UiT The Arctic University of Norway

Department of Engineering Design

# Optimization of Conveyor Roller, LKAB

Manoj Bhusal

Master’s thesis in Engineering Design END-3900 May 2024

# ABSTRACT

The aim of this thesis is to optimize the performance, material, life and accessories for the conveyor rollers installed in longest conveyor section “15101 smst TR010” of LKAB Narvik, which is under the SILO with overall conveyor length of 620 meter. The report outlines the problems created by groundwater falling inside the tunnel, dust and spoilage of iron ore pallets outside the conveyor belt to create a major problem for the carrying side idlers. LKAB, produces iron ore and in Narvik, a big facility is installed in which, heavy duty steel conveyor rollers are used to move the iron pallet by receiving from train to moving towards the shiploader area inside the plant.

The major problem for roller stucking due to above mentioned environmental conditions has been studied and a new concept of idlers are recommended considering all the problems, requirements and constraints. One of the satisfactory result is 13.3 % overall weight reduction in whole conveyor unit. In troughing idler set, two different sized rollers are used namely central roller and side roller, where, central roller is horizontal and side rollers are troughed by 45°. Central roller has designed with 20.38% weight reduction in rotating parts( cylinder, bearings, bearing hub and seals), where as 11.6% total weight is reduced in one single roller. In case of side roller, 26.34% of weight is reduced in rotating parts, likewise, 15.5% total weight is reduced in a single roller. The conveyor rollers are equipped with high performance bearings, but in existing case, bearings are stucking frequently due to dust and water issue, so that bearings with more feature are used in a new design to reduce the repair and maintenance cost. One positive result of weight optimzation is that 6944 watt electricity has been saved. Likewise, the bearing life has been increased to 50900 hour from 15000 hour after selection same sized but extra featured bearings.

Overall, the new concept of rollers-LKAB are generated as a optimization in weight, performance, and life consideration in co-operation with LKAB Narvik and the department of Engineering Design UiT Narvik Campus.

## ACKNOWLEDGEMENT

Firstly, I want to extend my heartfelt thanks to my supervisor Guy Beeri Mauseth and co-supervisor Andrei Karzhou for their unwavering support, guidance, and encouragement during this period. Their invaluable assistance was instrumental in making this thesis possible. I would like to thank department of Engineering Design for allowing me to use necessary resources that let me to complete the literature study and design providing required guidelines, reference materials and the effective ideas to use the resources available in the library. I am also very grateful to Adrian Verhoef, a member of library communications team Uit Narvik, for providing a special training on scientific report writing, literature search and references giving full access of the library resources which, without doubt, was very crucial to my thesis work.

Secondly, my earnest gratitude to LKAB Narvik for offering me this thesis topic in which I was interested and is a topic of my background of mechanical engineering. Special thanks to Daniel Valen Nilsen and Fred Yang, both are Maintenance Engineer at LKAB Narvik, for their invaluable support providing all the necessary technical data and information about conveyor system installed in LKAB’s facility. Moreover, it is highly regarded that they gave me an opportunity to visit the plant for understanding the problem, identifying the research areas, and witnessing all the principles related to heavy duty conveyor system with full access by spending their precious time.

Additionally, I extend my gratitude to my parents for their invaluable guidance and understanding. Completing this dissertation wouldn't have been possible without the unwavering support of my friends.

# Table of Contents

# List of Tables

## List of Figures

# 1 Introduction

Conveyor systems represent the backbone of mordern industrial operations, faciliating the seamless transportation of goods across various stages of production and distribution. From manufacturing to warehousing, conveyors play a pivotal role in enhancing efficiency, reducing labor costs, and ensuring consistent product flow. However, as industries evolve and technology advances, the design and optimization of conveyor systems undergo continuous refinement to meet the growing demands of efficiency, flexibility, and sustainability.The use of conveyor systems in the field of engineering industries, manufacturing plants, extraction and processing units has been improving day by day with the use of more and more optimized techniques all over the world. It has been a effective and efficient way to move the products, raw materials, goods, construction materials, ores and all kind of items from one place to another either to process further or to finishing up the production process which leads the final products towards shipping unit in any kind of industries. The reasons behind using the conveyor systems in product handling are in huge numbers such as to lower the material handling cost, to save the time, increase the production rate, to save the workers from possible accidents, etc. Conveyors are one of the most flexible types of industrial equipment which is mostly installed with an aim to improve the productivity, efficiency and safety in the industry. These are also used in order to transport bulk materials too, for instance, gravel or iron ore [1].

Among various kinds of product moving techniques conveyors are the more famous one which are also in different types on the basis of types of material used, environmental conditions, boundary conditions, loading conditions and the outlay of the space inside the processing unit in all type of industries. The highly used conveyor types are belts, buckets, chains and rollers. And more specifically the common conveyor types are[1], belt conveyor, gravity roller conveyor, chain driven live roller, motorized roller conveyor, live roller conveyor, power and free conveyor, chain conveyor, chain conveyor, over under conveyor and slat conveyor.

To enter into the actual topic information it is highly neccessary to talk about LKAB who provided this topic and precisely LKAB is an international group that sells sustainable iron ore, minerals and special products. LKAB is leading the green transformation of the iron and steel industry by developing carbon free processes and products like sponge iron. The corporate history of LKAB was started in 1890, however the story began more than 200 years earlier. It was the ignition of what eventually became an international mining and minerals group. LKAB has the mine in Kiruna, Sweden and the track is connected by rail up to Narvik port where the train carries the ore from Kiruna to Narvik port, after doing some pre-extraction, ore is loaded into the ship for different countries. In Narvik many mechanical and industrial equipments are installed to receive the iron ore from train and to load it into the ship, among these, transport conveyor rollers are one of the most critical and important parts which are used to move the ore towards the loading area. The problem here is rollers are damaging frequently which has been creating a problem in smooth operation of plant since long. So in this thesis part the reason of rollers failure will be studied and also the optimized design of these parts will be presented with all the design steps, methods and tools used.

## 1.1 Background and overview

According to H.W. Guderjahn, the first recorded use of a conveyor system dates back to ancient times, around 4000 years ago, in ancient Egypt. These early conveyors were simple systems consisting of wooden belts or ropes used to transport goods and materials such as stones and construction materials for the building of the great pyramids. Historical evidence suggests that such systems were employed to ease the manual labor involved in moving heavy loads over long distances. The concept of conveyor systems has since evolved significantly, with advancements in technology leading to the development of more sophisticated and efficient designs[2].

Transport conveyor systems have played a pivotal role in revolutionizing industrial processes, providing efficient movement of goods within factories and warehouses. The inception of conveyor systems can be traced back to the late 18th century when primitive conveyor belts were utilized in mining operations to transport coal and ore. However, it wasn't until the early 20th century that conveyor technology truly burgeoned, with the introduction of powered belt conveyors by Richard Sutcliff in 1905, followed by Henry Ford's assembly line innovations incorporating conveyors for automobile manufacturing in 1913. These advancements laid the foundation for the widespread adoption of conveyor systems across various industries, leading to increased productivity and cost-effectiveness[3].

The evolution of conveyor technology continued throughout the 20th century, with developments such as roller conveyors, overhead conveyors, and automated sorting systems. Major advancements in materials and engineering further enhanced conveyor efficiency and durability. Today, conveyor systems are indispensable in industries ranging from manufacturing and logistics to food processing and packaging. They faciliate seamless material handling, streamline production processes, and optimize workflow, contributing significantly to the global economy[4].

## 1.2 Scope

the optimization of conveyor roller is expansive and critical for industries reliant on robust material handling solutions. Optimization encompasses various aspects including material selection, weight reduction, structural design, and operational efficiency. Firstly, choosing high-strength materials such as low carbon steels or some composite polymers ensure durability and longevity under heavy loads. Structural design optimization involves enhancing load-bearing capacity while minimizing weight, reducing stress concentrations, and improving resistance to wear and tear. Moreover, weight optimized design of rollers makes ease on assemble and disassemble during maintenance as well as plays a vital role to decrease the power requirement to run the conveyor system[5]. In heavy duty application for example in mining industry like LKAB, the environmental conditions effect the performance of roller by decreasing the life and also can cause the damage of bearings. Likewise, conveyor rollers are mainly sealed with specially designed labyrinth seals from both ends to protect the bearing and inner surface of roller from dust, stone and water, but in harsh conditions seals cannot perform well and the outide dirt can rust the bearing which can loss huge amount of money and time to replace. The roller in LKAB Narvik has been facing same problem of dust, stone and water which is promoting roller to stuck. In this thesis part an extra solution to stop the access of dirt will be purposed with optimized design of whole conveyor roller. The main optimization steps followed in this thesis are,

- Weight reduction
- Reduction of shaft deflection to prevent bearing failure.
- Material selection for optimum stiffness requirement.
- Selection of the bearing and sealing elements.
- Finite element analysis of roller components to study stress distribution and deformation.
- Comparison of power and belt pull requirement with existing system in LKAB.

# 2 Literature survey

## 2.1 Basic conveyor mechanism

The fundamental concept of conveyors, which serve to transport materials across various industries such as mining, food, automotive, manufacturing, and agriculture is showing in figure 1 below. Conveyors consist of three primary components: a moving loop (chain, belt, wire), a support system (rollers/idlers), and a pulley system. The moving loop carries materials along a predetermined path, facilitated by the support system ensuring smooth movement. The pulley system maintains tension in the loop and drives it using energy. Figure 1 provides a clear illustration of a basic belt conveyor, where a looped belt is tensioned between two pulleys, one connected to a drive motor. As the motor runs, it drives the belt, transporting goods placed on it. While this example is simplified, industrial conveyor systems are highly mechanized and optimized with numerous accessories.[6]

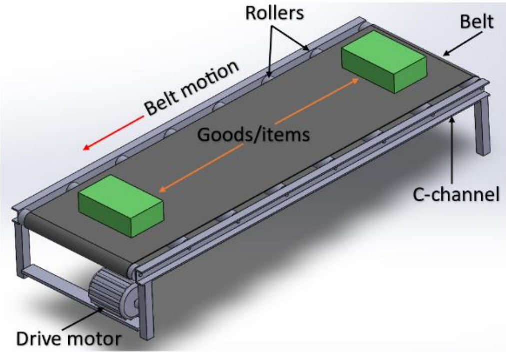



*Figure 1: Common conveyor belt system with rollers, powered* [6]

## 2.2 Conveyor rollers/ idlers

Conveyor rollers, also known as idlers, are essential components in belt conveyor systems that play a crucial role in supporting and guiding the conveyor belt. They are cylindrical rollers that rotate on bearings and are strategically placed along the length of the conveyor belt to ensure smooth and efficient material handling. Basically, rollers/idlers are mounted perpendicular to the direction of conveyor belt's travel. Their peimary purpose is to support the weight of the conveyor belt and the material being transported, preventing sagging and ensuring proper belt alignment and tracking [7].

## 2.3 Construction of idlers

As mentioned in earlier section that conveyor idlers are cylindrical in shape and can rotate about the longitudional axis with the help of bearings placed on both ends and supported by a stiff shaft. Conveyor rollers typically consists of the following components.

**A. Roller tube:** The roller tube is the cylindrical body or the roller, which is typically made of steel such as 1.0037 S235 JR structural steel, composite materials, polymer materials etc for durability and strength. [8].

**B. Roller shaft:** The roller shaft is the central axle that runs through the roller tube and is supported by bearings. It is often constructed from high-strength steel such as 1.0037 S235JR structural steel to withstand the loads and stresses of the conveyor system[8].

**C. Bearings:** conveyor rollers are equippped with either single or double-row deep groove ball bearings, which facilitates smooth rotation and reduce friction. Common bearing option include 2RZ (sealed on one side) and 2Z (sealed on both sides), with C3 clearance for enhanced efficiency[8].

**D. Seals:** To prevent the ingress of dust, debris, and contaminants, conveyor rollers are fitted with sealing systems. These typically include a grease-retaining inner seal with a multi-stage labyrinth design and a retention cap**[8].**

**E. Lubrication:** Conveyor rollers are lubricated with lithium soap-based grease containing rust inhibitors to ensure smooth operation and extend the service life of the bearings[8].

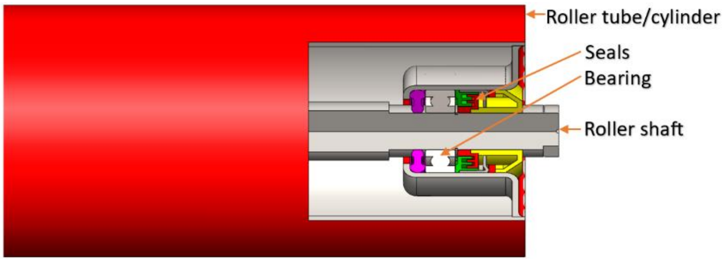


*Figure 2: Major components in a conveyor roller*[8]

## 2.4 Idler types:

There are various types of conveyor rollers, each designed for specific applications and conveyor system requirements. Some common types include[7].

**A. Carrying idlers:** These idlers support the weight of the conveyor belt and the material being transported.

**B. Return rollers:** These rollers support the underside of the conveyor belt on the return path, preventing sagging and ensuring proper belt tracking.

**C. Impact idlers:** Impact idlers are designed to absorb the impact of materials falling on to the conveyor belt, protecting the belt and other components from damage.

**D. Troughing idlers:** Troughing idlers are used to create a trough shape in the conveyor belt, which helps contain and guide the material being transported. Basically troughing idlers are used in heavy duty applications such as, mining, construction and bulk material handling.

**E. Flat rollers:** These rollers are used in applications where the conveyor belt needs to remain flat, such as in sorting or packaging operations.

## 2.5 Schematic overview of full conveyor belt unit

Figure 3 below illustrates a detail system of belt conveyor, as described in earlier sections about the components and types of idlers we can see most of the parts in a single unit. To describe the working of this belt sonveyor unit, the is the belt loop tensioned by two pulleys, head or drive pulley and tail pulley. The head pulley is powered and drives to the belt to move in its predetermined loop. There are troughing (carrying) idlers which guide the belt and allows to move smoothly. Belt is guided by return idlers and maintained proper tension by using a couter weightbelow the take up pulley. The material dropped from load chute moves towards dribble chute through belt and the whole process continues[9].

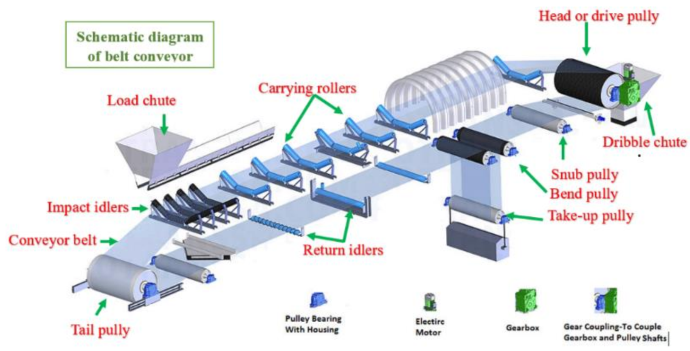


*Figure 3: Schematic overview of full conveyor belt unit*[9]

## 2.6 Role in optimizing roller performance

Optimizing conveyor roller performance is essential for efficient material handling operations in industries such as manufacturing, distribution, and logistics. Coveyor rollers play a critical role in the smooth and reliable movement of goods along the conveyor ssytem. Figure 4 shows that, decreasing the mass of roller maintaining proper safety factor for  stress and failure criterias can help to increase the energy efficiency, likewise selection of maaterial, bearing and selas also promote the performance. Some design ideas to consider the environmental effect on the roller can also help to prevent failure of bearings and roller stuck.

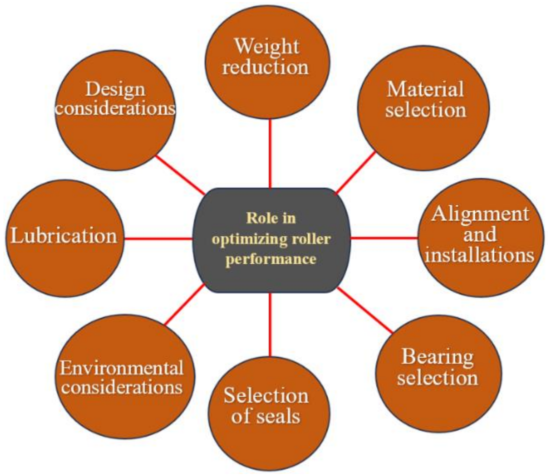



*Figure 4: Roles in optimizing roler performance*[10]

## 2.7 Description of rollers/idlers-LKAB

LKAB Narvik has all together 15 KM long conveyor system including all zones, SILO area, screening unit, crushing unit, storage unit and shiploader area. In a single unit of conveyor it consists exactly same components and works in the same principle discussed in section 2.4 and also shown in figure 3. Our main goal is to optimize the troughing idlers in SILO area which is also known as "15101 smst TR010" section. It has a single horizontal belt conveyor unit of length 620 meter and 516 trough carrying idler set. Each idler set has one central roller (Ø194N) and two side idlers (Ø159N). The side rollers are 45° inclined where the central roller is horizontal. A belt of 2000 mm width and 17 mm thickness has been used which conveys the iron ore starting from SILA to the shiploader area. A schematic drawing of longest conveyor in LKAB or "15101 smst TR010" section is shown in the figure below[11].

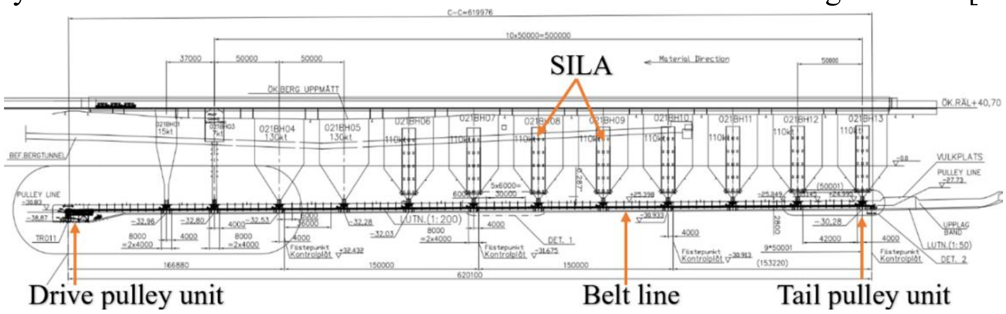



*Figure 5: Schematic drawing (15101 smst TR010) under the SILO-LKAB*[11]

### 2.7.1 Conveyor system practice

LKAB's plant in Narvik has various types of roller assemblies as mentioned in earlier section in different zones, and the overall conveyor system is of around 15 KM starting from receiving pallet from SILO to the ship loading area. We can be clear from the graphical representation of main sections used in LKAB in figure 5 below. First the pallets from mine site (KIRUNA SWEDEN) are carried througn train and store in SILA (12 SILOs). From SILO, pallets are dropped in conveyor for screening area, after seperating good sized and oversized pallet, good size pallet goes to shiploader unit and oversized pallet goes towards crushing unit then goes to its separate storage unit. Finally the crushed oversized pallet is also goes towards shiploader unit. Here all these units are connected each other by conveyor belt, of total length 15 KM as mentioned before [11].

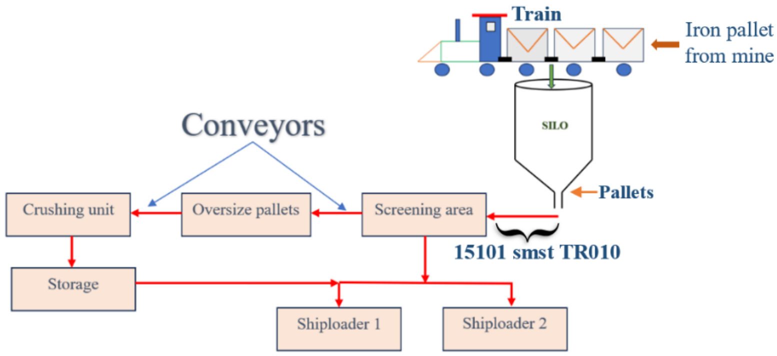



*Figure 6: Schematic diagram of different sections in LKAB Narvik*[11]

### 2.7.2 Type and alignment of rollers

In LKAB Narvik, different types of rollers are installed in each conveyor belt unit, and on the basis of alignment of rollers there are various kinds of roller assembles. Following figure shows the type and assembly system used in LKAB Narvik.

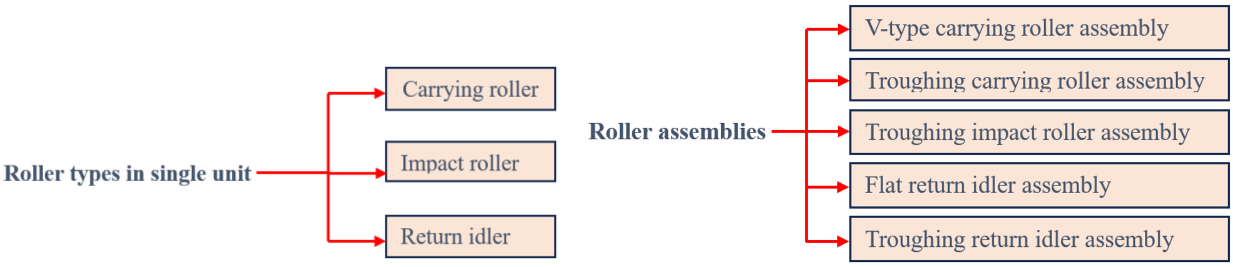



*Figure 7: LKAB Narvik- roller types and assemblies*[11]

The most common and that one where roller is damaging and stucking frequently is troughing carrying roller assembly in "15101 smst TR010" section which happens mostly under the silo, where the pallet drops on the belt from SILO. Here in this thesis, all the analysis will be on troughing carrying roller asembly of "15101 smst TR010" section with its redesign and simulation condisering all the needed engineering design requirements.

### 2.7.3 Identification of the problem and solution requirements

The conveyor system (mainly in 15101 smst TR010 section) in LKAB, there are some issues with the idlers because of some environmental conditions, which has been creating a major cause to bearing failure and roller stuck, as a result the conveyor belt wears frequently and needs huge money and time to replace the idler. They are looking for any alternative solution to prevent the idler from stucking either by optimizing the roller design or by designing any external mounting features to prevent the rollers from outside contaminations. In this thesis part an optimized roller design will be presented and some alternative solution to prevent from outside contaminations will also be prepared considering all the engineering requirements. From a visit to the plant, we have obsorbed main three issues or problems which are discussed below.

### A. Ground water problem

As we mentioned in earlier section that LKAB has 12 SILOs where iron ore(pallet) is stored, the SILOs are build underground and the conveyor under the silo has set along a big tunnel as shown in figure below. It means conveyor belt is inside a tunnel and there is a major problem due to falling ground water, which is creating big problem for the rollers. The ground water directly drops on the belt and mixes with pallet, finally it goes to screening unit through coveyor belt and creates mud in huge amount and falls on the floor. The mud deposits on the bearing side of the roller and enters inside the bearing to make rusting.

*Figure 8: Ground water leakage on the tunnel floor*-LKAB[11]

### B. Pallet droping problem

Under the SILO, while droping the pallet on the conveyor belt some pieces of pallet go outside the belt and drop on the floor, also the pallet goes on the space between roller and bracket near the bearing area. Which helps to stop the roller and it requires best solution to make safe the roller from these dropped pallet particles. Normally in product loading zones, skirtboards are used to prevent the falling of material outside belt but it can not stop all the falling pallet and the pieces of iron ore insetrs in the gap between roller and bracket.


*Figure 9: Dropped pallets fron conveyor on the ground*[11]

### C. Dust deposition on roller

As mentioned in section a) ground water creates mud and it finally converts on dust and deposits all alound the roller, as well as it goes inside the bearing through seal material and helps to stuck the roller. Once any roller is stucked, it starts to wear the conveyor belt and finally makes big waste of money for maintenance and disturbance for the operation of plant. The condition of roller after depositing dust can be seen in the picture below.

*Figure 10: Dust deposited on conveyor roller-LKAB*[11]

# 3 Methodology

The aim of this master's thesis is to optimize the troughing idlers on the carrying side of longest conveyor (620 meter) installed at LKAB Narvik. The project is being conducted in collaboration with LKAB and the Department of Engineering Design at Uit Narvik. As an engineering product design thesis, it should follow the product design process as directed by Uit. Given the connection with LKAB, international standards, guidelines, accessory selection, design criteria, and constraints must be considered effectively. For instance, LKAB uses DIN (Deutsches Institut für Normung) standards for conveyor equipment [11], which are manufactured following the standards and guidelines provided by CEMA(Conveyor Equipment Manufacturers Association). Due to certain constraints, it is not feasible to redesign all conveyor accessories, such as the belt, roller bracket, drive unit, scrapers, impact idlers etc. Therefore, this thesis will focus soley on optimizing the roller components and will adhere to the product design process outlined by UiT and consider the relevant international standards, particularly DIN and CEMA.

## 3.1 Case study of the conveyor roller optimization

Engineers focus on optimizing roller weight to enhance efficiency and reduce power consumption, especially when loads are below capacity. Standard rollers typically maintain consistent dimensions and materials, but advancements in technology allow for lighter products. A notable study by Prasad M Patil and Gajanan C Koli in October 2020 explored weight optimization of support rollers using theoretical, finite element, and experimental analyses [12].

About research paper:

**Topic: "Weight optimization of support roller by using theoretical, finite element and experimental analysis"**

**Author:** Prasad M Patil, Gajanan C Koli

**Published journal:** "International journal of science and technology research volume-9"

**Published date:** october 2020

**Objectives, methodology and findings:**

i. Changing roller dimensions, but using the same materials as it is.

ii. By keeping same geometrical dimensions and modifying material of components.

iii. Changing both material and parts dimensions.

| Analysis type | Material properties | | | stress (Mpa) | Deflection(mm) | wt.(kg) | SF |
|---|---|---|---|---|---|---|---|
| Existing roller parameters | Material | | Mild steel | 6.68 | 0.0575 | 83.20 | 29.5 |
| | Density | | 7860 | | | | |
| | Young's modulus(kg/m$^3$) | | $2.10 \times 10^5$ | | | | |
| | Yield strength(Mpa) | | 370 | | | | |
| | Ultimate strength(Mpa) | | 590 | | | | |
| | Poisson's ratio | | 0.3 | | | | |
| Results after changing dimensions | Sr no. | OD (mm) | ID (mm) | 38.53 | 0.414 | 17.0 | 5.10 |
| | 2 | 60 | 59 | 75.16 | 0.807 | 8.6 | 2.62 |
| | 3 | 76.1 | 74.1 | 23.70 | 0.200 | 21.7 | 8.30 |
| Changing material | ASTM A 554 | 76 | 68 | 6.695 | 0.061 | 87 | 25 |
| | IS 4923 | 76 | 68 | 6.695 | 0.056 | 84.36 | 22 |
| Changing material and dimensions | ASTM A 554 | 76.1 | 74.1 | 23.70 | 0.216272 | 22.6 | |
| | | 76.1 | 72 | 12.05 | 0.109985 | 45.8 | |
| | IS 4923 | 76.1 | 74.1 | 23.70 | 0.210865 | 22.0 | |
| | | 76.1 | 72 | 12.05 | 0.107236 | 44.5 | |
| Selection | OD (76.1 mm), ID (74.1 mm), material: mild steel | | | | | | |

*Table 1: Data obtained by M Patil and C Koli to optimize weight for conveyor roller*[12]

## 3.2 Standardization and guidelines

CEMA, or the Conveyor Equipment Manufacturers Association, is an organization dedicated to establishing standards, technical publications, and engineering guidelines for conveyors across industries like mining, construction, and manufacturing. Its goal is to ensure safety, efficiency, and consistency in conveyor design, operation, and maintenance. Members of CEMA include conveyor equipment manufacturers, designers, suppliers, engineering firms, and end users. One prominent conveyor parts manufacturer adhering to CEMA standards is RULMECA, an Italian company producing conveyor rollers of various sizes, standards, and series. RULMECA aligns with CEMA and ISO standards for standard rollers. These rollers, tailored for the European region, comply with DIN 15207 and ISO 1537 standards, offering a range of series based on materials used, application areas, load ratings, capacities, and sizes. Below are the list of series of RULMECA's rollers[13].

1. Rollers in steel series- PSV
2. Rollers in plastic series- PL
3. Rollers in steel series – MPS
4. Rollers in thermoplastic polymer series – TOP
5. Rollers in steel series – RTL

Among all the above series, PSV rollers are quite famous in heavy duty applications. PSV rollers excel in demanding conveyor environments characterized by heavy loads and large material sizes, requiring minimal upkeep. They find ideal application in industries such as, mining, cement production, coal-fired power generation, and dock operations. Their sealing system effectively addresses challenges posed by dust, dirt, water, and extreme temperatures, making them suitable for harsh conditions where temperature fluctuations are significant. Standard components are rated for temperature ranging from -20℃ to +100℃, but specialized grease, bearings, and seals can extend this range as needed[13].

**Nomenclature of the PSV series**

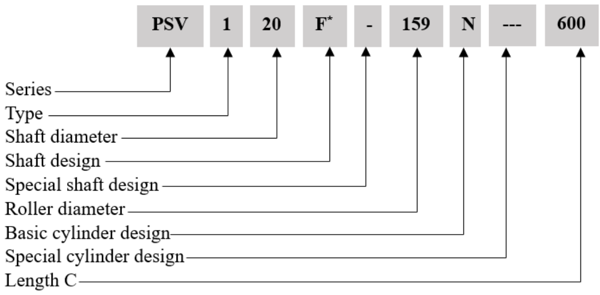
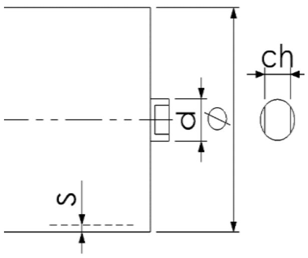



*Figure 11: Nomenclature of PSV rollers develpoed by RULMECA*[13]

The abbreviation N refers to cylinder material and is different for different materials, in PSV series there are seven different sub classes and each sub class has a set of rollers according to dimension, material and bearing used. Below are the list of all basic abbreviations for cylinder on the basis of material used.

| Basic abbreviation | Description | Note |
|---|---|---|
| **N** | Steel S235JR(EN10027-1), St37(DIN 17100) | Standard |
| **I** | Stainless steel AISI 304 | Optional |
| **PE** | HDPE high density polyethylene-black color | Standard |
| **V** | Rigid PVC-color grey-RAL 7011 | Standard |
| **S** | Spiral metal cage | Standard |
| **J** | Electrolyte zinc-color grey-10 micron thickness | Standard |

*Figure 12: Basic abbreviations for PSV rollers developed by RULMECA*[13]




| Roller type | Ø mm | Basic design | S | Shaft D | Shaft ch | Bearing |
|---|---|---|---|---|---|---|
| PSV -1 | 63 | N | 3 | 20 | 14 | 6204 |
| | 89 | N | 3 | | | |
| | 108 | N | 3.5 | | | |
| | 133 | N | 4 | | | |
| PSV -2 | 89 | N | 3 | 25 | 18 | 6205 |
| | 108 | N | 3.5 | | | |
| | 133 | N | 4 | | | |
| | 159 | N | 4.5 | | | |
| PSV -3 | 89 | N | 3 | 25 | 18 | 6305 |
| | 108 | N | 3.5 | | | |
| | 133 | N | 4 | | | |
| | 159 | N | 4.5 | | | |
| PSV -4 | 89 | N | 3 | 30 | 22 | 6206 |
| | 108 | N | 3.5 | | | |
| | 133 | N | 4 | | | |
| | 159 | N | 4.5 | | | |
| PSV -5 | 89 | N | 3 | 30 | 22 | 6306 |
| | 108 | N | 3.5 | | | |
| | 133 | N | 4 | | | |
| | 159 | N | 4.5 | | | |
| PSV -7 | 108 | N | 4 | 40 | 32 | 6308 |
| | 133 | N | 4 | | | |
| | 159 | N | 4.5 | | | |
| | 194 | N | 6.3 | | | |
| | 219 | N | 6.3 | | | |

*Table 2: Different dimensions and series for PSV rollers developed by RULMECA*[13]

*Note: The cylinder and shaft in steel S235JR (EN 10027-1), exFe360(EN10025), St37(DIN 17100), the highlighted numbers in above tables are the rollers used in LKAB.*

# 4 Technical information of idlers-LKAB

## 4.1 Specification of the conveyor system

In this thesis we mostly focus on troughed carrying rollers in the section "15101 smst TR010", their failure mechanisms, optimization techniques and solution for the problem that LKAB has been facing since long which will be discussed in this paper in later sections. The overall specification of LKAB's facility in Narvik which we are going to use for optimization is listed below. All the data was provided by LKAB Narvik.

| No | Item | Parameters | | Measurement |
|---|---|---|---|---|
| 1 | SILA | Number of iron ore SILOs | | 12 |
| | | Total storage capacity | | 1.5 Mega ton |
| | | Total capacity per year | | 30 Mega ton/year |
| | | Loading capacity for pallers per hour | | 9000 Ton/hour |
| | | Loading capacity for fines per hour | | 11000 Ton/hour |
| | | Depth of each SILO | | 60 m |
| | | Width of each SILO | | 32 m |
| 2 | Conveyor belt | Number of conveyor belt unit | | 70 |
| | | Total length | | 15 KM |
| | | Length of longest conveyor unit | | 620m |
| | | Belt thickness | | 17 mm |
| | | Belt width | | 2000 mm |
| | | Speed | | 2.9 m/s |
| | | Load carrying capacity | | 9000 Ton/hr |
| 3 | Roller | Parameters | Side idler - **ϕ159N** | Center idler - **ϕ194N** |
| | | Roller type | PSV/5-FHD | PSV/7-FHD |
| | | Material | SteelS235JR | Steel S235JR |
| | | Alignment | Troughed 45° | Horizontal |
| | | Length | 600 mm | 950 mm |
| | | Diameter | 159 mm | 194 mm |
| | | Wall thickness | 4.5 mm | 6.3 mm |
| | | Weight | 16.7 kg (total) | 43.9kg (total) |
| 4 | Bearing | Type | Deep groove ball | Deep groove ball |
| | | Manufacturer | SKF international | SKF international |
| | | Part no | 6306 | 6308 |
| | | Dimension | 30×70×19 | 40×90×23 |
| | | Bore type | Cylindrical | Cylindrical |
| | | Basic dynamic load rating | 29.6 KN | 42.3 KN |

| | | | | |
|---|---|---|---|---|
| | | Basic static load rating | 16 KN | 24KN |
| | | Material | Bearing steel | Bearing steel |
| | | Net weight | 340 gm | 630 gm |
| | | Reference speed | 20000 RPM | 17000 RPM |
| | | Limitting speed | 13000 RPM | 11000 RPM |
| 5 | Roller shaft | Diameter | 30 mm | 40 mm |
| | | Length | 632 mm | 982 mm |
| | | Material | Steel S235JR | Steel S235JR |
| | | Material standard | EN10027-1 | EN10027-1 |
| 6 | Iron ore | Maximum size | | 20 mm |
| | | Minimum size | | 1 mm |
| | | Density | | 3 Ton/m$^3$ |

*Table 3: Full specification of conveyor parts installed in LKAB Narvik facility*[14]

## 4.2 Schematic drawing of rollers-LKAB

### A. Side roller-Ø159 N

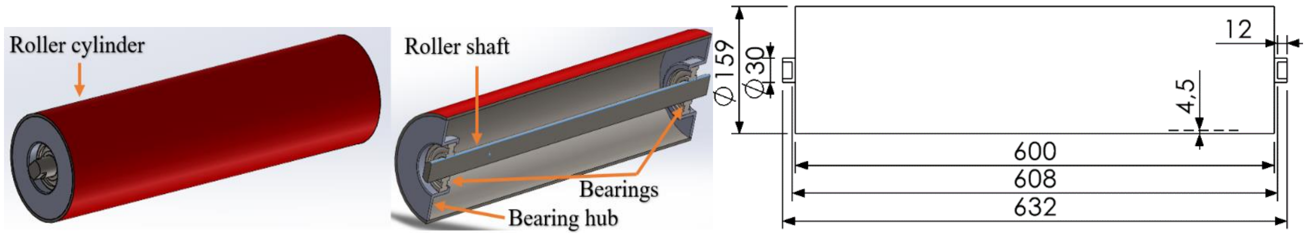



*Figure 13: 3D model of existing side roller with section view and 2D drawing*[14]

### B. Side roller shaft-for Ø159 N

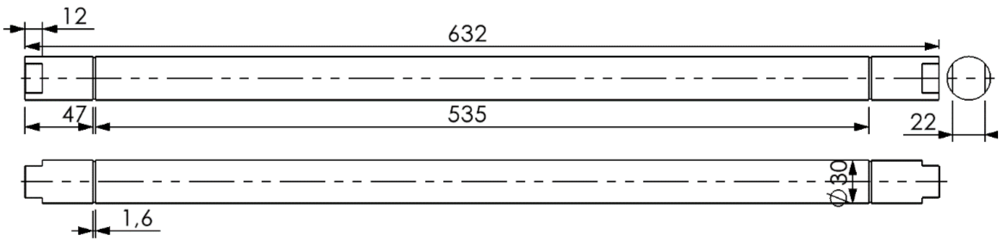
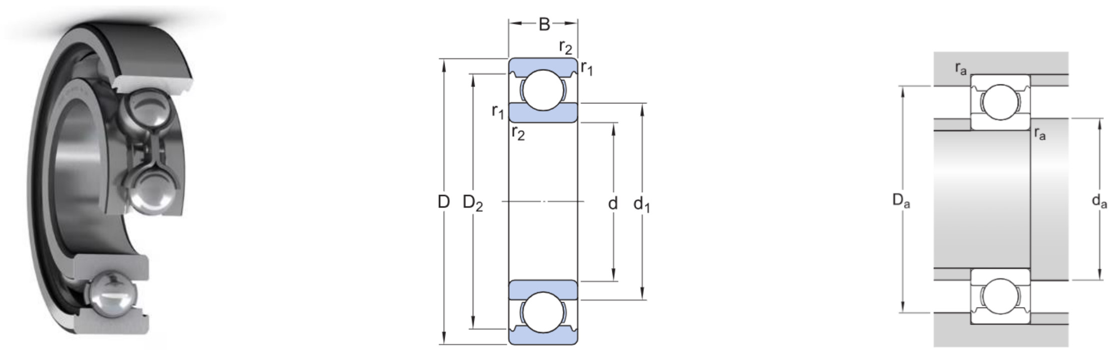



*Figure 14: 2D drawing of roller shaft for side roller-LKAB with dimension*[14]

### C. Bearing-for Ø159 N & Ø194 N




**(a)** **(b)** **(c)**

*Figure 15: (a) 3D view of Roller bearing (b)&(c) section view with different dimensions* [15]

| Dimensions(mm) | | For Ø159 N | For Ø194 N | Abutment dimensions(mm) | | | |
|---|---|---|---|---|---|---|---|
| | | SKF 6306 | SKF 6308 | SKF → | | 6306 | 6308 |
| d | Bore diameter | 30 | 40 | $d_a$ | Diameter of shaft abutment | min.37 | Min.49 |
| D | Outside diameter | 72 | 90 | | | | |
| B | Width | 19 | 23 | $D_a$ | Diameter of housing abutment | max.65 | Max.81 |
| $d_1$ | Shoulder diameter | 44.6 | 56 11 | | | | |
| $D_2$ | Recess diameter | 61.88 | 77.7 | $r_a$ | Radius of shaft/housing fillet | max.1 | Max.1.5 |
| $r_{1,2}$ | Chamfer dimension | min.1 | Min 1.5 | | | | |

*Table 4: All the dimensions of bearing SKF 6306 & 3608 used in Rollers -LKAB*[15]

### D. Central roller-Ø194 N

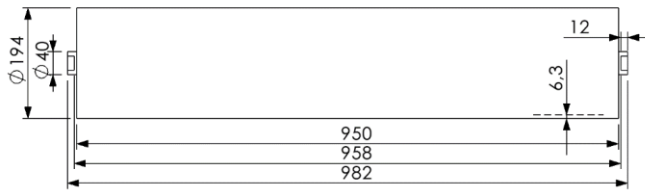



*Figure 16: 2D drawing of existing central roller-LKAB*[14]

### E. Central roller shaft-for Ø194 N

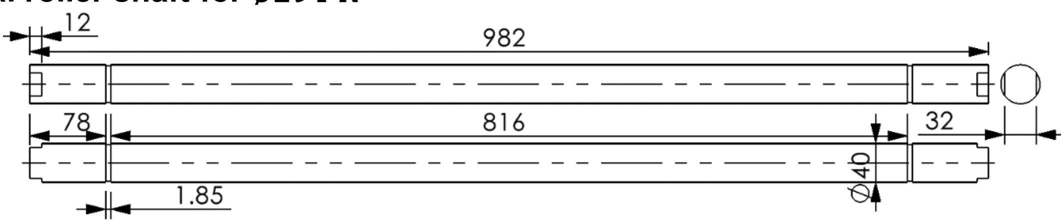



*Figure 17: 2D drawing of existing central roller shaft-LKAB*[14]

### F. Load carrying configuration with full assembly

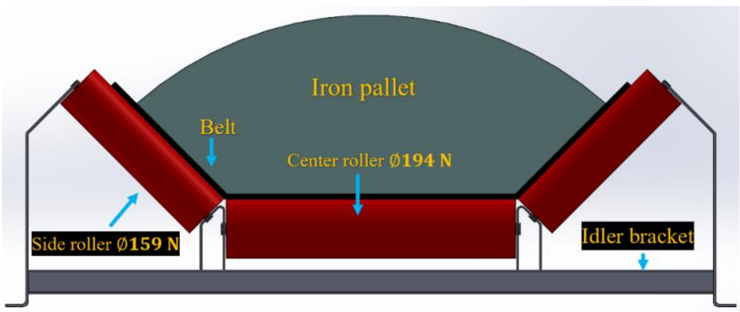



*Figure 18: Load carrying configuration for troughed roller assembly-LKAB*[14]

## 4.3 Loading and boundary conditions

### A. Load distribution by material flow

According to data provided by LKAB the calculation for load distribution on the roller is as below.

Distance between each idlers = 1200 mm

Belt speed (v) = 2.9m/s ............................(a)

Mass flow rate(Q) = 9000 Ton/h = 2500 kg/s .........................(b)

From (a) and (b) we can find that, in 1 sec belt moves by 2.9 m, also 2500 kg material moves with belt, which means 2500 kg material distributes over 2.9m length of belt section.

The total mass of material between two idlers = $1200 \times \frac{2500}{2900}$ ~ **1035 kg = 10153N**

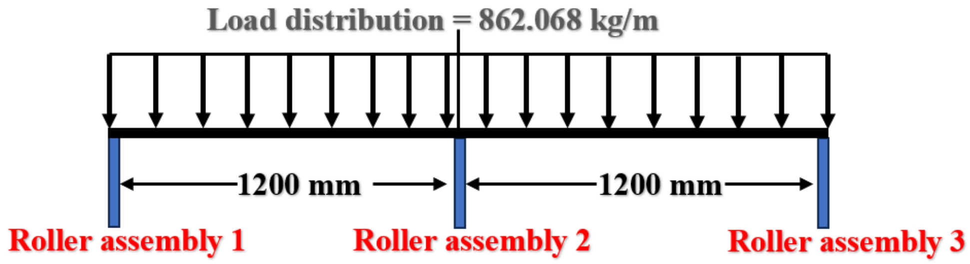



*Figure 19: Load distribution in three consequtive roller assmblies*

As shown in above picture, every roller assembly bears the 1035 kg of static load from material which is the maximum possible load in each assembly in LKAB's conveyor facility. In each roller assembly, one is horizontal and two are troughed by 45°, so the load in horizontal one and troughed roller is different which will be calculated here.

### B. Final load rating with belt weight

Selection of the bulk handling idlers is based on the idler load, generally on the center idler of a 3-roll set. Also, there is some weight contributed by belt itself and it needs to be considered while calculating the overall load on the roller. The belt weight has been calculated in section **4.3 C.**

$I_V(Q)$ = Conveyor capacity = 9000t/h

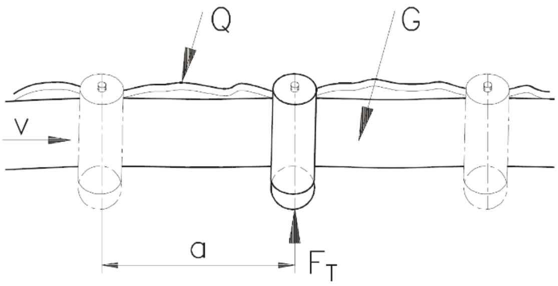



*Figure 20: Three consequtive set of 3 idlers showing total load on a single set of idler*[16]

V = Belt speed = 2.9 m/s
G = Belt weight = 34 kg/m (Table 1 appendix A) (estimation)
$F_T$ = Total load of one idler assembly(N)
$F_Q$ = Total material load on the center roller (N)
$F_R$ = Normal radial load for side rollers
$F_P$ = Participation factor of roller
under greatest stress = 0.72(Table 5 appendix A)
$F_d$ = Impact factor = 1 (Table 6 appendix A)
$F_s$ = Service factor = 1.1 (Table 7 appendix A)
$F_m$ = Environment factor = 0.9 (Table 8 appendix A)
B = Belt width = 2000 mm
L = Central roller length = 950 mm
D = Side idler diameter = 159 mm
d = Side idler shaft diameter = 30 mm
$\alpha$ = Troughing angle = 45°
$\beta$ = Surcharge angle of material in motion for iron ore= 20°[17]
Max. Particle size = 20mm

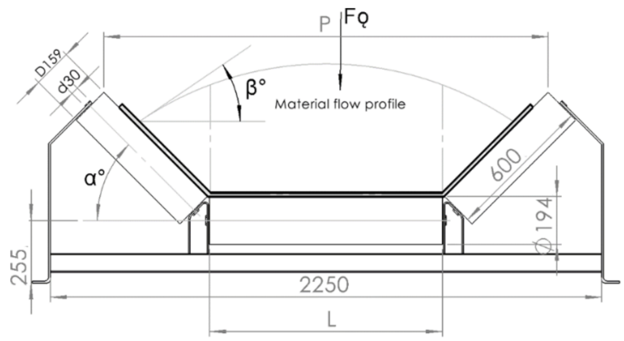



*Figure 21: Loading profile with trough and surcharge angle for a center idler***[16]**

### C. Load calculation with belt weight

G(2000) = 34 kg/m (belt weight per unit length)

$F_T = a_o \times \left(G + \frac{I_V}{3.6 \times V}\right) \times 0.981$ daN= **10556 N** (Load for one idler set) [18]

$F_Q = F_T \times F_p \times F_d \times F_s \times F_m$ = **7524 N[18]**

Vertical load on side rollers $(F_v) = \frac{10556-7524}{2}$ = **1516N**

For material 1.0037 S235 JR structural steel, density of material = 7800kg/m$^3$

Weight estimation for central roller cylinder = 251 N

Weight estimation for bearing hubs = 2.185 Kg = 21.4N

Total load on central roller shaft = 7524N+251N+21.4N = **7796 N**

The load on the central roller cylinder is a uniformly distributed load of 7920N/m, which we can simplify to a point load at the center. This adjustment accounts for the bending load the cylinder bears. We'll apply the uniformly distributed load on the upper surface during simulation to ensure accurate results.

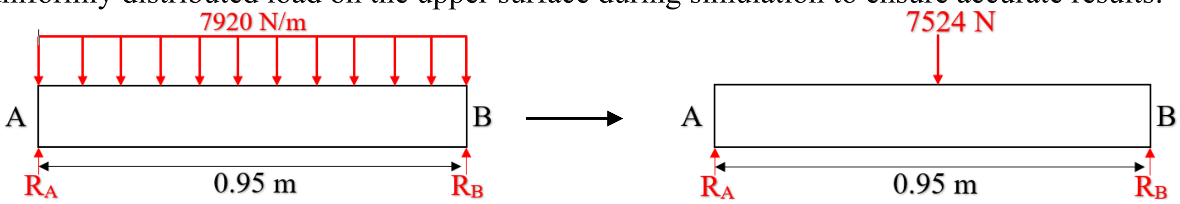



*Figure 22: Free body diagram of central roller with load distribution*

*Note: weight of bearings and sealing elements are neglected*

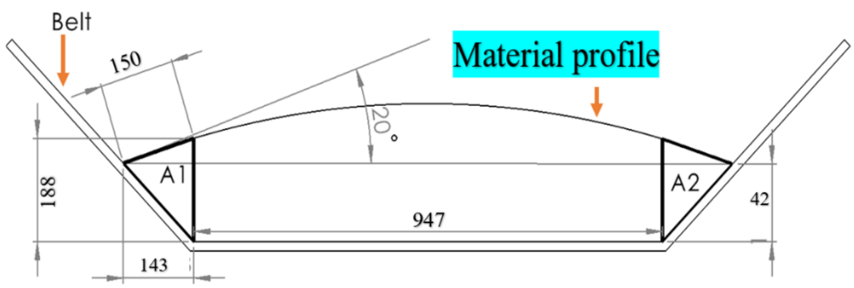



*Figure 23: Material load profile on side rollers*

The figure depicts sections A1 and A2, representing the material profiles on the side rollers. The load from these areas applies vertically on the side rollers, assumed from the center of mass of A1 and A2. In triangle ABC, the center of mass is 110 mm from the bottom end of the side roller, where a vertical load of 1516 N applies, aiding in determining the exact radial load on the roller. Additionally, there's an axial load applied on the roller cylinder, which is calculated separately.

$F_V = 1516$ N
$F_R = F_V \sin 45° = 1072$N radial load
$F_a = F_V \cos 45° = 1072$N axial load

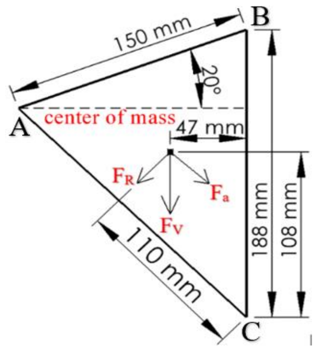



*Figure 24: Triangular material profile for side roller*

Likewise in central roller above, we have to consider the roller cylinder weight of side roller to calculate the shaft load because cylinder itself has some weight and can make some effects on loading condition.

Mass of side roller cylinder for material (S235JR structural steel, $\rho$ = 7800kg/m$^3$)

$$m = 9115 \text{ gm} = 9.115 \text{ kg}$$

Radial force generated by cylinder weight = mgcos45° = 65 N
Axial force generated by cylinder = mgsin45° = 65N
Total axial load on bearing or shaft ($F_{a,\, total}$) = 65+1072 = **1137N**
Total radial force on shaft ($F_{R,\, total}$) = 1072+65 = **1137N**
Total axial load divides and applies equally on each bearings
Axial load on each bearing = 568.5 ~**569N**

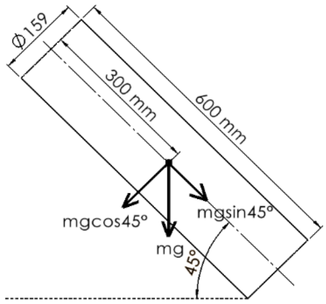



*Figure 25: Load from cylinder's mass to side roller shaft*

### D. Bearing loads on side rollers

Like in central roller, we can assume an uniformely distributed load for side roller over a distance of 220 mm from end A, which after resolving to a point load, applies 1072 N from 110 mm. So while doing ANSYS simulation, we will apply uniformely distributed load for accurate results.

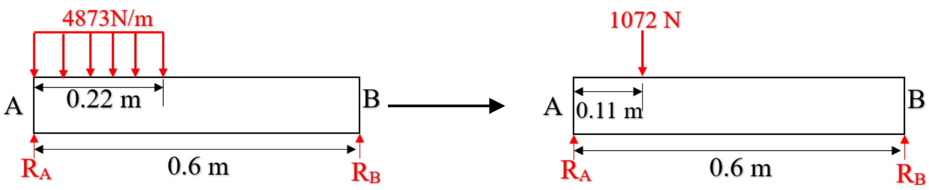



*Figure 26: Load distribution in side rollers*

The bearings on a roller experience different loads due to the troughing angle. One end bears a point load of 1072 N while the other end receives 65 N. Typically, the bearing closer to the heavier load bears more weight. The cylinder's ends are welded with the bearing hub, affecting load distribution. We can treat the cylinder as a beam to calculate bearing loads.

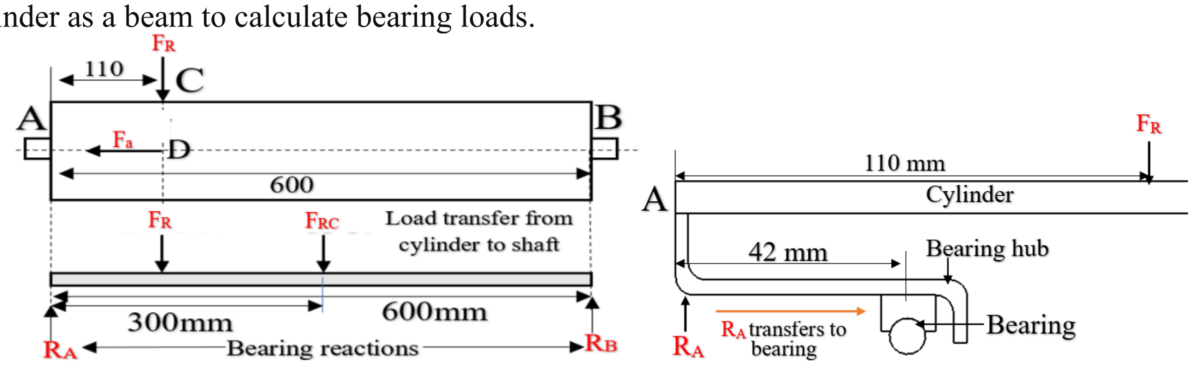



*Figure 27: Load transfer from cylinder to shaft in side roller*

Point A = lower roller end
Point B = upper roller end
$F_R$ = 1072 N
$F_a$ = 1072 N
$F_{RC}$ = 65 N (load from cylinder weight)
Taking moment equilibrium at point A we get,
$R_A \times 0 + 1072 \times 110 + 65 \times 300 - R_B \times 600 = 0$...........(1)
$R_A + R_B = 1137N$..................(2)

Solving (1) and (2)

$R_A$ = **908 N**
$R_B$ = **229 N**

It clarified that the load is more on lower end bearing (bearing on side A) than upper end bearing

**Free body diagram of shaft**

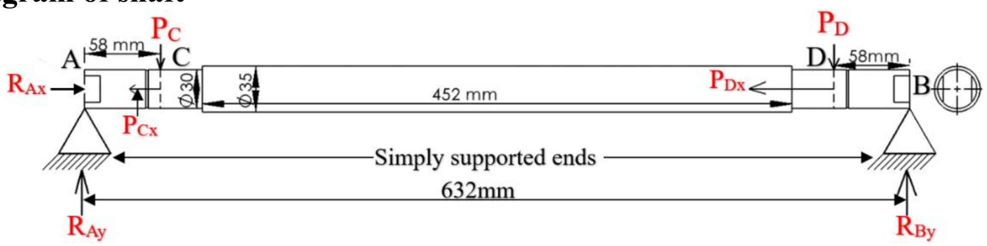



*Figure 28: Free body diagram of side roller shaft*

$P_C$ = 908 N, $P_{Cx}$ = 569 N
$P_D$ = 229 N, $P_{Dx}$ = 569 N
Taking moment equilibrium at point A we get,
$R_{Ay} \times 0 + 908 \times 58 + 229 \times 574 - R_{By} \times 600 = 0$.............(1)
Taking vertical force equilibrium we get,
$R_{Ay} + R_{By} = 1137N$........................(2)

Solving (1) and (2) we get, $R_{Ay}$ = **846 N,** $R_{Ax}$ = **1137 N** $R_{By}$ = **291 N (bracket reactions)**

*Note: The notations $R_A$ and $R_B$ for cylinder and shaft are different*

## 4.4 Simulation of existing rollers

Before devising new dimensions for improved design, we need to assess the stresses, deflections, and safety factor in current rollers. If the safety factor is excessively high, we can reduce material; if it's too low, we'll increase material to balance deflection and prevent failure. Our priority is weight optimization, so we'll primarily examine stress distribution in the roller via static structural analysis in ANSYS.

### A. Central idler

Boundary conditions:

- Fixed supported roller ends
- Load = uniformely distributed load (7920N/m) = 7524 N at centre
- Simulation tool: ANSYS workbench 2023 R2
- Simulation type: static structural
- Material: S235JR (E = 200 Gpa, v = 0.3)
- Mesh (size = 10mm, type = tetrahedron, Nodes = 77404, elements = 38532)

Results:

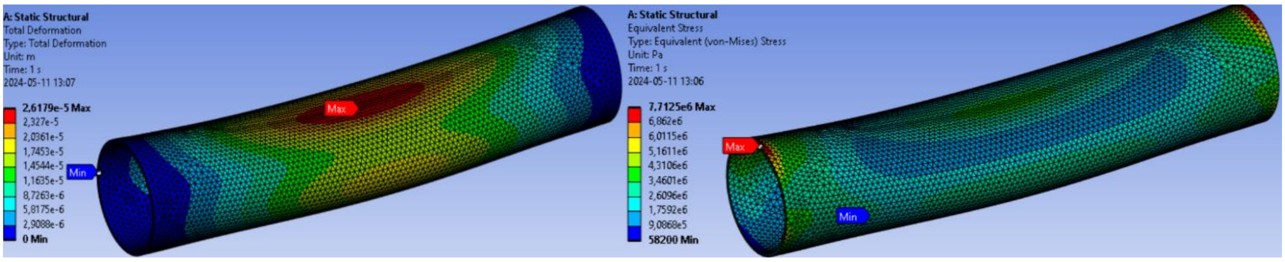



*Figure 29: Simulation results from ANSYS for existing central roller*

- Maximum stress($\sigma_{max}$) = 7.71 Mpa (at fixed end)
- Safety factor (SF) = 30.4
- Maximum deflection ($\delta_{max}$) = 0.0261 mm(at centre)

## B. Side idler

Boundary conditions: all the boundary conditions are same as for central roller except load.

- Load = uniformely distributed load (4873N/m)

  *Note: 4873 N/m ×0.22 = 1072 N*

- Mesh (size = 10mm, type = tetrahedron, Nodes = 39862, elements = 19780)

There is axial load in the side roller cylinder also so we did separate simulation for both cases, the point of load is same in both cases.

Case 1: only axial load of 1072 N

Case 2: axial load = 1072 N, radial load = 1072 N

Results:

Case 1:

- Stress($\sigma_{max}$) = 2.49 Mpa (at fixed end)
- Safety factor (SF) = 94.37
- Deflection ($\delta_{max}$) = 0.00365 mm

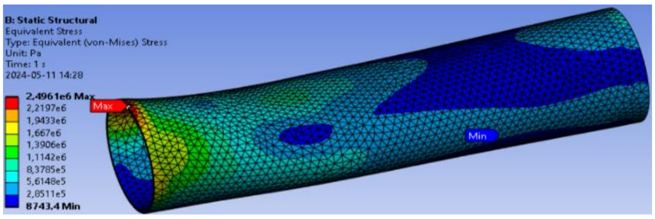



*Figure 30: Simulation results from ANSYS for existing side roller case-1*

Case 2:

- Stress($\sigma_{max}$) = 2.34 Mpa (at fixed end)
- Safety factor (SF) = 100.42
- Deflection ($\delta_{max}$) = 0.00358 mm

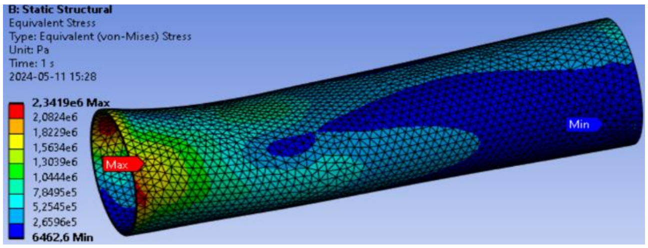



*Figure 31: Simulation results from ANSYS for existing side roller case-2*

From the above results after simulation in Ansys workbench, we have got minimum possible safety factor of 30.4 for central roller and 94.37 for side roller, which illustrates that the applied stress is 30.4 times less than the yield strength of S235JR(structural steel ≥235 Mpa), which is witnessed at top surface of the fixed end A for both roller cylinders. In the context of LKAB's conveyor system the loads we calculated are for maximum possible cases when it operates in full capacity of 9000 ton/hour. Even in such a high loading condition we have got very high safety factor, which means that we can significantly decrease the material for roller cylinder with decreasing safety factor to its optimum point. We will consider these simulation result while giving best optimized design.

# 5 Design requirements

Majority of the prerequisites have been detailed in preceding sections, and this segment serves to consolidate all design specifications, technical prerequisites, environmental factors, independent variables, and limitations. Once all essential elements are accounted for, we'll proceed to formulate the conceptual or preliminary design. The key requirements are as follows.

## 5.1 Constraints: the rollers should be designed in a way that , there are some contraints which should be considered and are listed below[14].

- The dimensions of the roller bracket remain consistent and cannot be altered, rollers must precisely match the bracket.
- Loading and boundary conditions are set and cannot be modified.
- The conveyor's capacity, set at 9000 tons per hour is fixed, requiring rollers capable of continuous 24-hour operation
- The coneyor belt's type, size, and material are predetermined.

## 5.2 environmental considerations

- LKAB's plant is at the beach, there is very high humidity in air, it means rollers can rust quickly so any solution for rust prevention is neccessary[14].
- Plant has dust, ground water and irol pallet pieces everywhere and affect to the roller so prevention is necessary[14].
- In winter season temperature drops up to -15℃ and heavy snowfall, rollers should be perfect for cold weather.

## 5.3 Other requirements[14]

- Should be light in weight.
- Should be easy to assemble and disassemble.
- Should be easy for maintenance and repair manually.
- Seals and bearings should be of good quality because main cause of roller stuck is due to having problem in bearing and seals.
- While optimizing the weight, it should also be considered that rollers wear continuously while using so the weight should not be too low.

## 5.4 Technical requirements

- Maximum material flow ($Q_{max}$) = 9000 ton/hr
- Central roller load ($P_c$) = 7524 N
- Side roller load ($P_s$) = 1072 N
- Maximum shaft deflection ($\delta_{s,max}$) = 1°
- Minimum bearing life ($L_{10h,\ min}$) = 30000 hr
- Minimum factor of safety for cylinder ($SF_{min}$) = 10
- Minimum roller life = 1 year

# 6 Concept development

## 6.1 Central roller

Based on the simulation findings in section 4.4 and the design specifications outlined in chapter 5, there's an opportinuty to modify the central roller to reduce material usage to some degree. With a factor of safety of 30.4, well above the minimum value of 10, there's leeway for weight optimization. So we have developed preliminary concept for roller components as below.

### A. Roller cylinder

Based on simulation results, we reduced the length and thickness of the roller cylinder. Shorter lengths decrease deflection and stress. Thus, we've designed a central roller with dimensions: 930 mm long, 5 mm thick, and 194 mm outer diameter. Bearing slot details will be included in the detailed design phase.

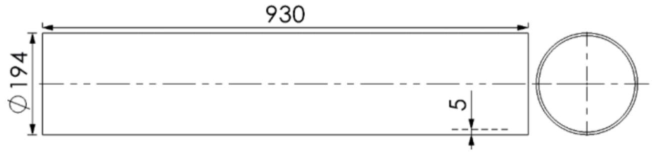



*Figure 32: Preliminary dimension of central roller cylinder*

### B. Bearing hub

The selection of the bearing hub depends on the bearing model, size, shaft diameter, and loading conditions. We've initially estimated its shape, size, and thickness based on existing rollers in LKAB. The final length will be determined after finalizing sealing elements. We'll opt for a bearing with higher strength but the same dimensions, resulting in a hub diameter of 90 mm.

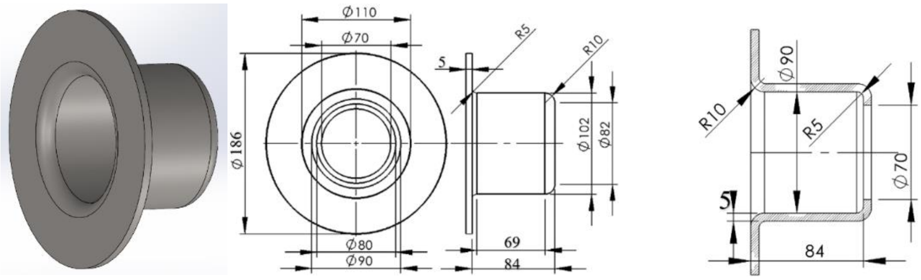



*Figure 33: Preliminary concept of bearing hub for central roller*

### C. Shaft

The design for the shaft has some limitations that we cannot increase or decrease the length because of fixed sized roller bracket. We have to design shaft keeping constant length and end shapes, moreover, the total vertical load from material finally applies to the shaft and it starts to bend which can create significant vibrations on the system and bearing can damage frequently so that for new concept we have designed stepped cross section of 45 mm diameter and of 720 mm long as shown in figure below. Total length is same as 982 mm and the end diameter is also same as 40 mm, the material and final dimension will be fixed after simulation

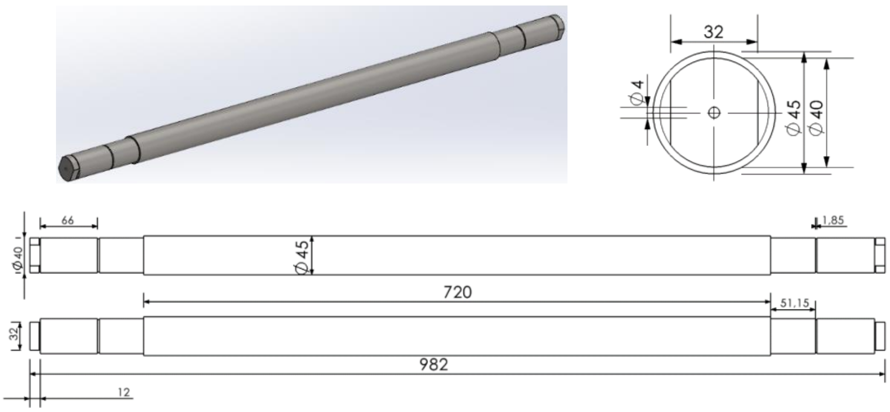



*Figure 34: Preliminary design of central roller shaft*

## 6.2 Side roller

After ANSYS simulation in section 4.4, we found a safety factor of 94.37 for stress in the side roller, indicating it's overweight. However, we need to ensure it lasts at least a year considering continuous wear. Thus, we devised a new concept to address these requirements.

### A. Roller cylinder

Previously, roller was of 600 mm long, 4.5 mm thick and diameter was 159 mm. Having mimimum safety factor of 94.37, it is better to decrease the material and for new concept we decreased the wall thickness from 4.5 mm to 3 mm keeping length and outer diameter constant of 600 mm and 159 mm respectively. The length is kept constant because of fixed roller bracket size and belt width. Isometric view and 2D drawing of new concept of roller cylinder for side roller is shown in figures below.

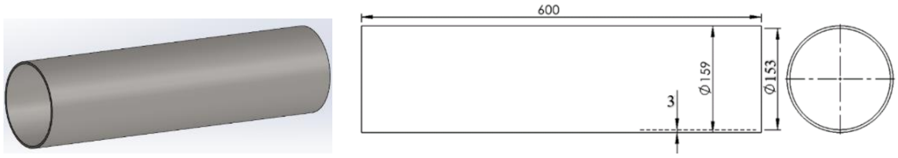



*Figure 35: Preliminary design of side roller cylinder*

### B. Bearing hub

The bearing hub for the side roller is crucial as it transfers all material loads to the bearing. It requires careful design to address potential failure mechanisms. Ideally, it's made of the same material as the cylinder (S235JR EN 10026-1)/N (basic design) with a thickness of 4.5 mm. In the new concept, the shape remains the same, but the thickness is reduced to 4 mm, and the exact diameter of the bearing hub is 72 mm. Below are preliminary concepts with 3D models and 2D drawings.

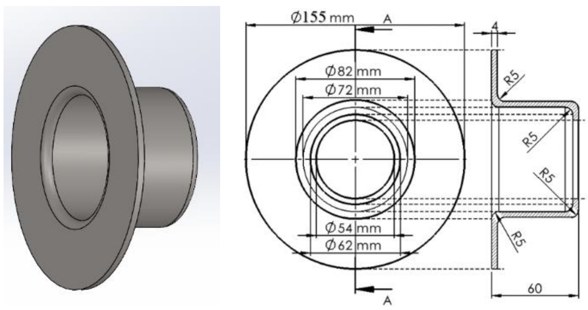



*Figure 36: Preliminary design of bearing hub for side roller*

### C. Shaft

The total load from belt and cylinder applies finally to the roller shaft through bearings so it should be strong and stiff enough. If the shaft deflects significantly, vibration can create and it can make bearing to fail rapidly. Previously, shaft for side roller was 632 mm long, 30 mm unifrom diameter with solid cross section. For the new concept we have designed steeped cross section as shown in figures below with keeping constant length of 632 mm. Stepped diameter is of 35 mm and 452 mm long symmetric from both ends.

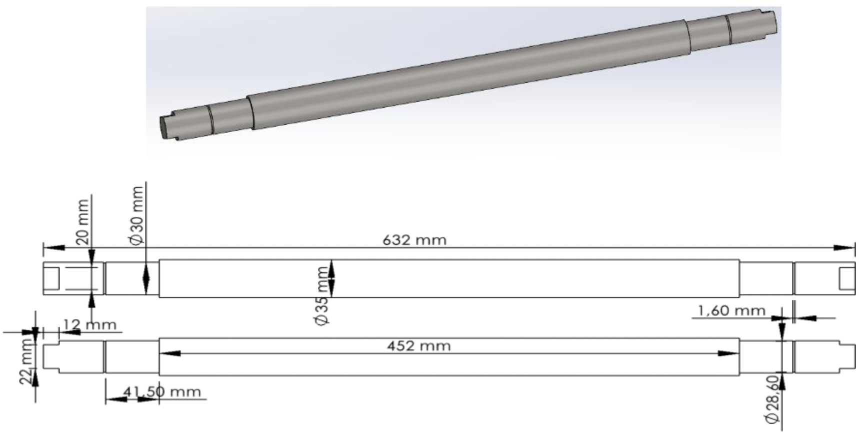



*Figure 37: Preliminary design of side roller shaft*

## 6.3 Sealing elements

**Inner lip seal:** To stop the leakage of lubricant, hydraulic fluid from actuators, air leakage from pneumatic pistons lip seals are widely used and are very suitable in rotating components such as bearings and shafts **[19],** as well as in conveyor roller it is used to stop the leakage of lubricant from bearing. So we can select any suitable lip seal from the market which can help to protect the bearing. Type, material and working principle for both side and central roller lip seal is same and below are their dimension.

| Parameters | For central roller | For side roller |
|---|---|---|
| Outside diameter | 90 mm | 72 mm |
| Inside diameter | 40 mm | 30 mm |
| Thickness | 12 mm | 7 mm |
| Material | nylon | nylon |

*Table 5: Desired dimensions of inner lip seals*[20]

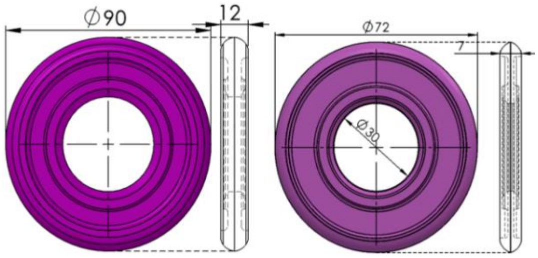



*Figure 38: Inner lip seals*

**Labyrinth seals (female and male):** labyrinth seals are popular in conveyor rollers to work effectively against the intrusion of outside contaminants towards bearing. Labyrinth seals are female and male, filled with lithium lubricating grease between them**[20].**

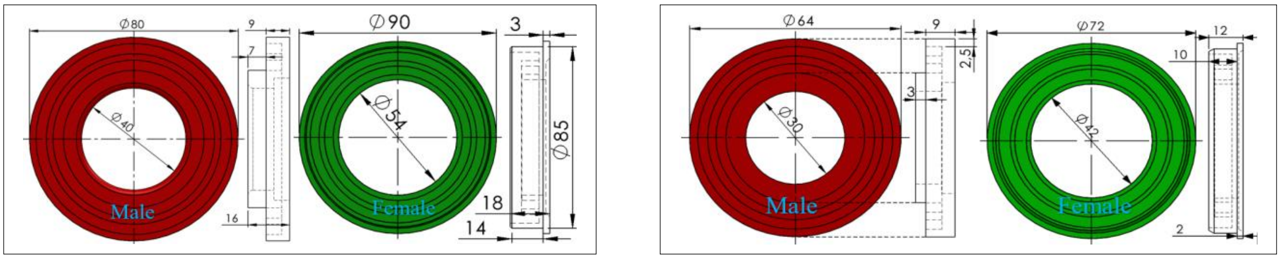



*Figure 39: Labyrinth seals (male and female for central and side roller respectively)*

| Parameters | For central roller | | For side roller | |
|---|---|---|---|---|
| | Male | Female | Male | Female |
| Inside diameter (mm) | 40 | 54 | 30 | 42 |
| Outside diameter (mm) | 80 | 90 | 64 | 72 |
| Total thickness (mm) | 16 | 18 | 12 | 12 |
| Material | nylon ( polyoxyethylene is also used) | | | |

*Table 6: Desired dimensions of Labyrinth seals*[20].

**Galvanized metal dust cover:** in idler roller a metal dust cover is used to protect the bearing from dust, is made from metal specially steel and rotates with bearing hub and do not touch the shaft**[20].**

| Parameters | For central roller | For side roller |
|---|---|---|
| Inside diameter | 52 mm | 38 mm |
| Outside diameter | 90 mm | 72 mm |
| Width | 23 mm | 16 mm |
| thickness | 2.5 mm | 2 mm |
| Material | Carbon steel sheet | |

*Table 7: Desired dimensions of metal dust cover*[20].

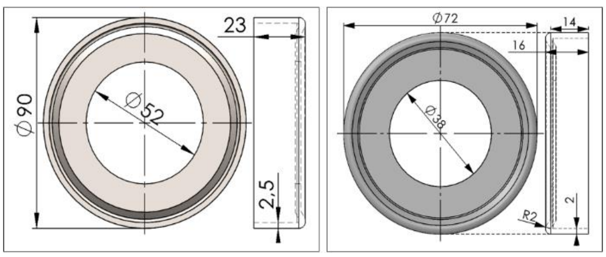



*Figure 40: Galvanized metal dust covers*

**Protective cover:** this cover protects the roller and sealing parts from stones, big pallets of conveying material**[20].**

| Parameters | For central roller | For side roller |
|---|---|---|
| Inside diameter | 40 mm | 30 mm |
| Outside diameter | 90 mm | 72 mm |
| Width | 40 mm | 23 mm |
| thickness | 2.5 mm | |
| Material | Natural rubber | |

*Table 8: Desired dimensions of protective cover*[20].

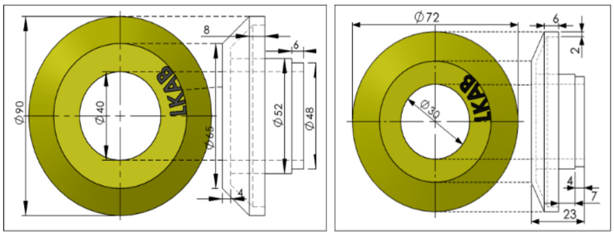



*Figure 41: Protective outer covers*

**Circlip pin:** Circlip pin is neccessary to stop the movement the bearing from its place in axial direction, it is normally fitted in roller shaft with the help of small groove[20].

| Parameters | For central roller | For side roller |
|---|---|---|
| Inside diameter | 36.5 mm | 27.9 mm |
| Outside diameter | 40.5 mm | 31.9 mm |
| Width | 46.75 mm | 35.9 mm |
| thickness | 1.75 mm | 1.5 mm |
| Material | Stainless steel | |

*Table 9: Desired dimensions of circlip pin*[21].

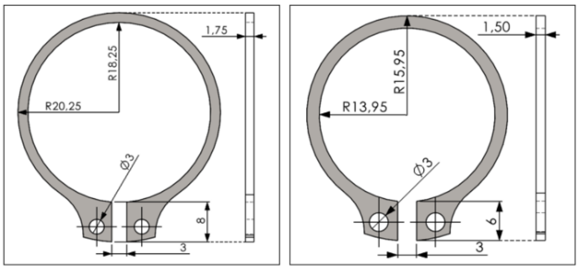



*Figure 42: Circlip Pins*[21].

# 7 Material selection criteria

Material selection is a crucial aspect of product design, guided by systematic processes and techniques. Professor Michael F. Ashby's "Material selection in mechanical design" offers valuable insights, including the use of Ashby charts for material selection. In our conveyor roller design, critical components like the roller cylinder, shaft, and bearing housing require optimization for mass, cost, volume, strength, and other parameters. We'll focus on identifying the best materials for these components using Ashby's methodology outlined in the 5th edition of his book.

## 7.1 Material selection for roller cylinder and shaft

The shaft and roller cylinders in the conveyor system act as beams, bearing vertical loads at their positions. Since both components endure similar loads, using the same material for both is practical and cost-effective. Michael Ashby's material index (M) aids in selecting materials that maximize design efficiency by minimizing mass. Considering existing materials and engineering property requirements set based on references like Rulmeca, we'll utilize material index and property criteria to identify the most suitable material for our design.

### Technical requirements for material

| No | Material properties | Minimum | Maximum | Remarks |
|---|---|---|---|---|
| 1 | Young's modulus | 100 Gpa | 220 Gpa | Most essential feature |
| 2 | Shear modulus | 40 Gpa | 100 Gpa | |
| 3 | Bulk modulus | 100 Gpa | 190 Gpa | |
| 4 | Yield strength(elastic limit) | 150 Mpa | 360 Mpa | Essential feature |
| 5 | Tensile strength | 250 Mpa | 600 Mpa | Essential feature |
| 6 | Surface hardness | 60 HV | 155 HV | Most essential feature |
| 7 | Bending shape factor($\emptyset_B^e$) | 15 | 66 | Most essential factor |
| 8 | Fatigue strength at $10^7$ cycles | 160 Mpa | 300 Mpa | Essential feature |
| 9 | Price | 10 NOK/kg | 30NOK/kg | Essential feature |
| 10 | Service temperature | -19°C | 200°C | Essential |

*Table 10: Technical requirements for material selection*

### General requirements for material

- Priority 1: The shaft should be stiff enough.
- Priority 2: The shaft material should be corosion resistant.
- Priority 3: The shaft material should have high durability and wear resistance.
- Priority 4: The shaft material should have high impact resistance.
- Priority 5: The shaft material should be easy for fabrication.
- Priority 6: The shaft material should be cost effective and easily available in required size.

We won't solely prioritize minimizing mass; instead, materials lighter than existing roller weights will rank higher. We'll consider all requirements to finalize the material choice. Starting with the priority for a stiff shaft design, the design requirement is as follows:

Design requirements for light stiff roller shaft and cylinder for minimum mass

| | Roller shaft | Roller cylinder |
|---|---|---|
| Function | ■*Light stiff beam(shaft)* | ■*Light stiff beam(pipe element in bending)* |
| Constraints | ■*Length L specified (982 mm for central and 632 mm for side idler* | ■*Length L specified( 930 mm for central and 600 mm for side idler)* |
| Objective | ■*Minimize mass* | ■*Minimize mass* |
| Free variables | ■*Choice of material* | ■*Choice of material* |
| | ■*Shaft diameter* | ■*Cylinder diameter and thickness* |

*Figure 43: Design requirements for material selection*[22].

The steps for finding material index is given below.

| | |
|---|---|
| Objective function | Mass(m)= $\rho AL$ |
| The deflection ($\delta$) of beam | $\delta = \frac{FL^3}{C_1 EI}$, $C_1 = 48$ for simply supported beam |
| Second moment of area | $I = \frac{\pi r^4}{4} = \frac{A^2}{4\pi}$ |
| Critical stiffness $S_{cr} \leq S = \frac{F}{\delta} = \frac{48E\left(\frac{Ar^2}{4}\right)}{L^3}$ | Area of cross section $A \leq \left(\frac{S_{cr}L^3\pi}{12E}\right)^{\frac{1}{2}}$ |
| Masss (m) | $= \rho AL = \rho\left(\frac{S_{cr}L^3\pi}{12E}\right)^{\frac{1}{2}} L = \rho\left(\frac{S_{cr}L^3\pi}{12}\right)^{\frac{1}{2}} L \frac{1}{E^{\frac{1}{2}}} = \left(\frac{S_{cr}L^3\pi}{12}\right)^{\frac{1}{2}} L \frac{\rho}{E^{\frac{1}{2}}}$ |
| Material index $(M_1) = \frac{E^{\frac{1}{2}}}{\rho}$ | $(M_1) = \frac{E^{\frac{1}{2}}}{\rho}$ |
| Material property chart | Young's modulus(E) vs density($\rho$) |
| Equation for coordinates | $\underbrace{\log E}_{y-axis} = \underbrace{2\log\rho}_{x-axis} + \underbrace{2\log C}_{constant}$ |
| Slope of the guideline | 2 |

*Table 11: Steps to find the material index for beam in bending*[22].

Here,

$$\text{Mass(m)} = \text{performance matrix(P)} = f_1(F)\times f_2(G)\times f_3(M) \rightarrow (P) \leq f_1(F)\times f_2(G)\times f_3(M)$$

$$\text{Functional requirement } f_1(F) = \left(\frac{S_{cr}\pi}{12}\right)^{\frac{1}{2}}$$

$$\text{Geometry requirement } f_2(G) = L^{\frac{5}{2}}$$

$$\text{Material properties } f_3(M) = \frac{\rho}{E^{\frac{1}{2}}}$$

The figure below shows possible materials for minimum mass design considering technical requirements set in earlier section. Five materials are listed by CES GrantaEdupack on the basis of minimum mass, cast iron.ductile(nodular), low carbon steels, Nickel, Bronze, and Brass.

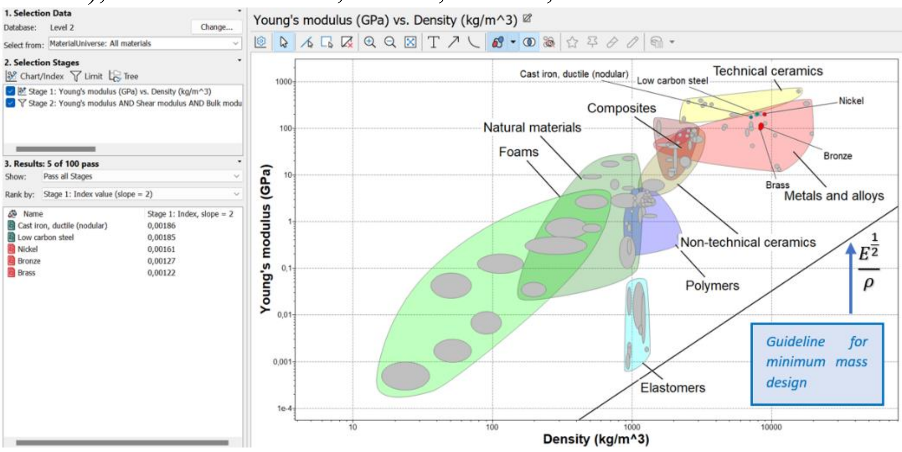



*Figure 44: List of materials got from CES granta*[23].

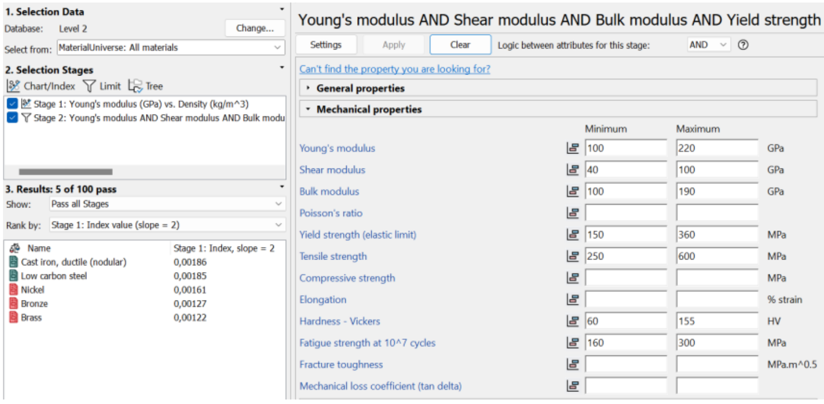



*Figure 45: Minimum and maximum parameters for material slection from CES granta*[23].

The guideline in the bottom right corner of the figure indicates that despite exploring various materials on CES Granta, only five are viable for a light and stiff beam design. Other materials didn't meet the minimum technical requirements even when set low. Some composites, while used in conveyor rollers, were deemed unsuitable for our design load and boundary conditions, as they would deflect excessively and fail. Among the five materials, we'll compare their cost, availability, and other necessary physical and mechanical properties to make a selection.

| Properties | Materials | | | | |
|---|---|---|---|---|---|
| | Cast iron | Low carbon steel | Nickel | Bronze | Brass |
| Young modulus | 180 GPa | 200-220 GPa | 190-220 GPa | 130 GPa | 110 GPa |
| Density(Kg/$m^3$) | 7150 | 7820 | 8950 | 8700 | 8480 |
| Availability | Easily available | Easily available | Available | Limited | limited |
| Cost(NOK/Kg) | 5.8(raw) | 13.6 | 252 | 129 | 88.2 |
| Processability | Good | Good | Satisfactory | good | Excellent |

*Table 12: Comparison of properties of materials from CES Granta*[23].

The table indicates that cast iron, low carbon steel, and nickel have higher Young's modulus than bronze and brass, meeting our first engineering requirement. Considering cost, low carbon steels are chosen over nickel, bronze, and brass. Low carbon steel, including mild steels, structural steels, drawing quality steels, and highly formable steels, is specific. Michael F. Ashby's shape factor (ϕ) aids in material selection for shaped sections like rods and pipes, enhancing stiffness while reducing mass. According to Ashby's book, structural steel exhibits superior bending shape factors, with values up to 65, making it suitable for roller components.

**Specific**

| Material | Grade | Number | Classification | Composition |
|---|---|---|---|---|
| 1.0044<br>(S275JR)<br>Non-alloy<br>Structural<br>steel | S275JR | 1.0044 | Structural | Carbon(C): ≤0.21%<br>Manganese(mn):≤0.5%<br>Phosphorus(P):≤0.045%<br>Sulfur(S): ≤0.045%<br>Silicon(Si): ≤0.04%<br>Nitrogen(N): ≤0.012%<br>Copper(Cu): ≤0.55% |
| | Young's modulus | Density | Yield strength | |
| | 210 Gpa | 7800 Kg/m$^3$ | 275 Mpa | |
| | Shear modulus | Bulk modulus | Standard | |
| | 79 Gpa | 170 Gpa | EN 10025-2 | |
| | Melting point | Tensile strength | Surface hardness | |
| | 1480℃ | 410 Mpa | 140 HV | |
| Category | DIN Steel(Structural), St44-2 | | Poission's ratio | 0.28 |

*Table 13: Chemical composition and mechanical properties of S275JR*[24]

### 7.1.1 Possibility discussion for hollow shaft

Hollow shafts, due to their shape factor, are both lightweight and stiffer than solid shafts. Michael F. Ashby defines the shape factor (∅) as a dimensionless factor representing the mechanical efficiency achieved by combining material with macroscopic shape. This factor varies under different loading conditions, such as bending ($\emptyset_B^e$) and torsion ($\emptyset_T^e$). In conveyor rollers, shafts primarily undergo bending, and increasing the bending shape factor enhances stiffness. Bending shape factor compares the bending stiffness of a shaped cross-section to that of a solid square reference of the same length and cross-section [22].

$$\emptyset_B^e = \frac{S}{S_0} = \frac{EI}{EI_0} = \frac{12I}{A^2}$$

The figure illustrates that the left tube with the same cross-section is 2.5 times stiffer than the square one, while the right-side tube achieves the same stiffness with only one fourth of the cross-section. This demonstrates that modifying the shape can significantly enhance strength and stiffness.

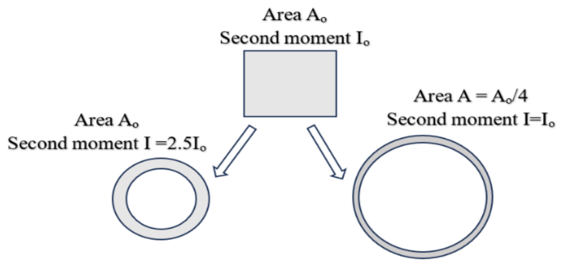



*Figure 46: The effect of section shape on bending stiffness*[22]

### Limitation for conveyor roller shaft

Obviously the hollow shaft is more stiffer and lighter than solid shaft but we have a constraint of bracket size (22mm×30mm) to fit the shaft, if we design hollow shaft, the outer diameter should be very large which is not possible because it  will occupy more space and create other troubles to design the bearing hub, selection of bearing and sealing elements. We will find here the size of hollow shaft required to achieve same stiffness as 30 mm solid circuar shaft.

Bending stiffness for 30 mm solid shaft ($S_{solid}$) = EI = $E\frac{\pi r^4}{4}$ = 39760.78E [22]

Bending stiffness for hollow shaft($S_{hollow}$) = $E\frac{\pi(r_o^4 - r_i^4)}{4}$, if $r_o$ = 15mm, $r_i$ = 0 (same as solid shaft)

if $r_o$ = 20mm, $r_i$ = 18.18 mm( not possible due to 22mm bracket size

To achieve same stiffness of 39760.78E, outsode diameter of hollow shaft should be greater than 30mm as well as inside diameter can not be more than 5 mm. To achieve same stiffness, the internal radius should be greater than wall thickness according to Michael F. Ashby($\emptyset_B^e$ for tube = $\frac{3}{\pi}\left(\frac{r}{t}\right)$, r≫ $t$). So for our conveyor roller hollow shaft design is not possible.

# 8 Bearing selection

Roller accessories are the important part which help roller to perform well giving optimized, reliable, efficient and economic output for the belt conveyor system. The main accessories for the roller are bearing, sealing elements, and lubrication. In this chapter we will select the perfect accessories for both side roller and central roller by discussing all the neccessary requirements, design parameters, boundary conditions, market availability, price and dimensions.

## 8.1 Bearing for side roller

In conveyor roller, bearings play a cruical role to decrease the friction between cylinder and shaft for the smooth spinning of roller cylinder on the shaft. There are generally two bearings are used in a single roller set which are installed inside the cylinder to fit the shaft into bore of the bearing. Moreover there is a specific part called bearing hub, which holds bearing and connects the roller cylinder. We have some requirements for side roller bearing which are put prioritywise below.

- Bearing to withstand 908 N radial load and 569 N axial load
- Bore diameter: 30mm to fit on 3 mm shaft.
- Durable, resistance to fatigue, wear and impact.
- Capable of handling 350 RPM rotational speed.
- Corrosion resitant.
- Easy installation, proper alignment with roller, appropriate mounting feature and compatibility.
- Ease of maintenance and replacement option.

On the basis of above requirements, we have some options from SKF with various specifications and material properties as below. We will compare all the possible bearings and select best one.

| Specification | Bearing models | | | |
|---|---|---|---|---|
| | 6306 (existing) | 6306-2Z | W 6306 | W 6306-2RS1 |
| Type | Deep groove ball | Deep groove ball | Deep groove ball | Deep groove ball |
| Manuafacturer | SKF | SKF | SKF | SKF |
| Dimension | 30×72×19 mm | 30×72×19 mm | 30×72×19 mm | 30×72×19 mm |
| Basic dynamic load rating | 29.6 KN | 29.6 KN | 22.9 KN | 22.9 KN |
| Basic static load rating | 16 KN | 16 KN | 15 KN | 15 KN |
| Limiting speed | 13000 RPM | 11000 RPM | 14000 RPM | 6300 RPM |
| Reference speed | 20000 RPM | 20000 RPM | 22000 RPM | NA |
| Lubricant | None | Grease | None | Grease |
| Bearing material | Bearing steel | Bearing steel | Stainless steel | Stainless steel |
| Relubrication feature | Without | Without | Without | Without |
| Sealing | Without | On both sides | Without | On both sides |
| Tolerance class | Class P6 | Class P6 | Normal | Normal |
| Product weight | 0.34 kg | 0.353 kg | 0.346 kg | 0.346 kg |
| Average price | 315 NOK/Pc | 364 NOK/Pc | 4872 NOK/Pc | 5603 NOK/Pc |
| Terms used in model no | 6306 = indicates type, size, dimansion, series and variation<br>2Z = indicates shield on both side<br>2RS1 = indicates two rubber seals on one side of bearing<br>W = indicates specific material (stainless steel in this case) | | | |

*Table 14: Comparison of the bearings 6306, 6306-2z, W6306 and* W6306-2RS1[25]

### Comparison

From the table, it's evident that stainless steel bearings like W 6306 and W 6306-2RS1 are excessively expensive despite meeting general requirements. However, W 6306-2Z aligns closely with stainless steel bearings in features, yet it's more cost-effective. Hence, we'll opt for W 6306-2Z for the side roller, considering its affordability and suitability, with all relevant features, pros, and cons detailed.

### Selection

| Selection | Reason of selection | Advantages/Disadvantages |
|---|---|---|
| SKF 6306-2Z | SKF 6306-2Z boasts a simple, robust, and versatile design, offering low friction and high-speed capability. It efficiently accommodates radial and axial loads in both directions while demanding minimal maintenance. Unlike the existing 6306 bearing, SKF 6306-2Z features double side seals and pre-filled grease, essential for heavy-duty conveyor rollers. Its P6 tolerance class ensures compatibility with existing shafts, eliminating the need for modifications. This bearing saves time by providing ready-to-use grease and prevents failures, rusting, and maintenance costs with its double seals. Although slightly more expensive than the 6306, it's approximately 13 times cheaper than W6306 and W 6306-2RS1. | Bearing steel, or high carbon-chromium steel, isn't naturally corrosion-resistant like stainless steel. However, its chromium content allows it to develop a protective chromium oxide layer on its surface when exposed to oxygen, which helps deter further oxidation and corrosion. |

*Table 15: Advantages and limitations of bearing 6306-2Z*[25]

**Basic rating life of bearing 6306-2Z**

The fatigue life of a bearing represents the number of revolutions or operating hours at a constant speed before the first indication of metal fatigue, such as rolling contact fatigue (RCF) or spalling, appears on either the bearing's rings or rolling elements. There are significant variations in fatigue life observed among identical bearings operating under the same conditions, as evidenced by both laboratory tests and real-world experience. If we consider only load and bearing speed, we can find basic rating life for any kind of bearings. According to SKF, the basic rating life in accordance with International Organization for standardization (ISO) 28 is represented by $L_{10}$. If the speed is constant, it is preferable to calculate the life in term of $L_{10h}$. Below is the calculation of basic rating life for side roller bearing which has higher radial load. **[25]**

$$L_{10} = \left(\frac{C}{P}\right)^{p},\ L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^{p} \quad \text{......................} \ (2)\ [25]$$

Where,

$L_{10}$ = basic rating life (at 90% reliability) [millions of revolutions]
$L_{10h}$ = basic rating life (at 90% reliability) [operating hours]
C = basic dynamic load rating [kN] = 29.6 kN
P = equivalent dynamic bearing load [kN] = X $F_r$ + Y $F_a$ = 1.58 kN
*(see table 3 appendix C)*
$F_r$ = actual radial bearing load = 0.908 kN
$F_a$ = actual axial bearing load = 0.569 kN
X, Y = radial, axial load factors respectively = unknown
n = rotational speed [rev/min] = 349
p = exponent of the life equation
= 3 for ball bearing and 10/3 for roller bearings

Putting all these values in equation (2) we get,

$L_{10}$ = 6575 million revolutions
$L_{10h}$ = 313998 hours

Above life estimations are calculated for SKF 6306-2Z ball bearing with following lubrication conditions.

Actual coefficient of viscosity (v) = 25.3mm$^2$/s
Rated coefficient of viscosity ($v_1$) = 40.9mm$^2$/s
Rated coefficient of viscosity at 40°C ($v_{ref}$) = 163mm$^2$/s

*Note: bearing temperature considerations are: inner ring = 70°C and outer ring = 65°C. For more information about this bearing see table 3 appendix C.*

### Lubrication system and performance

SKF, the manufacturer of the 6306-2Z ball bearing, states that the bearing comes pre-lubricated with LGNL-2 grease. This grease, known for its high load capacity, consists of a unique blend of mineral oil and anhydrous calcium thickener. It provides exceptional water resistance and outperforms traditional lithium-based greases, ensuring reliability and durability even in extreme conditions. LGNL-2 offers superior protection and performance, making it the preferred choice for demanding applications [25].

- Excellent corrosion protection
- Outstanding water resistance
- Good mechanical stability
- Excellent wear protection
- Operating temperature range -30°C to 110°C.

## 8.2 Bearing for central roller

In the central roller, the existing SKF 6308 bearing lacks lubrication and sealing. All the general requirements are same for side roller bearing, to improve upon this, we'll seek a bearing with enhanced features and higher quality. The requirements for the central roller bearing, prioritized, are as follows.

- Bearing to withstand 3898 N radial load.
- Bore diameter: 40mm to fit on 40 mm shaft.

On the basis of these requirements we have sorted four bearings from SKF with their specification and will compare each other on the basis of design requirements and select one.

| Specification | Bearing models | | | |
|---|---|---|---|---|
| | 6308 (existing) | 6308-2Z | 6308-2Z-C3 | 6308-2RS1-C3 |
| Type | Deep groove ball | Deep groove ball | Deep groove ball | Deep groove ball |
| Manuafacturer | SKF | SKF | SKF | SKF |
| Dimension | 40×90×23 mm | 40×90×23 mm | 40×90×23 mm | 40×90×23 mm |
| Basic dynamic load rating | 42.3KN | 42.3 KN | 42.3 KN | 42.3 KN |
| Basic static load rating | 24 KN | 24 KN | 24 KN | 24 KN |
| Limiting speed | 11000 RPM | 8500 RPM | 8500 RPM | 5000RPM |
| Reference speed | 17000 RPM | 17000 RPM | 17000 RPM | NA |
| Lubricant | None | Grease | Grease | Grease |
| Bearing material | Bearing steel | Bearing steel | Bearing steel | Bearing steel |
| Relubrication feature | Without | Without | Without | Without |
| Sealing | Without | On both sides | On both sides | On both sides |
| Tolerance class | Class P6 | Class P6 | Class P6 | Class P6 |
| Product weight | 0.611 kg | 0.635kg | 0.635 kg | 0.63 kg |
| Average price | 584 NOK/Pc | 671 NOK/Pc | 671NOK/Pc | 760NOK/Pc |
| Terms used in model no | 6308 = indicates type, size, dimansion, series and variation<br>2Z = indicates shield on both side<br>2RS1 = indicates two rubber seals on one side of bearing<br>C3 = indicates internal clearance of the bearing | | | |

*Table 16: Comparison of the bearings 6308, 6308-2z,6308-2Z-C3 and* 6308-2RS1-C3 [25]

Among the four bearing types, 6308-2Z, and 6308-2Z-C3 offer more features than the existing 6308 bearing, including seals and lubrication. Although slightly more expensive, 6308-2Z-C3 provides better value for money compared to 6308-2RS1-C3. Thus, 6308-2Z-C3 is selected for its cost-effectiveness and advantages.

**Selection**

| Selection | Reason of selection | Advantages |
|---|---|---|
| SKF 6308-2Z-C3 | • Simple, robust and versatile design.<br>• Low friction and high-speed capability.<br>• Load capacity 24 KN ≫ 3.898 KN<br>• Comes with double side seals and grease.<br>• Same tolerance class P6 as 6308, no modification required for shaft.<br>• Saves money on seeking other lubricants.<br>• Ruduces friction and heat production due to having C3 internal toerance.<br>• Grease inside prevents from failure, rusting and prevents maintenance cost. | Bearing steel, or high carbon-chromium steel, isn't naturally corrosion-resistant like stainless steel. However, its chromium content enables the formation of a protective chromium oxide layer on the surface when exposed to oxygen, which inhibits further oxidation and corrosion. With seals, grease, and C3 class internal tolerance, 6308-2Z-C3 emerges as a promising choice. |

*Table 17: Advantages and limitations of bearing 6308-2Z-C3* [25]

### Basic rating life of bearing 6308-2Z-C3

$$L_{10} = \left(\frac{C}{P}\right)^p,\ L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^p \ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)\ [25]$$

C = basic dynamic load rating [kN] = 42.3kN

P = equivalent dynamic bearing load [kN] = X $F_r$ + Y $F_a$ = 3.9 kN[25] (see table 1 appendix C)

$F_r$ = actual radial bearing load = 3.898 kN

$F_a$ = actual axial bearing load = 0 kN

X, Y = radial, axial load factors respectively = unknown

n = rotational speed [rev/min] = 286

p = exponent of the life equation

= 3 for ball bearing and 10/3 for roller bearings

Putting all these values in equation (2) we get,

$L_{10}$ = 1276 million revolutions

$L_{10h}$ = 74400 hours

Above life estimations are calculated for SKF 6306-2Z-C3 ball bearing with following lubrication conditions.

Actual coefficient of viscosity (v) = 25.3 $mm^2/s$

Rated coefficient of viscosity ($v_1$) = 43.3 $mm^2/s$

Rated coefficient of viscosity at 40°C ($v_{ref}$) = 175 $mm^2/s$

*Note: bearing temperature considerations are: inner ring = 70°C and outer ring = 65°C. For more information about this bearing see table 1 appendix C.*

### Lubrication system and performance

SKF, the manufacturer of the 6308-2Z-C3 ball bearing, confirms that it comes pre-lubricated with LGNL-2 grease. LGNL-2, a high load industrial bearing grease, is renowned for its exceptional performance and water resistance. Its unique blend of mineral oil and anhydrous calcium thickener surpasses traditional lithium-based greases, ensuring reliability and durability even in extreme conditions. With LGNL-2, superior protection and performance are guaranteed, making it the preferred choice for demanding applications [25].

- Excellent corrosion protection.
- Outstanding water resistance.
- Good mechanical stability.
- Excellent wear protection.
- Operating temperature range -30°C to 110°C.

# 9 Numerical analysis

## 9.1 Central roller cylinder

### 9.1.1 Static structural analysis

Boundary conditions:

- Component type: Fixed ended beam element
- Cross section: hollow tube (OD 194, ID 184)
- Length: 930 mm
- Analysis type: Static structural
- Software: ANSYS workbench 2023 R2
- Load ($P_C$) = 7524 N (uniformely distributed 8090 N/m)
- Material: 1.0044 (S275JR) structural steel (E = 210 Gpa, v = 0.28)
- Mesh: type = automatic, size = 10 mm, Nodes = 39736, elements = 5952

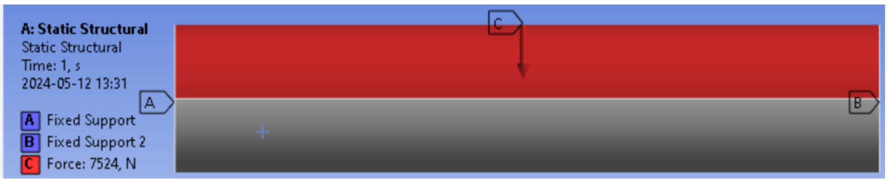



*Figure 47: Loading and boundary condition of central roller in ANSYS workbench*[26]

Results

- Maximum deformation ($\delta_{max}$) = 0.0363 mm at centre
- Minimum deformation ($\delta_{min}$) = 0 mm at fixed ends
- Maximum stress ($\sigma_{max}$) = 11.01 Mpa at top surface of fixed ends
- Safety factor (SF) = 25
- Reaction forces ($R_A$ & $R_B$) = 3762 N

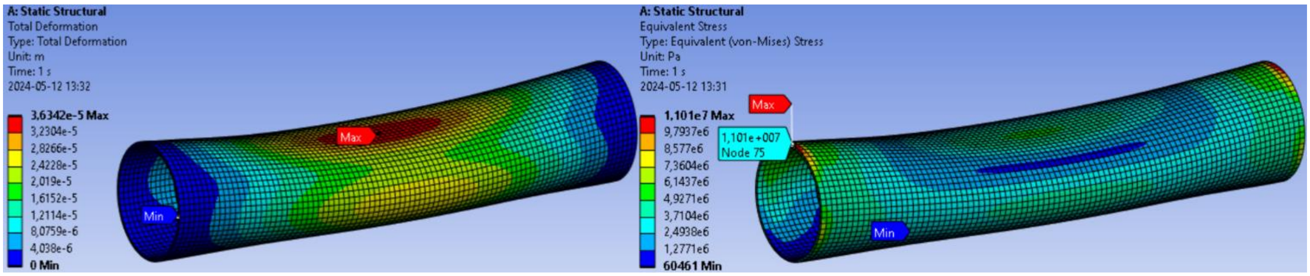



*Figure 48: Deformation and stress distribution for central roller obtained from ANSYS* 26]

Comment on result:

The roller cylinder, being a thin-walled structure, has a higher load-bearing capacity compared to a solid square reference. Even with a thickness reduction from 6.3 mm to 5 mm, the maximum deformation at point C is only 0.0363 mm, allowing weight optimization. Despite a low stress of 11.01 MPa and a minimum safety factor of 25, further thickness reduction isn't feasible due to constant wear from friction with the conveyor belt. The cylinder can withstand a static load of 7524 N without yielding.

### 9.1.2 Modal (vibration) analysis

It is most important that the roller should be safe from vibration, if the vibration due to load and motion of conveyor belt exceeds the natural frequency of the system, the roller will fail in first mode shape which we get from Ansys modal analysis. In this section we will perform a modal analysis for central roller cylinder having same boundary condition as in static structural analysis and will compare the first mode shape frequency with hand calculated value.

**Boundary conditions:**

- Fixed ended beam element.
- Mesh element type: tetrahedrons.
- Mesh element size = 10 mm
- Total nodes = 92546
- Total elements= 45840
- Tool used= Ansys workbench (modal analysis)
- Material = 1.0044(S275JR structural steel, E =210G

Result:

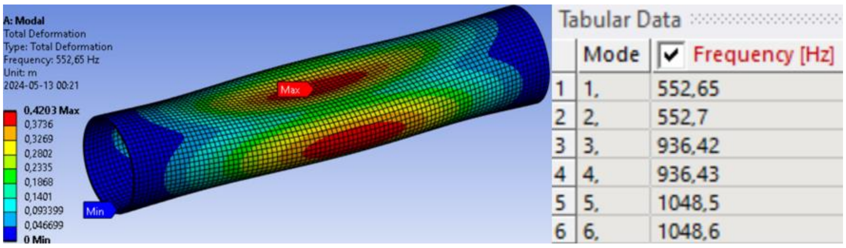
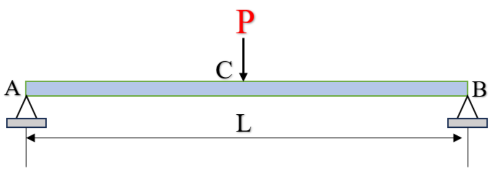



*Figure 49: Modal analysis result for central roller cylinder from ANSYS workbench*[26]

From figure we can see that the first mode will occur at 552.65Hz frequency which is quite high, but we will calculate the natural frequency generated due to deflection because of 8090N/m uniformly distributed load.

### Analytical calculation

*Assumption: We can resolve uniformely distributed load into a point load at centre of 7524 N and the cross-section is cylindrical, we are going to use a line element with fixed supported beam having thin-walled cross section.*




load (P) = 7524 N
length (L) = 0.930 m
diameter (Ø) = (OD 0.194 m, ID 0.184m)
Young's modulus (E) = 210 GPa
Second moment of area (I) = $1.3265\times10^{-5}$m$^4$

Spring constant of fixed supported beam having point load at center (K) = $\frac{192EI}{L^3}$ [27]

$$= \frac{192\times210\times10^{9}\times1.3265\times10^{-5}}{0.930^{3}}$$

= 664952645 N/m

External mass from load (m) = 7524/9.81 = 767 Kg

Natural frequency of the cylinder due to vibration ($f_n$) = $\frac{1}{2\pi}\sqrt{\frac{K}{m}} = \frac{1}{2\pi}\sqrt{\frac{664952645}{767}}$ = 148.18 Hz [27]

Safety factor in first mode = 552.62/148.18 = 3.73 (safe)

*The natural frequency of the system with external load is 148.18 Hz with safety factor of 3.73, it means cylinder is safe from vibration.*

### 9.1.3 Fatigue stress analysis

Fatigue strength indicates the maximum stress a material can endure for a specified number of loading cycles before failing under cyclic loading. In conveyor rollers, the cylinder experiences fluctuating or alternating stress due to its revolving motion matching the belt velocity. Fatigue failure accounts for 90% of mechanical service failures. In analyzing cyclic loading, we'll assess stresses and safety factors under consistent boundary and loading conditions as in static structural analysis.

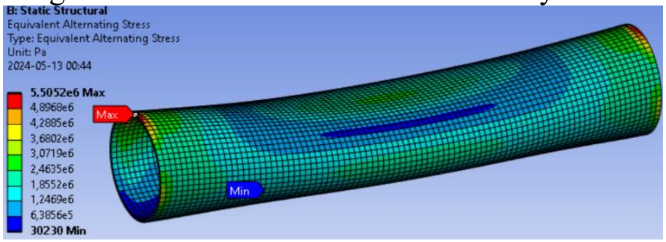
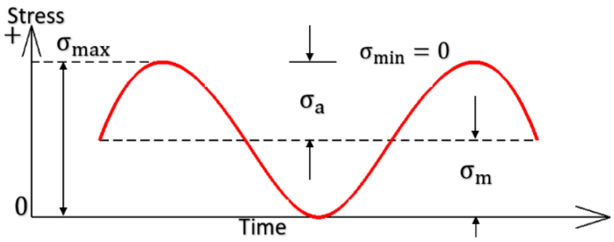



*Figure 50: Equivalent alternating stress distribution for central roller cylinder*[26]

**Stress cycle:** Stress cycle refers to the repetitive loading and unloading of a material that occurs during cyclic loading conditions. It involves the fluctuation of stress levels within the material over time. Graph shows the exact stress cycle nature in roller cylinder, the values of $\sigma_m$ and $\sigma_a$ is calculated below.




$\sigma_{max}$ = 5.5 MPa
$\sigma_{min}$ = 0.030 MPa ~ 0 MPa
$\sigma_m = \frac{1}{2}(\sigma_{max} + \sigma_{min})$ = 2.75 MPa (mean stress)
$\sigma_a = \frac{1}{2}(\sigma_{max} - \sigma_{min})$ = 2.75 MPa (stress amplitude)
Fatigue strength of S275JR = 180 Mpa
Minimum safety factor =32.72 (safe from fatigue failure)

## 9.2 Central roller shaft

### 9.2.1 Static structural analysis

Boundary conditions:

| Parameters | | Loading |
|---|---|---|
| Component type | Simply supported beam on both ends | Point A: reaction |
| Analysis type | Static structural | Point B: reaction |
| Software | Ansys workbench 2023 R2 | Point C: 3898N radial |
| Cross section | Solid cylindrical steeped(40 mm and 45mm) | Point D: 3898N radial |
| Load | $L_C$ = 3898N(radial), $L_D$= 3898 N(radial) | Both ends are simply supported and only able to move axially |
| Material | 1.0044 (S275JR) Non-alloy structural steel | |
| Mesh element size | 20mm tetrahedron (minimum possible) | |

*Table 18: Boundary conditions for central roller shaft*

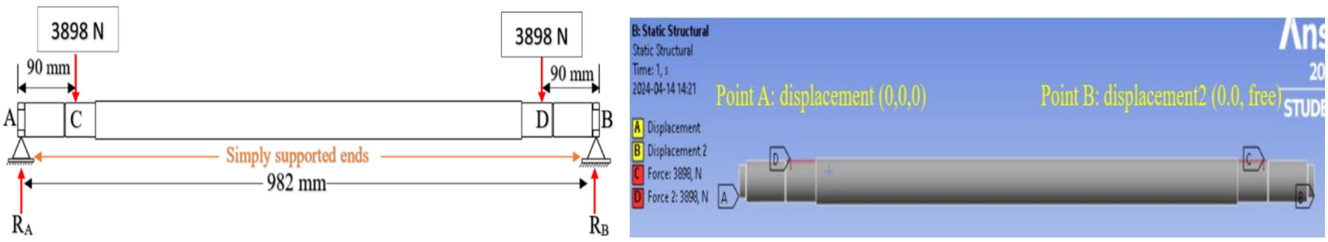



*Figure 51: Loading and boundary conditions for central roller shaft* [26]

Results

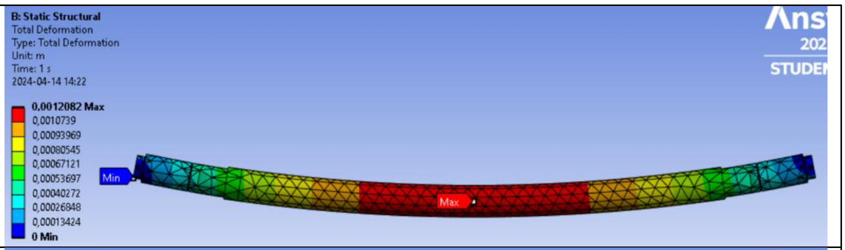
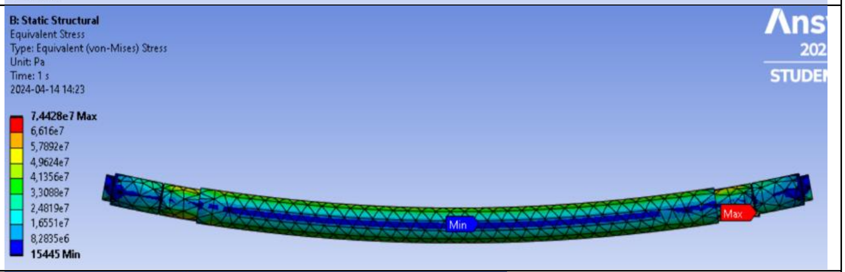
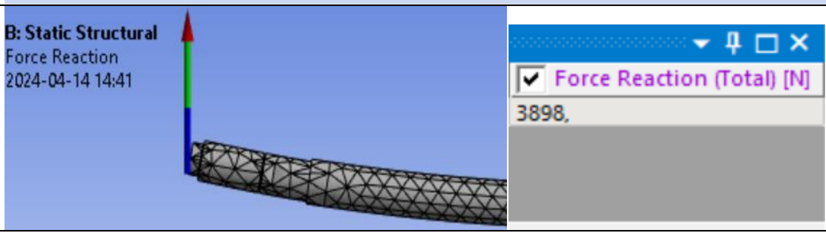
<table>
<tr><td>Total deflection:<br>• Max: 1.2082mm at centre<br>• Min: 0 mm at ends A and B<br>Ends are simply supported so deflection is zero and center of beam has maximum deflection.</td><td>
</td></tr>
<tr><td>Total equivalent stress(von misses):<br>• Maximum: 74.42 Mpa<br>• Minimum: 0.01544 Mpa<br>Maximum stress is recorded at section of circlip pin(ϕ37.5mm), min at neutral axis.</td><td>
</td></tr>
<tr><td>Reaction forces:<br>Reaction forces are equal on both ends with 3898N vertical due to symmetric loading and cross section of shaft from middle of length of the shaft.</td><td>
</td></tr>
</table>

*Table 19: Results from ANSYS for central roller shaft after static structural analysis*[26]

Comment on result:

The new concept of shaft has non uniform cross section and designed solid circular like in existing rollers-LKAB due to some constraints for the bracket size. 720 mm length from middle section of shaft has diameter 45 mm and rest ends are kept constant of diameter 40mm. We tried to decrease the deflection of shaft due to static load which will help bearing to increase its life and improve its performance and decrease transverse vibration due to deflection of the shaft. 1.2082 mm maximum deflection is witnessed at centre and 0 at both ends. The maximum stress of 74.4228 Mpa is recorded at the position of circlip pin(ϕ37.5mm) which is due to having maximum bending moment at that point

Minimum factor of safety ($SF_{min}$) = $\sigma_y/\sigma_{max}$ = 275/74.42 = 3.69

### 9.2.2 Modal (vibration) analysis

Modal analysis is a crucial step in dynamic analysis, providing insight into natural frequencies, mode shapes, and damping ratios. It's vital to examine the vibration behavior of the central roller shaft, especially due to its higher load compared to the side roller shaft. Ansys Workbench is utilized for numerical analysis to determine natural frequencies for different mode shapes. The first mode shape occurs at a frequency of 93.929 Hz, indicating potential failure if the natural frequency exceeds this value. To calculate the natural frequency under load, the dynamical systems formula for elastic members will be used.

**Boundary conditions:**

- Simply supported ends.
- Mesh element type: tetrahedrons.
- Mesh element size = 10 mm
- Total nodes = 24768
- Total elements= 15305
- Tool used= Ansys workbench (modal analysis)
- Material = 1.0044(S275JR structural steel, E =210G

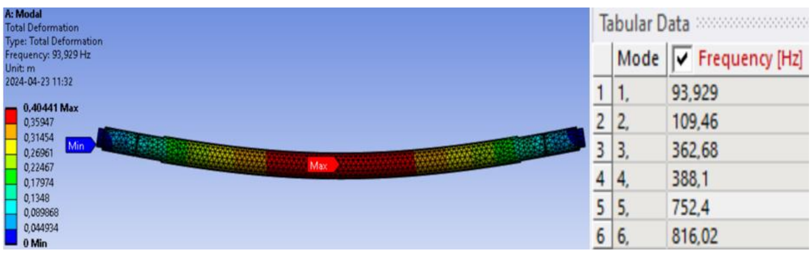
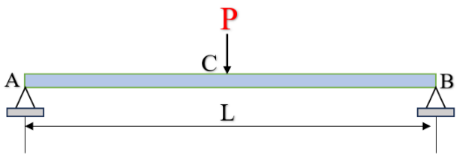



*Figure 52: Six mode shapes for central roller shaft from Modal analysis*[26]

**Analytical calculation:**

***Assumption:*** *The load on the shaft isn't centered, making it critical to determine its spring constant. Assuming the total load applies at the center can help find the maximum natural frequency and predict vibration levels. Using a uniform cross-section with the maximum diameter (45mm) simplifies calculations, as natural frequency is proportional to the radius squared ($f_n \propto r^2$).*




Load (P) = 7796 N

Length (L) = 0.982 m

Diameter (Ø) = 0.045 m

Young's modulus (E) = 210 GPa

Second moment of area (I) = $2.013\times10^{-7}$ m$^4$

Spring constant of simply supported beam

having point load at center (K) = $\frac{48EI}{L^3}$**[27]** $= \frac{48\times210\times10^9\times2.013\times10^{-7}}{0.982^3}$

= 2142742 N/m

External mass from load (m) = 7796/9.81 = 794.6 Kg

Natural frequency of the shaft due to vibration ($f_n$) = $\frac{1}{2\pi}\sqrt{\frac{K}{m}} = \frac{1}{2\pi}\sqrt{\frac{2142742}{794.6}}$ = 8.26 Hz [27]

*Here, the natural frequency ($f_n$) of the shaft due to 7796 N central load is very less than the frequency of first mode shape shown by Ansys, so the shaft is totally safe due to vibration. ($f_n$ = 8.26 Hz≪ 93.929 Hz).*

## 9.3 Side roller cylinder

### 9.3.1 Static structural analysis

Boundary conditions:

- Component type: Fixed ended beam element
- Cross section: hollow tube (OD 159, ID 153)
- a = 0.22 m, L = 0.6 m
- Analysis type: Static structural
- Software: ANSYS workbench 2023 R2
- Load(q) = (uniformely distributed 4873N/m)
  = 4873×0.22 = 1072 N (at 110 mm from end A)
- Material: 1.0044 (S275JR) structural steel (E = 210 Gpa, v = 0.28)
- Mesh: type = tetrahedron, size = 10 mm, Nodes = 40136, elements = 19922

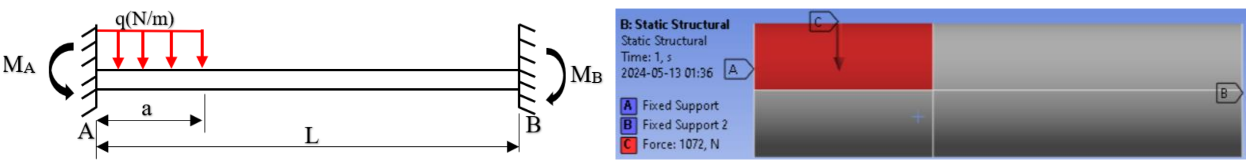



*Figure 53: Loading and boundary conditions for side roller cylinder*[26]

Results:

Case 1:

- Stress($\sigma_{max}$) = 3.63 Mpa (at fixed end)
- Safety factor (SF) = 75.75
- Deflection ($\delta_{max}$) = 0.0057 mm

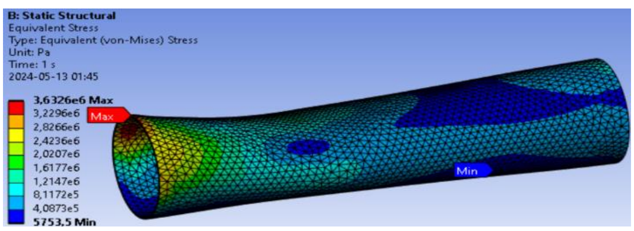



*Figure 54: Stress distribution for side roller Cylinder from ANSYS-case1*[26]

Case 2:

- Stress($\sigma_{max}$) = 3.63 Mpa (at fixed end)
- Safety factor (SF) = 75.75
- Deflection ($\delta_{max}$) = 0.0056 mm

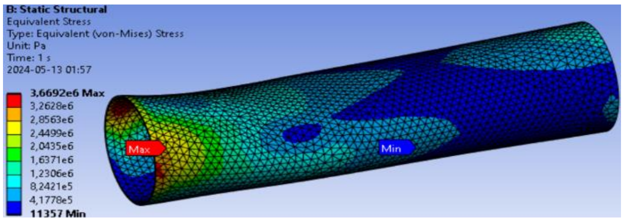



*Figure 55: Stress distribution for side roller Cylinder from ANSYS-case2*[26]

*Note: in case of side roller cylinder, the load is very low and the deflection is also low so it is not necessary to check for the vibration, we simply can assume that it is also safe from vibration.*

### 10.3.2 Modal (vibration ) analysis

In modal analysis of the side roller and bearing hub assembly, the bore of the bearing hub serves as a cylindrical support. This support is treated similarly to a fixed support, effectively making the entire part behave like a fixed-ended beam element.

**Boundary conditions:**

- Fixed ended beam element.
- Mesh element type: tetrahedrons.
- Mesh element size = 10 mm
- Total nodes =51115
  Total elements= 25188
- Tool used= Ansys workbench (modal analysis)
- Material = 1.0044(S275JR structural steel, E =210GPa, v = 0.28

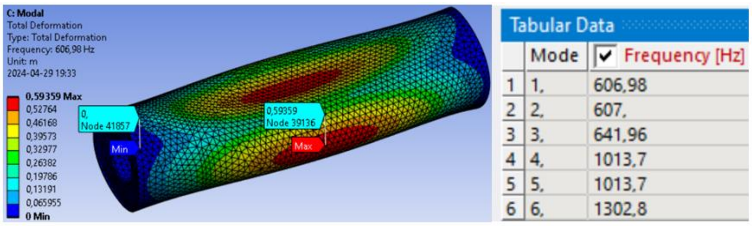



*Figure 56: Six mode shapes for side roller cylinder from modal analysis* [26]

Based on the figure, it appears that the first mode will occur at a frequency of 606.98 Hz. This frequency is relatively high, suggesting that the assembly is likely safe from vibration induced by the radial load of 1072 N. The reason for this safety assumption is that the applied load is substantially less than the load required to generate the natural frequency of 606.98 Hz.

## 9.4 Side roller shaft

### 9.4.1 Static structural analysis

Boundary conditions:

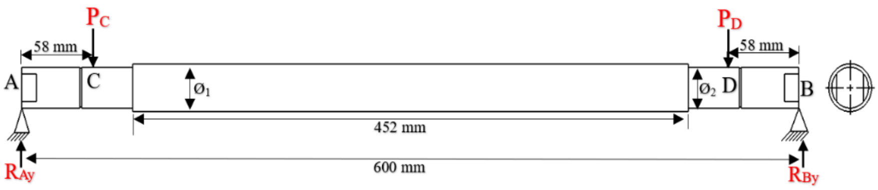



*Figure 57: Free body diagram of side roller shaft*

| Parameters | | Loading |
|---|---|---|
| Component type | Simply supported beam on both ends | $P_C$ = 908 N<br>$P_D$ = 229 N<br>$R_{Ay}$ = 846 N<br>$R_{By}$ =291 N<br><br>The axial load is not considered because shaft acts as a beam |
| Analysis type | Static structural | |
| Software | Ansys workbench 2023 R2 | |
| Cross section | Solid cylindrical steeped(30 mm and 35mm) | |
| Diameters | $Ø_1$ = 35 mm<br>$Ø_2$= 30 mm | |
| Material | 1.0044 (S275JR) Non-alloy structural steel | |
| Mesh element size | 20 mm tetrahedron (minimum possible) | |

*Table 20: Boundary conditions for side roller shaft*

Results

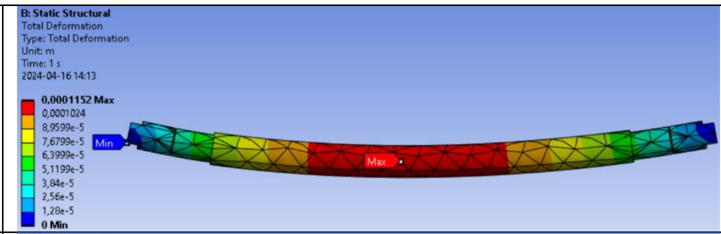
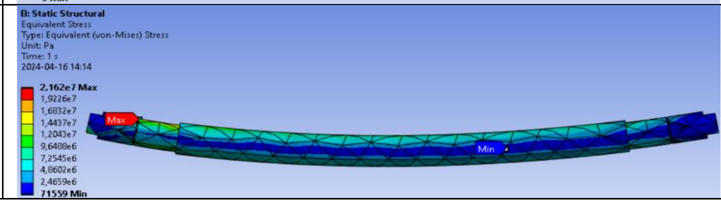
<table>
<tr>
<td>Total deflection:<br>• *Max:0.1152mm at centre*<br>• *Min: 0 mm at ends A and B*<br>Ends are simply supported so deflection is zero and center of beam has maximum deflection.</td>
<td>
</td>
</tr>
<tr>
<td>Total equivalent stress(von misses):<br>• *Maximum: 21.62 Mpa*<br>• *Minimum: 0.0.0715 Mpa*<br>Maximum stress is recorded at section of circlip pin(ϕ28.6mm), min at neutral axis.</td>
<td>
</td>
</tr>
</table>

*Table 21: Results from ANSYS for side roller shaft after static structural analysis* [26]

Minimum factor of safety ($SF_{min}$) = $\sigma_y/\sigma_{max}$ = 275/21.62 = 12.71

Comment on results:

The side roller shaft shares a similar design to the central roller shaft, with dimensions adjusted to minimize deflection and stress. After simulation, only 0.1152 mm deflection is observed at the center of the shaft, meeting design requirements. Maximum stress is recorded at the circlip pin location (ϕ28.6mm), with a minimum safety factor of 12.71, slightly above requirements. Despite concerns about potential failure in heavy-duty applications due to small deflections, further weight reduction is not feasible. Increasing shaft weight does not impact conveyor power requirements, as the shaft is stationary and does not consume power. Additionally, increased mass may help reduce vibration caused by deflections.

### 9.4.2 Modal (vibration) analysis

Likewise for central roller shaft in section 9.2.2, it is necessarry to do the modal analysis for side roller shaft to find its mode shapes in vibration. All the procedures and boundary conditions are same as for central roller shaft except load and diameter of the shaft. We have found following result from ansys workbench modal analysis.

**Boundary conditions:**

- Simply supported ends.
- Mesh element type: tetrahedrons.
- Mesh element size = 10 mm
- Total nodes = 9525
- Total elements= 5539
- Tool used= Ansys workbench (modal analysis)
- Material = 1.0044(S275JR structural steel, E =210 GPa

In the figure below, if the natural frequency of the disturbance reaches 175.2 Hz, the first mode shape will occur, leading to shaft failure. There are six different mode shapes, but priority is given to the first mode shape as it occurs first. If the first mode shape occurs, the shaft will deflect by 647 mm at the center. Analytical calculations will be conducted to verify its safety Under load.

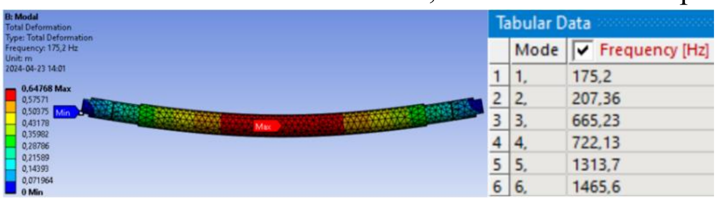



*Figure 58: Six mode shapes for side roller shaft obtained From modal analysis*[26]

**Analytical calculation:**

***Assumption:*** *the load on shaft is not at the center, so it is a bit critical to find its spring constant, we can assume the total load applies at the center and can help to find the minimum natural frequency. If we assume load at center the deflection will be more as well as the vibration due to it and helps to find the ultimate natural frequency. Moreover, the cross section of the shaft is not uniform so we can use uniform* *cross section with maximum diameter (35mm) because the natural frequency of spring-mass system directly proportional to the radius ($f_n \propto r^2$).*

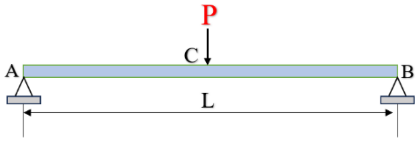
Load (P) = 1137 N

Length (L) = 0.632m

Diameter (Ø) = 0.035 m

Young's modulus (E) = 210 GPa

Second moment of area (I) = $7.366\times10^{-8}m^4$




Spring constant of simply supported beam

having point load at center (K) = $\frac{48EI}{L^3} = \frac{48\times210\times10^9\times7.366\times10^{-8}}{0.632^3}$**[27]**

$= 2941312$ N/m

External mass from load (m) = 1137/9.81 = 116 Kg

Natural frequency of the shaft due to vibration ($f_n$) = $\frac{1}{2\pi}\sqrt{\frac{K}{m}} = \frac{1}{2\pi}\sqrt{\frac{2941312}{116}} = 25.34$ Hz [27]

*Here, the natural frequency ($f_n$) of the shaft due to 7796 N central load is very less than the frequency of first mode shape shown by Ansys, so the shaft is totally safe due to vibration. ($f_n$ = 25.34 Hz≪ 175.2 Hz).*

# 10 Analytical calculations

## 10.1 Central roller

### A. Roller cylinder

Step 1: Calculation of reaction forces $R_A$ and $R_B$

Due to symmetric loading condition both reaction forces are equal

q = 8090 N/m  $R_A$ = 3762N, $R_B$ = 3762N

Step 2: Calculation of shear forc↑( ↓, -)

- Shear force at A to its left = $SF_{AL}$ = 0
- Shear force at A to its right = $SF_{AR}$ = 3762N
- Shear force at C to its left = $SF_{CL}$ = 3762N
- Shear force at C to its right = $SF_{CR}$ = 3762-7524 = -3762N
- Shear force at B to its left = $SF_{BL}$= -3762N
- Shear force at B to its right = $SF_{BR}$ = -3762+3762 = 0N

Step 3: Calculation of bending moment ( sagging effect= +, hogging effect = -)

- Bending moment at point A ($M_A$) = $-\frac{qL^2}{12}$ = -583 Nm
- Bending moment at point B ($M_B$) = $-\frac{qL^2}{12}$ = -583 Nm
- Bending moment at C, ($M_C$) = $\frac{qL^2}{24}$ = 291Nm

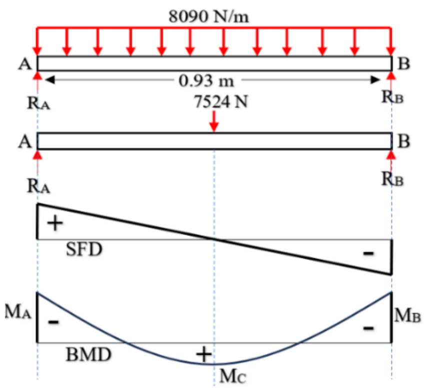



*Figure 59: Shear force and bending moment diagram for central roller cylinder*

## B. Roller shaft

### Shear force and bending moment

Step 1: Calculation of reaction forces $R_A$ and $R_B$

In case of roller shaft, the loads are applied on points.

Due to symmetric loading condition both reaction forces are equal to a single load that is,

$P_C = 3898$ N, $P_D = 3898$ N, $R_A = 3898$N, $R_B = 3898$N

Step 2: Calculation of shear forc↑( ↓, -)

- Shear force at A to its left = $SF_{AL} = 0$
- Shear force at A to its right = $SF_{AR} = 3898$N
- Shear force at C to its left = $SF_{CL} = 3898$N
- Shear force at C to its right = $SF_{CR} = 3898\text{-}3898 = 0$N
- Shear force at D to its left = $SF_{DL} = 0$N
- Shear force at D to its right = $SF_{DR} = 0 - 3898 = -3898$N
- Shear force at B to its left = $SF_{BL} = -3898$N
- Shear force at B to its right = $SF_{BR} = -3898+3898 = 0$N

Step 3: Calculation of bending moment ( sagging effect= +, hogging effect = -)

- Bending moment at point A = 0
- Bending moment at point B = 0
- Bending moment at C, $BM_C = + R_A \times 0.09 = 350.82$Nm
- Bending moment at D, $BM_D = + R_B \times 0.09 = 350.82$Nm

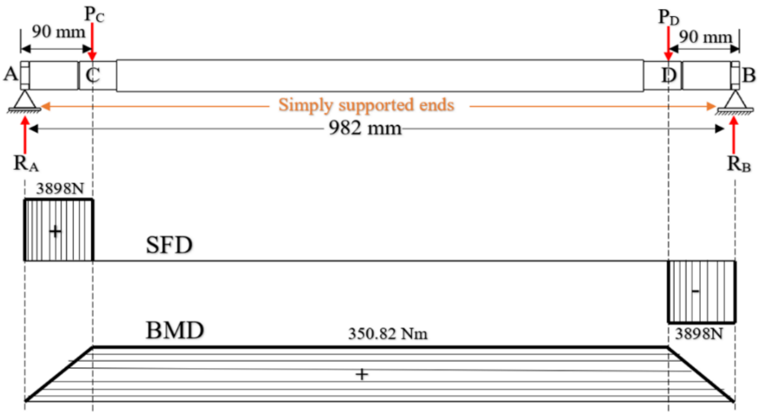
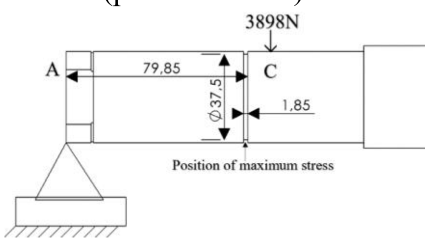



*Figure 60: Shear force and bending moment diagram for central roller shaft*

### Stress calculation in critical point

In bending of beams, the the stress generates along the cross section due to bending moment and depends on magnitude of bending moment(M), cross sectional area(A) and second moment of area(I).

In our beam AB the diameter is different along the length so the magnitude of stress is also different for each cross section. The stress at free ends A and B is zero, it is required to calculate the stress in minimum cross section (37.5 mm) and point of load (point C and D) to find the maximum stress.




We know,

$$\text{Stress}(\sigma) = \frac{\text{Mc}}{\text{I}}$$

where, M= bending moment, c= distance from neutral axis, I= second moment of area

At point C and D: Bending moment (M) =350.82 Nm (due to symmetric loading)

Distance from neutral axis (c) = 0.02 m

Second moment of area (I) = $1.256\times10^{-7}\text{m}^4$

$$\text{Stress } (\sigma_{C,D}) = \frac{Mc}{I} = \frac{350.82\times0.02}{1.256\times10^{-7}} = 55.86 \text{ MPa}$$

At section (Ø 37.5 mm):

Bending moment (M) = 3898 ×0.07985 = 311.25 Nm

Distance from neutral axis (c) = 0.01875 m

Second moment of area (I) = $9.70\times10^{-8}\text{m}^4$

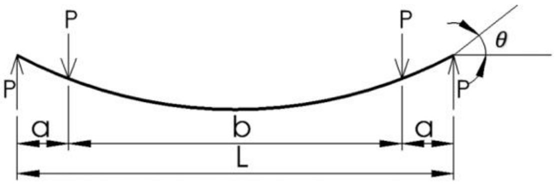
$$\text{Stress } (\sigma) = \frac{Mc}{I} = \frac{311.25\times0.01875}{9.70\times10^{-8}} = 60.16 \text{ MPa}$$

$$\text{Minimum safety factor (SF)} = \frac{\text{Yield strength}(\sigma_y)}{\text{Generated stress}(\sigma_{max})} = 4.57$$ *(safe from yield failure)*

The maximum stress observed at this point in ANSYS simulation was 74.42 MPa. There's a discrepancy between analytical and simulation results due to factors like element size, boundary conditions, and geometry complexity. Ansys simulations often provide approximate results, and in this case, the difference in maximum stress is 14.3 MPa. Despite this difference, the ANSYS result shows a sufficient safety factor of 3.69, indicating the stress calculated above is accurate.

### Deflection limitation on the basis of CEMA standard:

CEMA (Conveyor Equipment Manufacturers Association) recommends that the roller shaft should not deflect more than one degree**[28]**, the deflection of shaft can be a major cause for bearing failure. Here from a advisory paper by CEMA, we have used formula to find the shaft deflection as below.




For central roller shaft:

P = 3898 N

A = 90 mm = 0.09 m

L = 982 mm = 0.982m

B = 802 mm = 0.802m

E = 210 Gpa

I = $9.708\times10^{-8}\text{m}^4$

Considered minimum cross section at Circlip pin that is 37.5 mm diameter

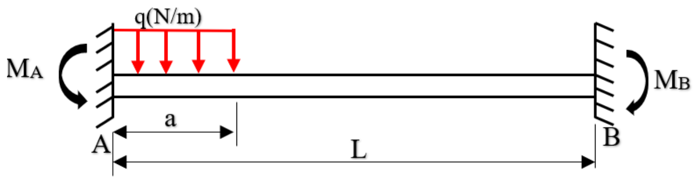
$$\theta = \frac{57.3Pab}{2EI} = 0.4° \text{ [28]}$$

(the angle of deflection is less than 1° so our design is within the CEMA limitation)

## 10.2 Side roller

### A. Roller cylinder

### Shear force and bending moment




q = 4873 N/m, a = 0.22 m., L = 0.6 m

$$(M_A) = -\frac{qa^2}{12L^2}(6L^2 - 8aL + 3a^2) = -68\text{Nm [26]}$$

$$(M_B) = -\frac{qa^3}{12L^2}(4L - 3a) = -21\text{Nm}$$

Step 1: Calculation of reaction forces $R_A$ and $R_B$

$R_A + R_B = 1072$ ...............(1)

$R_A \times 0.6 - 1072\times0.49 - M_A + M_B = 0$.......(2) $(\sum M_B = 0)$

On solving (1) and (2) we get,

$R_A = 954$ N

$R_B = 118$ N

Step 2: Calculation of shear force( ↑↓, -)

- Shear force at A = 954 N
- Shear force at C = 954-1072 = -118 N
- Shear force at B = -118 +118 =0N

Step 3: Bending moment

Considering simply supported beam

- The reaction forces, $R_A + R_B = 1072$ ............(1)
- $\sum B_B = 0$, $R_A \times 0.6 - 1072 \times 0.49 = 0$..............(2)
- On solving (1) and (2) $R_A = 875$ N, $R_B = 197$ N
- Beanding moment at A and B($M_A$, $M_B$) = 0
- Bending moment at C ($M_C$) = $R_A \times 0.22 - 1072 \times 0.11 = 75$ Nm

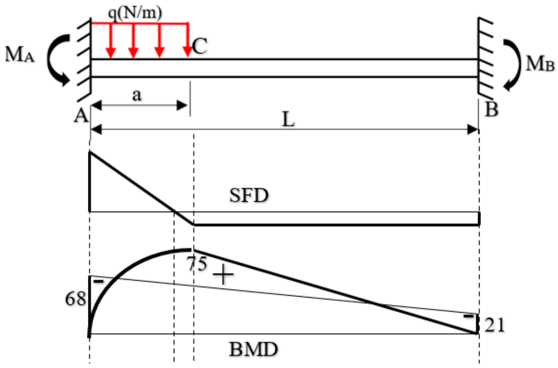



*Figure 61: Shear force and bending moment diagram for side roller cylinder*

## A. Roller shaft

## Shear force and bending moment

Step 1: Calculation of reaction forces $R_A$ and $R_B$

$P_C$ = **908 N**, $P_D$ = **229 N** $R_{Ay}$ = **846 N**, $R_{By}$ = **291 N**

Step 2: calculation of shear force( ↑↓ +, -)

- Shear force at A to its right = $SF_{AR}$ = 846N
- Shear force at C to its right = $SF_{CR}$ = 846-908 = -62N
- Shear force at D to its left = $SF_{DL}$ = -62N
- Shear force at D to its right = $SF_{DR}$ = -62-229 = -291N
- Shear force at B to its left = $SF_{BL}$ = -291N
- Shear force at B to its right = $SF_{BR}$ = -291+291 =0N

Step 3: Calculation of bending moment ( sagging effect= +, hogging effect = -)

- Bending moment at point A = 0
- Bending moment at point B = 0
- Bending moment at C, $BM_C = + R_{Ay} \times 0.058 = 49.068 \sim 49$Nm
- Bending moment at D, $BM_D = + R_{By} \times 0.058 = 16.878 \sim 17$Nm

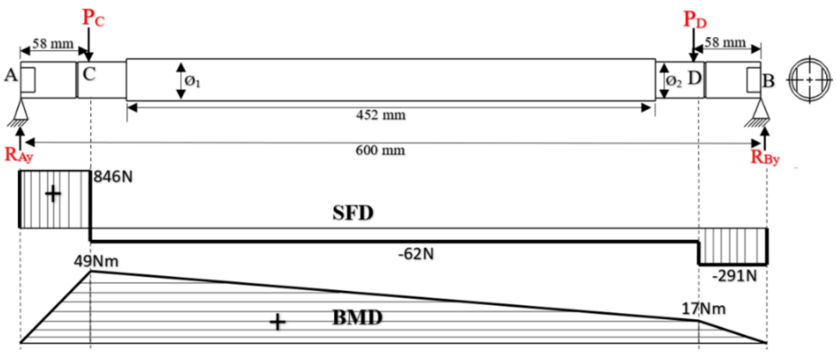
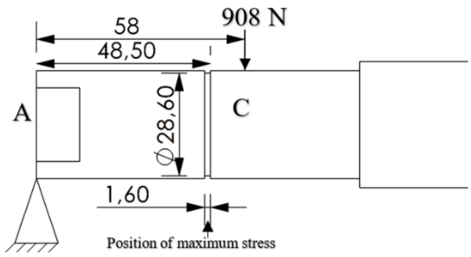



*Figure 62: Shear force and bending moment diagram for side roller shaft*

### Stress calculation in critical point

As  for the central roller shaft, it is required to calculate the stresses in critical locations analytically, the side roller shaft has same geometry as central roller shaft but the dimension and loading condition is different, here the minimum diameter at circlip pin is 28.6 mm so we will calculate the stress in point C and in the section of 28.6 mm diameter. There are two different sections having diameter 28.6 mm near point C and near point D but we found that bending moment is maximum at point C and ANSYS also gave the maximum stress at 28.6 mm diameter section near point C, so we can simply understand that the stress may be maximum near maximum bending moment.




We know,

$$\text{Stress}(\sigma) = \frac{Mc}{I}$$

At point C: Bending moment (M) = 49 Nm

Distance from neutral axis (c) = 0.015 m

Second moment of area (I) = $3.976\times10^{-8}m^4$

$$\text{Stress } (\sigma_C) = \frac{Mc}{I} = \frac{49\times0.015}{3.976\times10^{-8}} = 18.5 \text{ MPa}$$

At section (Ø 28.6 mm):

Bending moment (M) = 846 ×0.0485 = 41.031 Nm

Distance from neutral axis (c) = 0.0143 m

Second moment of area (I) = $3.284\times10^{-8}m^4$

$$\text{Stress } (\sigma) = \frac{Mc}{I} = \frac{41.031\times0.0143}{3.284\times10^{-8}} = 17.86 \text{ MPa}$$

We have maximum stress ($\sigma_{max}$) = 18.5 Mpa at point C, there is some difference between ANSYS result and analytical result that ANSYS gave maximum stress of  21.6 Mpa at (Ø 28.6 mm) this is because in ANSYS simulation we used the boundary condition at flat end in place of cylndrical surface due to some difficulties to apply boundary condition. So the analytical calculations are correct and the ANSYS results.are also considerable for best approximate results.

$$\text{Minimum safety factor (SF)} = \frac{\text{Yield strength}(\sigma_y)}{\text{Generated stress}(\sigma_{max})} = 14.86 \textit{ (safe from yield failure)}$$

## 10.3 Weight comparison of existing and new rollers

| | Components | | Side roller(gm) | | Central roller(gm) | |
|---|---|---|---|---|---|---|
| Existing Roller | Rotating parts | Cylinder | 10166 | Total 13272 | 27432 | Total 34334 |
| | | Bearing hub | 2076 | | 4634 | |
| | | Bearings | 680 | | 1260 | |
| | | Sealings | 350 | | 1008 | |
| | Non rotating parts | shaft | | 3462 | | 9596 |
| | Total | | | 16734 | | 43930 |
| New roller | Rotating parts | Cylinder | 6880 | Total 9776 | 21436 | Total 27334 |
| | | Bearing hub | 1840 | | 3620 | |
| | | Bearings | 706 | | 1270 | |
| | | Sealings | 350 | | 1008 | |
| | Non rotating parts | shaft | | 4361 | | 11470 |
| | Total | | | 14137 | | 38804 |

*Table 22: Weight comparison of existing and noe roller components*

Central roller:

- Mass saved in rotating parts = 7 Kg (20.38%).
- Total mass saved in one roller = 5.12 Kg (11.6%)

Side roller:

- Mass saved in rotating parts = 3.5 Kg (26.34%).
- Total mass saved in one roller = 2.6 Kg (15.5%)

Overall:

- Mass saved in rotating parts for one assembly (2 side rollers + 1 central roller) = 14 Kg (29.4%).
- Total mass saved in one roller assembly = 10.3 Kg (13.33%).

## 10.4 Belt tension(pull) and power calculation

To calculate the force or power needed to move the conveyor belt at a constant speed, we must first determine the length of the conveyor unit. In LKAB's facility, various carrying conveyor units are used with different lengths, inclinations, roller and belt sizes, capacities, and purposes. Our focus is on the "15101 smst TR010" section, which comprises a 620-meter-long conveyor belt. Longer conveyor units require more power due to increased material to move, heavier idlers, friction, and longer belts. The power consumption of the existing system in a horizontal configuration will be calculated and compared to the power required for the new roller design.

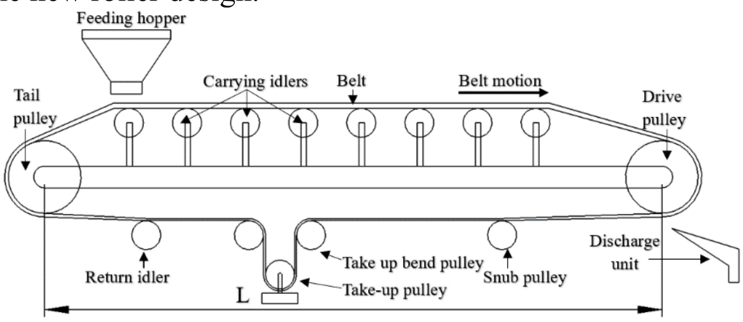



*Figure 63: Graphical representation of "15101 smst TR010" section-LKAB*

L = conveyor length in LKAB at "15101 smst TR010" section
= 620 meter

### Calculation of overall mass in "15101 smst TR010"

| Parameters | New design | Existing system |
|---|---|---|
| Length of longest conveyor | 620 meter | 620 meter |
| Idler spacing (a) | 1.2 meter | 1.2 meter |
| Total no of idler set(N) | 516 | 516 |
| Total no of side rollers($N_S$) | 1032 | 1032 |
| Total no of central roller($N_C$) | 516 | 516 |
| Total weight of side rollers | 14626 Kg | 17270 Kg |
| Total weight of central roller | 20022 Kg | 22668 Kg |
| Overall weight of rollers | 34648 Kg | 39938 Kg |

*Table 23: Calculation of the overall mass in "15101 smst TR010"*

### Belt tension:

To determine the maximum tension, it's essential to compute the effective tension, which represents the force needed to maintain constant speed for both the conveyor and the load. This calculation considers the foundational conditions required to overcome frictional resistance and tension forces. We will find the belt pull requirement and power consumption for existing case and new design to find the magnitude of energy efficiency for new design. Essentially. Effective tension comprises:[29]

- The tension to move with an empty belt, $T_x$
- The tension to move the load horizontally, $T_y$
- The tension to increase or decrease the load, $T_z$ (in inclined conveyor)
- The tension to overcome the resistance of accessories, $T_{us}$
- The tension to overcome the resistance of scrapers, $T_{uc}$

**A. The tension to move with an empty belt $T_x$** [29]

$$T_x = G \times f_x \times L_c$$

G = mass per unit length of the moving parts (belt+ rotating roller components)
=34.21+39 = 73.21 ~ 73 kg/m
=34.21+51 = 85.21 ~ 85 kg/m ***(existing case)***
$f_x$ = coefficient of friction for empty belt = 0.020 (from Table 4: appendix A)

$L_c$ = belt length correction: Short belt conveyors require relatively more power to overcome the resistance to friction than long ones and therefore an adjustment is made to calculate.
The effective tension, $L_c$ = L + 70 m = 620+70 = 690 meter
$T_x$ = 73 × 0.020 ×690 = 1007.4 Kg~ 1007 Kg
$T_x$ = 85 × 0.020 ×690 = 1173Kg ***(existing case)***

**B. The tension to move the load horizontally $T_y$.**

$$T_y = Q\ (W/m) \times f_y \times L_c\ [29]$$

Q(W/m) = mass per unit length of conveying material (iron ore) = 862 Kg/m
$f_y$ = coefficient of friction for loaded belt = 0.020 (from Table 4: appendix A)
$T_y$ = 862× 0.020 × 690 = 11896 Kg ***(same in new and existing case)***

**C. The tension to increase or decrease the load $T_z$** (in inclined conveyor) = 0 (no inclination)[29]

**D. The tension needed to overcome the resistance of the skirtboards $T_{us}$.**

$$T_{us} = \frac{fs \times Q \times Ls}{V \times b_s}\ [29]$$

$f_s$ = coefficient of friction for skirtboards = 0.650 (from Table 4: appendix A)
Q = 862 Kg/m
$L_s$ = length of skirtboard = 6 meter (considered only one SILA section out of twelve)
$b_s$ = height of skirtboard = 0.970m
v = belt speed = 2.9m/s

$$T_{us} = \frac{fs \times Q \times Ls}{V \times b_s} = 1195\ \text{Kg}$$

**E. The tension to overcome the resistance of scrapers $T_{uc}$.**
No scrapers are indicated by LKAB, so the tension (Tuc) to overcome the resistance of scrapers is zero.

Effective tension ($T_e$) = $T_x + T_y + T_z + T_{us} + T_{uc}$ = 1007+ 11896+ 0+ 1195 = 14098 Kg
($T_e$) = $T_x + T_y + T_z + T_{us} + T_{uc}$ = 1173+ 11896+ 0+ 1195 = 14264 Kg ***(existing case)***
Other tensions to be considered are:
**Slack side tension $T_m$:** minimum tension to prevent belt slip with drive pulley:

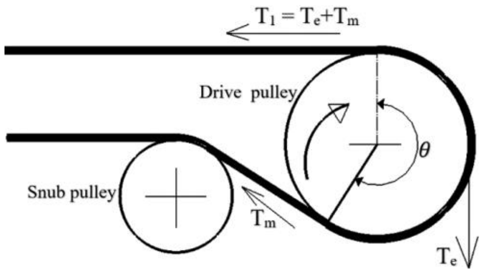



*Figure 64: Slack tension $T_m$ in conveyor belt*[29]

$T_m = k \times T_e$ where k = wrap factor = 0.38 for 210° wrap and lagged pulley (Table 5: appendix A)
$T_m = 0.38 \times 14098 = 5357$ Kg
$T_m = 0.38 \times 14264 = 5421$ Kg ***(existing case)***

**Power loss in bearing due to friction.**

**New design**

Power loss in single bearing 6308-2Z-C3 = 3.3 W *[see table 1 appendix C]* [30]
Power loss in whole conveyor with 6308-2Z-C3 = 3405 W
Power loss in single bearing 6306-2Z = 4.5 W *[see table 3 and table 4 appendix C]* [30]
Power loss in whole conveyor with 6306-2Z = 4644 W
Overall power loss in bearings ($P_{loss}$) = 8049W

**Existing bearings**

Overall power loss in bearings $(p_{loss})_{existing}$ = 8049 W *[see table 2, table 5 and table 6 in appendix C]* [30]

Now the total tension required to pull the conveyor belt,

$T_1 = T_e + T_m = 14098 + 5357 = 19455$ Kg = 190853 N
$T_1 = T_e + T_m = 14278 + 5421 = 19699$ Kg = 193247 N ***(existing case)***

Power required (P) = $T_1 \times$ belt speed + $P_{loss}$ = 561522 watt
Power required (P) = $T_1 \times$ belt speed +$(p_{loss})_{existing}$ = 568466 watt ***(existing case)***
Power saved = 568466 - 561522 = 6944 watt
Power saved in one hour in KWH (kilowatt-hour) = 6.944~7

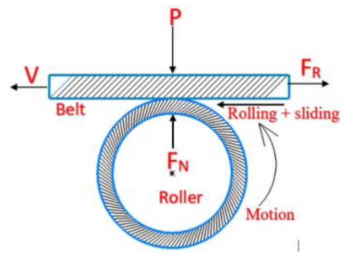
Horsepower required (HP) = 753
Horsepower required (HP) = 762 ***(existing case)***

## 10.5 Roller cylinder wear analysis:

Wear is the gradual loss of material from a surface due to relative motion between that surface and another substance. It encompasses various processes, including material displacement within the surface and material removal. This occurs when the surfaces slide or roll against each other, with or without lubrication, and when the surfaces are abraded by hard particles or eroded by solid or liquid impacts. In case of conveyor roller, there is rolling and sliding motion between belt and roller cylinder which results wear of belt and roller material. Archard's wear equation is commonly used to quantify wear under dry rolling-sliding conditions, expressing wear volume(W) per unit sliding distance(L) as the product of a non-dimensional wear coefficient(K), applied load(P), and reciprocal of material hardness(V) [31].

P = vertical load on roller = 7524N
V = belt speed = 2.9m/s
$F_R$ = rolling friction
$F_N$ = normal force = 7524 N
Archard's wear equation,

$$\frac{W}{L} = K \times \frac{P}{Hv} \text{ .............(1) [31]}$$




Archard's wear equation is usually an approximation since there are a number of assumptions into it.
W = wear volume ($mm^3$)
Hv = Vickers hardness of structural steel (in MPa) = 1373 MPa
K = wear coefficient of structural steel = $1.7\times10^{-5}$ (approximate) [32]
L = circumference of cylinder for 1 rotation of central roller = 609.4 mm

*Note: there is no any specific procedure to calculate the exact amount of sliding velocity between roller and belt because it depends upon the various factors such as, belt surface roughness, hardness, coefficient of friction, environmental condition etc., the equation above gives the wear for unit sliding distance, first we will calculate wear for pure sliding case and estimate that there is 90% rolling and 10% sliding.*

So, from (1), for pure sliding case, $W = 1.7 \times 10^{-5} \times \frac{7524\times609.4}{1373} = 0.05677\ mm^3$
For 10% sliding, $W_{final} = 0.005677\ mm^3$ per revolution of central roller
Wear per hour = 97.24 $mm^3$
Wear per day = 2334 $mm^3$
Wear per year = 851822 $mm^3$
Original volume of central roller cylinder = 2748202 $mm^3$
Material left after one year = 1896380 $mm^3$
Percentage wear per year = $\frac{851822}{2748202} \times 100\% = 31\%$
Cylinder wall thickness wear in one year = 1.53 mm

In case of side roller, total vertical load (P) = 1072 N
Belt speed (V) = 2.9 m/s
Normal force ($F_N$) = 1072 N
L = circumference of cylinder for 1 rotation of side roller = 500 mm
from (1), for pure sliding case, $W = 1.7 \times 10^{-5} \times \frac{1072\times500}{1373} = 6.636\times10^{-3}\ mm^3$
For 10% sliding, $W_{final} = 6.636\times10^{-4}\ mm^3$per revolution of central roller
Wear per hour = 13.93 $mm^3$
Wear per day = 335 $mm^3$
Wear per year = 122086 $mm^3$
Original volume of side roller cylinder = 882159 $mm^3$
Material left after one year = 760073 $mm^3$
Percentage wear per year = $\frac{122086}{882159} \times 100\% = 13.83\%$
Cylinder wall thickness wear in one year = 0.40 mm

*Note: this wear prediction is calculated for the case if the conveyor runs 24 hours a day, 30 days a month and 365 days a year.*

# 11 Proposed solution for dust and water

In response to the three major issues identified at LKAB, we'll propose protective covers for roller ends. We'll utilize a systematic product design process and methodology to develop these solutions.

### Problem identification:

- Excessive groundwater causes roller bearings to rust quickly, leading to roller jamming.
- Iron ore pallets dropping outside the belt during transfer can become lodged between the roller end and bracket support, causing further jamming.
- Groundwater creates muddy floors, which dry and turn into dust, promoting rust and bearing failure around the roller.

### General requirements:

- Compact, easy to use and maintenance, cost effective.
- Very simple in concept and easy to mount and dismount.
- Rust free, working temperature (-20°C to 100°C).
- No alterations can be made to the roller bracket, requiring the product to feature self-alignment.

### Concept development:

After careful consideration of the issues and needs outlined above, we've created roller caps to affix to the end of each roller. These caps can be mounted onto the bracket without affecting the roller's functionality or requiring any modifications to the bracket itself. We have distinct caps for the central roller and the side rollers, each with its own mounting mechanism. These caps have been designed in a straightforward manner, making them incredibly easy to install.

- **Cap for cetral roller**

The cap shown in the picture below is for the central roller, it has 5 parts namely, covers(left & right), bolts, nuts and lock plate. It is very simple and can mount easily without taking off the roller, can mount on the bracket supporting with shaft's end without any disturbance to the conveyor operation.

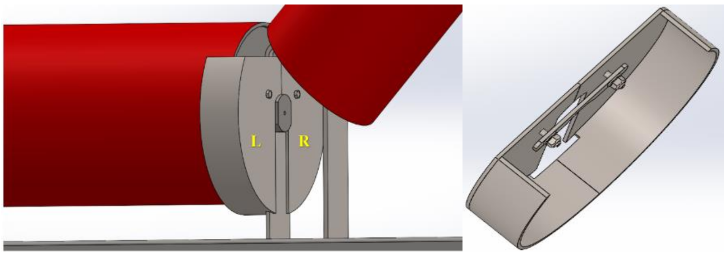



*Figure 65: Central roller cap assembled with roller*

### Working principle and reliability:

As in figure above, the covers written L and R are left and right respectively, these covers are supported on shaft and bracket with a slot provided, both parts are separate and can mount easily. There is a lock plate having two 5 mm holes and used to tight covers with bolt and nut from inside as in figure. The main function of this concept is to prevent the rollers end and bearing from dust, water and dropped pallet. It is not fully round due to having belt over the roller and belt will somehow help to prevent of entering water and dust from top side. Material used is AISI 316 stainless steel sheet(3mm ), which is totally rust free and very strong. AISI 316 stainless steel has very good processibility and raw material is easily available. It can manufacture easily from sheet metal and relatively cheaper to produce. Other features and parameters are mentioned below.

- Sheet thickness: 3mm
- Overall weight: 1150 gm
- Bolts (2 pieces, M5)
- Nuts(2 pieces, M5)
- No of item in single roller = 2
- No of item in whole conveyor (15101 smst TR010) =1032

### Components:

- Left and right covers:
  Figure 3(a) in appendix A are two side covers and are separate each other. Both are of same material, dimension and thickness.

- Covers lock plate
  Figure 3(b) in appendixA is the cover lock plate, is 3 mm thick and 100 mm long, made of stainless steel.
- Bolts and nuts
  Figure 4 in appendix A are a set of bolt and nut of same standard(M5) **.**

➢ **Cap for side roller**

For the side roller, a protective cap made of 2 mm thick AISI 316 stainless steel is proposed to address dust, water, and pallet ingress. Similar to the central roller cap, it features a single main part with two top slots to fit the bracket's top end and a cap lock plate for secure fastening. This solution aims to mitigate environmental hazards without impeding conveyor system functionality.

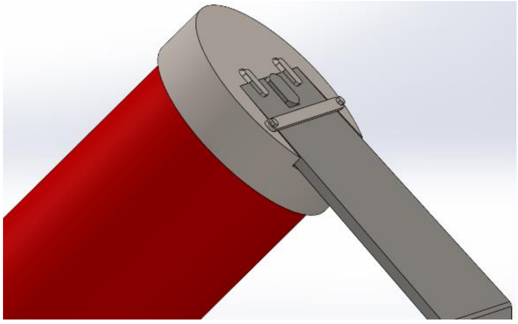


*Figure 66: Side roller cap assembled with roller bracket*

**Working:**

The cap is affixed to the roller bracket's top end using bolts and secured with a cap lock plate. It shields the sensitive internal components from dust and water ingress without impeding roller function, particularly at the bearing hub side. Additional details and specifications are outlined below.

- Sheet thickness: 2mm
- Material: AISI 316 stainless steel sheet
- Overall weight: 685 gm
- Bolts (2 pieces, M5)
- Nuts(2 pieces, M5)
- No of item in single roller = 2
- No of item in whole conveyor (15101 smst TR010) =2064

**Components:**

- Circular plate section with rectangular four strips (figure 5: Appendix A)
- Cylindrical tube section (figure 6: appendix A)
- Cap lock plate (figure 7: appendix A)
- Bolt and nut set (figure 5: appendix A)

# 12 Detailed design

## 12.1 Product specifications

### A. Shaft, bearing and housing tolerances

The table below outlines the crucial tolerance specifications necessary to ensure proper mating of parts, particularly for achieving an intereferenc fit, as required in conveyor rollers. This data has been derived from SKF's tolerance specifications for bearings, shafts and bearing housings. SKF has designed the barings specifying tolerance for all the mating components with their bearings. We have used SKF 6306-2Z and 6308-2Z-C3 bearings so need to use the tolerance provided by SKF for shaft and bearing housing too.

| Side roller | Shaft outer diameter(mm) | | Bearing bore(mm) | | Bearing outer diameter(mm) | | Housing bore (mm) | |
|---|---|---|---|---|---|---|---|---|
| | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. |
| | 30+0.041 | 30+0.028 | 30 | 30-0.01 | 72 | 72-0.013 | 72-0.026 | 72-0.045 |
| Fit | Interference(P6, r6) | | | | Interference(P6, r6) | | | |
| Central roller | 40+0.05 | 40+0.034 | 40 | 40-0.012 | 90 | 90-0.015 | 90-0.03 | 90-0.052 |
| Fit | Interference(P6, r6) | | | | Interference(P6, r6) | | | |

*Table 24: Shaft, bearing and housing tolerances*[33]

## 12.2 Bill of materials

| No | Item name | Part no | Qty. | Material | Dimension |
|---|---|---|---|---|---|
| 1 | Cylinder-1 | 1 | 1 | S275JR | Ø(194,184) |
| 2 | Shaft-1 | 2 | 1 | S275JR | L 982 |
| 3 | Shaft-2 | 3 | 2 | S275JR | L 632 |
| 4 | Cylinder-2 | 4 | 2 | S275JR | Ø(159,151) |
| 5 | Bearing hub-1 | 5 | 2 | S275JR | |
| 6 | Bearing bub-2 | 6 | 4 | S275JR | |
| 7 | Bearing-6306-2Z | 7 | 4 | Bearing steel | 72×30×19 |
| 8 | Bearing-6308-2Z-C3 | 8 | 2 | Bearing steel | 90×40×23 |
| 9 | Inner lip seal-1 | 9 | 2 | Nylon | 90×40×12 |
| 10 | Inner lip seal-2 | 10 | 4 | Nylon | 72×30×7 |
| 11 | Circlip pin-1 | 11 | 2 | Stainless steel | Ø(40.5,36.5) |
| 12 | Circlip pin-2 | 12 | 4 | Stainless steel | Ø(31.9,27.9) |
| 13 | Female labyrinth-1 | 13 | 2 | Nylon | 90×54×18 |
| 14 | Female labyrinth-2 | 14 | 4 | Nylon | 72×42×12 |
| 15 | Male labyrinth-1 | 15 | 2 | Nylon | 80×40×16 |
| 16 | Male labyrinth-2 | 16 | 4 | Nylon | 64×30×12 |
| 17 | Metal dust cover-1 | 17 | 2 | Carbon steel | 90×52×23 |
| 18 | Metal dust cover-2 | 18 | 4 | Carbon steel | 72×38×16 |
| 19 | Protective cover-1 | 19 | 2 | Natural rubber | 90×40×40 |
| 20 | Protective cover-2 | 20 | 4 | Natural rubber | 72×30×23 |
| 21 | Central roller cap | 21 | 2 | Stainless steel | 216×50×3 |
| 22 | Side roller cap | 22 | 2 | Stainless steel | 174×43×2 |
| 23 | Nuts | 23 | 8 | Stainless steel | M5, |
| 24 | Bolts | 24 | 8 | Stainless steel | M5, 16 |
| 25 | Cap lock plate -1 | 25 | 2 | Stainless steel | 160×15×3 |
| 26 | Cap lock plate -2 | 26 | 2 | Stainless steel | 145×15×3 |

*Table 25: Bill of materials*

## 12.3 Engineering drawings

All the drawings of main parts and their components is attached in Appendix. C.

# 13 Fabrication and assembly

## 13.1 Process selection and cost

Ashby emphasizes the significance of process selection in engineering design, highlighting its role in realizing the desired product efficiently. He presents a strategy in his book outlining the correlation between material properties, processes, and costs (Figure 7.1 5$^{th}$ edition). This strategy serves as a guide for selecting the most suitable manufacturing process for the rollers we've designed. With four main components to fabricate, we've identified processes that can be applied to groups of components for efficient production.

Components to manufacture[34].

- Roller shaft
- Roller cylinder
- Bearing hub
- Roller cap

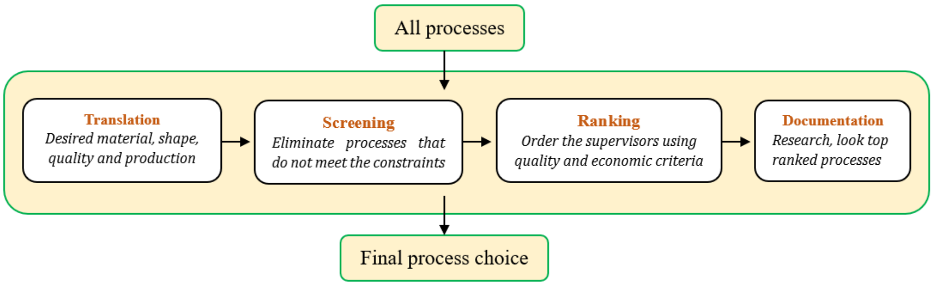



*Figure 67: Flow chart of the procedure for process selection*[34].

**Translation: (**See figure 1 Appendix D)

| Function: | | Shaft | Cylinder | Bearing hub | Cap |
|---|---|---|---|---|---|
| | | *Bending shaft* | *Cylinder* | *Bearing hub* | *Roller end cap* |
| Objective: | | *Minimize cost, time* | *Minimize cost, time* | *Minimize cost, time* | *Minimize cost, time* |
| Constraints: | *Material:* | *1.0044(S275JR structural steel), ferrous metal* | | | *Stainless steel* |
| | *Shape:* | *3D solid* | *Prismatic circular* | *Dished sheet* | *Folded sheet* |
| | *Mass:* | *4.36 – 11.4 Kg* | *9-21.4 Kg* | *0.9-2.1 Kg* | *0.6-1.2 Kg* |
| | *Min section:* | *1.5 mm* | *1 mm* | *4 mm* | *2 mm* |
| | *Tolerance:* | $< 0.013\ mm$ | $< 0.019\ mm$ | $< 0.019\ mm$ | $< 1mm$ |
| | *Roughness:* | $< 5\mu m$ | $< 5\mu m$ | $< 5\mu m$ | $< 5\mu m$ |
| | *Batch size:* | *1200* | | | |
| Free variable: | | *Choice of manufacturing process, and process operating conditions* | | | |

*Table 26: Translation for selecting process on the basis of material matrix*

**Screening:** (See figure 2 Appendix D)

| Screening of the process compatibilities for translation | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Compatible with ferrous metals (Figure 1 Appendix D) | | | Compatibility with shape (Figure 2 Appendix D) | | | | Compatibility with batch size 1200 (Figure 3 Appendix D) | | Quality comment |
| Shaft, cylinder | Bearing hub | Roller cap | Shaft | Cylinder | Bearing hub | Roller cap | Shaft, cylinder, bearig hub and roller cap | | |
| *Sand casting* | | | *Pass* | *Pass* | | | *Pass* | | *Meets requirements for finish and tolerance* |
| *Investment casting* | | | *Pass* | *Pass* | | | | | |
| *Centrifugal casting* | | | | *Pass* | | | | | |
| *Forging* | | | *Pass* | *Pass* | | | *Pass* | | |
| *Sheet forming* | | | | *Pass* | *Pass* | *Pass* | *Pass* | | |
| *Powder methods* | | | *Pass* | *Pass* | | | *Pass* | | |
| *Electro-maching* | | | *Pass* | *Pass* | | *Pass* | | | |
| *Machining* | | | *Pass* | *Pass* | *Pass* | *Pass* | *Pass* | | |
| *Additive manufacture* | | | *Pass* | *Pass* | *Pass* | *Pass* | | | |

*Table 27: Eliminating non fevourable processes on the basis of process-shape matrix*

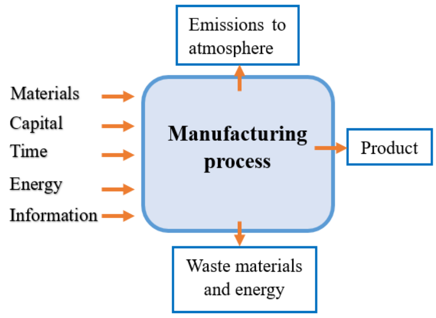
**Selection:**

| Component | Shaping process | Joining process | Finishing process |
|---|---|---|---|
| Roller shaft | *Maching* | *Not required* | *Precision maching* |
| Roller cylinder | *Maching* | *Welding, metals* | *Grinding, polishing* |
| Bearing hub | *Sheet forming* | *Welding, metals* | *Polishing* |
| Roller cap | *Sheet forming* | *Welding, fastners* | *Polishing* |

*Table 28: Final selection of fabrication process for each roller components*

## Ranking: The cost modeling:

According to Michael F. Ashby, cost prediction is integral to engineering design, reflecting the consumption of resources throughout production. While conducting a detailed cost analysis for our roller project may be complex, we can estimate costs by considering the primary factors outlined in Ashby's book. With four main components shaft, cylinder, bearing hub, and roller cap. we'll estimate costs individually, factoring in the shaping process and component size for both side and central rollers. [35].




## Material cost ($C_1$):

We have two different category of materials, 1.0044 (S275JR) structural steel and AISI 316 stainless steel and need to find unit price of each material, before that it is required to calculate the total amount of material required for each component for 1200 unit of batch size which is given in the table below.

| Material required (kg) (1.0044 S275 JR) (For 1200 unit) | | | | | | Material required(kg) (AISI 316 stainless steel) | |
|---|---|---|---|---|---|---|---|
| Roller shaft | | Cylinder | | Bearing hub | | Roller cap | |
| Central | Side | Central | Side | Central | Side | Central | Side |
| 14677 | 5760 | 25956 | 8256 | 2640 | 1080 | 1800 | 1200 |
| Total = 58369 Kg | | | | | | Total = 3000 Kg | |

*Table 29: Cost of material \$/Kg for 1200 unit of each component*

1.0044(S275JR) structural steel price per unit $(C_m)_1$ = 0.95 \$/Kg [36]

AISI 316 stainless steel sheet price per unit $(C_m)_2$ = 4.4 \$/Kg [37]

Total material cost (for 1200 unit of each component) ($C_1$) = 55450 \$ + 13200 \$ = 68650 \$

## Capital cost ($C_2$):

| Equipment | | | Tools | | |
|---|---|---|---|---|---|
| Item name | Quantity | Price (\$) | Item name | Quantity | Price (\$) |
| Band sawing machine | 1 | 7700 | Band sawing blade | 5 | 100 |
| CNC lathe machine | 1 | 15000 | Turning tool | 5 | 150 |
| CNC welding machine | 1 | 10000 | Welding electrode | 20 Kg | 300 |
| Sheet stamping machine | 1 | 10000 | - | - | - |
| Deep drawing machine | 1 | 4000 | Punching die | 2 | 2000 |
| Sheet bending machine | 1 | 900 | - | - | - |
| Hydraulic press machine | 1 | 2000 | - | - | - |
| Enamel painting machine | 1 | 2000 | Enamel | 10 Kg | 50 |
| Total cost of equipment ($C_c$) = 51600 \$ | | | Total cost of tooling ($C_t$) = 2600 \$ | | |

*Table 30: Estimation of tooling cost for each roller componets* [38]

Capital cost ($C_2$) = equipment cost ($C_c$) + tooling cost ($C_t$) = 51600 \$ + 2600 \$ = 54200 \$

*Note: the capital cost calculated above is for batch size of 1200 for each component where the equipment cost is initial set up cost, the maintenence cost for equipment has not included and tooling cost is variable cost. Every batch will require 2600 \$ for tools.*

**Time cost ($C_3$):** Time cost is the cost of space or rent, labor cost, administration cost which is normally calculated in per hour as overhead rate but we are going to estimate overall time cost for 1200 batch size. Considering maximum 15 minute required to produce one roller set including cylinder, shaft, bearing hubs and roller caps it requires 18000 minutes for 1200 sets. So the total hour required to complete one batch is 300 hr.

No of labor required in one hour = 10
Labor cost per hour in Norway = \$19
Total labor cost for 300 hour = \$57000
Total administration and rent cost = \$10000
Overall time cost ($C_3$) = \$67000

**Energy cost ($C_4$):** energy cost is the cost of fuel and electricity.

Energy cost ($C_4$) = \$10000

**Information cost ($C_5$):** it is the cost of research & development, royalty payments and we have roughly estimated for one batch of 1200 unit of each roller component. Normally royalty cost is calculated in per year system but we can divide it into fraction time which required to complete one batch.

Information cost ($C_5$) = \$ 10000

Total cost (C) = $C_1 + C_2 + C_3 + C_4$ = \$209850

## 13.2 Manufacturing processes of roller shafts

- Step 1: Cutting steel rod
- Step 2: Facing the end faces
- Step 3: Turning the shaft
- Step 4: Flatening the ends
- Step 5: Cut groove for circlip pin
- Step 6: Cut groove of 0.5 mm radius for both ends and for both shafts and 4 mm hemishperical hole at both end faces

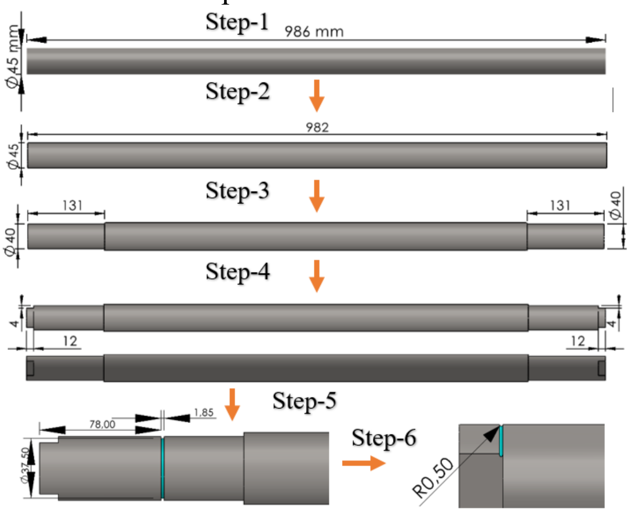



*Figure 68: Manufacturing proces of roller shafts*

## 13.3 Manufacturing process of roller cylinders

- ➢ Step 1: Cut 1.0044 S27JR steel pipes in to required length for side roller and central roller.
- ➢ Step 2: Face 2 mm from both ends to make perfect length for both cylinders.
- ➢ Step 3: Cut end slot for bearing hub of 1 mm thickness from inside diameter of cylinder.

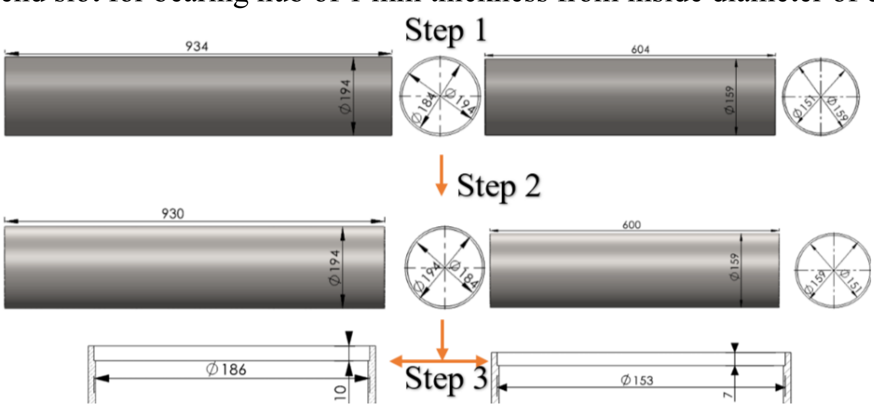



*Figure 69: Manufacturing process of roller cylinders*

## 13.4 Manufacturing process of bearing hub

Method: deep drawing punching

As shown in the figure, at first the steel plate is cut in circular disc shape having diameter bigger than the diameter of bearing hub (this is because after deep drawing the diameter decreases). The pate is then placed over the die concentrically and hold by a retainer block or any mounting component. After this, the drawing punch presses the plate to fill inside die giving the proper shape, finally bearing hub will be ready.

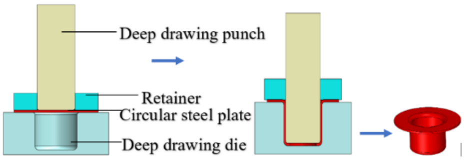



*Figure 70: Deep drawing punching method for bearing hub manufacture*

## 13.5 Assembly of bearing hub and cylinder

Initially, the bearing hub is aligned centrally with the cylinder end, as depicted in the illustration. Subsequently, the hub is inserted into the Ø186 mm slot for the central roller and the Ø153 mm slot for the side roller. Following this, the hub is securely attached to the cylinder by welding along the outer surface, as illustrated in the third image provided.

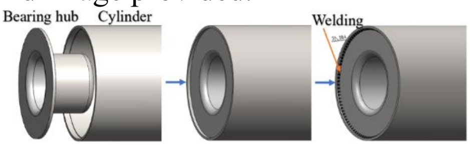



*Figure 71: Cylinder-hub assembly process*

## 13.6 Painting

To prevent the roller from rusting, it is neccessary to apply the coatings on the outer surface. The best way to protect roller from rusting is using the paint and normally some enamel paints are reliable and mostly used in the conveyor rollers.


*Figure 72: Spray painting of cylinder-hub assembly*

## 13.7 Assembly of seals, bearing, shaft and cylinder-hub

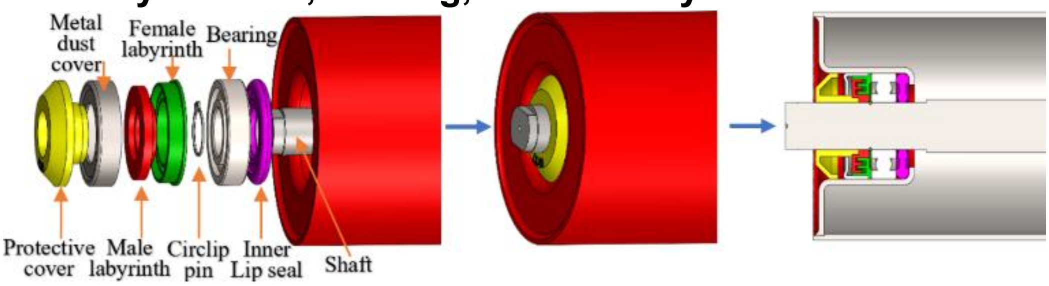



*Figure 73: Assembly method for seals, bearing, shaft and cylinder-hub*

## 13.8 Fully assembled view

*Figure 74: Left-central roller, right- side roller (full assembly)*

## 13.9 Roller - bracket assembly

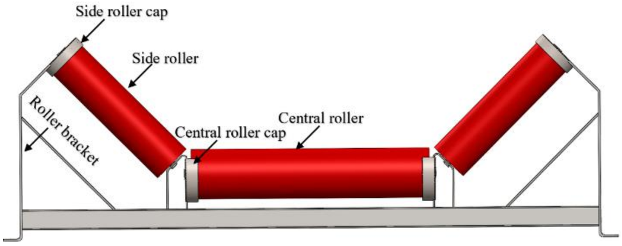



*Figure 75: Roller-bracket assembly in a 45°trough configuration*

# 14: Results

Throughout this dessertation, we optimized the roller components with alternative solution for accesories such as bearings and sealing elements. We have achieved most of the portion of requirements we set as design requirements. Each component of roller had thier own requirements and parameters to consider and we covered most of them to prepare optimized design of rollers for carrying side of the longest conveyor in LKAB. We will discuss all the outcomes we have got in each component below with the technical data and specification.

## 14.1 Weight reduction in roller

Our initial objective was to reduce material while ensuring safety, confirmed through ANSYS simulations. In the central roller, the safety factor was 30.4 with a weight of 27.43 Kg. By reducing thickness to 5 mm and length to 930 mm, we achieved a weight reduction to 21.43 Kg, saving 6 kg. Similarly, for the side roller, the safety factor was 94.37. By reducing thickness from 4.5 mm to 3 mm, we reduced weight by 3.28 kg. Outer diameter remained constant due to belt and bracket constraints.

### A. In central roller

- Overall weight of existing roller = 43.9 Kg
- Overall weight of new roller = 38.8 Kg
- Percentage of weight reduction= 11.6 %
- Weight reductio in rotating parts = 20.38%

### B. In side roller

- Overall weight of existing roller = 16.7Kg
- Overall weight of new roller = 14.1 Kg
- Percentage of weight reduction= 15.5 %
- Weight reductio in rotating parts = 26.3%

*Note: The weight reduction in rotating parts reduces the belt power requirements and decreases the chance of stuck, the detail of belt pull and power savings will be discussed in next section.*

## 14.2 Energy saving

In section 10.4, we calculated the belt pull and power requirements to run the conveyor in both existing and new design cases. Weight reduction in rotating parts helped to decrease the power requirement by 6944 watt. It can save 7 KWH of electricity per hour and 61320 KWH in a year. It can save around 613200 NOK per year of electricity if we consider 10 NOK per KWH.

## 14.3 Bearings and roller life

One of the main outcome is from the selection of bearing, in case of central roller, reviously SKF 6308 was used and it doesnot have shields and lubricant, according to SKF, this bearing needs to relubricate in every 15000 hour (see table 2 Appendix C), which needs lot of repair and maintenance cost. but in the new bearing SKF 6308-2Z-C3, double sields with lithium based grease is used and the grease life is

50900 hour (see table 1 Appendix C), which saves the repair and maintenance cost for the bearing. Likewise, the cost for the external lubricant is not required in 6308-2Z-C3
Similarly, in side roller, SKF 6306-2Z is used in place of SKF 6306, in 6306-2Z, the grease life is 73100 hr, while in existing 6306 it is required to relubricate in every 21600 hour (see table 5 Appendix C).

# 15 Furture work

Due to some limitations and challenges, we could not work on certain areas, which may be important parameters in roller design. First of all in this thesis part there were some limitations such as duration, department requirements for product design guidelines and methodologies, so that below are the description of subjects in which further research can do in the future.

### A. Heat generation in bearing

Heat generates inside the bearing during operation, and it is a lenthy and critical process to calculate analytically without exact mechanical properties of bearing material. Due to our time limitation we could not work in this area and in the future this may be a good research topic. We can compare existing and new bearing on the basis of heat production, which can show the effectiveness of shields used in bearing.

### B. Detail study of enamel paint

To prevent the rollers from rusting, we have recommended to use the enemel paint but the detail about paint has not covered in this dessertation. From our research, most of the roller manufacturers use paint to keep it safe from rust and wear, paint not only saves from rusting but can play a major role to prevent roller from wear.

### C. Surface hardening of roller components

Surface hardening is the most essential process for rollers cylinder to increase the surface hardness, in future the reliable method, its cost and guidelines can study to complete the whole manufacturing process of roller components.

### D. Modeling and testing of the roller caps

The roller caps we have recommended are the basic concept, but we have shown its effectivness and ease for use. Caps are designed in a simple way and can fabricate from sheet metal (AISI 316 stainless steel sheet), to check its reliability, we can develop a model to test or simply fabricate one sample and test.

# 16 Conclusion

The optimization of heavy-duty conveyor rollers aims to reduce weight while enhancing performance in harsh environmental conditions. This reduction not only decreases material usage but also lowers power consumption. In this study, a weight reduction of 20.38% for central rollers and 26.34% for side rollers led to a significant decrease in electricity consumption by 6944 watts per second. Additionally, while the existing design required 762 HP, the new design only needs 753 HP.
Furthermore, careful bearing selection not only extended roller lifespan but also minimized repair and maintenance costs. The selected SKF6308 bearing for central rollers requires lubricant changes every 15000 hours, while the SKF6306 bearing for side rollers needs maintenance every 21600 hours. The new bearings have extended lubricant life to 50900 hours for SKF6308-2Z-C3 and 73100 hours for SKF6306-2Z, reducing downtime. The chosen sealing elements, noted in section 6.3, are reliable, cost-effective, and durable, providing optimal performance for the new design while conforming to standard sizes for bearings, shafts, and bearing hub bores.

## References


[1] Löwe Andreas, (2023, November 21). Conveyor systems: A complete guide. Online: https://www.autostoresystem.com/insights/conveyor-systems-a-complete-guide

[2] H.W. Guderjahn, "Conveyors: Application, Selection, and Integration," CRC Press, 2010.

[3] David A Hounshell, "From the American Systems to Mass Production, 1800-1932: The Development of Manufaacturing Technology in the United States." John Hopkins University Press, 1984.

[4] Womack, James P., Jones, Daniel T., and Roos, Daniel. "The Machine That Changed the World: The story of lean production." Free Press. 2007.

[5] D.K Nannaware, R.R Kharde, (2014, july 7). Design and optimization of roller conveyor system. Online: https://www.scribd.com/document/317597762/Design-and-Optimization-of-Roller-Conveyor-System

[6] B, B. (2023, may 15). How does a conveyor belt works?-Storage equipment systems-Inc. Online: https://storageequipmentsystems.com/how-a-conveyor-belt-works/

[7] Smith, J. (2024, April 15). Continental conveyor idlers: Optimal material handling, GRAM conveyor. Online: https://www.gramconveyor.com/continental-conveyor-idlers/

[8] JIENASI, Detailed conveyor idler feature, Baoding Zhuoli Machinery Co., Ltd. Online: https://www.conveyoroller.com/product/Conveyor-idler.html

[5] Austin, L.S. "Conveyor Systems: A Summary of the Basic Types of Conveyors." Industrial Press Inc., 2001

[6] Patric M. McGuire, "Conveyors: Application, Selection, and Integration",-2nd edition, CRC Press, 2017

[7] Niazi, M.A. (2022, November 2). Introduction to conveyor belt systems. Technical Article. https://control.com/technical-articles/introduction-to-conveyor-belt-systems/

[8] Automated conveyor systems, Inc. Online webpage Available at: https://www.automatedconveyors.com/image/HPB.jpg

[9] Art and science of belt conveyor lubrication, by Vas Tribology solutions, Published by: Machinery Lubrication India, online: https://www.machinerylubricationindia.com/magazine/2017/mar-apr/art-and-science-of-belt-conveyor-lubrication/

[10] Narcop. (2023, may 25). Key Factors for Optimizing conveyor belt motor performance-NAR. Nar Automation Engineering(Wuxi) Co.Ltd. online: https://www.nar-robotics.com/key-factors-for-optimizing-conveyor-belt-motor-performance/

[11] Daniel Valen Nilsen, Maintenence Engineer-LKAB Narvik

[12] Prasad M Patil, Gajanan C Koli (2020). Weight optimization of support roller by using theoretical, finite element and experimental analysis. Online:https://www.academia.edu/67792725/Weight_Optimization_Of_Support_Roller_By_Using_Theoretical_Finite_Element_And_Experimental_Analysis?email_work_card=view-paper

[13] RULMECA holding S.P.A, catalog bulk handling, online: https://www.rulmeca.com/en/complete-catalogues?idp=&Dir=bulk#listingDownload

[14] Daniel Valen Nilsen, Fred Yang-Maintenance Engineer LKAB Narvik. Online: www.LKAB.com

[15] SKF International-bearing 6306, online: https://www.skf.com/group/products/rolling-bearings/ball-bearings/deep-groove-ball-bearings/productid-6306

[16] ROXON OY,(A conveyor equipment manufacturer), Product catalog, online: https://roxon.com/wp-content/uploads/2023/11/Roxeng.pdf

[17] Jurandir Primo, PE (2020), Belt conveyor for bulk materials-practical calculations PDH online course M344(2PDH), online: https://pdhonline.com/courses/m344/m344content.pdf

[18] Rulmeca Group, Technical Information-project and design criteria for belt conveyors, Online:https://www.rulmeca.com/download/catalogo/macrofamiglia_eng/1_Catalog_Bulk_Technical_Information_EN

[19] Raparelli T., Mazza L., & Trivella A. (2023). Experimental analysis and priliminary model of non-conventional lip seals. Tribology International-181-108311 Online: https://www.sciencedirect.com/science/article/pii/S0301679X23000981?via%3Dihub

[20] Steel conveyor roller/Joyroll. (2023, August 26), Joyroll sealing systems. Online: https://www.joyroll.net/product/steel-conveyor-roller/

[21] TR Fastings.com, produst catalog/shaft external circlip
Online: https://www.trfastenings.com/Products/Catalogue/Washers-Circlips-Rings-Seals-O-Rings/Circlips/External-Circlips/TR00020162-000?rank=15
[22] Michael F. Ashby,(2017). Material selection in Mechanical Design-5th edition,
[23] Granta EduPack 2023 R2, Level 2 (a material selection software in engineering design)
[24] European Steel and Alloy Grades/Numbers Database (www.steelnumber.com)
Online: https://www.steelnumber.com/en/steel_composition_eu.php?name_id=3
[25] SKF International (online website) : https://www.skf.com/group
[26] ANSYS Workbench 2023 R2 (Static structural analysis, Modal analysis)
[27] S. Rao, Mechanical Vibrations, 5th edition, Prentice Hall-2011
[28] CEMA, Standard no 401-2019, Online: ANSI-CEMA-401-Draft-1.pdf (cemanet.org)
[29] Jurandir Primo, PE (2020), Belt conveyor for bulk materials-practical calculations PDH online course M344(2PDH), online: https://pdhonline.com/courses/m344/m344content.pdf
[30]SKF Product selection and calculations, Online: https://skfbearingselect.com/#/size-lubrication/single-bearing
[31] Gautham Kasori, 2010-027 PB, Rolling/sliding wear resistance of steels with different microstructures. (A master's thesis in Advanced material science and engineering)
[32] Tunde Kovacs-Coskun/Peter Pinke, (April-2011). The effect of microstructure on the local wear behavior of heat treated structural steel. Online:
https://www.researchgate.net/publication/260795033_The_Effect_of_Microstructure_on_the_Local_Wear_Behavior_of_Heat_Treated_Structural_Steel
[33] SKF product select, Fits & Tolerances. Online: SKF Product select (skfbearingselect.com)
[34] Michael F. Ashby,(2017). Material selection in Mechanical Design-5th edition/ Process selection and cost/process selection strategy-chapter7 -figure7.1
[35] Michael F. Ashby,(2017). Material selection in Mechanical Design-5th edition/ Process selection and cost/Ranking: the process cost figure 7.17
[36] Nordic steel prices/ MEPS. MEPS International Ltd.
Online:https://mepsinternational.com/gb/en/products/nordic-steel-prices
[37] Europe Stainless steel prices/3 year's historical Data/MEPS. MEPS International Ltd.
Online: https://mepsinternational.com/gb/en/products/europe-stainless-steel-prices
[38] Made in china .com. manufacturers, suppliers and products in china.
Online: https://www.made-in-china.com/?pv_id=1ht47tgk7d3c&faw_id=1ht48u8c47bb
[39] Michael F. Ashby,(2017). Material selection in Mechanical Design-5th edition-figure 7.3
[40] Michael F. Ashby,(2017). Material selection in Mechanical Design-5th edition-figure 7.4
[41] Michael F. Ashby,(2017). Material selection in Mechanical Design-5th edition-figure 7.6

## Appendix A: Conveyor design factors

Table 1: Estimated belt mass-B(Wb)-kg/m [29]

| Belt width. (mm) | Operating conditions | | |
|---|---|---|---|
| | Light duty Kg/m | Medium duty Kg/m | Heavy duty Kg/m |
| 500 | 4.1 | 6.2 | 10.3 |
| 600 | 5.0 | 7.4 | 12.3 |
| 750 | 6.2 | 9.3 | 15.5 |
| 900 | 7.4 | 11.1 | 18.5 |
| 1050 | 8.6 | 13.0 | 21.6 |
| 1200 | 9.8 | 14.8 | 24.7 |
| 1350 | 11.0 | 16.7 | 27.8 |
| 1500 | 12.3 | 18.6 | 30.9 |
| 1650 | 13.5 | 20.5 | 33.9 |
| 1800 | 14.7 | 22.3 | 37.0 |

Table 2: Sag factor **[29]**

| Percentage sag | Sag factor $S_f$ |
|---|---|
| 3% | 4.2 |
| 2% | 6.3 |
| 1.5% | 8.4 |

Table 3: Recommended sag percent

| Trough angle (degree) | Fine material | Lumps up to max lump size | Max lump size |
|---|---|---|---|
| 20 | 3% | 3% | 3% |
| 35 | 3% | 2% | 2% |
| 45 | 3% | 2% | 1.5% |

Table 4: Friction factors [29]

| Symbol | Description | Value of the friction factor | | |
|---|---|---|---|---|
| | | Normal operating conditions. Horizontal length Up to 250m | Normal Operating Conditions Horizontal length More than 250m | Very well Aligned Structure Tilted idler horizontal length more than 500m |
| $f_c$ | Friction coefficient for scrapers | 0.600 | 0.600 | 0.600 |
| $f_s$ | Friction coefficient for skirtboards | 0.650 | 0.650 | 0.650 |
| $f_x$ | Friction coefficient for empty belt | 0.022 | 0.020 | 0.020 |
| $f_y$ | Friction coefficient for loaded belt | 0.027 | 0.022 | 0.020 |

Table 5: Participation factor [29]

| Participation factor $F_P$ - loaded rate on the most loaded roller | | | | | | | |
|---|---|---|---|---|---|---|---|
| $\alpha$ | $0^\circ$ | $20^\circ$ | $30^\circ$ | $35^\circ$ | $45^\circ$ | $30^\circ - 45^\circ$ | $60^\circ$ |
| $F_P$ | 1.00 | 0.60 | 0.65 | 0.67 | 0.72 | 0.52-0.60 Shorter central idler | 0.47 5 rollers |

Table 6: Impact factor [29]

| Impact factor "$F_d$" | | | | |
|---|---|---|---|---|
| Material lump size | Belt speed m/s | | | |
| | **2** | **2.5** | **3** | **3.5** |
| **0-100 mm** | 1 | 1 | 1 | 1 |
| **100-150 mm** | 1.02 | 1.03 | 1.05 | 1.07 |
| **150-300 mm** | 1.04 | 1.06 | 1.09 | 1.12 |
| **300-450 mm** | 1.06 | 1.09 | 1.12 | 1.70 |

Table 7: Service factor [29]

| Life | Service factor $F_s$ |
|---|---|
| Less than 6 hours per day | 0.8 |
| From 6-9 hours per day | 1.0 |
| From 10-16 hours per day | 1.1 |
| Over 16 hours per day | 1.2 |

Table 8: Environment factor [29]

| Environment factor $F_m$ | |
|---|---|
| Conditions | $F_m$ |
| Clean and regular maintenance | 0.9 |
| Abrasive or corrosive material present | 1.0 |
| Very abrasive or corrosive material present | 1.1 |

## Appendix B : Dimension of roller cap components

Figure 1: Cutting dimension of stainless steel plate for central roller cap
Item name: left plate, right plate

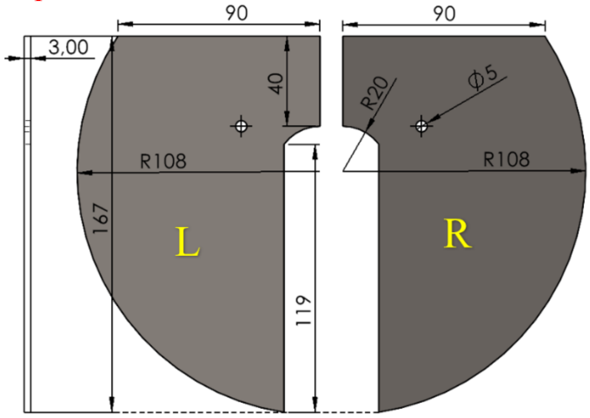



Figure 2: Curved plate for central roller cap
Item name: curved plate( left & right)

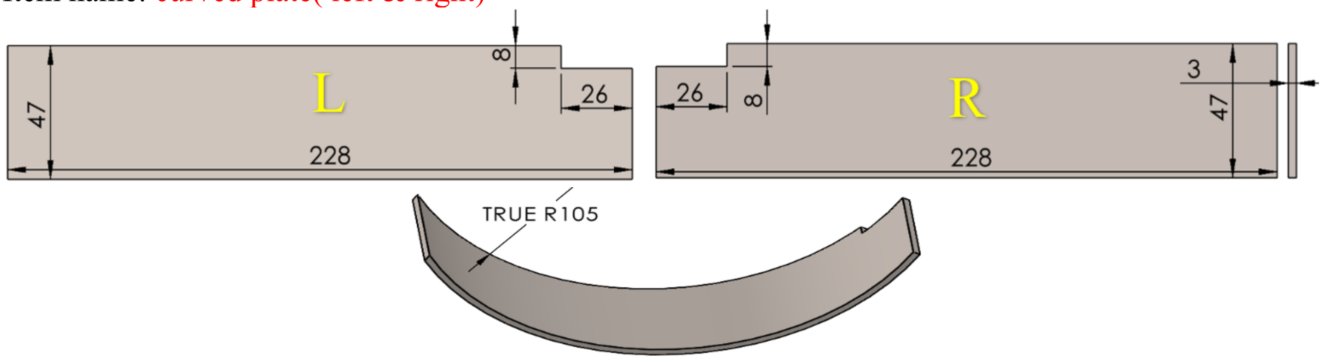



Figure 3: Fabricated view of left and right covers with cap lock plate for central roller cap

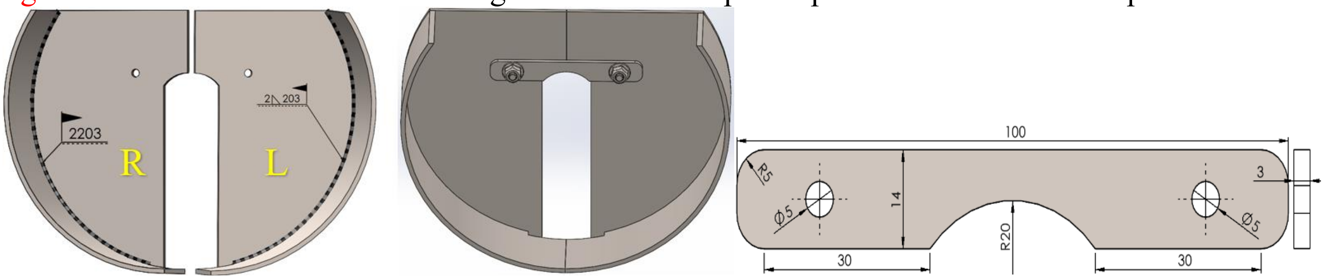



Figure 4: Bolt and nuts for central roller cap (M5)

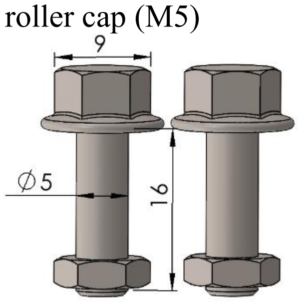


Figure 5: Circular plate section for side roller cap

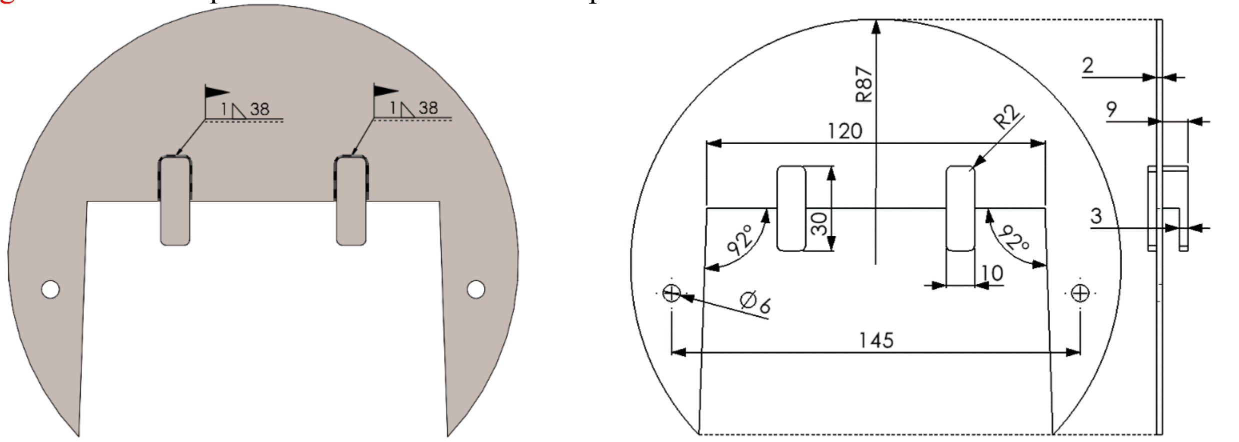



Figure 6: Cylindrical tube section for side roller cap

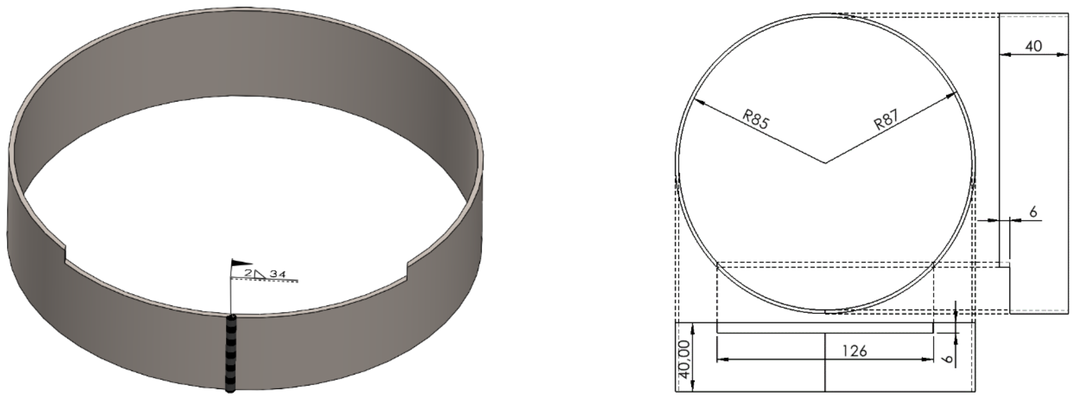



Figure 7: Cap lock plate for side roller cap

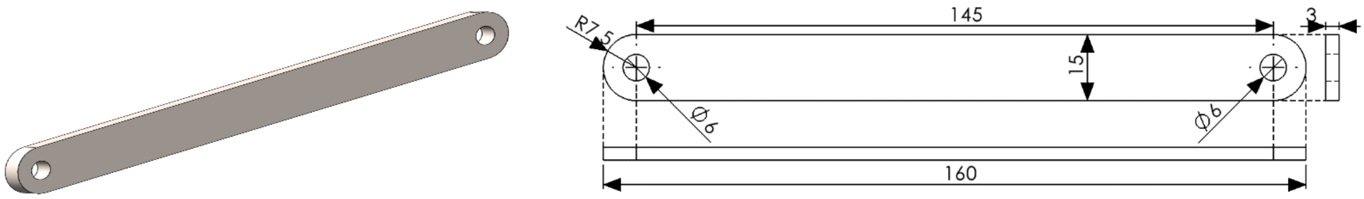



Figure 8: Flat sheet dimension for cylindrical tube of side roller cap

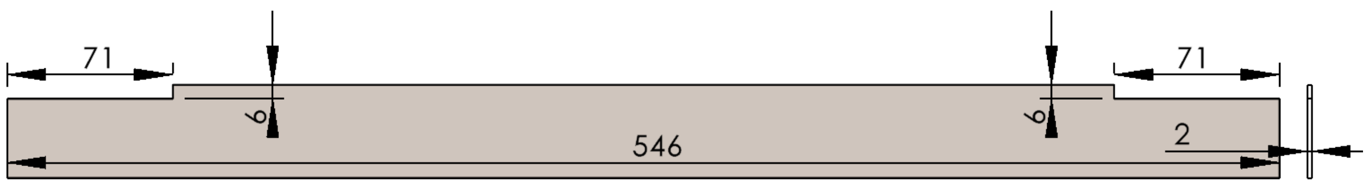
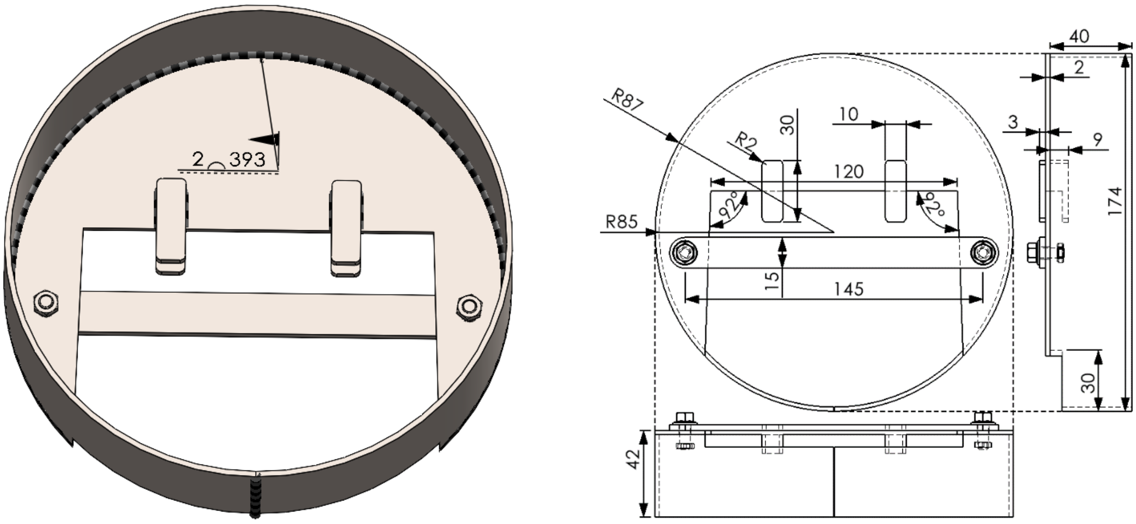


Figure 9: Full assemble view of side roller cap with dimensions




## Appendix C: Calculations of the bearing parameters

Table 1: Bearing parameters for central roller (SKF 6308-2Z-C3) new design[25]

<table>
<tr><td>Parameters</td><td colspan="6">Designation = SKF 6308-2Z-C3 (deep groove ball bearing)</td></tr>
<tr><td rowspan="2">Viscosity</td><td>Actual v<br>(mm²/s)</td><td colspan="2">Rated $v_1$<br>(mm²/s)</td><td colspan="2">Rated at 40°C $v_{ref}$<br>(mm²/s)</td><td>Viscosity ratio<br>(k)</td></tr>
<tr><td>25.3</td><td colspan="2">43.3</td><td colspan="2">175</td><td>0.58</td></tr>
<tr><td rowspan="2">Bearing loads</td><td colspan="3">Equivalent dynamic load P (kN)</td><td colspan="3">Load ratio (C/P)</td></tr>
<tr><td colspan="3">3.9</td><td colspan="3">10.85</td></tr>
<tr><td rowspan="2">Grease life and relubrication interval</td><td colspan="3">Grease life<br>$L_{10h}$ (h)</td><td colspan="3">Speed factor<br>$nd_m$ (mm/min)</td></tr>
<tr><td colspan="3">50900</td><td colspan="3">18500</td></tr>
<tr><td rowspan="2">Bearing rating life</td><td>Basic<br>$L_{10h}$ (h)</td><td colspan="2">Basic<br>$L_{10mh}$ (h)</td><td colspan="2">Life modification factor ($a_{skf}$)</td><td>Contamination factor $\eta_c$</td></tr>
<tr><td>74400</td><td colspan="2">91200</td><td colspan="2">1.23</td><td>0.5</td></tr>
<tr><td rowspan="2">Friction moment and power loss</td><td>Total<br>M (Nmm)</td><td>$M_{start}$<br>(Nmm)</td><td>Rolling<br>$M_{rr}$ (Nmm)</td><td>Sliding<br>$M_{sl}$ (Nmm)</td><td colspan="2">Power loss<br>$P_{loss}$ (W)</td></tr>
<tr><td>110</td><td>138</td><td>24.8</td><td>86</td><td colspan="2">3.3</td></tr>
</table>

Table 2: Bearing parameters for central roller (SKF 6308) existing case[25]

<table>
<tr><td>Parameters</td><td colspan="6">Designation = SKF 6308 (deep groove ball) ($F_r$ = 3898N, $F_a$ = 0N)</td></tr>
<tr><td rowspan="2">Viscosity</td><td>Actual v<br>(mm²/s)</td><td colspan="2">Rated $v_1$<br>(mm²/s)</td><td colspan="2">Rated at 40°C $v_{ref}$<br>(mm²/s)</td><td>Viscosity ratio<br>(k)</td></tr>
<tr><td>26.2</td><td colspan="2">43.3</td><td colspan="2">175</td><td>0.6</td></tr>
<tr><td rowspan="2">Bearing loads</td><td colspan="3">Equivalent dynamic load P (kN)</td><td colspan="3">Load ratio (C/P)</td></tr>
<tr><td colspan="3">3.9</td><td colspan="3">10.85</td></tr>
<tr><td rowspan="2">Grease life and relubrication interval</td><td colspan="2">Relubrication interval<br>$t_f$ (h)</td><td>Grease quantity $G_p$<br>(gm)</td><td colspan="3">Speed factor.<br>$nd_m$ (mm/min)</td></tr>
<tr><td colspan="2">15000</td><td>10</td><td colspan="3">18500</td></tr>
<tr><td rowspan="2">Bearing rating life</td><td>Basic<br>$L_{10h}$ (h)</td><td colspan="2">Basic<br>$L_{10mh}$ (h)</td><td colspan="2">Life modification factor ($a_{skf}$)</td><td>Contamination factor $\eta_c$</td></tr>
<tr><td>74400</td><td colspan="2">49200</td><td colspan="2">0.66</td><td>0.22</td></tr>
<tr><td rowspan="2">Friction moment and power loss</td><td>Total<br>M (Nmm)</td><td>$M_{start}$<br>(Nmm)</td><td>Rolling<br>$M_{rr}$ (Nmm)</td><td>Sliding<br>$M_{sl}$ (Nmm)</td><td colspan="2">Power loss<br>$P_{loss}$ (W)</td></tr>
<tr><td>110</td><td>138</td><td>25.3</td><td>84.9</td><td colspan="2">3.3</td></tr>
</table>

Table 3: Bearing parameters for side roller (SKF 6306-2Z) new design side A[25]

| Parameters | Designation = SKF 6306-2Z (deep groove ball) side A ($F_r$ = 908 N, $F_a$ = 569 N | | | | |
|---|---|---|---|---|---|
| Viscosity | Actual v ($mm^2/s$) | Rated $v_1$ ($mm^2/s$) | | Rated at 40°C $v_{ref}$ ($mm^2/s$) | Viscosity ratio (k) |
| | 25.3 | 40.8 | | 162 | 0.62 |
| Bearing loads | Equivalent dynamic load P (kN) | | | Load ratio (C/P) | |
| | 1.58 | | | 18.68 | |
| Grease life and relubrication interval | Grease life $L_{10h}$ (h) | | | Speed factor $nd_m$ (mm/min) | |
| | 73100 | | | 18200 | |
| Bearing rating life | Basic $L_{10h}$ (h) | Basic $L_{10mh}$ (h) | | Life modification factor ($a_{skf}$) | Contamination factor $\eta_c$ |
| | > 200000 | > 200000 | | 2.1 | 0.45 |
| Friction moment and power loss | Total M (Nmm) | $M_{start}$ (Nmm) | Rolling $M_{rr}$ (Nmm) | Sliding $M_{sl}$ (Nmm) | Power loss $P_{loss}$ (W) |
| | 61.4 | 64.4 | 21.9 | 39.5 | 2.3 |

Table 4: Bearing parameters for side roller (SKF 6306-2Z) new design side B[25]

| Parameters | Designation = SKF 6306-2Z (deep groove ball) side B ($F_r$ = 229 N, $F_a$ = 569 N | | | | |
|---|---|---|---|---|---|
| Viscosity | Actual v ($mm^2/s$) | Rated $v_1$ ($mm^2/s$) | | Rated at 40°C $v_{ref}$ ($mm^2/s$) | Viscosity ratio (k) |
| | 25.3 | 40.8 | | 162 | 0.62 |
| Bearing loads | Equivalent dynamic load P (kN) | | | Load ratio (C/P) | |
| | 1.2 | | | 24.57 | |
| Grease life and relubrication interval | Grease life $L_{10h}$ (h) | | | Speed factor $nd_m$ (mm/min) | |
| | 73100 | | | 18200 | |
| Bearing rating life | Basic $L_{10h}$ (h) | Basic $L_{10mh}$ (h) | | Life modification factor ($a_{skf}$) | Contamination factor $\eta_c$ |
| | > 200000 | > 200000 | | 2.98 | 0.45 |
| Friction moment and power loss | Total M (Nmm) | $M_{start}$ (Nmm) | Rolling $M_{rr}$ (Nmm) | Sliding $M_{sl}$ (Nmm) | Power loss $P_{loss}$ (W) |
| | 59.5 | 63.7 | 20.5 | 39 | 2.2 |

Table 5: Bearing parameters for side roller (SKF 6306) existing case side A[25]

| Parameters | Designation = SKF 6306 (deep groove ball) side A ($F_r$ = 908N, $F_a$ = 569N) | | | | |
|---|---|---|---|---|---|
| Viscosity | Actual v ($mm^2/s$) | Rated $v_1$ ($mm^2/s$) | | Rated at 40°C $v_{ref}$ ($mm^2/s$) | Viscosity ratio (k) |
| | 26.2 | 40.8 | | 162 | 0.64 |
| Bearing loads | Equivalent dynamic load P (kN) | | | Load ratio (C/P) | |
| | 1.58 | | | 18.68 | |
| Grease life and relubrication interval | Relubrication interval $t_f$ (h) | Grease quantity $G_p$ (gm) | | Speed factor. $nd_m$ (mm/min) | |
| | 21600 | 7 | | 18200 | |
| Bearing rating life | Basic $L_{10h}$ (h) | Basic $L_{10mh}$ (h) | | Life modification factor ($a_{skf}$) | Contamination factor $\eta_c$ |
| | >200000 | >200000 | | 0.96 | 0.2 |
| Friction moment and power loss | Total M (Nmm) | $M_{start}$ (Nmm) | Rolling $M_{rr}$ (Nmm) | Sliding $M_{sl}$ (Nmm) | Power loss $P_{loss}$ (W) |
| | 61.4 | 64.4 | 22.3 | 39 | 2.3 |

Table 6: Bearing parameters for side roller (SKF 6306) existing case side B[25]

| Parameters | Designation = SKF 6306 (deep groove ball) side B ($F_r$ = 229N, $F_a$ = 569N) | | | | |
|---|---|---|---|---|---|
| Viscosity | Actual v ($mm^2/s$) | Rated $v_1$ ($mm^2/s$) | | Rated at 40°C $v_{ref}$ ($mm^2/s$) | Viscosity ratio (k) |
| | 26.2 | 40.8 | | 162 | 0.64 |
| Bearing loads | Equivalent dynamic load P (kN) | | | Load ratio (C/P) | |
| | 1.2 | | | 24.57 | |
| Grease life and relubrication interval | Relubrication interval $t_f$ (h) | Grease quantity $G_p$ (gm) | | Speed factor. $nd_m$ (mm/min) | |
| | 21600 | 7 | | 18200 | |
| Bearing rating life | Basic $L_{10h}$ (h) | Basic $L_{10mh}$ (h) | | Life modification factor ($a_{skf}$) | Contamination factor $\eta_c$ |
| | >200000 | >200000 | | 0.96 | 0.2 |
| Friction moment and power loss | Total M (Nmm) | $M_{start}$ (Nmm) | Rolling $M_{rr}$ (Nmm) | Sliding $M_{sl}$ (Nmm) | Power loss $P_{loss}$ (W) |
| | 59.5 | 63.7 | 20.9 | 38.6 | 2.2 |

## Appendix D: Material property and process selection charts

Figure 1: The process-material matrix[39].

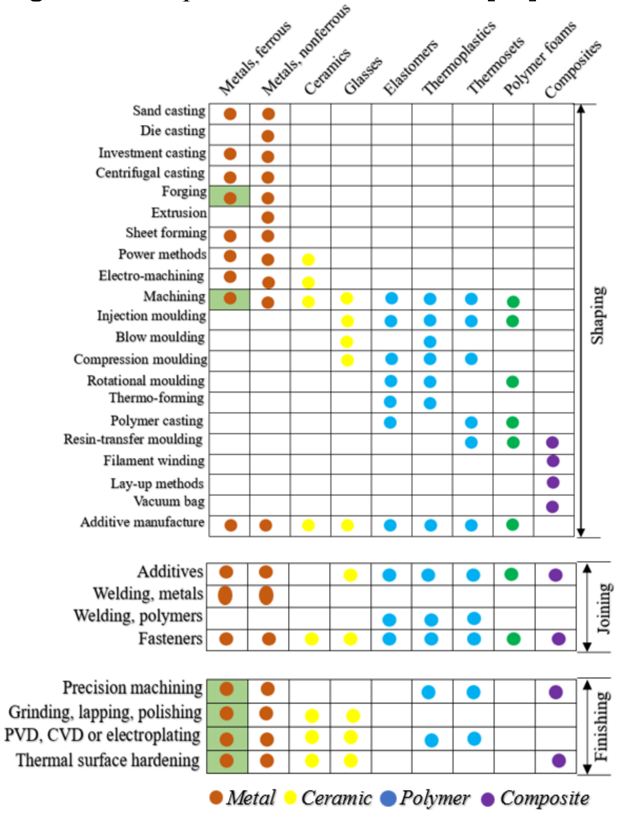



Figure 2: The process-shape matrix[40].

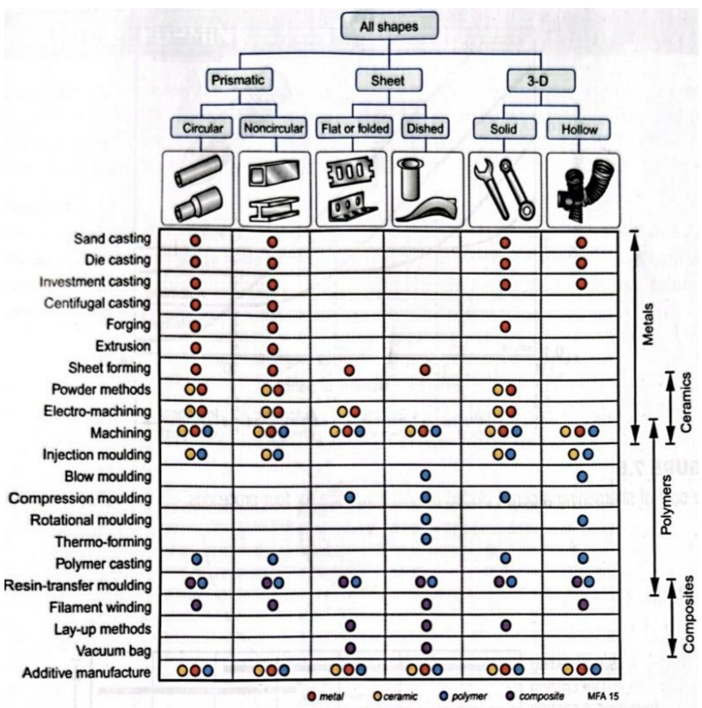
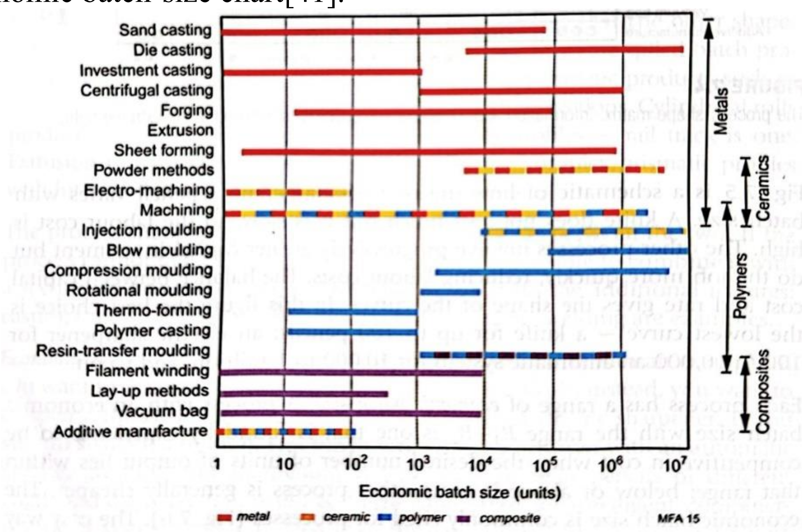
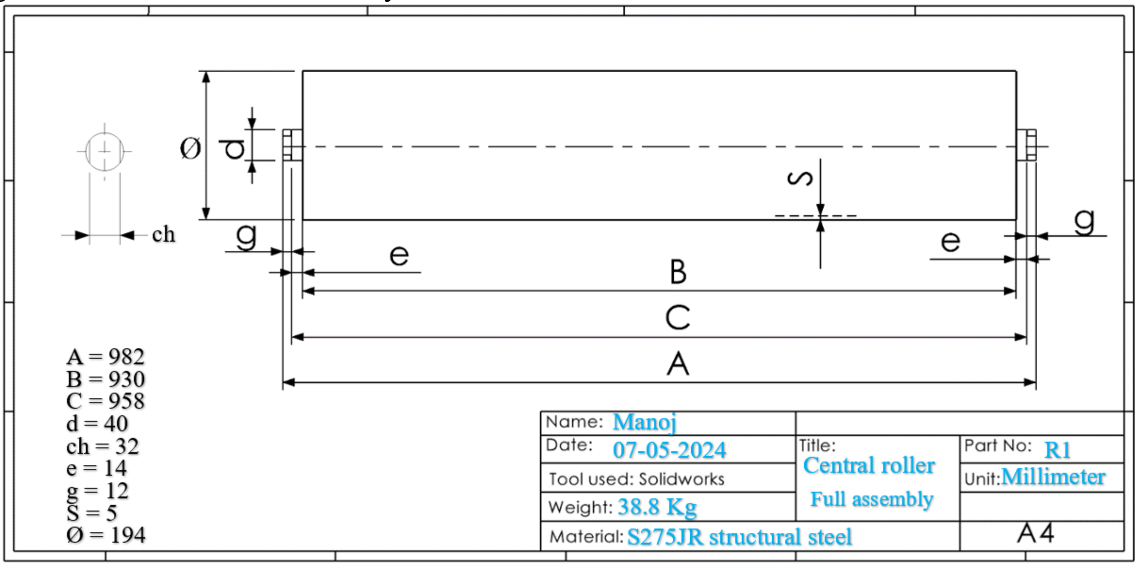


Figure 3: The economic batch-size chart[41].

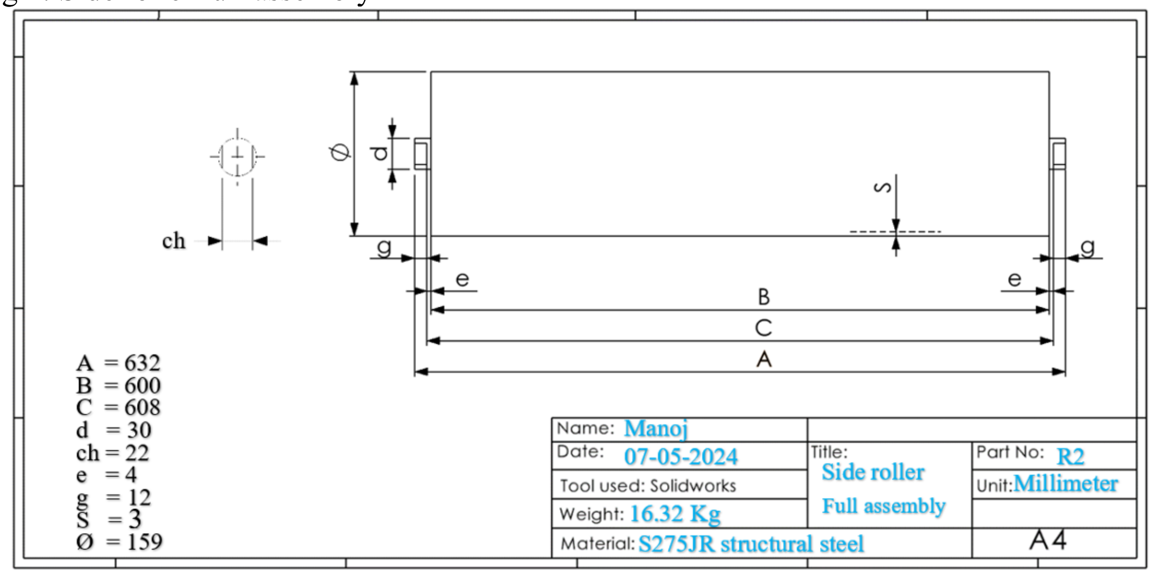



## Appendix E: Drawings of roller components

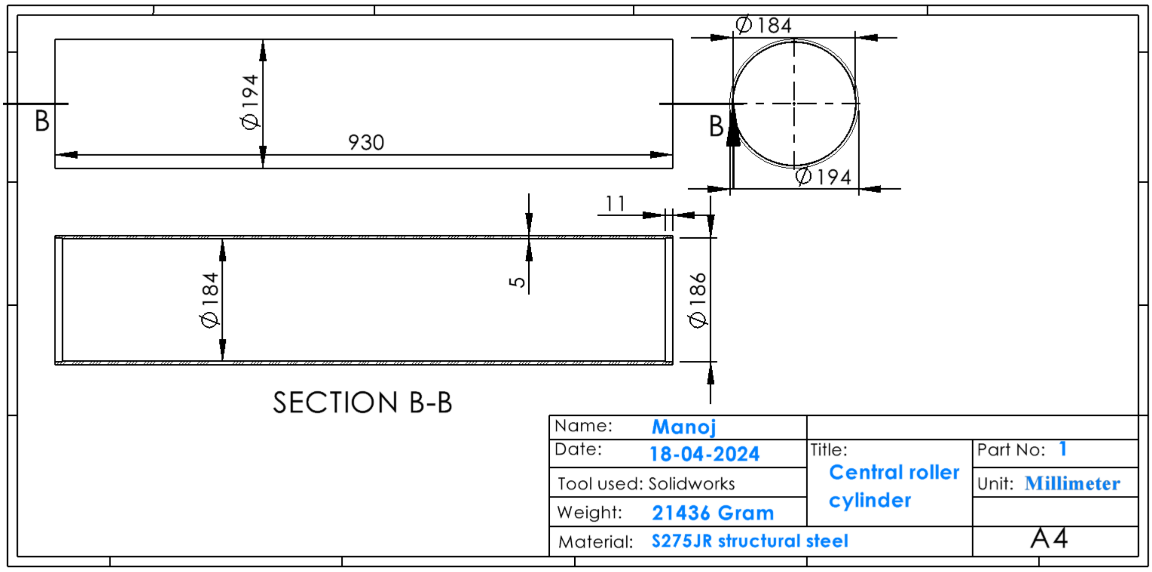
Drawing 1: Central roller full assembly



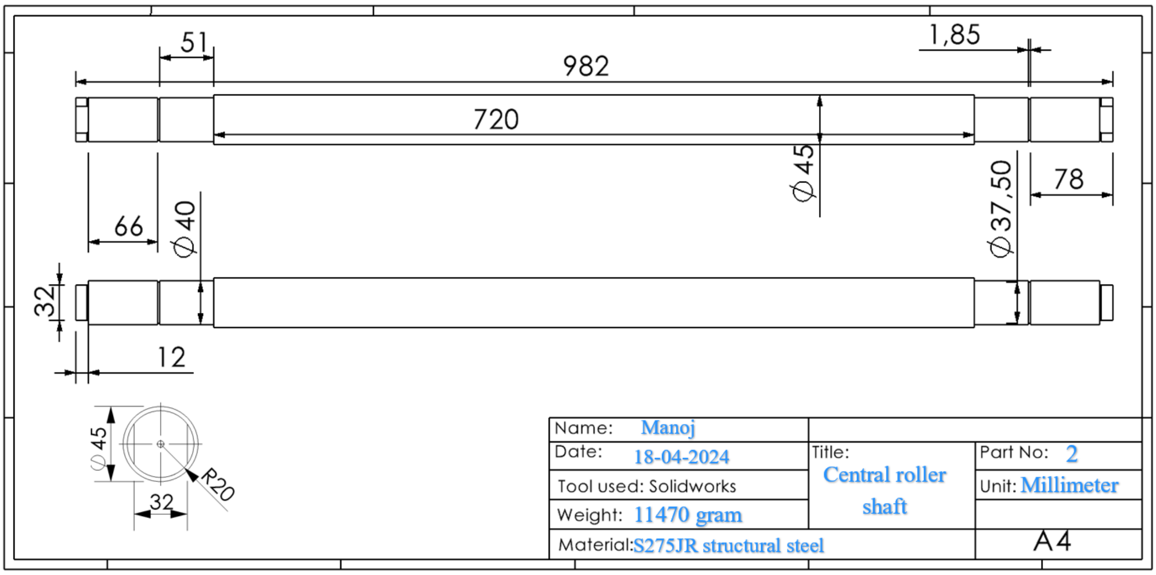

Drawing 2: Side roller full assembly



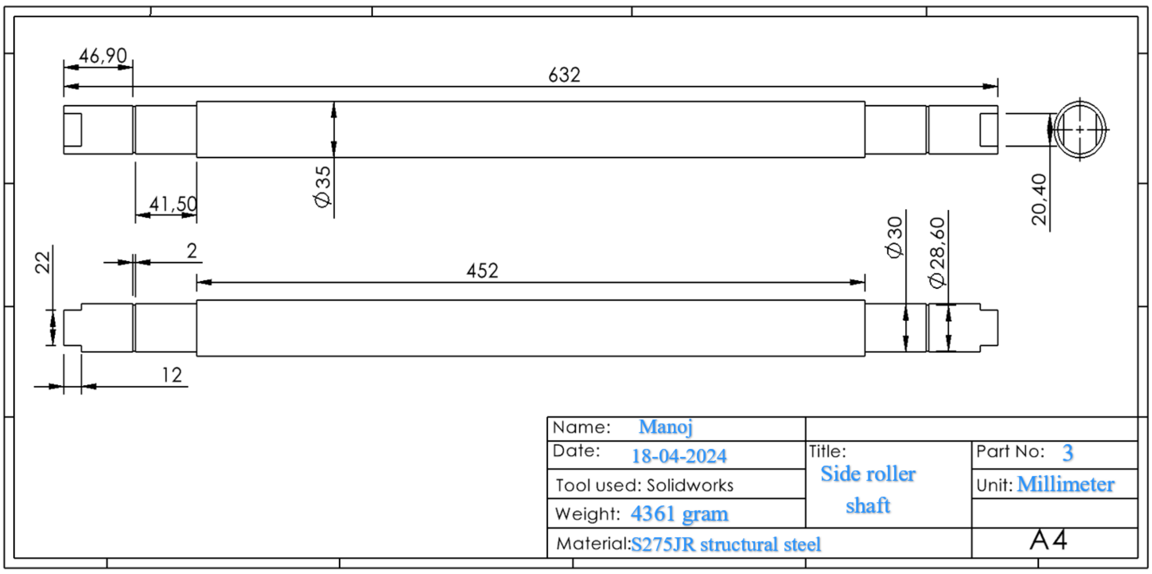
Drawing 3: Central roller cylinder



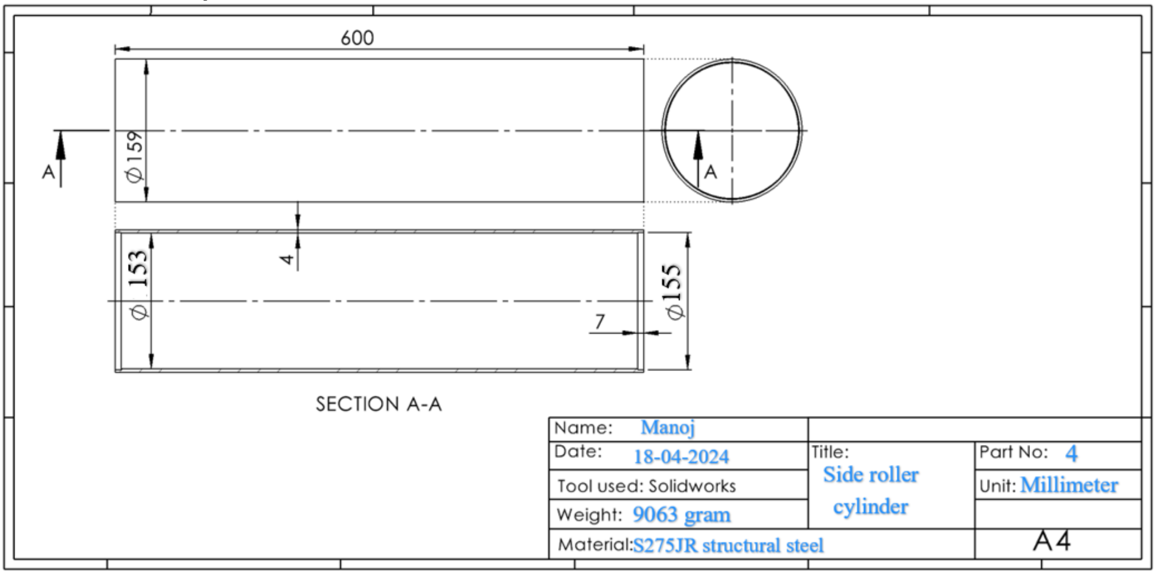

Drawing 4: Central roller shaft



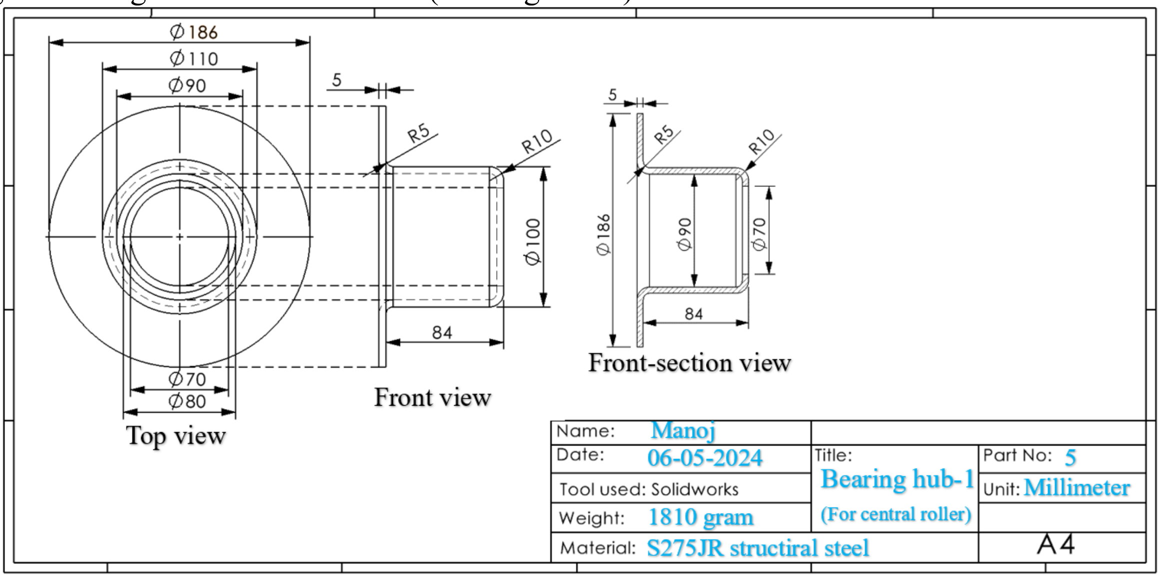

Drawing 5: Side roller shaft



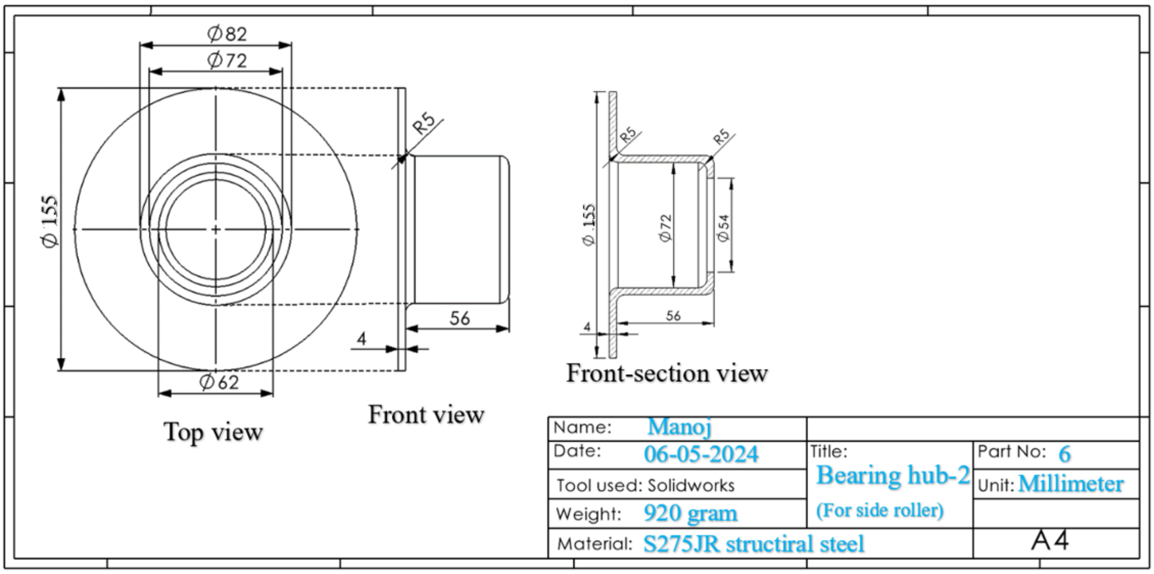
Drawing 6: Side roller cylinder



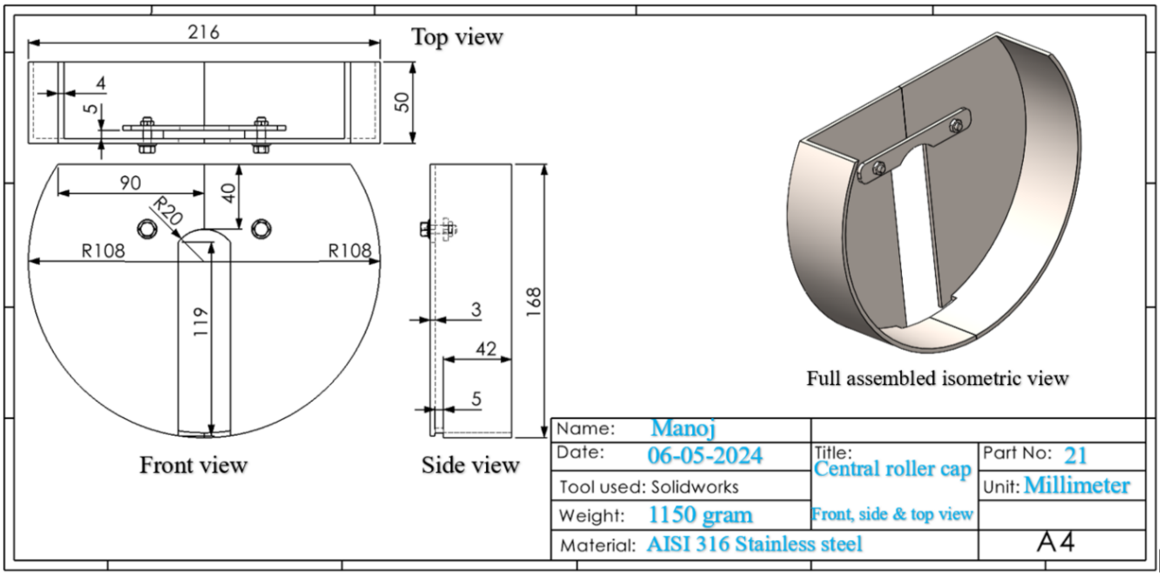

Drawing 7: Bearing hub for central roller (Bearing hub-1)



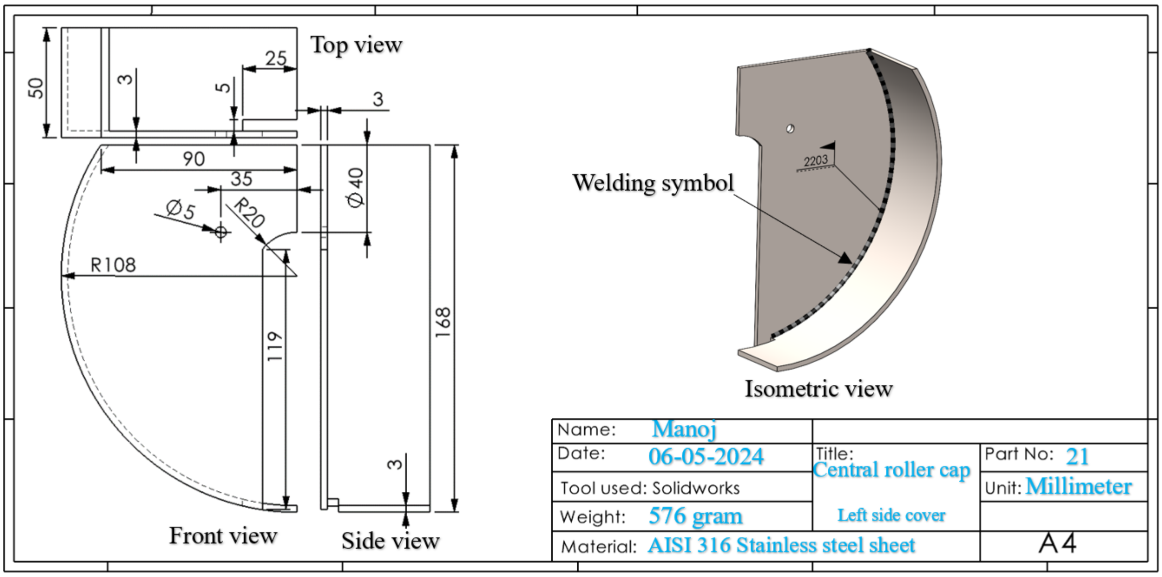

Drawing 8: Bearing hub for side roller (Bearing hub-2)



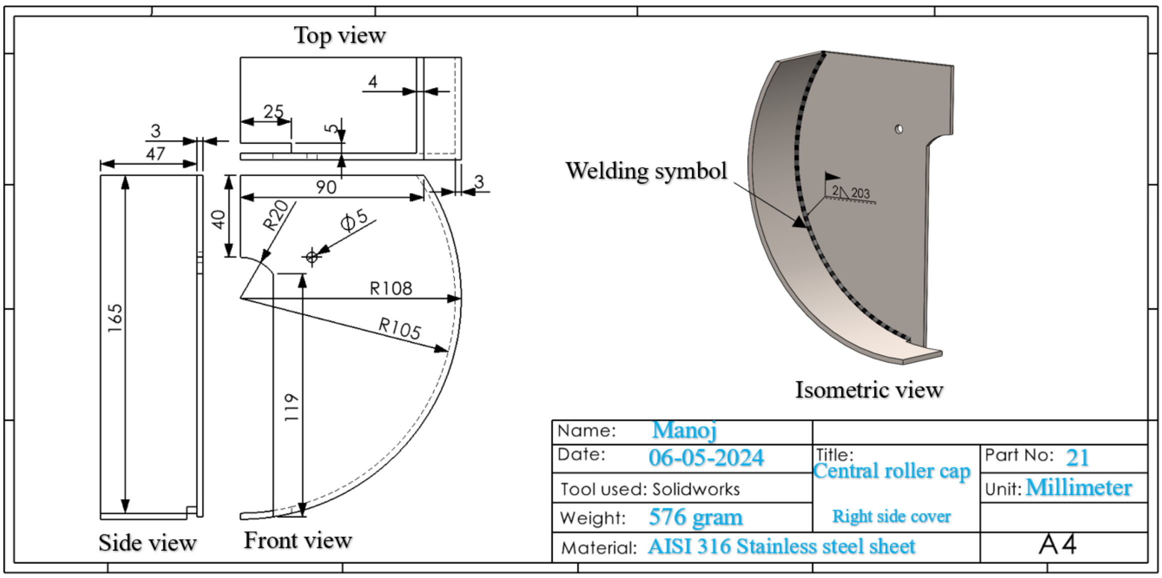
Drawing 9: Central roller cap (full assembled isometric view with 2D drawings)



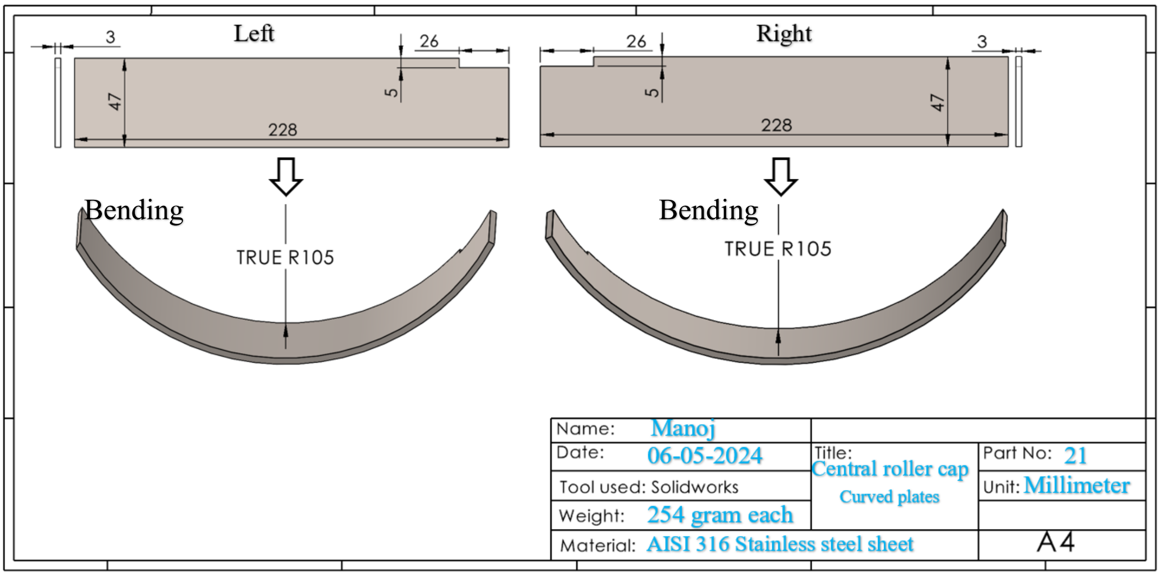

Drawing 10: Left side cover of central roller cap



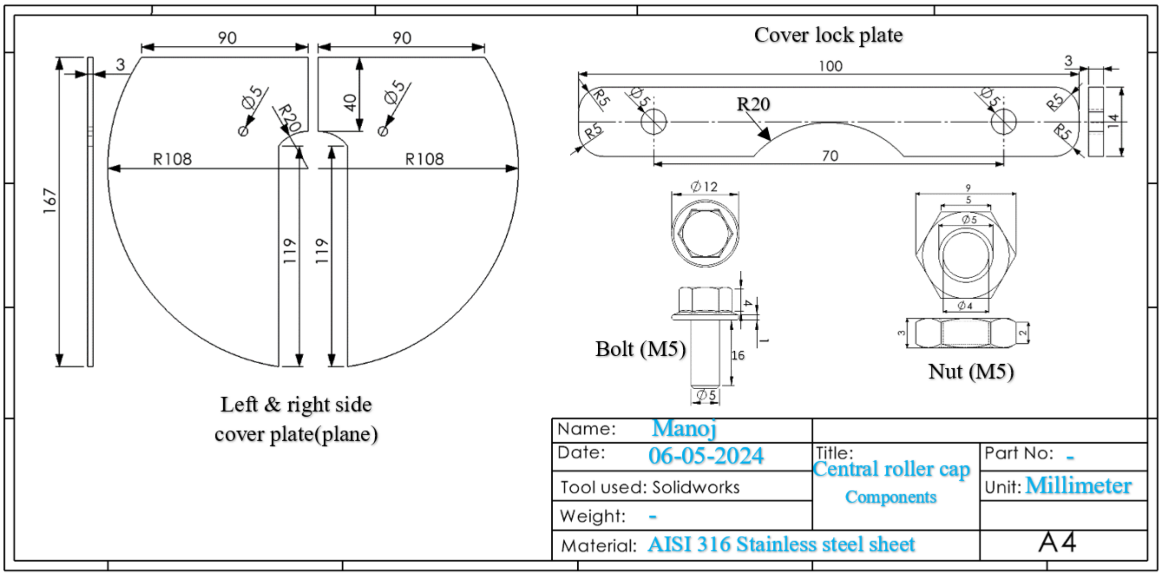

Drawing 11: Right side cover of central roller cap



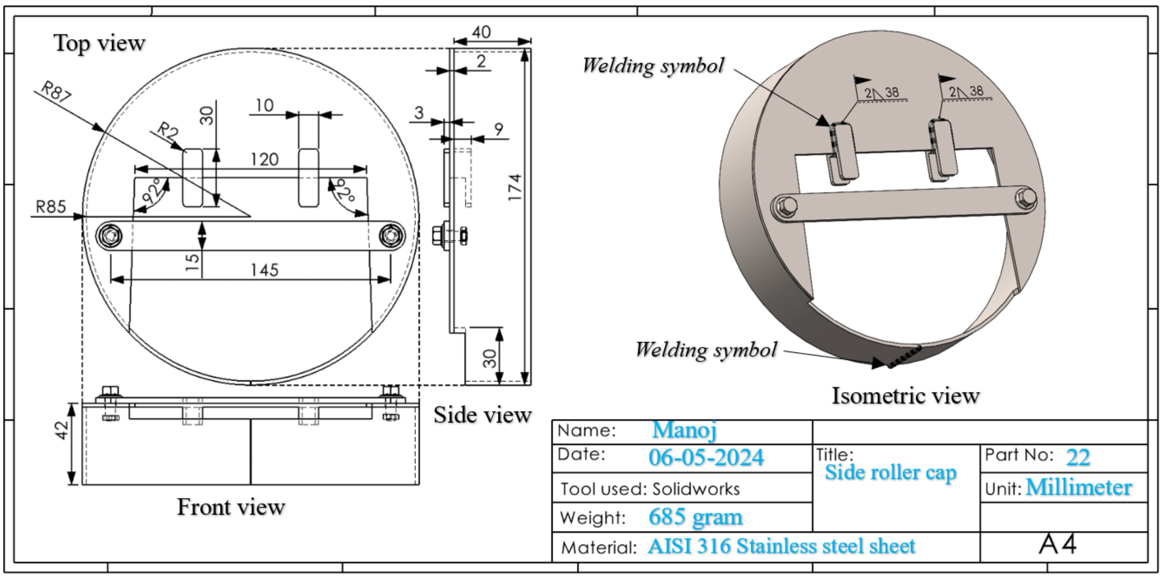
Drawing 12: Curved plate for central roller cap



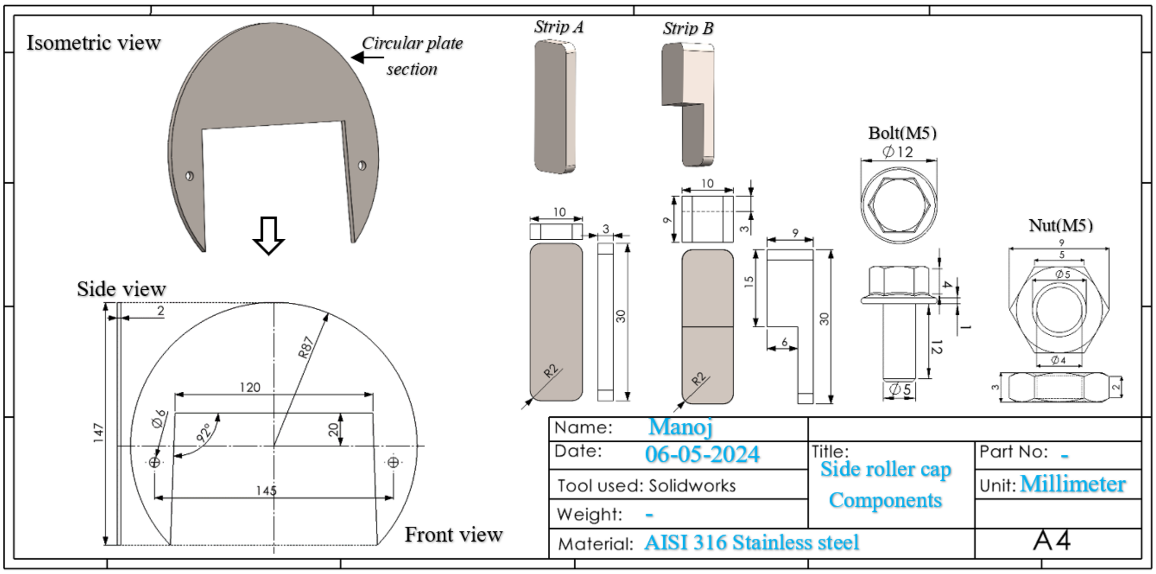

Drawing 13: Central roller cap components




Drawing 14: Side roller cap full assembled siometric view with 2D drawings



Drawing 15: Side roller cap components




Drawing 16: Side roller components