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Specialization: Transport Engineering and Logistics

Report number: 2014.TEL.7875

Title: **Improved Snagload Protection system for STS container cranes**

Author: A.N.M. van der Bijl

Title (in Dutch) Verbeterd Snagload bescherming systeem voor STS container kranen

Assignment: Master thesis

Confidential: yes (until October 2015)

Initiator (university): prof.dr.ir G. Lodewijks

Initiator (company): ir R.J.G. Kleiss (Cargotec, Rotterdam)

Supervisor: ir W. van den Bos

Supervisor (company): ir H. W van Ingen Schenau

Date: September 18, 2014

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Supervisor (Cargotec):	H.W. van Ingen Schenau	Specialization:	TEL
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		Confidential:	Yes

Subject: Snag Load Protection: early snag detection

Cargotec (Kalmar) offers the widest range of cargo handling solutions and services to ports, terminals, distribution centers and heavy industry. One in four container movements around the world is handled by a Kalmar solution. This assignment will only focus on Ship to Shore container cranes.

During container unloading it may occur that the spreader or container gets jammed in the vessel, this is called a snag load. This sudden stop causes high forces in the crane. There is special equipment available on the market to reduce the overload situation once snag has occurred.

Previous studies have shown that snag load reduction systems are inevitable with current heavy loads and high hoisting speeds.

A new proposal by Kalmar and Sibre (a German brake manufacturer) is to detect the snag earlier and take quicker measures (braking) to reduce the snagload itself rather than fighting the consequences like current systems do.

Your assignment is to study at least the following issues:

- Theoretical background of snag and conventional protection systems
- Event scheme of snag: time available to detect, react and inter(act)
- Overview of possible protection systems, ways to:
    - Detect: *measuring of torque, force, inclination, deceleration etc.*
    - React
    - (Inter)act: *de-couple, no special intervention, brake*
- Investigate feasibility of new proposed Snag Load Protection system by Sibre
- Create a validated model (multi model dynamics) to make impact of different systems visible
    - Compare: conventional systems with new proposal
- Come up with quickest/best way to detect snag.

The report should comply with the guidelines of the section. Details can be found on the website.

The professor,

Prof. dr. ir G. Lodewijks

Preface

I am proud to present you this master thesis, a symbolic stack of paper, containing my graduation project and final work in order to achieve my master degree in Mechanical Engineering from the Technical University of Delft.

In line with my mastertrack Transportation Engineering I searched for a practical assignment related to mechanical lifting equipment. Together with Kalmar Ship To Shore division this assignment was formulated: the investigation and improvement of Snag Protection for container cranes.

First of all I want to thank Kalmar for offering me the opportunity to increase my knowledge and ability to analyze and work on a real practical and current issue. Even though it were turbulent times at Kalmar STS due to reorganizing, I was not put aside but helped wherever possible and involved in the process; taken up in the shrinking team. Besides Kalmar a lot of extern people provided assistance. At the end of the report is a list of extern people who were of great help and a great source of information! Thank you all!

I would also like to thank my university, this finalizes my time in at the Delft University of Technology, I achieved a lot, this is the basis for a large part of my (future) life.

Finally thanks to all my involved supervisors and leading professor: Rene Kleiss, Walter de Jong, Cock Angevaren, Huub van Ingen Schenau, Wouter van de Bos and prof. G. Lodewijks, thank you for your time and efforts to make this possible.

I am convinced that this has been a relevant research, especially for me, I hope this investigation contributes to the further development and implementation of an improved snag protection system for STS cranes.

"For now enough theory and text, time to bring it into practice… The toolbox is waiting for me.."

Ton van der Bijl

September 2014

Summary

This graduation and report focusses on an extreme load scenario of ship to shore container cranes. Containers are standardized steel boxes used for multi-modal transport of material all over the world. To (un)load big container ships, ship to shore container cranes are being used.

In the ships the containers are stacked in cellguides. During hoisting of the load inside the ship, for some reason, the load gets stuck. This sudden stopping is called snag, and this causes huge loads on the crane: snagload. There are several systems available on the market which claim to reduce this snagload and protect the crane, these systems however have an often non-proven working principle and are expensive, heavy and complex. Kalmar, a producer of container handling equipment, is concerned about this snagload and searches for a better way to protect their cranes and therefore initiated this graduation.

Rope force during snag

Cranes use ropes to lift the container from the vessel, the hoisting system can be seen as a winch: motors drive a gearbox which is connected to a rope drum. The force on the ropes is an indication of trouble during snag. The load is stuck and the only parts connecting this load to the crane are the ropes. A too large force in the ropes directly means too much load on the crane structure and mechanical components. Therefore the focus is on the rope force which undergoes 3 phases in case of a snag event: static force, increase due to motor torque and increase by rotational energy.

Static ropeforce

During normal hoisting the hoistspeed is constant and therefore also the ropeforce is constant.

Torque increase

Then a sudden jamming stops the container or spreader: snag! This means the load suddenly stops and the ropes cannot raise the load anymore (this is most often the result of a damaged cellguide). At the moment of snag the winch, driveline and system are not aware of this and therefore still maintain speed and try to lift the load by pulling on the rope. The ropes can be seen as long springs, thus the elongation results in a linear force increase. The drives/motors increase torque to maintain speed until maximum torque is reached, then the torque goes to zero.

Rotational energy

The drives shutdown: torque goes to zero but there is still residual rotational velocity in the driveline. All rotating components therefore still contain kinetic energy related to their moment of inertia and continue to stretch the ropes. All rotational energy is transferred into the ropes and thereby increase the ropeforce even more.

This maximum ropeforces by snag are much higher than the allowable ropeforce and calculated maximum loads and therefore put the crane and terminal at risk. In normal operation the safety factor for ropes is 6 times the normal ropeforce, in case of snag the safety is less than factor 2. Therefore snag protection is inevitable!

Calculations and modelling have shown that the entire snag event, from snag initiation up to complete stop, takes only 400 to 600 milliseconds. Therefore detection and snag protection has to be very quick.

Important factors for snag

The rotational velocity of the motors is very important for the snag events. High speed rotation means a lot of rotational energy. The actual speed of hoisting depends on the load; a light load is lifted with twice the speed of a heavy lead. This means that light loads are subjected to a lot of rotational energy due to the high rotational velocity and that the maximum torque is reached very quickly. For a heavy load the speed is much slower thus less rotational energy, the time involved in the torque increase for the motors is much longer. The amount of reserve torque at low speed is significantly more than for high speed hoisting, meaning this causes a large force increase for the low speed snag. The rotational energy is the biggest problem of high speed snag.

Snagload protection

The ropeforce may never exceed the elastic limit, because this would mean permanently damaging the ropes and putting extreme forces on the crane structure and mechanical components. Typically the elastic limit of hoist rope is about 50% of the minimal break strength; this limit may never be exceeded. In practice the limit of 1/3 the break strength is used for extreme loads. Limiting the ropeforce and thereby protecting the crane structure is the goal of a snagload protection system, the maximum ropeforce must be far below 50%; the target is to reach maximally 33% of the break strength. Additionally the system should protect the mechanical components for too large torque values.

There are basically three options for limiting the ropeforce and protecting the crane:

- Stopping the drivetrain
- Increasing ropelength
- Decoupling the drivetrain

The last two principles are used by conventional systems: hydraulic snag protection by Rima and ZPMC and a new system of Pintsch Bubenzer that uses a breakcoupling called SOS. These systems can be huge, heavy and complex and are expensive but yet have a non-proven working.

Therefore Kalmar and Sibre (a German brake supplier) came up with the idea to protect differently: early snag detection & fast stopping, without additional equipment.

Early detection

Two factors are very important for snag detection: speed and reliability. Reliability because no snag may be missed and the system may not give false detection, resulting in delays in operation and likely system shutdown. Since snag happens in 400-600ms speed is very important, the earlier it is detected the more measures can be taken. The best options to measure seem acceleration or rope force measurement.

Acceleration has to be measured at the headblock/spreader, Sibre came up with the idea of horizontal force measurement, it is believed that the damaged cell guide causes a horizontal acceleration first, even before the load is stuck. If this horizontal acceleration can be detected, the snag can be predicted. The headblock and spreader are subjected to much impact causing noise for the measurement and damage to the components, therefore this is not considered to be a suitable place for measurement of acceleration nor force.

Conventional load measuring is done at the utmost back of front of the crane far from the actual snag. Due to the long rope lengths this is a non-detailed and very fluctuating signal caused by whipping of the cable. For ropeforce measurement the best place to measure is therefore the trolley. Together with Pat-Kruger the development of fast measuring and processing is initiated. This system will detect snag fast and reliable based on the force increase in time. The measuring processing and transferring the snag signal to the machine house should be done within 35 milliseconds after snag initiation.

Fast stopping

Once snag is detected, actions can be started. The idea is to initiate stopping as soon as possible and brake as fast as allowable. Deceleration causes moments on the driveline that may not exceed component limitations. Sibre is developing ultra-fast brakes, fast emergency brakes are currently being tested and are able to apply 90% braking torque within 80ms, compared to 300-400ms for conventional brakes. The operational fast brakes are also under development, they are likely to close faster but for now is assumed that also these close within 80ms. A third option for stopping the hoist is by reversing torque on the motor by the drive, ABB and Siemens can apply a reverse torque within 50ms after a snag detection signal.

For every project/crane an analysis has to be made to determine the maximum allowable deceleration and then determining the maximum allowable breaking torque. The three options for braking should be combined and optimized up to the allowable deceleration for every crane.

By doing this the ropeforce can be limited as much as possible without adding complicated or expensive equipment.

Rope force reduction

With only the brakes and fast detection it is possible to reduce the rope force with at least 30% for the new APMT cranes on the second Maasvlakte. Depending on the detection- and closing times of the brakes this may even be more. Reducing the inertia of the driveline and limiting the torque increase in the drive will further reduce the force, because these limit the potential energy in the driveline.

Concluding

Snag is a problem and current protection systems have an often non proven working method; are complex, heavy and expensive. Therefore Kalmar and Sibre came up with a new idea for a snag protection system based on early detection and fast stopping. This new proposed snag protection system has a lot of potential.

Sibre is testing their brakes and will come up with final closing times. After development of the detection system by Pat-Kruger, practical test will start, when all exact times are known final calculations can be made to establish the real effect. Testing should be started to determine the bandwidth setting of the snag detection in practice.

An integrated system that works with the brakes, the drives and the crane PLC is important, this way torque increase can be limited for low speed snag and quick stopping can be executed especially for high speed snag.

By taking all into account and by engineering with straight forward common sense, the consequences of snagload can be made acceptable.

Summary (in Dutch)

Het afstuderen en dit rapport focussen op een extreem belastingscenario van container kade kranen. Containers zijn gestandaardiseerde stalen dozen gebruikt voor inter-modulair transport van materiaal over de hele wereld. Om schepen te beladen en lossen worden kade kranen gebruikt.

In schepen zijn de containers opgestapeld tussen cel geleiding. Tijdens het heffen van de last in het scheepsruim, komt het voor dat de last vast komt te zitten. Dit plotseling stoppen van de last noemt men snag en veroorzaakt grote belastingen op de kraan. Er zijn verschillende systemen op de markt beschikbaar, welke beweren deze belasting te reduceren en de kraan beschermen. Van deze systemen is echter niet bewezen dat ze werken en daarnaast zijn ze zwaar, groot of complex en duur. Kalmar is producent van container kranen en maakt zich zorgen om deze snagload. Kalmar zoekt naar een betere manier om de kranen te beschermen en heeft daarom deze opdracht opgezet.

Kabelkracht tijdens snag

De kranen gebruiken kabels om de containers uit de schepen te tillen, dit hijssysteem kan men zien als een lier, motoren drijven via een tandwielkast een kabeltrommel aan. De kabelkrachten zijn een goede indicatie voor snag en de gevolgen. De kabels zijn namelijk de enige connectie tussen de last en de kraan. Een te hoge kabelkracht, bedreigt niet alleen de kabel maar geeft ook een te hoge belasting op de kraan en de mechanische componenten. In geval van snag ondergaat de kabel drie fases: statische belasting, toename als gevolgd van motorkoppel en een toename door rotatie energie.

Statische kabelkracht

Gedurende normaal hijsen is de hefsnelheid constant, daarom is ook de kabelkracht constant.

Koppel toename

Als de last plotseling tot stilstand komt: snag, dan kunnen de kabels de last niet verder omhoog bewegen. De oorzaak van het vastslaan is vaak een beschadigde scheepscel. Op het moment van snag is de lier, de kraan en het hele systeem zich nog niet bewust dat de last tot stilstand is gekomen en proberen daarom de hijssnelheid vast te houden en rekken daardoor de hijskabel uit. De kabels kan men beschouwen als lange veren, de uitrekking resulteert in een lineaire krachttoename in de kabel. De motoren blijven koppel opvoeren, om de snelheid te behouden, totdat het maximale koppel bereikt is, het aangebrachte koppel gaat dan naar nul.

Rotatie energie

De motoren schakelen af als het maximale koppel is bereikt, maar de motoren staan nog niet stil. De overgebleven rotatiesnelheid samen met de massa traagheid resulteert in een grote hoeveelheid rotatie energie welke nog steeds de kabel uitrekt. Pas als alle rotatie energie is overgedragen aan de kabels is de rotatiesnelheid nul, de kabelkracht is dan maximaal.

Deze maximale kabelkracht als gevolg van snag is veel hoger dan is toegestaan en veroorzaakt een belasting op de kraan groter dan de ontwerpbelasting. Daardoor brengt snag de kraan en terminal in gevaar. Tijdens normaal bedrijf is de veiligheidsfactor voor kabels zes maal de normale kabelkracht. In geval van snag is deze factor nog minder dan twee. Daarom is snagbeveiliging onvermijdelijk.

Calculaties en modellen hebben laten zien dat een snag incident, vanaf initiatie tot stilstand, slechts 400 tot 600 milliseconden duurt. Snag detectie en bescherming moeten daarom erg snel zijn.

Belangrijke factoren voor snag

De rotatiesnelheid van de motoren is erg belangrijk voor snag, hoge snelheid betekent namelijk een grote hoeveelheid rotatie energie. De werkelijke snelheid van heffen hangt af van de belasting: een licht last wordt tweemaal zo snel getild dan een zware last. Dit betekend dat bij een lichte last een de gevolgen door rotatie energie groter zijn. De koppeltoename bij zware last is groter doordat de motoren op lage snelheid een groter koppelreserve bezitten en is daarom bij lage snelheid een groter risico.

Bescherming tegen snag belasting

De kabelkracht mag de elasticiteitsgrens van de kabel nooit overtreffen, dat zou namelijk de kabel permanent beschadigen en daarnaast extreme belasting op de kraan en mechanische componenten geven. De elasticiteitsgrens van een kabel is normaal gesproken circa 50% van de minimale breeksterkte van een kabel. In de praktijk, ook bij het ontwerp van de kraan, wordt een kabelkracht van 33% van de breeksterkte als maximum aangehouden.
Het beperken van de kabelkracht en daarmee de kraan beschermen is het doel van een snag bescherming systeem, het doel is om de kracht ver onder de 50% te krijgen, het streven is om 33% van de breeksterkte te benaderen. Daarnaast moet het snag systeem ook de mechanische componenten beschermen tegen een te groot koppel.

Er zijn in principe drie verschillende manieren om de kabelkracht te limiteren tijdens snag:

- Het stoppen van de aandrijflijn
- Toevoegen van kabellengte
- Ontkoppelen van de aandrijflijn.

De laatste twee opties worden toegepast in bestaande systemen. Hydraulische snag bescherming van ZPMC en Rima gebruikt cilinders om kabellengte toe te voegen om zo de kabelkracht terug te reduceren. Pintsch Bubenzer heeft een nieuw systeem ontwikkeld genaamd SOS op basis van een breekkoppeling. Beide systemen hebben nadelen, ze zijn zwaar, groot of complex en duur; toch is de effectiviteit is niet bewezen.

Kalmar en Sibre (een Duitse Remmenfabrikant) hebben daarom een alternatief bedacht: vroeg detecteren van snag & snel stoppen, zonder bijkomende apparatuur.

Vroege detectie

Twee factoren zijn erg belangrijk voor het detecteren van snag: snelheid en betrouwbaarheid. Het systeem mag geen snag missen maar mag ook geen valse meldingen geven, dat zou vertraging betekenen en als dat vaker voorkomt zal de terminal het systeem uitschakelen. Aangezien snag in circa een halve seconde is voltrokken moet het systeem snel werken om nog op tijd in te kunnen grijpen. Twee reële opties lijken het meten van acceleraties en kabelkracht.

Acceleraties moeten gemeten worden op de spreader of headblock, Sibre wil horizontale acceleraties meten. Het idee hierachter is dat door de beschadigde cellguide de last opzij wordt gedrukt en dat dit de snag veroorzaakt. Als men deze verplaatsing kan meten kan snag worden voorspelt en nog eerder worden ingegrepen.
Headblock en spreader zijn tijdens normaal gebruik onderhevig aan veel impact, dit brengt twee nadelen met zich mee: ruis op de metingen en beschadigingen van meet en verwerk apparatuur. Daarom wordt het meten op het headblock van acceleraties of krachten afgeraden.

Kabelkracht meting wordt in het algemeen voorop of helemaal achterop de kraan gedaan. Dit is ver van de optredende snag, circa 100m kabellengte, met als gevolg niet gedetailleerde en fluctuerende meting door het slaan van kabels. Voor kabelkracht meting is de aangewezen plek daarom op de kat. Samen met Pat-Kruger is de ontwikkeling van een snelle detectie opgezet. Het meten, analyseren en signaleren naar het machinehuis zou in minder dan 35ms volbracht kunnen zijn.

Snel stoppen

Zodra de snag gedetecteerd kan de stop worden ingezet. Het doel is om zo snel mogelijk te beginnen met stoppen en zo hard te remmen als toelaatbaar is. Deceleratie veroorzaakt momenten op de aandrijving, deze mogen de toelaatbare waardes van de componenten niet overschrijden. Sibre ontwikkelt extreem snelle remmen, de noodremmen worden momenteel getest deze produceren 90% remkoppel binnen 80ms, vergeleken met 300-400ms voor conventionele remmen. Ook operationele remmen worden ontwikkeld welke nog sneller zouden kunnen sluiten. Een derde optie om te stoppen is door een tegenkoppel in de motor te genereren, dit kan volgens Siemens en ABB binnen 50ms na een noodsignaal.

Voor elk project/kraan zal geanalyseerd moeten worden wat de toegestane maximum deceleratie is, dit bepaald het toe te passen remkoppel. De drie opties voor remmen moeten gecombineerd en geoptimaliseerd worden voor elke kraan, zodoende kan de snagbelasting worden beperkt zonder een duur en complexe apparatuur toe te voegen.

Kabelkracht reductie

Met bovenstaande is onderzocht wat dit kan betekenen voor bestaande APMT kranen op de tweede Maasvlakte. Met enkel vroeg detecteren en snel stoppen kan de maximale kabelkracht gereduceerd worden met 30%. Daarmee komt de maximale kabelkracht al ver onder de elasticiteitsgrens en benaderd de 33% breeksterkte. Het reduceren van traagheidsmomenten en het limiteren van koppeltoename in de motor zal de kabelkracht nog meer beperken aangezien dit de potentie energie in het systeem reduceert.

Concluderend

Snag is een probleem voor container kade kranen, huidige snag bescherming systemen zijn complex of zwaar en duur en ervan is niet bewezen dat ze effectief zijn. Daarom hebben Kalmar en Sibre samen het idee opgepakt van snag bescherming op basis van vroeg detecteren en snel stoppen van de aandrijflijn. Dit voorgestelde systeem heeft veel potentie.

Sibre is momenteel remmen verder aan het ontwikkelen en testen. Pat-Kruger ontwikkeld een detectie systeem, zodra dit systeem klaar is zal met praktijk testen de bandbreedte worden bepaald. Dit zal definitieve detectie en rem tijden geven.

Het is belangrijk dat het snag systeem compleet geïntegreerd is en samenwerkt met remmen, aandrijving en de kraan PLC. Op deze manier kan het systeem snel ingrijpen en remmen bij snag en de koppeltoename beperken voor lage snelheid snag.

Met inachtneming van al het voorgaande en door logisch te engineeren kunnen de gevolgen van snag belasting worden beperkt en acceptabel zijn.

List of symbols and abbreviations

I	Inertia [kgm^2]
i	Gearbox ratio
n	Number of revolutions [rpm]
E	Young's modulus [N/mm^2]
A	Area [m^2]
d	Diameter [m]
l	Length [m]
α	Angular acceleration [rad/s^2]
ω	Angular velocity [rad/s]
θ	Angular displacement [rad]
T	Torque [Nm]
F	Force [N]
k	Spring constant [N/m]
u	Wire elongation [m]
TEU	Twenty foot Equivalent Unit – container size
MVII	Second Maasvlakte (Harbor site in Rotterdam)
STS	Ship To Shore
F.E.M.	Federation Europeenne de la Manutention

Contents

1 Introduction

This chapter is an introduction into my graduation assignment and gives enough background information to understand the project. First the company who initiated the assignment is introduced. Subsequently container cranes will be shortly explained, followed by the actual problem to investigate: "**Snag**" and the systems to protect the crane. The introduction is concluded with the methodology, approach and goals of this graduation.

1.1 Cargotec

Cargotec is the initiating company of this assignment. Cargotec is a Fins company with many subsidiaries; part of the Cargotec is Kalmar, with an office located in Rotterdam. Kalmar offers the widest range of cargo handling solutions and services to ports, terminals, distribution centers and to heavy industry. One in four container movements around the world is handled by a Kalmar solution. This assignment will only focus on Ship to Shore container cranes, this is now part of the Cargotec group, but formerly produced by Kalmar and before that by Nelcon.

Sibre (Siegerland Bremsen) is a supplier of Kalmar and specialized in industrial brakes and drive components. This assignment is executed under Kalmar supervision but in cooperation with Sibre.

1.2 Containers: boxes and ships

Containers are the main mode of global transportation. The containerization started around 1950's. The big advantage is the modality; the standardized measurements and corner castings make handling, storing and transporting them a lot easier and quicker than loose cargo [1]. Containers can be stored on top of the deck or inside the ship´s hull. Inside the hull there are metal strips welded to align the containers and keep them in place, these are called cell guides. Due to standardizations of containers the clearance inside the cell guide can be quite limited to hold as many containers as possible.

The economy of scale really applies to container transportation, especially in these times of high fuel price. Container ships are ever increasing, currently multiple Maersk Triple E class ships are being built and used with a capacity of 18.000 Twenty foot Equivalent Units (TEU) containers. The biggest transport in one vessel so far is 17.603 TEU from Algeciras to the East [2].

1.3 Ship to shore container cranes

To make sure the containers can get on and of the ships the ports are equipped with ship to shore container cranes. Simply said a container cranes grabs the container in the ship's hull with a spreader and takes it to the shore, loading happens in opposite order.

A typical structure of a container crane is displayed in Figure 1, with the names for several components.

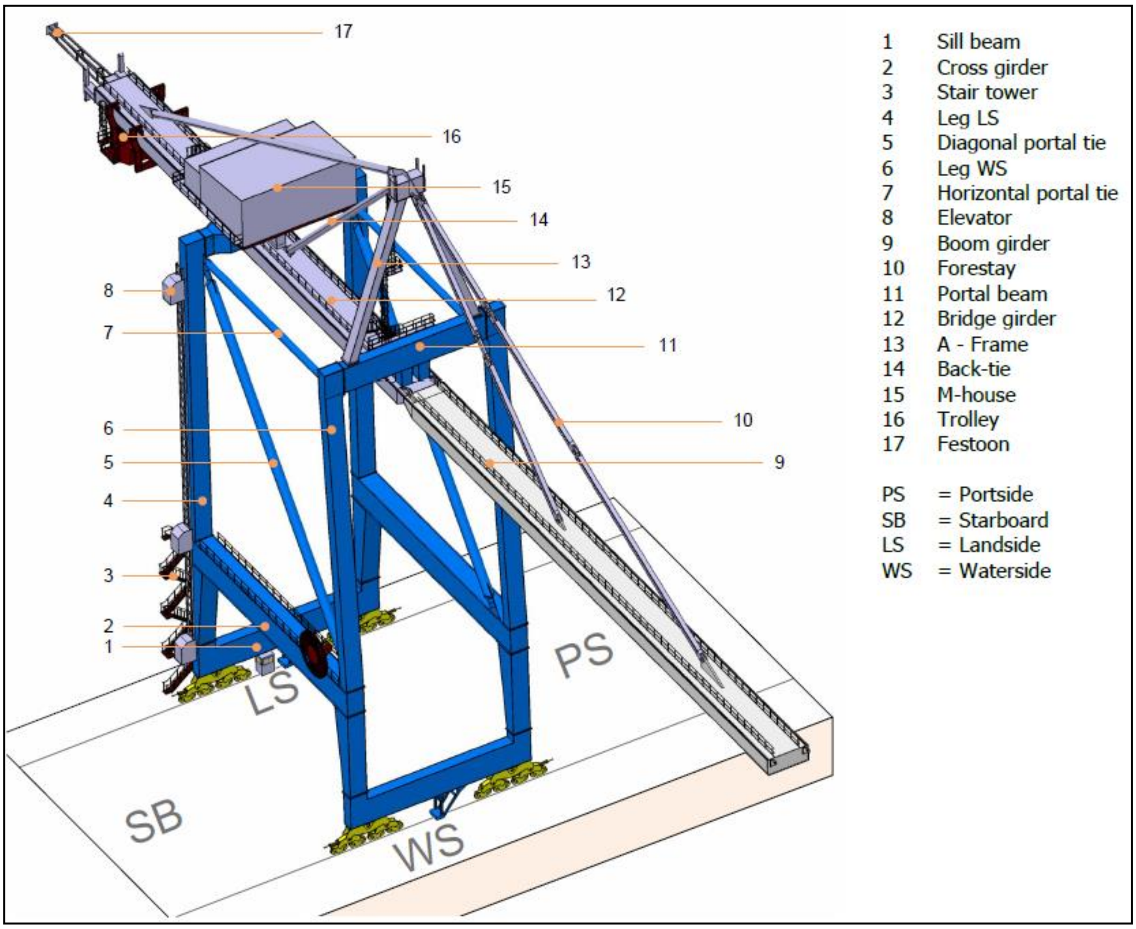

Figure 1: Overview container crane [3]

Depending on the shipsize and requirements of the customer the crane is designed, therefore a STS crane can be a small barge crane up to the biggest container crane in the world for the new Maersk Triple E class. Obviously these cranes are very different. Since the ships are increasing in size and volume also the cranes have to increase in height and outreach. The demand from the ports and shipping lines is to ever reduce throughput times meaning that the hoisting and moving speeds have to increase as well. The hoisting capacity can vary from one up to four containers per move with hoisting speeds up to 3 m/s. Obviously load capacity and speed have big influence on mechanical and structural parts of the STS crane. More about the crane characteristics will be dealt with in chapter 2.

1.4 Snagload

A problem that might occur during the container unloading of a ship is that the spreader and/or container gets jammed in the vessel. This is called a snag load, and this is the topic of this research. As said before containerships contain cells in which the containers are placed inside the hull, these cellguides keep the containers in place. Containers are manufactured according to ISO standard and cellguides have a very small clearance, therefore the allowed container rotation is very limited and if there is a irregularity or dent in the cell guide a container can get stuck and suddenly stop: “snag” is initiated. Figure 2 shows an example of snag.

Figure 2: Snag [Kalmar Archive]

This sudden stop causes high forces in the crane, which of course is unfavorable. There is special equipment available on the market to reduce the overload situation once snag has occurred. Some are a passive system with hydraulic cylinders, others are equipped a break coupling/toque limiter in the driveshaft. These will further be explained in chapter 5.
Previous studies have shown that snag load reduction systems are inevitable with current heavy loads and high hoisting speeds [4].

In case of a snag without an overload reduction system, from the moment a container gets stuck a couple of things happen: first the drive is still turning and stretching the ropes, resulting in more rope tension. Once the motors are shut down at a certain maximum torque (torque becomes zero, speed is not zero!), then there is still energy stored in the inertia of the drivetrain, which also goes into the ropes increasing the rope force. This snag event from the moment the container touching something up to a completely stuck container takes only about 0.5 seconds. The effect of the snag and the consequences depend on a lot of factors, some very important ones are cable length, inertias and speed.

The type and length of the cable determine the energy they can dissipate, before reaching critical values.

1.4.1 Hoist mechanism on trolley or rope trolley

In general a very important deviation can be made regarding the hoisting equipment location: this can be placed on the trolley itself (often done for smaller cranes) or the hoist mechanism can be placed is a separate machine-house with the ropes guided through the entire crane. This is called machinery on trolley cranes vs. full (or semi) rope trolley cranes. This has a large influence on the rope length which is relevant for snag, as B de Vette concluded in his research [4] snag protection systems are unavoidable for machinery on trolley cranes. For rope trolley cranes due to the longer cable length an alternative might be possible [4]. In this report and graduation will first be focused on long cabled (and therefore rope trolley) cranes only.

1.5 A new concept snag protection

Current available snag systems are considered to be expensive and have a not always proven effect. Besides that these systems limit the consequences rather the actively reducing the impact of snag. A new proposal by Kalmar and Sibre is a new Snag Load Protection system, which consist of two parts: 1, earlier detection of snag or even snag prediction. 2, faster stopping. As said everything from detection to interaction has to happen within a very short amount of time to have effect, as guideline take 0.5 seconds for the total event.

1.5.1 Early detection

The idea of detecting snag is currently based on load measuring pins or torque limiters. This means one always measures the consequence of the jammed container.
The idea of Sibre is that by measuring horizontal accelerations and angle changes of the load, the snag can be detected earlier, or even be predicted. Then measures can be taken earlier and the consequences might be reduced.

1.5.2 Fast braking

Once snag is detected, one can start taking measures to reduce the consequences. Since the entire snag event happens very quickly, the response of the entire system has to be very fast, that means the processing of the signal, the activation of the brakes and the closing of the brakes itself. Special brakes are being developed to make this possible.

The question is whether this system will be feasible and effective enough as a snagload protection system. Chapters 7 & 8 are completely about early detection and fast stopping according to this new concept.

1.6 Methodology, approach and goal of graduation

First step in the process is understanding the crane, the working principle and components involved in the snag event. Subsequently we have to investigate snag, how does it occur, what are the different scenarios, where does all energy come from and go to? Calculation and modelling will be used to understand it and make it visible.

The next step will be looking into the different systems to prevent snag or to reduce the consequences. Several products are already available on the market which claim to be the best snag protectors, how do they work and are they effective?

Kalmar and Sibre together came up with a new approach for a snag protection device, early detection and then very fast braking. Focus of this assignment will be on two things mainly:

1) How to detect snag as early as possible (or even predict it)
2) How to stop the hoisting mechanism as fast as acceptable

This entire snag protection system will be further worked out and evaluated in the section "a New Idea".

Finally the complete new concept will be evaluated and compared with conventional systems.

In short are the study goals for this graduation are:

- Understanding of crane
- Understanding of snag
    - Initiation
    - Energies
- Conventional snag systems
    - Working principle
- Detection of snag
    - Possible ways to detect snag
    - Best way: Load vs. acceleration
- Fast Stopping
    - How to limit the forces on the crane
    - Limit rope force
- Evaluation and comparison
    - Conventional vs. New concept
    - Best system related to crane characteristics

2 Crane Characteristics

All equipment and parts related to the hoist function of the crane are explained in this chapter. This is important for an understanding of the components that might be related to the snag events. Focus is on the hoist motion only, because this is relevant for snag. Crane driving, boom hoist and trolley driving etcetera is not involved in snag and is therefore not in scope of this investigation and report.

Every relevant aspect will be dealt with in a separate paragraph and at the end of the chapter a table will be given with a short overview of common values for several components since this is important for later calculations in chapter 4.

A Ship To Shore container crane uses wire ropes to lift the container out of the vessel and onto the quay. There are many different possibilities and configurations possible for the crane but in general it is just a winch with hoisting rope, attached to a huge steel structure [5].

2.1 Load

Ship to Shore container cranes are designed for basically one type of load: containers. Containers are designed and produced according to standard ISO 1496-1 [6]. Containers sizes are expressed in Twenty-foot Equivalent Units in short TEU's. Common sizes are displayed in Figure 3, also 10ft and 30ft containers exist. Most common size handled by STS cranes are 20, 40 and 45 foot containers.

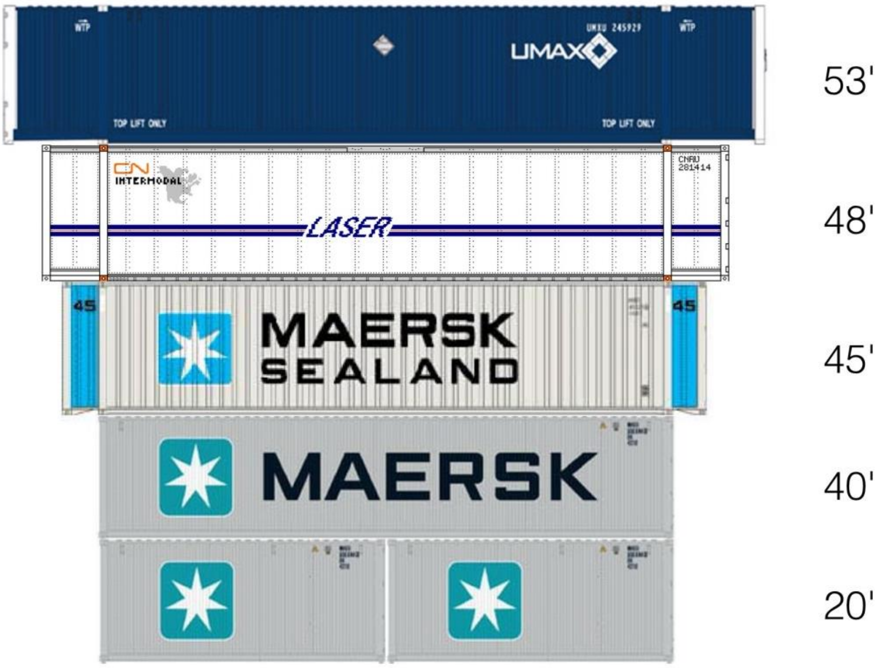

Figure 3: Common container sizes

The standard permissible maximum load of a single standard container goes up to 32.5 metric ton [7]. The main reason why containers are developed and so successful for transport is standardization, all containers have standard corner castings to allow pick up and handling by standard equipment like spreaders.

Figure 4: Container load, spreader and headblock

Figure 4 shows the spreader as the connection between the crane and the container. On the crane side the spreader is attached to the headblock, which is with sheaves connected to the wire ropes. The container is attached to the spreader with use of Twistlocks, a standardized way of connecting containers, displayed in Figure 5. The pin slides in the corner casting and by twisting it locks itself to the container.

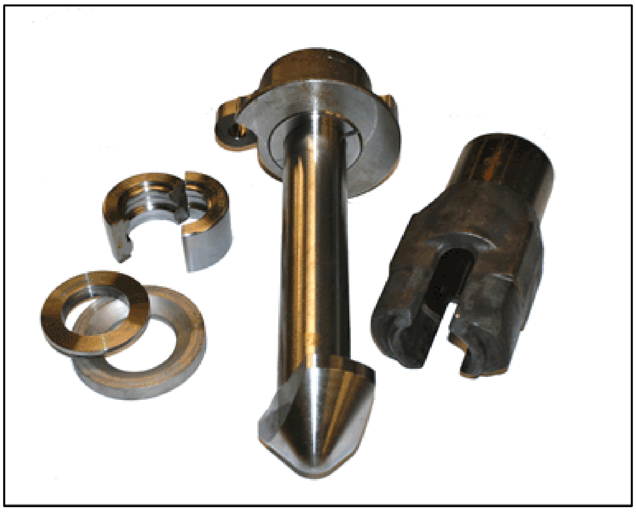

Figure 5: Stinis twistlock set [8]

Spreaders are available in multiple configurations, dedicated to a single size of container or flexible spreader like displayed in Figure 6 capable of handling one 20/40 and 45 foot containers or two 20 foot containers at once, this is currently the most used spreader for STS cranes. New developments have even made it possible to handle up to two 45 foot containers or four 20 foot containers, as displayed in Figure 7.

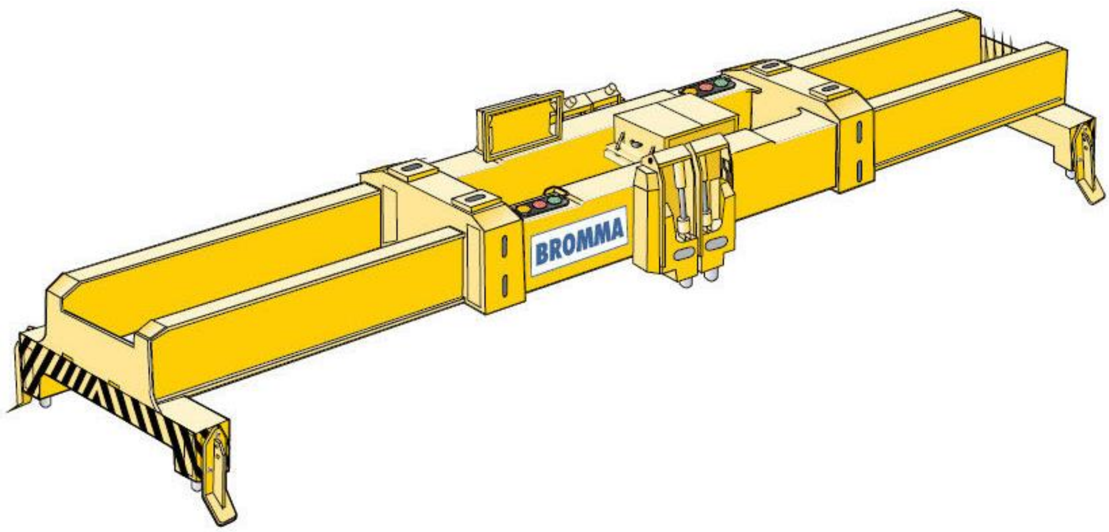

Figure 6: Twinlift spreader, STS 45 capable of lifting: 1TEU, 2x1TEU and 1x2TEU (40/45) [9].

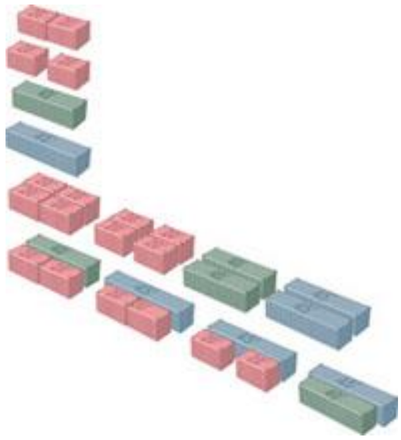

Figure 7: Bromma Tandem Quattro spreader and possible lifting configuration [9].

Sometimes cranes are also equipped with a hook under the headblock to lift special cargo, currently the maximum load under the headblock is 105t for the new APMT cranes at MVII, with the headblock weighing about 9t, total load on ropes becomes maximum 114t.

2.2 Structural

The structural part of the crane is not considered most important for the snag analysis. The crane structure is designed with taking in account overload situations including snag. Changing the crane structurally is not seen as an option to solve snag problems within Kalmar. Therefore this is not in the scope of this investigation and report. Size of the crane is however relevant for the rope lengths as will be discussed later.

2.3 Reeving and ropes

2.3.1 Trolley

Reeving is the route of the wire rope throughout the crane. A major design choice concerns the trolley: a machinery trolley vs (semi-)rope trolley. The trolley is driving on the boom, it drives from land to waterside and vice versa, attached to the trolley is the headblock and subsequently the load.

A machinery trolley is self-propelling and has the hoisting winch mounted on the trolley, this type of trolley has therefore very short hoisting ropes, only from the trolley to the headblock.
A crane with a rope trolley has a separate machine house where the hoist winch is located. If the trolley is pulled forward and back by ropes it is called a full rope trolley; if the trolley is self-driven it is called a semi-rope trolley. For both rope trolleys the hoisting winch is in the separate machine house and the reeving goes throughout the entire crane to the trolley and from there down to the headblock. This type of crane has therefore much longer hoisting ropes.

Figure 8 shows the reeving scheme of the hoisting ropes for a rope trolley crane, upper left one can see the hoist drums, in the middle the trolley and at the bottom the square headblock. Due to the way of reeving the load is carried by 8 pieces of rope. Currently the maximum load on a Kalmar crane is 114t on the ropes, which equals a maximum static load of 140kN per rope.

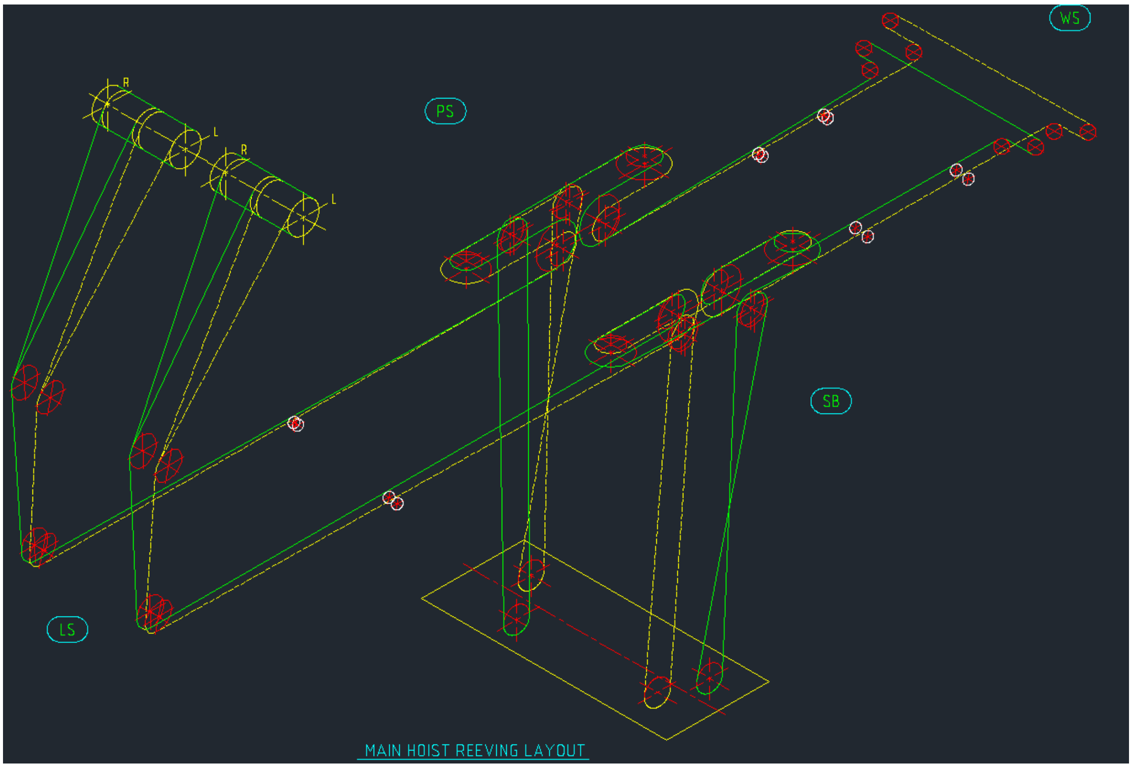

Figure 8: Hoisting rope reeving scheme for rope trolley crane [KALMAR]

2.3.2 Wire rope

Obviously the rope length depends on the reeving as discussed before, varying from approx. 70m (machinery trolley) to 300m (rope trolley). The diameter and specification depends on the design requirements.

The wire ropes work as springs, depending on the load they stretch, the spring constant can be expressed as: $k = \frac{EA}{l}$ where E is the Young's modulus, A is the effective cross sectional area of the rope and l is the length. The force in a rope is equal to the spring constant times the elongation: $F = k\,u$.

Safety factors: European standard FEM

There is a European standard: The Federation Europeene de la Manutention [FEM]1.001 which gives rules for the design of hoisting appliances, booklet 4 describes checking for fatigue and choice of mechanism components. This booklet also describes, related to the mechanism group, what safety factor is required when selecting cables, a minimal breaking load of cables related to the tensile force in the cable. During normal operation a safety factor of 5 to 6 is required, for extreme cases which seldom occur safety factor 3.35 is applicable [10]. With these safety factors and the load spectrum of the crane suitable wire rope is chosen. This means for a maximum load of 140kN the rope selection is a minimal rope strength of approximately 840kN.

According to technical manual of Teufelberger, a wire rope supplier, the elastic limit of their wire ropes is about 50% of the minimal breaking strength [11]. They state that in no case this limit may be exceeded. For the worst case scenario of snag this limit must be used. In practice is striven to not exceed 1/3rd of the breaking strength for extreme load cases. These values are very important for designing and selecting snag protection systems.

Due to the long cable lengths there is a sagging of the cable, causing whipping of the cable during hoisting. Due to the spring working of the cable it takes time to transfer the force from one place to the other end of the rope.

2.4 Hoist Mechanism

The drive train of the hoist mechanism is schematically displayed in Figure 9, in fact it is nothing different than a winch. The figure is not on scale, but gives a good impression. Starting at the motors, obviously driving the system, in this configuration there are two motors installed and attached to the so called high speed shaft or ingoing shaft (RED) by a clutch/coupling. The ingoing shaft is equipped with operational disk brakes (in blue) to keep the load in position when the motors are switched off.

The gearbox reduces the ingoing speed tremendously and has a low speed outgoing shaft(GREEN). The outgoing shaft is attached to the rope drums responsible for hoisting the load. Both rope drums are equipped with one or two sets of emergency brakes (blue).

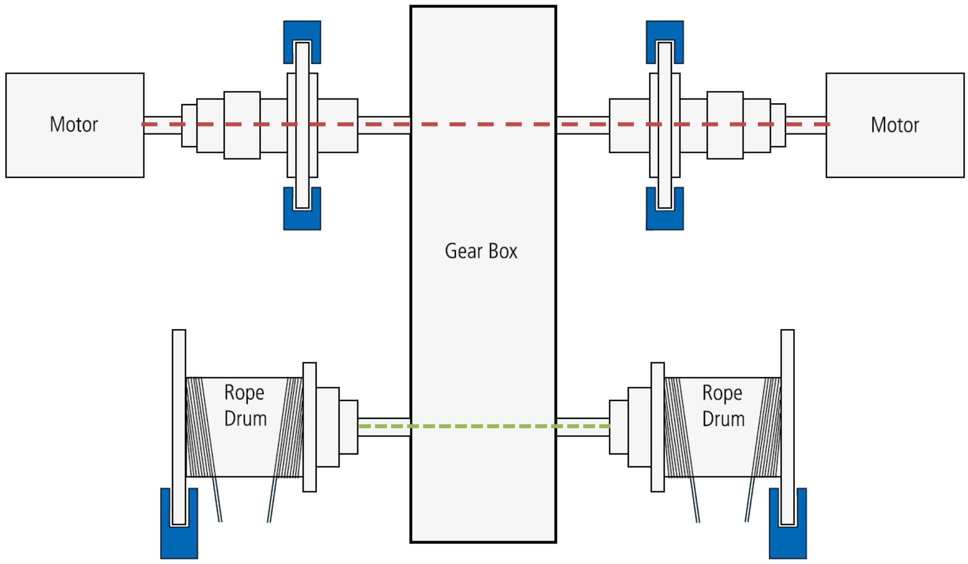

Figure 9: Schematic view of hoist drivetrain, RED: high speed, GREEN: low speed, BLUE: brakes.

Depending on the crane configuration, reeving scheme, desired hoisting loads and speeds one chooses appropriate drums, gearbox, brakes and motors. This can mean one or two drums and one or two engines. With current drivetrains the speed depends on the load actually hoisted. For a light load the empty container or empty spreader the hoisting speed can be twice as high as with full load. As indication speeds up to180m/min (3m/s) for empty containers are currently not an exception anymore!

2.4.1 Motors & Drives

In the past often direct current motors were used, nowadays mostly asynchronous alternating current motors are installed to drive the cranes. These AC motors are fed by variable frequency drives and these are controlled by the crane PLC (Programmable Logic Controller). The crane PLC is in fact the brain/computer controlling the entire crane, the PLC will be further explained at the end of this chapter. Common partners for Kalmar are Siemens and ABB.

Motors

The motors have variable speeds, depending on the amount of load the speed is set, for example the new APMT cranes at Maasvlakte 2, hoist 1.5 m/s with a load of 105mt and 3.0 m/s with 46mt on the ropes. Depending on motor choice the inertia can range from 4 up to 20 kgm^2 per motor.

The capacity of the motors depends, amongst others, on the speed. The motor is designed for a certain speed and related torque available at that speed: nominal speed and nominal power. With a variable frequency drive it is possible to run at faster speed but then less torque is available. Common used motors for STS are Siemens, General Electric and Wölfer. Where Wölfer is known to make very low inertia motors, the relevance of this will be shown in chapter 4.3.

Drives

Variable frequency drives are responsible for feeding the motors, the drive transfers a direct current into an alternating current with a desired frequency. This frequency, by pulse width modulation, is responsible for achieving the desired speed of the motor. Setpoints based on the load and desired speed determine the curve for startup, power input and frequency [12].

The drives-motor combinations installed by Kalmar are speed driven, this means according to load a speed is set. The drive tries to follow this set speed and curve and if the actual measured speed deviates from the setpoint the drive adjusts power to regain the set speed. The response/processing time of a drive internally to a deviating speed is about 15ms. The response time of the drive to an external signal (new setpoint or emergency signal) is about 50ms according to Siemens Engineer Hans Borst.

This combination of variable frequency drives with motors have also the possibility to reverse torque, then instead of putting power to the motor energy is taken from the motor and is fed back into the power grid. Siemens uses their Active Line Module for this. The amount of reverse torque that can be applied depends on the capacity and amount of installed equipment.

2.4.2 Gearbox

The gearbox is responsible for transmitting the motor power from the drives to the ropes. Since the motors run at relative high speeds the gearbox reduces the speed and thereby increases the torque on the rope drum to a required value.

The relationship between hoisting speed and motor rotational speed can be expressed in a simple formula. The rope displacement on the drum is the drum circumference times rotational speed of the drum, which is the gearbox ratio times the rotational speed of the motors: $v_{rope} = d_{drum}\pi \, \frac{1}{i} \, \frac{\omega_{motor}}{2\,\pi}$. Typically the reduction factor in the gearbox is in the range of 10-25.

2.4.3 Hoist Drum

The hoist drum or rope drum is used to actually pull and spool the wire ropes. Obviously the size and diameter depends on the design requirements: hoisting height, required load etc. Typically the diameter of a hoist drum can be up to 1 meter.

2.5 Stopping the hoist

Stopping the hoist motion and keeping the load in place can be done with brakes or by the drive/motor combination, both are explained in this part. There are several processes to stop the hoist, these are related to the reasons of stopping. The types of stopping are put into categories and are explained. Last part explains the braking limitations, it seems logical to stop as quickly as possible but that is actually not always true.

2.5.1 Brakes

The hoist drivetrain is equipped with two sets of brakes, one on the ingoing shaft of the gearbox, called the operational brakes and one set at the drum, the emergency brakes, as was visible in Figure 9, the schematic hoist drivetrain. Both brakes nowadays are clamping brakes, pressing brakepads onto a brakedisk, an example of the current applied operational brakes is given in Figure 10.

Figure 10: Sibre operational disk brakes

The operational brakes are used during normal operation, often the motor with drive takes care of the slowing down (this is possible with the variable frequency driven AC motors), the brakes are applied just before or even after standstill to keep everything in place and as backup/emergency braking.

Often the rope drums are equipped with brakes too: the emergency brakes, if the gearbox would fail, or axles would break, then these brakes can always hold the load. In case of an emergency stop, these brakes are also applied. The brake disk here is actually the drumwall.

2.5.2 Reverse torque

With the modern AC frequency driven motors and drives it is possible to reverse torque. This means that in normal condition the drives/motors apply torque in one direction and that when stopping is desired or an emergency occurs the drives can reverse the power, and apply torque to the shaft in the opposite direction, Siemens and ABB both claim that their drives can change power direction within 50ms. This time is required by the drive and does not include the control and switching time in the PLC.

2.5.3 Stop categories

The crane and control system handles several protocols for stopping, based on the urgency and kind of stop. The different protocols use different approaches/equipment and are described in categories:

Category 0:	Emergency stop, purely mechanical, not help of the motors only on both brakes To be used in case of power shutdown or manually initiated emergency.
Category 1:	Quick stop, uses reversed torque to stop the hoist, purely with the drive/motor. Often stop the hoist quicker than cat. 0 but this requires power.
Category 2:	Normal operation, drive initiates and controls slow down and stop. Mechanical brakes are applied after (or just before) standstill.

2.5.4 Braking limitations

The most logical thing to do seems to brake as quickly as possible, especially in case of an emergency, but that will have some drawbacks. Every part in the drivetrain has mechanical limitations: maximum forces and maximum allowable torques. Force is equal to mass times acceleration, the same holds for rotating parts, the moment of inertia times the radial acceleration equals the applied torque: $M = I\alpha$. Take for example the Motor and high speed shaft, the motor has a large inertia, if theoretical the motor is suddenly stopped (very high (de)celebration value) the moment in the shaft is also very high and this might break the shaft. This obviously has to be avoided and kept in mind when designing the brake protocols and a snag protection system!

2.6 Crane Brain

A Programmable Logic Controller (PLC) is the brain of the crane, this "computer" collects data from all sensors of the crane and uses the input and settings to control and drive the crane. Nowadays the cranes are equipped with many sensors, signals and protocols, to measure load, speeds, wind etcetera and protect the crane, people and terminal. This extensive monitoring and control has as consequence that the PLC needs quite some time to process everything. There is a certain delay in decision making and controlling.

Sometimes a crane is equipped with dedicated fast PLCs or emergency PLCs, these PLCs have a designated task and use only a few input/output channels and are therefore capable of responding much faster.

Table 1: Indication of relevant data of former Kalmar projects

Hoist load	1x1 TEU	32.5t
	1x2 TEU	52.5t
	2x1 TEU	65t
	2x2 TEU	105t
	Hook	105t
Hoisting speed	Depending on load	30-180m/min
Motor	Speed	900-2000rpm
	Power	300-750kw
	Inertia	5-20 kgm^2
Operational Brakes	Brake torque	9500-12000Nm
Gearbox	ratio	10-27
Emergency Brakes	Closing Time	300-400ms
	Brake torque	80000-160000Nm
Ropes	Length	70-300
	Diameter	26-31mm
	Young's modulus	Ca. 1.05e11
	Min. breaking strength	500-900kN

3 Snag event

This report is not an investigation into the reasons for snag neither into avoiding snag. Snag happens! This assignment is about understanding snag and how to reduce the impact with possibly a different system.

This chapter starts with the likely cause of snag, obviously this depends on a lot of variables but a general description is given here, followed by a simple snag event overview. Subsequently will be explained what the consequences of the jammed hoist are for the hoist mechanism, including a calculation and analysis of the energies in the next chapter.

Figure 11: Snag [Kalmar archive]

Snag is considered to only be the jamming and sudden stopping of the load during hoisting inside the ship, Figure 11 shows an extreme example. The case of overloaded containers or container attached to ship is therefore not taken into account in this report. This is an overload during start of the lifting and something completely different than sudden stopping.

According to an article in Cargo Handling there are six important factors that affect snag load: [13], behind the factor is indicated how this is represented in this report.

1) **Rotating components**: Inertia's and speed in the drivetrain
2) **Control design**: response times, plc protocols, overload detection
3) **Ropes**: Length, E-modulus, diameter, minimal break strength, safety factors.
4) **Brakes**: Closing times, friction factors and applied clamping force
5) **Centric or eccentric snag**: how many ropes are being stretched during snag
6) **Snag protection device**: is there a system or device applied to reduce effects

All these aspects have to be taken into account and together determine the snag event consequences. Additional there is also a large influence of the speed and loads. First take a closer look into the actual cause of snag.

3.1 Snag initiation

There are many possible causes for a container or spreader to get jammed, for example getting stuck underneath a hatch, or other container. According to amongst others Kalmar, Sibre, Rotterdam Shortsea Terminals (RST) and Europe Container Terminals (ECT) the most common cause of snag is believed to be getting stuck in the cell guides due to some damaged cell guides.

3.1.1 Cell guides

As explained in chapter 2.1 containers are the common load of STS cranes and are designed according to ISO 1496-1 [6]. Germanischer Lloyd's set Rules for Classification and Construction of ships and cell guides [14]; Rules I describe ship technology, part 1 concerns with seagoing ships, chapter 20 is about stowage and lashing of containers. Lloyd describes the Below-Deck Stowage of Containers in cell guides: "C.1.5.1. Clearance of standard containers in guide rails shall not exceed 25 mm athwart-ships and 38mm in the fore-to-aft direction. Maximum clearance in the fore-to-aft direction includes the deformation of the cell-guide system itself. Where containers are stowed in less than six layers, larger clearances can be permitted, provided container strength has been proven to be sufficient." This shows that the clearance is very limited compared to the size of a container. The flange thickness of the cell guides should be at least 12mm.

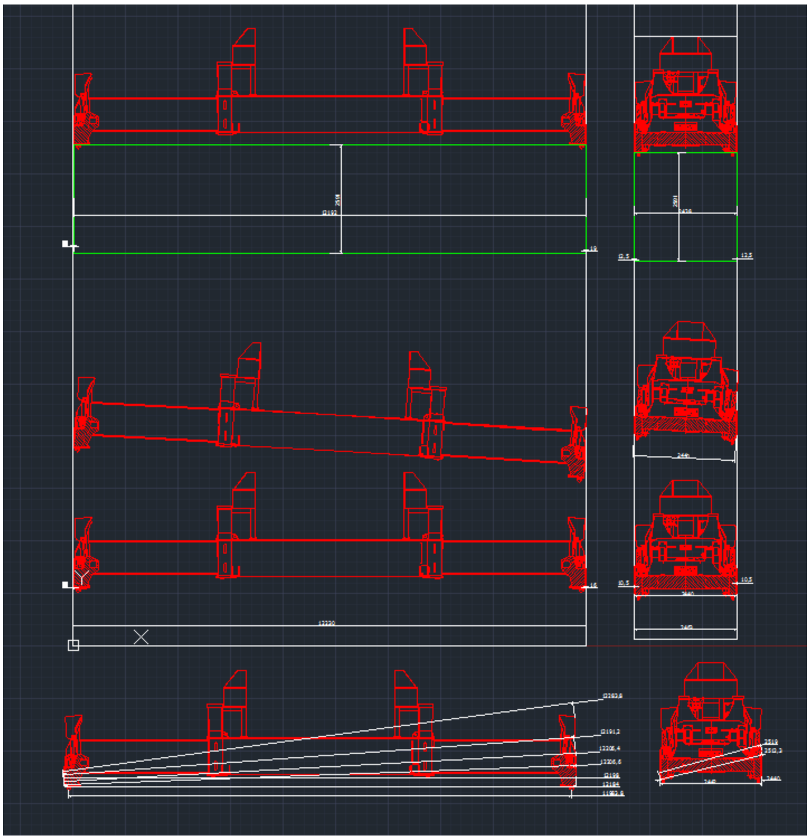

Figure 12: Cell guide clearance

To investigate whether it is possible for a spreader, with or without a container, to snag inside the cell guides, a drawing in AutoCAD is made, see Figure 12. This drawing shows that a spreader fits diagonal in a ship cell in longitudinal direction, however the flipper (the corner guide on the spreader) might get stuck depending on the walls and location of stiffeners. Crosswise the diagonal of the spreader will not fit in the cell guide. With a container attached to the spreader (drawn in green), the possibility to rotate is very limited (<1 degree) due to the small clearance.

Note: it must be mentioned here that this is in the perfect world. In real life it could for example happen that a flipper may extend and thereby gets stuck in the cellguide.

3.1.2 Cell guide damage

The previous showed that there is very limited room to rotate with a container in a ship's hull, this means snag is not easily caused by simply rotating. According to several terminals often snag is initiated by a cell guide disturbance: a dent or damaged cell guide near a stiffener. This means on one side the container or spreader is blocked or slowed down and starts rotating but due to the very limited clearance the container gets stuck on the other side as well side very soon.

The rotation of the spreader gives the number of ropes affected by the snag, indicated in Figure 13. If in theory the container is stopped at one side only the two left ropes are stretched at first. Pure 4 rope snag might for example occur when hitting a hatch. In practice often snag is initiated at one side due to the cell guide disturbance, because of the small clearance it soon turns into 4 ropes being stretched.

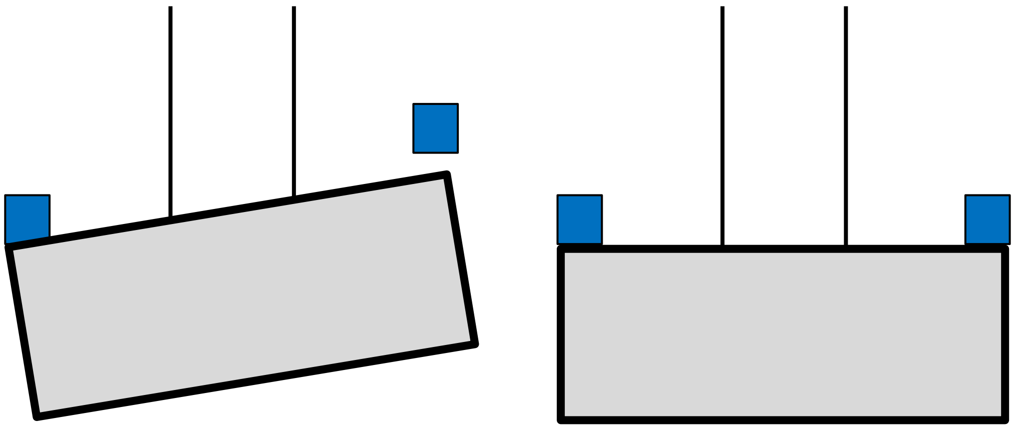

Figure 13: Eccentric (2 rope) vs. centric (4 rope) snag

3.2 Snag event overview

Once the snag is initiated a couple of things happen in time, this paragraph gives a simple overview of that. The easiest and most clear understanding can be achieved by looking at the ropeforce.

Figure 14 gives an indication of the trend of the ropeforce related to time in case of snag. Start at the left:

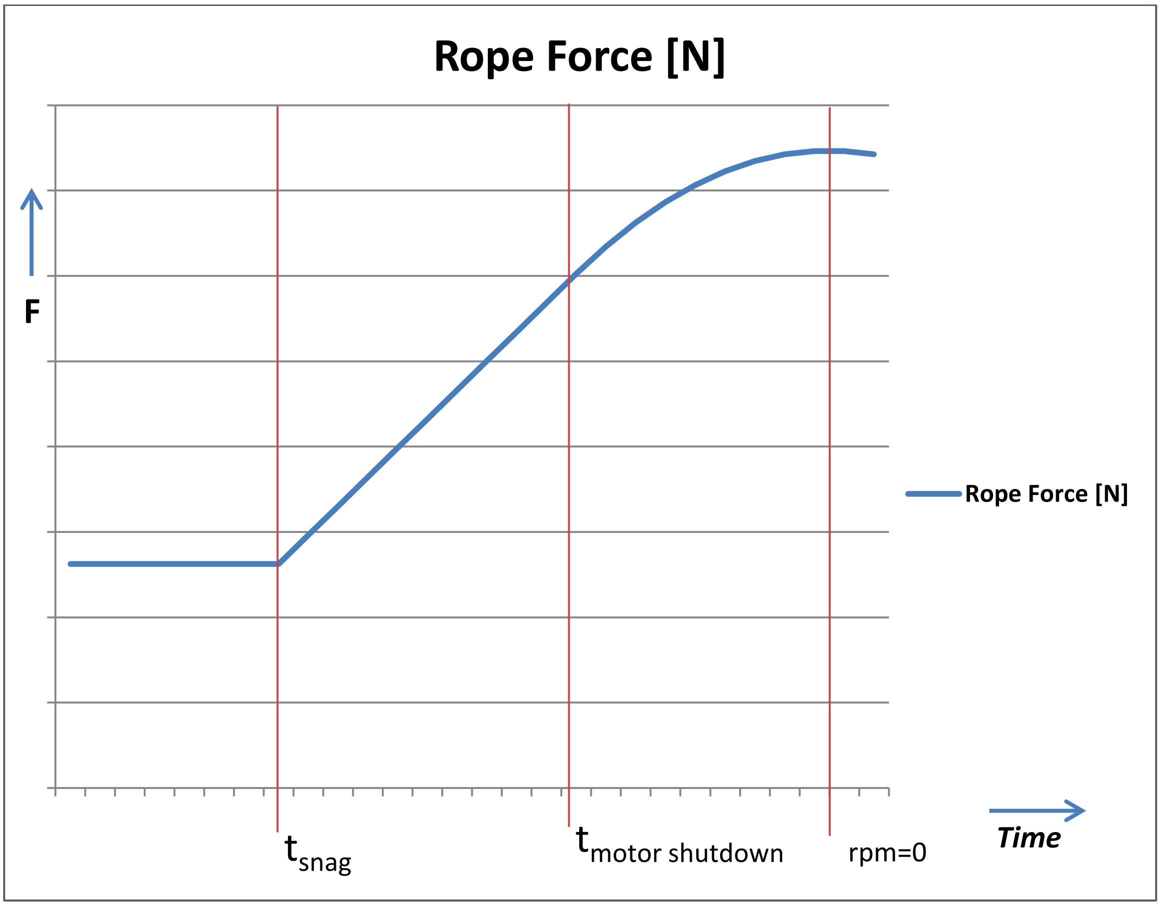

Figure 14: Rope force throughout snag event

$t<t_{snag}$: Torque is constant

Snag has not been initiated yet, the hoisting speed is still constant. The rope force is therefore also constant and equal to the static rope force; everything is in equilibrium.

t_{snag}: Torque increase to maintain speed

At this instant the container gets stuck; snag is initiated.

This in not noticed by the operator or crane plc directly, the crane therefore remains in normal hoisting mode. Because the container is stuck, the rope length cannot change at the headblock, but the hoist drum still pulls on the rope thereby stretching it and increasing the rope force according to $F = k\,u$, where k is the spring constant, and u is the elongation depending on the hoist speed.

The force on the hoist drum increases and tries to slow down the motor. The drives notice this slowing down and try to maintain set speed by adding power. And keeping the speed constant until maximum torque is reached.

The rope elongation is equal to the speed on the hoistdrum multiplied by time and therefore linear. The spring constant is constant at this length and therefore the force increase is a straight linear line.

$t_{motor\ shutdown}$: Maximum torque: motor shutdown

After a while the maximum torque the drive/motor can deliver is reached and the motor shuts down, this means the applied torque goes to zero. All rotating parts still have a rotational velocity and therefore potential energy related to their inertia.

All this rotational energy is transferred into further stretching the ropes. The ropes apply a force to the hoistdrum, creating a moment and thereby slowing down the drivetrain, however the ropes are being stretched until the speed becomes zero. At the moment the motor speed is zero the ropes are maximally stretched and maximum rope force is reached.

This maximum rope force should at all times be less than the elastic limit of the ropes: no more than 50% of the minimal breaking strength. In practice is striven to keep this value below 1/3 of the breaking strength.

4 Energy and rope force calculation

Energy is involved in the snag event, at the moment before the snag there is kinetic energy in the load moving upwards and in the rotating bodies of the drivetrain. Once snag has occurred the load is put to a hold by transferring its kinetic energy into the vessel/cellguides. It is assumed that all kinetic energy of the load goes into the ship. The rotation of the drivetrain/hoist drums is not yet put to a stop and therefore keeps on stretching the ropes as explained in chapter 3.2. All this additional kinetic energy of the drivetrain doesn't go into lifting the load but directly into stretching the cables. Until the drives shut down there is energy added to the system. Once the motors are shut down, the torque goes to zero but there is still residual speed. The rotational energy in the inertias of the rotating parts is then transferred into the cables until the motors reach speed zero, at that moment the cable force has reached the maximum value. The rotational speed of the driveline determines how much energy is in the system.

The energy concerning the ropes during the snag event can be represented by three simple formulae:

E_{drives}	$= T \times \theta$	- The energy added by the drives till maximum torque is reached
$E_{kinetic\ rotational}$	$= ½\ I\ \omega^2$	- The energy stored by inertia of rotating components
$E_{poptential\ spring}$	$= F\ u$	- The energy related to rope elongation

In this chapter the calculation of these energies and the rope elongation and force is worked out. It is important to understand how much energy is in the system and all timing related to the snag event so an alternative drain for this energy can be found.

The energy calculation is split into three parts: first is explained how the rope takes up energy, secondly is calculated how much energy and ropeforce is added until the motor shuts down and last is a calculation of the rotational energy related to the inertia and residual speed.

4.1 Energy absorption by ropes

Since a snag event happens very quickly (in about 0.5 seconds) normal brakes will not apply early enough to take up energy. This means if no additional snag protection device is installed all energy has to be absorbed by the ropes.

It was already said that ropes act as long springs and potential energy of a spring is related to the spring constant (k) and the elongation (u): $E_{spring} = \frac{1}{2} k\ u^2$. The number of snagged ropes tell how many ropes share the total amount of energy. The spring constant of a rope is determined by the Young's modulus, the area and length: $k = \frac{EA}{l}$.

The calculation of energy is a bit too generalized since there is a certain amount of pretension in the cable. Therefore a better formula is related to the rope force and additional displacement. Since the rope force keeps on increasing this has to be an integral:
$E = \int F_{rope}\, \Delta u = \int (ku)\, \Delta u.$

4.2 Motor energy during torque increase

The sequence of events during snag was already explained, first thing that happens after jamming of the load is a torque increase in the drive/motor to try to maintain speed. This is done until maximum torque is reached or until a safety setting in the drive shuts it down first.
As long as the motors have not shut down energy is added, this is equal to the amount of torque times the rotational displacement: $E_{motor} = T_{motor}\, \theta$. Since the torque in the motor keeps on increasing an integral is better:$E_{added\ before\ shutdown} = \int T_{motor}\, \Delta\theta$.
With motor shutdown is meant maximum torque is reached, at that moment the applied torque goes to zero and the motor remains free spinning by inertia.

4.2.1 Limit added Energy

As can be deducted from the formula there are two options to reduce the added energy: limit the amount of rotations or limit the torque. Limiting the amount of rotations depends on when the drive shuts down and that depends on the shutdown/maximum torque, therefore not much can be done about the rotations. The torque increase is however something that can be taken into account, by monitoring the drive and allowing just a small increase of torque to maintain speed instead of the maximum torque. This would mean earlier shutdown, limiting the torque and the rotations and thereby the energy.

As example for the new APM cranes at the second Maasvlakte: in case of lifting a load of 46 ton at 3m/s the constant torque is about 3600Nm. if the drive would shut down after an torque increase of 30% it would shut down at 4700 Nm instead of a maximum torque of 5730Nm.

4.3 Rotational Energy by inertia

Once the hoist motors are shut down, the addition of energy into the drivetrain stops, but the rotational velocity is not equal to zero yet meaning that there is still energy stored in the rotating components. The rotating kinetic energy is related to the moment of inertia of the parts and the velocity: $E_{rotational} = \frac{1}{2} I\omega^2$. The inertia of the drivetrain is a constant based on the choice of components, the initial rotational speed is set based on the load. Reducing the hoisting speed reduces the energy directly, especially since it is to the second power. However terminal and shipping lines keep on pushing to increase the speed so this is not an attractive option. This leaves a wise choice of drive components.

4.3.1 Choice of components

The components determine the inertia therefore one should choose carefully. If we look at for example look at the hoist drum on the outgoing shaft: the drum has a very large inertia but a very limited rotational speed. On the outgoing shaft: $E = \frac{1}{2} I \omega_{out}{}^2 = \frac{1}{2} I (\frac{\omega_{in}}{i})^2 = \frac{1}{2} I \omega_{in}{}^2 \frac{1}{i^2}$ with a reduction factor of 15, the inertia on the high speed shaft has 225 times more energy per kgm^2 compared to the low speed shaft. This shows the relevance of selecting low inertia rotating parts especially on the ingoing/high speed shaft thus the motors, coupling and brake.

Worst case is that all this rotational/inertia energy goes into the ropes, no brakes are applied nor braking with the drives. With this we can calculate the rope maximum elongation and thereby also approach the deceleration and time used to the stop the rotation. This time tells us the available time to interact if a snag protection system would be applied.

4.4 Basic calculation

Now the energy sources and rope energy absorption formulae are known a basic calculation can be made. For this the crane data is taken of the new APMT cranes currently under construction at the second Maasvlakte. These are the biggest STS container cranes built at the moment.

Table 2 on the next page shows the relevant data of these cranes used for calculation. The calculation is split in 3 parts: 1. The static load, defining the pretension; 2. The increase of motor torque; 3. Rotational energy.
The calculation is done for 4 snag scenarios:

1. 2 snagged ropes, 46 ton load, 180m/min hoisting speed
2. 4 snagged ropes, 46 ton load, 180m/min hoisting speed
3. 2 snagged ropes, 105 ton load, 90m/min hoisting speed
4. 4 snagged ropes, 105 ton load, 90m/min hoisting speed

During this chapter the calculation is shown for scenario 2. At the end of the chapter the data is for all 4 scenarios and a force comparison is given.

Table 2: Relevant crane data SQC APMT MVII

Hoist load and speed	46t	180m/min
	105t	90m/min
Ropes [15]	**Teufelberger Perfekt**	**QS 816 V**
	Tensile strength	1960N/mm^2
	Minimal breaking load	846kN
	Max. allowable load	50% of breakload [11]
	Nominal rope diameter	30mm
	Young's Modulus	10.5e4N/mm^2 [16]
	Length at snag	260m
Motor	**Wölfer ODRKF 335X-6**	
	Inertia	13.6kgm^2
	Speed	46t: 1800rpm 105t: 900rpm
	Maximum motor torque	46t: 5732Nm 105t: 15422Nm
Motorcoupling	**Malmedie MSC AKNXSE 0.88**	**SOS break coupling**
	Inertia	6.636kgm^2
Operational Brakes	**Pintsch Bubenzer**	**SB 28-1000x30**
	Inertia	8.25kgm^2
Gearbox	Ratio i	15.8
Drumcoupling	**Malmedie TTXs21 Inertia**	13.5kgm^2
Hoistdrum		
	Inertia	946.5kgm^2
	Effective radius drum	0.5m

Additional calculated rope data

$$Area = \frac{minimal\ breaking\ load}{tensile\ strength} = \frac{846e3\ N}{1960\ N/mm^2} = 431.6mm^2$$

$$d_{eff} = \sqrt{\frac{Area}{\frac{1}{4}\pi}} = 23.443mm$$

Spring constant at time of snag: $k = \frac{EA}{l} = \frac{45.3e6N}{260m} = 174.3e3\ N/m\ \ per\ rope$

The maximum allowable load which is considered to be in elastic field of ropes and to not do permanent damage is 50% of the breakload: 423kN.

Inertia

All inertias from Table 2 have been summed together with regard to the gearbox ratio according to chapter 4.3.1, the total inertia of the APMT driveline calculated towards the high speed shaft is equal to 65 kgm^2.

4.4.1 Step 1: Static load

The static load on the ropes determines the pretension on the ropes. The total load is carried by four cables on the drums but due to the reeving the load is spread over 8 parts of rope (2x number of ropes). Resulting in the following static rope elongation and torque:

$$F_{rope\ static} = \frac{Load * \ gravity}{2 * \ \#_{ropeparts}} \rightarrow u_{static} = \frac{F_{rope\ static}}{k}$$

$$T_{drum\ static} = F_{rope} * \ \#_{ropes} * r_{drum} \rightarrow T_{motor\ static} = \frac{T_{drum}}{i_{gearbox}}$$

Indication APMT crane with 46ton load, high speed, 4 rope snag.

$$F_{rope\ static} = \frac{Load * \ gravity}{2 * \#_{ropes}} = \frac{46e3kg * 9{,}81m/s^2}{2 * 4} \rightarrow u_{static} = \frac{56.4e3N}{174e3N/m} = 0.32m$$

$$T_{motor\ static} = \frac{F_{rope} * \ \#_{ropes} * r_{drum}}{i_{gearbox}} = 7.14kNm = 3.57kNm\ per\ motor$$

4.4.2 Step 2: increased motor torque

A maximum motor torque per motor is given, due to equilibrium this has to be transferred onto the ropes; therefore it can be calculated into force in the number of snagged roped (2 or 4) and then be related to additional rope elongation.

$$T_{drum} = T_{motor\ max} * \#_{motors} * i_{gearbox} \rightarrow F_{rope} = \frac{T_{drum}}{r_{drum} * \#_{snagged\ ropes}} \rightarrow u_{add\ step2} = \frac{F_{rope}}{k} - u_{static}$$

The speed is considered to remain constant, that's the principle of a speed driven motor. By combining the hoist speed and added rope length we get the time involved in this step.

$$v_{hoist} = \frac{v_{motor}}{60 * i} * 2 * \pi r_{drum} \rightarrow t_{step2} = \frac{u_{add\ step2}}{v_{hoist}}$$

Note that here is assumed that all force is taken by either 2 or 4 snagged ropes, more realistic might be distributed over 4 ropes but depending on angle in cell guide. Here is however choses for worst case.

Indication APMT crane with 46ton load, high speed, 4 rope snag.

Maximum torque high speed is 5732Nm

$$F_{rope} = \frac{T_{m\ max} * \#_m * i_{gearbox}}{r_{drum} * \#_{snagged\ ropes}} = \frac{5732Nm * 2 * 15.8}{0.5m * 4} = 90.6kN \rightarrow u_{add\ step2} = \frac{F_{rope}}{k} - u_{static} = 0.20m$$

$$v_{hoist} = \frac{v_{motor}}{60 * i} * 2\pi r_{drum} = \frac{1800rpm}{60 * 15.8} * 2\pi * 0.5m = 5.96\frac{m}{s} \rightarrow t_{step2} = \frac{u_{add\ step2}}{v_{hoist}} = \frac{0.20m}{5.96\frac{m}{s}} = 0.034s$$

From the calculation is visible that in case of 4 rope snag the motors are shut down in 0.034s, in this case 4 ropes are being stretched at high speed thus the torque limit is reached soon. When stretching only 2 cables or at lower speed the time involved in the torque increase is much more.

4.4.3 Step 3: Rotational energy

By now the applied torque on the motors is zero, the driveline is still rotating though and the residual speed of the drivetrain resulting in a lot of kinetic energy still in the system:

$$E_{total\ stored\ in\ rotation} = \frac{1}{2} I * \omega^2$$

It is assumed that all this rotational energy has to be transferred into the snagged ropes. Energy absorption by the snagged ropes is equal to the force in the cable times the elongation:

$$E_{total\ stored\ in\ rotation} = E_{ropes} = \#_{snagged\ ropes} * \int F_{rope}\ \Delta u$$

The rope elongation is initially the static stretch plus the elongation after step two, the additional elongation in step 3 determines the absorbed energy as expressed in next formula:

$$E_{total\ stored\ rotation} = E_{ropes} = \#_{snagged\ ropes} \left[k\left(u_{static} + u_{step2} + \frac{1}{2}u_{add}\right)\right] * \mathrm{u}_{add} \rightarrow \text{iterate} \rightarrow u_{add}$$

This total elongation means that all energy is transferred into the rope. At that moment the motor has stopped and the ropeforce is maximal. If by now no brakes are applied the ropes will start to pull back the drums and rotation in reverse direction would start.

Indication APMT crane with 46ton load, high speed, 4 rope snag.

$$E_{total\ stored\ in\ rotation} = \frac{1}{2} I * \omega^2 = \frac{1}{2} * 65kgm^2 * (188.5rad/s)^2 = 1.15MJ$$

This should match the additional energy absorption in the ropes

$$E_{ropes} = \#_{snagrope} \left[k\left(u_{static} + u_{step2} + \frac{1}{2}u_{add}\right)\right] * \mathrm{u}_{add} \rightarrow \text{iterate} \rightarrow u_{add}$$

$$E_{ropes} = 4\left[174e3N/m\left(0.32m + 0.20m + \frac{1}{2}u_{add}\right)\right] * \mathrm{u}_{add} = 1.15\mathrm{MJ} \rightarrow \text{iterate} \rightarrow u_{add} = 1.37\mathrm{m}$$

$$F_{rope} = u_{total} * k = (0.32 + 0.20 + 1.37)m * 174e3N/m = 328kN$$

The maximum rope force for scenario 2 (4 rope, 46 ton, 180m/min) is 328kN. This is below the maximum elastic allowed force of 423kN. Worst case is however scenario 1 with 466kN as will be discusses at the end of this chapter.

4.4.4 Calculation program

Based on this basic calculation an Excel calculation sheet is made. The following variables have to be inserted:

- Rope/spring constant
- Number of snagged ropes
- Static load
- Engine speed
- Total inertia of drivetrain
- Gearbox ratio
- Effective drum radius
- Maximum motor torque

Based on this input the sheet calculates the first two steps: the static elongation, elongation due to torque increase, time during torque increase, kinetic energy of rotating parts and rope force. The form is completed by (iteratively) filling in the field of additional elongation to match the rope energy with the kinetic energy. The sheet uses the values of times and elongations to calculate speeds, deceleration, times and rope force not with integrals but with small steps to make everything visible in time.

The rope elongation is directly linked to the rotation of the drum; this rotation can therefore be calculated according to the following formula:

$$u_{add} = \frac{\theta_{drum}}{2\pi} * circumference = \frac{\theta_{drum}}{2\pi} * 2\pi * r_{drum} = \theta_{drum} * r_{drum} \rightarrow \theta_{drum} = \frac{u_{add}}{r_{drum}}$$

$$\theta_{motor} = i_{gearbox} * \theta_{drum}$$

For step 3 torque on the high speed shaft is equal to the total inertia times the rotational acceleration, the inertia and the amount of torque are known, therefore the rotational acceleration can be calculated:

$$\alpha = \frac{T_{high\ speed\ shaft}}{I_{total}}\ with\ T_{high\ speed\ shaft} = i_{gearbox} * T_{drum} = i_{gearbox} * F_{rope} * r_{drum}$$

For every step the residual kinetic energy in the rotating parts is equal to the initial kinetic energy minus the energy already taken by the ropes. From this residual kinetic energy the rotational velocity can be calculated:

$$E_{kinetic\ residual} = E_{rotational\ initial} - E_{ropes}$$

$$E_{kinetic\ residual} = \frac{1}{2} I * \omega^2 \rightarrow \omega = \sqrt{\frac{E_{kinetic\ res.}}{\frac{1}{2} I}}$$

With the change in velocity and the average acceleration for every calculation step, we can calculate the step time and the total time:

$$t = \sum \Delta t = \sum \frac{\Delta\omega}{\bar{\alpha}}$$

The calculation sheet is shown in Appendix 2 Calculation of Energy. The calculation is done for the four scenarios (2 vs. 4 rope snag; high speed vs. low speed). Figure 15 to Figure 17 show the calculated data for 4 rope snag, high speed as calculated as indication. In the appendix the graphs for all 4 scenarios can be found. Figure 18 shows the rope force of all 4 scenarios in one diagram.

In Table 3 at the end of this chapter the forces and times for all 4 snag scenarios are given.

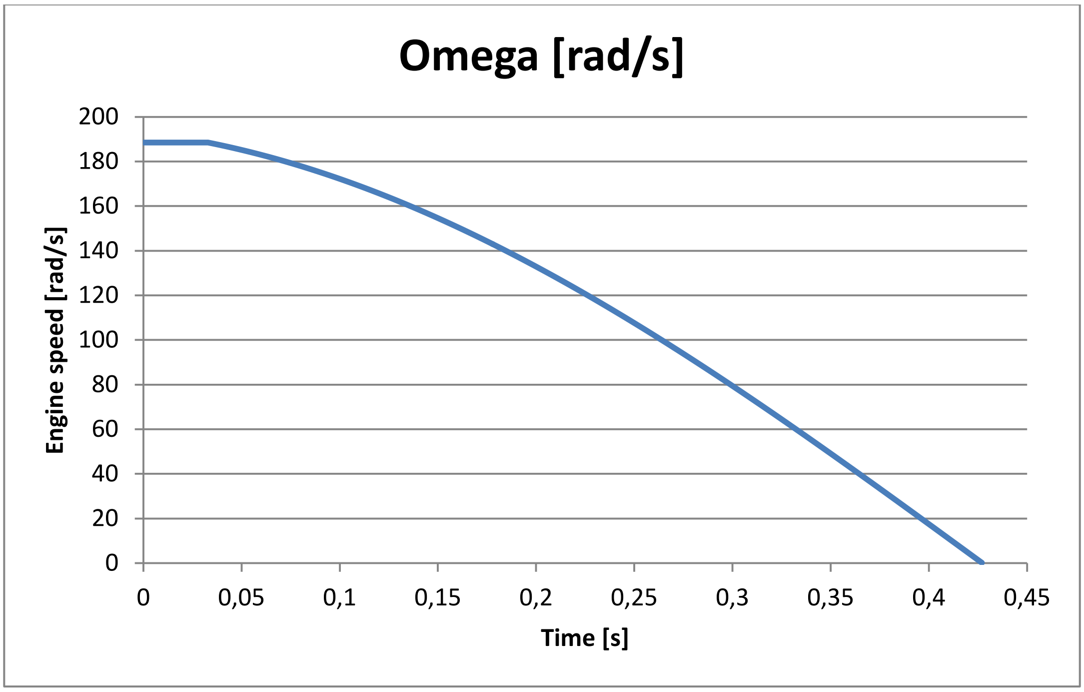

Figure 15: High speed shaft velocity for 4 rope snag, high speed

As can be seen in Figure 15 the rotational velocity is constant from the moment snag is initiated at t=0 until the motor shuts down after 33 milliseconds. From that moment on the ropes slow down the motors, after 0.42 seconds the speed becomes zero.

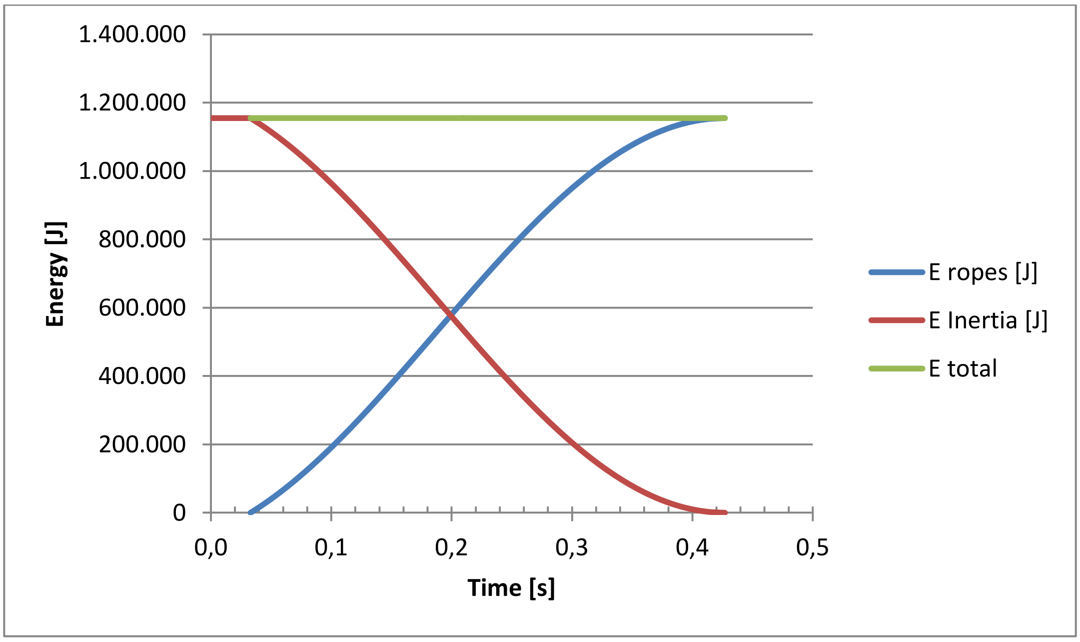

Figure 16: Kinetic energy for 4 rope snag, high speed

Figure 16 shows the kinetic energy in the stored in the rotating components. Until 0.033s the rotational velocity is constant and therefore also the kinetic energy is constant. The ropes start slowing down the motors after motor shutdown, all this rotational energy is then being transferred into the ropes.

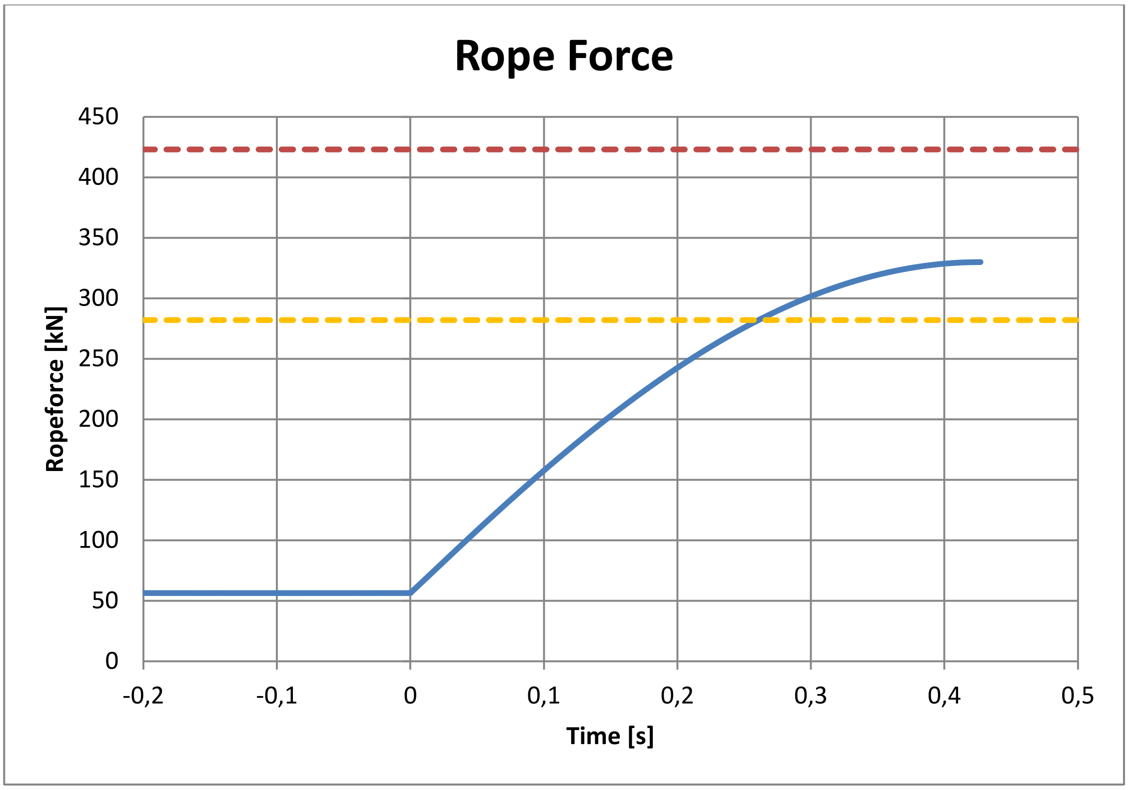

Figure 17: Single rope force for 4 rope snag, high speed

Figure 17 displays the ropeforce in case of a snag event for scenario 4: 4 rope high speed snag. The graph also includes the rope limits, in red the elastic limit at 50% of the minimal break strength, in orange the target limit, 33% of the minimal break strength. With no snag protection this limit is exceeded.

4.5 Analysis of calculation results

Figure 18 shows the calculated ropeforce for all 4 scenarios, in the graph is displayed what the limits for ropeforce are, the red dotted line represents elastic limit: the absolute maximum which may never be exceeded and the orange dotted line shows the 33% line which is the desired maximum.

The previously calculated scenario 2 (4rope snag, high speed) is displayed in red, what is immediately visible is that this is not worst case. Scenario 1 and 3, both 2 rope snags, have a much higher maximum ropeforce. All four scenarios exceed the desired limit and the both 2rope snags even reach ultimate maximum. This means for this crane a snagload protection system is required!

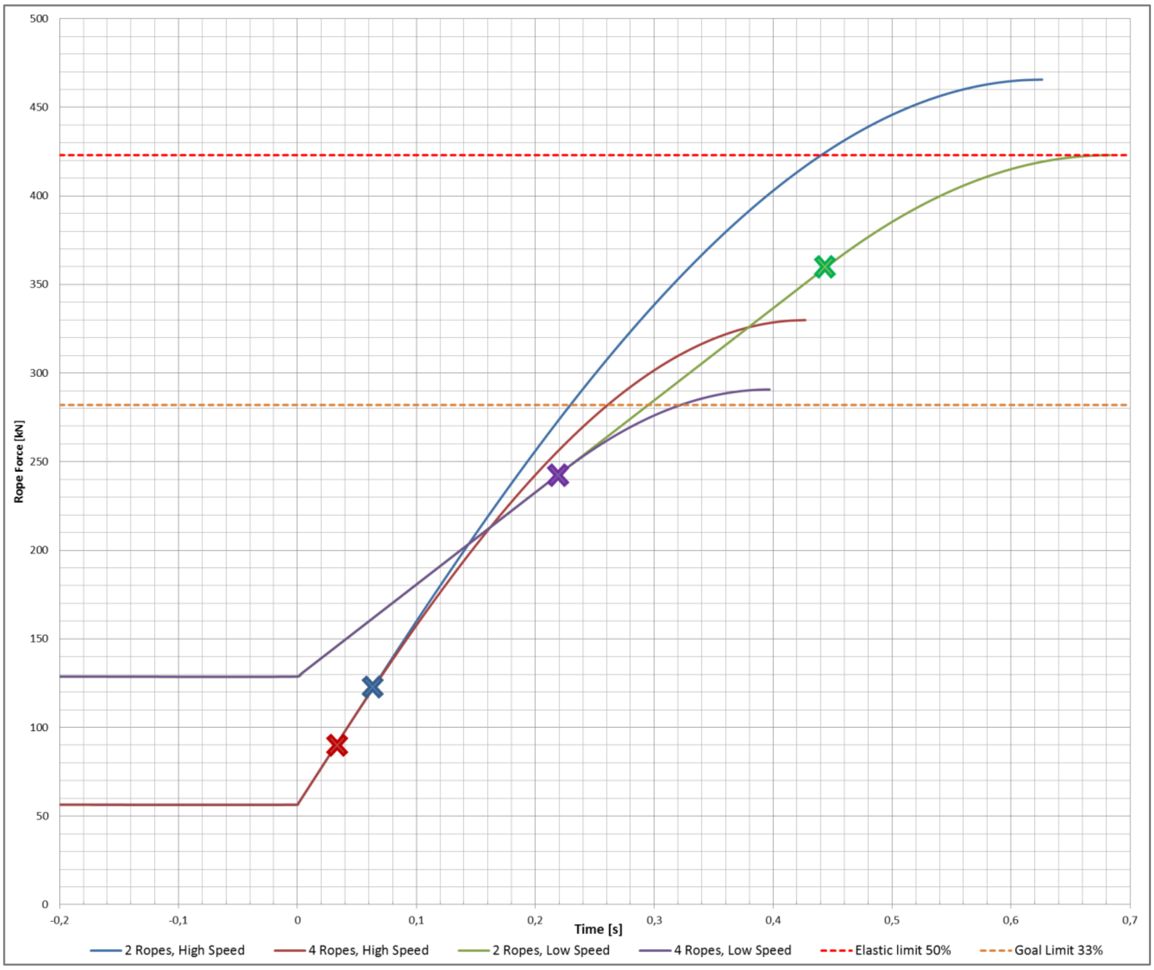

Figure 18: Rope force in case of snag for all 4 scenarios based on AMPT SQC

Table 3: Calculated data for all 4 scenarios

	Snagged ropes	Load	Hoist speed	$F_{rope\ static}$	$t_{motor\ shutdown}$	$F_{rope\ shutdown}$	$t_{speed=0}$	$F_{rope\ max.}$
1	2	46t	180m/min	56kN	0.066s	125kN	0.63s	466kN
2	4	46t	180m/min	56kN	0.033s	91kN	0.43s	330kN
3	2	105t	90m/min	129kN	0.442s	360kN	0.68s	423kN
4	4	105t	90m/min	129kN	0.221s	244kN	0.40s	291kN

Table 3 gives some important data to accompany Figure 18. The crosses on each line indicate when maximum motor torque is reached for that scenario. For both slow speed snags, the time until motor shutdown is significantly larger than for high speed snag, this is due to the fact that the reserve torque is much more in case of slow speed snag and because the force increase factor in time is double for high speed snag, due to double speed. Therefore the relative effect of the torque increase on the ropeforce is much larger in case of slow speed snag. By only limiting this torque increase for slow speed the force in the ropes could already be reduced tremendously.

In practice there is no real two rope snag, it will always be a combination, the 2 rope snag is displayed because it would be the utmost worst case scenario. Still all scenarios reach too high forces thus snag protection is required!

5 Model: multi body dynamics

The calculation in the previous chapter give a good indication of the energies and forces, to get an even better insight and also a visual image of what happens a simulation model is made in MSC Adams. The multi body dynamics simulation gives the possibility to change parameters quickly: sizes, speeds, loads but also applied braking characteristics etcetera. Also by simulating the snag event itself, rather than just assuming the load gets stuck, it gives a good indication of accelerations and forces during the snag initiation.

This chapter starts with a description of the model and the components, followed by a validation of the model by comparing it to hand calculations. Subsequently all parameters of the final model for an analysis are given with their results.

5.1 Model and components

MSC Adams is a multi-body dynamics simulation tool developed for dynamic analysis of moving parts and interaction. A simulation model is created to get a visual understanding of what happens with snag, this model also allows fine tuning of applied forces and torques. The model is better than the basic calculations for taking into account the working of several parts of ropes, multiple sheaves and their efficiency and the actual snag initiation. Of course a simulation is never identical to the real world but this give a lot of insight and valuable information.

For the model the dimensions of STS rope trolley cranes are used, the actual steel crane structure is not modelled, this is not part of the snag analysis. The analysis is limited to the mechanical components: the drivetrain, reeving and load. Figure 19 shows an overview of the crane model. The reeving, dimensions and components are according to the new APMT cranes as used in the previous chapter.

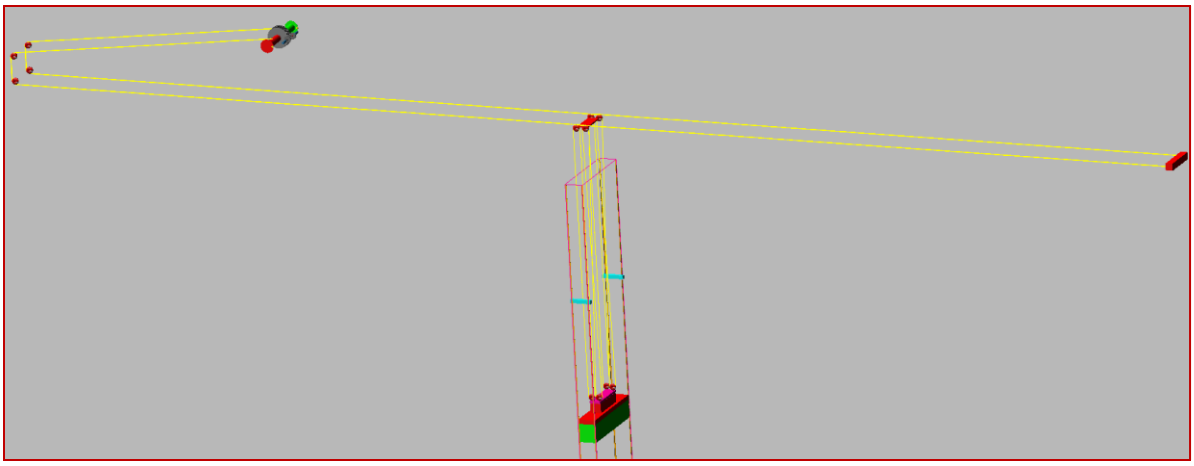

Figure 19: Overview of crane model in MSC Adams

5.1.1 Drivetrain

Since it is a rope trolley crane the model has a separate machine house with the hoisting equipment. Figure 19 shows the location of the machine house at the top, this is where the ropes (in yellow) start. A close up of the driveline is given in Figure 20. As can be seen all components as discussed in chapter 2.4 are present. The blue bars represent the rotors of the motors on the ingoing high speed shaft, in dark grey the coupling, followed by the grey brake disk with in yellow the brake pads. The gearbox is simplified in the model, only transferring torque and speed according to a ratio, in reality the gearbox contains multiple gear sets. Both ropedrums are connected to two hoistingropes each as displayed by the yellow strings. The outer drumwall is the emergency brakedisk with in yellow the brakepads.

Brakes

The brakes in the model cannot be based on applying torque, because this would mean that after standstill the brake torque tries to accelerate the motors in the opposite direction. By applying clamping force on brakepads, times the friction factor and radius, we get a brake torque. Which is actually a more realistic approach and keeps the drive in place after standstill.

Motors

In the model motor (and therefore speed) is controlled by applying torque, this is preferred over defining speed. This allows applying zero torque and letting the motors "freewheel", this is not possibly by defining velocity: speed zero would instantly stop the motors and defining a free speed is not possible.

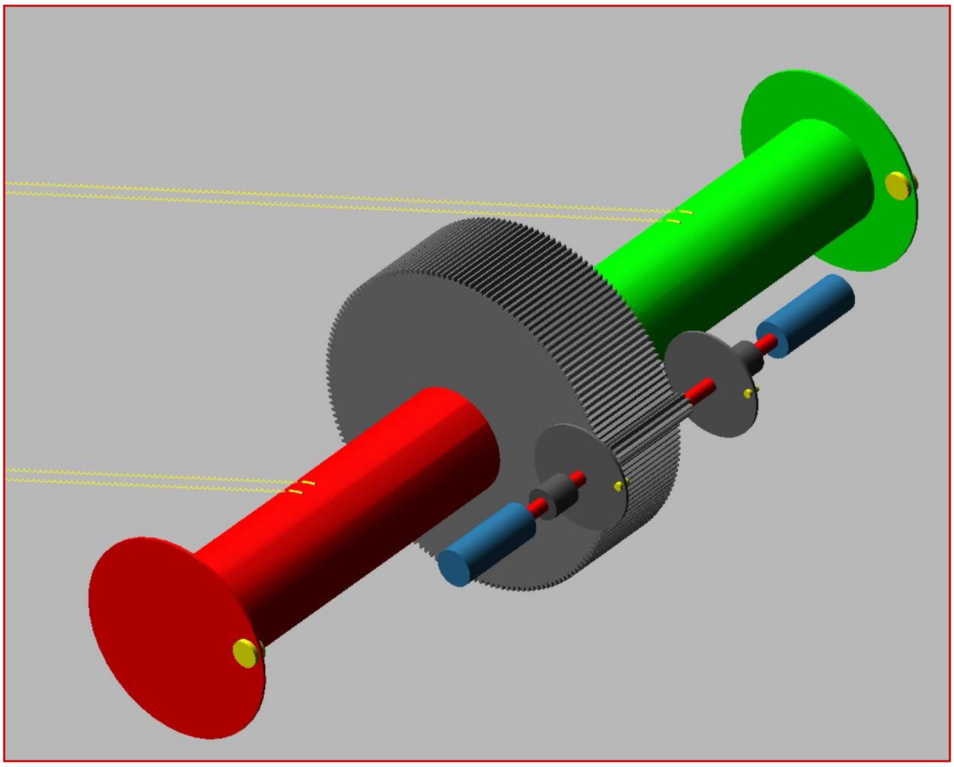

Figure 20: Close-up of the driveline as modelled in MSC Adams

5.1.2 Trolley and load

Figure 21 shows a close-up of the trolley, spreader, load and cellguides as modelled. The trolley/headblock connection is made with 8 rope parts and 12 sheaves, this is according to the reeving of real cranes. This way of reeving makes that the outer rope parts, the ones close to the cellguides, don't move; the ropeparts in the middle have a speed of two times the hoisting speed. This reeving also allows driving with the trolley without lowering or hoisting the load.
As one can see the trolley, headblock, spreader and container are drawn simplified, all outside dimensions and weights are however realistic.

5.1.3 Cellguides

The cellguides and clearance are modelled according to Lloyd's register [14], as discussed in chapter 3.1.1: 150x150 mm corner profiles with a flange thickness 12mm are used. It is important to model the cellguides and the disturbance since this is thought to cause the snag. By trying to model as realistic as possible, forces, impact, decelerations and times can be found. This is very important for understanding how to detect snag and the requirements for the detection sensors as will be discusses later.

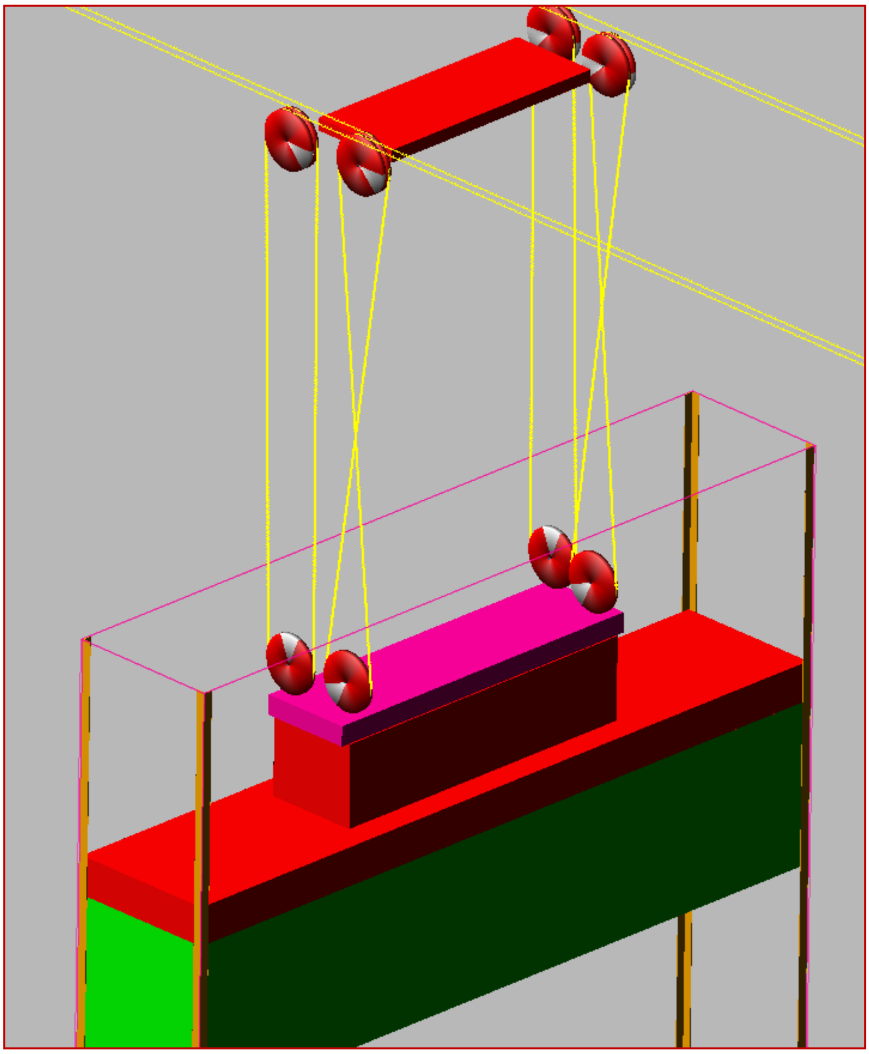

Figure 21: Trolley, Load and Cellguides

Cellguide disturbance

The cellguide disturbance is considered to be the most occurring cause of snag. The disturbance is modelled as displayed in Figure 22. This shows that the side of the cellguide (the side in contact with the short containerside) is split in two parts, determining the depth on the dent. By adding a third part (in light blue) the cell disturbance is given an angle. This is considered to be similar to a real dent cellguide at the location of a stiffener. The depth and angle are adaptable.

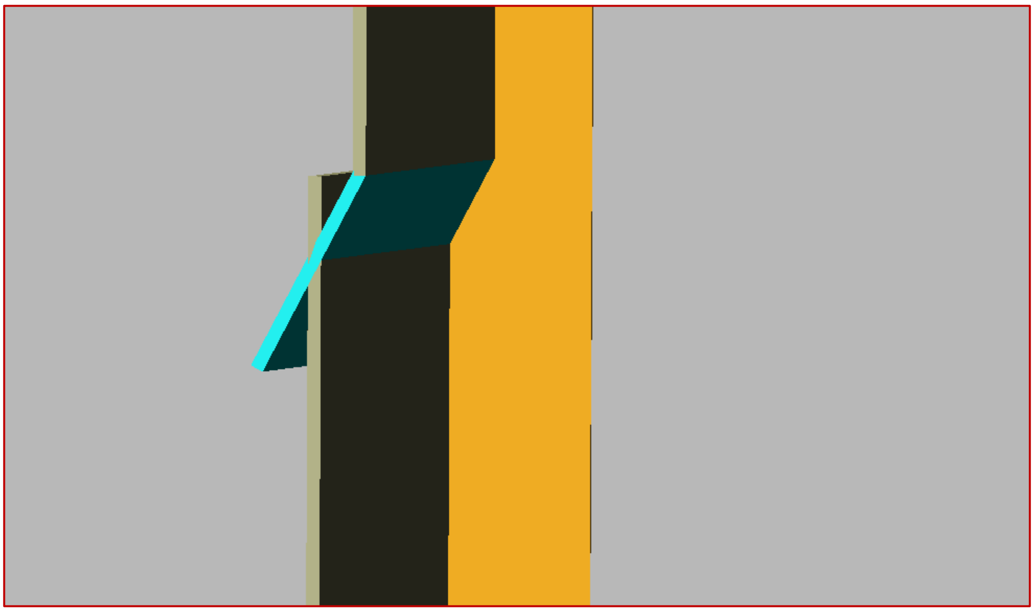

Figure 22: Cell guide disturbance in Adams model

Above the dent, the cell guide dimensions are according to Lloyds this means that below the dent the container has additional longitudinal space. To ensure a collision with the cell guide the container is hoisted with an offset relative to the centre of the cell to ensure contact with the misplaced lower cellguide.

5.1.4 Variation of crane characteristics

Every crane is different and depending on the crane characteristics it might be relevant to equip a crane with a snag protection or not. The model can quite easily be adapted to fit the crane specification by changing the following variables:

- Inertias of all rotating components
- Trolley position
- Crane boom length
- Ropes: thickness, length, E-modulus
- Brakes: times, forces
- Cellguide disturbance: angle and depth
- Impact collision: stiffness and damping
- Drives: speed/torque
- Container load: empty/loaded

5.2 Validation and verification

This part of the chapter describes the validation and verification of the model. It is easy to create something that looks nice but it also should be true. For the model several parts will be separately validated. Two very important things are rope's elongation and energy transfer from rotating components. For this simple MSC Adams models will be made and compared with basic calculations. A separate model is made because the complete model will be too complex to compare and not possible to assess the separate aspects.

5.2.1 Rope validation

In MSC Adams an additional toolkit is used: TKC Cables Toolkit, this is a plugin created by Chris Verheul, this toolkit has the possibility to create, pulleys, winches and cables. This plugin is used in the model because it works easier and better with long rope lengths then the standard parts in MSC Adams. This part describes the validation of the ropes of this plugin, compared with a simple hand calculation.

Situation: a 10ton load attached to 100m of rope, what is the rope elongation by gravity? The cable data is equal to the cables used in the new quayside cranes for APM at the second Maasvlakte and are given in Table 4. The wireropes used are Teufelberger Perfect 8-Strand Ropes Q816V with outer diameter 30mm.

Table 4: Cable data APMT MVII [15]

Length cable	100m
Young's Modulus	10.5e4N/mm^2
Tensile strength	1960N/mm^2
Minimal breaking load	846kN
Nominal rope diameter	30mm
Effective rope diameter (chapter 4.4)	23.433mm
Weight	4.37kg/m
Load	10000kg
Gravity	9.80665m/s^2

Hand Calculation

This is the same data as used in chapter 4, for the basic calculation, therefore the effective diameter is 23.433mm and the spring constant (k) is equal to:

$$k = \frac{EA}{l} = \frac{45.3e6N}{100m} = 45.3e4\ N/m$$

This gives a calculated elongation of:

$$u = \frac{F}{k} = \frac{10000kg * 9.80665\ \frac{m}{s^2}}{45.3e4\frac{N}{m}} = \underline{\mathbf{0.2165m}}$$

MSC Adams

This handcalculation will be compared with a simple model in MSC Adams. One single rope is connected to a fixed winch, the other side of the 100m rope is connected to a 10ton beam, the shortened model is displayed in Figure 24. Figure 23, on the right, shows the data input for the cable in MSC Adams, the data is equal to that used for the hand calculation.

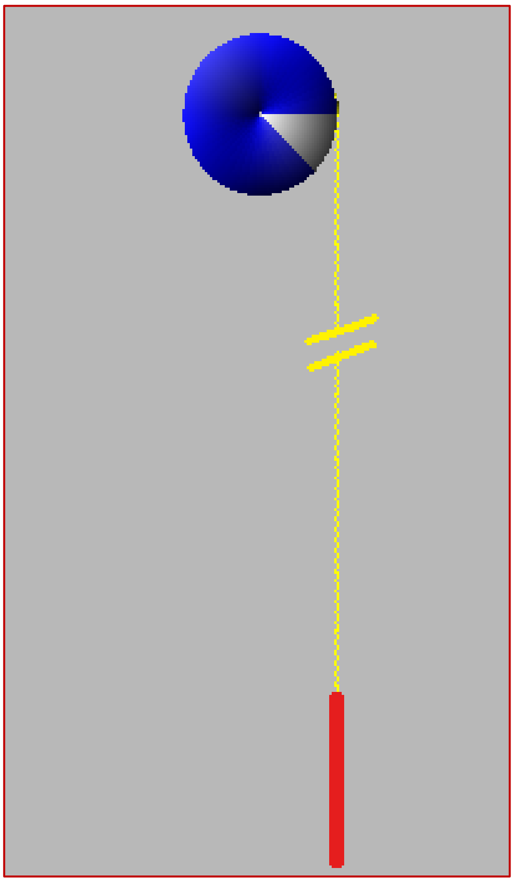

Figure 24: Rope validation model

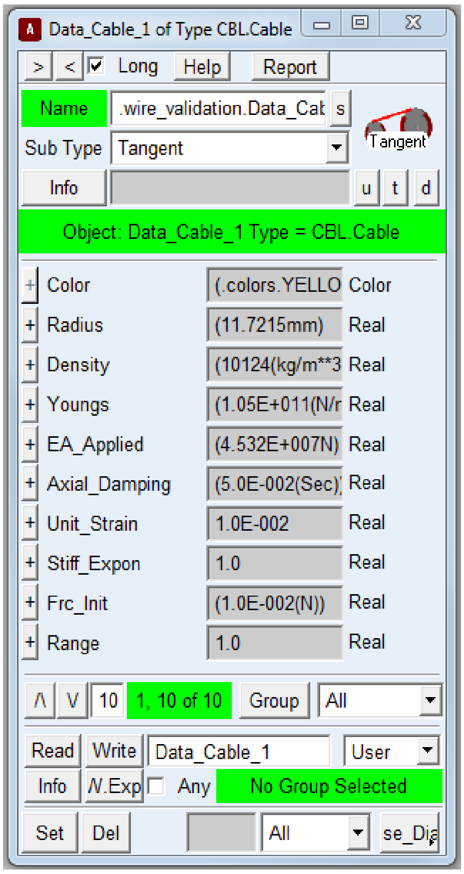

Figure 23: Rope data

Figure 25 shows the plot of cable length in time of the model. As can be seen the load starts at a height of -100, equal to the rope length. At t=0 the load is released and due to the gravity pulls on the rope, it damps out after a few cycles and reaches an equilibrium at -100.2169 meter.

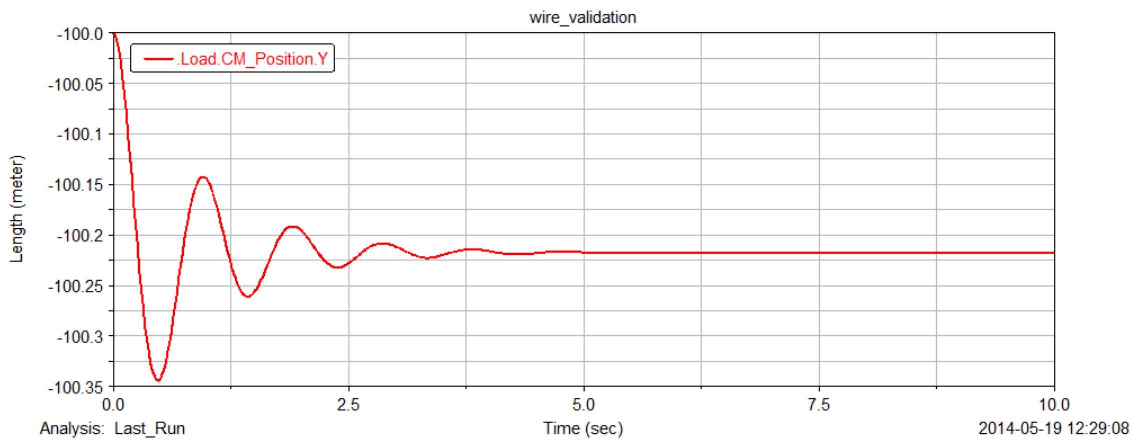

Figure 25: Plot of cable length from model

The elongation of the rope in the model is 0.2169m, this is only 0.20% difference with the hand calculation, therefore the rope elasticity in the toolkit is considered valid for this setup.

5.2.2 Kinetic energy in rotating parts

In chapter 4.4.3 the influence of the kinetic energy in the rotating parts was shown, the rope elongations, angular acceleration and velocity, times and energy were the output of the calculations. Here this will be compared to a similar model in MSC Adams to validate the working of ropes, gearboxes and inertia.

Table 5: Cable data for Adams model

Cable length at snag	250m
k-factor	722kN/m
I	35kgm^2
i	15.8
Radius drum	0.5m
Initial rotational velocity	209 rad/s
Static load	46 ton → 225.6kN pretension

The simplified model consist of only one rope, but with the characteristics of four: double diameter (→four times area) and the spring constant equal to the sum of 4 ropes. One winch, a gearbox and a single cylinder representing the total inertia, all other inertias are set to zero.

Figure 26 displays the model, the cable is 100 meters long and fixed to the ground on the other end, the winch drum, visible in blue, is connected to the slow output shaft of the gearbox, the ingoing shaft is connected to a rotating cylinder with an inertia of 35 kgm^2. At t=0 the applied cable pretension is equal to 225.6 kilo Newton and the initial speed of the motor is 209rad/s. Figure 27 on the next page shows the graph of kinetic energy in the of the rotating mass in MSC Adams.

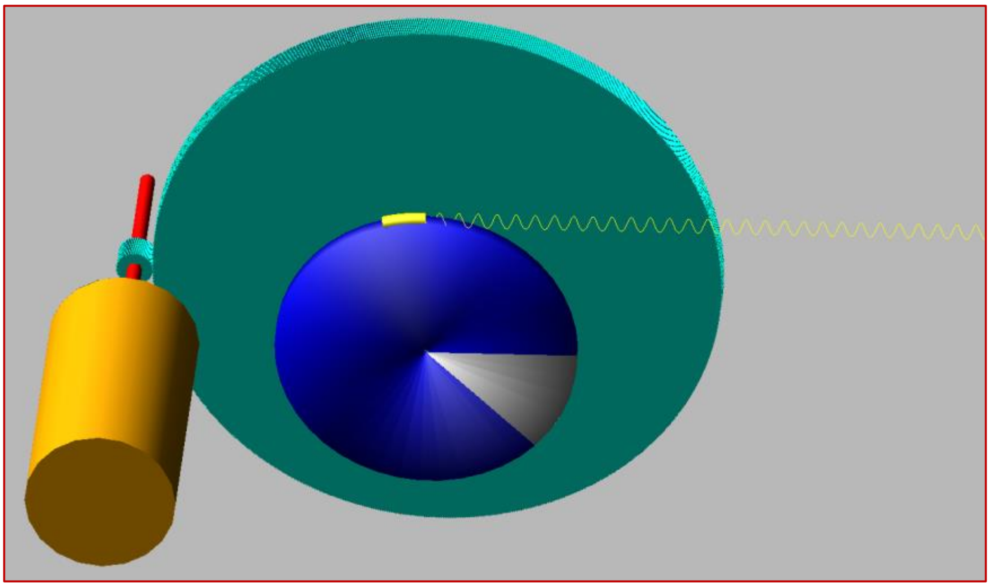

Figure 26: Energy transfer validation model

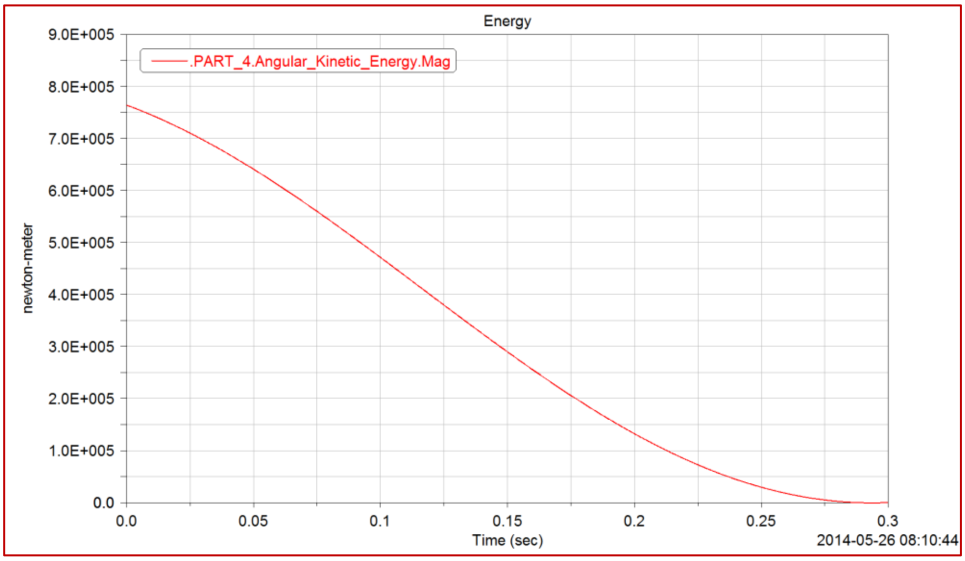

Figure 27: Kinetic energy in rotor MSC Adams

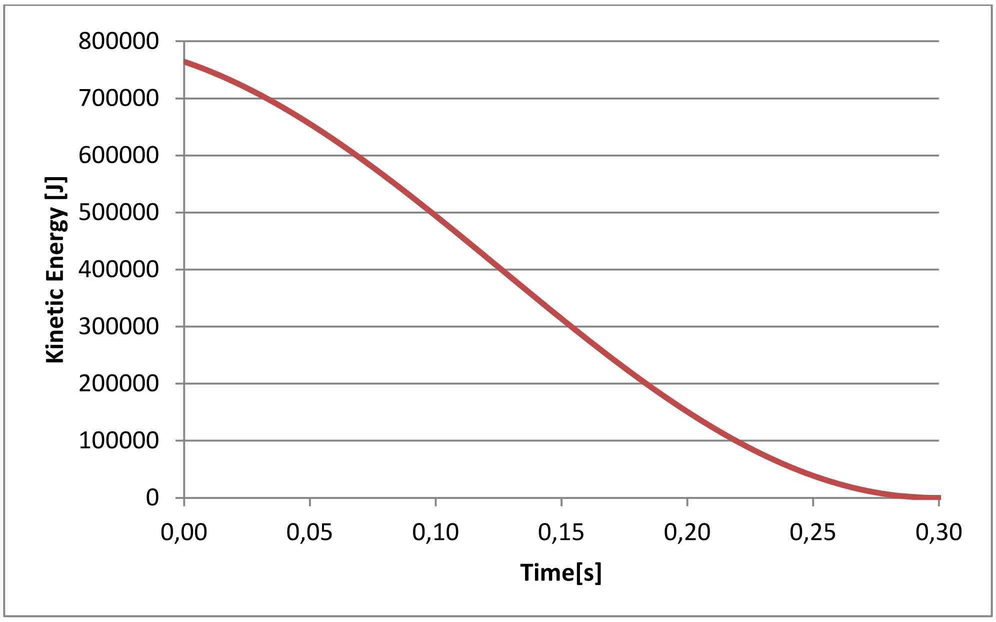

Figure 28: Calculated rotational energy

To validate this energy transfer into the ropes a calculation is made with the same data as the model, the calculation is similar as in chapter 4.4.4. The graph in Figure 28 shows the result, the initial kinetic energy is 764kJ, same as was found in MSC Adams. The trendlines of the graphs are similar and both velocities reach zero after 290 milliseconds.

5.3 Parameters in final model

Table 6 gives the data used in the MSC Adams model to simulate the APMT crane. This is the same crane as used for the calculations in chapter 4.4.

Table 6: Parameters for MSC Adams

Cable		*Teufelberger perfect*
	Radius	11,7215mm (d=23.443)
	Density	10125kg/m^3
	EA-factor	4.532e7N
	Axial Damping	0.05s (iteratively determined)
	Force initial *(depends on load)*	57160N (empty container)
Load		
	Headblock (excl. 4 sheaves)	9300kg
	Spreader (Bromma STS45)	12600kg
	Container *(depends on load)*	24100kg (max weight at max speed)
Sheaves		
	Mass	156kg
	Inertia	19.5kg/m^2
	Radius	465mm
	Width	80mm
	µTangential	0.002
Operational brakes		
	Effective friction radius	450mm
	Friction factor	0.4
	Adjusted torque	19200Nm
Emergency Brakes		
	Effective friction radius	805mm
	Friction factor	0.4
	Clamping force	500kN *Adjusted torque is 193200Nm*
Gearbox		
	Ratio	15.8
Inertia drivetrain		
	Rotor	13.6kg/m^2
	Brake disk	8.25kg/m^2
	Coupling (SOS MALMEDIE!!!)	6.636kg/m^2
	Drums	946.5kg/m^2 →3.79kg/m^2
	Drum coupling	13.5kg/m^2 → 0.054kg/m^2
Cellguide disturbance		
	Angle	65 degrees
	Depth	40mm

As discussed before the motors are torque driven, in case of snag the force in the cables increase linearly since speed in still constant (u=2x hoist speed). Due to equilibrium this must be equal to the torque increase in the motors. After snag the torque increase is:

$T_{incr.} = 2 * v_{hoist} * k_{rope} * \#_{snagged\ ropes} * r_{drum} * i_{gearbox}$ until shutdown torque is reached.

5.4 Results from model

This chapter describes the result from running the model. The model is run with the same parameters as the calculation of chapter 4 as shown in Table 6. This allows us to compare the complete calculations but does show some differences since the calculations are based on theoretical sudden complete stop and the model uses a more realistic collision and jamming of the load.

5.4.1 Hoist motion + snag

The entire simulation from lifting up to snag of a 46 ton load is displayed in Figure 29. The figure contains 4 lines: in red the cable force, in blue the headblock height, in dark blue the total applied motor torque and in green the driveshaft velocity. With these 4 measurements we can describe the entire event according to 8 points/intervals, indicated in the figure with orange vertical lines. Table 7 describes for the event the measures at all these intervals.

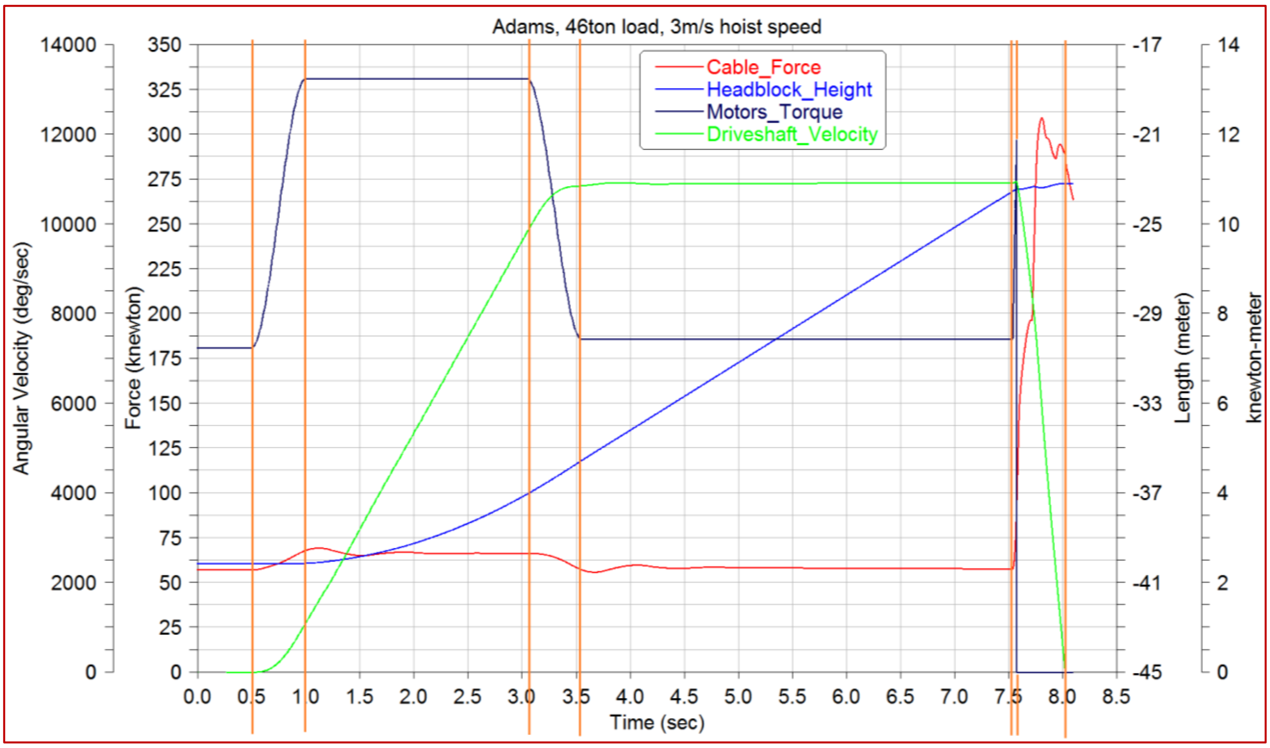

Figure 29: Hoisting of 46 ton until snag

Table 7: Event description

Time	Hoist speed/acc.	Motor Torque	Motor speed	Cable force
0-0.5s	v=0 a=0	Static load 7kNm	0	Static 56kN
0.5-1.0s	v=increasing a=increasing	Ramp up to acceleration torque	Accelerating from 0-1800 rpm	Increases to accelerate mass
1.0-3.0s	v=increasing a=constant	Constant acceleration torque	Accelerating from 0-1800rpm	Constant for F=m.a
3.1-3.55s	v=increasing a=negative	Ramps down to static torque	Accelerating from 0-1800rpm	Decreases to static
3.55-7.53	v=constant 3m/s	Static torque	1800rpm	Static 56kN
7.53-7.56	v=decreasing a= very negative	Increases to Tmax to maintain speed	1800rpm	Increases with F=k.u (u=3m/s)
7.56-8.1s	v~0	T drops to 0	Decreases from 1800-0rpm	Increase due to elongation by inertia
8.1	v~0	T=0	0	Maximum reached

For this research the last part is most interesting, from snag initiation up to engine stall. Therefore a close-up is given in Figure 30. The scaling of the height has changed to zoom in, other scaling has remained the same.

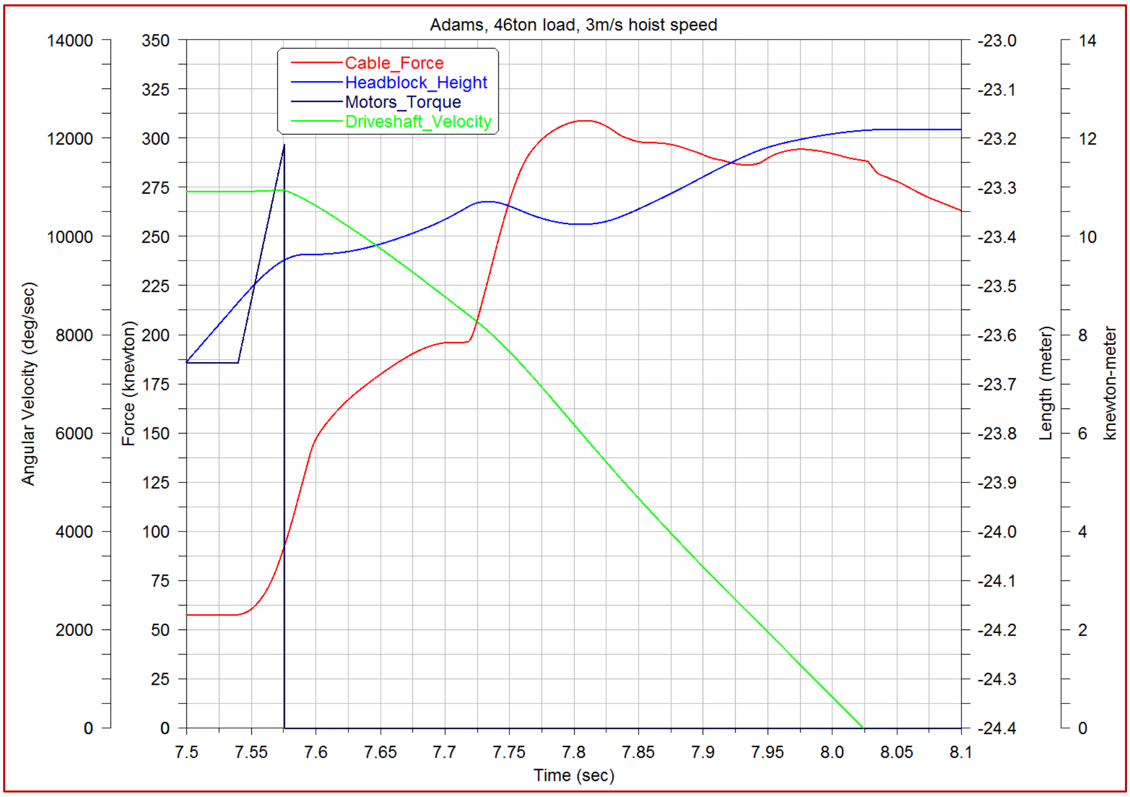

Figure 30: Close up of snag event

Due to penetration of the load into the cellguides the load does not instantly stop, it has a certain deceleration. As can be seen in Figure 30 the rope force starts increasing before the load is stopped. The ropeforce fluctuates with the headblock displacement. It is also visible that there is an increase in torque but the engine speed remains nearly constant as it was designed.

5.4.2 Rope Force

In theory snag is pure on 2 or 4 ropes with sudden stopping, in practice the container moves a little before reaching complete standstill and after tilting all 4 ropes will be stretched, even if snag is initiated on one side. Figure 31 shows the container in the complete snagged position for 46 ton load on the ropes and the high speed hoist of 3 m/s.

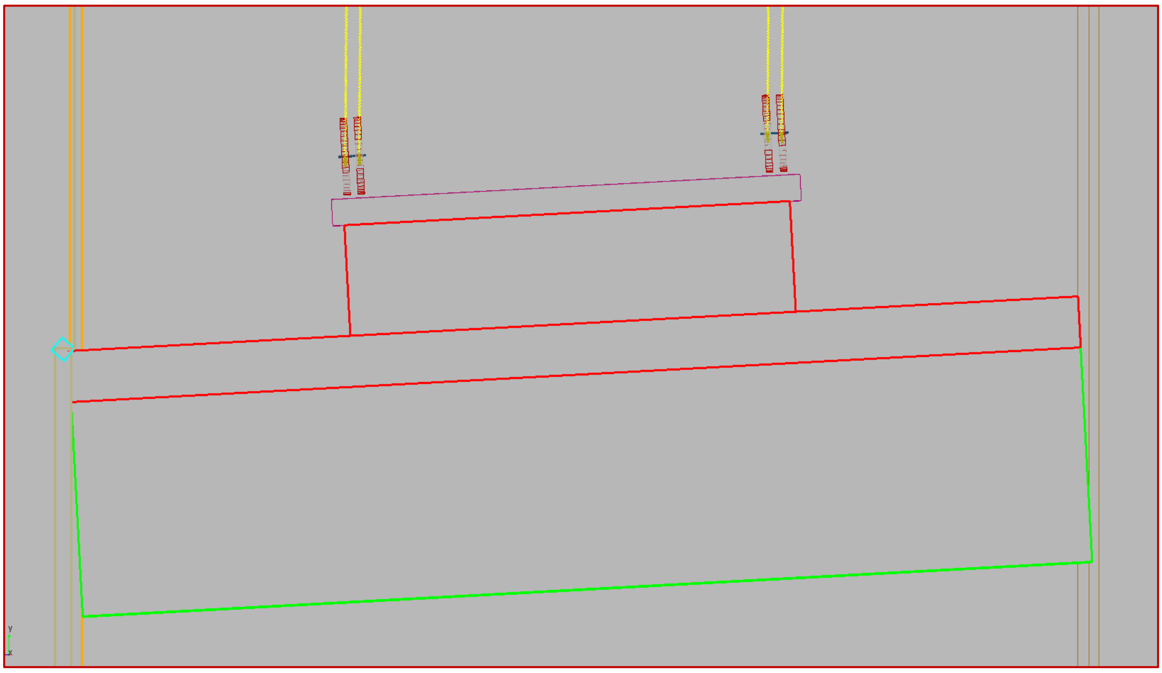

Figure 31: Snagged load in model, 45 ton on the ropes, 3m/s hoist speed

As can be seen in Figure 31, the snag is not a pure 2 or 4 rope snag. All 4 ropes are being stretched only the left more than the right. Therefore we would expect that the ropeforce will end up in the middle of the calculated 2 rope and 4 rope scenario of chapter 4.4.

In Figure 32 the maximum found rope force from the Adams simulation is compared with the calculations from chapter 4.4. It is immediately visible that the rope force increase does not follow the same ideal curve as given in chapter 4. Due to the interaction with all components, the collision, the sheaves and all parts of rope there is a overshoot and levelling effect, the trend-line of the ropeforce is however very similar. What seems weird is that the eventual ropeforce is not between theoretical 2 and 4 rope snag as we would expect. This can be explained: if we look at Figure 30 the light blue line indicates the headblock height, from snag initiation until complete stop the displacement is about 0.3m due to the reeving this is 0.6m rope length (in theory this would be zero due to the instant stop). The corresponding rope force is 0.6m times the spring constant: 104.6kN, resulting in a maximum rope force of 394kN. This value and end times fit perfectly between the calculated values, as expected.

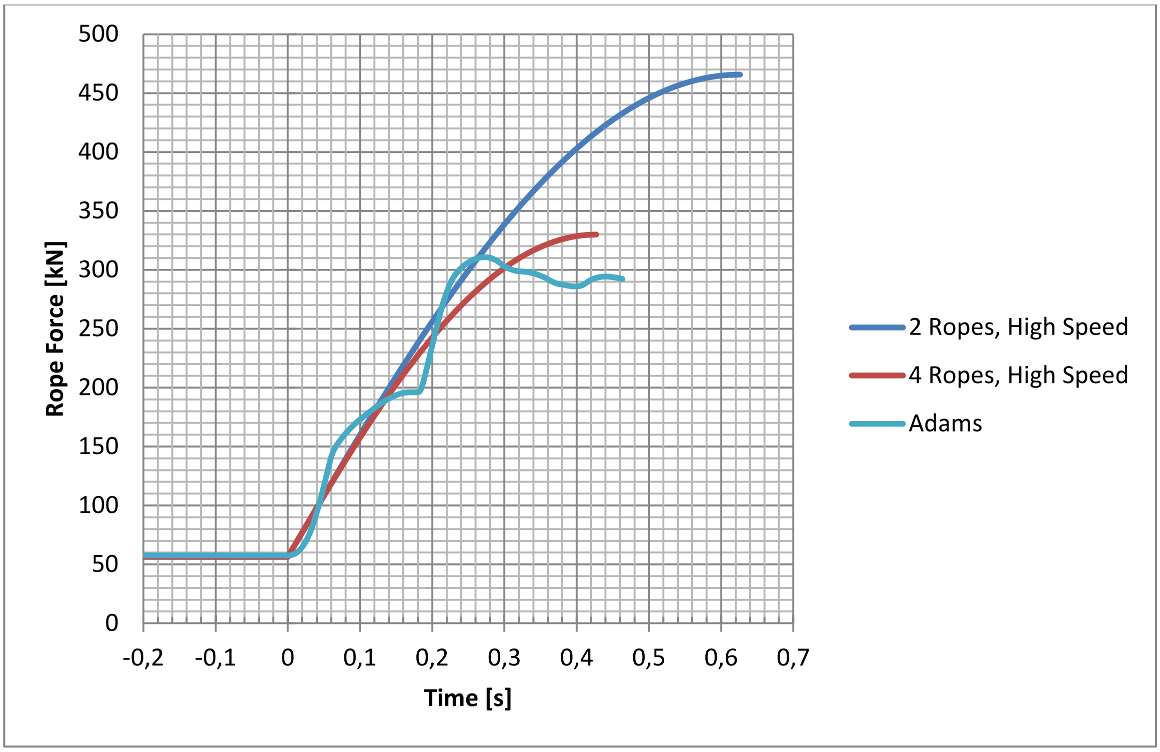

Figure 32: Rope force Adams model compared to calculations

5.4.3 Snag detection

In chapter 6.1 will be discussed how we can detect snag. We can only detect and measure things that happen, for example accelerations: forces changes etc. currently within Kalmar is no representative or valuable data available of headblock movement or impact. The model is obviously not identical to reality but is the best indication available right now. Basically there are two changes interesting to detect: Load/impact and speed/acceleration.

Figure 33 shows the graphs of all 4 ropeforces during snag for high speed hoisting. As can be seen the increase is very steep, 20% increase is within 12.5 milliseconds. Figure 34 shows accelerations in all directions Y is vertical, X is horizontal along the crane and Z is horizontal in the longitudinal direction of the container. The figure displays the accelerations at middle of the headblock and at the starboard side of the spreader, the location where the spreader hits the cell guide. This can be relevant for the location of measurement as will be discussed in next chapter. The accelerations in X direction are very small, in Y direction the deceleration is of course huge due to the initial velocity, values above 7.5 g are reached.
The Z acceleration is the container being pushed from one side to the other side by the cellguide dent. These horizontal accelerations are up to 3 g.

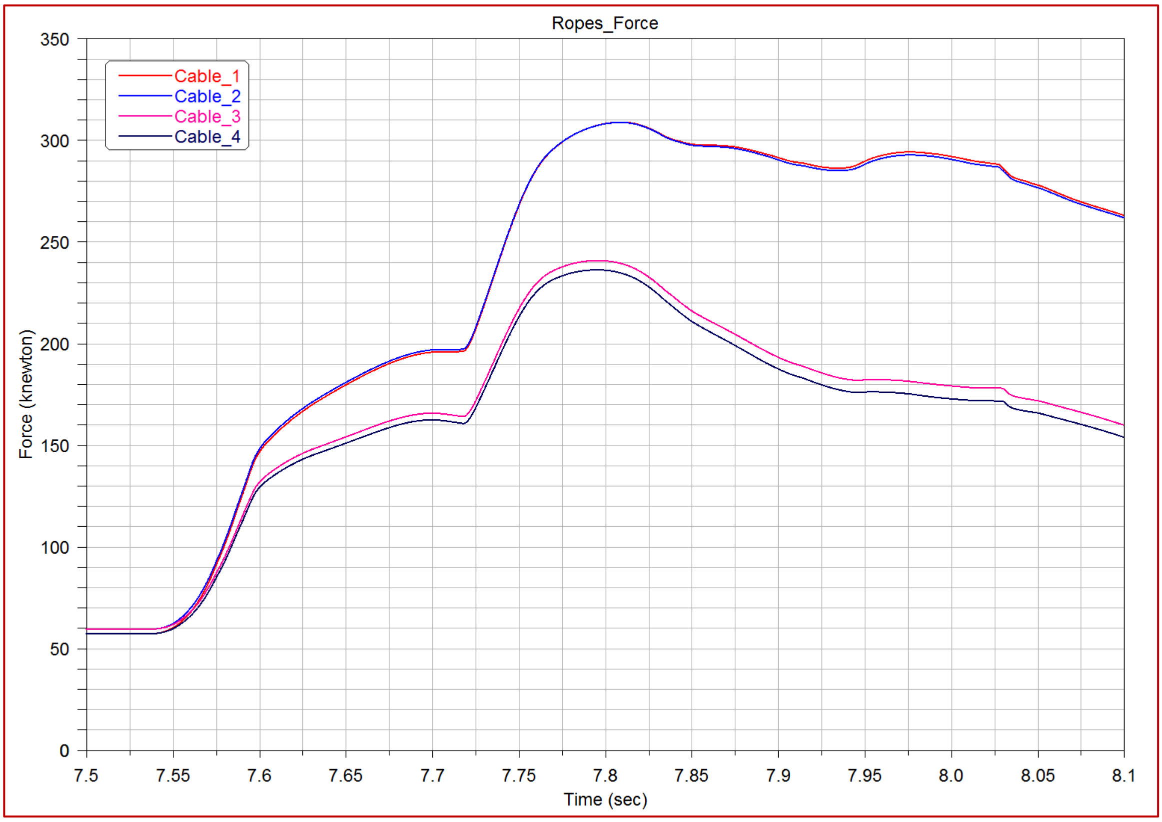

Figure 33: Rope forces during snag

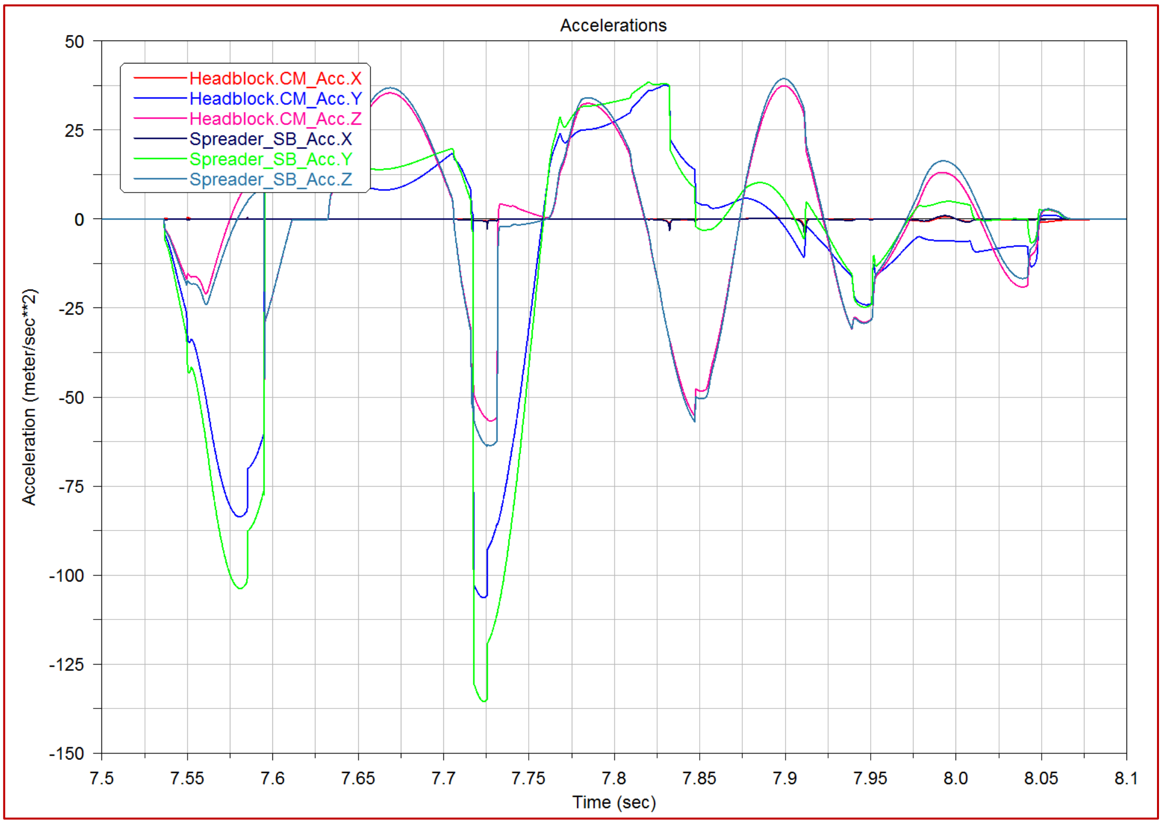

Figure 34: Accelerations during snag

6 Snag systems overview

Precious chapters have shown what components are involved in the snag event and what forces and energies are the present. There are systems and ideas to reduce the impact of snag. The goal of this system is to protect two things: the ropes and the mechanical components. The steel structure is protected by limiting the rope force.

Limits

The rope force has an upper limit, a maximum allowable force based on the safety factor and elastic limit of the ropes. In general one can never exceed 50% of the break strength. In practice often a factor 3 is used as safety limit and a factor 6 for normal operation.

The components of the drivetrain are the second limit: maximum allowable torque on each part. This determines a maximum deceleration of the drivetrain in case of snag. Both limits need to be taken into account for every snag system!

Another important aspect is speed, in previous chapters was already shown that an entire snag event may take less than 0.5 seconds, a snag protection system has to work fast.

Principle

The working principles of systems to prevent snag or reduce the consequences of snag can be split up in three parts: 1 detect, 2 react, 3 (inter)act. First the snag has to be noticed, second step is to process this and decide whether to do something or not, last step is take measures.

Predict → Detect → React → (inter)Act

This chapter will start with a short description of the possibilities for the 3 parts. Subsequently several systems available on the market will be explained. Some of these systems are passive, they have no active control system and thus only contain the action step. Active systems measure and really interact with the crane and mechanisms. The snagload protection systems can consist of multiple simultaneous working combinations to detect and/or interact.

6.1 Detect/predict?

As long as you do not know snag is occurring or initiated you cannot take any measures, that's why the first thing is to detect snag or better would be to even predict an upcoming one. The earlier you know, the more influence you can have on preventing or reducing the consequences.

What happens in case of a snag is that the container/spreader gets stuck caused by an irregularity in the cellguides. This means a motion is changing: there might be a movement but there is definitely a change in velocity. Of course forces are involves force in this acceleration or deceleration. The force in the ropes increase so there must be a change in the torque on the ropedrums, gearbox and motors. In theory all these forces and accelerations can be measured.

Here several options for detection and measurement will be generally discussed, these give basic knowledge to understand several protection systems and are a good introduction to chapter 7 concerning the choice of an early snag detection system.

6.1.1 Measuring rope force

Most cranes are equipped with some way of load measuring to prevent the crane from overload and to define load and hoisting speeds. This is often done by means of load measuring pins. The best way to measure is at a location where the forces work in one direction, therefore often the sheaves at the back of the boom (backreach) or at the front end are used. Other sheaves or locations might also be possible and suitable. Cables on STS cranes are quite long, resulting in sagging and whipping of the cables and due to the spring-working it takes time to transfer force from one place to another. The measurement location is important and decides if you actually see what you want to see.

6.1.2 Torque

Another way to measure force is to measure torque, the easiest way to do this is by the motor/drive. The modern AC motor with frequency drive is quite intelligent and actually knows based on speed and input power which amount of torque is exerted. This can be transferred to the crane PLC. A force exerted at the load has to travel through the entire rope before it affects the motors, then the motors still have to respond to the change in load before the torque increases, this all takes time.

6.1.3 Twistlocks

Twistlocks is the standardized way of connecting in the container shipping business. There are multiple sets of twistlocks on the STS crane, one set between the container and spreader and often one set between the spreader and headblock. There are special twistlocks available on the market that can measure the force/load on the twistlock. These measuring twistlock pins are designed to measure the load of a container, to detect overload and thereby to prevent the crane from picking up containers when still attached to a trailer etc. Figure 35 displays such a system produced by Lasstec. If a load during hoisting suddenly starts to increase then something must be wrong, maybe a snag. The twistlocks between container and spreader are unfortunately less suitable for this because the spreader might actually be part of the snag, thus the twistlocks between headblock and spreader should be used.

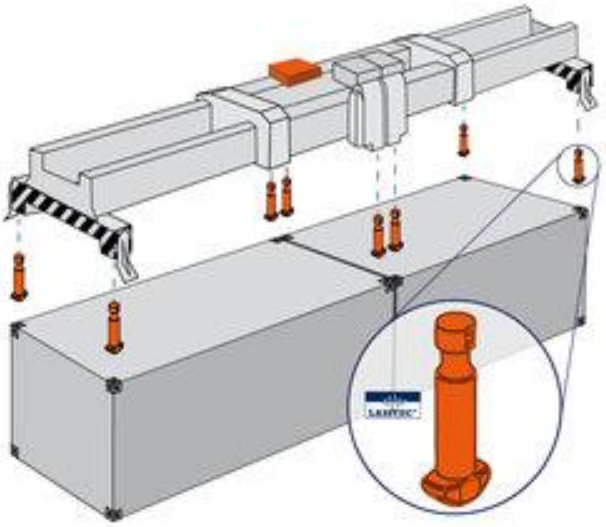

Figure 35: Load sensing twistlocks [Lasstec]

6.1.4 Angle

With an inclination sensor on the headblock or spreader it is possible to measure the angle and/or angle change of the spreader. When snag is initiated at one side the spreader obviously gets tilted and this can be measured. This is only suitable to detect one side initiated snag, not centric snag!

6.1.5 Accelerations

When snag occurs the container/spreader gets stuck, so there is change in speed: an acceleration. With acceleration meters this can be measured quite easily. Deceleration in vertical plane is quite obvious, the spreader used to go up but is suddenly stopped. The idea of Sibre is that there might be an acceleration in horizontal plane. Due to a bent cell guide it might be that the load is first moved in horizontal plane, actually initiating the snag, if one can measure this, snag could be predicted instead of detected. This will be further explained in chapter 7.
The location of the sensors is quite important, measuring in the middle of the headblock might give different values then measuring at the corners of the spreader where the snag actually is initiated.

6.2 React

After the snag is detected the next step is to react, process the signal and decide whether the sensors measure a real snag or not. The crane is standard equipped with a so called crane PLC (programmable logic controller) as had been shortly explained in chapter 2.6, this is the brain behind every sensor, signal and movement. Since cranes are equipped with numerous sensors, extensive safety and advanced control systems the crane PLC is not very quick, it knows everything but is not dedicated to emergency events only. Since snag and all consequences happen in a very short amount of time another way of signal processing is desired.

A separate "safety" PLC dedicated to emergency events or even better only to snag will react a lot quicker. It is also possible to not use a PLC but simple relays that might even be quicker but is more difficult to adjust to settings because it is a simple on-off module depending on the incoming signals.

In short, based on the input from the detection step, the reaction decides whether to do something or just stick to normal operation procedure.

6.3 (inter)Act

Once the system has detected snag and in the reaction step is decided to take measures several things can be done or happen. Here some possible measures are explained basically the system wants to reduce the energy input in the ropes, said there are three options to do so: 1. Stop the drivetrain, 2. Increase ropelenght, 3. decouple the drivetrain.

6.3.1 Stop the drivetrain

Braking takes up energy, as described in previous chapters there is a lot of energy in the drivetrain with the potential to go into the ropes and stretch them. If the drivetrain is slowed down by brakes or the drive then they take up energy and therefore reduce the energy flow into the ropes. Braking can be done in different ways: apply operational brakes, apply emergency brakes and/or reverse torque as described in chapter 2.5. This stopping obviously has to be initiated as fast as possible to take up as much energy as possible.

6.3.2 Increase ropelength

The energy in the drivetrain tries to stretch the ropes, if we add ropelength from somewhere else and if that requires a certain force, then energy is dissipated. With hydraulic cylinders this can be achieved quite easily. Open a valve, it takes force to push out the oil: energy dissipation. All energy goes through the ropes but is not takes by the rope, it is party transferred to another medium. By doing this the total rope stretching energy is reduced and therefore the maximum rope force is less.

6.3.3 Decouple drivetrain

As was seen in chapter 4 about the energies, the inertia is responsible for a major amount of energy, the motors represent the biggest part of the inertia. If it is possible to separate this component from the drivetrain, the total of amount of additional energy can be reduced tremendously. This can be a passive system with a break coupling or maybe even with a controllable electromagnetic clutch. A passive break coupling, has one major drawback: there is only one breaktorque, but the load on the crane can vary a lot (30 t-105 t), and therefore it only works in one specific case. It does limit the total amount of force in the rope. Obviously after decoupling the drive it is not possible anymore to use reverse torque to stop the system.

6.4 Available/conventional systems

This part describes some conventional snag load protection systems that are available on the market. All of the here described systems are still being installed or used on cranes.

6.4.1 Hydraulics

This system is based on hydraulic cylinders, an example is displayed in Figure 36 on the left, a schematic drawing is given at the right. This is an installation produced by Rima installed on a Kalmar crane in Antwerp. The principle of hydraulic snag protection is increasing rope length: the hoisting rope is reeved around an extended cylinder and when the rope tension increased the pressure in the cylinder becomes higher. An expansion valve is set to a certain pressure, when this pressure is reached the valve opens and the cylinder retracts and thereby adds rope length. Pushing the oil out of the cylinder requires force and thereby takes up energy from the ropes. This system is a passive and reacts **only** on one setting for the overload based on a safety factor over the maximum load. There is no PLC so no response time, a quick valve opens after an overload [17]. Opening the valve can give a signal to the drive and PLC to initiate braking and shut down the motors.

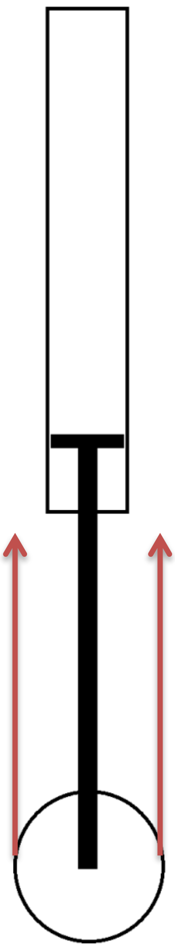

Figure 36: Hydraulic snag protection device, Antwerp

It is very important that a signal shuts down the drive and there is work done by the outflow of the oil otherwise no energy is dissipated and all energy still goes into increasing the ropeforce. In that case only time would be won not less rope force.
The main disadvantages of the system are the weight and complexity of the installation and the fact that the system uses hydraulics, in general everything that can leak oil eventually will.
There are several suppliers of these systems ZPMC and Rima are two of them.

6.4.2 Pintsch Bubenzer SOS [18] [19]

Another snag protection system which is growing in popularity last years is the Pintsch Bubenzer-Malmedie Snag Overload System (SOS). Figure 37 shows the system installed on a crane. This system combines multiple aspects, their SOS system contains a Malmedie torque limiter/break coupling and fast brakes. The brakes are applies once an overload condition is reached or when the coupling disengages. The coupling breaks when the torque in it becomes too high: the rope force is then at already at a high value and the inertia of the motors still try to stretch them. The break coupling is a passive system, the sensing of overload and braking is an active system.

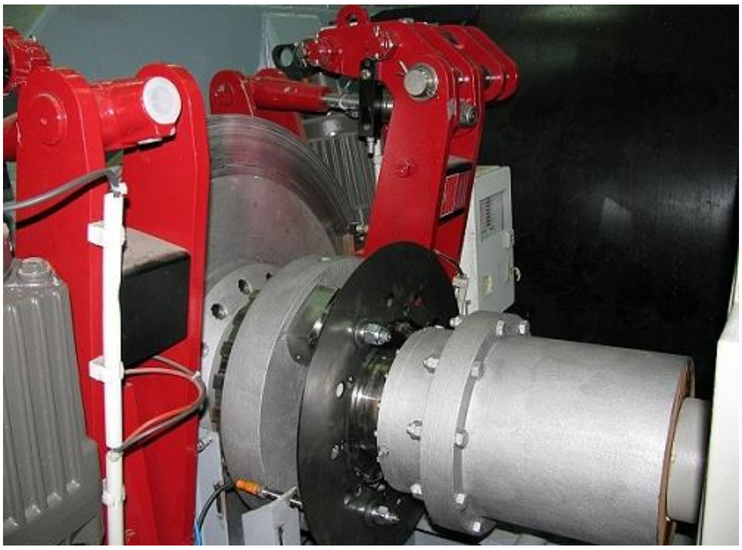

Figure 37: Malmedie breakcoupling and Bubenzer brakes

The coupling can only be set for one particular break torque, the crane however is confronted with many load cases. If the coupling is set for the high load and one lift a light load a lot of additional force is needed before it will break. In practice the coupling is set for a heavy hoist and a certain safety factor: often 150% of maximum required motor torque. In practice this means for the APM cranes a value of 17.8kNm per coupling. The load on one coupling during continuous hoisting of an empty container is only 3.6kNm, which is only $1/5^{th}$ of the limit. The breaktorque effectively means at least 280kN ropeforce is present before the coupling breaks.

Once the coupling breaks the system is quick and the brakes fall within approximately 70ms to 90% braking torque, note that now the ropeforce was already 280kN before the braking starts. The inertia of all components except the motor still stretched the ropes.

The SOS system also uses overload to detect snag, with use of load measuring pins they determine an overload, and if they can safely sense this the brakes are also applied. The load sensors at APM are located at the backreach, about 120m away from the Snag event, this is not the best place to measure. The discussing about placing load sensors will be continued in chapter 7.2.1.

A major disadvantage of the break coupling is the high inertia it has compared to a fixed coupling. This means in normal operation the coupling adds a significant energy demand to the drive and introduces more initial energy into the ropes in case of snag before the coupling breaks. To partly compensate for this increase in inertia Pintsch Bubenzer applied so call LiTec Brake disks for the operational brakes, although these brakes have a lower inertia they have some quality issues when getting hot, the disk material tends to damage and break sometimes.

An important advantage of the system is that it protects the gearbox and axles, as explained in chapter 2.5.4 concerning the brakes: if acceleration is too quick, one risks breaking something due to the high moment caused by the high deceleration. The motors are the parts with the highest inertia on the high speed shaft, and therefore the highest risk to break an axle or gear. This introduced torque by deceleration has to go through the SOS Malmedie break coupling, and this now actually preventing the too high moment to further go into the drivetrain.

In practice there are still a lot of problems with the correct setting for the torque, the drive and motor really have to take care of the coupling and use soft starts and stops to prevent unwanted breaks during normal operation. This makes the system liable and the terminals and crane/drive manufacturers are not yet convinced by the working principle of this system.

6.5 Other possibilities

There are also alternative ideas, Bart de Vette listed several patents in his report [4]. So far none of them have been largely applied in the STS cranes, therefore they are not discussed here. Next chapter describes a new idea for snag protection, proposed by Kalmar and Sibre. The idea is based on early detection of snag, and then a quick response start braking as soon as possible. This idea is that this would make the hydraulic cylinders or high inertia break coupling superfluous, removing these additional components would reduce cost and complexity.

“A new Idea”

Early detection

&

Fast Stopping

7 *Part I* : Early detection of snag

Kalmar and Sibre together came up with a new idea to protect STS cranes against snag. The basis for the new idea of snag load protection consists of two parts, the early detection of snag and fast stopping. The combination of noticing snag early and then braking quickly should reduce the amount of energy input into the ropes and bring back the maximum ropeforce to an acceptable level below the safety and elastic limit of the ropes. The goal is to reduce the ropeforce in case of snag to below the elastic limit, this is mostly 50% of the minimal break strength [15], striven is to bring the rope force below the 1/3 limit. The possibilities for fast/early snag detection are dealt with in this chapter, the fast stopping is worked out in the next chapter.

As said before conventional methods use mainly forces in the cables for actively detecting snag, however currently not quick enough, can these measurements be improved or do we need an alternative way of detection. Sibre proposed to predict snag by measuring horizontal accelerations, the idea behind this is that snag occurs due to an irregularity in the cell guides, a dent for example, this causes a horizontal displacement initiating the snag event. This means there are two main ways of snag detection, the first is based on acceleration the second is based on force measurement. The best way to actually detect snag is investigated in this chapter.

The first and seemingly most important aspect of snag detection is speed: the earlier snag is noticed the earlier measures can be taken, increasing effect. The second aspect actually equal importance is reliability, the system may not miss a snag but also may not give false alarms. In practice, if the system gives multiple false alarms and shuts down the crane too many times without a reason, then the system will be shut down by the terminal.

This chapter describes and evaluates the methods of detecting by acceleration and by (improved) force measurements, followed by an evaluation and choice of the most suitable way to detect snag.

7.1 Detection by **acceleration** at the headblock/spreader

A possibility to detect snag is by measuring accelerations. As said before this can be in two directions, vertical: this is a deceleration based on the stopping of the load; horizontal: acceleration caused by the cellguide disturbance. Acceleration can only be measured at the headblock, spreader or load since these are the only parts that move. This is actually where the snag occurs and therefore seems to be quite a suitable spot.

7.1.1 Vertical deceleration

Vertical deceleration always is present when snag occurs, otherwise there simply is no snag. Depending on the structure jamming the container the deceleration can be done in a very short time or it might take a little longer if the container is gliding until it really gets stuck. From chapter 5.4.3 is known that we can expect deceleration of up to 10g for snag.

7.1.2 Horizontal acceleration

Measuring horizontal acceleration to detect snag is an idea of Sibre. It is believed that since cellguide damage is the reason for snag, first a horizontal impact/acceleration is done and this actually initiates the snag. This would mean by horizontal measurement it could be possible to predict an upcoming snag, and therefore this would be the quickest way of snag detection.

Unfortunately by measuring horizontal acceleration one cannot detect centric snag.
Due to the very limited clearing in cell guides the space to speed up and slow down is very short, therefor the measurements have to be done in with very short intervals to come up with a safe signal.

7.1.3 Sensor location

Different locations might be suitable to measure accelerations, at the headblock or on the spreader, displayed in a schematic drawing in Figure 38. Sensors on the headblock (yellow crosses) have the advantage that in case of multiple spreaders it does not require additional cabling and sensors on all spreaders.

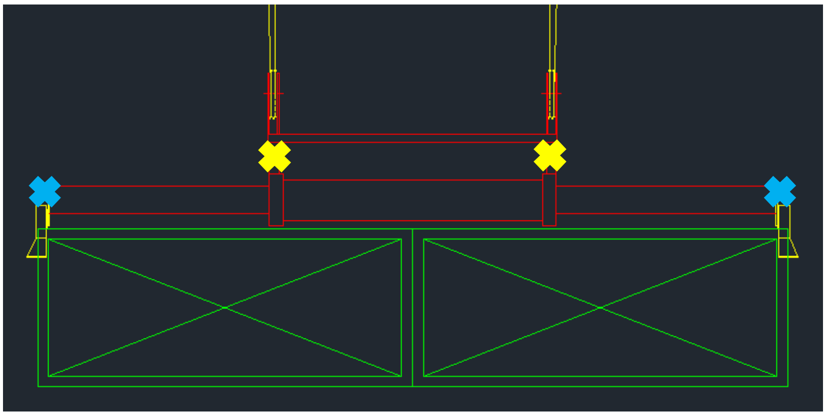

Figure 38: Sensor locations headblock/spreader

For vertical acceleration it is important to measure on the outside of the spreader, in case of a one side snag initiation only one side of the spreader decelerates first (indicated with the blue crosses in Figure 38), the middle of the headblock and other side of the spreader do not slow down first but follow later.

Horizontal acceleration can also be measured in the headblock or center of the spreader, but the stiffness of the spreader then has to be taken into account. Horizontal acceleration cannot be measured at the headblock when using a tandem spreader! Because a tandem spreader (4x1 TEU or 2x2 TEU) is connected to the headblock by chains, which is a flexible coupling and therefore horizontal accelerations are not well transferred and impossible to measure correctly.

7.1.4 Noise, impact during normal operation

During normal operations spreaders are subjected to high impacts, smacking of the load against cellguides, containers etc. according to Bromma and Kalmar accelerations up to 40 g no exception in normal operation. This causes two problems: 1. Noise in measurements; 2. Damaging measurement equipment.

Noise

The noise cause by the impact on the spreader makes it difficult to set a safe bandwidth or filter for snag detection. This increases the risk of missing snags or false detections. False alarms must be avoided since this will very likely cause that the protection system is shutdown by the container terminal/operator. For example when by a damaged cellguide the container is bumped from one side to another, so there is an acceleration but no snag, it still should not cause a snag shutdown.

Impact

The impact on the measurement components is also important with these severe shocks it is very likely that sensors or processing equipment breaks down. According to Kalmar: in general everything on the headblock or spreader breaks down in time due to the impact of normal operation. The sensors might survive but especially the processing equipment or PLCs are not able to take it.

These facts make it very difficult to safely detect snag. At this moment Sibre is testing a measurement setup at STS cranes in Algeciras. For now it is assumed that it is wise to look for an alternative, if the test shows positive results this can always be used to improve the detection and working of the snag protection system.

7.2 Measuring **Forces**

An alternative to measuring accelerations is measuring forces. This means not measuring initiation but consequences of snag, because forces are capable of doing damage. Question is whether we want to measure the impact peak or do we want see the force increase afterwards? Measuring a force (increase) in for example the cables can be a good indication for snag. Also now it is very important to measure quickly and reliable. The possibilities are investigated in consultation with Pat-Krüger and Brosa, both world-wide known suppliers of high quality force measuring equipment and safety systems.

There are several locations possible for force measuring, these will be analysed first, subsequently we will look into different types of sensors and signal processing.

7.2.1 Location of measuring rope force

Current rope force measurements are mostly done at the backreach or at the rope end at the front of the crane. At these locations the measurements are greatly influenced by rope sagging and whipping; the signal has a lot of noise and undergoes a certain delay. Figure 39 shows a graph of rope measurement at a Kalmar crane in Antwerp, the red line is the load measured at the headblock, the line in blue is the output of the load measuring pin at the backreach.

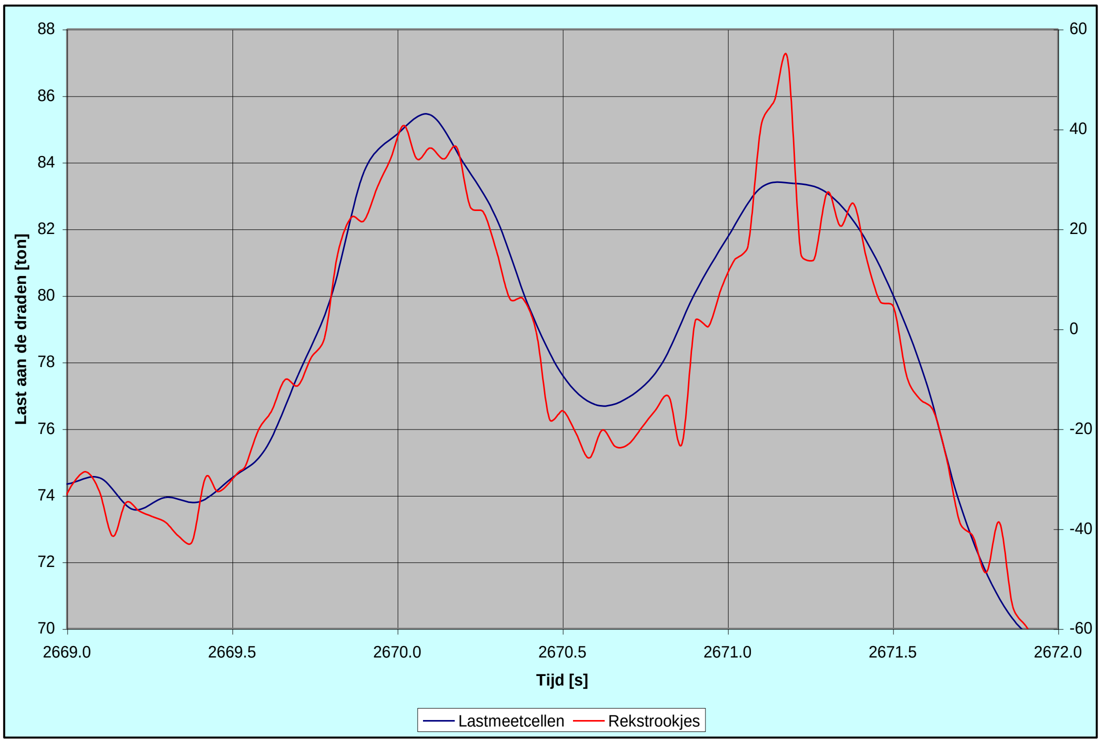

Figure 39: Measurements at P&O crane in Antwerp, backreach LMP vs. actual load

As we can see the load measuring has a similar trendline as the actual load but it filters and damps the peaks, this is great for determining the mass of the load but these peaks are very important for snag detection. If force increase rapidly (for example as at t=2671s) we want to know that as soon as possible not after it is transferred through a long rope. This shows the relevance of measuring as close to the actual snag as possible.

Figure 40 shows data of picking up and hoisting a load of 45 ton at the same crane in Antwerp. The graph describes the hoist speed in yellow, and output of load measuring pins at backreach in dark blue. The graph shows that after acceleration the speed is constant but still the rope force keeps on varying quite a lot. This is due to the whipping of the cables; a result of horizontal long and sagging cables under changing tension. This makes it hard to set a bandwidth for snag detection, 10% overshoot is common in normal operation without bumps. Then at least 20 or 30% or even more has to be chosen as shutdown value to be safe and avoid false alarms.

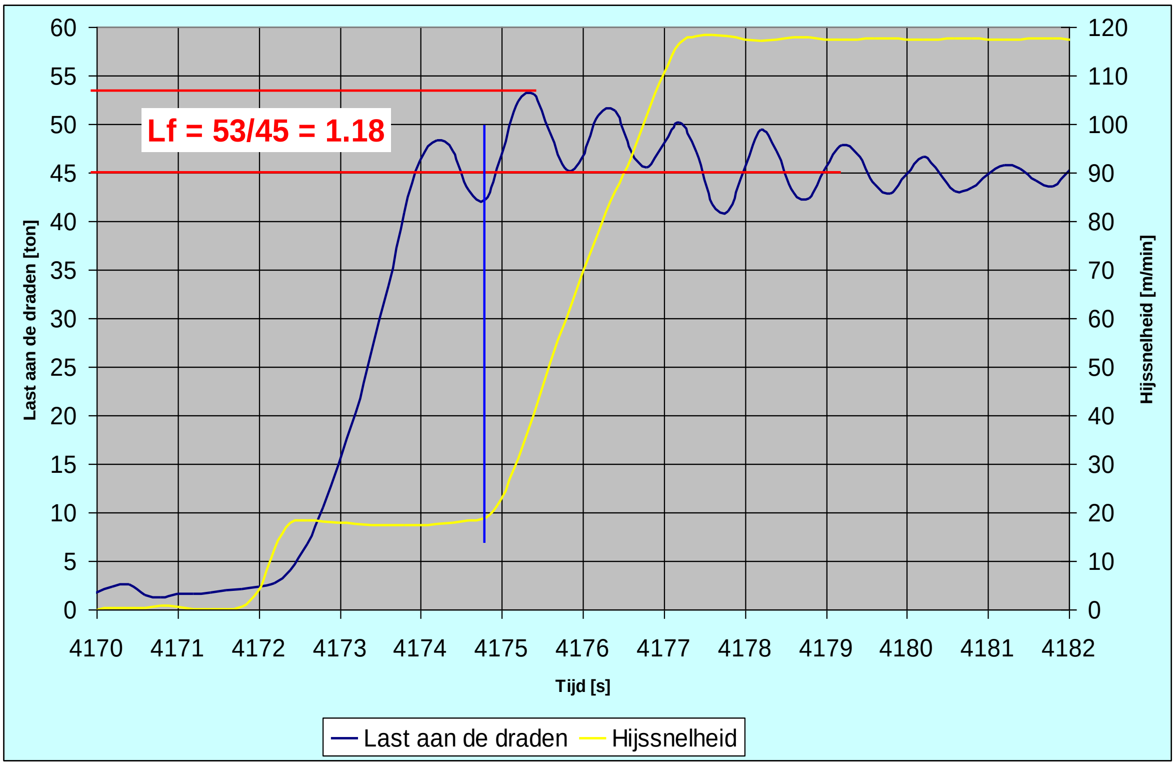

Figure 40: P&O crane, load hoisting data

7.2.2 Headblock/spreader

To avoid measuring ropeforce with all previous described problems, it is proposed to measure at the headblock, this is where the snag occurs. Measuring between spreader and container is not possible because an empty spreader can get stuck too and this would not be detected then. Measurement therefore must be between headblock and spreader or between the ropes and headblock in the sheaves. The force between the headblock and spreader can be measured by force measuring twistlocks, for example by equipping a twistlock with a force measuring washer as displayed in Figure 41 on the right. Measurement of rope force in the sheaves can be done by equipping the sheaves with load measuring pins as displayed in Figure 41 on the left.

Figure 41: Force measuring pins (left/mid) washer (right) [BROSA]

Influence

The placement of sensors at the headblock seems to be the perfect spot because this is where the impact happens, this also means that the sensors and all related equipment is exposed to the severe impact conditions. As said shocks up to 40g at a headblock are no exceptions. The sensors probably will survive that but the processing equipment/chips/PLCs probably not. This means another location has to be found!

7.2.3 Rope force in trolley

From the previous is concluded we have to measure somewhere close to the snag location but not at the headblock. If from the headblock we follow the rope then the trolley is the next place where one could place the sensors. The trolley is not exposed to the severe impact like the headblock and is connected to the headblock by only with single rope parts, so the influence by the wire rope is quite little compared to measurements at the backreach sheaves. There is no wire sag in vertical cables and therefore also less whipping.

Conclusion

In consultation with many people within Kalmar/Cargotec and after several meetings with load measuring suppliers Brosa and Pat-Krüger it was concluded that taking everything into account the best place to measure load is on the trolley.

7.3 Types of sensors

There are different types of force sensors available on the market, intelligent digital versions with internal processing but also analogue sensors. The “simple” analogue sensors are more quick in responding to changing loads and speed is exactly that is what we want, the reaction time for the output signal to a change in load can be in the order of 1 millisecond. But this signal still has to be processed.

7.4 Processing of measurements

As explained in previous paragraph the sensors are analogue because this is much quicker. The signals coming from these sensors have to be filtered and analyzed and then has to be concluded whether there is snag or not.

There is always some noise on the signal this means the signal has to be filtered. If there is a one peak signal it should not mean there is snag. Multiple measurements should lead to a snag conclusion.

Detection should be based on a force increase rather than a set overload. This means the system detects a rapid force increase if this is substantial snag is concluded and a signal will be sent to the E-house, brakes and drives.

There have been discussions with multiple suppliers of Brosa and Pat-Krüger, they both conclude that it should be possible to get an output signal in a short time approximately 20ms from measuring. Brosa has developed a Limit switch (grenzwertschalter) capable of very quickly shutting down after an overshoot. This limit switch is based on shutting down when the load exceeds a set limit. Brosa also filed a patent on the early overload detection for load lifting [20] and work on the development of early detection equipment. Brosa's patent is based on a variable limit, the system measures the normal load during every event and then sets a shutdown value based on a set safety margin, thereby the overload is believed to be detected earlier.

Together with Pat-Kruger, developer of crane safety systems and load measuring equipment an investigation and development of this system has been started. Pat-Kruger focusses on force increase in time not on a set limit. There have been multiple discussions with Pat-Kruger and they are now developing a system that can process and decide quickly, once a first setup is done they will initiate tests to fine-tune a suitable bandwidth and filter. Taking more time to detect, results in more certainty for snag decisions. As first attempt their target is 30ms from snag initiation up to a safe output signal, this seems very realistic. If possible this might be reduced and if necessary this might be increased to 45ms.

Trigger signal

The snag protection does not always need to be active only when there is a risk, therefore the system will receive an trigger signal from the crane PLC to tell the system when to look for snag. Snag is only a concern for lifting inside the ship's cellguides and especially with high speeds. This means the system should be activated when there is a positive hoist speed, for example>20%, this avoids detection due to impacts during take-off. The system can stop once the load left the cellguide, this is generally when the trolley starts driving.

The hoist on landside does not need to activate the snag system, there are no cellguides to get stuck in. Therefore only activate the system when the trolley is on the boom, which is the movable bridge part above the vessel.

In short start when the trolley is on the boom and the hoistspeed is more than 20% stop when the trolley starts driving or when the hoistspeed is below 20%

Output signal

The output of the processing unit will be sent to all involved active components: the equipped brakes, the drive and the crane PLC. To make it possible to fine-tune the timing of these components the output signal can get a delay from 0 up to 50ms.

7.4.1 Times

As said the load measuring sensors take about 1ms and the processing of time of the signal can be very quick. However to ensure that the force increase is real, multiple measurements are done to be sure no false alarm is give therefore the total processing time, from snag until output signal is set at 30ms for now. This is found to be realistic according to Brosa and Pat-Kruger.

From the measurement processing location, the signal has to be transferred to the E&M-house to activate brakes, drives and the PLC, this traveltime will not be more than 5ms.

7.5 Conclusion: Choice of detection

Basically there are two ideas for faster snag detection which seem to be suitable to: acceleration measurement at headblock/spreader and improved load measurement on trolley. The advantage of acceleration detection is that it can detect the impact before the forces in the cables actually increase. This option might therefore be faster, but can maximally save 10ms or less because by then the force increase in tremendously and measurable.

Horizontal acceleration is not capable of detecting 4 rope snag, and is less reliable than force measurements. Vertical acceleration does not have the drawback as horizontal, to get a snag the vertical motion of the headblock/spreader always has to stop.

All measurement at the headblock have a very big disadvantage: the impact during normal operation is severe, measurements by Kalmar and Bromma has shown that acceleration up to 40g are not unusual. This results in two important aspects: 1, noise, disturbing the measurement. 2, the impact on everything installed on the headblock is huge. Often equipment and installations on the headblock have a short live.

Taking everything into account, but especially the fact it does not require vulnerable equipment at the headblock and it is more reliable, results in the decision that for now force measuring on the trolley is the best way to detect snag.

Early detection of snag will be done by force measuring at the trolley, this provides fast and reliable measurements. In combination with the fast signal processing device a shutdown signal given within 20ms.

8 *Part 2* : Ultra-fast stopping

The second part of the new way of snag protection is braking. Once snag is detected using the sensors a signal is given to initiate stopping as soon as possible and brake as hard as allowable. We want to limit forces in the ropes and on the structure, requiring very fast stopping but on the other hand we also need to protect our mechanical parts in the drivetrain from too much torque, it is therefore not as simple as slamming on the biggest brakes one can find.

8.1 Timing is everything

As soon as the brakes apply they take up energy, this energy would otherwise be transferred into the ropes and that is exactly not where we want it. The sooner the brakes start consuming energy the better, therefore the development of ultra-fast brakes was initiated, the idea comes from Rene Kleiss, former vice president of Kalmar STS division, the development of the brakes comes from Sibre.

8.1.1 Reduce total system energy

If the detection of snag is quick enough to detected snag and provides a shutdown signal before maximum torque is reached, then the motors can be shut down earlier and thereby the total energy involved in the snag is reduced. This is only possible when the snag system interact with the crane PLC and drives. This might not work for all snag scenarios, but at least for the slow speed snag this can reduce the torque increase tremendously, since there is a lot of time used for the torque increase (ca. 0.3 s).

8.2 Brake factors

Brakes make sure the hoist can stop in every condition, that's why the cranes are equipped with two sets of brakes, operational and emergency brakes as already explained in chapter 2.5.

There are several factors that are important for the effect of braking:

- Clamping force
- Friction coefficient → Brake force
- Effective braking diameter → Braking torque
- Closing time of the brakes.

The first three parameters are chosen in the crane design according to standards mainly regarding safety and operational conditions.

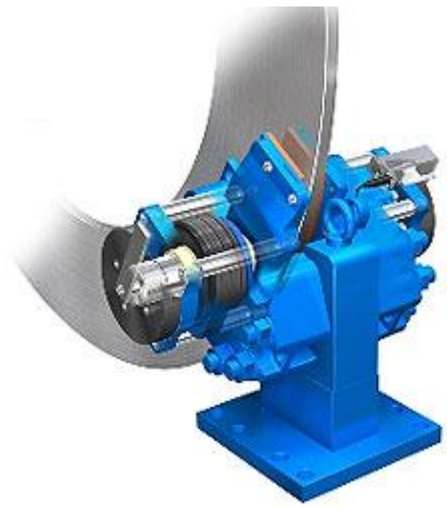

Figure 42: SHI Brake [Sibre]

8.2.1 Closing time: to limit max rope force

The closing time is very important to cut-off the increase in rope force. The closing time is the time the brakes need from activation to actually applying a brake torque (often 90%). The kinetic energy stored in the inertia has to go into the brakes and not into the cables. As already shown in previous chapters for snag initiation up to complete standstill only takes about 0.5 seconds, thus the brakes should close very fast. Conventional emergency brakes close in 300-400ms, which is too slow, considering also the detection and control part has to be executed. For this very purpose of snag protection Sibre developed, after suggestions of Rene Kleiss (Kalmar), faster closing brakes.

8.2.2 Limiting factor for braking

Only focusing on the rope force, it would be the best to slam on the biggest brakes and stopping the drivetrain in a split second. Because this would simply mean no more drum rotation and therefore no more stretching of the ropes. This would protect the ropes but very likely destroys the entire drivetrain. It was mentioned before in chapter 2.5.4, that for fast braking one has to take into account the inertias and maximum moment on components, these give a maximum allowable decelerations: $M = I * \alpha \rightarrow \alpha_{allowable} = \frac{M_{max}}{I}$.

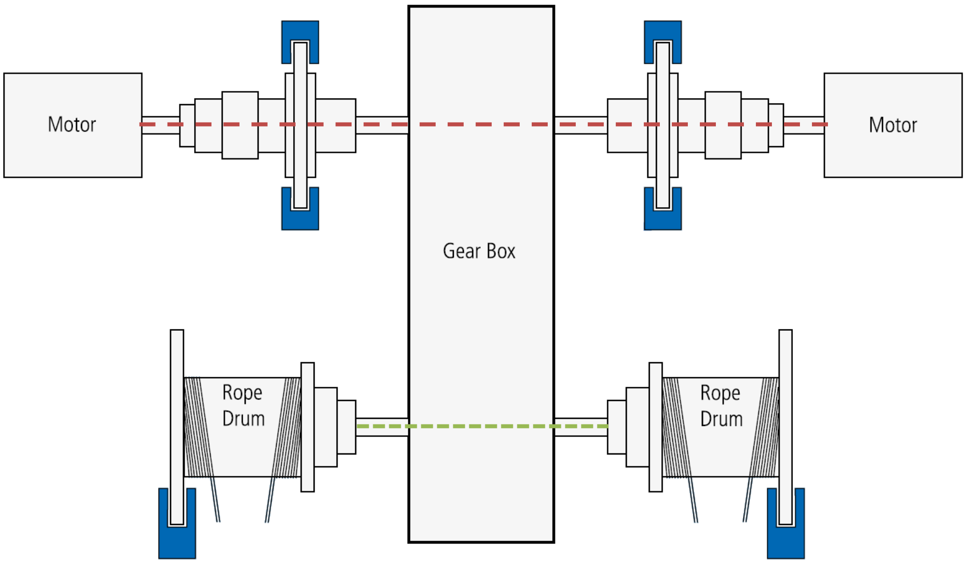

Figure 43: Schematic view of hoist drivetrain, RED: high speed, GREEN: low speed, BLUE: brakes.

If we look at Figure 43 one can see that especially the motors play an important role in this limitation, they are big lumps of inertia on the end of an axle, since the idea is to start braking on the drum the moments required to decelerate all the rotating parts have to go through the entire system: *motor->axle->coupling->axle->gears->axle->drumcoupling->drum*. For now the motors therefore limits the possible deceleration. Braking with operational brakes or by reversing torque would keep the torque on the high speed shaft and thereby protect the gearbox.

8.3 Braking

The drivetrain shows that there are three options to brake: using the emergency brakes, the operational brakes and/or to reverse torque.

8.3.1 Emergency brakes

The SHI brakes from Sibre are applied as emergency brakes on the hoist drum. In case of snag these brakes should apply to stop the hoist motion. The brakes are normally opened by oil pressure, so to close the brakes the oil has to go out, taking 300-400ms. It is possible to make this faster by using of a special Hydraulic Pressure Unit (HPU). A vacuum "expansion" vessel and quick valves. The fast setting SHI brakes close within 80ms and by then apply 90% of the braking torque. These brakes are produced and being tested at the site of Sibre at the moment

8.3.2 Operational brakes

Applying brakes on the high speed shaft would take up much energy and thereby reduce the energy that goes into other axles and especially the gearbox. This would mean that besides very quick emergency brakes also the operation brakes have to be ultra-fast. Sibre also works on faster operational brakes, their improved SLP Texu brakes should achieve a closing time of 40ms and are currently under investigation.

8.3.3 Reversing Torque

When reversing torque, slowing down the motor internally, there is no external moment on the high speed shaft or coupling caused by the motor inertia. This obviously depends on the capabilities of the installed drive and motors. Modern sophisticated variable frequency drives are able to act very quickly. This means they could help in the braking process, by shutting down very quick and also by absorbing some rotational energy of the inertia by applying a reversed torque.

Siemens

This idea was discusses with Siemens Netherlands, since this is a partner for Kalmar for drives and motors. This is possible but depends on the installed equipment, especially the Active Line Module (ALM) this is responsible for feeding back the regenerated power to the circuit/grid. Since snag, happens in a short period also the overload would be short and it is possible to do this. The reaction time for a drive after a new instruction signal (for example Snag) is 50ms until the torque is applied. According to Siemens it is best to apply a set torque until the speed reaches zero and then shut down.

ABB

ABB, the partner for Kalmar or the APMT MVII project confirms the possibility and claim to be able to apply a reverse torque of 1.8 times the nominal motor torque in 40ms. Applying this on the SOS break coupling causes a break since the torque change in a short time is from 1positive to 1.8 negative: 2.8 times the nominal torque which is more than the break torque. A fixed coupling is designed for 4 times the nominal torque and should therefore have no problems with this.

8.3.4 Drive included in Snag system

It would be best to also include the drive within the snag detection system. Setting an allowable band on the torque increase bases on path, speed and acceleration can help detecting snag and reduces the torque increase in case of snag. For example during constant hoist speed allow a maximum torque increase of 20% or a maximum gradient lower than allowed during acceleration. If this limit is exceeded the drives reverses torque and tries to bring the motor to zero speed, this can be earlier than waiting for the snag system to detect and give the shutdown signal.

8.4 Conclusion

There are three possible ways to stop: emergency-, operational brakes and reversing torque. The ultra-fast brakes of Sibre have a closing time of maximum 80ms, within this time 90% of the braking torque is reached. Reversing torque is possible but depends on the installed equipment. For braking it is important to always keep in mind the maximum allowed moments on the different component. These give a maximum deceleration.

For every crane and drivetrain a suitable combination of emergency brakes, operational brakes and if possible reversed torque has to be selected to match this allowable deceleration.

9 Evaluation of New Proposed snag System

This chapter starts with the principle of the new proposed snag protection system and the related components. Subsequently the design/selection related to crane characteristics, what has to be taken into account. Followed by an evaluation of the system and example based on the APMT cranes for the second Maasvlakte.

9.1 Snag protection system

Taking into account the required reliability for noticing snag and the impact exposure for everything on the headblock make it the best choice to measure rope force at the trolley.

The ultra-fast emergency brakes developed by Sibre close within 80ms. Operational brakes can probably close faster although here the same closing time is taken to be safe since they are still under development.

Detection: Ropeforce measurement in trolley, shutdown in case of rapid force increase.

Reaction: Brosa or Pat-Krüger processing within 30ms.

Time: Including some time for the force to increase and processing it should take maximum 30ms to have a snag detection output signal after initiation, add 5ms for the signal transfer from the trolley to the machine house. The activation signal is therefore within 35ms at the brakes and PLC.

Interaction: Sibre ultra-fast SHI (emergency) brakes closing time 80ms to 90% brake torque
Sibre Texu brakes (operational) brakes closing time 80ms to 90% brake torque
Drives shutdown 50ms after snag signal (if still running) or after reaching max torque

Components: Select low inertia components in the driveline: Wölfer motor, normal clutch.

9.2 Design steps and selection

To select the brakes and investigate the possibility of protection by this proposed system the following steps should be taken in the process of the crane design:

- Reduce inertia of driveline components
- List allowable torques for all components
    - Calculate maximum allowable deceleration
- Calculate static ropeforce
- Calculate force increase due to torque increase
    - limit torque increase in drive as possible
- Calculate rotational energy in the system
    - determine required rope elongation to stop

- If ropeforce is too high apply brakes of proposed snag protection system:
    - 35ms to detect and signal to components
    - 80ms closing time of Emergency brakes
    - 80ms closing time of Operational brakes
    - 50ms to reverse torque
  - Apply as much stopping capacity as required without overruling the maximum deceleration. If the ropeforce is then acceptable the system can be applied.

9.3 Evaluate by APMT example

To determine the effect and possibility of this proposal this is applied to the calculated example of the APM cranes at the second Maasvlakte, as used in calculations and model in chapters 4.4 and 5.4.

First just by applying the Emergency brakes and if it is permitted by torque limitation also by applying operational brakes. The part is concluded by additional measurement to further improve the snag situation.

9.3.1 Emergency brakes

The emergency brakes are replaced by Sibre ultra-fast SHI 282 brakes [21], these have a clamping force of 500kN. For the same model as chapter 5.4 snag initiated at 7.54s so brakes will apply 90% of torque after 115ms (35ms detecting, 80ms closing). Figure 44 shows the result of the Adams simulation with applying the emergency brakes. In the graph the rope force is displayed in red, the blue dotted line represents the cable force without braking, as simulated in chapter 5.4. The maximum rope force now reaches 250kN instead of 300kN, this is a reduction of 20%, at static state after shutdown the difference is even more.

Torque limitation

The green line in the graph represents the deceleration of the high speed shaft. The maximum value is 60500 deg/s^2=1056 rad/ s^2. This is important for calculating the maximum moments in the components, the drive axle is subjected to: $M = I * \alpha = 13.5 * 1056 = 14.255kNm$. This is less that allowable torque in the SOS coupling, (used as system guideline). So it is possible to apply the operational brakes as well. (However if we add half the SOS coupling inertia $M = I * \alpha = 17 * 1056 = 17.9kNm$ which is about the break value, still we will apply operational brakes to see what happens).

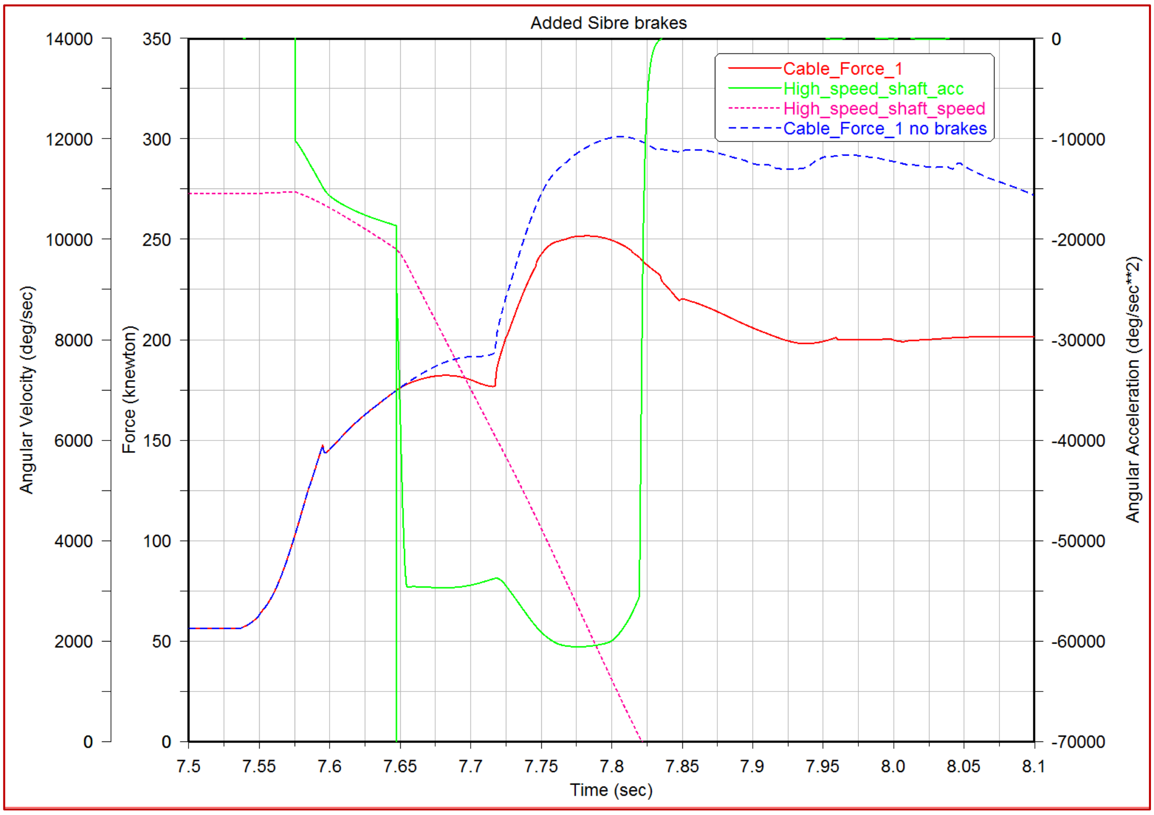

Figure 44: Adams simulation with emergency brakes (red)

9.3.2 Emergency and Operational Brakes

For this simulation the emergency brakes and operation brakes are applied, added Sibre Texu 710-E brakes or equivalent. These brakes can provide a braking torque of 20000Nm per brake. Again assumed that 90% or the braking torque is applied after closing time of 80ms, to be on the safe side. Figure 45 shows the simulation results, in red the cable force of only the emergency brake is shown, the dark blue dotted line gives the cable force when both brakes are applied. The maximum cable force is now less than 225kN(-25%), effectively approximately 200kN(-33%). In practice the operational brakes are likely to close faster and therefore could reduce the force even more.

The cable force fluctuates due to the movement of the headblock, as was also discussed in chapter 5.4, the graph including headblock displacement is given as Appendix 3. Taking this distance into account and calculating towards a sudden stop, a maximum of 85kN has to be added. This would bring the maximum ropeforce to 285 to 310kN, far below the elastic limit and close to the target! This shows the great potential of this proposed system.

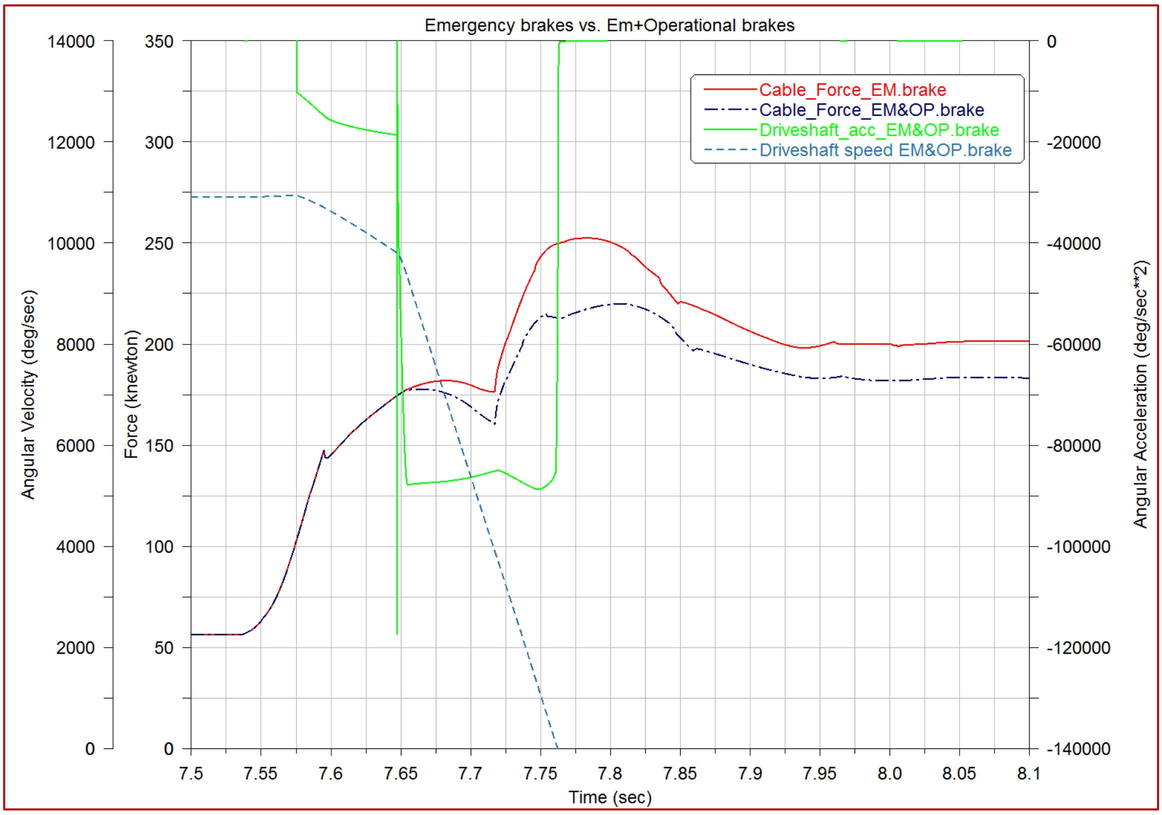

Figure 45: Adams simulation, emergency and operational brakes

Torque limitation

The green line in the graph represents the deceleration of the high speed shaft. The maximum value is 88000 deg/s^2=1536 rad/ s^2. The drive axle is now subjected to: $M = I * \alpha = 13.5 * 1536 = 20.7kNm$. $M = I * \alpha = 17 * 1536 = 26.1kNm$, when including half the SOS coupling Only now most of this is taken by the operational brake so it does not go into the gearbox. The SOS coupling would break at this torque though, therefore suggested to replace this with a fixed coupling, this will be dealt with underneath.

9.3.3 Reduce Inertia, remove SOS coupling

A lower inertia means less energy to dissipate and therefore less rope force. The APMT cranes are equipped with the SOS coupling from Malmedie which has an inertia of 6.636kgm^2, this is far more than a normal coupling. To compensate for this they installed a Pintsch Bubenzer low inertia break disks of 1000mm diameter and 8.25 kgm^2 unfortunately they do have quality problems so suggested is to replace this by a normal coupling and normal brakedisk of smaller diameter.

A Sibre AFC-140 coupling with 710x30 brakedisk has a total inertia of 9.434 kgm^2 and when equipped with the Sibre Texu brake 710 E the brake torque can remain the same. The coupling has a maximum permissible torque of 38.4kN so no problem with handling the brake torque. The reduction of the inertia is at least 10kgm^2 in total, this is a reduction of 15% of the total inertia. Figure 46 shows what this means for the calculation of a scenario 1 snag (2 ropes, high speed), the eventual rope force almost 10% lower, which nearly brings below the elastic limit of the ropes allowable level.

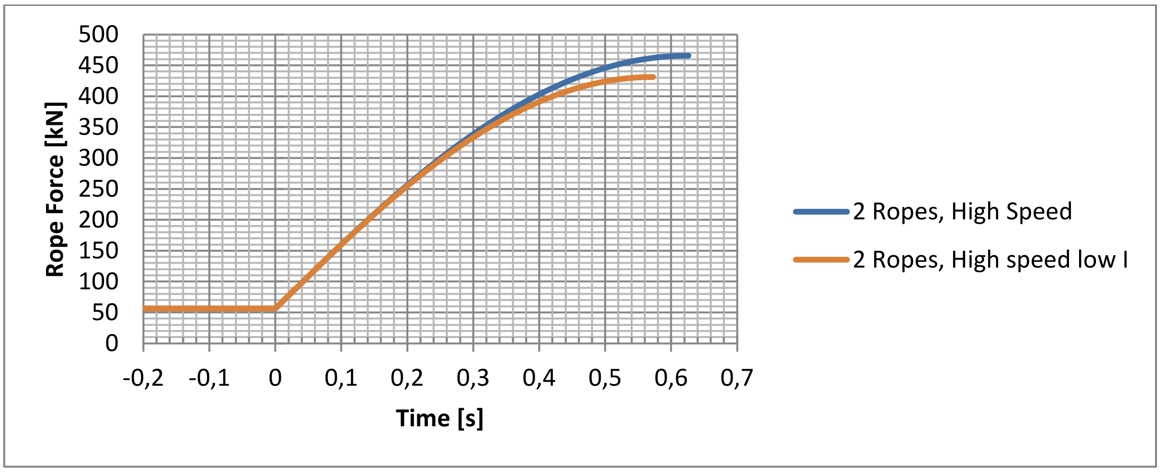

Figure 46: Calculation for reduced I, no brakes

9.3.4 Integrate drive system

Another improvement could be found by integrating the drive in the snag system. A more sophisticated drive control can detect snag and reduce the added energy. If the drives receive the snag signal, they can help slowing down the motors, by applying reverse torque as discussed in chapter 8.3.4 and 2.5.2. This is especially interesting for low speed hoisting since torque increase is large but slow, it takes a lot of time, enough to interact.

The drive always monitors the speed and tries to maintain it, this means the torque increase in case of snag. But if we allow the drive to only increase to torque by a little amount in a short time at constant speed the added energy can be less. For example only 30% increase on nominal torque during constant hoisting speed. For high speed 46 ton hoist, the nominal motor torque is 3.6kNm, an increase of 30% would be 4.7kNm instead of the maximum torque of 5.7kNm. Figure 47 shows the outcome, a reduction only 1.6% compared to initial set up. However for a scenario 2 snag with a 105t load, snag on 2 ropes, the reduction is 28.7%. The maximal rope is now 305kN, just above the target limit by only limiting the torque increase to 30%.

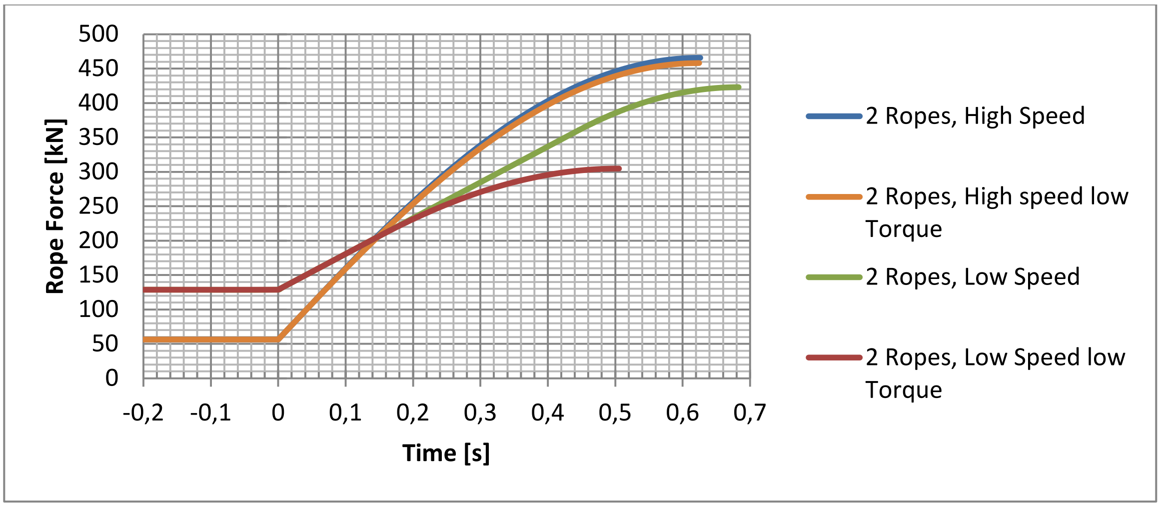

Figure 47: Calculation with limited torque increase, scenario 1 and 3

9.4 Sensitivity analysis of detection time

It is important to investigate what the consequence are when the timing as not as quick as intended. If for some reason more time is needed for detection or the brakeforce is applied slower this can have consequences for the effect of the system.

The model is run with several detection times, the results and detection times are given in Figure 48. All simulations are for high speed snag with only applied emergency brakes. This has to be taken into account when timing changes. Until 50ms the effect is still considered 14%

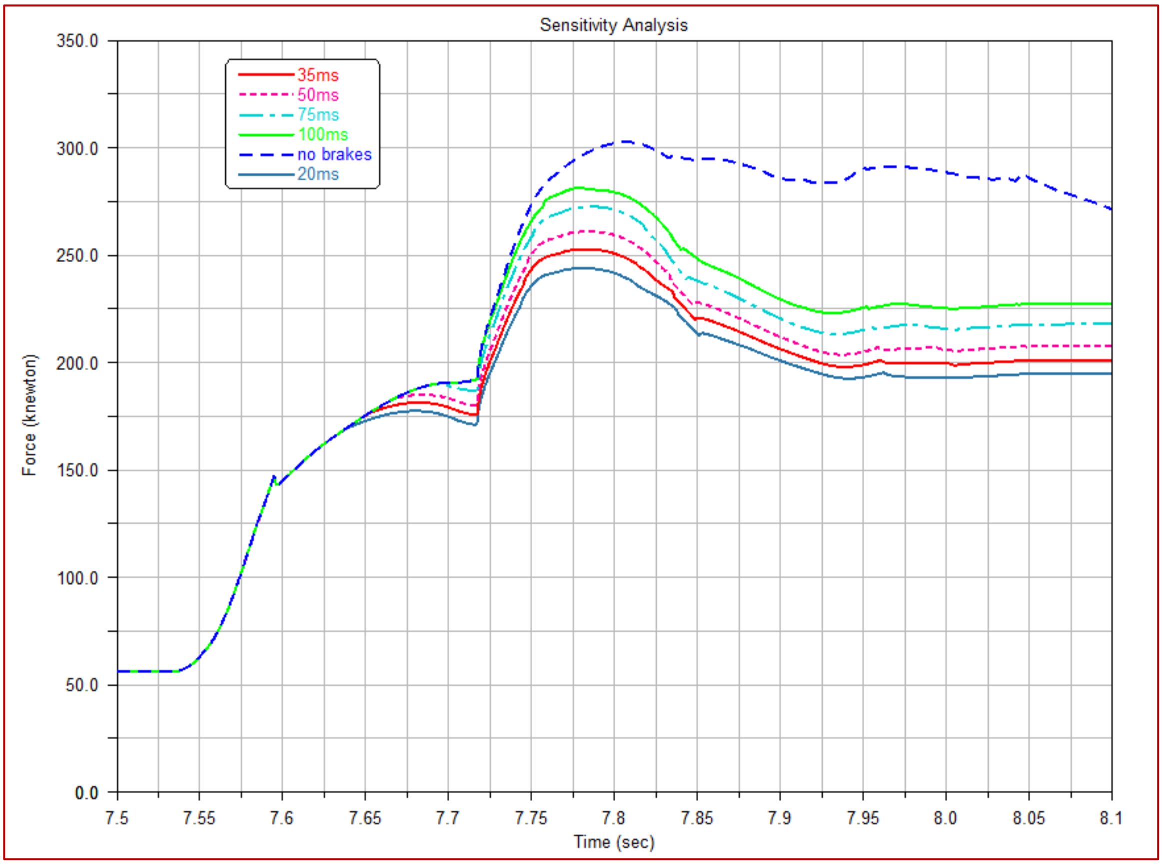

Figure 48: Sensitivity analysis detection time

9.5 Conclusion

The proposed system uses quick detection by ropeforce measurement at the trolley, the detection, processing and signal transfer to the brakes, drives and PLC takes maximum 35ms, probably less. The fast brakes are able to apply 90% of the brake torque within 80ms.

This quick response and actions have a positive effect on the rope force, this obviously decreases. For every application has to be investigated what the allowable deceleration is for the components in the driveline, taking that into account the maximum deceleration can be set by the brakes.

Reducing the inertia is very important to reduce the kinetic energy in the system. By limiting the torque increase in the drives during snag, the rope force can be reduced even more.

The times given (35ms for detection and 80ms for brake closing) are on the safe side, the brakes are currently being tested at the Sibre site, Pat-Kruger and Brosa both work on developing a set-up for detection, which probably will be faster. Although this already has shown to have great potential of this proposed snag protection.

10 Conclusion

Snag is a real problem on STS cranes with current heavy loads and high speeds. A snag event from initiation to complete stop takes only 400 to 600 milliseconds. The ropeforces reach unacceptable values above the elastic limit of the ropes. A snag protection system has to reduce the ropeforce and protect the crane structure and mechanical components. Current snag protection systems are expensive, not proven to work and not easy to implement due to the complex settings. A new proposed snag protection system by Sibre and Kalmar is to detect snag quicker, and then stop the driveline as quick as possible, without special additional equipment. Speed is the key, the earlier we start braking the more energy is dissipated and doesn't go into the ropes.

Detection

First objective is to detect snag, it is important to do this fast but also reliable. The system may not miss a snag event but also it may not give false alarms since this would probably mean system shutdown. Taking everything into account it turns out the best way to do this is by measuring rope force at the trolley. A reliable measurement and detection of snag can be done and transferred to the M-house within 35 milliseconds.

Stopping

Sibre works on ultra-fast brakes, fast emergency brakes are currently being tested and are able to close within 80ms. This means 90% of the braking torque is then applied. The operational fast brakes are also under development, they are likely to close faster but for now it is assumed that also these close within 80ms. A third option for stopping the hoist is by reversing torque on the motor by the drive, but depends on the applied equipment on the crane.

Limitation for deceleration

For stopping the drive line one has to take into account the torque limitation on all components. Torque=Inertia x deceleration. This means for every project/crane an analysis has to be made to determine the maximum allowable deceleration and thereby the maximum allowable breaking torque. Selecting strong components with low inertia is important.

Rope force reduction

With only the brakes and fast detection it is possible to reduce the rope force with at least 30% for the new APMT cranes on the Maasvlakte, this means the maximum ropeforce is far below the elastic limit. If the detection and closing time of the brakes turns out to be even faster, this will even be more. Reducing the Inertia of the driveline and limiting the torque increase in the drive will help lower the force even more.

Concluding

The new proposed snag protection system has a lot of potential, Sibre will test their brakes and come up with final closing times. After development of a force measuring system by Brosa and Pat-Kruger we can know all final times and can make final calculations to see the real effect. Also testing can be started to determine the bandwidth setting of the snag detection in practice.

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Appendix 1. Scientific Research Paper

Improved Snagload Protection System for STS container cranes

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Abstract

Kalmar is a global manufacturer of Ship-To-Shore container gantry cranes. With STS cranes containers are loaded and unloaded from a ship. During unloading it may occur that the load is jammed and suddenly gets stuck inside the ship, this is called snagload. Snagload protection systems are inevitable due to heavy loads and high speeds. The protection systems have to limit the ropeforce to an acceptable level, thereby protecting the ropes and the cranestructure, without damaging the mechanical components by taking into account the maximum allowable deceleration.

Together with Sibre (a German brake supplier) Kalmar came up with the idea to develop a new snag protection system based on early detection and fast braking, without requiring additional equipment. This system has shown great potential by reducing the maximum ropeforce tremendously. The development of the early detection and fast braking components is ongoing.

1. Introduction

Kalmar, part of Cargotec, is specialized in container handling equipment. Containers are the most used standardized box for shipping[1]. For (un)loading of seagoing ships, ship to shore container cranes are used. These STS cranes are designed and developed by a specific division within Kalmar.

During the unloading of a ship it happens that the load gets stuck inside the cellguide. This sudden jamming is called snag, which causes huge ropeforces and thereby extreme loads on crane structure and mechanical components.

Previous studies have indicated that with current heavy loads and high hoisting speeds a snag protection system is inevitable [2]. There are several systems available on the market that claim to protect a crane, however these often have a non-proven working principle, are complex, heavy and expensive.

Kalmar and Sibre think that by early detection and fast breaking these systems/components are unnecessary. This study is executed to get an understanding of the crane, snag and subsequently establish if this ideas has potential.

2. Crane

The function of a ship to shore container crane is to load and unload containers on and of seagoing ships. With ever increasing shipsizes the sizes of the cranes also increase. The terminals and shipping lines demand ever shorter turnover times pushes the performance and speed of container cranes to higher levels.

The STS cranes uses ropes to lift the load out of the ship, the ropes are attached to the hoistdrum and driveline which can simply be seen as a winch. For snag only the hoist mechanism is important and is schematically displayed in the Figure 1. The relevant components are[3]: on the ingoing (high speed) shaft (in red): The motor(s) connected by a coupling to the axle and brakedisk equipped with operational brakes. The gearbox reduces the speed shaft to increase torque on the outgoing shaft(in green). The ropedrums are located on the outgoing shaft and are equipped with emergency brakes on the drum wall.

The motors are currently alternating current motors, fed by a variable frequency drive, capable of lifting heavy loads on nominal speeds and lifting lighter loads up to double this hoisting speed. The available maximum torque depends on the speed.

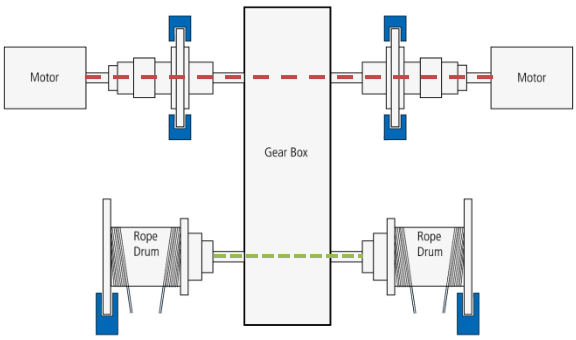

Figure 1: Schematic driveline

The ropes are responsible for lifting the load and can be seen as long spring. The elongation is directly linked to a force increase proportional with the spring factor.

The maximum ropeforce in any circumstance may never exceed the elastic limit, this would permanently damage the ropes and put the crane structure at risk since it is not designed to meet this high loads.

The elastic limit is in general approximately 50% of the minimal breaking strength [4]. In practice is striven to keep the ropeforce below 33% of this break strength. In normal operation the safety factor in the ropes is about 6 (16.7% of minimal breakstrength).

3. Snag event

According to an article[5] in Cargo Handling there are six important factors that affect snag load: behind the factor is indicated how this is represented:

1. **Rotating components**: Inertia's and speed in the drivetrain: rotational energy.
2. **Control design**: response times, plc protocols, overload detection.
3. **Ropes**: Length, E-modulus, diameter, minimal break strength, safety factors.
4. **Brakes**: Closing times, friction factors and applied clamping force.
5. **Centric or eccentric snag**: how many ropes are being stretched during snag.
6. **Snag protection device**: the applied system supposed to reduce effects

All these aspects have to be taken into account and together determine the snag event consequences.

The cause of snag is considered to mostly be a damaged cellguide. Containers are standardized and haves strict dimensions[6]. Lloyd set up regulation for cellguides for storing the containers in, inside the ship's hull. The cellguides have limited clearing and in case of a dent of small damage the load gets jammed and gets stuck: Snag!

For an analysis of the effect of snagload the focus is on ropeforce, this is a good indication for the load on crane and mechanical components and indicates the consequences.

Ropeforce

The ropeforce during a snag event consists of three parts: static, increase due to torque increase and increase by rotational energy. Indicated in Figure 2.

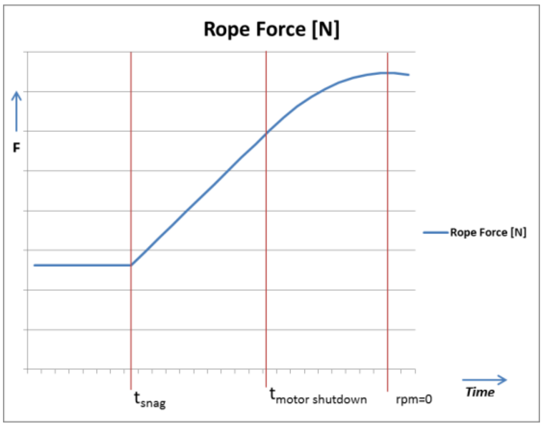

Figure 2: Rope force during snag

Static

During normal operation the hoist speed is constant and the ropes carry the load, until snag the rope force is therefore considered static.

Torque increase

When snag is initiated the load is suddenly stuck, the ropes cannot lift the load anymore. This is not immediately known by the operator, crane system or drives and therefore the motor/drives still try to maintain hoisting speed and keep on pulling on the ropes. The elongation in time multiplied by the spring constant cause a linear force increase. When the torque in the motors reaches maximum, the motors shut down: torque goes to zero.

Rotational energy

Once the motors are shut down the applied torque is zero but there is still residual speed in all rotating components. Due to the moment of inertia these component have a lot of kinetic rotational energy. This keeps on stretching the ropes until all this energy is transferred into the ropes, at that moment the ropeforce has reached its maximum value and the rotational speed reaches zero.

The entire event from snag initiation up to complete standstill takes only about half a second.

4. Energy and force calculation

To get an understanding and indication of these energies, ropeforce and times, calculations have been performed. Based on the new cranes for the APMT terminal at the second Maasvlakte, 4 different theoretical scenarios for snag are analyzed. The scenarios are based on load/speed and stretched wires, they listed below. The rope force is show in the graph.

1. 2 rope snag, 46t load 180m/min (blue)
2. 4 rope snag, 46t load 180m/min (red)
3. 2 rope snag, 105t load 90m/min (green)
4. 4 rope snag, 105t load 90m/min (purple)

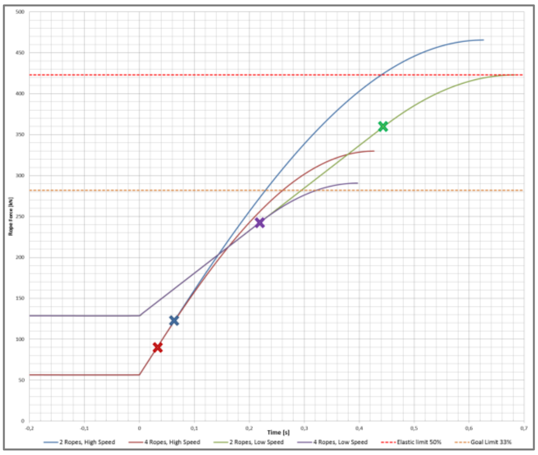

Figure 3: Calculated ropeforce for APMT crane MVII

Indicated in Figure 3 with the crosses is the moment of motor shutdown. The red dotted line indicated the absolute maximum allowable ropeforce: the elastic limit of the ropes. The orange line indicated the target upper limit of ropeforce. This means actions are necessary to reduce the force: snag protection is inevitable.

The times from snag initiation until maximum ropeforce for this crane are between 400 and 600ms. The two low speed snags (purple and green) clearly take much longer for a far bigger torque increase, this is due to the large motor reserve and the slow speed. Limiting the torque increase during normal speed can have a positive effect here. For both high speeds snags this time available to take measures against torque increase is too short.

5. Model

A model is made in MSC Adams, this is a multi-body dynamics simulation program. MSC Adams is used to get more insight in the initiation of snag and a more realistic snag event then the theoretical pure 2 or 4 rope snag. The model is displayed in Figure 4.

The model is based on the realistic reeving and characteristics of a STS container crane. The driveline has a simplified gearbox but contains all drive components with realistic data: speeds, inertia's etc.

In the model a damaged cell guide initiates the snag, this is shown in Figure 5. This is better than the assumption of sudden complete stop as calculated. This will give acceleration and forces that will more match the reality.

The depth and angle of the disturbance can be adapted. Snag in MSC Adams is more realistic, the container is stuck under an angle and therefore stretches all 4 ropes, however two ropes more than the other two. Figure 6 shows the maximum ropeforce for the model relative to theoretical 2 and 4 rope snag.

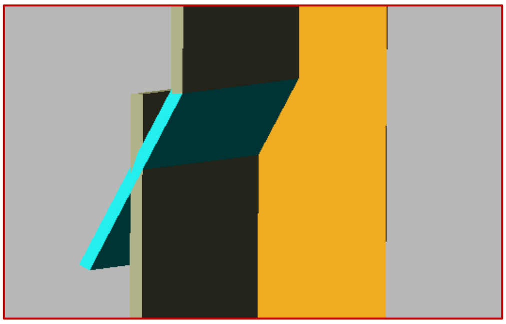

Figure 5: Cell guide disturbance in Adams model

Expected is that the ropeforce of the model is between the two calculated values however it is lower. In the model the headblock does not immediately stop but during snag penetrates and moves a bit into the cellguides. This gives additional ropelength, the distance times the spring constant is 100kN, add this to the value of the model and it fits perfectly between the theoretical values as expected.

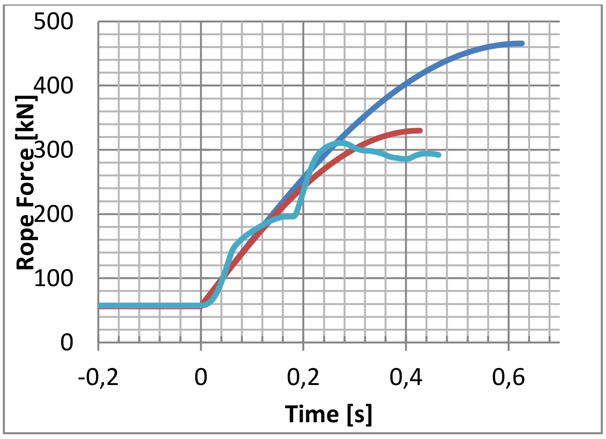

Figure 6: Model compared to calculation

6. Snag protection systems

A snag protection system has to limit the ropeforce to an acceptable level, thereby protecting the ropes and the cranestructure, without damaging the mechanical

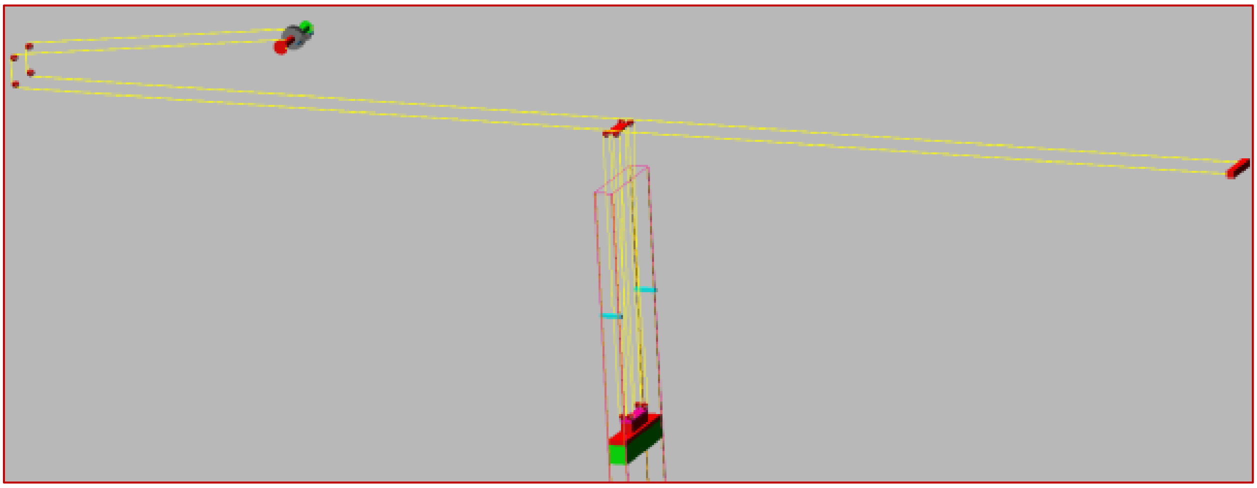

Figure 4: Crane model in MSC Adams

components by taking into account the maximum allowable deceleration. There are basically three options to do this:

1. **Increase ropelength:** reduce ropeforce and increase time for braking
2. **Decouple drivetrain:** reduce rotational energy
3. **Stop drivetrain:** reduce energy input in ropes by dissipation

Conventional systems are often based on hydraulic cylinders that add ropelength, these systems are huge heavy and in time may start leaking. A recent development by Pintsch Bubenzer and Malmedie is called Snag Overload System[7], in short SOS system. This system is based on a break coupling that decouples the motors when the ropeforce gets too high. In theory this is a nice idea, in practice the coupling does not break early enough for snag and adds a lot of inertia to the normal driveline. The system turns out to be quite complex in cooperation with the motors and drives.
Both systems are expensive and have doubtful working principled, therefore Kalmar and Sibre came together to look for a way to protect the crane without additional components.

"New Idea"

Kalmar and Sibre are working on a new approach of snag protection: early detection and fast stopping, without additional components as break couplings and hydraulic cylinders.

7. Early Detection

There are two general options for early snag detection: measuring accelerations or forces. Speed and reliability are very important for snag detection. Speed to be in time to interact and reliability because it is not permitted to miss a snag or give false alarms, this will result in system shutdown by the terminal.

Accelerations
When the load snags the vertical hoist speed goes to zero, a deceleration is always present in case of snag. Sibre came up with the idea of horizontal force measurement, it is believed that the damaged cell guide causes a horizontal acceleration first before the load is stuck. If this horizontal acceleration can be detected, the snag can be predicted. All measuring of accelerations has to be done on the headblock.

Force measurement
Ropeforce measurements can be done at many locations. Currently on most cranes this is measured at the back side of the crane or at the utmost front of the crane. These measurement locations are far from the actual snag location (100-150 m rope). Since the ropes work as spring and due to their weight also tend to sag over the long horizontal distance they tend to whip. Resulting in a lot of fluctuations of force and damping of impact peaks. This is therefore not considered as the good location for snag detection. Measurement on the headblock or spreader would be better.

The spreader and headblock are in normal operation subjected to severe impact. This results in two negative aspects: noise in measurement and damaging of equipment. Therefor this is not seen as a suitable spot either. For that reason acceleration measurement nor force measuring on the headblock will not be done.

The next spot to measure would be the trolley, this is relative close to the snag, and due to the short and vertical cables not subjected to sagging or whipping of the cables. This is therefore seen as the best place to detect snag quick and reliable.

Signal processing
The signal from the installed load cells has to be processed and analyzed. Together with Pat-Kruger the development of this is initiated. The detection is based on analogue detection of a rapid increase of ropeforce in time. This should be safely detected and transferred to the machine house within 35ms.

8. Fast Stopping

Once snag is detected actions can be done, since the snagevent is passed in 400-600ms also this has to quick.

There are three possibilities to stop the hoist motion and dissipate energy: apply emergency brakes, operational brakes or reverse torque in the motors. These can be combined to match the desired deceleration for each crane.

Conventional emergency brakes close in 300-400ms and are therefore not quick enough. Sibre developed ultra-fast emergency brakes which can deliver 90% braking torque in 80ms. These brakes are currently being tested. Also operational brakes are in development, these are likely to close faster but for now it is assumed that the same closing time will be achieved, until proven otherwise.

Reversing of the torque is possible in current sophisticated AC motors with frequency drives. ABB and Siemens claim that a reverse torque can be applied within 50ms. This can help reducing torque on components by slowing down the motor internally.

Limitation of deceleration
To limit the ropeforce it would be best to stop the driveline in a split second. If this was theoretically possible it would mean also the drive stops this fast as well, this huge deceleration in combination with the inertia of the components results in too high

moments, breaking gears, couplings or axles. The maximum deceleration should therefore be calculated and be kept in mind when designing and selecting the stopping equipment.

9. Evaluation

In the model in MSC Adams the brake possibilities are experimented with. The graph underneath shows the cable force without braking in blue, and with emergency brakes in red. The motor angular acceleration is displayed in green as well as the motor speed in pink.

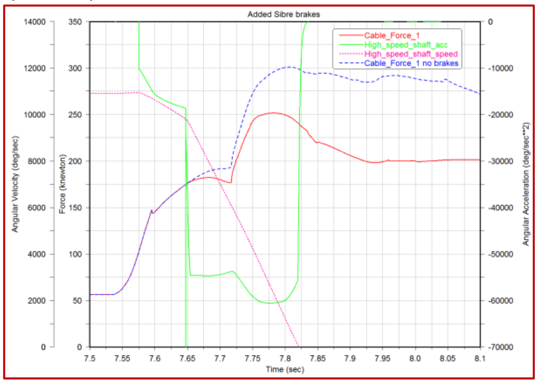

Figure 7: Adams simulation rope force when no brakes [blue] and applied Emergency brakes [red]

By only applying the emergency brakes the maximum ropeforce is reduced with 20%. The acceleration results in acceptable torque on the mechanical components. Therefore also the operational brakes are applied resulting in an effective reduction of almost 30%. The maximum ropeforce is now only 225kN, far below the target limit, if this is calculated towards a more sudden stop with no headblock movement the force would just be over 300kN which is only 20kN over target limit and far below the elastic limit. This system has shown its potential. For every project/crane an analysis should be made to determine the applicable brake torque.

Reduction of torque increase

Especially for low speed snag the torque increase by the drives causes a large force increase, if in the sophisticated drives the allowable torque increase is limited to 30% for constant speed, instead of the maximum available a significant reductions can be made. The graph in Figure 8 shows both 2 rope snag scenarios, high speed in blue, low speed in green. The same calculation is shown with limited torque increase (30%), orange shows the ropeforce for high speed, the red line represents reduced torque for low speed hoist. Note that this is without applying brakes.

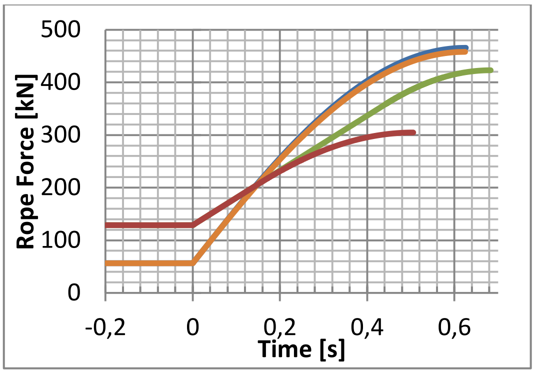

Figure 8: Calculated effect of torque limitation

As one can see the consequences are small for high speed snag but for low speed snag the effects are huge. This is due to the fact that for low speed snag the effect of motor torque was far more than for high speed snag. This should be taken into account in a sophisticated drive. For high speed snag the rotational energy is the biggest source of ropeforce.

Reducing inertia

If for the analyzed crane the special SOS breakcoupling would be replaced by a normal coupling the total driveline inertia decreases by 15%. This causes a drop in the ropeforce of almost 10%, without braking. This is illustrated by the graph in Figure 9, the orange line shows ropeforce for reduced inertia compared to the blue line of scenario 1 snag with SOS coupling.

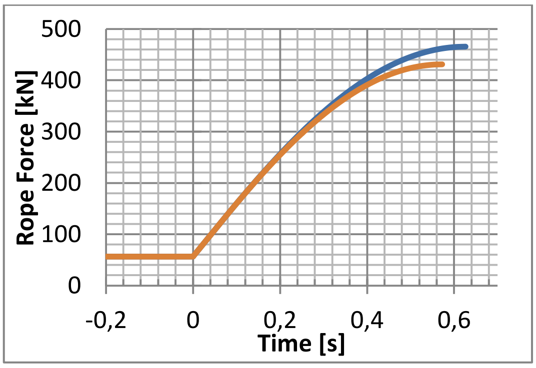

Figure 9: reduced inertia high speed 2 rope snag

10. Conclusion & Recommendations

Snag is a problem and current protection systems have an often non proven working method, they are big, complex, heavy and expensive. Therefore Kalmar and Sibre came up with a new idea for a snag protection system based on early detection and fast stopping.

Calculations and modelling have shown that an entire snag event only takes 400-600ms. The theoretical calculations show higher ropeforces than the model due to the difference in sudden stop and actual jamming of the load. Both however show the need of protecting the crane against too high ropeforces.

For the detection of snag speed and reliability are very important. Based on current ways of measuring and drawbacks of measurements at the headblock (noise and impact), it is decided that the best way to detect snag is by rope force measurement at the trolley. Including the detection, analyzing and processing, snag can be detected in 30ms after initiation. This means snag signal can be at the interacting components within 35ms.

There are three options for fast stopping the driveline: emergency brakes, operational brakes and reversing torque. Sibre's ultra-fast brakes reach 90% torque within 80ms. ABB and Siemens can apply reverse torque after 50ms.

The mechanical components in the driveline have maximum permissible internal torques. These give allowable maximum deceleration. Stopping faster will result in damaging components.

The stopping/braking should be designed to not exceed this maximum allowable deceleration. Apply as much braking as required to reduce the ropeforce to an acceptable level and preferably the target limit of 33% of the minimal break strength.

For the new APMT cranes on the second Maasvlakte this snag protection proposal has been evaluated and show significant improvements, the maximum ropeforce was reduced by at least 30% and thereby far below the elastic limit!

Recommendations

Pat-Kruger and Sibre both work on the further development of the components, when finalized real life testing can start followed by implementation.

More carefully selecting components for the drivetrain should result in a decrease in inertia and thereby reducing rotational energy.

Limiting the torque increase of the motors during constant speed has shown to have great potential, especially for low speed snag.

By taking all into account and by engineering with straight forward common sense, the consequences of snagload can be made acceptable.

13. References

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Appendix 2. Calculation of Energy

Calculation sheet Ropeforce & Energy

Specs for APMT		
k 1 rope	174308	[N/m]
# rope	4	
# snag rop	2	
Load	105000	kg
rpm	900	rpm
I	65	kgm2
i	15,8	
r drum	0,5	m
w	*94,25*	*rad/s*
k static	*697231*	*[N/m]*
k snagged	*348615*	*[N/m]*

Part 1: static		
F rope tota	514849	N
u static	0,738	m
u by vhoist	2,983	m/s

Part 2: increased motor T		
# motors	2	
M max	15422	Nm
F drum	974670	
F increase	459821	N
u T increas	1,3190	m
time	0,4422	s
u s+ t incr.	2,0574	

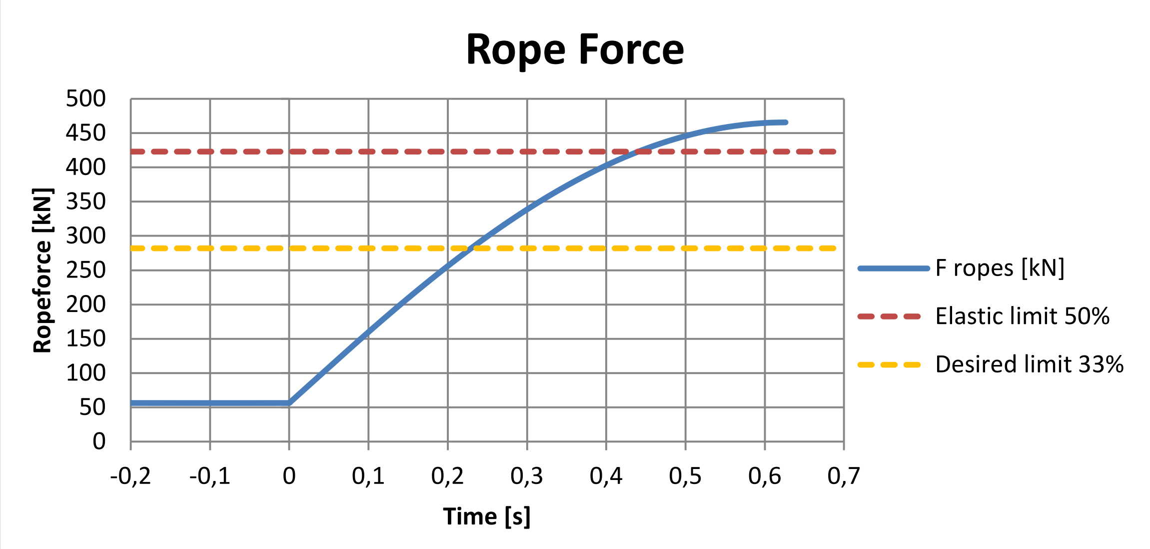
Part 3: Inertia		
E kin	288686	J
u add	**0,3693**	m
E pot	288685	J
Eleft	-0,68	J

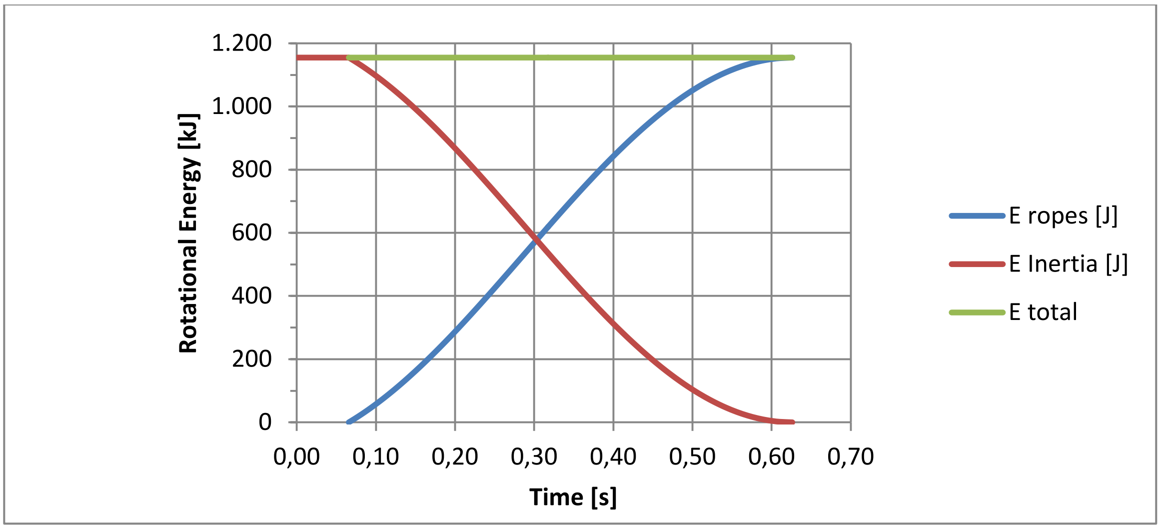
Rope force limit	
Elastic	50%
Target	33%

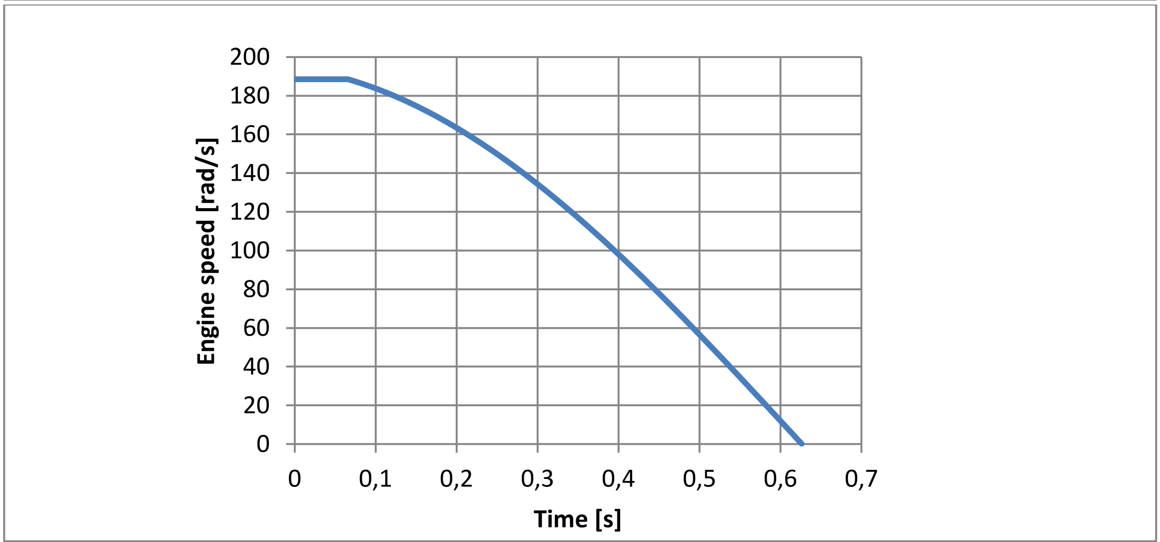
	u added	Theta drur	Theta mot	E add rope	Mdrum	M motor	Alpha	E total	E Rotation	w eng	t step	t total	F rope	*step*	U totaal
	[m]	[rad]	[rad]	[J]	[Nm]	[Nm]	[rad/s2]	[J]	[J]	rad/s	[s]	[s]	[kN]	#	[m]
Part 1		0	0	0				288686	288686	94,25		-0,2	**128,71**	*0*	0,7384
Part 2				0	358623	22698	349,1947	288686	288686	94,25	0	**0,4422**	**358,62**	*100*	2,0574
Part 3	0,36934	0,73868	11,671	288685	423002	26772	411,8809	288686	0,683	0,145	0,007185	**0,6831**	**423,00**	*1000*	2,4268

Graph calculation scenario 1

2 rope snag, 46 ton load, hoistspeed 180m/min

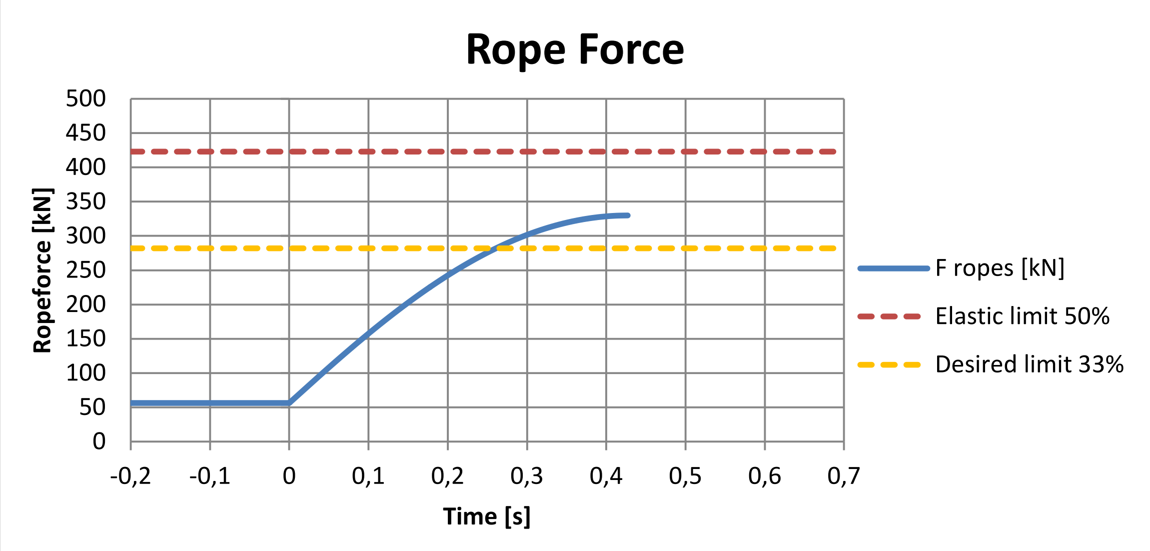


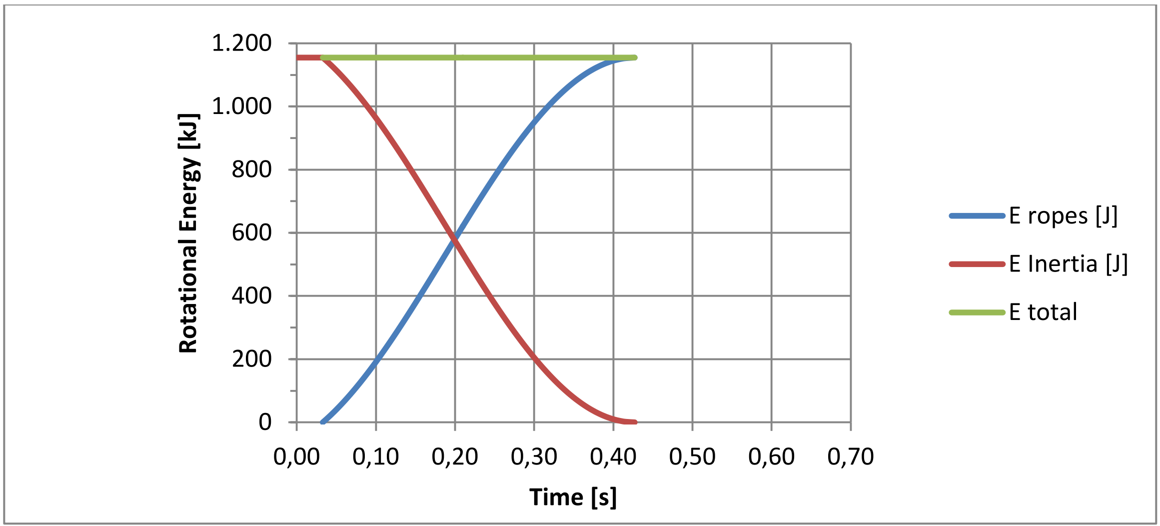


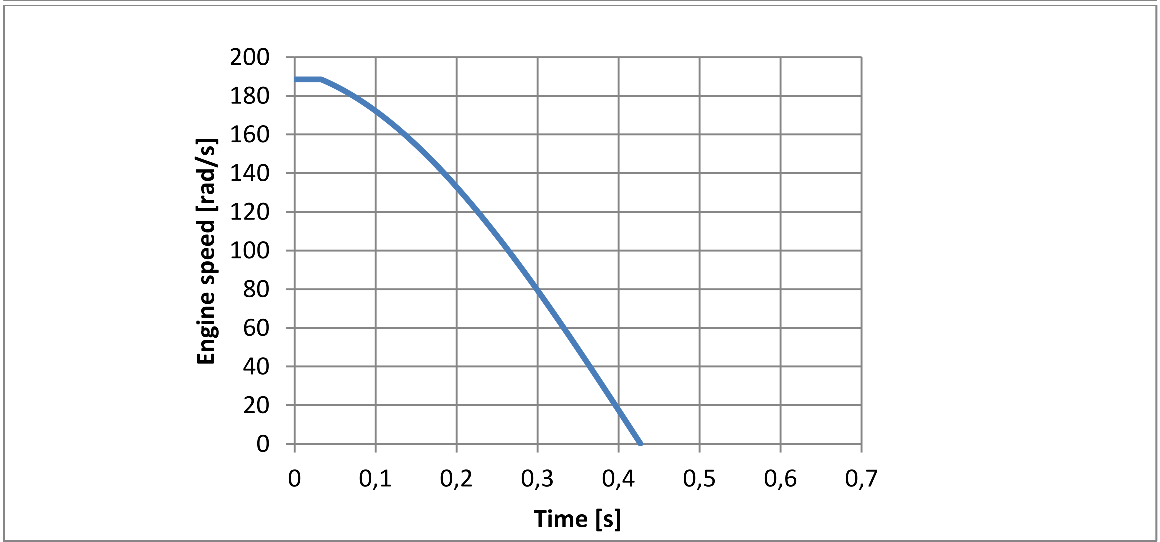


Graph calculation scenario 2

4 rope snag, 46 ton load, hoistspeed 180m/min

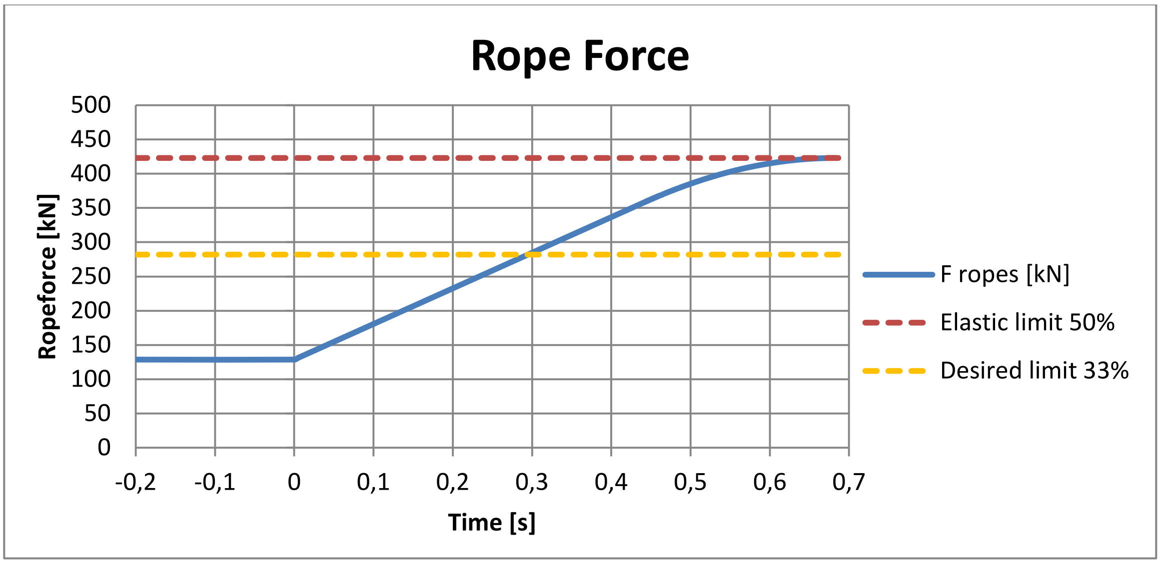


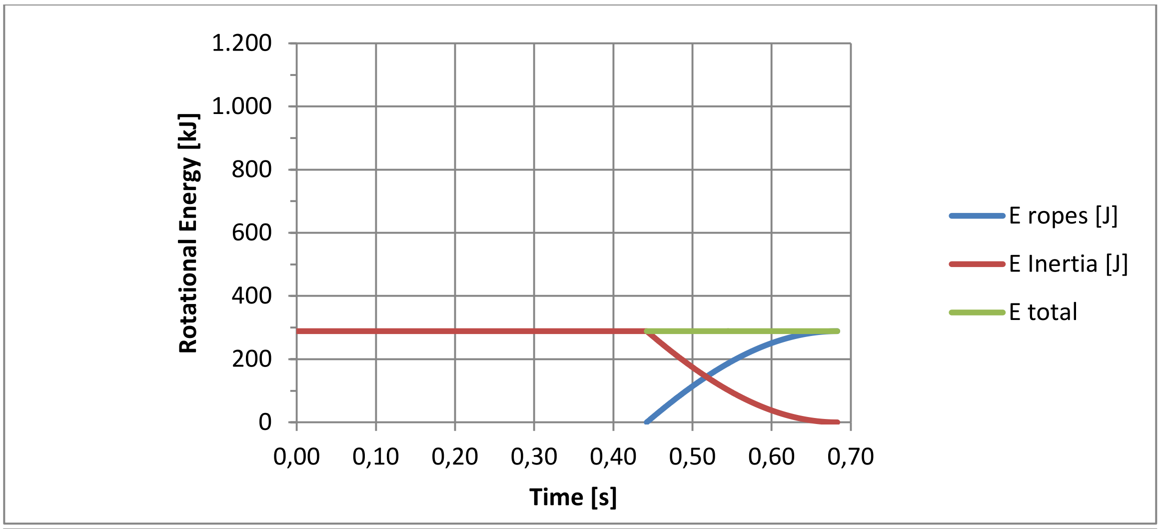


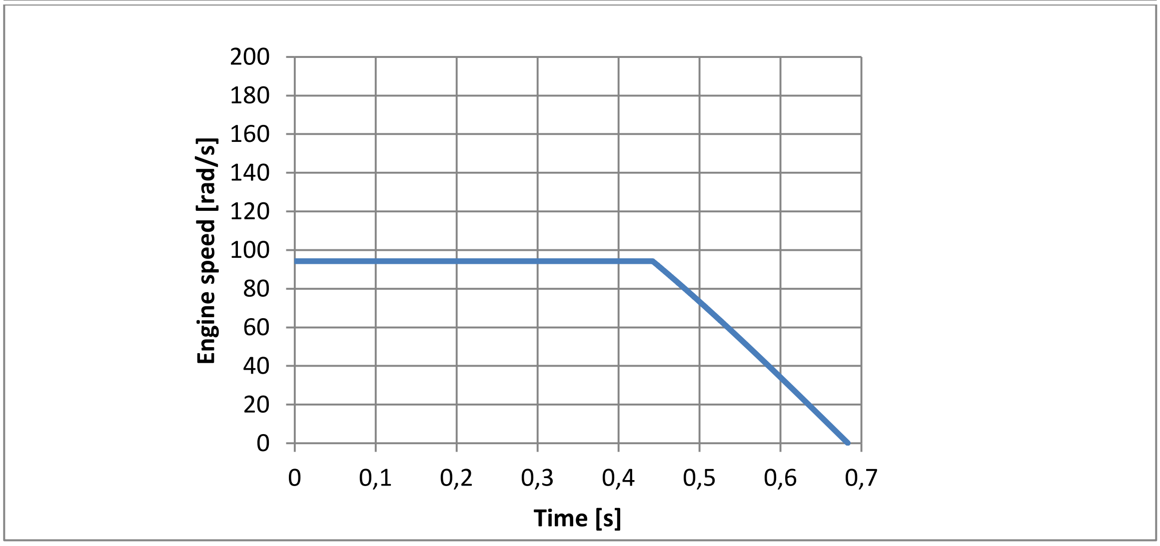


Graph calculation scenario 3

2 rope snag, 105 ton load, hoistspeed 90m/min

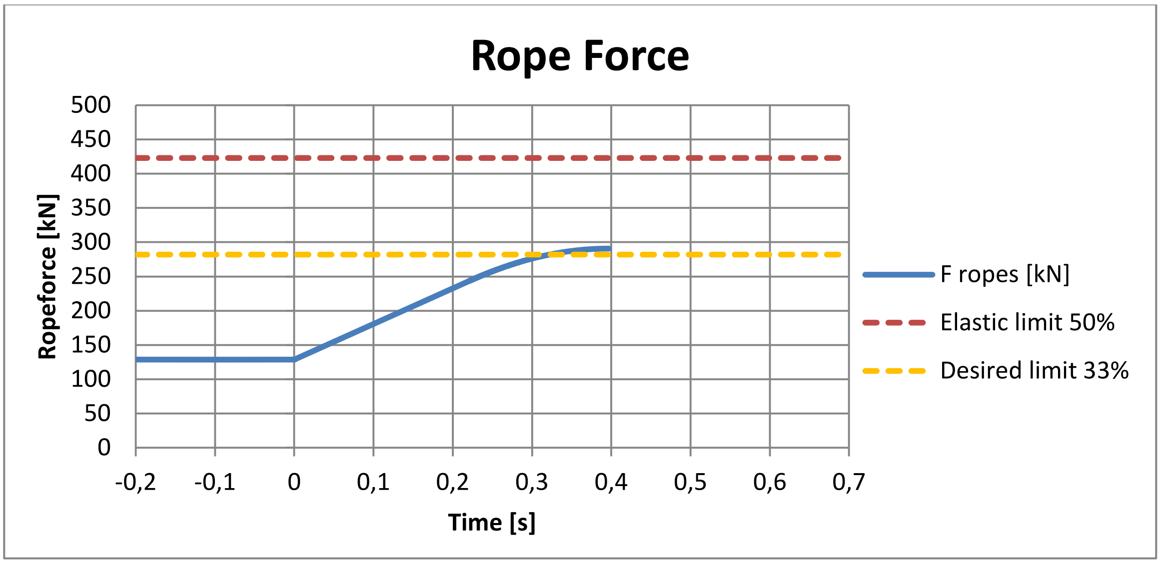


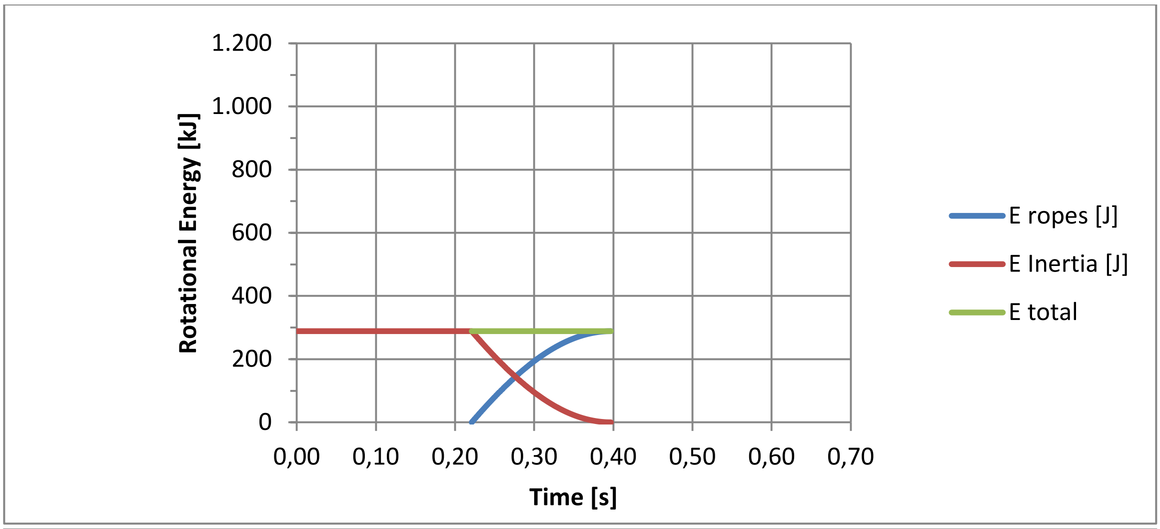


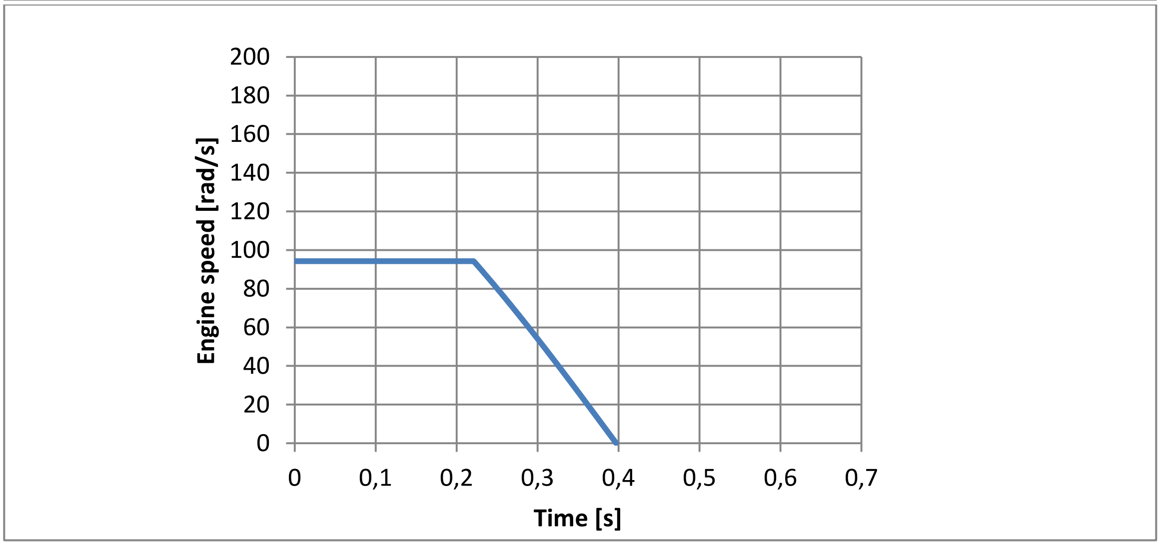


Graph calculation scenario 4

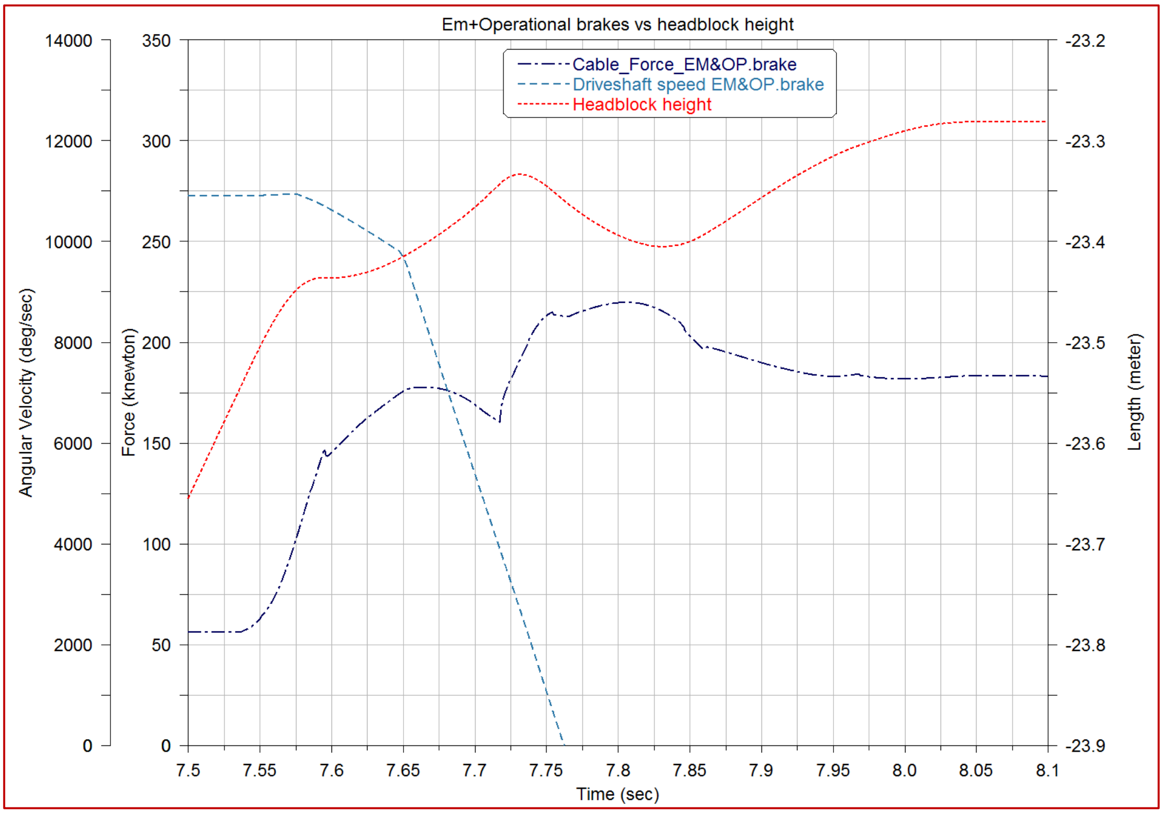
4 rope snag, 105 ton load, hoistspeed 90m/min







Appendix 3. Adams Graphs



Ropeforce fluctuation and headblock displacement