Schedler Michael, BSc

# Deduction of an analytic design calculation for the dimensioning of chain conveyors based on multibody simulation systems

## Masterarbeit

zur Erlangung des akademischen Grades

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Graz, am 16.02.2017

## STATUTORY DECLARATION

I declare that i have authored this thesis independently, that I have not used other than the declared sources / resources, and that i have explicitly marked all material which has been quoted either literally or by content from the used sources.

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# Abstract

Nowadays nearly everything can be simulated. No matter which field the problem statement is following, nearly always a specialised tool is available which gives engineers the opportunity to generate data which in former times, had to be measured in experiments from already existing systems, machines or test benches. This has the effect that in many cases the development costs are dropped massively because just virtual, so no physical existing machines and system have to be built to get the needed data. The concept "simulation" has on one hand for sure a lot of advantages but on the other hand also disadvantages and emerging problems can lead to major troubles within a project.

In many cases, complex design calculations get validated through tests on test-benches or simulations and often the complexity of the problem statement requires more than one simulation and sometimes even additional measurements to ensure the required accuracy of the design calculation. For this reason, a detailed planning of the tasks, the needed simulations and the measurements has to be done to fulfil the problem statement in an economic and efficient way. This thesis deals exactly with this kind of problems, so how to find the best fitting path between analytic calculation, simulation and measured data.

So, the main task of the thesis is to develop a user friendly analytic design calculation for chains which are used in long conveyors (up to 300m) as traction elements within a failure tolerance of ±5%. As foundation for the validation of it, multibody simulations of the needed conveyor systems were used to bring the usage of data which has to be measured on existing system or test benches to a minimum. It is obvious that an analytic calculation with a low grad of complexity will not reach the required accurateness without using several approaches for corrective factors. Accordingly, also the corrective factors where validated with the help of different simulations.

Even with modern simulation techniques and equipment, multibody simulations of conveyors with the required length are hard to realize because of the high number of single parts and the high grade of complexity. Within the thesis several approaches are shown how to handle this kind of simulation problems.

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# Kurzfassung

In der heutigen Zeit kann nahezu jedes physikalische und technische Problem simuliert werden. Für fast jede Problemstellung sind bereits spezialisierte Simulationsprogramme am Markt erhältlich. Dadurch ist es für die Ingenieure der heutigen Zeit möglich, Daten welche früher mühsam durch Versuche und Messungen an bestehenden Anlagen und Prüfständen generiert wurden durch virtuelle Versuche (Simulationen) zu "generieren", wodurch eine flexiblere und Kosteneffektivere Alternative zu den klassischen Versuchen und Messungen zugänglich wurde. Das Konzept der Simulation hat viele Vorteile, jedoch auch einige Nachteile, welche zu großen Komplikationen innerhalb eines Projektes führen können.

In vielen Fällen werden komplexe Auslegungsberechnungen mit Hilfe von Versuchen, Messungen und Simulationen validiert, um einen Nachweis und auch eine Bestätigung der Genauigkeit der Ergebnisse zu erhalten. Dadurch ist es vor allem bei komplexen System nötig mehrere Messungen oder Simulationen an verschiedenen Systemen durchzuführen. Aus diesem Grund ist es wichtig die genaue Vorgehensweise bei der Durchführung der Messungen und Simulationen zu planen (Versuchsplanung), um die Problemstellung in einer effektiven und wirtschaftlichen Art und weise zu lösen. Die vorliegende Arbeit beinhaltet genau solch eine Problemstellung, nämlich den best geeignetsten Weg zwischen analytischer Auslegungsberechnung, Mehrkörper Simulationen und gemessenen Daten zu finden.

Weiters beinhaltet die Aufgabenstellung, die Erstellung einer anwenderfreundlichen Auslegungsberechnung für verschiedene Variationen von Kettenförderern mit Längen bis zu 300m zu erstellen. Die auftretende Fehlertoleranz liegt dabei im Bereich von ±5%. Als Grundlage für die Validierungen der Berechnungen dienen dabei Mehrkörper Simulationen der benötigten Systemvariationen um so die Verwendung von Daten, welche von Messungen an bestehenden Anlagen und Systemen resultieren, zu minimieren.

Auch mit modernsten Simulationsprogrammen und Computersystemen, sind Mehrkörper Simulationen von Kettenförderern dieser Längen aufgrund der großen Anzahl an Komponenten nur schwer realisierbar wodurch ein Einsatz von diversen Ersatzsystemen notwendig ist. Die Erstellung dieser Ersatzsysteme ist ein weiterer Eckpfeiler der vorliegenden Arbeit.

# Vorwort

Die vorliegende Masterarbeit wurde im Zeitraum vom September 2016 bis Februar 2017 im Rahmen meines Masterstudiums Production science and Management an der TU Graz gefertigt.

Mein besonderer Dank gilt dem Leiter des Institutes für Technische Logistik Herrn Prof. Jodin, der Firma Beumer und meinem Betreuer Ass.Prof. Dipl.-Ing. Dr.techn Landschützer Christian für das große Vertrauen, welches sie in meine Fertigkeiten gesteckt haben, um diese Arbeit zu meistern. Zusätzlich möchte ich Herrn Dipl. Ing. Ortner Pichler für die vielen Stunden der Betreuung und den fachlichen Diskussionen danken. Ich bin davon überzeugt, dass ich nur durch die kompetente Führung mein potential voll ausschöpfen und so die doch sehr anspruchsvolle Arbeit meistern konnte. Ich habe sehr viel Fachliches aber auch sehr viel über mich selbst gelernt.
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Des Weiteren möchte ich natürlich meiner Familie, besonders meiner Mutter und meinem Vater, für die jahrelange Unterstützung während meines Studiums danken. Zusätzlich möchte ich meiner Oma für die unzähligen Rosenkränze die sie für mich gebetet hat, wenn wieder eine schwere Prüfung anstand, meinen Onkeln Christof und Sigi, meiner Tante Sonja sowie Bernd und natürlich meinem Schwesterherz für ihre Unterstützung auch in nicht so rosigen Zeiten danken. Ich möchte zudem meine Masterarbeit meinem Opa Siegfried Rüscher widmen.

Natürlich möchte ich auch meinen Freunden Sto, Judithle, Tini, Alex, Gebat, Kjetil, Mario, Lukasino und meinem großartigen Schatz Betz dafür danken, dass sie alle mich in Zeiten der Freude aber auch in Zeiten der Enttäuschung mit diversen Aktivitiäten und auch Festivitäten abgelenkt bzw. erfreut haben und dafür, dass sie immer für mich da waren wenn ich sie gebraucht habe.

Ohne euch alle würde ich nicht dort stehen wo ich heute bin.

Vielen Herzlichen Dank!

*“As gaut alls, wenn ma werkle wett!”*

*Eine Lebensweisheit aus dem Bregenzerwald*

# Nomenclature

## *index of abbreviations*

| | |
|---|---|
| DoE | Design of Experiments |
| e.g. | exempli gratia |
| etc. | et cetera |
| FFT | Fast fourier transformation |
| MBS | Multi-Body-Simulation |
| rpm | rounds per minute |
| CAE | computer aided engineering |

## *Formula index*

| | | |
|---|---|---|
| $C_L$ | [mm] | cell length |
| $C_W$ | [mm] | cell width |
| d | [mm] | diameter guiding roller |
| f | [%] | filling level cell |
| $F_{Vor}$ | [N] | preload |
| k | [-] | number of levels (DoE) |
| $L_I$ | [mm] | plane length Section I |
| $L_{III}$ | [mm] | angular length Section III |
| $L_{IIB}$ | [mm] | arc length Section II |
| $L_{IVB}$ | [mm] | arc length Section IV |
| $L_V$ | [mm] | highten plane length Section V |
| $L_{total}$ | [mm] | total transportation length |
| p | [mm] | chain pitch |
| $p_{GR}$ | [mm] | pitch guiding rollers |

| | | |
|---|---|---|
| $R_{II}$ | [mm] | radius Section II |
| $R_{IV}$ | [mm] | radius SectionIV |
| v | [m/s] | transportation velocity |
| $C_{LI}$ | [-] | Corrective factor for parameter LI |
| $C_{RII}$ | [-] | Corrective factor for parameter RII |
| $C_{LIII}$ | [-] | Corrective factor for parameterLIII |
| $C_{RIV}$ | [-] | Corrective factor for parameter RIV |
| $C_{LV}$ | [-] | Corrective factor for parameter LV |
| $C_{\alpha}$ | [-] | Corrective factor for parameter α |
| $C_{RP}$ | [-] | Corrective factor fo run up process |
| α | [°] | angle pitch |
| ρ | [kg/L] | densitiy bulk |
| $\mu_{GR}$ | [-] | friction coefficient guiding roller |
| $\mu_{CB}$ | [-] | friction coeficient chain bolt |

# Table of Contents

# 1 Problem statement

Faster, higher, longer and more efficient, these are keywords that fit into nearly every field of our lives. In times where the community and especially industries are looking for “best” technological and at the same time the most economic solutions for their problems, engineers are facing problems that are very often not that easy to solve.
One of these fields is the field of logistics and transportation technologies.

In this thesis, the problem is discussed how to find an approach to calculate the chain forces of long distance (>300m) apron conveyors with the help of Multi Body Simulations (MBS) and measured data of an already existing system of this type.

The problem in former times was to find a way to calculate a large number of variants (>100) of designs of the conveying system, where the approach of the analytical calculation is still valid and within the expected accuracy. The engineers had an easy way to get rid of these insecurities in respect of design calculation of conveyors and added a security factor of 7 to 10 to ensure the confidence of the system.
It was not the ideal solution, however there were no other possibilities at that time to develop calculation approaches fitting to a high variant of designs.
To find a more suitable solution for this problem, it is absolutely necessary to find ways to validate the analytical calculation.
One possible solution is to do a lot of tests and measurements on already existing systems. This was, and is, not always possible due to the high amount of costs and effort, and even with a lot of measured data there would be no guarantee that all the required data is available to obtain the validation of the analytical calculation so that its suitable for “all” expected system variants. In other words, all variants which are necessary for validation are not already built to gain the needed
So, the idea was to generate “measured” data not of already existing systems, but rather out of multi-body simulations that have the same physical properties as the real system. This gives the engineers a flexible and cost effective way to get every type of data for every system variant they need.

Exactly this is the task of this thesis, to find the path between real system, simulation and analytical calculation to develop an approach for the analytical calculation of the chain forces for a high variant of system designs with an acceptable accuracy.

# 2 General procedure for solving the problem statement

The base for the solving procedure is the analytic calculation of the chain force. It is required that this calculation is as simple as possible and at the same time so accurate that it can be used as a design calculation for chain conveyors.

To ensure that the calculation is accurate with every needed variation of the system, it is necessary to take a close look on the validation and further on, to determine the corrective parameters.

Out of this requirement the following approach was developed:

- Creation of an analytic calculation of the chain force
- Modelling of a multibody model which is based on an already existing system
- Validation of the multibody model with the help of measured data of the existing system
- Analysis of all relevant parameters and developing of an experimental design plan with the help of Design of Experiments (DoE)
- Modelling of all needed multibody models
- Determination of corrective parameters for the analytic calculation

# 3 Theoretical Introduction

## 3.1 Explanation of the chain conveyor system

Chain conveyor systems belong to the family of continuous conveyors. They transport goods (mostly bulk material) continuously from one place to another. In this Thesis, only continuous conveyors with a chain as traction element, on one or more lanes, are discussed. The chain itself is also the carrier for different attachments that capture the goods or bulk that is transported. There is a large variety of different attachments that are dependent on the transportation path and transported goods. Figure 1 shows some examples of different attachment styles.

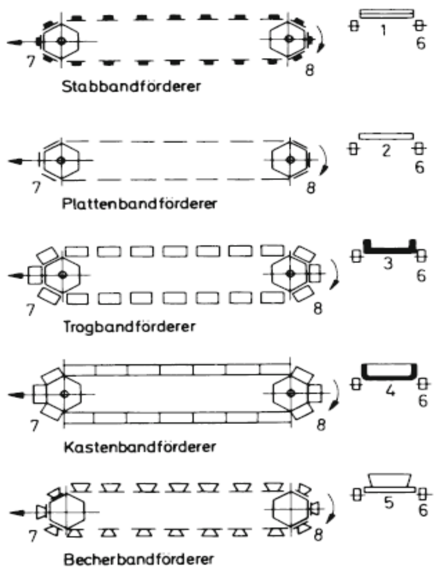


**Figure 1:Different attachment styles on chain conveyors [RÖM15]**

The main structure of chain conveyor systems is very similar to the main structure of belt conveyors.
They also consist of a drive pulley and a tail pulley where chain lanes are operating. Characteristic for chain pulleys is the possibility for a very compact design (compared to belt pulleys) which is the result of the form fit of the chain. This design ensures that a slip between chain and pulley is not possible
The major advantage of chain conveyors is the possibility of transporting the bulk over paths with very high pitches (up to 90°, depending on the design of the upper elements). This feature allows these constructions to manipulate bulk over pitches with more or less any movement of the material in the conveyor, ensuring constant and gentle transportation.

The most significant disadvantage of chain conveyors compared to belt conveyor is transportation velocity. Even with modern set-ups the transportation velocity is lower compared to belt conveyors. One reason for that is the so called “polygon effect” of running chains. (Vgl.[RÖM15])

Because of the chain design, it is not possible to loop the pulley in circle shape. The result is a loop with polygons that has the effect of an oscillating velocity and chain forces. This effect is very important for design calculations of chain conveyors and is therefore discussed in more detail in chapter 5.2.6.

A rough overview of technical data:

- Transportation capacity: up to 1000 t/h
- Distance from pulley to pulley: up to 400m
- Transportation velocity: 0,1…..1m/s (more is possible but not economical because of wear out of the chain)
  [RÖM15]

This thesis focuses on apron conveyors in the building style of the company Beumer. The basic structure of this conveyor is explained in 3.3.

## 3.2 Chain variants

Chains consist of a high number of links connected with a revolute joint. In general chains, can only transmit tractive forces. Special designs such as rigid backed chains can also transmit compressive forces up to specific level.

Depending on the field of application, the designs of the links and the chain bolts is varied (see figure 2).

Compared to ropes chains have the following advantages and disadvantages:

+ Smaller deflacting radii and from that smaller running moments
+ Less corrosive sensitive because of the smaller surface
+ Easier to repair and to maintain, links are exchangeable
+ High variety of link and bolt designs

– Lower grade of safety (the chain is only as strong as the weakest link)
– Higher weight per length
– Lower transport velocity because of higher level of wear out
– Higher dynamic forces evoked by oscillations
– Lower elasticity

(Vgl.[RÖM15])

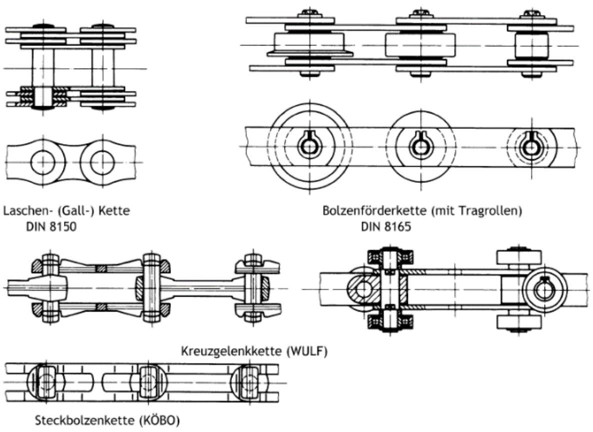


**Figure 2:Variants of chains [RÖM15]**

As clearly demonstrated, the disadvantages include very important parameters in view of safety and transportation. These are acutaully reasons to choose ropes (or belts) as traction elements because of better performances in specific transportation problems.

There are still a lot of applications where the overall performance of chains is much higher than of rope/belts.

If special attachments for carrying goods, high pitches in the transportation path and special environmental influences (e.g. heat) are required, chains will always have their place in the world of transportation systems as traction elements.

To reduce oscillations and sagging of the chain, a special design of the chain links is used to block the rotation of the links to one side. Chains with this design are called „rigid backed chains“.

**Figure 3: Transportation of glowing bulk (www.beumer.at)**

## 3.3 Basic structure of an apron conveyor

As mentioned before, the focus of this thesis lies on apron conveyors of the building style from Beumer. Figure 4 shows the most important parts of one of these apron conveyors.

The traction element is a double lined chain where every fourth link is realised as a rigid backed link. This design reduces the sag of the conveyor between the guiding rollers. This is important for reducing the possible radial movement of the chains (and cells) and further on to minimize the amplitude of the occurring oscillations. The general design of the chain is based on the standard steel bushed roller chain, and follows the Din Standard DIN8165/ DIN8167 (depending on the system variation).

The cell is directly attached to the chains and has the same pitch (so the distance from cell to cell) than the chain itself. It consists of overlapping laterally positioned side plates and a ground plate which is bet in a wavy shape to stop eventual movement of the bulk during transportation within an angular path. All this plates are welded together to one component.

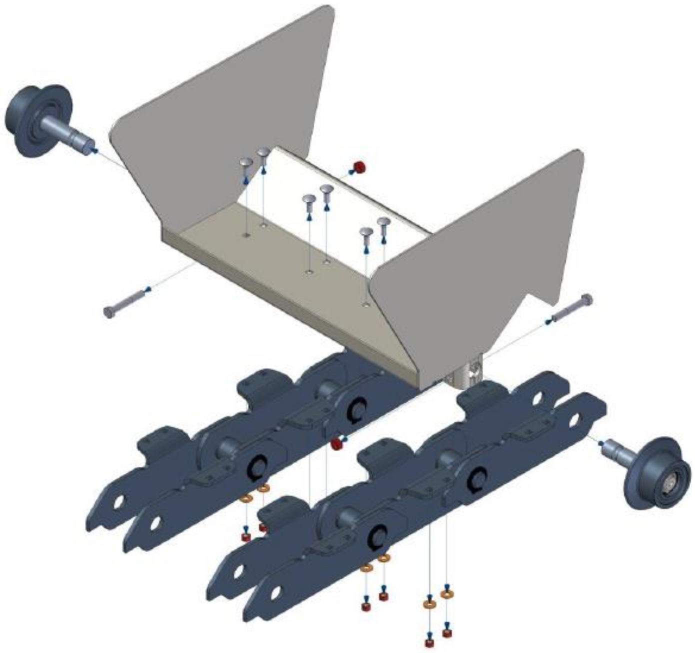

**Figure 4: Explosion view of an apron conveyor (www.beumer.at)**

The guiding rollers are mounted on every fifth chain link directly on the cell whereby the pitch of the guiding rollers can verify depending on overall weight of the conveyor. The rollers are assembled with sealed ball bearings and run on guiding tracks that are positioned sideways on the slack as well on the tight side of the conveyor (see Figure 5).

The drive of the whole conveyor is placed on the end, so at the out-feed station, which is generally located on a higher altitude. A detailed description of the drive unit as well the arrangement of the single components can be seen in 4.2.3.

On the other side of the conveyor, so on the feeding station, the preloading (or outfeed) device is located.

It is important to have a minimum load on the chains to ensure that the rollers do not leave the guide tracks and again to work against oscillations. Preload is mostly realized by concrete masses that works on the shaft of the sprockets in the area of the loading station. The whole system is based on kind of a sledge to ensure a movement (when necessary) in the transportation direction. This is also important for unplanned cases where the chain force rises over a calculated maximum (e.g. crashes).

Figure 6 shows a standard feeding station. Depending on the bulk which gets transported additional devices such as hoods for vacuuming dust can be installed. On the out-feed station devices like skimmers or strippers can be installed. Compared to hoods, these types of devices effect the chain force and have to be considered separately (see5.2.7).

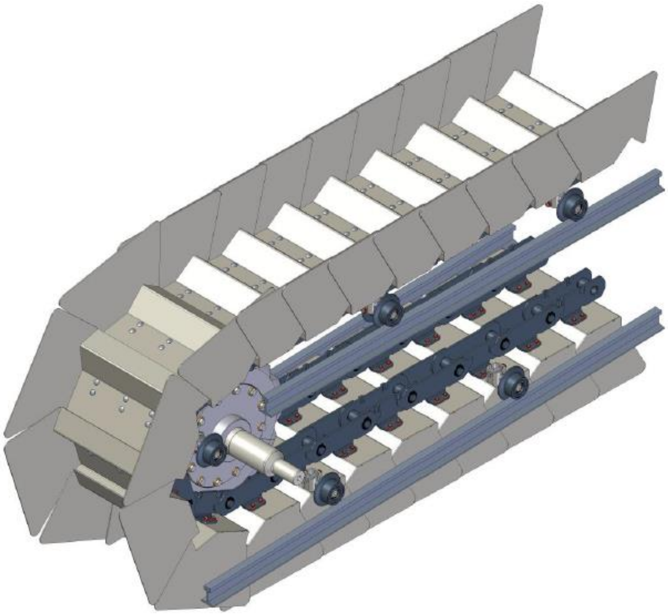

**Figure 5: Basic structure of an apron conveyor from Beumer (www.beumer.at)**

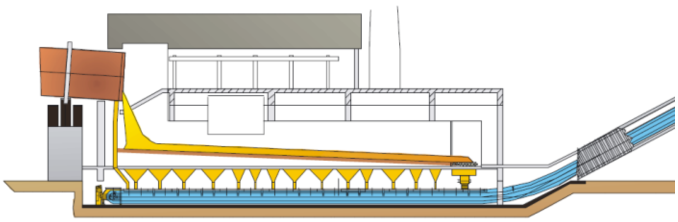

**Figure 6: Sketch of a standard loading station (Beumer catalogue)**

# 4 System variations

Nearly every application of bulk material transportation requires a new set of geometrical and technological (also indirectly economical) measurements to meet the customer's needs. This set of "initial" parameters is more or less fixed in the problem statement, which forces the developers to consider them in their design calculation.

In this thesis, the calculation of the chain force is focused, so each parameter that asserts an influence on this calculation, and could possibly be changed during the design process, has to be scrutinized in detail to find the grade of its impact on the analytical calculation and also on the multi-body simulation (see 4.3).

The parameters are often connected with each other, which makes it sometimes difficult to see their individual impact on the calculation of the chain force.

An example of this is the geometrical parameter "width" of the conveyor. It has undoubtedly an impact on the chain force as its raise the mass of the construction and also the mass flow, but it does not affect the model in the multi-body simulation. Due to the fact that just symmetrical and plain conveyors are consider in the simulations (so feeding and de-feeding station are in one line), the width can be taken in account as a simple enlargement of the cell masses. Connections like these are very important to know as they can be explained and also considered via other parameters.

This has huge influence on the number of multi-body models and furthermore on the experimental design and the corrective factors of the analytical calculation.

## 4.1 Geometrical Parameters

In this chapter the main geometrical parameters are shown and explained. Their influence on the calculation or the simulation is very often connected to a technological parameter and is not always directly recognizable.

In General, there are three different overall layouts that are mainly used by Beumer. The different system Variations are shown below and for the analytic calculation relevant parameters are described.
Further on the influence of the parameters on the chain force is roughly discussed.

### 4.1.1 System I: Horizontal transportation

Figure 7 shows a sketch of a system with a horizontal transportation path. It is the easiest and most simple system variation. This variation is  not often in use as the disadvantages from the chainconveyor in a system like this are compared to traditional belt conveyors to significant.
However, its still important to consider this system variation as it is the „basement“ of all the other systems.

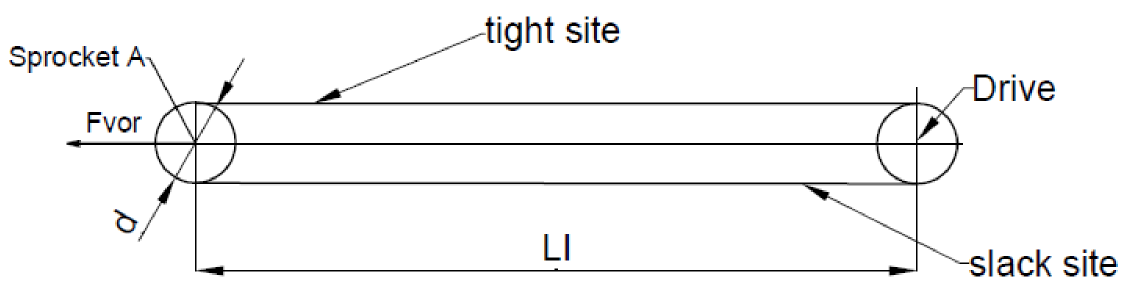


**Figure 7: Sketch of System I**

### 4.1.2 System II: Horizontal + pitch transportation

System II is broadly in use for applications where a difference in altitude is within the loading and unloading point. Examples of the application of this system is the loading process of silos or furnaces.

In comparison to system I, an angular part with a radius is added, This has the effect that pitch (or also called climbing) resistance appears. These resistances have compared to the resistances through friction a much stronger influence on the chain force. Depending on the parameters it can be up to 90%.

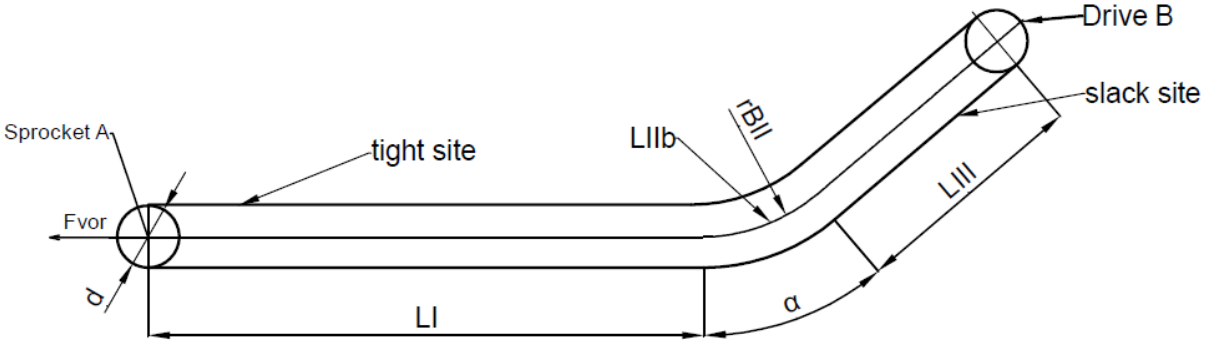


**Figure 8: Sketch of System II**

### 4.1.3 System III: Horizontal + pitch+ horizontal transportation

This variation is actually a combination between the systems I and II. In this thesis, the radius $r_{IV}$ and the heighten plane length $L_V$ are handled like a mirrored version of the radius $r_{II}$ and the plane length $L_I$ because of the similarity of both sections.

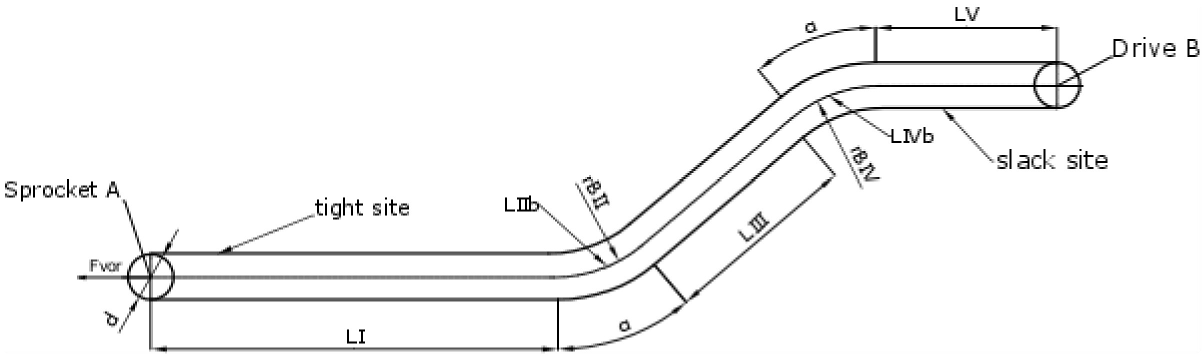


**Figure 9: Sketch of System III**

## 4.2 Technological parameters

The technological parameters are, as described in Chapter 4, not always directly choose able by the customer. They are mainly derived from necessity rather than out of the necessary technical devices or the system itself to fulfil the requirements in terms of a technical point of view. An example is the drive train, which results from the influence of the whole system, and under normal conditions is not directly chosen by the customer.

### 4.2.1 Mass flow of bulk material

The mass flow of the construction is one parameter that plays a major role for the customer and is therefore very important. It is dependent on the transportation velocity and the bulk volume per cell.
Furthermore, the cell volume depends on the cell width and height (the cell length is given through the chain pitch). The determination of the best combination of these parameters is crucial as a wrong set of the mass flow results in a system requires lot of power. This can lead quickly to a transport system that is no longer economic anymore.

### 4.2.2 Friction coefficients

Friction, no matter what kind, is one the strongest effects that influences the chain force. For this reason, these parameters (friction coefficients) are discussed briefly in the following paragraphs.

As previously explained, the apron conveyors of Beumer transport mostly dirty or dusty bulk. After a longer running period of the conveyor, a significant amount of dirt can be observed on all components of the system. This has an effect on the value of the resistance through friction which makes it necessary to consider this kind of soiling more in detail.

There are a lot of different norms and papers expressing the different friction coefficients for nearly all environmental conditions and material combinations. It is common to find a large discrepancy of the friction coefficients in this source. This could be due to the fact that when measuring these coefficients, different types of pollution were used as (dry dirt leads to a different coefficient than wet dirt and so on).

This makes it possible to always find a coefficient that suits well into the calculation and which makes the results look “good”.
Especially for the validation of the analytic calculation with the multibody simulation the gaps between these sources can be misused to “manipulate” the results with the help of the friction coefficient and so to close the gap between both results.
In this case, the friction coefficient is used as a kind of corrective factor which should definitely be avoided.

For this reason, a separate validation of the friction coefficient was done to develop an acceptable range for the friction coefficients. This reduces the risk of misuse and so the emerging failure (see 9.2)

### 4.2.3 Resulting parameters through drive train

In this chapter, parameters are discussed that do not have a direct influence on the analytic calculation of the chain force within the transient condition, but definitely have to be considered in different corrective factors.
The most significant one is the corrective factor for the run up process. This is the only factor where the impact of the whole drive train and its behaviour was considered. In general, it does not influence the chain force at a transient state.
This corrective factor is only with a lot of effort analytic calculable which is the main reason that it was simulated with Simulation X(see 7.2)

### 4.2.4 Parameters through additional devices

This group of parameters considers the impact of devices that have an impact on the chain force from outside. That basically means that these devices produce a local working resistance that effects the chain force. An example for this are brushes or strippers.

## 4.3 Definition of possible parameter combination and ranges

To ensure reproducibility and the required level of accurateness, it is necessary every parameter to set a value range. As described in the preceding chapters, the value of the chain force can strongly vary by the change of a parameter which is the reason why a defined range for the parameters must be set. The whole analytical calculation is just valid for system arrangement within this ranges. The company Beumer defined the range of the parameters and their combination in this instance.

Table 1 shows a summary of all the significant parameters which are necessary for an analytical calculation of conveyor systems. They are basically composed of the geometrical and technological parameters, the characteristics of the chain and the guiding rollers and the friction coefficient. Some of the parameters can be merged to get a simplification and lowering the number of parameters which is especially important when it comes to t multi-body simulations.

An example for this is the bright orange marked parameters “chain/cell data”. This group of parameters can be simplified to an arising of the transportation mass as they only raise the mass per transportation-path-length-unit. This allows that within the MBS parameters, such as the cell width and cell height, not having to be considered separately. So, only models with one cell width/length have to be modelled which is a dramatically reduction of necessary MBS models.

Another simplification regarding the analytic calculation is negligence of the run up process. Only the transient state is analysed in the analytical calculation within this thesis which makes it possible to set the transportation speed to a fixed value. The run up process itself was considered in a separated simulation (see chapter 7). Its impact is considered as a corrective factor in the analytical calculation. If the energy consumption or an analysis of the energy efficiency would be required, the velocity has for sure to be considered.

**Table 1: Overview of parameter ranges**

| variables | symbol | range of values | unit |
|---|---|---|---|
| **geometric data** | | | |
| total transportation lengt | $L_{total}$ | 50...300 | [m] |
| plane length | $L_I$ | 10...120 | [m] |
| length angular path | $L_{III}$ | 20...150 | [m] |
| heightens plane length LV | $L_V$ | 10...100 | [m] |
| radius II | $R_{II}$ | 5000...15000 | [mm] |
| radius IV | $R_{IV}$ | 5000...15000 | [mm] |
| angle of pitch | $\alpha$ | 20...40 | [°] |
| **technological data** | | | |
| transportation speed | v | 0...4 | [m/s] |
| filling level cell | f | 0...100 | [%] |
| density bulk | $\rho$ | 1,2...1,6 | [Kg/L] |
| **chain/cell data** | | | |
| chain type | | A,B,C | [] |
| chain pitch | p | 315;410 | [mm] |
| cell length | $C_L$ | 500...2000 | [mm] |
| cell width | $C_W$ | 300;400 | [mm] |
| **guiding roller data** | | | |
| diameter guding roller | d | 120;160 | [mm] |
| pitch guiding roller | $p_{GR}$ | 945;1260;1575 | [mm] |
| **friction coefficients** | | | |
| guiding rollers/rails | $\mu_{GR}$ | 0,001...0,01 | [] |
| chain bolt | $\mu_{CB}$ | 0,1...0,25 | [] |

# 5 Analytical calculation of the chain forces

In the last preceding paragraphs, the similarity of chain and belt conveyors was discussed in important points. In general, the physical behaviour of both, chain and belt, are regarding the analytic design calculation, the same. They both follow the same mechanical rules. There are unquestionably differences, especially in respect of the drive units and out of it the resulting oscillation (polygon effect), but for an analytical design calculation of the chain forces in a transient state the norms and approaches for belt conveyors are also valid for chain conveyors. Especially for calculating the different resistances during a transient state, the failure equals almost zero.

For this reason, the analytic design calculation follows the norm DIN 22101.

The resistances that appear in chains, for example bolt friction between the links, have to be considered separately to evaluate if they have a significant influence on the chain force or not. Not all resistances can be determined within an acceptable effort in the analytical calculation. This would raise the complexity tremendously and a guarantee that the result is more exact is also not ensured.

It is very important that no physical effects are forgotten to get a valid result with the requested exactness for a big variety of system designs.
The different effects should as be declared in the analytic calculation without any dependencies on each other. This is very important at the point of evaluation of the calculation itself, when the results of the multi-body simulation and the analytical calculation are compared. Failures can be easily determined by turning off different physical effects in both systems, the analytic and the simulation.

The triangle, analytic calculation (or also the mathematical model), multibody simulation and existing system have to be coherent at any time.

To achieve this, the analytic calculation should be as simple as possible in the beginning. If the differences of the results are too big, resistances which were not considered yet will get into the focus. Together with the determination of suitable corrective factors for specific system variants the results of the chain forces will converge to the measured data and the multi-body simulation. This requires a number iteration steps to fulfil this task.

As shown in earlier chapter, the whole analytic calculation is for an apron conveyor of the company Beumer. This also means that resistances which are resulting from the design of the apron conveyor itself have to be considered and discussed (e.g. chapter 5.2.3).

## 5.1 Segmentation of the system

To gain a flexible analytic calculation concerning a high quantity of different variants and initial parameters, it is important to split the conveyor into sections. That allows the addition or subtraction of single sections in cases where the path design varies, for example from System II (Figure 8), the angular part was then subtracted to get System I (Figure 7)and so on.

Another important reason for the segmentation is that it is much easier to calculate the chain force on a specific spot. This can also be useful within the validation with the multibody simulation by comparing the chain force values on different areas.

As a basic system, the design called System III (see 4.1.3) is used. The sections are districted in geometric significant areas. Each section is again divided into slack and tight site of the conveyor.

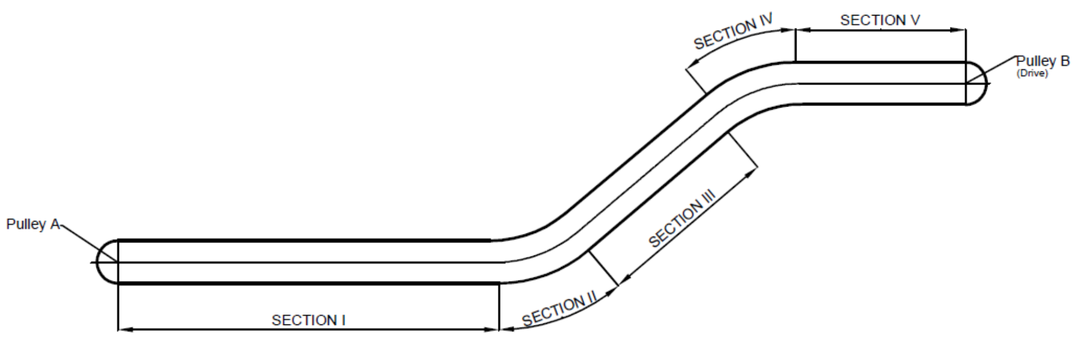


Figure 10: Overview of the segmentation

## 5.2 Evaluation of the relevant resistances

The analytic calculation of the chain force will only consider the relevant resistances. An evaluation if a specific resistance is significant or not is shown in the following chapters.

### 5.2.1 Friction between track rollers and guide rails

To reduce loses from friction resulting of the contact with ground and chain, and also for a smoother running of the whole system, the conveyor is guided with rollers which are a running on sidewise mounted rails. The distance between the rollers is in most cases constant and equals a multiple value of the chain pitch (mostly five).

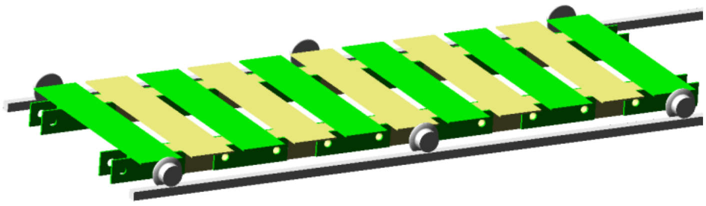

**Figure 11: Principle of guiding on a simple model**

The calculation of the rolling friction is strongly simplified on the usual standard formula (5.1) which is the resulting normal force multiplied by the rolling friction coefficient. Many studies have shown that this formula describes the effect of rolling friction very well which is in this case sufficient.

$$F_R = F_N . \mu \tag{5.1}$$

$$F_{R_{ges}} = \sum_{i=1}^{n} F_{N\_i} . \mu \tag{5.2}$$

The value of this resistance is strongly dependent on the length, the self-weight and the path (e.g. pitches) of the conveyor. The higher these parameters are, the higher the roller friction will be.

The assumption that the mass of the whole system will always be high is reasonable, due to the requirements and the initial conditions of the customer.
That implies that the roller friction is an important physical effect that has to be considered in the analytic calculation.

### 5.2.2 Friction in between chain bolt and chain links

Each radial movement between the chain links causes friction that appears as a resistance and therefore an enhancement of the chain force.
Because of oscillations caused by drive units and the polygon effect of the chain, a lot of tiny movements are conducted in the system. Rigid backed chains and also the guiding rollers are working against this movement, however thy still appear.
It is not possible to consider this occurrence in detail in the analytical calculation with an acceptable effort and a significant result.
The calculation of the chain bolt friction in the analytic will only take part in areas where the deflections between the chain links are significant. This occurs for example in the sprockets and also in the radii between the plane tracks and the pitch.

It can be assumed that in systems with long transportation paths and high pitches, the chain force will reach a value where the chain bolt friction will have a significant impact (even with very small friction coefficients) compared to other resistances.

The value of this friction strongly depends on the occurring chain force in the specific area, a friction coefficient and the diameter of the chain bolt. The chain bolt friction increases the chain force, that means that the chain bolt friction depends on its own value, which makes it necessary to iterate the result of it.
In the analytical calculation, this iteration is due to possible issues within the calculation program avoided. To minimize the rising failure, the average force from the chain force before and after the position where the chain bolt friction works, was used to calculate the chain bolt friction.

It is obvious that the result of this resistance will not depict the "real" system. This mainly results in the simplifications described in preceding paragraphs, but it is necessary to consider this effect in the analytical calculation.

### 5.2.3 Friction in between the cell plates

This friction only appears in apron conveyors. It is the result of the movement between the overlapping cell plates which are carrying the bulk and is only present in areas where the cells are moving relatively to each other (sprockets and radii).

As visible in Figure 12, the only load that works on the cell plates is the weight force of the bulk in each cell. This is true for plates mounted on the sides, the resulting normal force of the bulk is reaching a value where the impact on the overall chain force is almost zero.

On already existing systems, it was recognisable that a physical contact between the cell plates is not always guaranteed which is the result of the high tolerances during the assembly and that the bulk gets between the cells. So, it is not possible to ensure that the friction between the plates works in a predictable way or if it even exist.
This assumption is also be made for the ground plates of the cell.

Compared to the other forces and resistances of the system, the impact of cell friction is clearly negligible. Even with a "perfect" system, consisting of no gaps between the plates and continuous physical contact, the applied normal forces would be so small that the resulting friction forces would not influence the overall chain forces in a significant way.

**Figure 12:Loaded apron conveyor (www.beumer.at)**

### 5.2.4 Bearing Friction

All mechanical system bearings (especially roller bearings) play an important role regarding friction, efficiency and further on the economic viability. In the apron conveyors of Beumer bearings are also used. A standard sealed ball bearing is assembled in the guiding rollers.

It is assumable that under normal conditions the bearings are hermetically tight to a specific grade against the environment and are also lubricated within from the manufacturer recommended time periods. This makes it possible to take the assumption that the roller bearings will always have the same (so a constant) friction coefficient. Again, this does not depict reality, but if the maintenance of the conveyor is done within the recommended time periods it is a valid assumption.

Even if resistances through the roller bearings is considered, the impact on the chain force would be minimal (under 1% depending on the system). This is due to a very low friction coefficient (0,0005 to 0,001) and the relatively low load on the bearings.

With these assumptions and the assumption that the normal force on the bearings is the same as the one on guide rollers, we can simply increase the friction coefficient of the guiding rollers or even completely ignore this effect and assume that the failure is compensated by a corrective factor.

### 5.2.5 Climbing resistance

Compared to the other kinds of resistances which were described yet, the climbing resistance has the highest impact on the value of the chain force. Depending on the system variation, the share of it can rise up to 90% of the maximum chain force. That is the reason why this resistance is calculated much more accurately than the others, which is mainly visible in the fact that no assumptions that could make the analytical caluculation easier were done. This minimizes the risk of a lack in accuracy of the calculation.

Reasons for the high impact of the climbing resistance are on one hand the great angle of the angular part of the transportation path (up to 40°) and on the other hand the high mass per cell (>100kg loaded).
The fact that this combination leads to a high value of this resistance is also easily seen by the standard formula (5.4).

$$F_{ST} = m \cdot g \cdot \sin(\alpha) \tag{5.4}$$

A distinct feature of the climbing resistance is, that it has to be observed in detail on every part of the conveyor. For example, the climbing resistance of the slack site of the conveyor is actually not a resistance from the point of view of the drive momentum as it is working in transportation direction. So, the value of the climbing resistance in this part of the conveyor reduces the drive momentum and therefore the necessary power.

But from the point of view of the chain force the climbing resistance of the slack site does increase the value of the overall chain force. For this reason, it is handled like the other resistances.

### 5.2.6 Resistances through oscillations

As already mentioned in previous chapters, oscillations have an impact on nearly every kind of the explained resistance. A separate aspect about the occurring oscillation of chain conveyors is the already mentioned polygon effect. It mainly results through the wrapping of the chain around the sprocket in the shape of a polygon. This has the effect that oscillations of the speed and the acceleration in the chain occurs. (Vgl.[WMJ09])

These oscillations have the effect that all the occurring forces on the conveyor start to oscillate with the frequency conducted through the polygon effect into the system.
Due to the quite low speed of the sprockets (4-7 rpm), the great chain pitch (315mm) and the pitch circle diameter of the sprocket (876mm) the amplitudes of the oscillation through the polygon effect can be neglected in the analytical calculation.

The occurring failure compensated through all the other corrective factors. This assumption can be done because of the regularity of the oscillations.

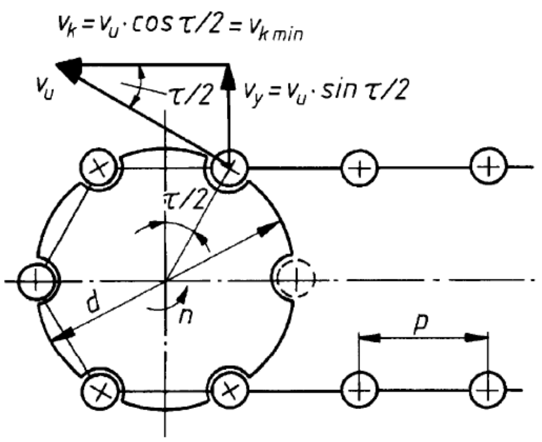


**Figure 13: Sketch of the polygon effect [WMJ09]**

### 5.2.7 Resistances through optional devices

This group of resistances is strongly dependent on the type and category that the optional device belongs to. It most cases it is chosen by the customer and is specially designed for the specific task or bulk.

This makes it necessary to calculate the emerging resistance separately due to the large variety of different optional devices.

An example of an optional device is the feeding station (Figure 14). Normally each conveyor needs a type of a feeding and de-feeding station, however also these devices can strongly vary from task to task which is the reason why also the feeding station is not considered within the chain force calculation.

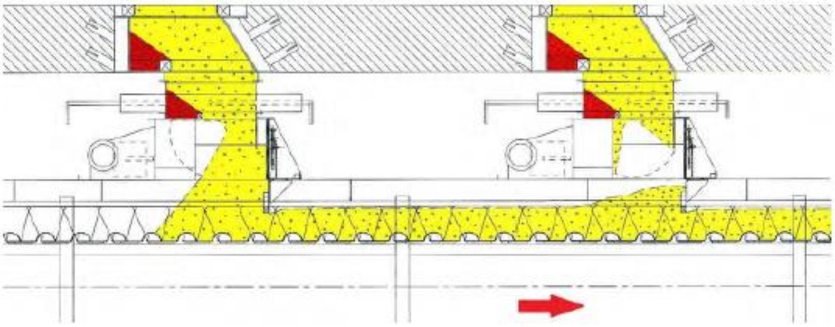

**Figure 14: Sketch of a silo pull off (www.beumer.at)**

## 5.3 Calculation of the chain forces

## 5.4 Corrective Factors

To get an analytic calculation approach which calculates the chain force within an acceptable accurateness, it is absolutely necessary to introduce corrective factors to the calculation to make the approach flexible against changes of the initial variables and parameters (especially the geometric measures).

As described in 5.2, each resistance which is considered in the analytic calculation is simplified to a specific grade. Without these simplifications, an analytic calculation would not be easy to handle and for sure not economic for the user anymore because of the increasing complexity. This is another signal that a deviation of the chain force between calculation and reality will occur without using any kind of corrective factors.

The corrective factors have to be flexible, which means that they have to change when the most significant variables and initial parameters are changed. This mainly results through the fact that the failure for a shorter system is much lower than that of a long systems. In longer conveyors the influence of the not considered effects in the analytic calculation are higher which is also the reason that the gap between calculation and simulation is raising by raising the initial parameter. So, the combination of the most significant initial parameters has a big influence on the corrective factors itself.

**Table 2: Overview of corrective factor**

| Symbol | Correction of |
|---|---|
| $C_{LI}$ | plane length LI |
| $C_{RII}$ | radius RII |
| $C_{LIII}$ | angular lenght LIII |
| $C_{RIV}$ | radius RIV |
| $C_{LV}$ | heightens plane length LV |
| $C_{\alpha}$ | angle of pitch |
| $C_{RP}$ | Run up |

For this reason, the influence of specific parameter combinations have to be observed in detail. The evaluation and weighting of the role of the parameters is described in chapter 8.6.
The output of this evaluation was the impact of one parameter on the chain force, therefore for each parameter that has a strong effect on the chain a corrective factor is introduced (See Table 2).

Due to the fact that these corrective factors consist of confidential data from Beumer just an overview of the emerging factors is given in this thesis. Numbers or graphs that are resulting of the validation will not be shown in detail. For giving the reader still a possibility to understand roughly the developed approach for the determination of the corrective factors, chapter 9.3 was introduced. There the systematicall approach for the determination of the corrective factor „$C_{RII}$“ is explained.

# 6 Multibody Simulation of chain conveyors

## 6.1 Introduction to Multibody Simulation (MBS)/ MSC Adams

A multibody simulation (MBS) system is a system that consists of solid bodies, or links, that are connected to each other by joints that restrict their relative motion. The study of MBS is the analysis of how mechanism systems move under the influence of forces, also known as forward dynamics. A study of the inverse problem, i.e. what forces are necessary to make the mechanical system move in a specific manner is known as inverse dynamics.

Motion analysis is important because product design frequently requires an understanding of how multiple moving parts interact with each other and their environment. From automobiles and aircraft to washing machines and assembly lines - moving parts generate loads that are often difficult to predict. Complex mechanical assemblies present design challenges that require a dynamic system-level analysis to be met. (Vgl. [MSC17])

MSC Adams is one of the most successful tools for realizing multibody simulation from the company MSC Software.

MSC Software is used for many types of motion analysis:

- Rigid and flexible multibody systems
- Sensitivity analysis
- Vibration analysis
- Coupled control/mechanical system analysis
- Kinematics and kinetics
- Contact and friction
- Loads and displacement
- Durability and life-cycle analysis
- Fracture or fatigue calculations
- Kinetic, static, and dissipative energy distribution
- Control system analysis

(Vgl. [MSC17])

This high grade of flexibility makes this tool to the perfect choice to simulate the different conveyor systems

## 6.2 Emerging problems by simulating chain conveyors

In general, MBS models (especially assemblies) need a lot of computational power and time to calculate a reasonable result. This mainly comes from the complex differential equations that need to be computed in each time step. For each object within the simulation (components, joints etc.) there has to be a specific number of differential equations computed numerical to simulate the dynamic behaviour of the whole system. This happens for every time step, which leads to a huge increase of the number of equations again. Even computers that have a lot of computing power can also have difficulties in solving these problem statements, and when they are able to do this, it can take several hours (or even days) to acquire a result.

Due to the fact that the conveyor system that has to be computed can have a transportation length of up to 300m, the number of components can be massive. Table 3 shows a rough calculation of the necessary components without any simplifications.

| | number | number of parts |
|---|---|---|
| chain segments | 3810 | 22860 |
| cells | 1905 | 7620 |
| guiding rollers | 382 | 1146 |
| guiding tracks | 6 | 6 |
| sprockets | 2 | 36 |
| **sum** | **6105** | **31668** |

**Table 3:rough calculation of components of a 300m conveyor**

Depending on the design of the conveyor, the components are connected to each other with a minimum of one joint and a variable number of constrains. A maximum is not describable as it is dependent on the dynamic condition and movements of the whole system.
In this case, it is assumable that each cell includes two revolute joints (in the chain links), one linear joint (to restrict an angular movement to transportation direction) and one planar constrain to keep the conveyor in between the centre of the guiding tracks.

Each guiding roller includes one revolute joint in the centre of mass, a linear joint to keep it in a defined positon against the cell and one 3D contact between roller and track.

Nearly all kinds of the available joints in MSC Adams have the possibility to consider friction in the connection. This has always the effect that the computational effort is raised again. Especially in the so called “3D contacts”, so contacts where bodies have contact with each other without geometric constrains (e.g. sprockets and chain bolts, guiding rollers and track), the computational time raises dramatically.

From these numbers and facts, the massive computational effort is easily replicable, and especially that systems in this size are not computable with the available resources and within a suitable time range.

Figure 15 shows the result of a simulation of a small conveyor (20m). Each line describes a Vector (most times a force). This figure makes the massive computational effort of simulations of this kind graphically visible.

For that reasons, it is necessary to simplify the whole system without losing any physical effects that occurs in the “real” system (see 6.2).

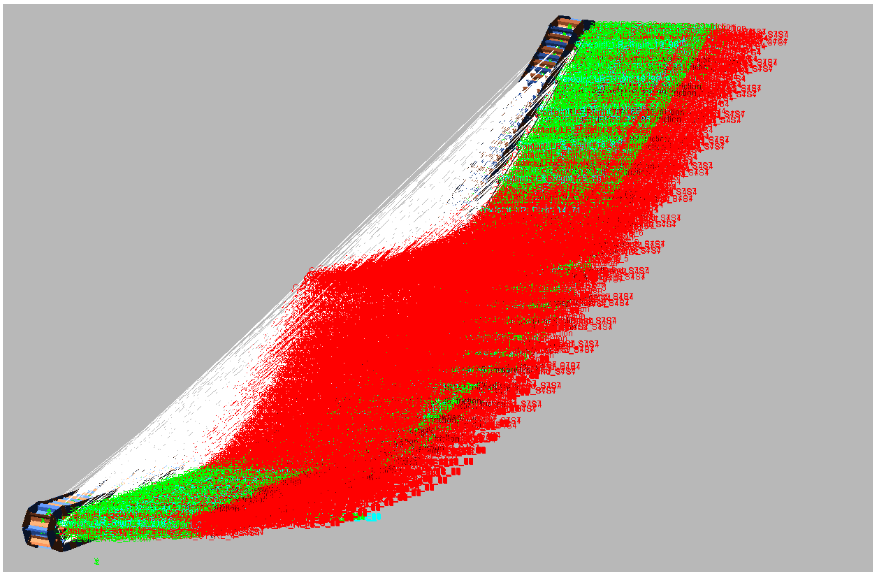

**Figure 15: Example for the computational effort of a MBS Simulation on a small conveyor**

## 6.3 The role of substitutional models

For the validation of the analytic calculation a minimum of MBS models are necessary to guarantee a constant accurate result of the chain force (see 8.6). Due to this, the structure of the MBS models has to be as simple as possible to allow a fast computing and also an easy approach for building up the models in MSC Adams when the system variables change (especially the geometric variables).

This is mainly done by introducing substitutional models of single joints, forces and even whole sections of the conveyor to describe the physical behaviour of the system with a minimal computational effort.

In general, there are three different kinds of substitutional models:

- Substitutional model for physical effects (e.g. chain bolt friction)
- Substitutional models for optimizing the simulation (e.g. run up models, feeding-de-feeding models ect.)
- Substitutional models for parts of the system (models with spring damper systems)

The risk of this method is, that dynamic behaviour of the Multibody simulation does not depict the "reality" anymore, and so failures occure. In particular for the oscillations and the natural frequencies of the conveyor the possibility of failures have to be determined in detail.

The final MBS models consist of big variety of different substitutional models. In this thesis, only the most significant ones will be mentioned in the following chapters. Not all of them were successful which means that not all of them had a positive impact on the stability and quality of the simulation. The success or failure of the model is unknown before the test, meaning that it is possible that there are models that do not have the expected impact

### 6.3.1 Substitutional model for physical effects

A major point of the MBS models is to depict the physical effect friction such as in reality. The usual formula to calculate the friction seems simple to calculate (FR=my · FN) but it is actually a challenge to simulate it in MBS models. This mainly comes from the fact that simple contacts (in MBS named as "3D contact") are very complicate to calculate for the algorithms. Generally, compared to other joints, it is the connection that needs the most CPU power and generally invcreases the simulation time.

For these reasons, it is often useful to substitute 3D contacts with other models where joints are used allowing easier calculation for the numeric algorithms (e.g. revolute or planar joints).
An example for this approach was the modelling of the constraint which realizes the block of movement against one direction of the rigid backed chains. On one hand, a simple 3D contact would work fine if the assembly size does not get too big as the problem is in reality solved by a geometrical solution. But on the other hand, so many 3D Contacts were introduced to the model that the performance of the model decreased.
One of the solutions is shown in Figure 16. A torque and a standard revolute joint were introduced into the centre of the chain bolt (red arrow). The angular movement of the revolute joint was measured and returned to torque definition. Together with the stiffness and damping coefficients of the link (working around the axle of the revolute joint), the torque describes a similar behaviour than solution with a 3D contact. One problem of this solution was that the oscillational behaviour compared to the reality was not satisfying, and the system reacted relatively "nervous" when specific oscillations occurred.

The stiffness of these types of problems can be calculated analytically or simulated with FEM simulations. In comparison, that the damping coefficients are not calculatable and have to be determined through tests. In chapter 6.3.3 this process is described briefly.

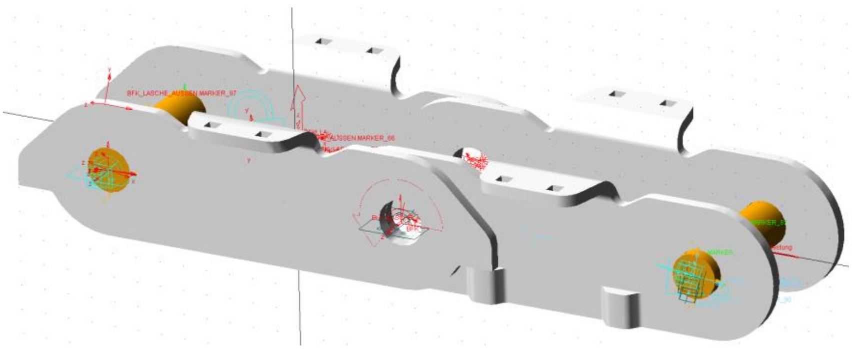

**Figure 16: Substitutional model for the behavior of rigid backed chains**

### 6.3.2 Substitutional models for optimizing the simulation

This group of substitutional models is mainly used for changes of the standard approaches for describing physical effects of the simulation program[1]. In particular for friction between two non-guided bodies, the section 6.3.1 has already mentioned standard feature “3D-Contact” that MSC Adams provides, every little detail of this effect (e.g. penetration depth of the contact etc.) is considered and calculated. Smaller models in particular require the correct and most accurate way to simulate the friction effects, but for huge models such as an apron conveyor, the 3D-contacts use too much computational power compared to the influence of resulting friction on the value of the chain force. For this reason, this very detailed simulation of the physical problem friction is not necessary and is also a less accurate approach for describing a suitable solution of the simulation.

An example in this approach is seen in the friction force through the guiding rollers. Figure 17 shows the concept of a substitutional model for optimizing the roller friction regarding the computational effort.

The guiding rails were replaced with splines (white) which are following the required transportation path. The guiding rollers are pictured as a simple cylinder (orange) with markers (Points) which are placed on both sides. These markers are describe the contact point between the rollers and the path.

In general, the contact between a line (in this case the white spline) and a point (the lateral positioned markers of the cylinder) is the easiest and most effective way to simulate a contact concerning the computational effort. This mainly results through the well-defined states “contact” and “no contact” between point and spline making it much easier for the numeric which runs in the background to calculate the behaviour of both elements.

[1] MSC Adams

It is possible to measure this contact force and use its value as a "run-time-variable"[2] during the simulation. With this feature, the "measured" run-time variable "Contact-Force" was multiplied by a simple value that describes the friction coefficient and is further used to introduce a force working against the transportation direction (red arrow). With this approach a resistance through friction was considered which has three to four times better performance than a "classic" approach with 3D-Contacts.

The major disadvantage is, that only simulations of the transient state can be done. The friction effects during the run up can just be considered with the help of IF-loops (e.g. IF velocity roller is zero use friction coefficient 1; ELSE use friction coefficient 2). By implementing these IF-loops, the performance becomes much worse until a point where even the 3D contacts are the better choice. As in all substitution models, a kind of compromise is necessary if the get used. In this case, only friction forces of the transient state could be considered without losing performance.

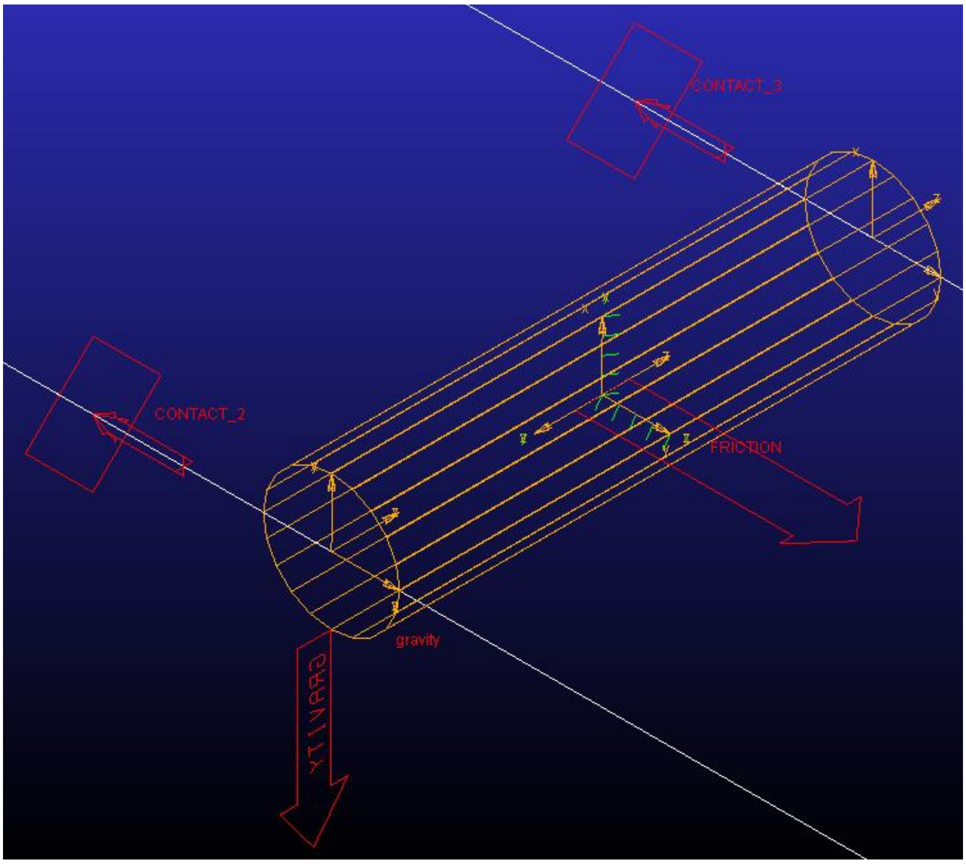


**Figure 17: Substitution model of contact Roller-Guiding Rail**

[2] Parameter which is not fixed and gets changed during the simulation

### 6.3.3 Substitutional models for parts of the system

This group of substitutional models reduces the computational effort for simulating big system the most. The idea is, to change parts or whole sections of the original design of the conveyor to a system that has much less components but still has the same properties of the original system.
Furthermore, also significant masses of some components are reduced to a point mass and joints to simple spring-damper systems for another optimisation of the computing effort.

The result is a system that looks compared to original quite abstract (see Figure 18). The difficulty of these substitutional models like these ones is to modify it in a way that all the specific characteristics which also the original model has are considered. This is not always possible and it is common that substitutional models generate inaccuracies which also have to be considered.
As in 6.2 explained, MBS of conveyors in the required length are not easy (or not even possible) to simulate which makes it necessary to take the inaccuracies in account. For sure, these inaccuracies have to be observed in detail to keep the failure in the final result as low as possible.

In the following paragraphs the development of a substitutional model for five cells (so from guiding roller to guiding roller) is explained in detail. During the whole thesis, there were several designs of substitutional models for these kinds of problems but just the one with highest grade of abstraction is shown in.

The structure is as simple as possible, the cells and chain links were summed together to one point mass which is located right in the middle between the guiding rollers. The mass is connected with spring-damper systems with the still “existing” cells. Further on, two constrains were set on the point mass, one which “holds” it in the centre of the guiding rails and one which prevents a rectangular movement to transportation direction. At this point it’s already discernible that a failure will occur because of the restriction of oscillations through the second constrain. But due to the fact that the system is that simple, the effort for validating it is small which leaded to the decision to continue with this substitutional model.

Because of the mentioned constrains which are necessary for the substitutional model, are usage in between the radii or the sprocket area is not possible. So just a substitution of sections which are positioned on a plane length is possible.

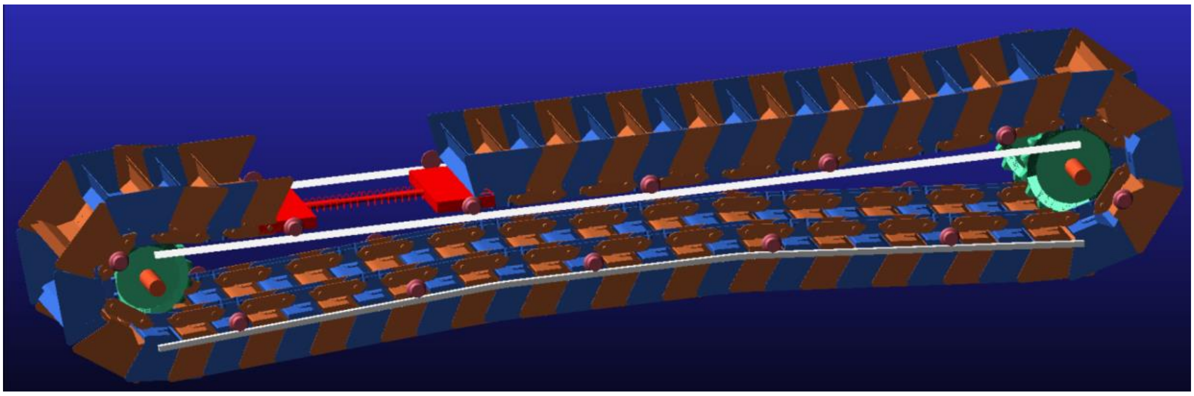

**Figure 18: Screenshot of a system with substitutions**

The main problem in creating system like the one of Figure 18 is the difficult determination of the stiffness and damping parameters of the spring damper connections.
Especially when movements which occur in the real system are restricted in the substitutional model a calculation of the parameters is just very roughly possible. The damping coefficient for example is in many cases not calculatable and has to be determined through experiments.

In this example, the stiffness coefficient of the spring was simulated with the help of an FEM simulation (see Figure 19) in the program called Ansys[3]. With the help of this simulation, the length difference which results through a constant working force could be measured and with the simple formula (6.1) the stiffness was calculated.

$$c = \Delta l / F \tag{6.1}$$

With this value, and the assumption that the cells do not affect the stiffness in a significant way, the stiffness for two cells could be calculated and afterward used in the spring damper connections.

[3] CAE Software of the company ANSYS Inc. which is specialiced on finite element analysis

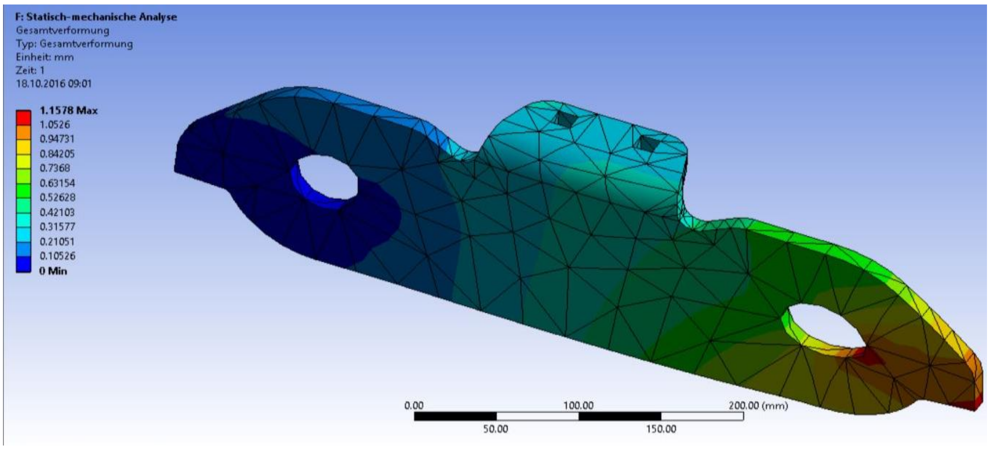


**Figure 19: Screenshot of a stress analysis of a chain link**

As mentioned before, there is no valid approach available from literature for determining the damping coefficient. This makes it necessary to determine it with the help of test or simulations.

During the thesis, the damper coefficient was changed and simulated until the result were similar and just small failures occurred. As reference model the same model as for the substitutional one was used just without simplification.
For validation, the chain force was measured in both models and compared. Further on, the eigenfrequencies of both models was calculated with the help of a FFT[4] analysis of the graph of the chain force and also compared. With this method, it is possible to check if the dynamic behaviour of both models are similar

In Figure 20 the characteristic of the chain force and a FFT analyses of it of the model without any modifications. Compared to that, Figure 21 shows the characteristics with substitutions. The figures show that the average chain force of both models is nearly equal (approx. 3% failure) and that characteristic frequencies also fit together. For sure, especially the amplitudes of the chain forces with substitutions are much higher, but due to the fact that the failure can be determined easily this results are satisfying.

[4] Fast fourier transformation, transformation from a time based graph to a frequency based one

To get this result 31 iterative steps, so changes of the value of the damping coefficient, simulation and then validation, were necessary. This method is called “change-one-at-once”. This sounds like a lot, but with this substitution the calculation time was reduced from 18 to 11 minutes which is a huge improvement.
But it is still a lot of effort to develop this kind of substitutions and it is not always guaranteed that the results can be used in a bigger scale, so for more than one segment.

This was one reason why bachelor thesis with the title was started. There the damper coefficient was not determined through trial and error but with a statistical method which is based on the principle of design of experiments. This method is reducing the number of iterative steps to 20 to 25.

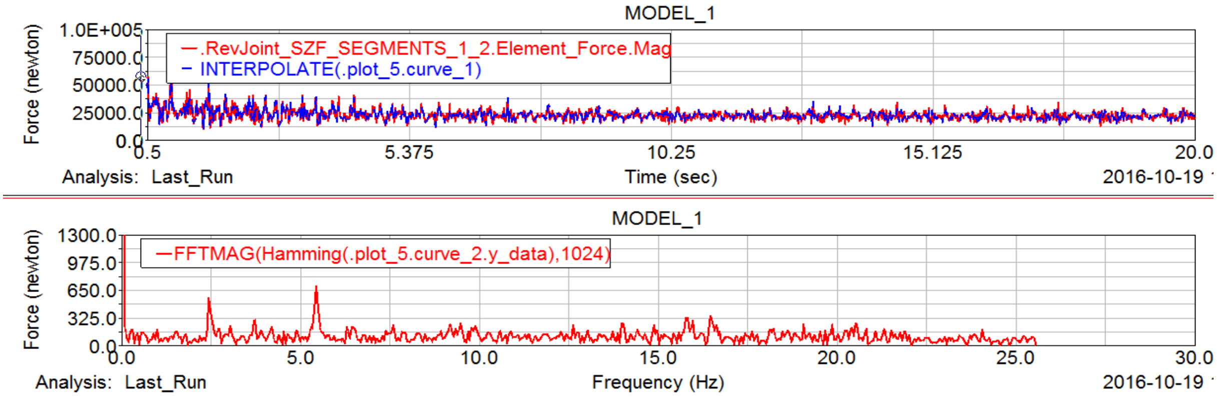


**Figure 20: Characteristic of the chain force + FFT of the original model**

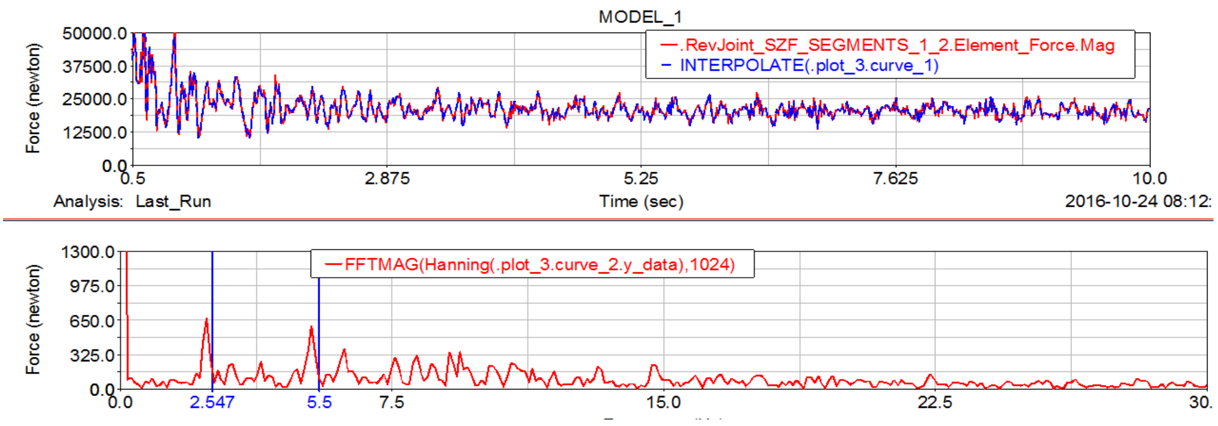


**Figure 21: Characteristic of the chain force + FFT of the model with substitutions**

# 7 System Simulations of chain conveyors

## 7.1 Introduction to SimulationX

SimulationX is CAE (Computer- Added- Engineering) Software of the company ESI ITI GmbH and is used to simulate physical and technical systems.

Simulation models are created on the basis of a discrete network approach. That means the system is broken down into logical parts which are linked through specific connections. These sub-systems are represented by preconfigured or custom model elements organized in domain-specific and custom model libraries. The sub-models are then parameterized and connected with each other. (Vgl. [ESI17])

Model libraries can be obtained as modular packages from the software producer. There are various libraries available with basic models for the corresponding physical domains as well as libraries with advanced models for specific applications and industries. Depending on the objective, the user can choose from 1D, 2D and 3D model elements with respect to the modelling, simulation and parameterization requirements. Experimental Design/ Determination of the corrective factors (Vgl. [ESI17])

These functionalities allow a quick and easy simulation of complex systems which makes it to a perfect tool to simulate specific behaviours of complex system (like an apron conveyor) within a small computational effort.
Other examples where SimulationX was used are different tasks in the automotive industry (e.g. shifting simulations of gearboxes, simulations of the drive train, analysation of occurring vibrations etc.) (Vgl. [ESI17])

For this thesis, SimulationX was used to simulate the run up process of the conveyor.

## 7.2 Simulation of the run-up-process

The run-up-process is the only state of the conveyor, where the subcomponents of the drive train are directly influencing the chain force. This effect mainly results by the inertia of the huge masses of the conveyor which have to get accelerated from stagnation.

The inertia of the drive train has actually not directly an influence on the chain force itself but rather on the run up speed of the whole system, so drive train and conveyor. The result is an increase of the chain force within the run up state which has to be considered as a corrective factor in the analytic calculation.

Run up simulations are often difficult to simulate with MBS. Especially assemblies (like the conveyors in this thesis) react very sensitive on bigger acceleration which leads to results which do not depict reality anymore.
This effects comes mainly from numeric solvers which are running in the background of the MBS tools. In cases of big acceleration, the "forecast" algorithms cannot predict a reasonable result anymore which leads very often to a crash of the simulation and further on to useless results. This is also the main reason why with MBS the whole conveyor has to be accelerated much slower than reality and just the transient state can be observed and measured within the

Further on its very difficult to simulate the behaviour of the single components of the drive train (e.g. fluid clutch, engine ect.) which leads also to inaccuracies.

This reasons made it necessary to use SimulationX for simulating the run-up-processes in a fast, flexible and economic way.

### 7.2.1 Description of the simulation

As in 7.2 described, the structure of SimulationX is based on sub components with specific characteristic which can be connected in several ways. This sub components can be chosen out of a model library. Each model signifies the characteristic of a component from the real system and has often one port for “physical” connections to other models (e.g. forces, torques, movements etc.) and ports for controlling the unit (e.g. speed controls etc.)
Out of this functions, the existing conveyor can be modelled in a strongly abstracted way but with nearly the same physical behaviour than it has in reality.

The whole simulation is structured in three segments:

- drive train
- simplified conveyor
- control loop

The drive train represents all the components which are necessary to run the conveyor, so from the drive until the sprocket. Most of the components are already existing in the model library of SimulationX and could be used directly (e.g. fluid coupling, gearbox). Others, like the drive, had to been modelled separately with standard components (e.g. momentums, masses ect.) even they are existing in the library. The reason for this was that the behaviour of the standard components like the asynchronous drive, did not follow the reality (e.g. the moment curve). This made it necessary to remodel this kind of components. Further on it is very important to consider all the inertias of the components because they have a strong influence on the run up performance of the conveyor.
All the necessary parameters for the subcomponents were available in the data sheets of the standard parts which are used from Beumer.

It is obvious that the conveyor itself can just be considered in the simulation in an abstract way So, a kind of a substitutional model was necessary to reproduce all the occurring physical effects which also appear in reality.

This substitution mainly consists of simple masses which are connected and direct linked to a resistance sub model. During the simulation, it was recognizable that the failure of the chain force out of this model is under 4% compare to the analytic calculation Detail effects within the conveyor like the polygon effect are not possible to consider in the simulation.

The mass "m_tightsite" for example contains the mass of all the elements, so cells, chains and bulk over the whole elongated length of the conveyor on the tight site (formula XXX). The same was done for the slack site just without the bulk mass. The mass "m_climbing" is actually not used as mass in the simulation but as a factor to consider the climbing resistance. It is calculated with formula (7.1) and considers the climbing resistance of the slack as well of the tight site. With the assumption that the pitch length of the slack site is the same than on the tight site formula (7.2) follows.

$$m_{slacksite} = \sum_{i=1}^{n} \frac{l_i}{p} (m_{cell} + m_{chain} + m_{bulk}) \tag{7.1}$$

$$\mathrm{m_{climb}} = \mathrm{m_{bulk}} \cdot \mathrm{l_{III}}/\mathrm{p} \cdot \cos(\alpha) \tag{7.2}$$

To consider the resistances the sub model "Resistances" was introduced. This sub model is usually used as a friction element but the algorithm of it allows to calculate the simulation Force by user defined formulas which made it possible that also other resistances like the climbing resistance could be considered with this element. formula (7.3) shows the generalized formula for this element.

$$\mathrm{F_{Resistance}} = (\mathrm{m_{tighsite}} + \mathrm{m_{slacksite}}) \cdot \mathrm{g} \cdot \mu + \mathrm{m_{climb}} \cdot \mathrm{g} \tag{7.3}$$

The inertias of the single masses were considered automatically by SimulationX which is one reason why m_climbing is not connected.

The control loop of the run up simulation is based on a standard control loop. In already existing conveyor the drive is not controlled which means that they have more or less just and on/off switch. The reason for that is that the usage of a frequency controller for asynchronous drives the overall efficiency factor gets reduced by 3-4%. This does not sound a lot but due to the high-power consumptions of conveyors in this size the running cost get raised immensely. So, caused by these economic reasons, the controlling of the drive units has to be as minimal as possible.

But in the run up simulation a control loop is still necessary to warrant a constant speed of the conveyor and to get a grade of flexibility to change the parameters of the drive (e.g. moment curve). Even so, it was important to keep the control loop as simple as possible which is the reason for the choice of a simple pT1-controller and not a PID.

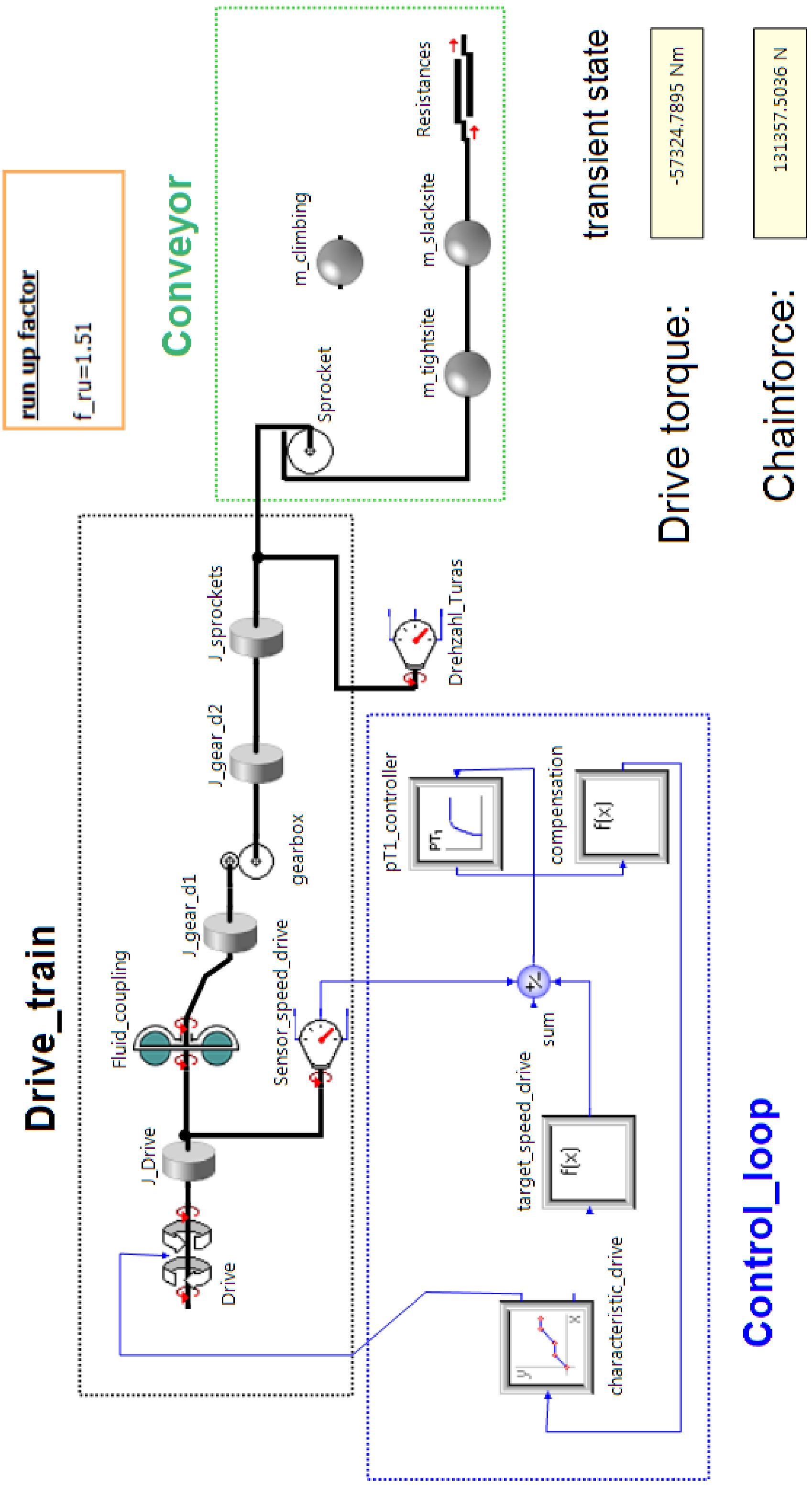


Figure 22: overview of the structure of the run up simulation out of SimulationX

### 7.2.2 Results of the run up simulation

The main result of this simulation was the corrective factor for the run up process $C_{RP}$. It considers all the additional forces which are appearing during the run up. Most of them are caused through the inertia of the conveyor masses others from the characteristic behavior of the drive train components. Figure 23: Characteristic chain force during run up shows the characteristic of the chain force during the run up. $C_{RP}$ is visible as the difference between maximum force during the run up and the working force of the transient state.

In general, $C_{RP}$ is the corrective factor with the highest value compared to all the others and is depending on the system arrangement between a range of 1,2….1,6.

Depending on constellation and adjustment of the drive train components $C_{RP}$ can strongly vary (see Figure 26: Run-up characteristic of the chain force regarding different Oil levels) which makes it necessary to simulate different set ups of the drive train to optimize the run up. An example for this is explained in the following chapter.

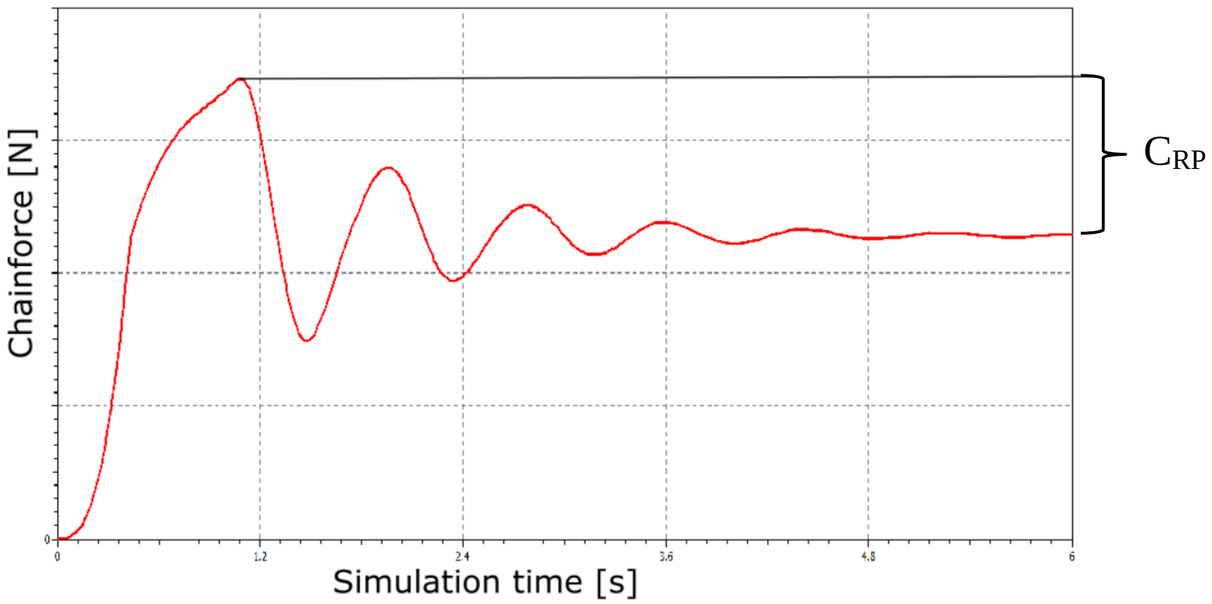


Figure 23: Characteristic chain force during run up

### 7.2.2.1 Analysis of the impact from different oil levels in the fluid coupling

As in 7.2.2 already mentioned, the simulation can be used for test with different parameters of the drive train. Because of the non-existing control of the drive during the run up it is possible that high peaks within the characteristic of the chain force appears which follows to high value of $C_{RP}$ and further on to an increase of the chain force. For this reason, a solution for avoiding this strong oscillating run up was required. The solution was a study about the oil level in the fluid coupling.

The results are visible in Figure 26 and Figure 29. A chart out of the datasheet of the fluid coupling which shows the impact of the oil level FG and the max. transmittable torque (Figure 24) was used to acquire the maximum of the transmittable torque of the used fluid coupling (see Table 4).

Already a little change of the oil level has a strong impact on the run up behavior and further on the high peaks of the chain force get “damped” to a smooth run up characteristic. Nevertheless, this simulation does not consider the impact of temperature and the changes of the flow effects in coupling. Especially for long running periods within the transient state problems like overheating ect. could possible occur.

Anyhow, the example of the different oil levels shows pretty good which kind of simulations are possible with a comparatively low computational effort (in this case just a view seconds).

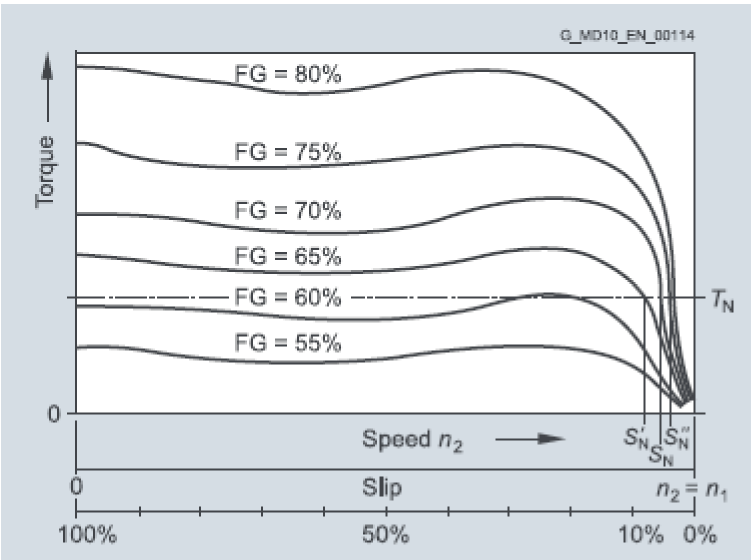


**Figure 24: Relationship between Oil level and transmittable torque of fluid couplings**

**Table 4: Impact of oil level on nominal torque**

| Oil level | reduction of nominal torque | color in chart |
|---|---|---|
| 100% | 0 | blue |
| 75% | 30% | green |
| 65% | 50% | red |
| 60% | 65% | turquoise |

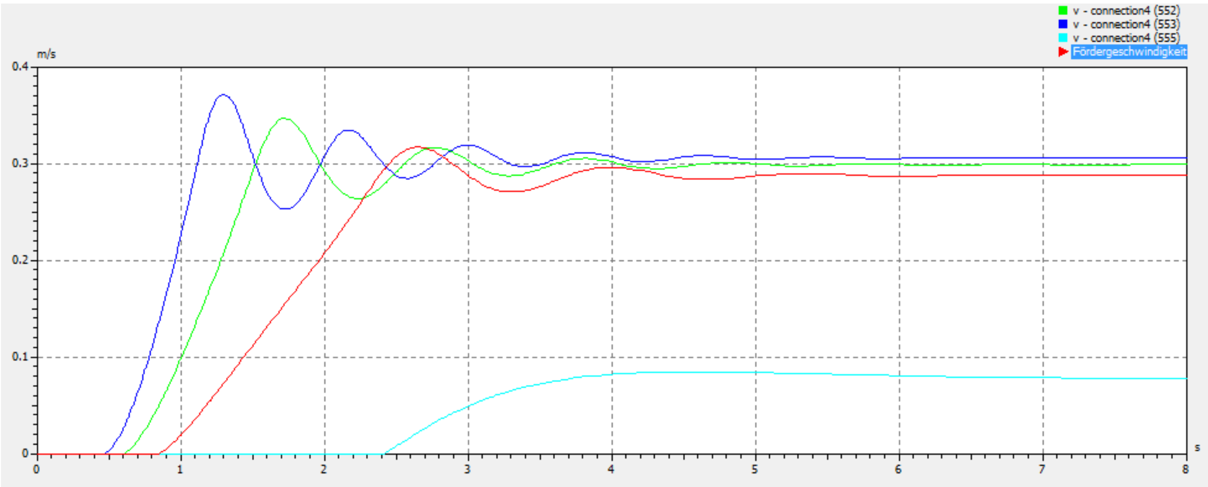


**Figure 25: Run-up characteristic of the transportation speed regarding different Oil levels**

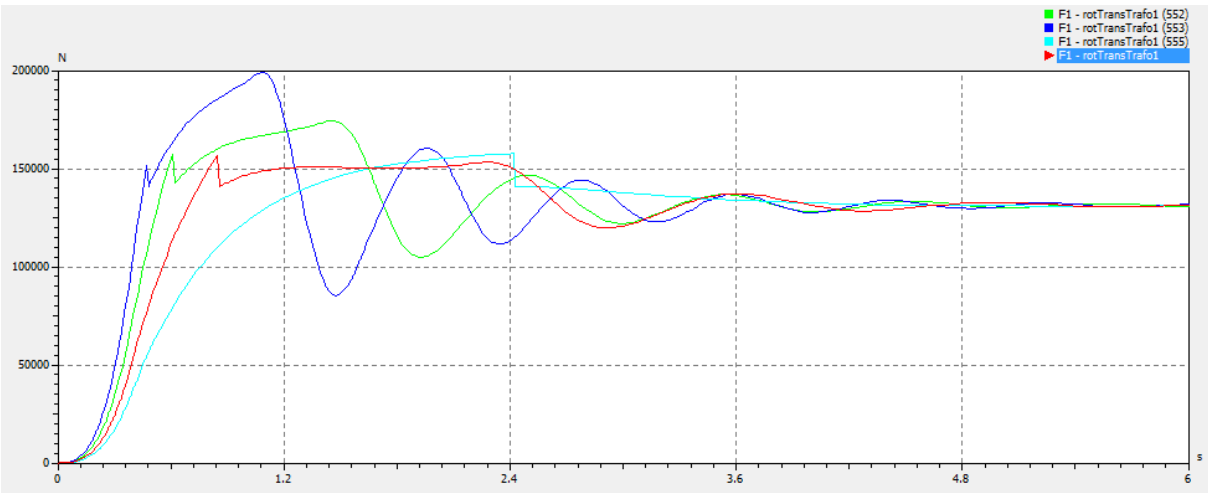


**Figure 26: Run-up characteristic of the chain force regarding different Oil levels**

# 8 Theoretical introduction to Design of Experiments (DoE)

Design of experiments belongs to the field of applied statistics and plays an important role in planning, conducting, analysing and interpreting data from engineering experiments. When several variables influence a certain characteristic of a product, the best strategy is then to design an experiment so that valid, reliable and sound conclusions can be drawn effectively, efficiently and economically.(Vgl.[ANT03])

In a designed experiment, the engineer often makes deliberate changes in the input variables (or factors) and then determines how the output functional performance varies accordingly. It is important to note that not all variables affect the performance in the same manner. Some may have strong influences on the output performance, some may have medium influences and some have no influence at all. Therefore, the objective of a carefully planned designed experiment is to understand which set of variables in a process affects the performance most and then determine the best levels for these variables to obtain satisfactory output functional performance in products..(Vgl.[ANT03])

A number of successful applications of DoE have been reported by many US and European manufacturers over the last fifteen years or so. The potential applications of DOE in manufacturing processes include:

- improved process yield and stability
- improved profits and return on investment
- improved process capability
- reduced process variability and hence better product performance consistency
- reduced manufacturing costs
- reduced process design and development time
- heightened morale of engineers with success in chronic-problem solving
- increased understanding of the relationship between key process inputs and output(s)
- increased business profitability by reducing scrap rate, defect rate, rework, retest, et

## 8.1 Explanation of the used experiment designs

Each test (or simulation) takes time and further on costs money. So, it is important to distinguish in detail which experimental design fits the best to the required task to minimize the required test. Is the size of the experiments to small, the relevant results (most times the impact of the parameters) cannot be shown with the expected accuracy. So, the possibility that relevant distinctions are overlooked is raising. (Vgl. [KLE13])

In this case, the DoE approaches where used generate knowledge if and how big the influence of the initial parameters on the calculation of the chain force is (screening). Further on, the most significant parameters where investigated again with a more detailed experimental design to find out how the influence of it looks like (e.g. linear ect.).

In the next two chapters the experimental designs which were used within the thesis are explained in detail.

### 8.1.1 Screening

Screening is the most common experimental design to find out of a large group of parameters the ones which have the biggest effect on the process or in our case the calculation, or to say it in other words: the purpose of screening is to identify those factors and parameters that demand further investigations. Depending on the size of the experimental design, e.g. full factorial or part factorial, also interaction of a higher grade can be determined. It is important to mention that the screening is just valid within the defined parameter ranges, so an extrapolation is not valid. .(Vgl.[ANT03])

The screening approach was used to find out which parameters have the strongest influence on the calculation. At this stage, it was not important to find out how big the influence of interactions of higher grades are.
For this reason, it is sufficient to start with a "rough" experimental designs (e.g. 1/2 factorial design for the screening). Further on a multi-level screening was done to get more detailed information about the impact of the parameters.

## 8.2 Introduction to Minitab

Minitab[5] is statistic software which helps to analyse data in the field of quality management. It provides a big variety of different statistical methods and graphical tools specially designed for tasks in industry and research. Especially for applications like Six Sigma and a big variety of different DOE application Minitab is the perfect tool for fast and accurate results. [MIN17]

In this thesis, Minitab was used for the weighting of the influence of the initial parameters of the conveyer and the construction and analysis and further on to generate the best fitting experimental design.

[5] Developed by Minitab Inc.

## 8.3 Construction of an experimental design

## 8.4 Two level screening designs

As in 8.1 already mentioned, the two-level screening was used to get a rough overview about the impact of the single parameters. The number of factors[6] k is directly influencing the number of test/simulation. For a full factorial design this would $2^k$ steps. This means that before a experimental design could be planned an analyses about the meaningfulness of the parameters has to be done. The output was group of five parameters which have the most impact: (Vgl.[KLE13])

- Plan length $L_I$
- Radius $R_{II}$
- Angular length $L_{III}$
- Angle $\alpha$
- Mass m

**Table 5: Half factorial design for 2-Level screening**

| Nr | LI | RII | LIII | alpha |
|---|---|---|---|---|
| 1 | 1 | 2 | 1 | 2 |
| 2 | 2 | 2 | 1 | 1 |
| 3 | 2 | 1 | 1 | 2 |
| 4 | 2 | 1 | 2 | 1 |
| 5 | 2 | 2 | 2 | 2 |
| 6 | 1 | 1 | 2 | 2 |
| 7 | 1 | 1 | 1 | 1 |
| 8 | 1 | 2 | 2 | 1 |

It is obvious the influence of the mass m is the most significant one which makes it unnecessary to consider this factor in the screening. This means that only four parameters will be considered in the first experimental design.

Further on, the assumption was taken that interactions of higher grades will not be considered in detail which reduces the design from a full factorial design to ½ factorial design (so from $2^4$=16 steps to $½ \cdot 2^4$=8 steps). This is another major reduction of the design range with the disadvantage that information about the interactions of higher grades will be lost.

Table 5 shows the experimental design with two steps for each factor (value 1 and value 2) which was developed and analysed with the program minitab. The results can be seen in Figure 27 and Figure 28.

[6] Factors is the nomination for parameter in the most literatures about DoE

## 8.5 Multi-level screening designs

As in 5.4 already explained, it is important to know how exactly the influence of a parameter looks in a mathematical way. This is especially then important when it is about to develop the reference curve of the corrective factors. If the impact of parameters is linear, two points are enough for fully describing the graph, if the impact is following a higher degree function 3 to 5 points are necessary to get a satisfactory result.

This is exactly the reason why a multi-level screening design is introduced. In general, this designs are very complex structured and solvable with a variance analysis. To keep the effort as low as possible, the analysis was also done with minitab. At this point, also the mass was considered.

For the design of the experiment a 5 factor with 4 levels design (each factor has four different values) was used (see Table 6). The result of this designed experiment can be seen in Figure 30

**Table 6: Multi-Level design**

| Nr | m | LI | RII | LIII | alpha |
|---|---|---|---|---|---|
| 1 | 1 | 1 | 1 | 1 | 1 |
| 2 | 1 | 2 | 2 | 2 | 2 |
| 3 | 1 | 3 | 3 | 3 | 3 |
| 4 | 1 | 4 | 4 | 4 | 4 |
| 5 | 2 | 1 | 2 | 3 | 4 |
| 6 | 2 | 2 | 1 | 4 | 3 |
| 7 | 2 | 3 | 4 | 1 | 2 |
| 8 | 2 | 4 | 3 | 2 | 1 |
| 9 | 3 | 1 | 3 | 4 | 2 |
| 10 | 3 | 2 | 4 | 3 | 1 |
| 11 | 3 | 3 | 1 | 2 | 4 |
| 12 | 3 | 4 | 2 | 1 | 3 |
| 13 | 4 | 1 | 4 | 2 | 3 |
| 14 | 4 | 2 | 3 | 1 | 4 |
| 15 | 4 | 3 | 2 | 4 | 1 |
| 16 | 4 | 4 | 1 | 3 | 2 |

## 8.6 Evaluation and weighting of the role of the parameters

As in 8 explained, one strength of DoE is to find out if and how big the influence of the input parameters and their combinations on the system is. As in 5.4 described, the number of corrective factors should be as low as possible which leads to the idea to just introduce one for a parameter if its influence is significant.

To find this out, a screening of the initial parameters was done with the help of minitab. As test plan a simple half factorial test plan was used to get a rough overview about the specific influences of the parameter. Out of this result, the parameters which have no, or just a small influence were sorted out and another sequence with a more detailed test plan was done.
The results of the final sequence can be seen in Figure 27and Figure 28.

Figure 27 shows the main effects plot for the chain force F_chain. The grade of the pitch of the single graphs shows how big the influence of the specific parameter is on the chain force. It is visible that the parameter LIII has the biggest and LI the lowest impact. Out of this chart the assumption could be taken that the influence of LI is nearly zero because of the nearly horizontal line but in reality, it is not. In this case, it is important to also have a look on the interaction of a higher grade[7] between these variables.

This interaction of higher grades are shown in the Pareto diagram in Figure 28. It is visible that the combination of $L_I$ and radius $R_{II}$ has a strong effect on the chain force which makes it necessary to consider also a corrective factor for $L_I$.

[7] Interaction by combining two parameters (e.g. LI + LIII)

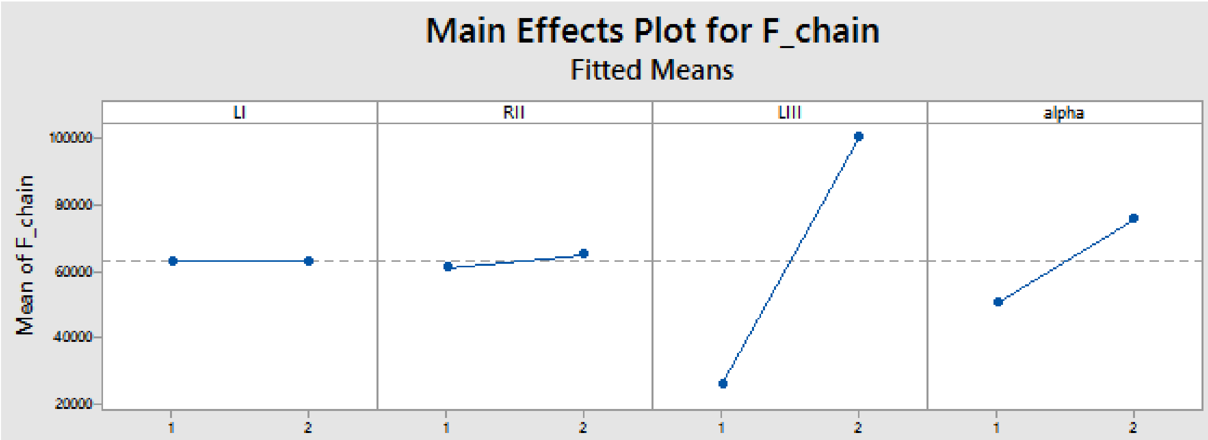


Figure 27: main effects Plot for the chain force

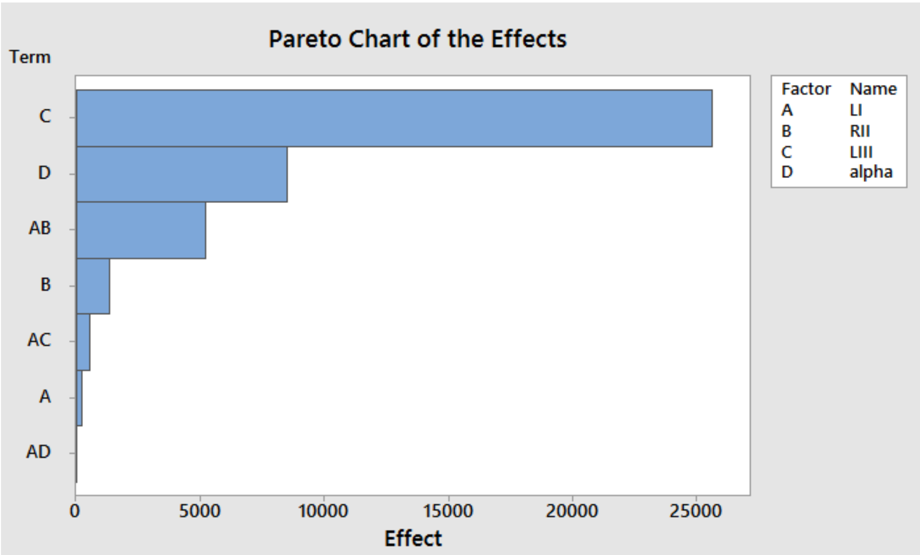


Figure 28: Pareto chart of the effects of the chain force

For the corrective factors $C_{RIV}$ and $C_{LV}$ the assumption was taken that the behaviour of the parameter $R_{IV}$ and $L_V$ are similar to the one of $R_{II}$ and $L_I$. For this reason, the corrective graphs were the same.

# 9 Validation of the analytical calculation

The validation is the most important step within the whole thesis. It is necessary to determine "how" accurate the analytic calculation is, so to find out how big the gap between the "real" chain force and the assumption through the analytic calculation is.

So, the validation is actually a comparison of the results of MBS, the analytic calculation and if available also data which was measured from an already existing conveyor. The measured data is in the beginning necessary to double check if the MBS is following the reality. Later on, the measured data is obviously not necessary anymore and the data which results out of the MBS can be seen as measured data from existing systems.

The approach for validation follows a simple guideline which is shown in Figure 29.

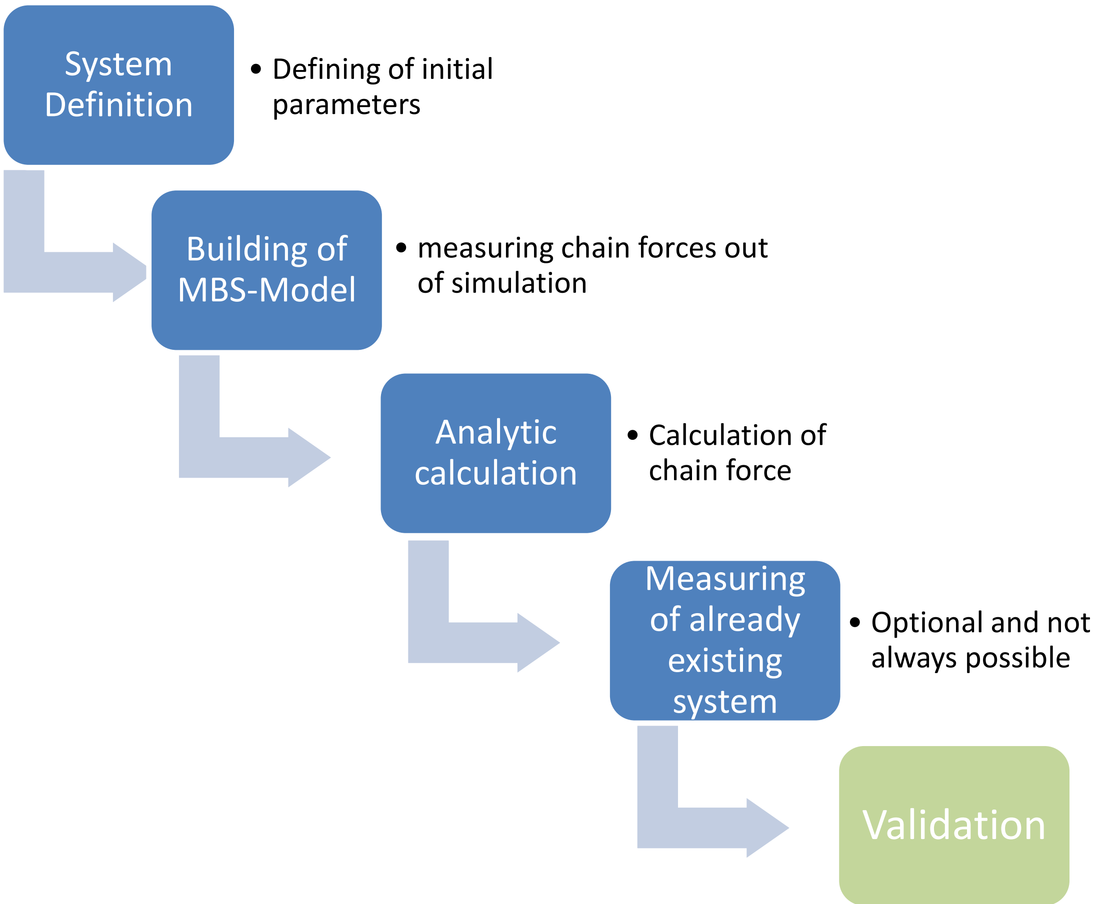


Figure 29: Approach for validation

It is estimative that the failure between the analytic calculation and the MBS will be not acceptable (>10%) without the consideration of several corrective factors. For this reason, the approach for the determination of corrective factors is shown in chapter 9.3 for one factor in detail.

**The validation process contains a lot of confidential data concerning the characteristic of the chain force and other parameters of the simulation, analytic calculation and especially the measurements out of existing conveyor.**
**For this reason, the approach for the validation is briefly described but without mentioning any specific values.**

## 9.1 Systematical approach reproducible results in MBS

It is important, that all the steps which are done within the validation are done always the same to reduce the risk of doing failures and so to ensure a reproducible result (e.g. measuring failures in the MBS). This makes it necessary to introduce a systematically approach especially for steps were a high number of results are possible. The best example for this are the results of the MBS. In general, the results of MBS are graphs Out of this graph the value of the chain force can be determined geometrical

To warrant a reproducible result the following checklist was created:

1. Is the transient state already reached?
2. Is the graph of the chain force following a alternating sequence?
3. Is the Joint[8] for measuring the chain force the correct one (two cells before the drive sprocket on the tight site)?
4. Are the working forces from every direction (X-,Y direction) summarized?
5. Is the average of the chain force over the simulation measured?

[8] MSC Adams provides the feature to measure the contact force from all directions of the coordinate-system in every joint

## 9.2 Development of a valid range for the friction coefficients

As in 4.2.2 already explained, the determination/choice of the value of the friction coefficients between the contacts guiding roller – track and in-between the chain bolts is crucial for an accurate and realistic calculation of the chain force. In literature, the standard values of this coefficients can strongly vary which can lead to strong variating results. For this reason, it is necessary to give the user a range for the friction coefficients where the calculation depicts the real system.

Table 8 shows an overview of possible ranges of the friction coefficients out of diverse literature. It is visible that the ranges strongly vary which can result to strongly varying results of the chain forces.

During the validation of the analytic calculation with the measured data the following range for the two friction coefficients for a polluted conveyor was developed (Table 7):

**Table 7: fixed ranges for friction coefficients**

| fixed ranges | |
|---|---|
| $\mu_{GR}$ | $\mu_{CB}$ |
| **0,002....0,005** | **0,2....0,4** |

**Table 8: Overview of the possible friction coefficients ranges**

| Source | range $\mu_{GR}$ | range $\mu_{CB}$ |
|---|---|---|
| [RÖM15] | 0.002.....0,003 | 0,3....0,5 |
| [WMJ09] | 0,001....0,005 | 0,1....0,2 |
| [BÖG07] | ------ | 0,01....0,15 |
| [GHS06] | ------ | 0,1....0,4 |

## 9.3 Determination of corrective factors and their characteristic

As in preceded chapters already mentioned, the corrective factors play a major role for the precision of the calculation. The specialty of all the corrective factors which are used in this thesis is, that they don't have a fixed value but further more they have to be determined new for each calculation which means that they have to get changed when the initial parameters change. This mainly comes through the fact that influence of single resistances of the conveyor can strongly vary when some of the initial parameters get changed.

If for example the pitch of the angular part of the transportation path gets raised, the influence of the climbing resistance is rising dramatically.
For this reason, the corrective factors were considered in a "dynamic" way.

As in chapter 5.4 already explained, each parameter which has a big influence on the chain force gets one corrective factor. Within the analytic calculation, all the forces where a parameter has influence on is separated summed and multiplied by the specific parameter (see 14.1).

To find the characteristic of the reference curve it is possible to use different approaches. One is to use the method "change-one-at-time" which basically means that just one parameters gets changed per test. This has to be done multiple times to get an characteristic of the influence of the specific parameters on the chain force.  This approach has the advantage that the result is accurate compared to other appoach but on the other hand it needs many iteration steps (so minimum for each parameter 3-4). This leads to experimental design with 20 steps.

A more elegant way is to use the benefits of DoE. Figure 30 shows the output of a designed experiment with the consideration of four levels per parameter. The experimental design is shown in Table 6. As it can be seen, the design includes 16 steps.

The problem of an approach with Taguchi is, that the result can not directly been used for development of the reference curve for the corrective factors because all the output data which is shown is based on the mean of the chain force. For this reason, the result of Figure 30 can be used to get overview of the possible characteristic of the influence of the parameters. For example, it can be seen that the influence of the mass is nearly linear and further on all the other parameters are following function with higher degrees.

Within the thesis, a mixture of both approaches, so the "change-one-at-time" and the DoE after Taguchi was used to determine an accurate characteristic of the reference curves. Depending on the grade of impact (which can be seen in Figure 30) the number of tests for an "change-one-at-time" approach varies between the parameters (e.g. for $C_{LIII}$ more steps than for $C_{LI}$).

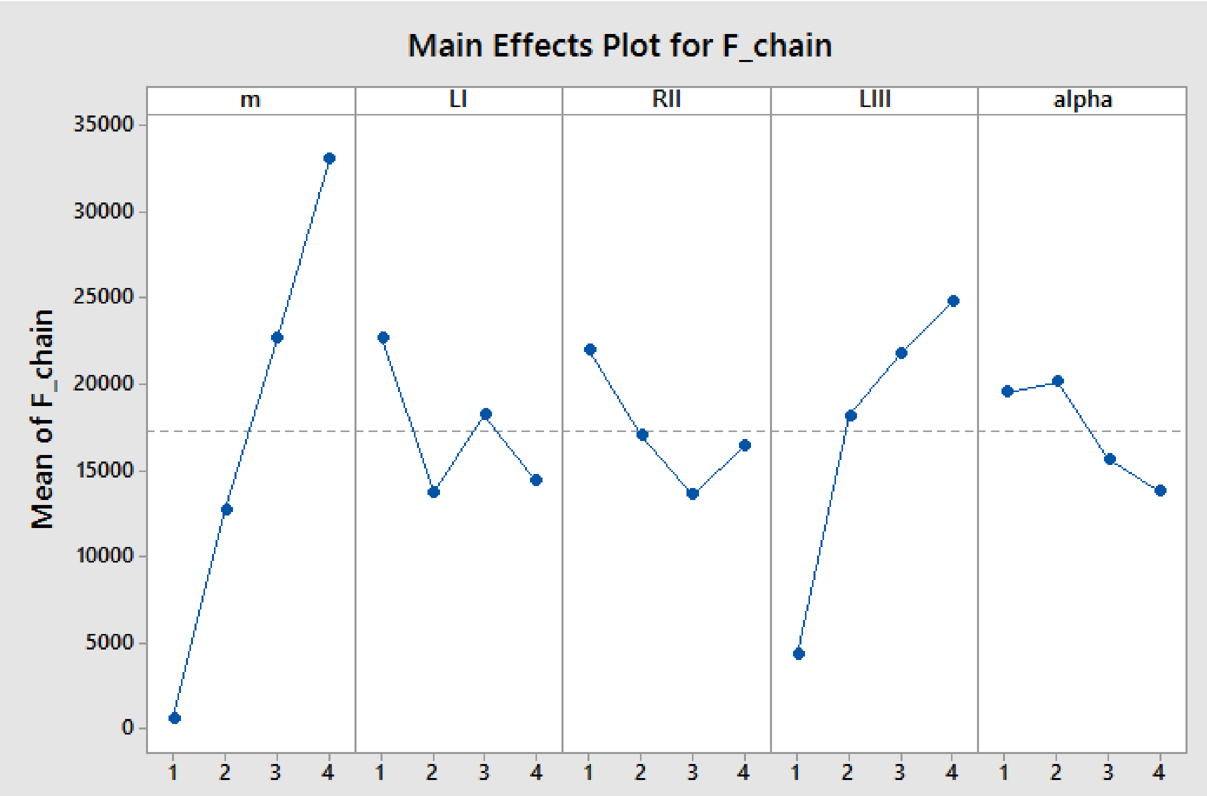


**Figure 30:Main effects Plot- linearity of Masses**

### 9.3.1 Example for developing the corrective characteristic of $C_{RII}$

In the example of the radius RII, all resistances where summed and multiplied by $C_{RII}$.

This is important because the corrective factors have just an influence on the parameter which it is based on.

Depending on the value of the initial parameter, the corrective factor is measured out of charts (see Figure 31). So, $C_{RII}$ equals the difference between the analytic calculation (blue) and the measured or simulated value (orange).

The measured/simulated characteristic was done with an average load of 58 kg. This is the load of a conveyor with a grade of filling with 75% and geometric measures which are laying in the middle of the defined ranges. It is obvious that for every change of the load, and indirect of the geometric measures of the cell, the characteristic curve (orange) had to be done again. But as in Figure 30 visible, the impact on the overall force of the parameter load (m) is nearly linear. This makes it possible to take the assumption that a rise of the value from the parameter mass can be considered as a simple parallel displacement of the reference curve.

The evidence for this assumption is visible in the validation of the analytic calculation (see 9.4)Validation of the analytical calculationThe parameter mass is for sure one of the parameter with the strongest impact (visible in Figure 30). If the assumption would be wrong, the deviation of the calculated chain force would be much higher.

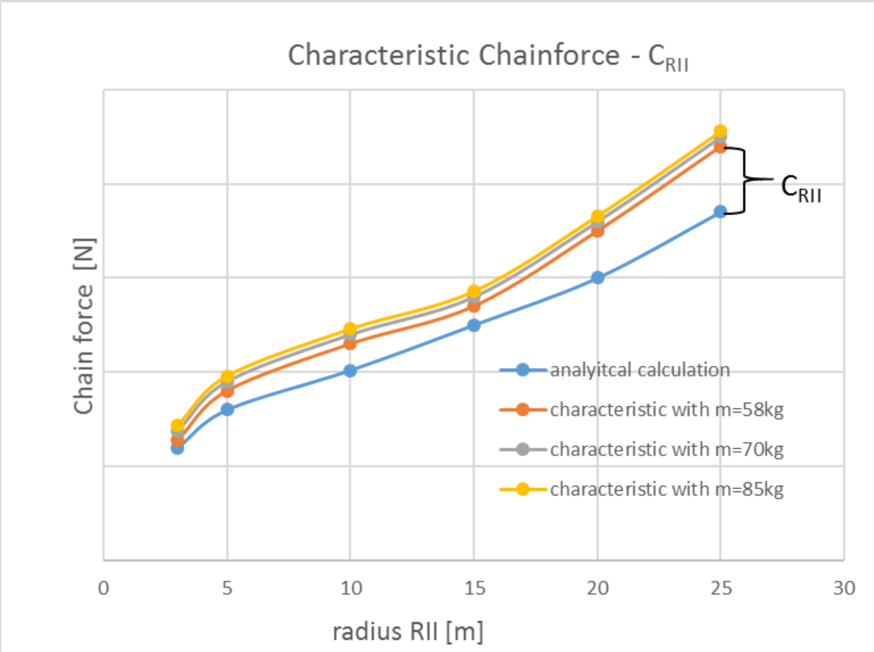


**Figure 31: Chart for determination of $C_{RII}$**

## 9.4 Validation of the analytical calculation

Table 9 shows an excerpt of the validation of the analytical calculation. The tests 1 to 8 where validated by the comparison of MBS and analytical calculation. Test number 9 results through the comparison of measured data from an existing system and analytical calculation.

Additionally, all the MBS models which were used to determine the corrective factors can also be used to validate the analytical calculation. In sum this are 48 models.
All in all, the mean of the occurring deviation of the analytical calculation is 2,1% which is in-between the required accuracy.
Tests without using corrective factors have shown a deviation of 9,3%.

**Table 9: Excerpt of the validation of the analytical calculation[9]**

| Nr. | Definition system | Deviation |
|---|---|---|
| 1 | 10m/40grd-10m/58kg | **0.64%** |
| 2 | 10m/40grd-10m/0kg | **1.55%** |
| 3 | 40m/0grd-0m/58kg | **-0.25%** |
| 4 | 40m/0grd-0m/0kg | **2.38%** |
| 5 | 90m/40grd-11m/70kg | **-1.42%** |
| 6 | 90m/40grd-90m/58kg | **-1.13%** |
| 7 | 90m/40grd-90m/30kg | **-0.39%** |
| 8 | 90m/30grd-90m/30kg | **0.30%** |
| 9 | Existing conveyor | **1.24%** |

[9] Naming schematic for definition system: LI[m]/alpha[°]-LIII[m]/mass[kg]

# 10 Conclusions

The analytical calculation of conveyer systems is and always was a big challenge especially when the transportation length is very long (up to 300m). The thesis has shown, that analytical calculations which are built with standard approaches for calculating mechanical system, can lead to results with an unexpected high level accuracy. Combined with well-defined corrective factors the results can have a higher an accuracy than the failures which are resulting through physical measurements out of existing systems (deviations of approx. 2%).

Even without any methods for correcting the calculated chain force, the results can be used as a design calculation for sure, and this also without using a safety factor of 10. So, a reduction of the safety factor to 8 or 7 is possible without doing any simulations. This is already a huge improvement compared to the "old" way of doing design calculation.

With the consideration of the corrective factors which were developed with the help of Multi body simulations (MSC Adams) and a run up simulation with a system simulation (SimulationX) the safety factor could be reduced by 50%.

These results can lead to a major reduction of the production costs (trough the saved material), the running costs (through the lower power consumption) and further on also to lower maintenance costs because of the lower loads and higher grade of reliability.

Nowadays simulations (no matter which kind) are bit like the holy grail for every kind of problem statement in the everyday life of engineers. Sometimes "modern" engineers have lost the trust and probably sometimes also the knowledge how problems can be solved without using complex simulation programs or to say it in other words: they lost the know-how of solving technical problems in the old-school way. The thesis has shown that analytical design calculation of complex system can also lead to acceptable result with failures of 5 to 10% even without any corrective factors.

For sure, the results with the help of the different simulation tools are better, but compared to the effort, the necessary know how of using these programs combined with the needed of special IT resources and the factor time, the decision if the path simulation or the path no simulation has be taken carefully and under consideration of all possible influences.

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# 12 List of tables

# 13 Table of figures

# 14 Attachements

## 14.1 Analytical calculation via Mathcad

### Definitions:

| | |
|---|---|
| "transportation velocity" | $v$ |
| "filling degree cell" | $f_g$ |
| "volume Zelle" | $V_Z$ |
| "plane path length" | $l_I$ |
| "arc length II" | $l_{IIb}$ |
| "Radius II" | $r_{B\_II}$ |
| "Radius IV" | $r_{B\_IV}$ |
| "angular path length III" | $l_{III}$ |
| "heighten plane length V " | $l_V$ |
| "angle of pitch" | $\alpha$ |
| "density bulk" | $\rho_S$ |
| "pitch guiding rollers" | $p_L$ |
| "preload force" | $F_{Vor}$ |

### Chain

| | |
|---|---|
| "pitch chain" | $p$ |
| "number chain lines" | $n_k$ |
| "mass per chain link" | $m_{k\_G}$ |
| "mass per cell" | $m_Z$ |
| "mass guiding roller" | $m_L$ |
| "pitch circle diameter" | $d$ |
| "diameter chain bolt" | $d_{B\_K}$ |

## Friction coefficients

| | |
|---|---|
| "guiding roller–rails" | $\mu_L$ |
| "chain bolt" | $\mu_{KB}$ |

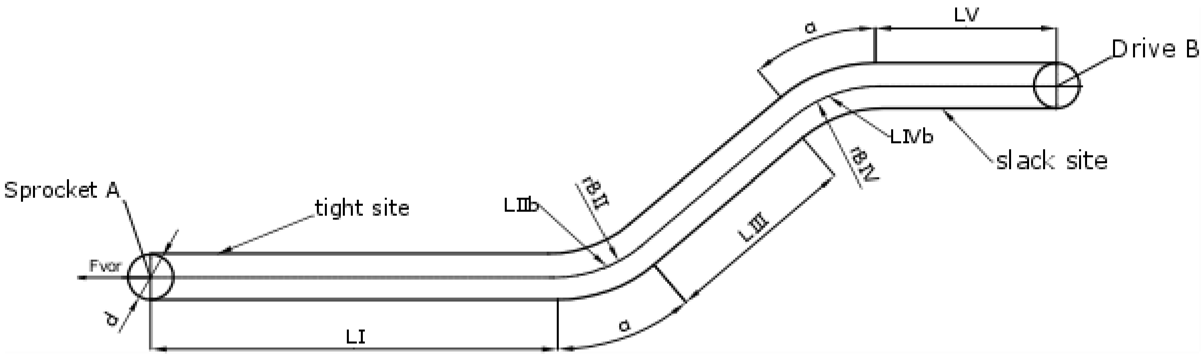
## Degrees of efficiency

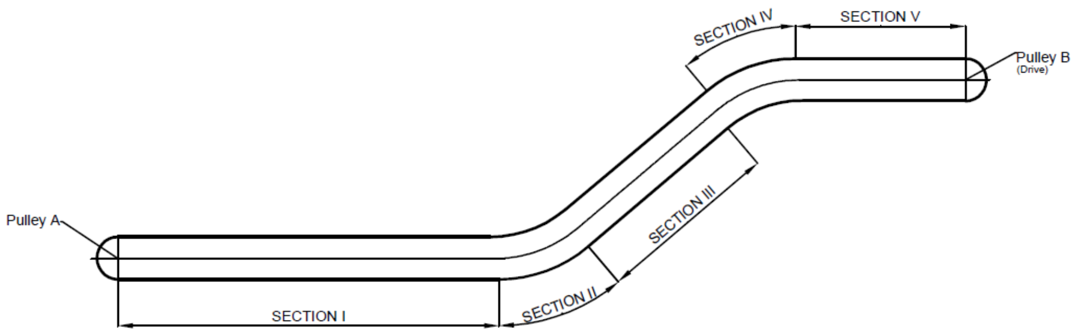
| | |
|---|---|
| "drive/motor" | $\eta_M$ |
| "gearbox" | $\eta_G$ |
| "transmission gearbox" | $i$ |

## Correctiv factors

| | |
|---|---|
| "for plane length I" | $C_{LI}$ |
| "for radius II" | $C_{RII}$ |
| "for angular length III" | $C_{LIII}$ |
| "for radius IV" | $C_{RIV}$ |
| "for heighten length V" | $C_{LV}$ |
| "for run up process" | $C_{RP}$ |
| "for angle of pitch" | $C_{\alpha}$ |

# System Sketches



# Resistance Calculation-SECTION I

## Guiding Rollers - Tight site I

"Normal–Force"

$$F_{N\_res\_I\_LA} := g \cdot \left[ \frac{p_L}{p} \cdot \left( m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G} \right) \right] + 2 \cdot g \cdot m_L$$

"Friction Force for the Length I"

$$F_{R\_II\_LA} := F_{N\_res\_I\_LA} \cdot \mu_L \cdot \frac{l_I}{p_L} \cdot C_{LI}$$

## Guiding Rollers - Slack site I

"Normal–Force"

$$F_{N\_res\_I\_LE} := g \cdot \left[ \frac{p_L}{p} \cdot \left( m_Z + n_k \cdot m_{k\_G} \right) \right] + 2 \cdot g \cdot m_L$$

"Friction Force for the Length I"

$$F_{R\_II\_LE} := F_{N\_res\_I\_LE} \cdot \mu_L \cdot \frac{l_I}{p_L} \cdot C_{LI}$$

# Resistance Calculation - SECTION II

## Guiding Rollers - Tight site II

"Normal–Force"

$$F_{N\_res\_II\_LA} := \int_0^{\alpha} \left( \left( \frac{g}{p} \right) \cdot \left( m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G} \right) + \frac{2}{p_L} \cdot g \cdot m_L \right) \cdot \cos(\beta) \cdot \left( r_{B\_II} - \frac{d}{2} \right) \mathrm{d}\beta$$

"Friction Force over arc length II – Tight site"

$$F_{R\_lII\_LA} := F_{N\_res\_II\_LA} \cdot \mu_L \cdot C_{RII}$$

"Climbing Resistance–Section II – Tight site"

$$F_{W\_ST\_II\_LA} := \int_0^{\alpha} \left( \frac{1}{p} \cdot g \cdot \left( m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G} \right) + \frac{1}{p_L} \cdot g \cdot 2 \cdot m_L \right) \cdot \left( r_{B\_II} - \frac{d}{2} \right) \cdot \sin(\beta) \cdot C_{RII} \, \mathrm{d}\beta$$

## Guiding Rollers - Slack site II

"Normal – Force"

$$F_{N\_res\_II\_LE} := \int_0^{\alpha} \left( \left( \frac{g}{p} \right) \cdot \left( m_Z + n_k \cdot m_{k\_G} \right) + \frac{2}{p_L} \cdot g \cdot m_L \right) \cdot \cos(\beta) \cdot \left( r_{B\_II} + \frac{d}{2} \right) \mathrm{d}\beta$$

"Friction Force over arc length II – Tight site"

$$F_{R\_lII\_LE} := F_{N\_res\_II\_LE} \cdot \mu_L \cdot C_{RII}$$

"Climbing Resistance–Section II – Slack site"

$$F_{W\_ST\_II\_LE} := \int_0^{\alpha} \left( \frac{1}{p} \cdot g \cdot \left( m_Z + n_k \cdot m_{k\_G} \right) + \frac{1}{p_L} \cdot g \cdot 2 \cdot m_L \right) \cdot \left( r_{B\_II} + \frac{d}{2} \right) \cdot \sin(\beta) \cdot C_{RII} \, \mathrm{d}\beta$$

# Resistance Calculation - SECTION III

## Guiding Rollers - Tight site III

"Normal Force"

$$F_{N\_res\_III\_LA} := \left(\left(\frac{l_{III} \cdot g}{p}\right) \cdot \left(m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G}\right) + \frac{2 \cdot l_{III}}{p_L} \cdot g \cdot m_L\right) \cdot \cos(\alpha)$$

"Friction Force for the Length III – Tight site III"

$$F_{R\_lIII\_LA} := F_{N\_res\_III\_LA} \cdot \mu_L \cdot C_{LIII}$$

"Climbing Resistance–Section II – Tight site III"

$$F_{W\_ST\_III\_LA} := \left(\frac{l_{III}}{p} \cdot \left(m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G}\right) + \frac{l_{III}}{p_L} \cdot 2 \cdot m_L\right) \cdot g \cdot \sin(\alpha) \cdot C_{LIII} \cdot C_\alpha$$

## Guiding Rollers - Slack site III

"Normal Force"

$$F_{N\_res\_III\_LE} := \left(\left(\frac{l_{III} \cdot g}{p}\right) \cdot \left(m_Z + n_k \cdot m_{k\_G}\right) + \frac{2 \cdot l_{III}}{p_L} \cdot g \cdot m_L\right) \cdot \cos(\alpha)$$

"Friction Force for the Length III – Slack site III"

$$F_{R\_lIII\_LE} := F_{N\_res\_III\_LE} \cdot \mu_L \cdot C_{LIII} \cdot C_\alpha$$

"Climbing Resistance–Section II – Slack site III"

$$F_{W\_ST\_III\_LE} := \left(\frac{l_{III}}{p} \cdot \left(m_Z + n_k \cdot m_{k\_G}\right) + \frac{l_{III}}{p_L} \cdot 2 \cdot m_L\right) \cdot g \cdot \sin(\alpha) \cdot C_{LIII} \cdot C_\alpha$$

# Resistance Calculation - SECTION IV

## Guiding Rollers - Tight site IV

"Normal Force"

$$F_{N\_res\_IV\_LA} := \int_0^{\alpha} \left( \left( \frac{g}{p} \right) \cdot \left( m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G} \right) + \frac{2}{p_L} \cdot g \cdot m_L \right) \cdot \cos(\beta) \cdot \left( r_{B\_IV} + \frac{d}{2} \right) \mathrm{d}\beta$$

"Friction Force over arc length IV – Tight site"

$$F_{R\_UV\_LA} := F_{N\_res\_IV\_LA} \cdot \mu_L \cdot C_{RIV}$$

"Climbing Resistance–Section IV – Tight site "

$$F_{W\_ST\_IV\_LA} := \int_0^{\alpha} \left( \frac{1}{p} \cdot g \cdot \left( m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G} \right) + \frac{1}{p} \cdot g \cdot 2 \cdot m_L \right) \cdot \left( r_{B\_IV} + \frac{d}{2} \right) \cdot \sin(\beta) \, \mathrm{d}\beta \cdot C_{RIV}$$

## Guiding Rollers - Slack site IV

"Normal Force"

$$F_{N\_res\_IV\_LE} := \int_0^{\alpha} \left( \left( \frac{g}{p} \right) \cdot \left( m_Z + n_k \cdot m_{k\_G} \right) + \frac{2}{p_L} \cdot g \cdot m_L \right) \cdot \cos(\beta) \cdot \left( r_{B\_IV} - \frac{d}{2} \right) \mathrm{d}\beta$$

"Friction Force over arc length IV – Slack site"

$$F_{R\_UV\_LE} := F_{N\_res\_IV\_LE} \cdot \mu_L \cdot C_{RIV}$$

"Climbing Resistance–Section IV – Slack site "

$$F_{W\_ST\_IV\_LE} := \int_0^{\alpha} \left( \frac{1}{p} \cdot g \cdot \left( m_Z + n_k \cdot m_{k\_G} \right) + \frac{1}{p} \cdot g \cdot 2 \cdot m_L \right) \cdot \left( r_{B\_IV} - \frac{d}{2} \right) \cdot \sin(\beta) \, \mathrm{d}\beta$$

# Resistance Calculation - SECTION V

## Guiding Rollers - Tight site V

“Normal Force”

$$F_{N\_res\_V\_LA} := g \cdot \left[ \frac{p_L}{p} \cdot \left( m_Z + V_Z \cdot f_g \cdot \rho_S + n_k \cdot m_{k\_G} \right) \right] + 2 \cdot g \cdot m_L$$

“Friction Force for the Length V – Tight site ”

$$F_{R\_lV\_LA} := F_{N\_res\_I\_LA} \cdot \mu_L \cdot \frac{l_V}{p_L} \cdot C_{LV}$$

## Guiding Rollers - Slack site V

“Normal Force”

$$F_{N\_res\_V\_LE} := 0.5 \cdot g \cdot \left[ \frac{p_L}{p} \cdot \left( m_Z + n_k \cdot m_{k\_G} \right) \right] + 2 \cdot g \cdot m_L$$

“Friction Force for the Length V – Slack site ”

$$F_{R\_lV\_LE} := F_{N\_res\_I\_LE} \cdot \mu_L \cdot \frac{l_V}{p_L} \cdot C_{LV}$$

# Resistance calculation through chain bolt friction

## Defection Sprocket A - Section I

"Theoretical chain force over sprocket A"

$$F_{K\_TH\_I} := \left(\left(\frac{F_{Vor}}{2} + F_{R\_II\_LE}\right)\right) \cdot \frac{1}{n_k}$$

"resistance through chain bolt friction – Deflection A"

$$F_{KB\_UI} := F_{K\_TH\_I} \cdot \frac{d_{B\_K}}{d} \cdot \mu_{KB} \cdot n_k \cdot 2$$

## Deflection radius II- Tight site

" Theoretical chain force during Radius II – Tight site"

$$F_{K\_TH\_II\_LA} := \left(\frac{F_{Vor}}{2} + F_{R\_II\_LE} + F_{KB\_UI} + F_{R\_II\_LA} + F_{R\_III\_LA} + F_{W\_ST\_II\_LA}\right) \cdot \frac{1}{n_k}$$

"resistance through chain bolt friction Radius II – Tight site"

$$F_{KB\_II\_LA} := F_{K\_TH\_II\_LA} \cdot \frac{d_{B\_K}}{d} \cdot \mu_{KB} \cdot n_k \cdot 2$$

## Deflection radius II- Slack site

" Theoretical chain force during Radius II – Slack site"

$$F_{K\_TH\_II\_LE} := \left(\frac{F_{Vor}}{2} + F_{W\_ST\_II\_LE} + F_{R\_III\_LE}\right) \cdot \frac{1}{n_k}$$

"resistance through chain bolt friction Radius II – Slack site"

$$F_{KB\_II\_LE} := F_{K\_TH\_II\_LE} \cdot \frac{d_{B\_K}}{d} \cdot \mu_{KB} \cdot n_k \cdot 2$$

## Deflection radius IV- Tight site

“ Theoretical chain force during Radius IV – Tight site”

$$F_{K\_TH\_IV\_LA} := \left( \frac{F_{Vor}}{2} + F_{R\_II\_LE} + F_{KB\_UI} + F_{R\_II\_LA} + F_{R\_UI\_LA} + F_{W\_ST\_II\_LA} \right) \cdot \frac{1}{n_k}$$

“resistance through chain bolt friction Radius IV – Tight site”

$$F_{KB\_II\_LA} := F_{K\_TH\_II\_LA} \cdot \frac{d_{B\_K}}{d} \cdot \mu_{KB} \cdot n_k \cdot 2$$

## Deflection radius IV- Slack site

“ Theoretical chain force during Radius IV – Slack site”

$$F_{K\_TH\_II\_LE} := \left( \frac{F_{Vor}}{2} + F_{W\_ST\_II\_LE} + F_{R\_UI\_LE} \right) \cdot \frac{1}{n_k}$$

“resistance through chain bolt friction Radius IV – Slack site”

$$F_{KB\_II\_LE} := F_{K\_TH\_II\_LE} \cdot \frac{d_{B\_K}}{d} \cdot \mu_{KB} \cdot n_k \cdot 2$$

## Deflection Sprocket B

“Theoretical chainvorce before deflection B”

$$F_{K\_TH\_III\_LA} := F_{K\_TH\_II\_LA} + \left( F_{W\_ST\_III\_LA} + F_{R\_UII\_LA} + F_{KB\_II\_LA} \right) \cdot \frac{1}{n_k}$$

“Theoretical chainvorce after deflection B”

$$F_{K\_TH\_III\_LE} := F_{K\_TH\_II\_LE} + \left( F_{R\_UII\_LE} + F_{W\_ST\_III\_LE} \right) \cdot \frac{1}{n_k}$$

“resistance through chain bolt friction – Deflection B”

$$F_{KB\_U2} := \left( F_{K\_TH\_III\_LE} + F_{K\_TH\_III\_LA} \right) \cdot \frac{d_{B\_K}}{d} \cdot \mu_{KB} \cdot n_k$$

# Summary of calculated Data

## Total resistance through guiding rollers

$$F_{R\_ges\_LA} := F_{R\_II\_LA} + F_{R\_UI\_LA} + F_{R\_UII\_LA} + F_{R\_UV\_LA} + F_{R\_IV\_LA}$$

$$F_{R\_ges\_LE} := F_{R\_UII\_LE} + F_{R\_II\_LE} + F_{R\_UI\_LE} + F_{R\_UV\_LE} + F_{R\_IV\_LE}$$

$$F_{R\_ges} := F_{R\_ges\_LA} + F_{R\_ges\_LE}$$

Input torque through guiding rollers

$$M_{AN\_RR} := F_{R\_ges} \cdot \frac{d}{2}$$

## Total resistance through climbing

$$F_{ST\_ges\_LE} := F_{W\_ST\_II\_LE} + F_{W\_ST\_III\_LE} + F_{W\_ST\_IV\_LA}$$

$$F_{ST\_ges\_LA} := F_{W\_ST\_II\_LA} + F_{W\_ST\_III\_LA} + F_{W\_ST\_IV\_LA}$$

"Sum climbing resistances – Tight site"

$$F_{W\_ST\_ges} := F_{ST\_ges\_LA} - F_{ST\_ges\_LE}$$

"resulting input torque"

$$M_{AN\_ST} := F_{W\_ST\_ges} \cdot \frac{d}{2}$$

"Input torque throug chain bolt friciton"

$$F_{KB\_ges} := F_{KB\_UI} + F_{KB\_U2} + F_{KB\_II\_LE} + F_{KB\_II\_LA}$$

$$M_{AN\_RR\_KB} := F_{KB\_ges} \cdot \frac{d}{2}$$

$$P_{AN\_RR\_KB} := M_{AN\_RR\_KB} \cdot \omega_{AN}$$

## Results

"sum input torque (at gear output)"

$$M_{KB\_ges} := M_{AN\_RR} + M_{AN\_ST} + M_{AN\_RR\_KB}$$

"Power gear output"

$$P_{Getr\_Ausg} := M_{KB\_ges} \cdot \omega_{AN}$$

"Power drive input"

$$P_{M\_Eing} := \frac{P_{Getr\_Ausg}}{\eta_G \cdot \eta_M}$$

"theoretical max. chain force – Tight site"

$$F_{K\_THEO\_LA} := \left( \frac{F_{Vor}}{2} + F_{R\_ges\_LA} + F_{ST\_ges\_LA} + F_{KB\_UI} + F_{KB\_U2} + F_{KB\_II\_LA} \right) \cdot \frac{1}{n_k}$$

"Corrected max. chain force – Tight site"

$$F_{K\_LA} := F_{K\_THEO\_LA} \cdot C_{RP}$$

"theoretical max. chain force – Slack site"

$$F_{K\_THEO\_LE} := \left( \frac{F_{Vor}}{2} + F_{R\_ges\_LE} + F_{ST\_ges\_LE} + F_{KB\_II\_LE} \right) \cdot \frac{1}{n_k}$$

"Corrected max. chain force – Slack site"

$$F_{K\_LE} := F_{K\_THEO\_LE} \cdot C_{RP}$$

## 14.2 Excerpt of the analytic calculation

**Kettenkraftberechnung SZF**

**SYSTEM:** **Eben+Steigung+Erhöhte Ebene**

Aus Dropdown wählen

**Ausgangsdaten**

**Allgemein**

| | | | |
|---|---|---|---|
| Fördergeschwindigkeit | v | 0 | [m/s] |
| Füllgrad Zelle | fg | 0 | [] |
| Volumen Zelle | Vz | 0 | [L] |
| Dichte Schüttgut | ρs | 0 | [kg/L] |
| Ebene Länge I | LI | 0 | [mm] |
| Bogenlänge II | LIIb | 0 | [mm] |
| Bogenlänge IV | LIVb | 0 | [mm] |
| Radius Bogen II | rBII | 0 | [mm] |
| Radius Bogen IV | rBIV | 0 | [mm] |
| Länge Steigung III | LIII | 0 | [mm] |
| Länge erhöhte Ebene | LV | 0 | [mm] |
| Steigungswinkel | α | 0 | [deg] |
| Teilung Laufräder | pL | 0 | [mm] |
| Vorspannkraft Förderer | Fvor | 0 | [N] |

**Kette**

| | | | |
|---|---|---|---|
| Teilung Kette | p | 0 | [mm] |
| Anzahl Kettenstränge | nk | 0 | [] |
| Masse pro Kettenglied | mk_G | 0 | [Kg] |
| Masse Zelle | mZ | 0 | [Kg] |
| Masse Laufrad | mL | 0 | [Kg] |
| Teilkr. Durchm. Kettenrad Antr. | dA | 0 | [mm] |
| Teilkr. Durchm. Kettenrad Uml. | dU | 0 | [mm] |
| Durchm. Kettenbolzen | dB_K | 0 | [mm] |

**Reibungskoeffizienten**

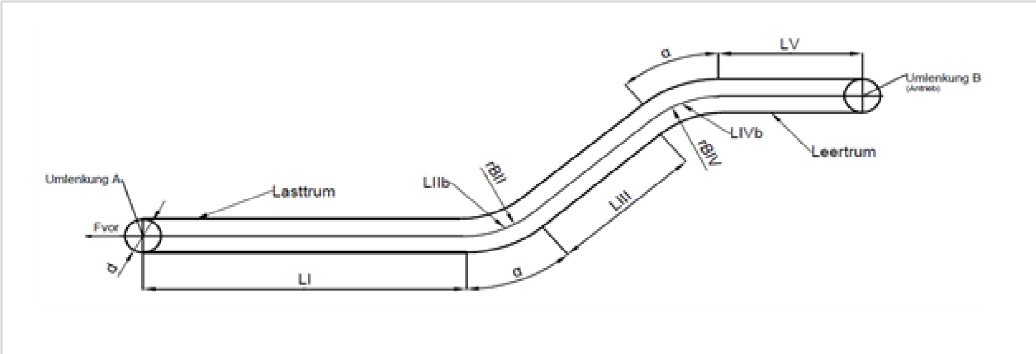
| | | | |
|---|---|---|---|
| Laufrad (Rollreibung) | μL | 0 | [] |
| Kettenbolzen | μKB | 0 | [] |

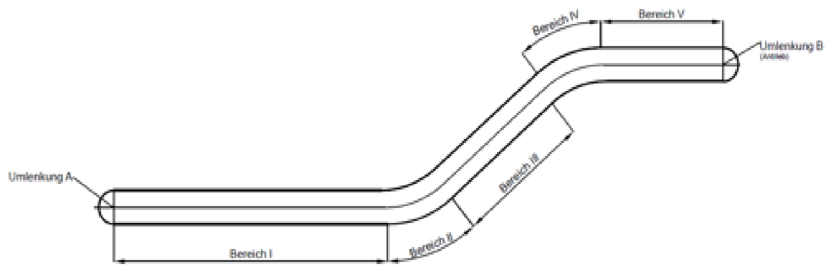
**Wirkungsgrade**

| | | | |
|---|---|---|---|
| Wirkungsgrad Motor | η_M | 0 | [] |
| Wirkungsgrad Getriebe | η_G | 0 | [] |

**Sonstiges**

| | | | |
|---|---|---|---|
| Erdbeschleunigung | g | 0 | [kg*m/s^2] |
| Masse Turas | m_turas | 0 | [Kg] |
| Zusatzleisung | P_zus | 0 | [KW] |



## Berechnete Ergebnisse System: Eben+Steigung+Erhöhte Ebene

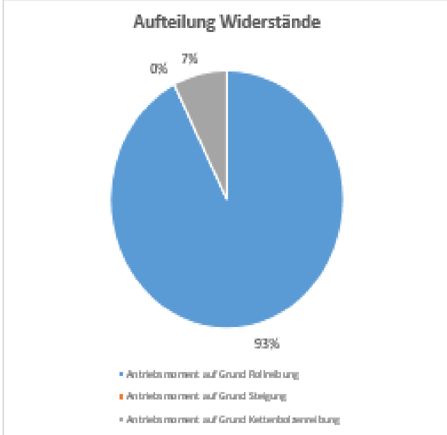
| samtwiderstand durch Rollreibu | 11208.2 [N] |
|---|---|
| **Lasttrum** | |
| Bereich I | 11208.2 [N] |
| Bereich II | 0.0 [N] |
| Bereich III | 0.0 [N] |
| Bereich IV | 0.0 [N] |
| Bereich V | 0.0 [N] |
| Summe Lasttrum | 11208.2 [N] |
| **Leertrum** | |
| Bereich I | 3284.4 [N] |
| Bereich II | 0.0 [N] |
| Bereich III | 0.0 [N] |
| Bereich IV | 0.0 [N] |
| Bereich V | 0.0 [N] |
| Summe Leertrum | 3284.4 [N] |
| Reibung durch Zusatzleistung | 0.0 |

| esamtwiderstand durch Steigun | 0.0 [N] |
|---|---|
| **Lasttrum** | |
| Bereich II | 0.0 [N] |
| Bereich III | 0.0 [N] |
| Bereich IV | 0.0 |
| Summe Lasttrum | 0.0 [N] |
| **Leertrum** | |
| Bereich II | 0.0 [N] |
| Bereich III | 0.0 [N] |
| Bereich IV | 0.0 |
| Summe Leertrum | 0.0 [N] |

| Widerstand Kettenbolzenreibung | 1170.0 [N] |
|---|---|
| **Umlenkung A** | 154.4 [N] |
| **Radius II** | 482.3 [N] |
| Lasttrum | 289.1 [N] |
| Leertrum | 193.2 [N] |
| **Radius IV** | 289.1 [N] |
| Lasttrum | 289.1 [N] |
| Leertrum | 0.0 [N] |
| **Umlenkung B (Antrieb)** | 244.3 [N] |

| Antriebsmoment | |
|---|---|
| Antriebsmoment auf Grund Rollreibung | 5612.0 [Nm] |
| Antriebsmoment auf Grund Steigung | 0.0 [Nm] |
| Antriebsmoment auf Grund Kettenbolzenreibung | 453.1 [Nm] |

| | |
|---|---|
| Gesamt Antriebsmoment | 5787.4 [Nm] |
| Leistung Getriebe Ausgang | 4483.7 [W] |
| Leistung am Motoreingang | 4968.1 [W] |
| maximale Kettenkraft Lasttrum (für einen Strang) | 12303.4 [N] |
| maximale Kettenkraft Leertrum (für eine Stang | 8179.6 [N] |




Für Vergleich mit Messung
**Umfangskraft (für alle Strän ###**

### Kettenkräfte (nach TGL 20-353 006)

| | |
|---|---|
| Vor Umlenkung A (Antrieb, Lasttrum) | 12249 |
| Nach Umlenkung A (Antrieb, Leertrum) | 8335 |
| Einlauf Radius IV (Leertrum) | 0 |
| Einlauf Schräge III (Leertrum) | 0 |
| Einlauf Radius II (Leertrum) | 8142 |
| Einlauf Ebene Länge LI (Leertrum) | 8142 |
| Umlenkung (Leertrum) | 6500 |
| Umlenkung (Lasttrum) | 6577 |
| Einlauf Radius II (Lasttrum) | 12181 |
| Einlauf Schräge III (Lasttrum) | 12181 |
| Einlauf Radius IV (Lasttrum) | 12181 |
| Einlauf erhöhte ebene V(Lasttrum) | 12326 |