only one bend is added per cycle and no external pressure is applied to the wire. A net positive on wire rope's lifetime can therefore be expected.	Pass
7 - Sheave with clamping jaws	The clamping forces on the wire rope result in low peak stresses compared to Concept 5. This concept also only adds 2 bends per cycle. A net positive effect is expected. Risks of wire rope damage during slippage between the tensioner and wire rope should be taken into account in the risk assessment.	Pass

After discussion of the first two requirements, it is clear that Concept 1 & 2 do not meet all requirements. These concepts are therefore not analysed further and excluded from the further selection process. The third requirement states that the system cannot create an unacceptable risk. This is both for operational and safety risks. In order to asses this a risk analysis is performed for all concepts. This risk assessment was done following the Huisman standard [71]. Which itself is based on multiple ISO standards. The risk assessment process is explained below.

Firstly, all potential risks associated with each system function are identified. The possible harm they could inflict and their causes are explored and explained. Subsequently, the severity and likelihood of occurrence are assessed for each risk. Multiplying these factors yields the risk score, which ranges from 1 to 25. Risks scoring between 1 and 4 fall into the low-risk category. Scores from 5 to 9 are classified as moderate risk, while scores from 10 to 25 are considered high risk. Risks in the low-risk category are deemed acceptable. Moderate risks are tolerated, but measures to mitigate these risks are strongly preferred. High risks are unacceptable, and measures should be implemented to reduce the risk to at least moderate levels. The results from the risk assessment indicate where and what measures are required to mitigate the risks of the concepts.

The risk assessments performed for the five concepts identified a total of 137 risks. It is impractical to provide a complete summary of all these risks. Instead, two risks identified for four of the five concepts are highlighted, along with some concept-specific risks discussed in Table 3.10.

In the risk assessment, the same two high-risk category risks were identified for every concept, except for Concept 3. The first risk is: ”The system activates during hoisting or lowering at SWL.” This would increase the tension in the wire rope beyond its safety limit, posing a significant safety hazard and potentially causing substantial damage to the wire rope. This risk can be mitigated by implementing a control system that prevents system activation during heavy loading of the crane. Additionally, warnings and alarms could be provided to the crane operator.

The second risk is: ”Insufficient tension generated during empty lowering.” If the active wire rope tensioner is used to lower a light lower block and insufficient tension is generated, the lower block could move upwards uncontrollably. This could result in a collision with the block catcher, causing damage or destruction to both the lower block and the block catcher. This represents a significant safety and operational risk that must be avoided. Potential solutions include adding an emergency brake that clamps the wire rope or ensuring the lower block maintains a minimum weight to prevent unintended upward movement.

Table 3.10 provides an overview of the concept-specific risks identified in the risk assessment. A concept fails this requirement if it creates a risk that cannot be mitigated and therefore cannot be accepted.

Table 3.10: Assessment of requirement 3 - The system cannot create an unacceptable risk

Concept	Performance	Result
3 - Double drum	This concept has no additional risk when compared to a regular crane.	Pass
4 - Clamping track	If the wire rope is not positioned correctly when to links close, it could deform and be damaged due to possible sharp edges on the links. This risk can be mitigated by using slack rope detection and guide rollers.	Pass
5 - Sheave with roller chain	The shape of the rollers will induce high peak stresses on the wire rope. It is currently unknown whether, and to what extent, these peak stresses may damage the wire rope. Experts within Huisman believe these risks are low. However, the impact of high peak pressure should be investigated if this concept is selected.	Pass
6 - Tracked sheave	If the track of the tracked sheave malfunctions and cannot rotate, the wire rope will slip across the track under high tension, resulting in wire rope damage. This risk is considered low and deemed acceptable.	Pass
7 - Sheave with clamping jaws	In the event a clamping jaw would not retract, the jaw would either break or force the wire rope off the sheave. This would have a severe impact on the crane. However, the likelihood of occurrence has been deemed low, and therefore it is considered an acceptable risk.	Pass

The final requirement ensures that the concept is compatible with various types of crane and rigging configurations. This is crucial because Huisman designs multiple types of cranes, all of which could be susceptible to wire rope damage depending on their usage. Table 3.11 provides an overview of the performance of the concepts.

Table 3.11: Assessment of requirement 4 - The system must work for different types of crane and rigging configurations

Concept	Performance	Result
3 - Double drum	This concept can be applied to all types of cranes and rigging configurations, except for knuckle boom cranes. These cranes only have a single fall, making it impossible to connect the other end of the wire rope to a drum, as it has an end connection to the hook.	Pass*
4 - Clamping track	The concept can be applied along any straight piece of wire rope and is not effected by the rigging configuration.	Pass
5 - Sheave with roller chain	As long as there is sufficient space in the boom for two additional sheaves, this concept can be applied to every type of crane and is not affected by the rigging configuration.	Pass
6 - Tracked sheave	As long as there is sufficient space in the boom for an additional sheave, this concept can be applied to every type of crane and is not affected by the rigging configuration.	Pass
7 - Sheave with clamping jaws	Equal to concept 5.	Pass

*Although the concept does not work for all types of cranes, it does work for a vast amount of them. Therefore it was decided to classify this requirement as a pass.

3.4.2. KPIs

Following the assessment of each concept's ability to fulfil the requirements, five concepts remain viable. This section will evaluate the performance of each concept with respect to the KPIs. The results of this analysis will be used to determine which concept has the most potential to be the optimal active wire rope tensioner.

The first KPI discusses the impact of the concept on the operating speed of the crane. This covers both the speed when the concept is used, but also it's impact on operating speeds when it is inactive. Table 3.12 shows and overview of each concept's impact on the operating speed.

Table 3.12: Assessment of KPI 1 - Impact on operating speed

Concept	Performance	Result
3 - Double drum	This concept does not influence the hoisting or lowering speed of a crane during operation and may even enhance these speeds when both drums are utilised simultaneously. However, the additional spooling operation requires a considerable amount of time (14 minutes for the 1600mt LEC). Consequently, this significantly impacts the overall operational speed of the crane.	+/-
4 - Clamping track	For all friction-based tensioner concepts, the system's speed when active is determined by the drive's power. Under normal operating speeds a power of 81 kW is required to generate the desired tension. When this concept is not utilised, the CAM guides can be retracted, and the system will be completely removed from the rigging. Therefore, it will have no impact on the operating speed when inactive.	++
5 - Sheave with roller chain	The impact of this concept on speed when in use is also determined by the power of the drives. As this concept requires two sheaves, both should have adequately sized drives (36 kW and 46 kW). When the system is inactive, its inertia affects the crane's speed. If the tensioner's drive is not engaged, the concept's inertia slows down the (de)acceleration of the main drum. This could be mitigated by controlling the drive of the concept to assist in (de)accelerating the sheaves, but this requires continuous use of an active control system.	+
6 - Tracked sheave	Similar to concept 5, as long as the drive is sufficiently sized, it has no impact on the speed when active. When inactive the inertia will affect the operating speed. To prevent this, a control system could continuously be used to match the sheave speed to the wire rope speed.	+
7 - Sheave with clamping jaws	Equal to concept 6.	+

In addition to the operating speed of the crane, it is important to consider whether the concepts introduce any additional procedures. Ideally, a concept should not require any thought or input from the crane operator. Whether an active wire rope tensioner operates automatically or at the operator's discretion is a control decision that can be made later by Huisman or the client. However, having the option for fully automatic operation is preferred. An overview of the concepts' performance can be found in Table 3.13

Table 3.13: Assessment of KPI 2 - Added procedures

Concept	Performance	Result
3 - Double drum	This concept would require the operator to decide whether to execute the tensioning process before each hoisting operation. This necessitates the operator making an informed decision prior to each hoisting task. It also adds an extra step which, as mentioned before, takes quite some time.	-
4 - Clamping track	If this concept is integrated with a proper control system and the necessary sensors, it could operate automatically. It would also add no procedural steps to the crane's operation, apart from the (de)activation of the system.	+
5 - Sheave with roller chain	Equal to concept 4.	+
6 - Tracked sheave	Equal to concept 4.	+
7 - Sheave with clamping jaws	Equal to concept 4.	+

The final KPI to be analysed is the ability of the concepts to decrease the lower block weight. This was assessed using the same parametric models employed to determine whether the concepts can generate a line pull of *Confidential*% MBL at the drum. The results are presented in Table 3.14. These results represent the required lower block weight, based on the concept configurations set to achieve the *Confidential*% MBL line pull.

Table 3.14: Assessment of KPI 3 - Decrease required lower block weight

Concept	Performance	Result
3 - Double drum	This concept cannot decrease the required lower block weight.	-
4 - Clamping track	In theory, this concept could facilitate a lower block of 0 mt. While this is not feasible, it allows for a lower block design optimized for low weight.	++
5 - Sheave with roller chain	Equal to concept 4.	++
6 - Tracked sheave	This concept requires a lower block mass of 24 mt, for the 1600 mt LEC. This represents a significant weight reduction of 65% compared to the original, although a minimum weight is still necessary.	+
7 - Sheave with clamping jaws	Equal to concept 4.	++

3.5. Decision

The previous section attempted to assess whether the concepts can fulfil the system requirements and quantify their performance with respect to the KPIs. This has proven challenging for some requirements and KPIs. The primary difficulty was determining whether the concepts would positively affect wire rope lifetime. Wire rope damage is highly complex, and assessing how the reduction of one damage factor relates to an increase in another has been challenging. Therefore, the assessment is somewhat subjective. It was ultimately based on the author's knowledge of wire rope damage, partly described in chapter 1 and 2, and discussions with wire rope experts within Huisman. The risk assessment is similarly subjective, as the severity and likelihood of occurrence are estimations.

During the requirement assessment, concepts 1 and 2 did not meet all requirements. Therefore, these two concepts are not considered in the further concept selection. The evaluation of each concept's performance for the KPIs is used to identify the most promising concept. This is done using the weighted sum model. This model assigns a weight to each KPI to signify its importance. This weight is then multiplied by a concept's performance regarding that KPI. Adding these weighted performances for a concept gives a total score for that concept [72]. Table 3.15 shows the weighted sum model resulting from the concept's performance.

Table 3.15: Weighted sum model used to select the most promising concept

Concept	KPI			Total
	1	2	3	
3 - Double drum	+/-	-	-	-3
4 - Clamping track	++	+	++	10
5 - Sheave with roller chain	+	+	++	7
6 - Tracked sheave	+	+	+	6
7 - Sheave with clamping jaws	+	+	++	7
Weight	3x	2x	1x	

From the weighted sum model, several conclusions can be drawn. Firstly, the double drum concept performs the worst among all concepts due to its poor performance with respect to each KPI. The concept is more 'out of the box' than concepts 4 to 7, which results in a less optimal solution based on the requirements and KPIs presented in this thesis. The same can be said for the other 'out of the box' concepts, concepts 1 and 2. These concepts did not meet all requirements and are not suitable to fulfil the current vision of the active wire rope tensioner's characteristics. However, if the vision on these characteristics changes in the future, the 'out of the box' solutions might become interesting, as they possess advantages that concepts 4 to 7 do not have.

Secondly, it is notable that concepts 4 to 7, the friction-based traction devices, all score very similarly. This is not surprising, given their similar working principles. They mainly differ in how they enhance their traction capabilities, but as described in section 2.2, there are only three main approaches to this. Because these concepts score so similarly, identifying the most promising one is challenging. Despite efforts to objectively quantify the performance of each concept, some level of subjectivity remains in the decision.

To reach a decision, all information described in this chapter was considered. Each concept has its own advantages and disadvantages compared to the others, which are not all apparent from the weighted sum model alone. After discussing all the pros and cons of each concept, using the weighted sum model as a basis for the discussion, the clamping track was determined to be the most promising concept.

The clamping track concept has a number of advantages over the other concepts. The tension generation capabilities of the clamping track can easily be scaled by adjusting the length of the track and the force applied to the wire rope. This makes it ideal for different (types of) cranes. By selecting the appropriate length and pressing force, the contact pressure on the wire rope can be kept lower compared to sheave-based designs. Also considering the fact that the clamping track does not introduce a bend into the crane's rigging, it is superior in extending the wire rope's lifetime. Another significant advantage of this concept is that it can be completely removed from the crane's rigging, ensuring no impact on the crane's operating speed when not in use. Its excellent tension generation capabilities also allow for a lighter lower block and the potential to increase the line pull at the drum above *Confidential*% MBL, if ever desired.

Lastly, as briefly mentioned in section 3.2, the installation location of Concepts 4 to 7 within the crane can significantly influence their performance. The models used in the previous section assumed the tensioner would be installed at the top of the boom. However, in practice, it is advantageous for the installation location to be flexible. Modelling indicate that the tension generation during hoisting for the clamping tracks is independent of its location, whereas the performance of the other concepts is constrained by their installation location. This advantage was not considered as a KPI, but it does increase the confidence in selecting the clamping tracks as the most promising concept.

4 Detailed design

The previous chapter concluded with the selection of the most promising concept, the clamping track concept. This chapter describes the detailed design of the clamping track concept, hereafter referred to as the tensioner. Firstly, the location of the tensioner within the system is discussed, followed by the determination of the tension generation requirements. These requirements are used to detail the mechanical design of the tensioner. Additionally, the potential improvement of the load curve is examined, along with the control system necessary for the tensioner.

The tensioner detailed in this chapter is designed to function on the 1600 mt LEC, the crane which was also used in chapter 3 to assess the requirements and criteria. However, the principles behind the tensioner are not crane-specific. This means that the tensioner can be scaled with relative ease for application in different cranes.

4.1. Tensioner location

As briefly mentioned at the end of chapter 3, the installation location of the system is a crucial factor to consider. It influences tension generation requirements of the tensioner itself, as well as the practicality of its application. This consideration is particularly relevant if the lower block weight is reduced, as this decreases the tension in the wire rope.

At first glance, a reduction in wire rope tension may not seem problematic, since the tensioner's purpose is to increase this tension. However, it does impose a limit on the flexibility of the installation location. This limitation arises because zero tension in the wire rope should be avoided. If zero tension occurs in the wire rope this can lead to slack rope issues, the wire rope running off the sheave, or the lower block being unable to lower. If the lower block weight is reduced and the tensioner is installed too low in the crane, parts of the wire rope could experience zero tension.

Lower block weight reduction

To determine the optimal location of the tensioner in the crane, it is necessary to assess the potential reduction in lower block weight. Currently, lower blocks are designed with a minimum weight requirement to ensure proper lowering and spooling during hoisting. This minimum weight is sometimes achieved by adding extra steel to the lower block, although this may not always be necessary.

Since there is currently no need to optimally design a lightweight lower block, estimating the realistic possible reduction in lower block weight is challenging. After reviewing the designs of multiple lower blocks previously made by Huisman and consulting with company experts, a potential reduction of 40% was chosen as the basis for further calculations.

Installation in the boom

Two potential installation locations are evaluated in the conceptual detailing. The first location is along the wire rope running atop the boom. This position offers the primary advantage of significantly reducing the lower block weight. This is due to the fact that the tension in the wire rope, induced by the lower block weight, cannot reach zero tension as previously mentioned. As the wire rope tension diminishes due to its own weight, the higher the system is installed on the boom, the lighter the lower block can become. However, this installation location has downsides, including difficulty in accessing the system for servicing and wiring. Additionally, the weight of the tensioner can negatively impact the crane's load curve.

The optimal location for the tensioner is as low as possible in the boom, while still accommodating a certain lower block weight. This positioning minimizes the impact of tensioner's weight on the centre of gravity on the boom. Using a model based on the previously utilized parametric model, it is possible to calculate the optimal location along the boom based on the chosen lower block weight. Figure 4.1 presents the results of this model, showing the required lower block weight as a function of the tensioner's position along the boom. Knowing that the lower block weight can be reduced by a maximum of 40%, the optimal location for the 1600mt LEC is determined to be 36 meters from the boom pivot. This installation distance can be increased if desired, but not decreased, as doing so would result in zero tension in the wire rope.

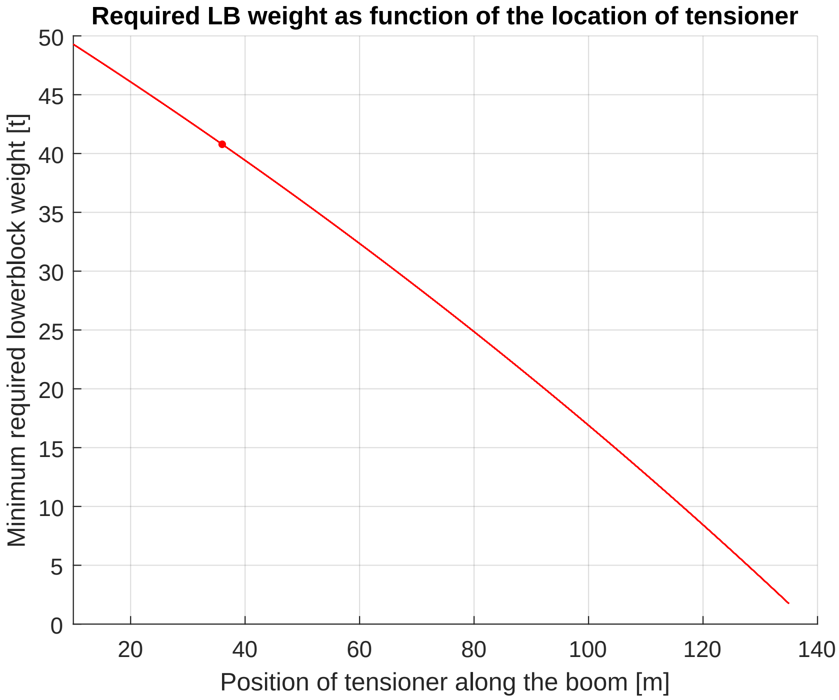


Figure 4.1: Visual representation of the model used to determine the optimal installation location along the boom.

Installation in the luffing frame

The second installation location considered is the luffing frame. The main advantage of installation in the luffing frame is that it does not add weight to the boom, thereby not negatively impacting the load curve. Additionally, installing the system in the luffing frame would make access to the tensioner easier, which is beneficial for servicing and wiring. However, the downside is that the reduction in lower block weight is limited.

Before the wire rope reaches the tensioner in the luffing frame, it traverses the entire length of the boom via a sheave located near the boom pivot point. It is crucial to maintain tension in the wire rope along its entire length. Therefore, the lower block must be sufficiently heavy to ensure tension at the sheave near the boom pivot point. For the 1600mt LEC, this determines that the lower block weight can be reduced by a maximum of 20%.

Three scenarios

Based on the information discussed above the design of the tensioner track is continued with three possible scenarios. These are the following:

1. Installation in the boom at 36 meters from the pivot, lower block weight reduction of 40%.
2. Installation in the luffing frame, lower block weight reduction of 20%.
3. Installation in the luffing frame, no lower block weight reduction.

Figure 4.2 visualizes the three scenarios. It shows a schematic drawing of a crane and the tension in the wire rope throughout its rigging. This figure illustrates several key aspects. Firstly, it highlights the change in wire rope tension caused by the wire rope's own weight. Secondly, it shows the increase or decrease in tension at each sheave, influenced by the sheave efficiencies. Additionally, it indicates the difference in lower block weight, as shown by the tension variation in the wire rope at the top of the crane. A higher lower block weight results in greater tension in the wire rope. The limitation to the lower block weight reduction is also visible in Figure 4.2b. Decreasing the lower block weight further would result in parts of the wire rope with zero tension.

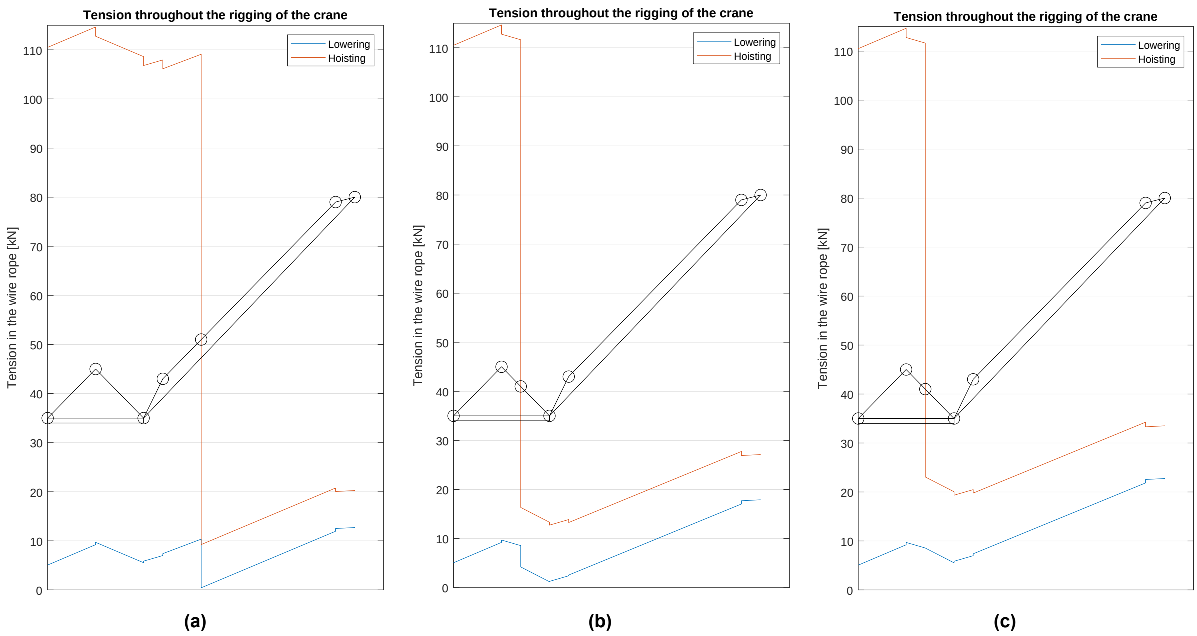


Figure 4.2: Schematics drawing of the tension throughout the reeving for the three scenarios: a) Installation in the boom, 40% weight reduction. b) Installation in the luffing frame, 20% weight reduction. c) Installation in the luffing frame, no weight reduction.

4.2. Tension generation requirements

In order to detail the design of the tensioner track it is important to determine the tension generation requirements of the system. As this will determine numerous aspects of the design. These requirements are influenced by a number of factors. These factors are itemized below:

- **The tension goals** are a critical factor as they dictate the amount of tension the combination of the lower block and the tensioner track should achieve. During hoisting, the target is set at *Confidential*% MBL, as discussed previously. During lowering, Huisman currently uses a tension goal of *Confidential*, a rule of thumb based on their experience.

  For detailing the tensioner track, a different value was chosen. The parametric model indicates that the current lower block is too light to achieve this tension target, and would therefore not lower properly. However, this crane does not experience these kind of issues in real-world usage, suggesting that this rule of thumb is not appropriate for this model. Instead, a value of 5 kN was chosen, which is the tension caused by the current lower block weight according to the model.

- **The lower block weight** significantly influences the tension generation requirements. Currently, the lower block weight is the primary method used to set the tension in the wire rope. As discussed in section 4.1, the design of the tensioner track is based on three scenarios, each with its own lower block weight.

- **The crane configuration** also influences the requirements of the tensioner track. The most important factors are the vertical distances between the sheaves, the wire rope's MBL, and its density. Other factors that impact the design include the wire rope diameter, sheave diameter, number of falls, and the speed of hoisting and lowering.
- **The sheave efficiency** has great influence on the tension in the wire rope as previously discussed. Based on the crane configuration and the lower block weight the sheave efficiency can be accurately determined.
- **The position of the system** also influences the tension generation requirements as discussed in the previous section.

With all these factors known, it is possible to calculate the required tension generation capabilities. This was done by combining two calculations. First, the required tension at the location of the tensioner, starting with the tension goals at the drum, was determined (Equation 4.2). Second, the tension at the location of the tensioner, caused by the lower block weight, was determined (Equation 4.5). The difference between these values should be generated by the tensioner (Equation 4.6). The equations below illustrate this for the tensioner placed in the luffing frame.

In these equations is F the tension in the wire rope, ρ the weight of the wire rope per unit length, dZ the vertical distance between two sheaves, x the location of the tensioner between two sheaves and m the lower block weight. The efficiency is noted as η for simplicity. However, efficiency is a function of multiple factors, more accurately denoted as $\eta(F,\, MBL,\, d_{wire},\, D_{sheave},\, state)$, where "state" indicates whether the crane is lowering or hoisting.

$$\begin{aligned} F_{drum} &= 5\,kN && \text{during lowering} \\ F_{drum} &= \textit{Confidential}\%\,MBL && \text{during hoisting} \end{aligned} \tag{4.1}$$

$$F_{track\,drum} = (F_{drum} + \rho g dZ_1)/\eta - \rho g x \tag{4.2}$$

$$\eta_{LB} = \sum_{i=0}^{n_{falls}-1} \frac{1}{\eta^i} \tag{4.3}$$

$$F_{first\,fall} = mg/\eta_{LB} \tag{4.4}$$

$$F_{track\,block} = \eta(\eta(\eta(\eta F_{first\,fall} - \rho g dZ_5) - \rho g dZ_4) - \rho g dZ_3) - \rho g(dZ_2 + x) \tag{4.5}$$

$$F_{track} = F_{track\,drum} - F_{track\,block} \tag{4.6}$$

The requirements for the tensioner installed in the studied crane following from these equations are presented in Table 4.1.

Table 4.1: Tension generation requirements for the three scenarios

	Scenario 1	Scenario 2	Scenario 3
Hoisting	99.9 kN	95.3 kN	88.6 kN
Lowering	9.9 kN	4.3 kN	0 kN

4.3. Mechanical design

With the location of the tensioner determined and the tension generation requirements known the tensioner itself can be designed. This was started with a risk analysis to gain insight into safety aspects of the design. After which all subsystems could be designed.

4.3.1. Detailed risk assessment

Safety is a critical aspect of the active wire rope tensioner. To design a system that is inherently as safe as possible, a second risk assessment was conducted at the subsystem level. For each subsystem, all risks were identified, and potential risk reduction measures were proposed. These identified risks are utilized in the further design process to optimize the inherent safety of the device. Such a risk assessment was performed for each of the three scenarios, as they design of the tensioner differs slightly based on the scenario. The complete risk assessment can be found in Appendix C.

In the risk assessment, several risks were identified that can be mitigated through the subsystem design philosophy. Firstly, it is crucial that the clamps of the tensioner open before rotating around the sprocket. Failure to do so may result in the clamps attempting to bend the wire rope, causing damage to either the wire rope or the clamps themselves. Secondly, it is important that the clamping mechanism, which presses the clamps onto the wire rope, moves and clamps uniformly along its length. Failure in this aspect can lead to the wire rope being displaced from its path and higher peak clamping pressures, resulting in damage to the wire rope or the clamps. Lastly, it is essential to study the emergency stop behaviour of the tensioner. Depending on this behaviour, slack rope or slip could occur, as well as an increase in wire rope tension beyond its safety limits.

4.3.2. Detailing of the track

The track is the subsystem at the heart of the active wire rope tensioner. It consists of a chain driven by a sprocket. The chain is supported by rollers and tensioned with a hydraulic cylinder. Each link of the chain has a clamp mechanism on top of it. By pressing the clamps on the wire rope and driving or braking the sprocket the tension in the wire rope is increased. This subsection will highlight multiple design aspects of the track.

Polygon effect

The clamping track system is driven by a chain drive. At both ends of the track are two sprockets that allow the chain to turn and drive the chain. In a chain drive system, the polygon effect occurs. This effect is caused by the fact that a chain wrapped around a sprocket does not form a perfect circle, but a polygon. Rotating the sprocket at a constant speed results in periodic speed fluctuations of the chain. These fluctuations can lead to undesired dynamic effects and may cause vibrations that resonate with the system's natural frequency [73].

The speed fluctuation caused by the polygon effect is often expressed using the coefficient of speed fluctuation δ. It can be calculated using Equation 4.7 [74], where z is the number of teeth on the sprocket.

$$\delta = \frac{v_{max} - v_{min}}{v_{avg}} = 2tan^2\frac{\pi}{2z} \tag{4.7}$$

Plotting this equation, as shown in Figure 4.3, reveals that it is an asymptotic function. The effect will never be zero, but after a certain number of teeth, the effect approaches its minimum. Both Mahalingam [73] and Li, Chai, Wang, *et al.* [74] suggest the usage of an absolute minimum of 15 teeth. Additionally, Iwis [75] states that after more than 19 teeth, the effect does not significantly decrease further. Therefore, a sprocket with 20 teeth was selected for the clamping track, resulting in a speed fluctuation coefficient of 0.014.

The polygon effect can further complicate the system's dynamics. The distance between the two sprockets and the number of teeth on both sprockets can cause additional dynamic effects. These effects can be mitigated by making both sprockets identical and positioning them at a distance that is an integral multiple of the chain length. This approach will be applied to the design of the clamping track.

Chain link design

With the number of teeth of the sprocket known the dimensions of the chain link and the sprocket can be determined. The relation between the number of teeth, sprocket radius and link length is shown in Equation 4.8. With a sprocket diameter of 300 mm the chain link length is 93.9 mm.

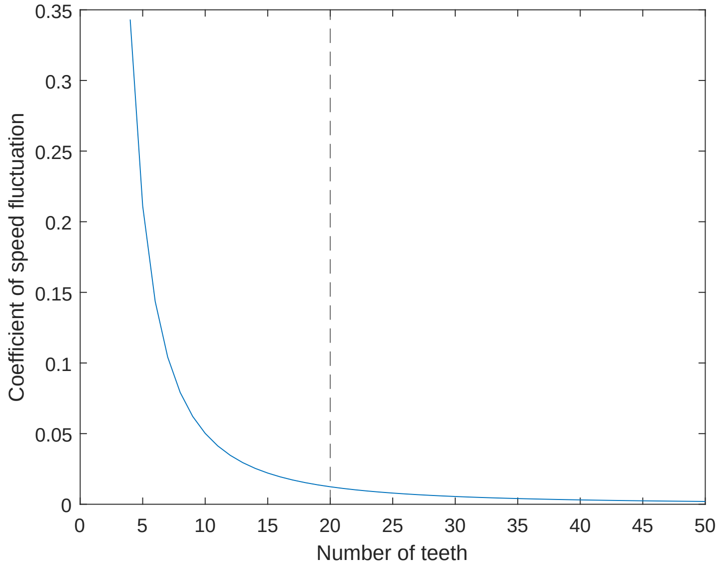


Figure 4.3: Relation between the number of teeth of the sprocket and the chain speed fluctuations caused by the polygon effect

$$l_{link} = 2r_{sprocket}sin(\frac{\pi}{z}) \tag{4.8}$$

Figure 4.4 illustrates the design of a chain link. At the base of the chain link are two plates connected by a link pin, which is used to connect links together and drive the chain, similar to a conventional chain drive. Positioned above this is the clamping mechanism, consisting of two clamp halves that can rotate around an axis. The clamps have a circular shape with a radius of $0.525d_{wire}$. Using a smaller radius would squeeze the wire rope, resulting in high stresses in the clamp, perpendicular to the clamping forces. Conversely, a larger radius could lead to deformation of the wire rope or uneven distribution of the load across the contact area. When fully closed, the clamps nearly encircle the wire rope, ensuring that the wire rope does not deform under high pressures.

On the back of each clamp is a roller with a long pin. This roller transfers the clamping force between the guide rail and the wire rope, while allowing smooth movement of the track. The pins are used to open the clamps when necessary, as further elaborated later in this section.

The width of the clamp is constrained by the limited chain link length. In the design, the clamps have a width of 50 mm, matching the wire rope diameter. This results in the contact area for each clamp half being $A_{clamp} = d_{wire}^2$.

Chain design

The tensioner increases the tension in the wire rope through the frictional forces between the clamps and the wire rope. One of the critical factors influencing this is the coefficient of friction between the wire rope and the clamps. Conventional tensioner tracks, used for pipe or cable laying, employ pads for the contact between the pipe or cable and the track. These pads are often made of materials with an improved coefficient of friction compared to steel, such as rubber and polyurethane (PU). However, the friction between rubber or PU and wire rope is not well-documented.

In subsection 2.2.2, additional materials were discussed to enhance the friction between the wire rope and the contact material. Research into the friction coefficients between these materials and wire rope shows minimum values of 0.15, with the potential for even higher friction. These materials, however, require the use of very low contact pressures. Due to this limitation and the lack of knowledge on the friction between rubber or PU and wire rope, the design of the tensioner assumes a frictional coefficient of 0.1, as discussed in subsection 2.1.3.

To determine the required clamping force, the Coulomb friction model is utilized. For scenario 1, this results in a required clamping force of 999 kN. This represents the total clamping force exerted by the

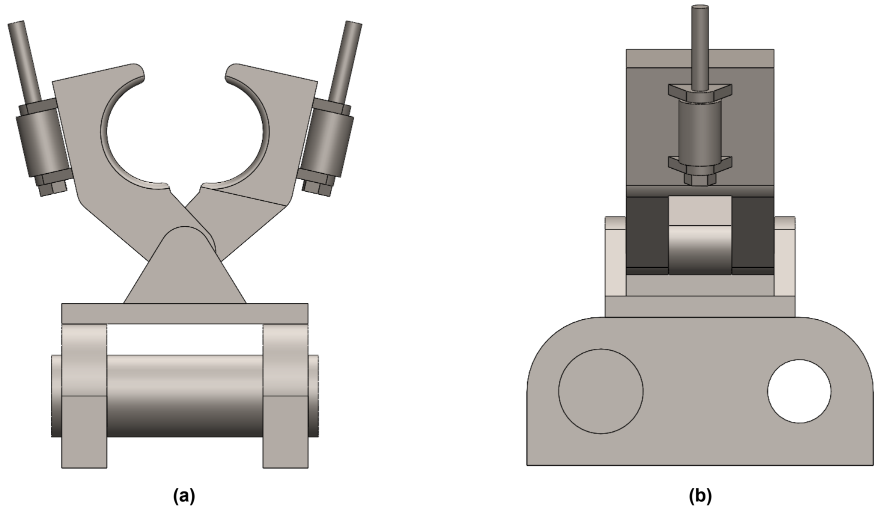


Figure 4.4: Clamp of the tensioner mounted on a chain link. a) Front view. b) Side view.

tensioner on the wire rope. Within the tensioner, this force is distributed across multiple clamps and both sides of each clamp. The number of clamps engaged with the wire rope simultaneously is dictated by the maximum allowable contact pressure.

Determining the maximum allowable contact pressure has proven to be challenging. Limiting the maximum clamping pressure is essential to prevent damage to the wire rope. However, research on wire rope damage caused by radial pressure is limited. High radial pressure damages the wire rope through two mechanisms. The first mechanism is the flattening of the wire rope, which occurs if the wire rope is radially loaded through a contact shape that is not complementary to the wire rope's shape. This can happen with a groove radius that is too large or flattening between two steel plates, leading to high internal stresses and eventually flattened portions of the wire rope [76]. Since the shape of the clamps is complementary to the wire rope's shape, this damage mechanism is likely negligible.

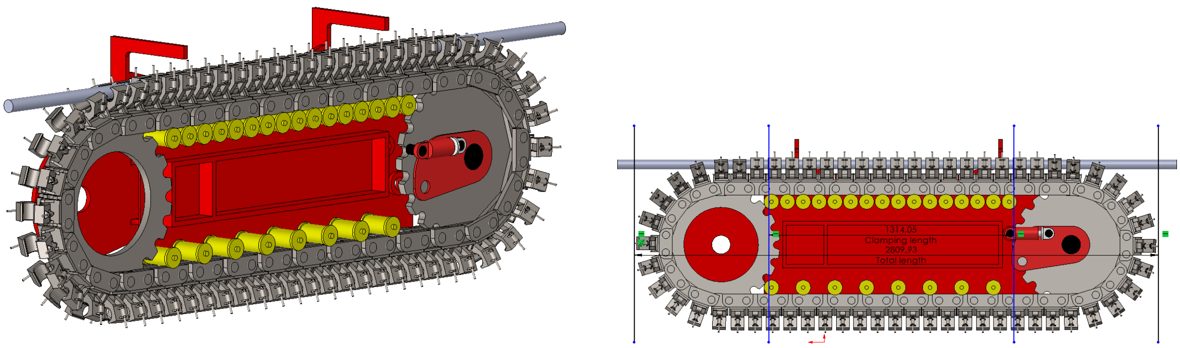
The second damage mechanism is the deformation of the outer strands of the wire rope caused by local peak pressures. Deformation of the outer strands makes them more susceptible to fatigue damage and creates a non-uniform distribution of the load across the strands, reducing the wire rope's lifespan [77] [78]. To mitigate the risk of local peak pressures damaging the wire rope, the nominal clamping pressure in the clamps has been set at a relatively low pressure of 15 MPa. This pressure is relatively low compared to the nominal pressure of 90 MPa that wire ropes experience when wound on the drum under full load.

The number of clamps that are engaged to the wire rope simultaneously is determined using Equation 4.9. Scenario 1 will required a minimum of 14 clamps under the set conditions. This sets the length of clamping part of the tensioner.

$$n_{clamps} \geq \frac{F_{clamptotal}}{2A_{clamp}p_{clamp}} \tag{4.9}$$

The design of the track is illustrated in Figure 4.5a, which shows the chain revolving around two sprockets. The sprocket on the left is connected to the drive system, while the sprocket on the right is connected to two hydraulic cylinders. These hydraulic cylinders provide pretension in the chain. Pretension is crucial to maintain continuous contact between the sprocket and the chain, as well as to smooth and stabilize the dynamic response of the chain [79]. The yellow cylinders represent a set of rollers. These rollers, along with the chain pretension, ensure that the clamps remain aligned with each other and the wire rope.

Figure 4.5b depicts the length of the track assembly. Due to the large sprocket diameter, the clamping distance in this design is approximately 50% of the total length. Reducing the clamping distance, for example by increasing the clamping pressure, would result in a shorter track length. However, there is a limit to how much the track can be shortened, imposed by the sprockets and the distance required to open and close the clamps.



(a) (b)

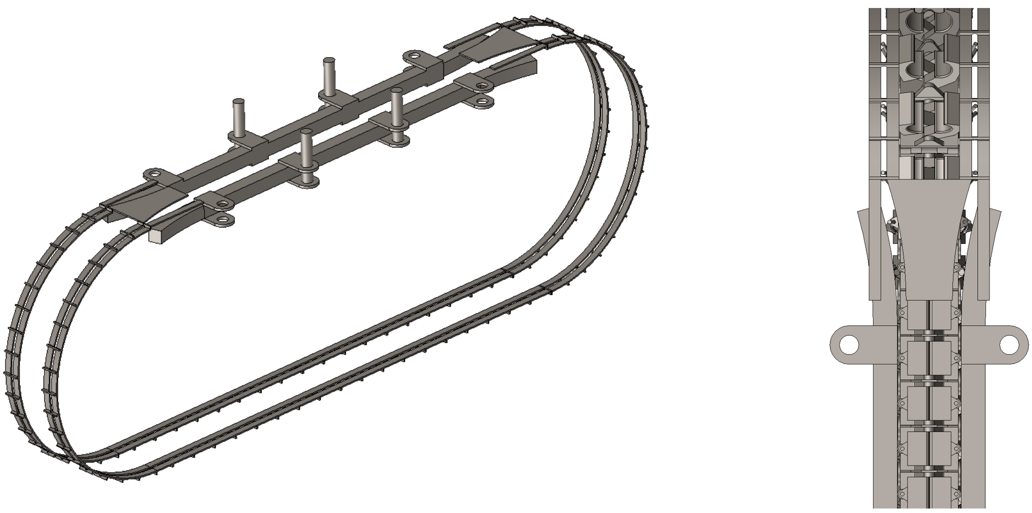
Figure 4.5: Chain of the active wire rope tensioner. a) Overview of the chain assembly. b) Detailed view of the tensioner length.

Guide rails

For proper functioning of the tensioner it is crucial that the clamps are positioned correctly. Along the clamping length of the tensioner the rollers of clamps roll along two guide rails, pressing the clamps on the wire rope. At the end of the clamping length the clamps must open away from the wire rope before they rotate around the sprocket. If this does not happen before this rotation is made this could damage the clamps and the wire rope. This is one of the critical risks identified in the risk assessment.

Two devices with similar functionality discussed in subsection 2.2.1 rotate the clamps with springs. These springs force the clamps open when they are not pressed onto the wire rope by the guide rail. This principle is very simple and also offers the advantage of compactness. But it comes with the risk that the springs do not open the clamps (fast enough). This could be caused by damaged springs or by contamination of the rotating parts.

Because the risk of the clamps not opening in time is unacceptable a more inherently safe design was chosen. The pins extending upwards from the clamps run trough a set of guide rails which force the clamps open or closed. This is shown in Figure 4.6.



(a) (b)

Figure 4.6: Guide rails to ensure proper opening and closing of the clamps. a) Overview. b) Detail of the opening with clamps.

4.3.3. Detailing of the clamping mechanism

The clamping mechanism is responsible for pressing the clamps to the wire rope with the desired force, as well as allowing the clamps to fully open, effectively disengaging the tensioner from the crane's rigging. The risk assessment identified a potential risk in this mechanism: if the guide rails that perform the clamping action move or apply pressure non-uniformly, the wire rope could be displaced from its path, and the clamping pressure could exceed the set maximum.

To mitigate this risk, a design was chosen that utilizes two mirrored parallelograms powered by a single hydraulic cylinder. This design is illustrated in Figure 4.7. Since actuation is performed with a single hydraulic cylinder, the force and movement of both parallelograms will always be equal.

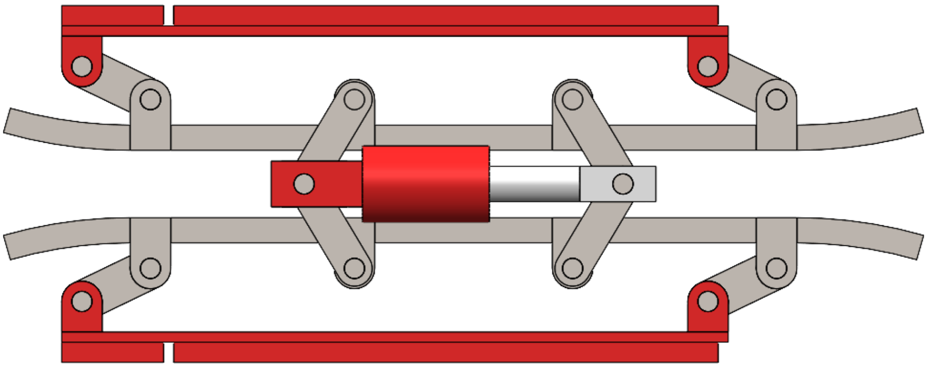


Figure 4.7: Clamping mechanism of the active wire rope tensioner

To dimension the hydraulic cylinder for the clamping mechanism, it is essential to determine the required force. Figure 4.8 presents a free body diagram of the clamping mechanism. From this diagram, equations can be derived to calculate the necessary cylinder force.

$$\sum F_y = 0 = F_{rod}cos(\alpha) - F_{rod}cos(\alpha) - 2F_{support}cos(\beta) \rightarrow F_{support} = 0 \tag{4.10}$$

$$\sum F_x = 0 = F_{clamp} - 2F_{rod}sin(\alpha) \rightarrow F_{rod} = \frac{F_{clamp}}{2sin(\alpha)} \tag{4.11}$$

$$F_{cylinder} = 2F_{rod}cos(\alpha) = \frac{F_{clamp}cos(\alpha)}{sin(\alpha)} \tag{4.12}$$

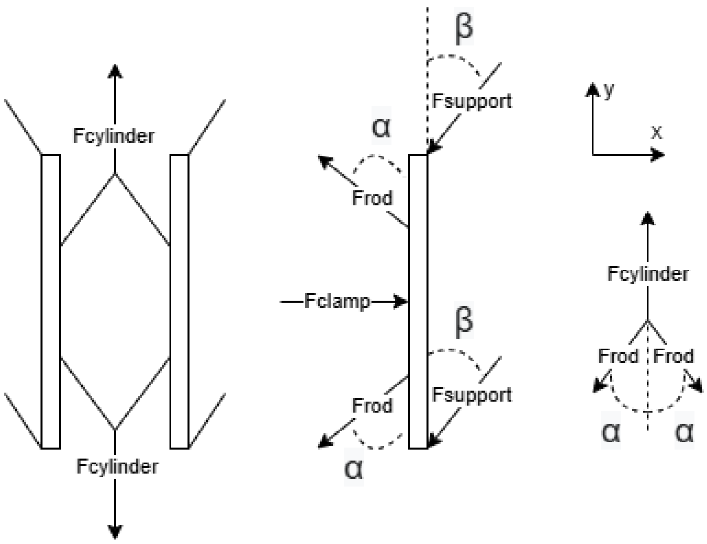


Figure 4.8: Free body diagram of the clamping mechanism

From equations 4.10-4.12, two extremes can be observed. If α approaches 90°, the cylinder force approaches 0. Consequently, the cylinder needs to exert minimal force to achieve the desired clamping force. This scenario presents a disadvantages. Namely, the tension within the rod and supports becomes very high, requiring their design to withstand these forces.

If α approaches 0°, the required cylinder force becomes infinitely high. A higher cylinder force necessitates a larger cylinder, complicating its integration into the tensioner and increasing the bending moment in the rod. For the tensioner, a value of 60° was chosen for α, striking a balance between both extremes.

Based on the total required clamping force discussed in subsection 4.3.2, the required cylinder force is $F_{cylinder} = 288$kN. For a working pressure of 270 bar, this necessitates a hydraulic cylinder with a diameter of at least 117 mm.

4.3.4. Detailing of drive system

The drive system, comprising the motor, gearbox, and brakes, ensures that the tensioner can generate the desired tension at the specified speed. Therefore, it is crucial to properly dimension the drive system. For this design the decision was made to employ a single motor with a single gearbox and single brake. A combination of multiple smaller motors and gearboxes could also be used, this would not fundamentally change the design of the tensioner.

The drive system was detailed for scenario 1, as the tensioner has the highest requirements for this scenario. In scenario 1, the tensioner is required to lower the lower block. It is important for Huisman and its clients that the tensioner can perform this operation at high speed. Tensioning the wire rope during hoisting to achieve *Confidential*% MBL of tension at the drum is permitted to occur at low speed. This results in two scenarios that the drive system must accommodate. The first scenario involves generating 99.9 kN at 0.96 m/s during hoisting. The second scenario involves lowering at 2.88 m/s while generating a tension of 9.9 kN. This speed is also refered to as the line speed.

Firstly the tension in the chain caused by the clamps exerting friction on the wire rope and the friction of the rollers was determined. Equation 4.13 shows this, in this equation $F_{WR,friction}$ is the tension generation goal as mentioned above. The frictional component consists of two parts. Firstly the friction in the rollers which support the chain caused by the weight of the chain. Secondly the friction in the clamp rollers caused by the rollers of the clamps rolling along the guide rail.

$$F_{WR} = F_{WR,friction} + m_{chain} g * \eta_{rollers} + \frac{F_{WR,friction}}{\mu_{clamp}} * \eta_{rolllers} \tag{4.13}$$

The tension in the return, or non active, side of the chain at the sprocket is described by Equation 4.14. Equation 4.15 is used to calculate the tension on the active side of the chain.

$$F_{return} = F_{pretension} + \frac{m_{chain} g}{2} \tag{4.14}$$

$$F_{active} = F_{return} + F_{WR} \tag{4.15}$$

The friction in the system caused by the chain support rollers and clamp rollers was already considered in Equation 4.13. There is however an additional friction component that needs to be considered. The friction in the link bearings (Equation 4.16). This is the friction caused by the rotation of the link bearings when the chain revolves around the sprocket.

$$T_{\eta,link} = (F_{return} + F_{active}) * \eta_{linkbearing} * \frac{d_{link}}{2} \tag{4.16}$$

The total torque on the sprocket can now be calculated using Equation 4.17.

$$T_{sprocket} = F_{WR} * r_{sprocket} + T_{\eta,link} \tag{4.17}$$

With the required torque on the sprocket determined, the next step is to consider the gearbox and motor. Selecting a suitable gearbox and motor can be an iterative process. Some characteristics of the motor are already known up front. Firstly, the motor power should be approximately 100 kW, based on the tension generation and speed requirements. Secondly, it is known that the maximum speed of the motor should be three times the nominal speed, as is the case for the motor of the main drum. For the motors used by Huisman, this means a nominal speed of 1000 rpm with a maximum speed of 3000 rpm. Based on this information a motor that can be used as starting point can be selected.

For the gearbox, a starting point for the selection can be found by determining the required transmission range and the required rated torque. This required rated torque is equal to the sprocket torque determined using Equation 4.17. The transmission range can be determined using Equation 4.18 and 4.19. The transmission of the gearbox should fall within this range. If the transmission is larger, the tensioner will not reach the required speed. If the transmission is lower, the required torque will not be achieved.

$$i_{torque} = T_{sprocket}/T_{motor} \tag{4.18}$$

$$i_{speed} = \omega_{motor}/\frac{linespeed}{r_{sprocket}} \tag{4.19}$$

With a first selection of both the motor and gearbox a number of calculations can be performed to determine whether the selected motor and gearbox are sufficient. Equation 4.20 calculates the required motor speed based on the selected gearbox. With this the maximum torque of the motor at this speed can be determined using Equation 4.21. The actual torque the motor experiences can be calculated using Equation 4.22. In this equation η is the efficiency of a single gearbox stage and k is the number of gearbox stages. Using Equation 4.23 the nominal power of the motor is determined.

$$\omega_{motor,required} = \frac{linespeed}{r_{sprocket}} * i \tag{4.20}$$

$$T_{motor,max} = \frac{P_{motor}}{max\{\omega_{motor}, \omega_{motor,required}\}} \tag{4.21}$$

$$T_{motor} = \frac{T_{sprocket}/\eta^k}{i} \tag{4.22}$$

$$P_{nominal} = T_{motor} * \omega_{motor,required} \tag{4.23}$$

To quickly assess whether a combination of motor and gearbox is suitable for use in the tensioner, a few checks have been performed. These checks determine the safety factor between critical values. A safety factor of 1 indicates that the requirements are exactly met. Higher factors suggest that the motor and gearbox are over-dimensioned, providing a safety margin. A factor lower than 1 indicates that the drive system is not properly selected. With the safety factors determined using the equations below, the motor and gearbox can be selected by testing different combinations.

$$SF_{torque,gearbox} = T_{gearbox,rated}/T_{sprocket} \tag{4.24}$$

$$SF_{torque,motor} = T_{motor,max}/T_{motor} \tag{4.25}$$

$$SF_{power} = P_{motor}/P_{nominal} \tag{4.26}$$

$$SF_{speed} = \omega_{motor}/\omega_{motor,required} \tag{4.27}$$

For the tensioner employed in scenario 1, the following motor and gearbox were selected using the selection process detailed above:

Cooling

One aspect often overlooked in the design of a drive system is the temperature of the oil in the gearbox. To ensure the desired lifespan of the gearbox oil, it must remain below certain temperatures. The maximum allowable temperature for mineral oil is 100 °C, with an ideal operating temperature below 60 °C. For synthetic oil, these values are 120 °C and 80 °C, respectively [80]. To determine whether cooling of the gearbox oil is necessary, the temperature increase of the gearbox oil is calculated.

Table 4.2: Specifications of the selected motor and gearbox

Motor		**Gearbox**	
Rated power:	132 kW	Rated torque:	42 kNm
Nominal speed:	1000 rpm	Transmission:	31.47
Maximum speed:	3000 rpm	Number of stages:	2
Rated torque:	1261 Nm	Efficiency per stage:	98%

In collaboration with its gearbox suppliers, Huisman has developed a model that approximates the oil temperature in the gearbox [81]. The model is presented in the equations below.

$$M_{loss} = (1 - \eta^k_{stage,mech})T_{drive} + M_{nom}\frac{n}{3400}(1 - \eta^k_{stage,speed}) \tag{4.28}$$

Equation 4.28 calculates the mechanical loss in the gearbox. This mechanical loss is not only a function of torque and efficiency, as often seen in simplified mechanical loss calculations, but also of the operating speed of the gearbox. Since the mechanical loss depends on both the driving torque T_{drive} and the driving speed n independently, the oil temperature can increase rapidly even under low loads.

The mechanical loss calculated in Equation 4.28 is then converted into the mechanical power lost as shown in Equation 4.29.

$$P_{loss} = M_{loss}\frac{2n\pi}{60} \tag{4.29}$$

Equation 4.30 shows the final calculation of the gearbox oil temperature. Based on the mechanical power loss P_{loss}, cooling capacity Q_{cool} and the following gearbox properties: Gearbox heat conductivity λ, outer area A, gearbox mass M and specific heat of the gearbox c.

$$T(t) = T_0 + \frac{P_{loss} - Q_{cool}}{\lambda A}(1 - e^{\frac{-\lambda A}{Mc}t}) \tag{4.30}$$

These equations were used to model the heat generation in the gearbox oil for two situations. Firstly, the lowering of an empty lower block in scenario 1 at high speed. Secondly is the hoisting of an empty lower block at nominal speed, while using the tensioner to achieve a tension of *Confidential*% MBL at the drum. In both situations the cooler power was set to 0. The result is presented in Figure 4.9.

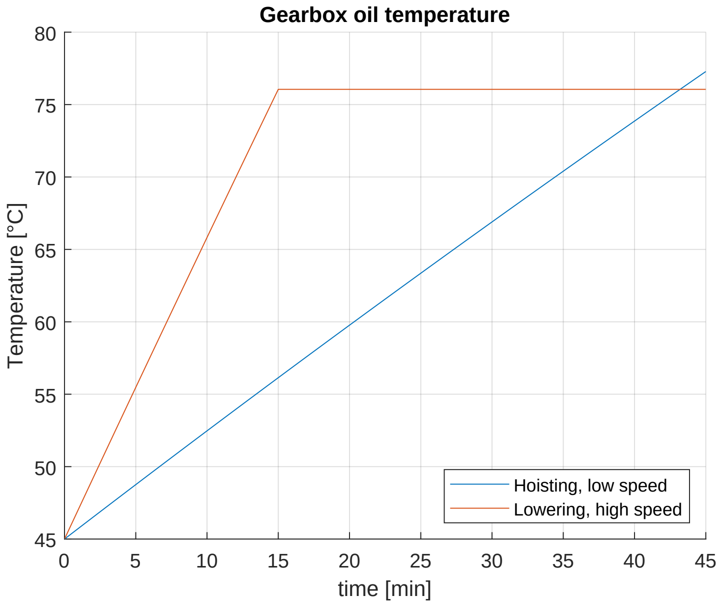


Figure 4.9: Temperature increase of the gearbox oil in the tensioner during high speed lowering and low speed hoisting

Lowering an empty lower block on the 1600 mt LEC takes 15 minutes, during which the temperature increases by 31 °C. The maximum environmental temperature considered by Huisman is 45 °C. Consequently, a single lowering operation will raise the gearbox oil temperature to 76 °C. A hoisting operation at low speed takes 45 minutes, during which the oil temperature increases to 77 °C. These values fall within the allowable ranges for oil temperature. However, a cooler will still be required, as long wait times after each hoisting or lowering operation are not acceptable.

Because the temperature increase during high-speed lowering occurs more rapidly than during low-speed hoisting, a larger cooler is required for high-speed lowering to maintain low temperatures. For example, a 3 kW cooler would reduce the maximum temperature during hoisting to 60 °C, while the temperature during lowering would still rise to 70 °C.

Brakes

The primary function of the tensioner is to increase the tension in the wire rope during hoisting. To achieve this, the chain clamped to the wire rope must be braked. During nominal operation, this is accomplished by the motor of the drive system. In the event of an emergency stop where the motor cannot be used, a mechanical brake is required to take over the braking action. This is critical for scenario 1, as the lower block would move uncontrollably upwards if the tensioner lacks a brake.

Several aspects should be considered during the selection and dimensioning of the brakes. Firstly, the position of the brakes is crucial. The brakes can be located behind the motor, between the motor and the gearbox, or directly on the sprocket. Huisman prefers brakes mounted behind the motor, as this location offers several advantages: ease of installation and maintenance, good protection from environmental factors and dirt, and a simple design [82].

For brakes installed behind a motor and used during emergency stops, negative caliper brakes are often employed. In this type of brake, a brake disc is connected to the axis running through the motor to the gearbox. One or multiple brake units can be installed on this brake disc. Each brake unit consists of a spring that pushes a brake piston onto the brake disc. The friction between the brake piston and the brake disc generates the braking torque. To release the brake, the brake piston can be moved hydraulically. This ensures that the brake is automatically applied when power, and thus hydraulic pressure, is lost, making a negative brake ideal for emergency stop scenarios. A schematic drawing of this is presented in Figure 4.10.

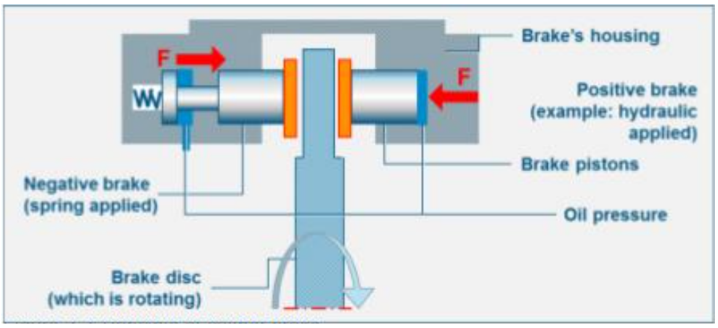


Figure 4.10: Schematic drawing of a caliper brake [82]

The minimal required static brake torque for the caliper brake can easily be determined as this is the same as the torque the motor experiences during the lowering of the lower block as calculated in Equation 4.22. The dynamic torque requirements of the brake are far more complicated. As further explained in subsection 4.5.2, during an category 0 emergency stop all power to the system is cut and the mechanical brake will bring the main drum and tensioner to a stop as quickly as possible. During this process the main drum and tensioner will influence each other. It is important to study this effect as a number of situation could occur that must be minimised or prevented.

Firstly, if the brake torque of the tensioner is sufficiently high, the tensioner could increase the tension in the wire rope beyond its safety limit of 1/3 of the MBL. This must be prevented for safety reasons. Secondly, if the tensioner brakes faster than the main drum, the drum's inertia could pull the wire rope through the tensioner, causing slip between the clamps and the wire rope, potentially damaging the wire rope. This should also be avoided if possible or otherwise minimised. Lastly, if the tension in the wire rope between the tensioner and the drum becomes too low, it can cause slack rope issues and the wire rope running off sheaves. It is also crucial to avoid or mitigate this.

To asses whether these issues occur, a simplified discrete dynamic model was made that models the interaction between the tensioner and the main drum during an emergency stop. This model simplifies the tensioner to a single sheave and the wire rope between the sheave and the drum to a spring. This makes it possible to asses the interaction under different load cases.

Firstly, the inertia of both the drum and the tensioner, including the drive systems, is determined. This is illustrated in the equations below. The inertia of the wire rope will vary depending on whether the drum is full or empty.

$$I_{drum,total} = I_{drum} + I_{wirerope} + (I_{motor} + I_{gearbox} + I_{brake}) * n_{drives} * i_{drum}^2 \tag{4.31}$$

$$I_{tensioner,total} = m_{chain} * r_{sprocket}^2 + (I_{motor} + I_{gearbox} + I_{brake}) * i_{tensioner}^2 \tag{4.32}$$

The wire rope between the main drum and the tensioner is modelled as a spring, as previously mentioned. This approach allows the determination of tension changes in the wire rope during an emergency stop. To determine the stiffness of the wire rope, the Ernst equation was employed [83]. This equation calculates an equivalent stiffness of the wire rope, accounting for the sag caused by its own weight. For a wire rope running between two points, the following equations can be used, where F is the tension in the wire rope determined by the load case, L is the distance between the points, α is the angle of the wire rope relative to the ground, $m_{wirerope}$ is the mass of the wire rope per meter, A is the cross sectional area of the wire rope and E is the modulus of elasticity.

$$k_{sag} = \frac{12F^3}{L(Lcos(\alpha))^2 * (m_{wirerope} * g)^2} \tag{4.33}$$

$$k_{rope} = \frac{EA}{L} \tag{4.34}$$

$$k_{ernst} = \frac{1}{\frac{1}{E_{sag}} + \frac{1}{E_{rope}}} \tag{4.35}$$

Using these equations, the equivalent stiffness for all the wire rope segments between the drum and the tensioner can be determined. Equation 4.36 is used to calculate the total equivalent stiffness.

$$k_{ernst,total} = \frac{1}{\frac{1}{k_{ernst}^1} + \frac{1}{k_{ernst}^2} + \frac{1}{k_{ernst}^3} + \frac{1}{k_{ernst}^4}} \tag{4.36}$$

With the inertia of the tensioner known, the following equations are used to determine the dynamics of the tensioner during an emergency stop. In these equations, F_{WR} represents the tension in the wire rope between the drum and the tensioner, at the tensioner. F_{LP} denotes the tension in the wire rope at the tensioner caused by the load on the crane. If v_0 is positive the crane is hoisting, otherwise the crane is lowering when the emergency stop is triggered, which influences the system's dynamics.

$$\text{if } v_0 > 0: \; T_{tensioner}[t] = T_{brake,tensioner} * i_{tensioner} - (F_{WR}[t] - F_{LP}) * r_{tensioner} \tag{4.37}$$

$$\text{if } v_0 < 0: \; T_{tensioner}[t] = T_{brake,tensioner} * i_{tensioner} + (F_{WR}[t] - F_{LP}) * r_{tensioner} \tag{4.38}$$

$$\alpha_{tensioner}[t] = -T_{tensioner}[t]/I_{tensioner} \tag{4.39}$$

$$\omega_{tensioner}[t] = \omega_{tensioner}[t - \Delta t] + \Delta t * \alpha_{tensioner}[t - \Delta t] \tag{4.40}$$

$$\Delta l_{tensioner}[t] = \Delta t * \omega_{tensioner}[t] * r_{tensioner} + \alpha_{tensioner}[t] * r_{tensioner} * \Delta t^2 \tag{4.41}$$

For the drum, similar equations can be used to calculate its dynamics. However, the equations to calculate the torque on the drum differ from Equation 4.37 and 4.38. In the drum calculations, the tension in the wire rope caused by the lower block is not relevant. Instead, the decrease in wire rope tension due to the wire rope's own mass and sheave efficiencies is. This is illustrated in Equation 4.42

and 4.43. In these equations, dZ represents the vertical distance between the drum and the tensioner, m is the mass of the wire rope per meter, and η denotes the sheave efficiency.

$$\text{if } v_0 > 0: \; T_{drum}[t] = T_{brake,drum} * i_{drum} + \frac{F_{WR}[t] - dZ * m * g}{\eta^3} * r_{drum} \tag{4.42}$$

$$\text{if } v_0 < 0: \; T_{drum}[t] = T_{brake,drum} * i_{drum} - \frac{F_{WR}[t] - dZ * m * g}{\eta^3} * r_{drum} \tag{4.43}$$

The model based on the equations above assumes that the full torque of the brakes is applied instantly when an emergency stop is triggered. In reality, there is a short period during which the brakes close the air gap and do not provide their designed braking torque. To model this, the brake torque for both the drum and tensioner is set to 0 for the first 0.2 seconds. The effect of this delay is clearly visible in Figure 4.11.

The equations above calculate the speed of both the drum and the tensioner at each time step. This allows for the determination of the elongation of the wire rope, which is modelled as a spring. This elongation affects the tension in the wire rope between the tensioner and the drum. The elongation of the wire rope is described in the equations below.

$$\text{if } v_0 > 0: \Delta l[t] = \Delta l_{drum}[t] - \Delta l_{tensioner}[t] \tag{4.44}$$

$$\text{if } v_0 < 0: \Delta l[t] = -\Delta l_{drum}[t] + \Delta l_{tensioner}[t] \tag{4.45}$$

The tension in the wire rope changes as described in Equation 4.46.

$$F_{WR}[t] = F_{WR}[t-1] + \Delta l[t-1] * k_{ernst,total} \tag{4.46}$$

There are three distinct limits to the change in wire rope tension. The first limit is the maximum frictional force of the tensioner. If the difference between the tension caused by the load and the tension between the tensioner and drum exceeds the maximum frictional force of 99.9 kN, the wire rope will slip through the clamps of the tensioner. During this the tension difference is maintained at 99.9 kN, assuming dynamic friction is equal to static friction.

While the wire rope can exert a braking force on the tensioner equal to the maximum frictional force, the tensioner can only exert as much braking force on the wire rope as its brake torque allows. If this torque is lower than the maximum frictional force, it becomes the limiting factor.

The third limit is the minimal required tension in the wire rope. As previously mentioned, it is crucial to maintain tension in the wire rope. In the model, this is ensured by adding Equation 4.47. Physically, this could be achieved by adding an extra free-floating sheave to the wire rope with a weight attached to it. This free-floating sheave takes up the wire rope to prevent slack and maintain minimal tension. The wire rope length that needs to be taken up by this free-floating sheave can be determined using Equation 4.48. A minimum tension value of 8 kN was chosen.

$$\text{if } F_{WR}[t] < 8\text{kN}: F_{WR}[t] = 8\text{kN} \tag{4.47}$$

$$\text{if } F_{WR}[t] \leq 8\text{kN}: s[t] = |\Delta l[t]| \tag{4.48}$$

The model described above can be used to test different load cases, facilitating the dimensioning of the tensioner's brake. The load cases consist of all combinations of the variables listed below, resulting in a total of 64 load cases. Appendix B provides an overview of all the tested load cases and a visualization for each. A visualisation of the results from the model for a load case is shown in Figure 4.11. This shows the interaction between the tensioner and the drum caused by the change in the wire rope tension.

- High speed or low speed
- Hoisting or lowering

- No load or maximum load
- Full drum or empty drum
- Maximum or minimum drum brake torque
- Maximum or minimum tensioner brake torque

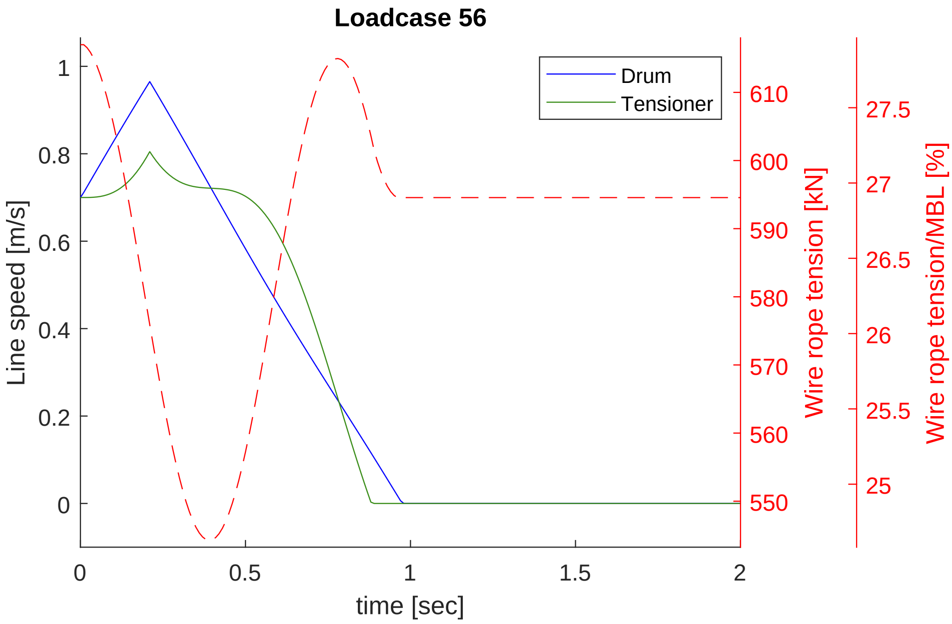


Figure 4.11: Dynamics of the drum and tensioner during an emergency stop

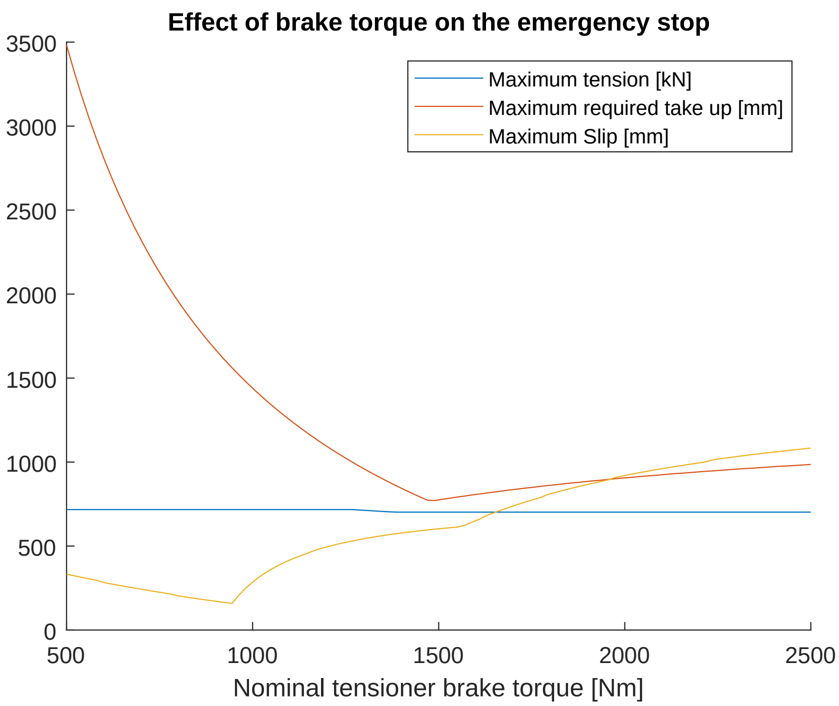


Figure 4.12: Result of dynamic analysis used to determine the optimal brake torque

To determine the optimal brake torque for the tensioner, the results from various load cases are analysed. From all 64 load cases, the maximum tension in the wire rope, the maximum slip, and the maximum wire rope length requiring take-up are extracted. Using the results shown in Figure 4.12, the tensioner's brake can be dimensioned to either minimise the maximum slip or reduce the maximum required take-up. The results indicate that slip and wire rope take-up are inevitable in some load cases, while it would be preferable to prevent both slip and wire rope take-up. A device capable of taking up slack wire rope is therefore necessary. Using a 944 Nm brake, which has a slightly lower torque than the motor's nominal operating torque of 1008 Nm, results in the lowest maximum slip across all load cases.

This simplified dynamic model can be used to illustrate the interaction between the tensioner and the drum during an emergency stop and to dimension the tensioner's brake. However, it remains a simplified model. One of the simplifications is the assumption of constant braking torque, whereas in reality, it can vary based on the brake's speed and environmental influences. Secondly, the inertia of the load is excluded from the model. While the tension loss due to the wire rope's own weight and sheave efficiency is considered, it is done so in a simplified manner compared to the parametric model discussed earlier. Lastly, it is assumed that a free-floating sheave can maintain a constant minimum tension, but the dynamics of this free-floating sheave are not modelled.

4.4. Improvement of load curve

One of the promising functions of the active wire rope tensioner is its ability to facilitate the use of a lighter lower block. The implementation of a lighter lower block enhances the crane's hoisting capacity. The hoisting capacity of a crane is characterized by its load curve, also known as load charts. This load curve determines the maximum SWL for a given boom outreach, with the SWL decreasing as the outreach increases.

The load curve of a crane is derived from the combination of load curves specific to its individual components. Components such as the hoisting system, slew bearing, and boom pivots each possess their own load limits, resulting in unique load curves for each component. The overall load curve of the crane is obtained by combining these component specific load curves. This is achieved by identifying the lowest maximum SWL for each component at every increment of boom outreach. The component with the lowest SWL at a given outreach increment becomes the limiting factor, thereby determining the crane's maximum SWL at that outreach.

In both scenarios 1 and 2, the reduction in lower block weight will influence the crane's load curve. In scenario 2, the lower block weight decreases by 20%. Since the tensioner is placed in the luffing frame in this scenario, its weight does not affect the load curve. In scenario 1, the lower block weight is reduced by 40%. Here, the tensioner is placed on the boom, meaning its weight could impact the load curve. Specifically, the addition of the tensioner increases the boom's total mass and shifts its centre of gravity (COG). Equation 4.49 shows how the addition of the tensioner shifts the booms COG. For scenario 1 the total mass of the boom increases from 515 t to 518 t and the COG moves from 81.5 m to 81.3 m.

$$m_{boom} * r_{COG,boom} + m_{tensioner} * r_{COG,tensioner} = (m_{boom} + m_{tensioner}) * r_{COG,new} \tag{4.49}$$

Figure 4.13 illustrates the resulting load curves, showing an expected increase in the maximum SWL with the reduction in lower block weight. The horizontal segment of the load curve is constrained by the crane's hoisting system, which lifts both the load and the lower block. Therefore, reducing the lower block weight directly enhances the SWL of the hoisting system. In the non-linear portion of the load curve, the reduction in lower block weight similarly results in an increased SWL.

In scenario 1, the addition of the tensioner increased the boom's mass and shifted its COG. This could potentially negatively impact the crane's load curve. Whether this is the case is determined by the limiting load curve. Evaluating the component specific load curves for the 1600 mt LEC reveals that the addition of the tensioner does not change the limiting load curve. However, if the tensioner is applied to a different crane, this impact should be carefully evaluated, as it may vary with different crane configurations.

The load curves for the two scenarios and the current situation closely follow each other. To better visualize the improvement in the load curve, this enhancement was plotted. Figure 4.14a illustrates

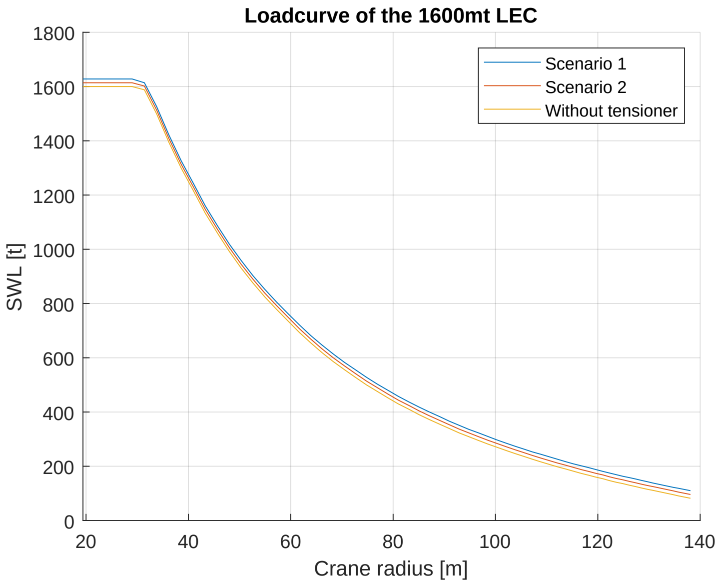
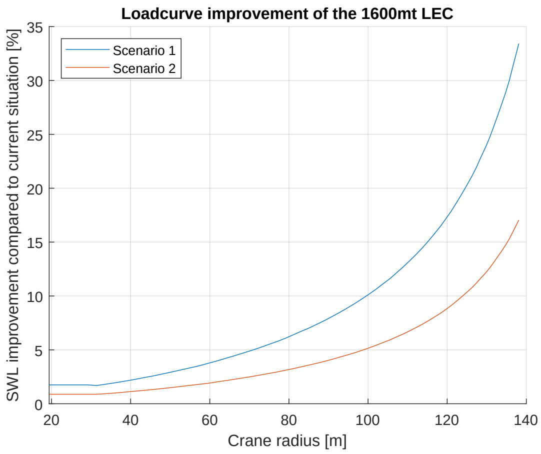


Figure 4.13: Load curves for the 1600 mt LEC under different tensioner employment scenarios

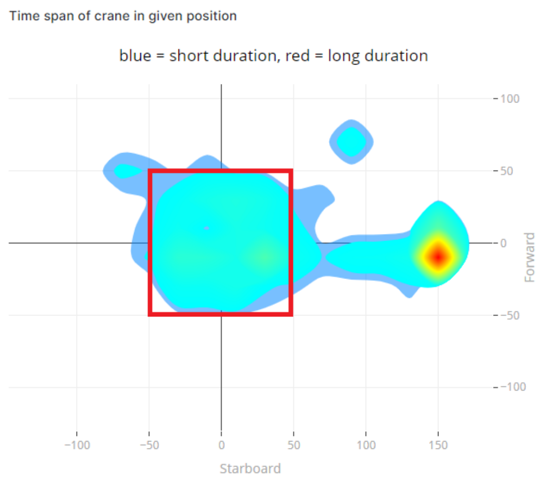
the improvement in the load curve as the percentage increase in SWL against the crane radius. It is evident that the tensioners have the most significant effect on the load curve when the crane's outreach is at its maximum.

In scenario 1, the load curve improvement reaches 33% at the maximum outreach. While this increase appears highly beneficial at first glance, it should be contextualized. The plot in Figure 4.14b illustrates the usage data of the 1600 mt LEC. The colour gradient from light blue to red indicates the amount of time the crane spends in specific positions. The red mark on the right side of the plot represents the crane's position when it is rested in the boom rest.

This data clearly shows that the crane is almost exclusively operated within an outreach of 50 meters, as indicated by the red box. In this region the load curve improvement is limited to 3% for scenario 1 and 1.5% for scenario 2



(a)



(b)

Figure 4.14: a)Relative improvement of the load curve. b) Overview of the outreach under which the crane is used.

4.5. Control system

To fulfil its intended purpose, the clamping track requires a robust control system. This control system is crucial not only during normal operation but also for detecting and responding to unexpected events. This section will describe the control system, the necessary sensors, and the emergency stop philosophy of the system.

One of the advantages of the clamping track concept is that it can be completely removed from the rigging, thereby not influencing the crane's performance. The crane control system distinguishes three main states for the tensioner: disengaged, inactive, and active. The control system can assist the crane operator in selecting the appropriate state or can automatically make this selection.

When the tensioner is disengaged, the guide rails are retracted, and the system is completely removed from the rigging. In the inactive state, the guide rails are in their operating position, causing the clamps to grab the wire rope, but the drive runs freely without active control. This state is advantageous when the operator wants to temporarily disable the system without waiting for the (dis)engaging process.

In the active state, the system is engaged with the wire rope, and its control system is active. In this state, it will control the tension in the wire rope to achieve the desired tension.

System disengaged or inactive

While the system is disengaged or inactive the control system has a number of key objectives. These are listed below:

- **Monitor tension during hoisting:** If the tension in the wire rope at the drum falls below the target of *Confidential*% MBL, the control system should advise the crane operator to activate the system. It could also suggest partly unwinding the drum first to ensure the entire wire rope is spooled with sufficient tension. These actions could be automated if desired.
- **Monitor tension during lowering:** If the lower block weight is reduced, the tensioning device is required to ensure proper lowering of the lower block. The control system should monitor the tension in the wire rope and advise the operator to activate the system when the tension reaches critically low values. If the tension becomes too low, the lower block will not lower properly and could even move uncontrollably upwards.
- **Slack rope detection:** It is important that the system does not engage if there is slack rope in the system. If there is slack rope in the system the wire rope is not properly positioned between the clamps. Engaging of the clamping track would damage the wire rope, which should be prevented. The control system can achieve this by detecting slack rope and preventing engaging when slack rope is detected.

System active

When the system is active the system has the following control goals:

- **Control tension during hoisting:** The primary objective of the tensioning system is to ensure that the entire drum is wound under a constant tension of at least *Confidential*% MBL. To achieve this, the control system must regulate the motor speed to match the speed of the wire rope and control the torque of the motor to achieve the desired wire rope tension.

  The control system must not only maintain a minimum tension during hoisting but also ensure that the wire rope tension does not exceed safety limits. When the tensioner is no longer required to maintain the target of *Confidential*% MBL, the control system should notify the operator, who can then decide to disable or disengage the system. It is crucial for safety that the tensioning device never increases the tension in the wire rope beyond its SWL.
- **Control tension during lowering:** Whether the tensioning system is required during lowering is dependent on the lower block weight. If the lower block weight is reduced below the current weight the tension system is needed to ensure proper lowering.

  If this is the case the control system must control the tension to ensure it reaches the target value of 5 kN at the drum. This should be done while also controlling the speed of the clamping track to follow the desired speed of the wire rope. If the tensioner moves to slow slack rope could occur, moving to fast results in slip between the wire rope and clamps.

When the lower block weight is not reduced the tensioner has no function during the lowering process. The control system should notify the operator that it is safe to disable or disengage the system.

Schematic overview

A schematic overview of the control system is shown below in Figure 4.15. This figure illustrates the proposed sensors and actuators and their communication within the crane and tensioner control system. subsection 4.5.1 provides a more detailed discussion of the possible sensors.

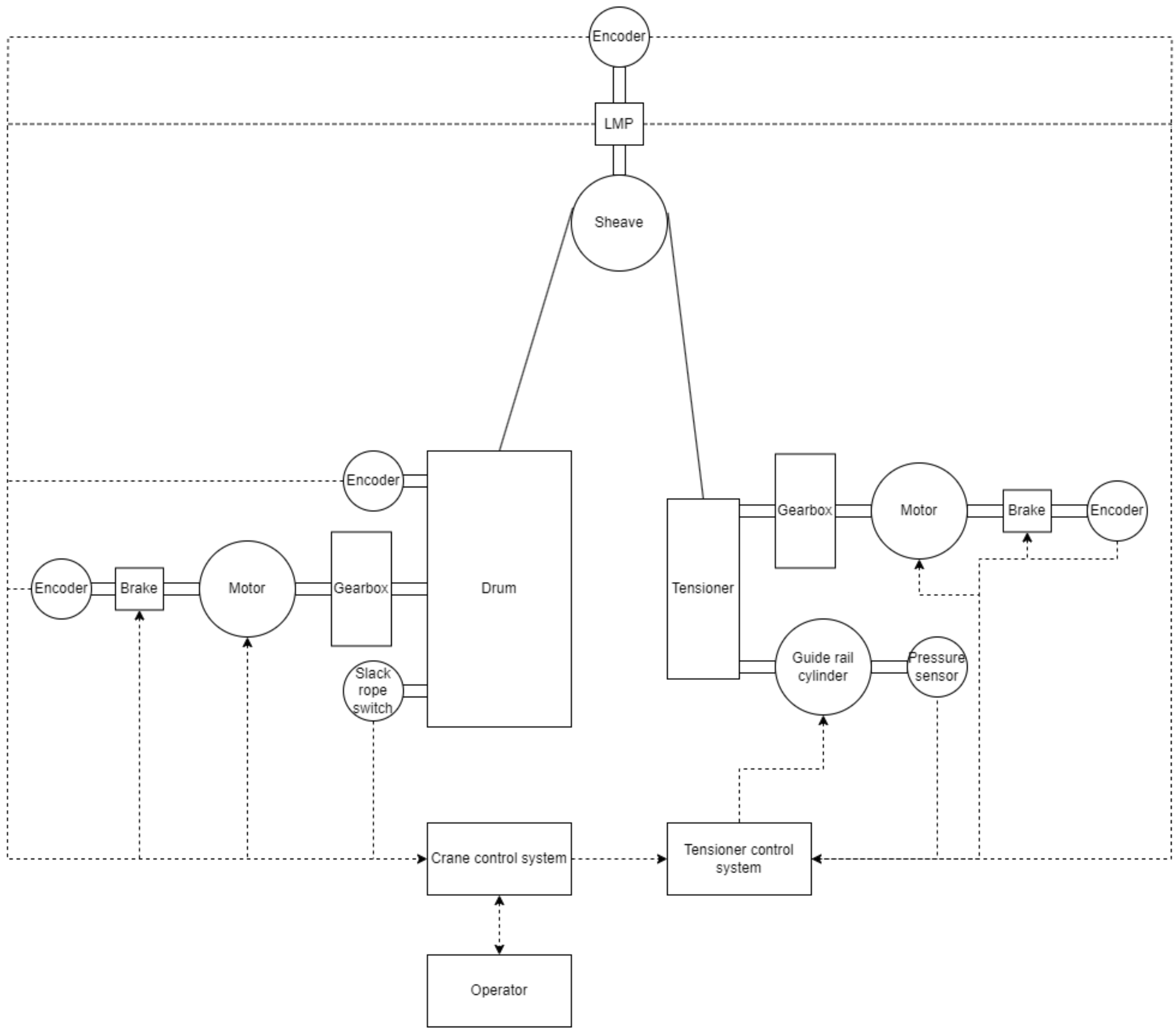


Figure 4.15: Schematic overview of the control system

4.5.1. Sensors

To achieve the control objectives, the control system requires knowledges of certain parameters. These parameters will be measured using sensors. Below is a list of the required parameters and the sensors that could be used to achieve this:

- **Torque in the wire rope at the drum:** Measuring the tension in the wire rope is often done with load measuring pins (LMP). These are hollow pins with strain gauges mounted inside of them, used to determine the load acting on the pin. If such a LMP is used as the shaft of a sheave, the tension in the wire rope can be determined based on the load on the LMP.

  An alternative method is to determine the tension in the wire rope based on the torque of the drum and the radius of current layer of the wire rope. This requires measurement of the torque on the drum and accurate measurement of the amount of layers.

- **Speed of the wire rope:** To ensure that the speed of the tensioner track matches the speed of the wire rope, it is crucial to accurately measure the wire rope's speed. A simple and reliable method involves attaching an encoder to a sheave. By knowing the radius of the sheave, the linear speed of the wire rope can be determined.

  If the current number of layers on the drum is accurately known, the rotational speed of the drum can also be used to calculate the linear speed of the wire rope. Utilizing an absolute multiturn encoder on the drum allows for the measurement of the drum's rotational speed and the determination of the current number of layers.

  If the tensioner track is placed in the boom and is used in combination with a light lower block, scenario 1, it is possible that the lower block moves upwards uncontrollable if the tensioner track fails. To detect this an encoder can be added to sheaves in the upper block to measure a mismatch between the sheave speeds.

- **Speed of the tensioner track:** In order to prevent slip between the wire rope and the clamps of the clamping track it is important to accurately control the speed of the track. Measuring the speed of the track can be achieved by connecting an encoder to one of the sprockets or drives connected to the sprocket.

- **Slack rope detection:** The main drum of a crane is often equipped with a slack rope switch. This is a mechanism consisting of a bar with a counterweight mounted to it. If the wire rope becomes slack it presses down on the bar, lifting the counterweight. This movement of the counterweight is measured by a proximity sensor.

- **Hydraulic pressure:** The hydraulic cylinders responsible for moving and pressurising the guide rails, as well as tensioning the track, should be equipped with pressure sensors to regulate the applied force. Huisman employs two types of pressure sensors. The first type utilizes strain gauges to measure the deflection of a thin film, offering a robust and cost-effective solution. However, more precise measurements can be achieved with piezoelectric pressure sensors, which detect voltage changes induced by pressure variations on the piezoelectric material. Although these sensors provide higher accuracy compared to strain gauge sensors, they are also more expensive [84].

4.5.2. E-stop philosophy

Safety is of paramount importance to Huisman and its clients. Therefore, it is crucial to consider the behaviour of the tensioner during emergency stops (E-stops). There are two distinct E-stop categories [85].

The first category is the Category 0 stop. A CAT-0 stop occurs in the event of a system or critical component failure, rendering it impossible to control the crane's braking. When a CAT-0 stop is triggered, all power to the system is cut off, resulting in the main brakes being applied to the drum, which brings the crane's movement to a halt [86].

The second category is the Category 1 stop, which is triggered either manually or by the safety systems. During a CAT-1 stop, control of the crane remains possible. The drum is decelerated as quickly as possible using the torque of the motors. After a predefined period, typically two seconds, the mechanical brakes are applied, and power to the drives is removed.

The desired behaviour of the tensioner during these two stop categories depends on the previously mentioned scenarios. In scenario 1, if the lower block weight is insufficient to overcome the weight of the wire rope, the lower block would move uncontrollably upwards without the tensioner. However, if the tensioner is placed in the luffing frame, as in scenarios 2 and 3, this issue does not arise. Consequently, two different desired system behaviours during E-stops are required. These behaviours are elaborated below.

Scenario 1
During an E-stop in scenario 1, it is crucial that the tensioner engages and brakes to prevent upwards movement of the lower block. The tensioner must have sufficient traction and braking capability to support the weight of the wire rope between the tensioner and the drum. Failure to do so could result in the lower block moving upwards. Therefore, the system must be equipped with brakes that activate upon a loss of power. A similar mechanism should be applied to the guide rails to ensure that the clamping force remains when the system loses power.

In a CAT-0 stop for scenario 1, the tensioner and drum brake as quickly as possible until they come to rest. The dynamics of this process have been modelled and discussed in subsection 4.3.4. Using this dynamic analysis, the brakes should be properly dimensioned. Proper dimensioning ensures that the maximum tension in the wire rope does not exceed its safety limits, slip is minimised, and slack rope is minimised and mitigated with a free-floating take-up sheave where necessary.

In a CAT-1 stop for scenario 1, the guide rails engage the clamps on the wire rope, similar to a CAT-0 stop. Since system control is still possible, the tensioner control system should brake the tensioner using the motor, while attempting to match the speed of the wire rope to prevent slack. Once the drum stops or the two-second window elapses, the tensioner brakes should be applied. If the tensioner cannot match the braking speed of the drum because the drum brakes faster, the free-floating take-up sheave is required to mitigate slack rope in the system.

Scenario 2 & 3
In scenarios 2 or 3, the tensioner is not required to prevent uncontrollable upward motion of the lower block. Therefore, it is desirable to disengage the clamping track during an emergency stop. Disengaging the system simplifies the E-stop process of the crane by removing the tensioner's inertia from the system.

Disengaging the tensioner during an E-stop can be achieved using a single-acting hydraulic cylinder. In such a cylinder, retraction is accomplished by a spring, similar to negative caliper brakes. If the oil pressure is removed during the emergency stop, the spring will force open the clamping mechanism, effectively removing the tensioner from the rigging.

It is realistic to assume that the retraction of the cylinder will not occur instantaneously. Consequently, the clamps will remain engaged to the wire rope for a certain period. While the cylinder's movement is not immediate, the loss of hydraulic pressure is nearly instantaneous [87]. As a result, the impact of the tensioner on the emergency stop of the hoist system is negligible. However, slip between the wire rope and clamps under low pressure may occur for a brief duration.

To bring the tensioner to a safe stop, a brake is still required. However, dimensioning this brake is less critical, as it does not affect the E-stop behaviour of the hoist system once the tensioner is disengaged.

4.6. Discussion

The detailed design of the clamping track concept described in this chapter has yielded a conceptual design for the optimal active wire rope tensioner. At the start of the detailed design phase three different employment scenarios where determined. These scenarios have their own advantages and disadvantages, but also require slightly different versions of the active wire rope tensioner. This section presents the final detailed design of active wire rope tensioner and highlights the differences between the employment scenarios.

Figure 4.16 presents a visualization of the active wire rope tensioner concept. This concept increases the tension in the wire rope by exerting a frictional force on it. It achieves this by clamping the wire rope along its length with multiple clamps. These clamps are connected to a chain, which is braked by a motor during hoisting and can be driven by the motor during lowering. The clamping action is facilitated by a hydraulic cylinder moving two guide rails. Rollers connected to the clamps roll along these guide rails, transferring the clamping force. A different set of guide rails ensures the clamps release the wire rope in time.

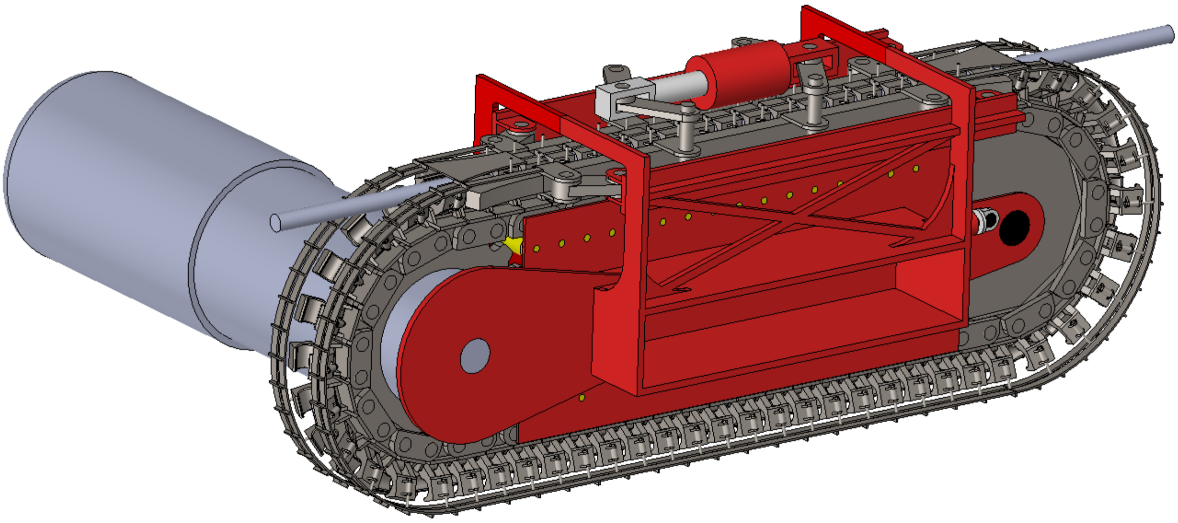


Figure 4.16: CAD model of the Active Wire Rope Tensioner. Working principle sketch for scenario 1

Table 4.3 presents the dimensions of the tensioner. A weight estimation of the tensioner is achieved by calculating the weight of all modelled parts using CAD software, combined with the known weight of the gearbox and motor. The result of this weight estimation is presented in Table 4.4. Adding the weights for the subcomponents yields a total weight of 3 tons.

Table 4.3: Dimensions of the Active Wire Rope Tensioner

Length	Height	Width excluding motor	Width including motor
2.8 m	1.2 m	0.7 m	2.0 m

Table 4.4: Weight estimation of the tensioner components

Motor	Gearbox	Chain	Other
765 kg	375 kg	500 kg	1360 kg

The general description of the concept provided above applies to all scenarios, but there are differences in the design of the tensioner based on the specific scenario. Table 4.5 presents an overview of various specifications of the tensioner for each of the three scenarios. From these specifications, it is evident that reducing the lower block weight, as suggested in scenarios 1 and 2, impacts the specifications. More tension generation is required with a lighter lower block, necessitating a higher clamping force. This results in a larger clamping cylinder and more clamps to distribute this higher clamping force, thereby increasing the length of the tensioner. Additionally, a larger drive system is required.

While these differences are not negligible, they are relatively small. More significant differences between the scenarios can be found in the control and safety aspects.

In scenario 3, the lower block weight is not reduced, and the tensioner's sole purpose is to increase the tension in the wire rope to *Confidential*% MBL during hoisting. This means that the system can be retracted most of the time and is only activated during empty hoisting. If the weight of the lower block is not reduced, the crane will still function as it currently does if the tensioner does not function due to

Table 4.5: Overview of the system specifications for the three scenarios

For the different scenarios		Scenario 1	Scenario 2	Scenario 3
Lower block mass	[mt]	40.8	54.4	68
During hoisting				
Tension generation	[kN]	99.9	95.3	88.6
Clamp force	[kN]	999	953	886
Nominal motor torque	[Nm]	1008	962	895
Nominal motor power	[kW]	101	97	90
During lowering				
Tension generation	[kN]	9.9	4.3	0
Nominal motor torque	[Nm]	107	51.5	0
Nominal motor power	[kW]	32	15.5	0
Tension at the drum during hoisting with empty block				
Without tensioner	[kN]	7.4 (0.33% MBL)	14.0 (0.54% MBL)	20.8 (0.94% MBL)
With tensioner	[kN]	*Confidential*% MBL	*Confidential*% MBL	*Confidential*% MBL

a malfunction. This scenario also has the advantage of allowing the use of a small gearbox cooler and having a negligible effect on the emergency stop process of the crane by removing it from the rigging during an emergency stop. The installation in the luffing frames has the benefit of easy access and it does not add any mass to the boom.

Scenario 2 introduces some complexities while retaining some advantages of scenario 3. In scenario 2, the tensioner is required to lower the lower block, but the lower block weight remains high enough to prevent uncontrollable upward motion. Consequently, the crane will not function if the tensioner fails, but no significant safety risk is posed. In this scenario, the tensioner can also be disengaged during an emergency stop, but a larger cooler will be necessary to allow high-speed operation. The main advantage of this scenario is the 20% reduction in lower block weight, resulting in an effective load curve improvement of 1.5% for the 1600 mt LEC.

Scenario 1 maximizes the advantages of decreasing the lower block weight but also introduces the greatest risks and complexities. Reducing the lower block weight by 40% results in an effective load curve improvement of 3%. This improvement could be further enhanced with additional lower block weight reduction. However, with such a light lower block, uncontrollable upward motion will occur if the tensioner fails, posing a significant safety and operational risk. Since the tensioner is required during an emergency stop to prevent upward motion of the lower block, dynamic interaction between the drum and tensioner during this emergency stop is inevitable. This interaction can cause slack rope issues, which need to be mitigated with the addition of a free-floating sheave or a similar mechanism.

All three scenarios achieve the primary design goal of increasing the wire rope tension during hoisting to *Confidential*% MBL. This reduces wear on the wire rope and is believed to prevent cutting-in. As discussed in subsection 1.1.1, this can result in an 85% improvement in the wire rope's service life. The optimal scenario is determined by the trade-off between load curve improvement and scenario-specific complexity.

5 Conclusion

5.1. Conclusion

To conclude, this thesis aimed to answer the research questions which led to achieving the research objective: *Design an active wire rope tensioner system to reduce wire rope damage and enable the usage of lighter lower blocks.* Based on the performed literature review seven concepts were created. Using an assessment of both the requirements and the KPIs the most promising concept was selected. Steps towards the detailed design of the clamping track have been performed, resulting in a conceptual design of the active wire rope tensioner.

The first research question, *"What are the requirements and KPIs for a successful system?"*, was addressed in chapter 1. As the active wire rope tensioner will augment an existing crane by enhancing its current functionality rather than adding new features, it is imperative to establish clear requirements and KPIs. Specifically, the tensioner aims to minimise wire rope damage and enable the use of a lighter lower block. To achieve this a discussions was conducted with multiple stakeholders within the company. An active wire rope tensioner must increase the tension to *Confidential*% MBL during hoisting, without increasing the weight of the lower block or causing damage to the wire rope. It is crucial that the system operates effectively across various types of cranes and rigging configurations, without introducing unacceptable safety risks. The optimal active wire rope tensioner is able to do this at high speed and without altering existing operational procedures procedures. Decreasing the required lower block weight as much as possible is also preferable.

Chapter 2 addresses the research question, *"How can the tension in the wire rope be manipulated?"* This chapter's literature and patent study investigates three potential methods. Firstly, the normal force in a tensioner can be increased by employing various clamping techniques, such as clamping the wire rope on sheaves with belts or rollers. Secondly, enhancing the coefficient of friction can be effective, although it is limited by maximum contact pressures and the potential for increased wire rope wear. Lastly, the study explores different techniques to increase the wrap angle for sheave-based traction devices.

The third research question, *"Which concept has the most potential based on the requirements and criteria?"*, is addressed in chapter 3. Initially, seven concepts were developed based on the literature study and innovative, 'out of the box' solutions. The workings of these concepts, along with their strengths, weaknesses, opportunities, and threats, are described. An assessment of the requirements for each concept resulted in five viable options. Using the KPIs, the most promising concept was identified: the clamping track concept. This selection is attributed to its superior tension generation capability, which is easily scalable. Additionally, it can be completely removed from the rigging, allowing uninfluenced crane operations.

This thesis presents a conceptual design for an active wire rope tensioner, which can be utilised for two purposes depending on the application scenarios. The primary design goal, achieving increased wire rope tension during hoisting to reduce wire rope damage and prevent cutting-in, is successfully met in each scenario. Previous studies show this can increase the wire rope lifetime up to 85%. However, the scenarios differ in their performance concerning the secondary design goal of reducing the required lower block weight. Reducing the lower block weight introduces certain disadvantages and limitations, necessitating the examination of multiple scenarios.

In Scenario 1, the use of the tensioner in a crane results in a 3% improvement in the effective load curve. However, it is more complex in design and usage compared to Concepts 2 and 3 and presents higher risks. Scenario 3 offers the simplest and safest design but does not allow a reduction of the required lower block weight. Concept 2 strikes a balance between the two, achieving a slight improvement in the load curve with some additional complications and risks.

The research presented in this thesis aimed to narrow down the wide array of possibilities to identify the optimal concept for the active wire rope tensioner. The conceptual design developed serves to increase confidence in the concept and provides a foundation for further research and development.

5.2. Discussion

This study holds significant scientific relevance in the field of crane operations, as it addresses the critical issue of wire rope damage and cutting-in. Although alternative solutions, such as employing a larger drum or a heavier lower block, are available, these options have drawbacks that render them undesirable. The device developed in this thesis aims to mitigate wire rope damage and cutting-in without adversely affecting crane designs. While this concept has been explored in literature, the principle of using the tensioner to reduce the lower block weight, is a concept not explored in existing literature. The findings of this study indicate that reducing the lower block weight while simultaneously increasing the tension during hoisting is feasible. However, this approach introduces increased complexity and associated risks.

The parametric model employed during the concept selection phase and for determining the system requirements during the detailed design phase is static and simplified. Consequently, inertias and dynamic effects between the drum and the tensioner concepts were not studied. Despite these limitations, the model is sufficiently robust to guide the selection process and determine the requirements during the detailed design phase. Its simplicity facilitates a clear comparison of different concepts, ensuring that the chosen design meets the necessary criteria. A dynamic analysis, which does incorporate the inertias and dynamic effects, was performed to detail the brake of the tensioner.

During the detailed design phase, several assumptions had to be made due to the unavailability of all required information in literature or Huisman internal documents. The primary design uncertainties include whether a tension of *Confidential*% MBL is sufficient to prevent cutting-in, the maximum allowable radial pressure on the wire rope, and the extent to which the lower block weight can be reduced. These uncertainties influence the design of the tensioner. However, their impact is limited, as one of the advantages of the clamping track concept is its scalability, achieved by adjusting the number of clamps. While the uncertainty in lower block weight reduction affects the maximum load curve improvement, the magnitude of this uncertainty is estimated to be approximately 1% for the effective load curve improvement.

The research objective of this study was to design a tensioning system that increases tension during hoisting and allows the use of a lighter lower block. While increasing the tension during hoisting to *Confidential*% MBL was a requirement of the designed concept, the reduction of the lower block weight was not. This decision was made because it was uncertain whether it would be feasible to design a tensioner capable of achieving both objectives.
As previously mentioned, the study demonstrated that achieving this dual objective is feasible, albeit with associated complexities and risks. If Huisman's goal is to have a 'one size fits all' concept that accommodates various client requirements, the proposed concept of the active wire rope tensioner is optimal based on the concept selection. Had the decision regarding the reduction of the lower block weight been made upfront, the concept selection and detailed design could have resulted in a different conceptual design for the active wire rope tensioner. This is because certain concept-specific advantages and disadvantages would have become more or less relevant.

5.3. Recommendation

For research into the active wire rope tensioner to progress, it is imperative to make a decision regarding the reduction of the lower block weight. As mentioned at the end of the discussion, this decision influences both the concept selection and detailed design. Based on the discussion in section 4.6, scenario 1 should not be chosen, as the load curve improvement of only 3% does not outweigh the increased complexity and risk.
Choosing between scenario 2 and scenario 3 is challenging due to them being similar in risk and complexity. However, designing the tensioner for scenario 2 offers a 1.5% improvement in the load curve, making it more advantageous. This dual-functionality of the tensioner in scenario 2, compared to the single function in scenario 3, makes it a more attractive option. Therefore, it is advisable to design the tensioner for scenario 2, paired with a modular lower block. If the lower block is designed to be 20% lighter than current requirements, with the option to add this 20% weight later, clients can switch to scenario 3 by adding weight to the lower block if ever desired.

It is also advised to perform tests to eliminate or reduce several design uncertainties. Specifically, testing the friction between the wire rope and friction materials such as PU and rubber is of interest, as this could decrease the required clamping force and, consequently, the size of the tensioner. This test should also include an assessment of the effect of pressure on the coefficient of friction. This is particularly relevant when determining the maximum allowable radial pressure on the wire rope. In this study, a conservative value of 15 MPa was used, although it is conceivable that significantly higher pressures may be permissible, which should be tested. With the results of these tests, an optimal combination of clamping pressure and friction material can be determined to optimize the size of the tensioner.

Constructing and testing a prototype can provide insights into design challenges that were not revealed by the model or whose effects are not fully understood. One such factor is the polygon effect. Although this effect is minimized by using a sprocket with a high number of teeth, it never completely disappears. It is essential to test or model in more detail whether the polygon effect causes undesired behaviour like vibration and noise. If linear speed fluctuations caused by the polygon effect negatively impact the tensioner's operations, it is advisable to design an alternative drive system or explore the possibility of using linear elongation chain links as described in subsection 2.2.1.

If, despite the disadvantages, scenario 1 is still considered and the lower block weight is reduced to the point where uncontrollable upward motion is possible, it is advisable to further detail the dynamic analysis of the emergency stop process. While the simplified model presented in this study was appropriate for illustrating the dynamic relationship between the tensioner and the drum and for dimensioning the brake, a more detailed study is desirable due to the critical importance of system safety. A good starting point would be to make the braking torque dynamic as well as adding the device used for wire rope take up to the model.

While the requirements and KPIs are partly based on the opinions of various stakeholders, the remainder of the design process was conducted on a purely technical basis. However, for the active wire rope tensioner to be successfully implemented in the future, it must be sold to a client of Huisman. For potential clients, the economic aspects of the concept are at least as important as the technical aspects. Therefore, it is advisable to develop a business case for the active wire rope tensioner. The capital expenditures and operational expenditures of the tensioners should be compared with the costs saved by increasing the wire rope lifetime.

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A

Research paper

Design of an Active Wire Rope Tensioner
Extending wire rope lifetime in offshore cranes

B. van Empel

Abstract— **This paper presents the design process of an active wire rope tensioner, a device designed for cranes, that serves two primary functions. Firstly, it increases the tension in the wire rope during spooling, ensuring neat and tight winding to prevent damage to the wire rope and cutting-in problems. Secondly, it reduces the required lower block weight, thereby improving the crane's load curve.**

The research has commenced with a literature and patent review, revealing a gap in the state-of-the-art of tensioning devices that both prevent cutting-in and reduce the required lower block weight. In the conceptual design phase, seven innovative concepts have been generated based on the literature review and current principles. These concepts have been assessed for their capability to meet the requirements and their performance against the key performance indicators (KPIs). The clamping track concept emerges as the most promising.

The clamping track concept utilizes a chain drive with clamps attached to it that press on the wire rope. By applying force to the chain, the tension in the wire rope is increased. Detailed development of the clamping track concept has yielded a conceptual design of the system, which successfully meets both design objectives. However, achieving the goal of reducing the lower block weight introduces associated risks and complexities.

Specifically, if the lower block weight is reduced by more than 20%, the risk of uncontrollable upward motion of the lower block is imposed, which also complicates the emergency stop process. This risk is not outweighed by the load curve improvement of 3%, making it inadvisable to reduce the lower block weight by more than 20%.

Reducing the lower block weight by 20% improves the load curve by 1.5% and eliminates the risk of uncontrollable movement. Therefore, it is advisable to design the tensioner to work with a 20% lighter lower block. By using a modular lower block, the 20% weight can be added back if needed, eliminating the need for the tensioner during lowering. This offers the most flexible and highest performing version of the active wire rope tensioner.

I. Introduction

The offshore industry has been at the forefront of technological advancements, particularly driven by the global shift towards renewable energy. As the demand for sustainable energy sources continues to rise, the offshore market, especially in wind energy, has seen remarkable growth. The increased demand for wind energy and the technological advancements in that field have increased the size of wind turbines significantly over the years. These enormous wind turbines require larger cranes with more lifting capacity and larger outreach to install them.

Wire ropes are critical components of cranes, facilitating the hoisting and lowering of loads. A long-standing issue in crane operations is the wear of the wire rope. As cranes increase in size and capacity, this problem is exacerbated. The wear mechanisms of wire ropes are highly complex, with many contributing factors still not fully understood [1]. Weiskopf [2] identifies 18 distinct factors influencing the lifetime of wire ropes in multi-layer winding systems. These factors can be broadly categorized into three groups: drum design, wire rope type, and the forces acting on the wire rope. The dominant wear mechanisms that determine a wire rope's lifetime are highly dependent on the crane's geometry and usage.

This study aims to reduce the wear of the wire rope in cranes by controlling the tension in the wire rope when it is wound on the drum under low tension. Low and uneven tension can result in irregular winding, leading to high localised pressures that damage the wire rope [3][4]. Properly arranged winding can significantly enhance the wire rope's lifespan. Previous studies have shown a lifetime improvement of 20% up to 85% [5][6].

Ensuring that the full length of the wire rope is wound under sufficient tension can prevent a phenomenon that is even more damaging to wire ropes: cutting-in. Cutting-in occurs when the wire rope forces itself between the lower layers of the winding, as illustrated in Figure 1.

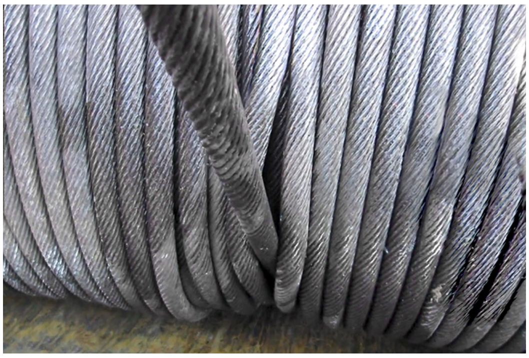

Fig. 1. Photo of a drum where cutting-in occurred [7]

This phenomenon typically arises during heavy lifting operations, where large tensional forces are exerted on the wire rope. The resultant pressure and friction from cutting-in can cause severe damage to the wire rope, leading to it being discarded prematurely.

The increase in crane size also affects another crucial aspect: the weight of the lower block. The lower block is a vital component of the crane, connecting the wire rope to the hook. Its weight causes it to descend during lowering, necessitating it to be heavy enough to overcome the weight of the wire rope and the inefficiencies in the sheaves. Its weight also ensures a minimal tension in the wire rope at the drum, required for proper spooling. The weight of the lower block directly impacts the lifting capacity of the crane. Therefore, reducing the required weight of the lower block would be highly beneficial.

These challenges can be solved with a single device: an active wire rope tensioner. This device can increase the tension in the wire rope during spooling, thereby improving the winding quality on the drum, reducing wire rope damage, and preventing cutting-in. Additionally, the device can be used to drive the wire rope, allowing for the use of a lighter lower block.

It is important to emphasise that these two goals inherently counteract each other. Specifically, reducing the weight of the lower block decreases the tension in the wire rope. While the principle of an active wire rope tensioner that achieves both goals is theoretically possible, determining the feasibility of designing such a system is the core focus of this research.

The research described in this paper aims to design an innovative wire rope tensioning system to increase the lifespan of wire ropes and enable the use of a lighter lower block. A literature and patent study forms the basis for exploring different concepts, as described in section II. The detailed design of the most promising concept is presented in section III.

II. Concept Exploration

A study of current literature and patents regarding tensioning devices was performed to allow the generation of multiple tensioner concepts. The literature study is described first, after which the concept generation is highlighted.

A. Literature and patent study

Devices designed to manipulate the tension in wire ropes often utilize frictional forces to achieve this. Examining both the standard Coulomb friction model and the Capstan equation, also known as the Euler–Eytelwein formula, reveals three approaches towards optimizing the active wire rope tensioner concepts.

Increasing the coefficient of friction is the simplest approach to optimising the tension generation capabilities of a tensioning device. Since the coefficient of friction is a material property, options for enhancing it are limited. However, research into optimal material pairings indicates that achieving a high coefficient of friction is feasible, albeit typically under low contact pressure. Additionally, the effective coefficient of friction can be increased by altering the shape of the object interacting with the wire rope, a method commonly applied in elevators [8]. While highly effective, this alternative shape increases the wear on the wire rope, which must be considered.

There are greater possibilities to increase the normal force between the wire rope and the tensioning device. One approach, described in many patents, involves clamping the wire rope between one or multiple tracks to create a high normal force, similar to tensioners used in pipe and cable laying. Increasing the normal force between a sheave and the wire rope has also been extensively studied. Some methods to achieve this include using external rollers, belts or claws, and magnets.

For tensioning devices that utilize a sheave or drum, increasing the wrap angle enhances traction capability. To achieve this, the wire rope needs to be axially offset to prevent contact between wraps. Devices designed to accomplish this include tracked drums, interlocking offset drums, lateral displacing drums, and the use of two connected drums.

A notable gap in the current literature and patents is the application of these systems to reduce the required weight of the lower block. While the concept of increasing wire rope tension has been described and studied in many patents and researches, there has been no significant focus on decreasing the lower block's weight. Although this reduction might not drastically impact a crane's capabilities, it is a small optimization step that leads towards achieving the best possible crane design.

B. Concept generation

To identify the optimal active wire rope tensioner concept, seven possible concepts were generated. Of these, three utilised mechanisms already installed in cranes to increase the tension in the wire rope, although these mechanisms are currently used for different purposes. The other four concepts are friction based tensioners, based on principles discovered in the literature study.

To assess each concept, a simplified parametric model was constructed to determine the tension in the wire rope throughout the rigging of a crane. This model incorporates the crane configuration, the weight of the wire rope, and sheave efficiency data, which accounts for efficiency variations depending on the tension [9]. The model is visualised in Figure 2

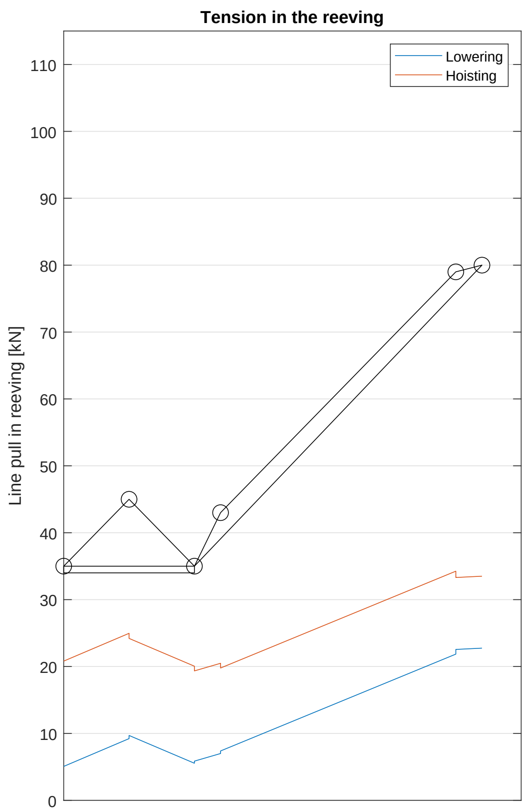

Fig. 2. Schematic drawing of the tension in the wire rope

Using a set of requirements and Key Performance Indicators (KPIs), the most promising concept was selected. The clamping track concept emerged as the most promising. This concept utilizes a single driven track with clamps positioned on top. These clamps are pressed onto the wire rope by rollers moving along a guide rail. By applying torque to the track's sprocket, the tension in the wire rope can be manipulated. A schematic drawing is shown in Figure 3.

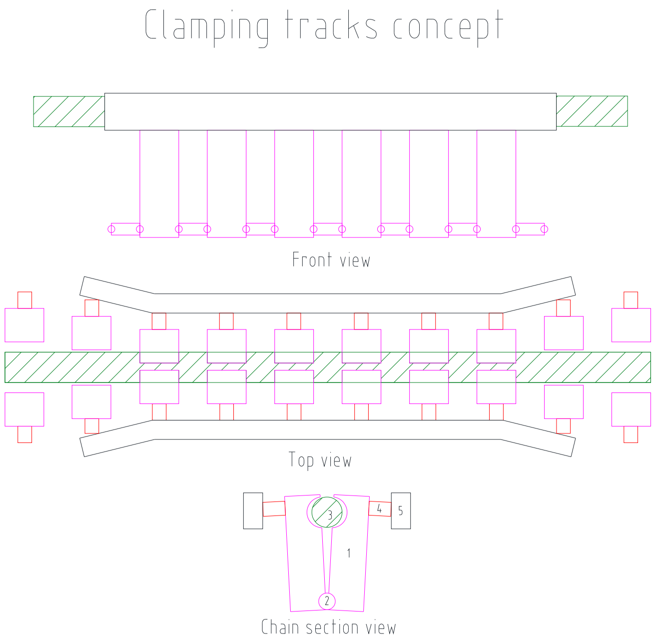

Fig. 3. Schematic drawing of the clamping track concept

The clamping track concept offers several advantages over other designs. Its tension generation capabilities can be easily scaled by adjusting the track length and the force applied to the wire rope, making it suitable for various types of cranes. By selecting the appropriate length and pressing force, the contact pressure on the wire rope remains lower compared to sheave-based designs. Additionally, since the clamping track does not introduce a bend into the crane's rigging, it is superior in extending the wire rope's lifespan. Another key advantage is that the clamping track can be completely removed from the crane's rigging, ensuring no impact on the crane's operating speed when not in use. Its superior tension generation capabilities also allow for a lighter lower block. Finally, the tension generation during hoisting is independent of the installation location, unlike sheave/drum-based tensioners.

Devices similar to this concept have been described in patents [10]. However, these devices are designed with different objectives and differ in several key aspects. These differences are presented in section III.

III. System design

Detailing the design of the clamping track concept has resulted in the design shown in Figure 4. One of the main innovations of this concept is the clamping mechanism. By using two mirrored parallelograms actuated by a single hydraulic cylinder, the movement and clamping force are always equal along the length of the guide rails. This eliminates the risk of the wire rope being pushed out of its path and prevents uneven force distribution, which could result in stresses exceeding allowable limits. Consequently, this design enhances the inherent safety of the system.

Another innovation of the clamping track concept is the use of guide rails along the full length of the track to control the position of the clamps. This design enhances inherent safety compared to using springs, as the performance of springs can be affected by fatigue and environmental influences.

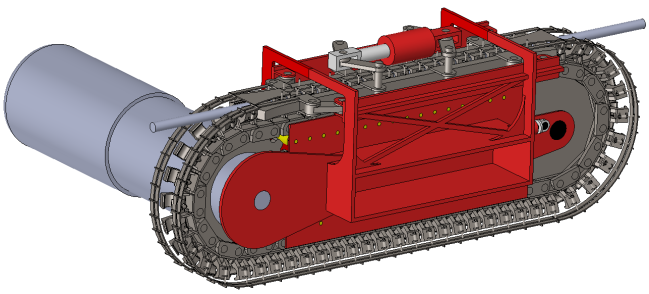

Fig. 4. Design of the clamping track concepts

Detailing the drive system of the tensioner has established the requirements for the motor and gearbox, as well as the need for a cooling system based on the modelling of the gearbox oil temperature. A dynamic analysis was conducted to examine the interaction between the tensioner and the main drum. This analysis was used to dimension the braking system of the tensioner.

Based on an analysis of currently used lower blocks, it is estimated that the lower block weight can be reduced by a maximum of 40%. This reduction would result in an effective improvement of the load curve by 3%. However, this also imposes significant safety and operational risks.

If the lower block weight is significantly reduced, its weight becomes insufficient to counteract the wire rope's own weight and sheave efficiency. Consequently, in the absence of the tensioner, the lower block would move uncontrollably upwards, posing a significant safety and operational risk. Additionally, in the event of a category 0 emergency stop, the inertia of the tensioner could cause slack rope issues. This issue necessitates the integration of a device capable of compensating for slack wire rope.

Reducing the lower block weight by only 20% would eliminate the risk of uncontrollable upward movement. However, this would reduce the load curve improvement to 1.5%, while still necessitating the tensioner to properly lower the lower block.

The chosen reduction in lower block weight influences the risks and other aspects of the tensioner, such as its length and the size of the drive. This raises the question of whether reducing the lower block weight justifies the additional risks and complications.

IV. Discussion

This study holds significant scientific relevance in the field of crane operations, as it addresses the critical issue of wire rope damage and cutting-in. Although alternative solutions, such as employing a larger drum or a heavier lower block, are available, these options present substantial drawbacks that render them undesirable. The device developed in this study aims to mitigate wire rope damage and cutting-in without adversely affecting crane designs. While this concept has been explored in literature, the principle of using the tensioner to reduce the lower block weight, is a concept not explored in existing literature. The findings of this study indicate that reducing the lower block weight while simultaneously increasing the tension during hoisting is feasible. However, this approach introduces increased complexity and associated risks.

The parametric model employed during the concept selection phase and for determining the system requirements during the detailed design phase is static and simplified. Consequently, inertias and dynamic effects between the drum and the tensioner concepts were not studied. Despite these limitations, the model is sufficiently robust to guide the selection process and determine the requirements during the detailed design phase. Its simplicity facilitates a clear comparison of different concepts, ensuring that the chosen design meets the necessary criteria. A dynamic analysis, which does incorporate the inertias and dynamic effects, was performed to detail the brake of the tensioner.

During the detailed design phase, several assumptions had to be made due to the unavailability of all required information in literature. The primary design

uncertainties include what tension is sufficient to prevent cutting-in, the maximum allowable radial pressure on the wire rope, and the extent to which the lower block weight can be reduced.
These uncertainties influence the design of the tensioner. However, their impact is limited, as one of the advantages of the clamping track concept is its scalability, achieved by adjusting the number of clamps. While the uncertainty in lower block weight reduction affects the maximum load curve improvement, the magnitude of this uncertainty is estimated to be approximately 1% for the effective load curve improvement.

The research objective of this study was to design a tensioning system that increases tension during hoisting and allows the use of a lighter lower block. While increasing the tension during hoisting was a requirement of the designed concept, the reduction of the lower block weight was not. This decision was made because it was uncertain whether it was feasible to design a tensioner capable of achieving both objectives.

As previously mentioned, the study demonstrated that achieving this dual objective is feasible, albeit with associated complexities and risks. If the goal is to have a 'one size fits all' concept that accommodates various applications, the proposed concept of the active wire rope tensioner is optimal based on the concept selection. Had the decision regarding the reduction of the lower block weight been made upfront, the concept selection and detailed design could have resulted in a different conceptual design for the active wire rope tensioner. This is because certain concept-specific advantages and disadvantages would have become more or less relevant.

V. Conclusion

In conclusion, this study aimed to design an active wire rope tensioner capable of enhancing the lifespan of the wire rope in crane operations up to 85% by increasing and controlling the wire rope tension during hoisting. Additionally, the tensioner can facilitate a reduction in the required lower block weight.

A study of current literature and patents investigates three potential methods. Firstly, the normal force in a tensioner can be increased by employing various clamping techniques, such as clamping the wire rope on sheaves with belts or rollers. Secondly, enhancing the coefficient of friction can be effective, although it is limited by maximum contact pressures and the potential for increased wire rope wear. Lastly, the study explores different techniques to increase the wrap angle for sheave-based traction devices. This literature study led to seven concepts out which the most promising was chosen based on a set of requirements and KPIs.

The development of the most promising concept resulted in a conceptual design of the active wire rope tensioner, which successfully meets both design objectives. However, achieving the goal of reducing the lower block weight introduces associated risks and complexities. Specifically, if the lower block weight is reduced by more than 20%, the risk of uncontrollable upward motion of the lower block is imposed, which also complicates the emergency stop process. This risk is not outweighed by the load curve improvement of 3%, making it inadvisable to reduce the lower block weight by more than 20%.

Reducing the lower block weight by 20% improves the load curve by 1.5% and eliminates the risk of uncontrollable movement. Therefore, it is advisable to design the tensioner to work with a 20% lighter lower block. By using a modular lower block, the 20% weight can be added back if needed, eliminating the need for the tensioner during lowering. This offers the most flexible and highest performing version of the active wire rope tensioner.

The research presented in this paper aimed to narrow down the wide array of possibilities to identify the optimal concept for the active wire rope tensioner. The conceptual design developed serves to increase confidence in the concept and provides a foundation for further research and development.

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B

Emergency Stop Process: Load Case Specific Plots

This appendix presents the results from the dynamic analysis of the CAT0 emergency stop process in Scenario 1 using a 1 kNm brake. Each plot illustrates the line speed of the drum and tensioner during the CAT0 stop. Additionally, the plots display the tension in the wire rope between the tensioner and the drum. The scenarios are displayed in Table B.1.

Table B.1: Load cases used in the dynamic analysis

Load case	Nominal speed	High speed	Empty hoisting	SWL hoisting	SWL/4 hoisting	Empty lowering	SWL lowering	SWL/4 lowering	Full drum	Empty drum	Drum max torque	Drum min torque	Tensioner max torque	Tensioner min torque
1	X		X						X		X		X	
2	X			X					X		X		X	
3	X		X							X	X		X	
4	X			X						X	X		X	
5	X					X			X		X		X	
6	X						X		X		X		X	
7	X					X				X	X		X	
8	X						X			X	X		X	
9		X	X						X		X		X	
10		X			X				X		X		X	
11		X	X							X	X		X	
12		X			X					X	X		X	
13		X				X			X		X		X	
14		X						X	X		X		X	
15		X				X				X	X		X	
16		X						X		X	X		X	
17	X		X						X		X			X
18	X			X					X		X			X
19	X		X							X	X			X
20	X			X						X	X			X
21	X					X			X		X			X
22	X						X		X		X			X
23	X					X				X	X			X
24	X						X			X	X			X
25		X	X						X		X			X
26		X			X				X		X			X
27		X	X							X	X			X
28		X			X					X	X			X
29		X				X			X		X			X
30		X						X	X		X			X
31		X				X				X	X			X
32		X						X		X	X			X
33	X		X						X			X	X	
34	X			X					X			X	X	
35	X		X							X		X	X	
36	X			X						X		X	X	
37	X					X			X			X	X	
38	X						X		X			X	X	
39	X					X				X		X	X	
40	X						X			X		X	X	
41		X	X						X			X	X	
42		X			X				X			X	X	
43		X	X							X		X	X	
44		X			X					X		X	X	
45		X				X			X			X	X	
46		X						X	X			X	X	
47		X				X				X		X	X	
48		X						X		X		X	X	
49	X		X						X			X		X
50	X			X					X			X		X
51	X		X							X		X		X
52	X			X						X		X		X
53	X					X			X			X		X
54	X						X		X			X		X
55	X					X				X		X		X
56	X						X			X		X		X
57		X	X						X			X		X
58		X			X				X			X		X
59		X	X							X		X		X
60		X			X					X		X		X
61		X				X			X			X		X
62		X						X	X			X		X
63		X				X				X		X		X
64		X						X		X		X		X

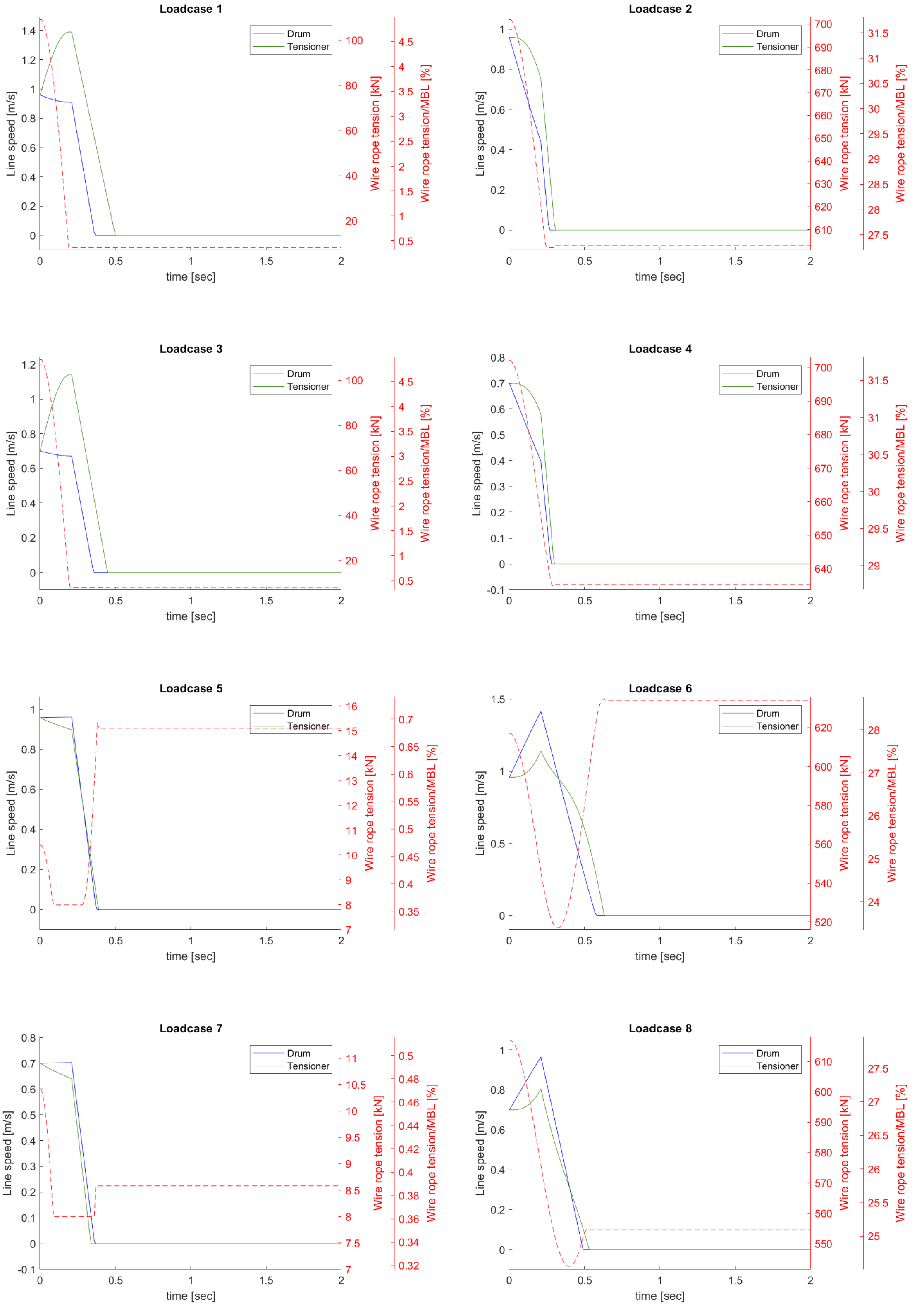

Figure B.1: Visualisation of the dynamic analysis for load cases 1-8

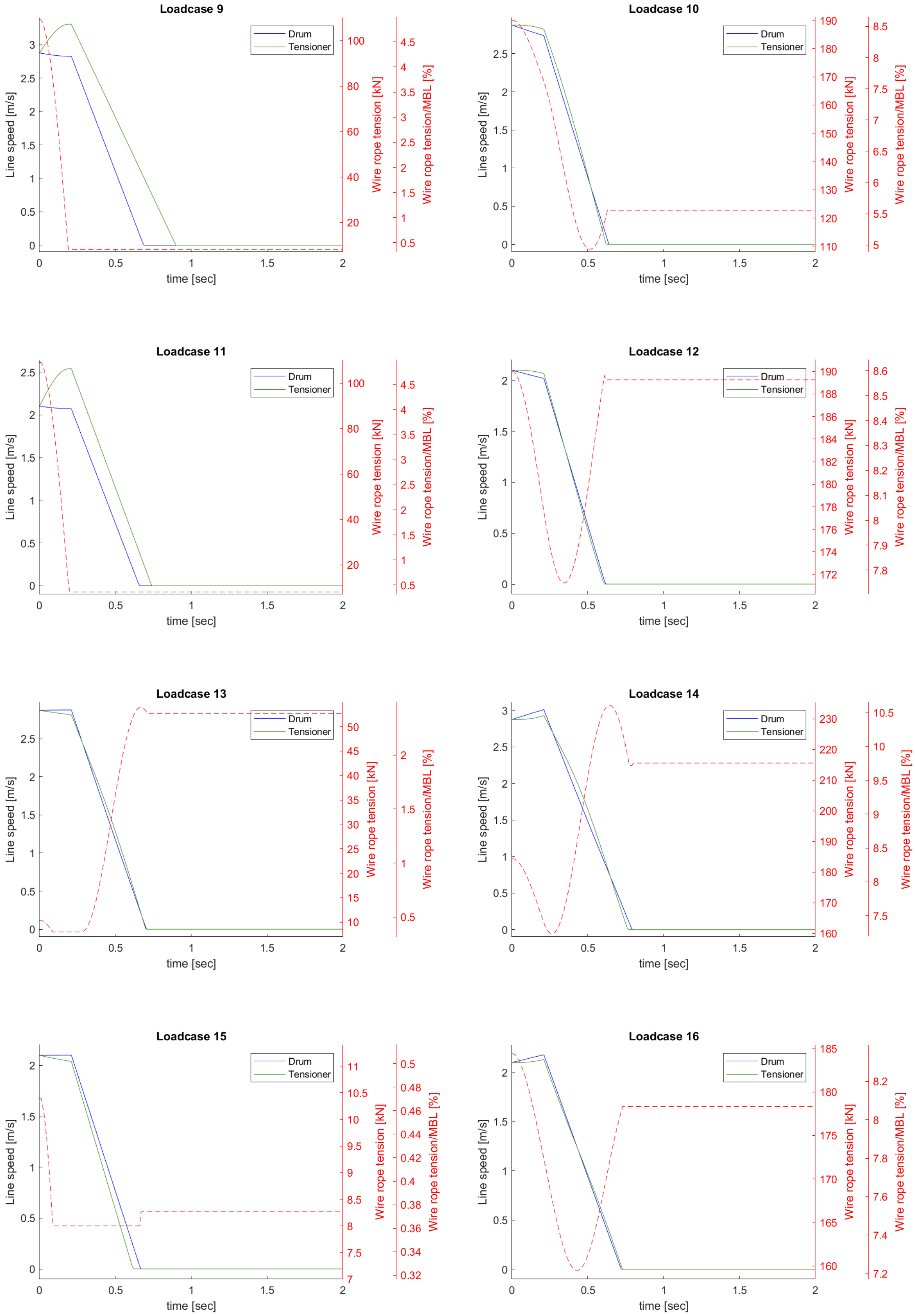

Figure B.2: Visualisation of the dynamic analysis for load cases 9-16

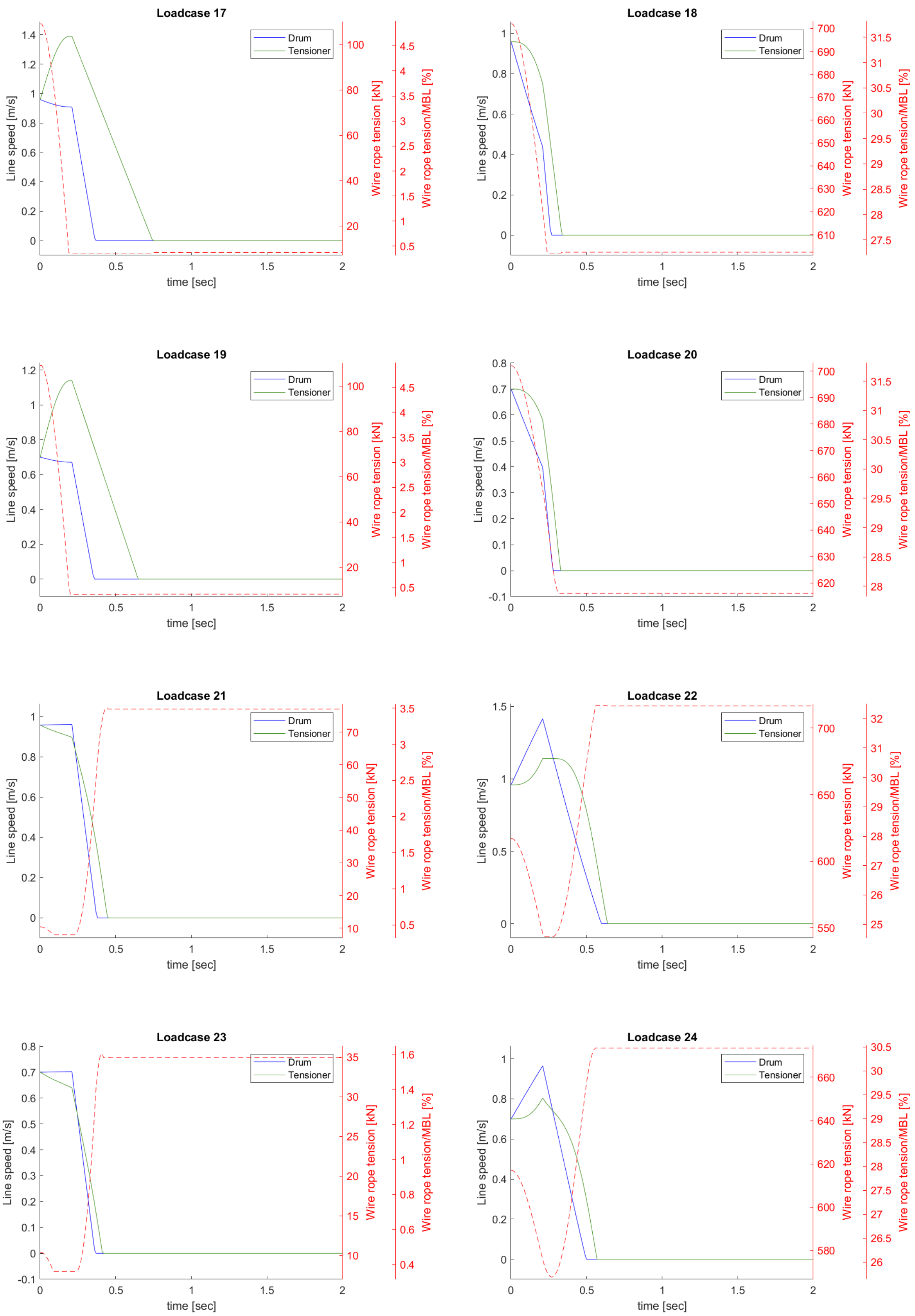

Figure B.3: Visualisation of the dynamic analysis for load cases 17-24

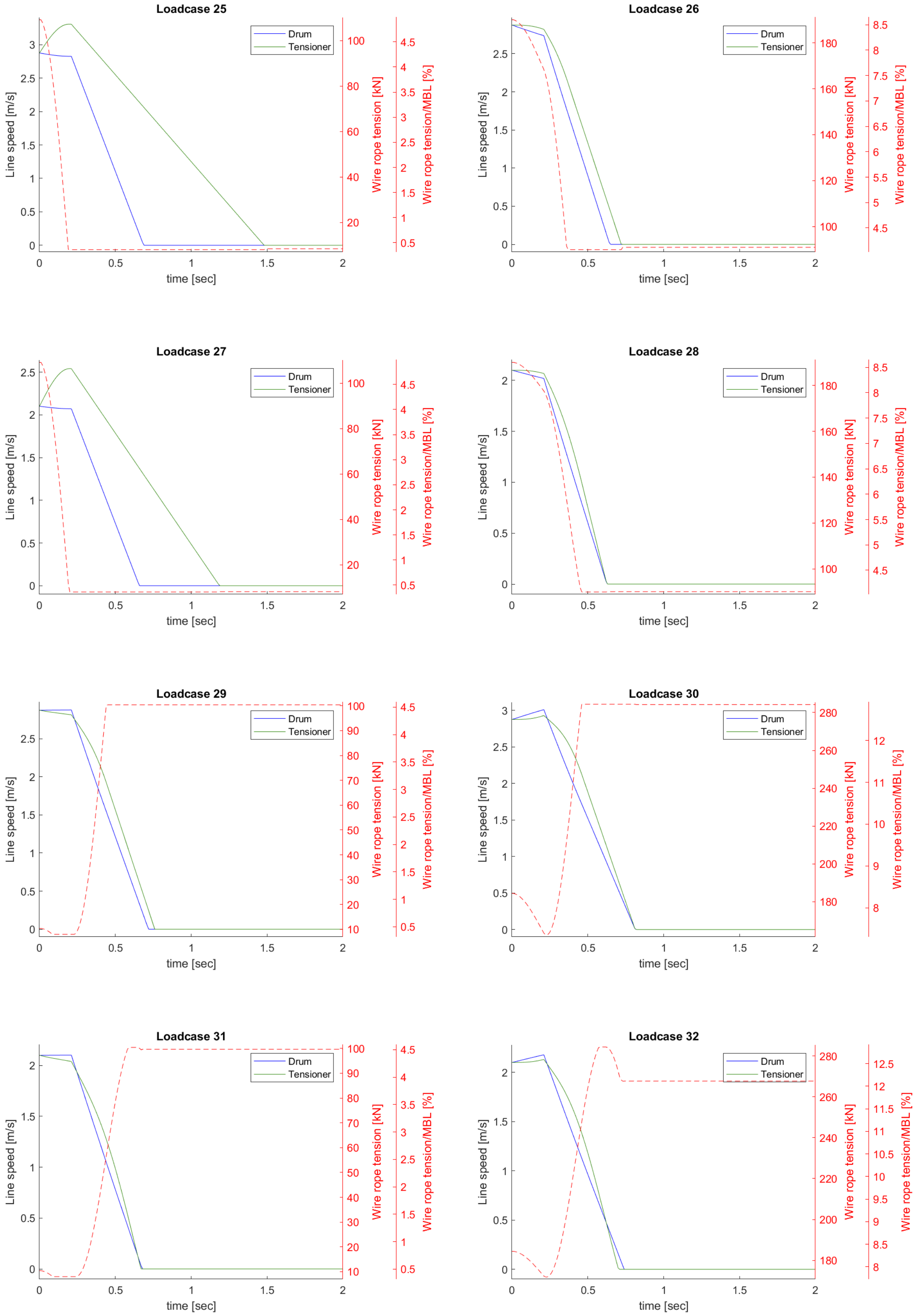

Figure B.4: Visualisation of the dynamic analysis for load cases 25-32

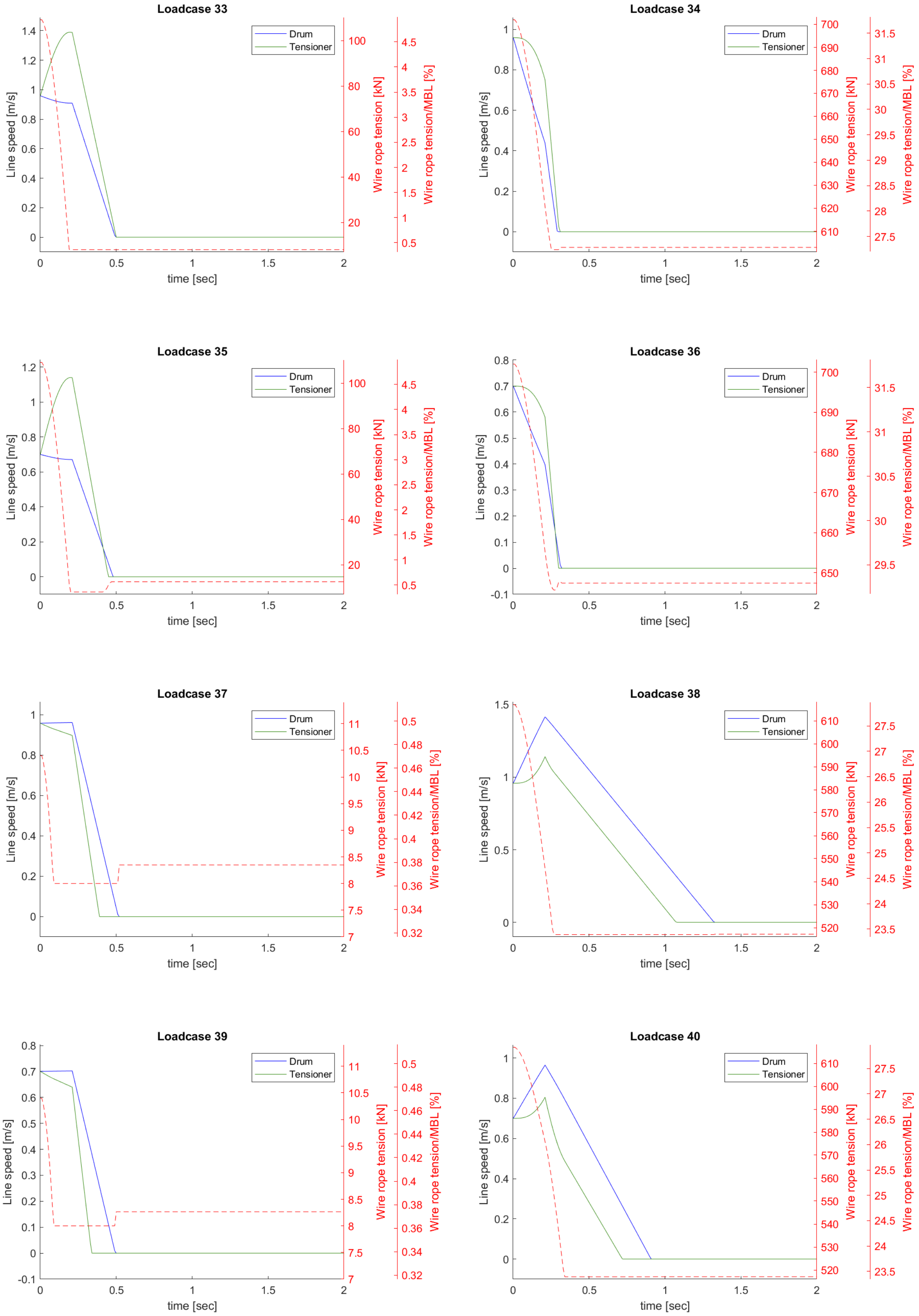

Figure B.5: Visualisation of the dynamic analysis for load cases 33-40

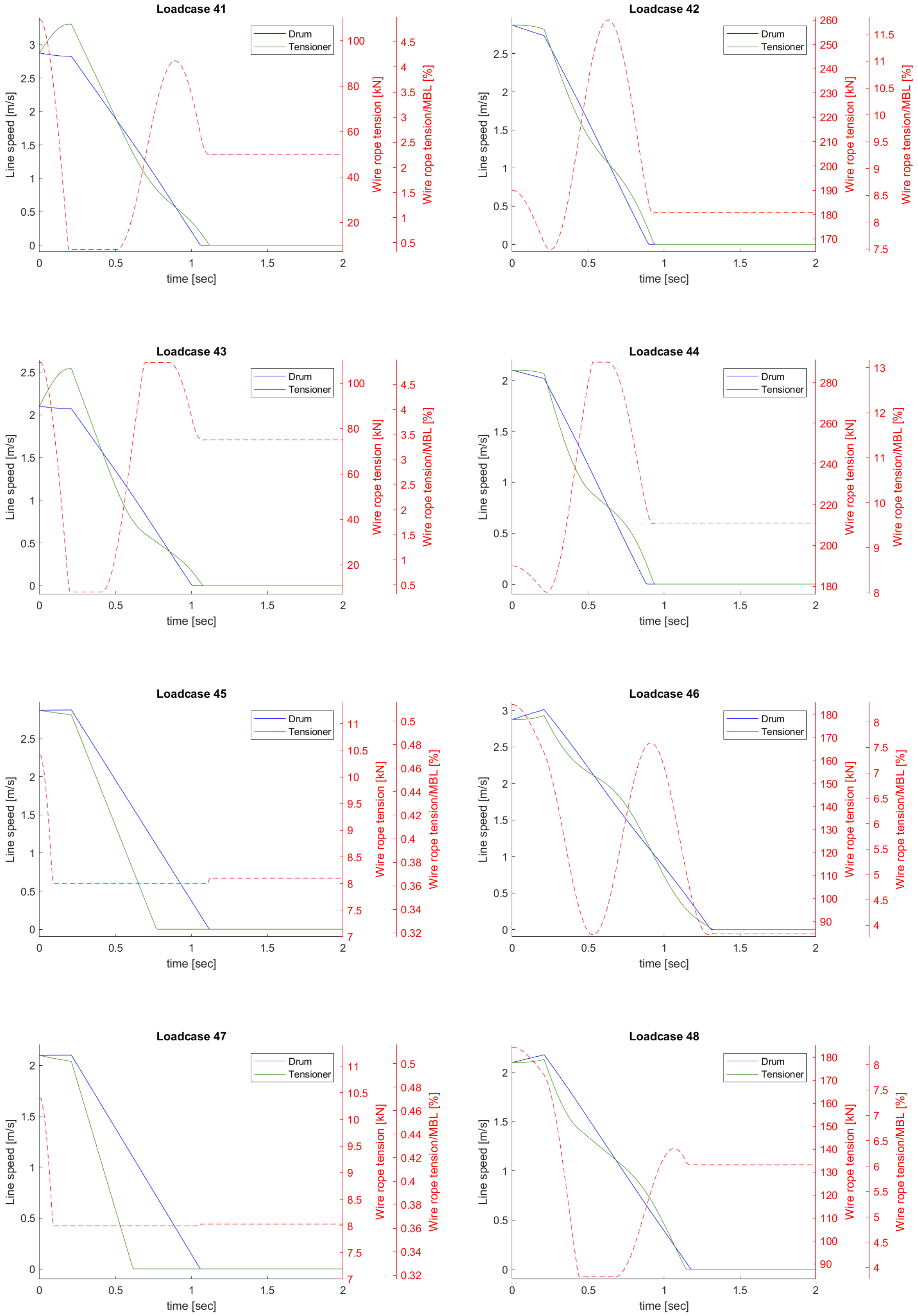

Figure B.6: Visualisation of the dynamic analysis for load cases 41-48

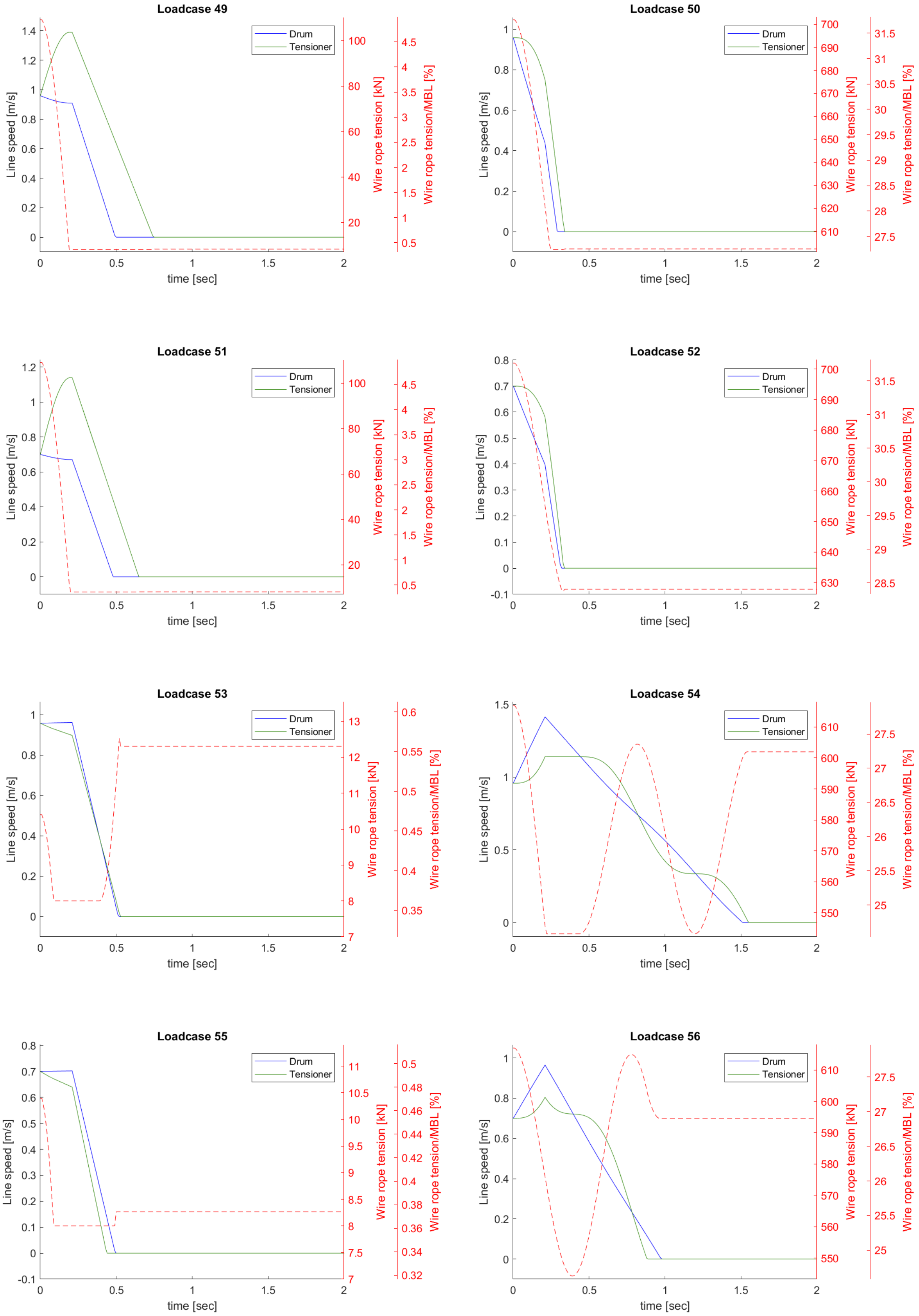

Figure B.7: Visualisation of the dynamic analysis for load cases 49-56

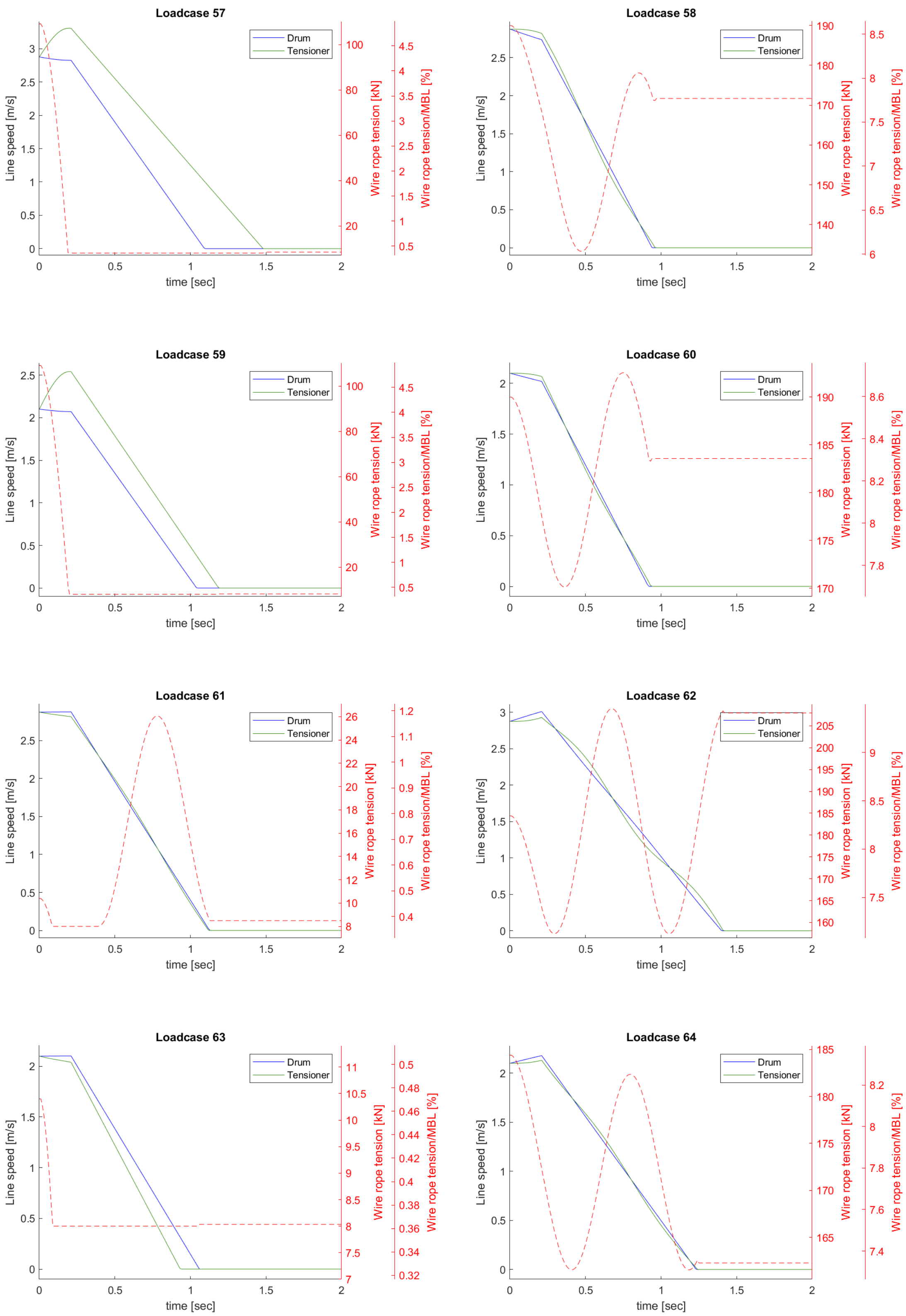

Figure B.8: Visualisation of the dynamic analysis for load cases 57-64

C
Risk assessment

Below the risk assessment made during the detailed design phase is shown. It displays a risk assessment for each of the three scenarios. Risks that are identical to previous noted risks are made grey for clarity.

RISK ASSESSMENT (TECHNICAL RISKS)

DOCUMENT NR	ORDER NR	CLIENT	PROCEDURE / SYSTEM	TEAM MEMBERS	REVISION	DATE
	A24-29000	Huisman	Active Wire Rope Tensioner		A	

NR	REV	SCENARIO	PROCEDURE (or) SYSTEM	PROCEDURE STEP (or) FUNCTION	SAFETY / DOWNTIME HAZARD	POSSIBLE HARM *(due to hazardous situation or event)*	CAUSE	ESTIMATION SAFETY SEVERITY	OCCURRENCE	RISK LEVEL	(POSSIBLE) RISK REDUCTION MEASURES Step 1: Inherently safe design (elimination of hazards / reduction by design) Step 2a: Safeguarding Step 2b: Complementary protective measures Step 3: Information for use (at machine / in manual)	EWI / Huisman standard	Safety function (Yes / No)	EXPECTED RESIDUAL RISK SAFETY SEVERITY	OCCURRENCE	RISK LEVEL	OPERATIONAL SEVERITY	OCCURRENCE	RISK LEVEL	REMARKS
1	A	3 (Luffing frame, no LB weight reduction)	Clamps	Close	Wire rope not positioned between clamps correctly	Flattening of wire rope, possible replacement needed	Slackrope	3	2	6	Prevent system activation when slack rope is detect by the control system									
2	A				Guide rollers not rolling	Roller wearing and not properly transferring the clamping force. Causes higher clamping force in other clamps	Bearing malfunction caused by the environment or wear	2	2	4	Regular inspection and maintenance on the rollers									
3	A			Open	Clamps not opening before rotation	Clamps forced open by wire rope, high contact pressures, possibly damaging the wire rope	Friction in clamp axis bearing very high	4	3	12	Use a form fitting mechanism to ensure the clamps always open									
4	A		Clamping mechanism	Move	Does not retract	Clamps stay clamped to wire rope, unnecessary wear of components	Failure of cylinder or system getting stuck	2	3	6	Regular inspection and maintenance on moving parts Measure the pressure in the cylinder									
5	A				Does not engage	Not possible to generate tension, tension goal at drum not reached	Failure of cylinder or system getting stuck	2	3	6	Regular inspection and maintenance on moving parts Measure the pressure in the cylinder									
					Uneven movement	Clamps push the wire rope away from its path, resulting in high peak pressures in the wire rope and clamps	Failure of cylinder or system getting stuck	3	3	9	Design clamping mechanism to prevent uneven movement									
6	A			Clamp	Clamp force too high	Clamp pressure increased beyond target. Possible damage to clamp components	Pressure in cylinder too high	2	3	6	Measure the pressure in the cylinder, control the pressure accordingly									
7	A				Clamp force too low	Desired tension generation can not be reached. Wire rope damage on drum not prevented	Pressure in cylinder too low	2	3	6	Measure the pressure in the cylinder, control the pressure accordingly									
					Clamp force uneven	Total clamping force not evenly devided over all clamps, pressure increased beyond target in some clamps	Failure of cylinder or system getting stuck	3	3	9	Design clamping mechanism to prevent uneven tensioner									
8	A		Drive system inc chain	Control speed of chain	Chain does not move	Slipping between wire rope and clamps, possibly causing damage. Tension generation fixed by dynamic friction	System failure	3	2	6										
9	A				Chain moves too slow	Slipping between wire rope and clamps, possibly causing damage. Tension generation fixed by dynamic friction	Control system not working properly	4	3	12	Measure speed of wire rope and of the chain, control the speed accordingly									
10	A				Chain moves too fast	Tension generation in oposite direction, possibly causing slack rope between tensioner and drum	Driving of motor in opposite direction instead of braking	4	3	12	Measure speed of wire rope and of the chain, control the speed accordingly									
11	A			Control chain driving torque	Tension in chain too high	If the drive brakes to hard slip will occur and dynamic friction will occur. The slip could cause damage to the wire rope	Control system not working properly	4	3	12	Measure tension in wire rope, control torque accordingly									
12	A				Tension in chain too low	Not enough tension generated in the wire rope, tension target at drum not reached	Control system not working properly	2	3	6	Measure tension in wire rope, control torque accordingly									
13	A		Chain pre tension cylinder	Control pre tension in chain	Chain pre tension too high	Friction forces in chain higher than required. Possible deformation in chain links or eyes	Pressure in cylinder too high	2	2	4	Measure the pressure in the cylinder, control the pressure accordingly									
14	A				Chain pre tension too low	If the pre tension in the chain is to low the clamps will rotate because of the friction forces. This could damage the wire rope by peak pressure spots.	Pressure in cylinder too low	3	2	6	Measure the pressure in the cylinder, control the pressure accordingly									
15	A		Clamps	Close	Wire rope not positioned between clamps correctly	Flattening of wire rope, possible replacement needed	Slackrope	3	2	6	Prevent system activation when slack rope is detect by the control system									
16	A				Guide rollers not rolling	Roller wearing and not properly transferring the clamping force. Causes higher clamping force in other clamps	Bearing malfunction caused by the environment or wear	2	2	4	Regular inspection and maintenance on the rollers									
17	A			Open	Clamps not opening before rotation	Clamps forced open by wire rope, high contact pressures, possibly damaging the wire rope	Friction in clamp axis bearing very high	4	3	12	Use a form fitting mechanism to ensure the clamps always open									
18	A		Clamping mechanism	Move	Does not retract	Clamps stay clamped to wire rope, unnecessary wear of components	Failure of cylinder or system getting stuck	2	3	6	Regular inspection and maintenance on moving parts Measure the pressure in the cylinder									
19	A				Does not engage	Not possible to generate tension, tension goal at drum not reached Not possible to lower the lower block	Failure of cylinder or one of the rotation points getting stuck	3	3	9	Regular inspection and maintenance on moving parts Measure the pressure in the cylinder									
20	A				Uneven movement	Clamps push the wire rope away from its path, resulting in high peak pressures in the wire rope and clamps	Failure of cylinder or system getting stuck	3	3	9	Design clamping mechanism to prevent uneven movement									

NR	REV	SCENARIO	PROCEDURE (or) SYSTEM	PROCEDURE STEP (or) FUNCTION	SAFETY / DOWNTIME HAZARD	POSSIBLE HARM (due to hazardous situation or event)	CAUSE	ESTIMATION (IN SAFETY) SEVERITY	OCCURRENCE	RISK LEVEL	RISK REDUCTION STEPS: (POSSIBLE) RISK REDUCTION MEASURES Step 1: Inherently safe design (elimination of hazards / reduction by design) Step 2a: Safeguarding Step 2b: Complementary protective measures Step 3: Information for use (at machine / in manual)	EWI / Huisman standard	Safety function (Yes / No)	EXPECTED RESIDUAL RISK SAFETY SEVERITY	SAFETY OCCURRENCE	SAFETY RISK LEVEL	OPERATIONAL SEVERITY	OPERATIONAL OCCURRENCE	OPERATIONAL RISK LEVEL	REMARKS
21	A	2 (Luffing frame, 20% LB weight reduction)	Clamping mechanism	Clamp	Clamp force too high	Clamp pressure increased beyond target. Possible damage to clamp components	Pressure in cylinder too high	2	3	6	Measure the pressure in the cylinder, control the pressure accordingly									
22	A				Clamp force too low	If clamping pressure is very low not enough tension can be generated to lower the lower block	Pressure in cylinder too low	3	3	9	Measure the pressure in the cylinder, control the pressure accordingly									
23	A				Clamp force uneven	Total clamping force not evenly devided over all clamps, pressure increased beyond target in some clamps	Failure of cylinder or system getting stuck	3	3	9	Design clamping mechanism to prevent uneven tensioner									
24	A		Drive system inc chain	Control speed of chain	Chain does not move	Slipping between wire rope and clamps, possibly causing damage. Tension generation fixed by dynamic friction	System failure	3	2	6										
25	A				Chain moves too slow	Slipping between wire rope and clamps, possibly causing damage. Tension generation fixed by dynamic friction	Control system not working properly	4	3	12	Measure speed of wire rope and of the chain, control the speed accordingly									
26	A				Chain moves too fast	Tension generation in oposite direction, possibly causing slack rope between tensioner and drum	Driving of motor in opposite direction instead of braking	4	3	12	Measure speed of wire rope and of the chain, control the speed accordingly									
27	A			Control chain driving torque	Tension in chain too high	If the drive brakes to hard slip will occur and dynamic friction will occur. The slip could cause damage to the wire rope	Control system not working properly	4	3	12	Measure tension in wire rope, control torque accordingly									
28	A				Tension in chain too low	If the tension generation is too low the lower block will not lower	Control system not working properly	3	3	9	Measure tension in wire rope, control torque accordingly									
29	A		Chain pre tension cylinder	Control pre tension in chain	Chain pre tension too high	Friction forces in chain higher than required. Possible deformation in chain links or eyes	Pressure in cylinder too high	2	2	4	Measure the pressure in the cylinder, control the pressure accordingly									
30	A				Chain pre tension too low	If the pre tension in the chain is to low the clamps will rotate because of the friction forces. This could damage the wire rope by peak pressure spots.	Pressure in cylinder too low	3	2	6	Measure the pressure in the cylinder, control the pressure accordingly									
31	A	1 (Boom, 40% LB weight reduction)	Clamps	Close	Wire rope not positioned between clamps correctly	Flattening of wire rope, possible replacement needed	Slackrope	3	2	6	Prevent system activation when slack rope is detect by the control system									
32	A				Guide rollers not rolling	Roller wearing and not properly transferring the clamping force. Causes higher clamping force in other clamps	Bearing malfunction caused by the environment or wear	2	2	4	Regular inspection and maintenance on the rollers									
33	A			Open	Clamps not opening before rotation	Clamps forced open by wire rope, high contact pressures, possibly damaging the wire rope	Friction in clamp axis bearing very high	4	3	12	Use a form fitting mechanism to ensure the clamps always open									
34	A		Clamping mechanism	Move	Does not retract	Clamps stay clamped to wire rope, unnecessary wear of components	Failure of cylinder or system getting stuck	2	3	6	Regular inspection and maintenance on moving parts Measure the pressure in the cylinder									
35	A				Does not engage	Not possible to generate tension, tension goal at drum not reached and lower block will move upwards uncontrollable	Failure of cylinder or one of the rotation points getting stuck	5	3	15	Regular inspection and maintenance on moving parts Measure the pressure in the cylinder									
36	A				Uneven movement	Clamps push the wire rope away from its path, resulting in high peak pressures in the wire rope and clamps	Failure of cylinder or system getting stuck	3	3	9	Design clamping mechanism to prevent uneven movement									
37	A			Clamp	Clamp force too high	Clamp pressure increased beyond target. Possible damage to clamp components	Pressure in cylinder too high	2	3	6	Measure the pressure in the cylinder, control the pressure accordingly									
38	A				Clamp force too low	Not possible to generate enough tension, tension goal at drum not reached and lower block will move upwards uncontrollable	Pressure in cylinder too low	5	3	15	Measure the pressure in the cylinder, control the pressure accordingly									
39	A				Clamp force uneven	Total clamping force not evenly devided over all clamps, pressure increased beyond target in some clamps	Failure of cylinder or system getting stuck	3	3	9	Design clamping mechanism to prevent uneven tensioner									
40	A		Drive system inc chain	Control speed of chain	Chain does not move	Slipping between wire rope and clamps, possibly causing damage. Tension generation fixed by dynamic friction	System failure	3	2	6										
41	A				Chain moves too slow	Slipping between wire rope and clamps, possibly causing damage. Tension generation fixed by dynamic friction	Control system not working properly	4	3	12	Measure speed of wire rope and of the chain, control the speed accordingly									
42	A				Chain moves too fast	Tension generation in oposite direction, possibly causing slack rope between tensioner and drum	Driving of motor in opposite direction instead of braking	4	3	12	Measure speed of wire rope and of the chain, control the speed accordingly									

ID			ACCIDENT SCENARIO					ESTIMATION (IN			RISK REDUCTION STEPS			EXPECTED RESIDUAL RISK						
								SAFETY			(POSSIBLE) RISK REDUCTION MEASURES			SAFETY			OPERATIONAL			
NR	REV	SCENARIO	PROCEDURE *(or)* SYSTEM	PROCEDURE STEP *(or)* FUNCTION	SAFETY / DOWNTIME HAZARD	POSSIBLE HARM *(due to hazardous situation or event)*	CAUSE	SEVERITY	OCCURRENCE	RISK LEVEL	Step 1: Inherently safe design (elimination of hazards / reduction by design) Step 2a: Safeguarding Step 2b: Complementary protective measures Step 3: Information for use (at machine / in manual)	EWI / Huisman standard	Safety function (Yes / No)	SEVERITY	OCCURRENCE	RISK LEVEL	SEVERITY	OCCURRENCE	RISK LEVEL	REMARKS
43	A			Control chain driving torque	Tension in chain too high	If the drive brakes to hard slip will occur and dynamic friction will occur. The slip could cause damage to the wire rope	Control system not working properly	4	3	12	Measure tension in wire rope, control torque accordingly									
44	A				Tension in chain too low	If the tension generation is too low the lower block will not lower	Control system not working properly	3	3	9	Measure tension in wire rope, control torque accordingly									
45	A		Chain pre tension cylinder	Control pre tension in chain	Chain pre tension too high	Friction forces in chain higher than required. Possible deformation in chain links or eyes	Pressure in cylinder too high	2	2	4	Measure the pressure in the cylinder, control the pressure accordingly									
46	A				Chain pre tension too low	If the pre tension in the chain is to low the clamps will rotate because of the friction forces. This could damage the wire rope by peak pressure spots.	Pressure in cylinder too low	3	2	6	Measure the pressure in the cylinder, control the pressure accordingly									
47	A		Brake	Keep tension in wire rope after E stop	Not enough brake torque to prevent upwards motion of lower block	Lower block moves upwards uncontrollable, crashing into block catcher. Possibly destroying it and causing harm	Brakes not strong enough, environmental effect on the brake material	4	2	8	Properly dimension brake and prevent contamination possibly add extra wire rope emergency brake in top of boom									
48	A			CAT0 stop	During CAT0 stop slip occues	Damage to wire rope due to the slippage	Big mismatch in braking time between drum and tensioner	3	3	9	Dimension brakes properly									
49	A				During CAT0 stop slack rope occurs	Wire rope running of sheaves	Big mismatch in braking time between drum and tensioner	4	3	12	Take up extra low tension wire rope with a mechanism									