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Re-do/upgrading Drive Unit for a Manure Cable Drum

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Summary

The main goal of this thesis is to develop a new power transmission design for a manure hose drum, manufactured and sold by HMR Voss AS. As of today, HMR delivers a manure spreader with an additional drum. The manure spreader continuously sprays manure across the acre, where the manure is delivered to the spreader through a long hose. Under operation, the drum initially spools out the manure hose and finally spools it back in after use.

Today's solution for rotating the drum consists of a roller chain system, where the roller chain is welded onto a plate in a circular pattern. The plate is then bolted onto the drum's side plate, making a rigid connection between the chain and drum. Under operation, the chain is rotated by a sprocket powered by a hydraulic engine. The bracket holding the engine experiences bending which creates an offset between the chain and sprocket. It has been confirmed that the sprocket could slip as the hose experiences great tension. There has also been confirmed that the sprocket teeth and chain have large clearances, resulting in a less smooth chain and sprocket engagement. Looking from a production perspective, the chain setup contains a lot of welding, which may influence the mesh between the sprocket and the chain. In addition, it's time-consuming. From the maintenance perspective, if the whole chain were to be replaced, the whole drum needs to be detached from its frame, which is not preferable. Even though replacing the sprocket is simple, it may not have the same mesh as before, which intensifies the wear. A cover is also missing, deviating from the basic EHS of rotating devices.

Throughout the design work, a synchronous belt was decided upon, eliminating a cumbersome power transmission design. Synchronous belt drive gave advantages such as eliminating lubrication need, obtaining a no-slip condition, smooth engaging, constant mesh, easier to maintain and change parts, more cost-effective towards the user, as well as being quieter and simpler than the old system. In addition, one of the belt pulleys was set to transfer torque only through clamping force, working as a torque overload protector. Prototype testing was completed without any load, and the functionality turned out good. Assembly time decreased, as well as own produced parts. Since this thesis has a very theoretical approach, testing and prototyping are essential for further design work and improvements.

In the future, HMR is looking to test the prototype with realistic loads to determine the next approach. Their goal is to update their old design, using this thesis as a base layer. Both the discussion and conclusion chapters mention relevant parts to consider for further design improvements.

Acknowledgements

This bachelor thesis has been written at the University of Stavanger, spring 2022. As soon to be a mechanical engineer, the knowledge obtained over the past three years, is put to test.
A summer job at HMR Voss AS gave me the opportunity to write my thesis with a company.
The thesis is based on upgrading an old design for operating a cable drum, which can deliver and spool in manure hose(s), used to deliver manure to e.g, a manure spreader.

I would like to express gratitude to the publishers *"Fagbokforlaget"* and *"Wiley"* for letting me reproduce their content, regarding the books *"Dimensjonering av maskindeler"*, written by Gunnar Härkegård, and *"Callister's Materials Science and Engineering"*, written by William D. Callister, Jr. and David Rethwisch. Also, a special thanks to Hirpa Gelgele Lemu for giving a permit to reproduce his compendium in the course *"MSK210 Maskinkonstruksjon"*, which is unpublished.

In addition, Rubix Group has provided with design manual and general information regarding the synchronous belts with a permit to use.

Regarding the semester, I would like to thank Hirpa for allowing me to work as a student assistant within *"MSK210 Maskinkonstruksjon"*. It has given me a chance to get more knowledge and understanding within the core subject of a mechanical engineer.

Finally, a big thank you to my supervisors from both HMR Voss AS and UiS. Audun Fauske and Tom Erik Vange from HMR has given me good assistance and insight along with the thesis, especially Tom Erik. Knowing that I have people to trust when it comes to gathering information around the design work and making things happen in the workshop, the workload tends to feel a bit lighter. Also, the rest of HMR Voss AS is responsible for the effort behind the physical aspects of making and testing the prototype.

Regarding the UiS supervisor, Dimitrios Pavlou has given directions on relevant books and theories, as well as given good feedback. When theories have been unclear to me, Dimitrios has not hesitated to meet up and talk through them, knowing that his time and effort helps his students through frustrating times.

Stavanger, May 2022

Olav Andreas Gjøstein

Abbreviations

3D	Three-dimensional
BMD	Bending moment
C-C	Centre to centre
FBD	Free body diagram
HTD	High Torque Drive
ID	Inner diameter
OD	Outer diameter
RPM	Rotations per minute
SFD	Shear force diagram
S-N	Strength - life

Nomenclature

α	Angle notation
γ_m	Material coefficient
δ	Mid-belt deflection
θ	Angle notation
μ_s	Static coefficient of friction
ρ	Distance from the neutral axis
σ	Normal stress
$\sigma_\perp$	Weld stress perpendicular to weld throat plane
σ_a	Amplitude stress
$\sigma_{allowed}$	Allowable normal stress
σ_{bmax}	Maximum bending stress
σ_{eq}	Equivalent amplitude stress
$\sigma_{equivalent}$	Equivalent normal stress on weld throat plane
σ_m	Midrange stress

σ_{max}	Maximum normal stress
σ_{min}	Minimum normal stress
σ_{max}^{vM}	Maximum von Mises stress
σ_{min}^{vM}	Minimum von Mises stress
$\tau_{\parallel}$	Shear stress along the weld throat plane
$\tau_{\perp}$	Shear stress perpendicular to weld throat plane
τ_{avg}	Average shear stress due to direct shear force
$\tau_{bending}$	Shear stress in weld throat plane due to bending
τ_{max}	Maximum shear stress due to direct shear force
$\tau_{\max weld}$	Maximum shear stress in weld throat plane
τ_{shear}	Shear stress in weld throat plane due to direct shear force
$\tau_{torsion}$	Shear stress in weld throat plane due to torsion
υ	Poisson's ratio
ω	Angular velocity

$8M$	Synchronous belt with 8 mm pitch
$14M$	Synchronous belt with 14 mm pitch
A	Area
A_{res}	Resultant vector in three dimensions
C_0	Static load rating
D_p	Pitch diameter larger pulley
F	Force
F_A	Resultant reaction force in bearing A
F_B	Resultant reaction force in bearing B
F_C	Circumferential force
G	Modulus of rigidity (Shear modulus)
H_a	Allowable power per V belt
H_d	Design power for V belt

H_{nom}	Required power to be transferred through the V belt
H_{tab}	Tabulated V belt power rating
I	2^{nd} moment of area
I_x	2^{nd} moment of area about X-axis
I_y	2^{nd} moment of area about Y-axis
J	2^{nd} polar moment of area
K_1	Angle of contact correction factor for V belt
K_2	Belt length correction factor for V belt
K_b	Belt width factor for synchronous belt
K_s	Service factor for V belt
L	Inside circumference V belt length
$L_{correction}$	Correction factor for V belt length
L_p	Pitch V belt length
M_T	Torsional moment
M_b	Bending moment
M_{res}	Resultant moment
N_b	Number of V belts
P_0	Equivalent static load
R	Stress ratio
R_1	Inner radius of the manure drum
R_e	Yield stress
S_0	Static safety factor
S_e	Endurance limit
S_f	Service factor for synchronous belt
S_u	Ultimate tensile stress
T	Tension force
V	Direct shear force
VV	Belt setup where both pulleys are V groove shaped

V_{res} Resultant shear force

W_b Section modulus

Z_1 Number of teeth for largest pulley

Z_2 Number of teeth for smallest pulley

a Weld throat width

d_p Pitch diameter smaller pulley

l Length

n_d Design factor for V belt

n_{safe} Safety factor

p Power

$p_{corrected}$ Corrected power rating on synchronous belt

p_{design} Design power for synchronous belt

$p_{required}$ Actual required power for synchronous belt

$p_{tabulated}$ Tabulated power rating for synchronous belt

r Radius

r_v Speed ratio between drum and hydraulic engine

t Weld leg

t_x Axial stress vector at throat width

v Tangential velocity

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1. Introduction

1.1 Background

After completing my summer job at HMR Voss AS, a bachelor thesis was interesting for both employer and employee (me). As the demand for their manure-spreader increases, production also increases. The spreader needs continuous manure feed through a manure hose which is always connected as it drives around the acre. Figure 1-1 shows the drum ready to be attached to a tractor. By having a stationary cable drum outside of the acre as it spools the manure hose in and out, it's easy to manoeuvre the hose. A drum is also convenient for storing the equipment in the off-season. HMR is under constant development of their spreader, aiming at more electronics and automation. All the work done on the spreader has resulted in no work done on the drum assembly. Since the drum is a very essential accessory, the demand will increase. A farmer needs solid and durable equipment. If something were to break, it should be simple to fix by the customer.

The main task is to upgrade the mechanical power transmission. Some of the old designs doesn't hold up when it comes to durability. Some issues are e.g., power transmission slipping, high clearance between the gearing teeth and deformed bracket. Low maintenance and cost-effective are important keywords. Area of interest is mainly agricultural, but HMR has also been delivering to docks. Having equipment that can be used across different disciplines is convenient. In this thesis, the focus will be on the agricultural side, assuming the customer uses the hydraulics from a tractor.

Figure 1-1 GS2 setup

1.2 Scope and limitations

A new design aims towards a solution with low-maintenance needs and high durability. Since the existing product is compatible with modern tractors, it will be the base layer for the new designs. The main tasks will be:

- New mechanical power transmission design
- Finished prototype within bachelor's deadline

Independent of the new design, challenges will occur. The following problems and challenges will be focused on when doing the new design work:

- Keeping the high torque and low speed
- Investigate parts that experience loads different from the existing model
- Analysis and calculation of different parts
- Keep the number of parts at a minimum
- Avoiding complex manufacturing and assembly methods
- Cost

The new designs will be built around the existing drawings of the current frame and drum. Through design development, 3D modelling will be used to find a suitable design. Use of FEA as an aid for analysing stresses and deformations. Based on the created design, technical drawings and parts list will be developed. The price estimate of a new design will be compared to the existing design. Production and testing of the prototype would include mounting procedure, production line, functionality, and strength test.

Production and testing are important for this thesis to determine if HMR should invest in a new design.

1.3 Prerequisite knowledge

The thesis will encounter several areas within the mechanical aspects. It is expected that the reader has knowledge of basic mechanical concepts, where maths and physics act as a base layer for most of the theories behind them. Together with the theoretical approach, Autodesk Inventor FEA will be aiding in analysing several parts.

1.4 Thesis Structure

First chapter will formulate the background of the thesis, as well as some limitations when it comes to the design and financial sides. Further on, relevant theory will be covered. Since this thesis covers many subjects, but not in-depth for all subjects, the theory part will be limited to the most relevant ones. The rest will appear in the methodology. Design work methodology is a keyword in the thesis structure and will be split up into several chapters, trying to have a decent overview of the different stages within it. This thesis will not have a specific chapter for results.

Results will be presented as the design work advances, with some relevant discussions and conclusions. The results will be generated by combining both design manuals and hand calculations to determine designs and solutions, together with knowledge from HMR and distributors. At some stage, a proposed design will be presented and sent to production. As this thesis is a very theoretical approach, discussions will be made correspondent to the many questions regarding the theoretical vs. physical aspects. Conclusions shall then be made, with their basis in the whole thesis. In the Appendices, pictures from the production and prototype, technical drawings, and both general information about and design manual for the synchronous belt is attached.

2. Formulation of the problem

2.1 Manure cable drum

HMR Voss AS has supported the agricultural area with a broad range of agricultural equipment of high quality. Product development, design and production takes place at their factory plant in a small town called Voss. The development of high-quality agricultural equipment has been one of HMR Voss' trademarks since its establishment [1].

The manure spreader developed by HMR can be attached to e.g., a tractor. By having a long hose attached to it, delivering manure continuously, it can drive around the field spraying manure, both effective and evenly distributed. Together with the spreader, a cable drum is used to spool in (and out) the manure hose after use. Hydraulic power is transferred to a mechanical power transmission, which then rotates the drum, which is connected to a stationary tractor. Different drum sizes can be ordered and are easy to replace [2].

As of today, the system exists of a roller chain system, as shown in Figure 2-1. The chain is welded onto the drum in a circular pattern, creating a lot of weld metres. An internally placed sprocket then drives the chain, rotating the drum. The sprocket is driven by a hydraulic engine, powered by the tractor's hydraulic system. The bracket attaching the hydraulic engine experiences bending, creating an offset between the chain and sprocket. One owner of a drum has confirmed that the sprocket slips when the hose experiences great tension. After stopping the operation, the sprocket will then start engaging again. There has also been mentioned large clearance between the sprocket teeth and chain, making start and stop less smooth.

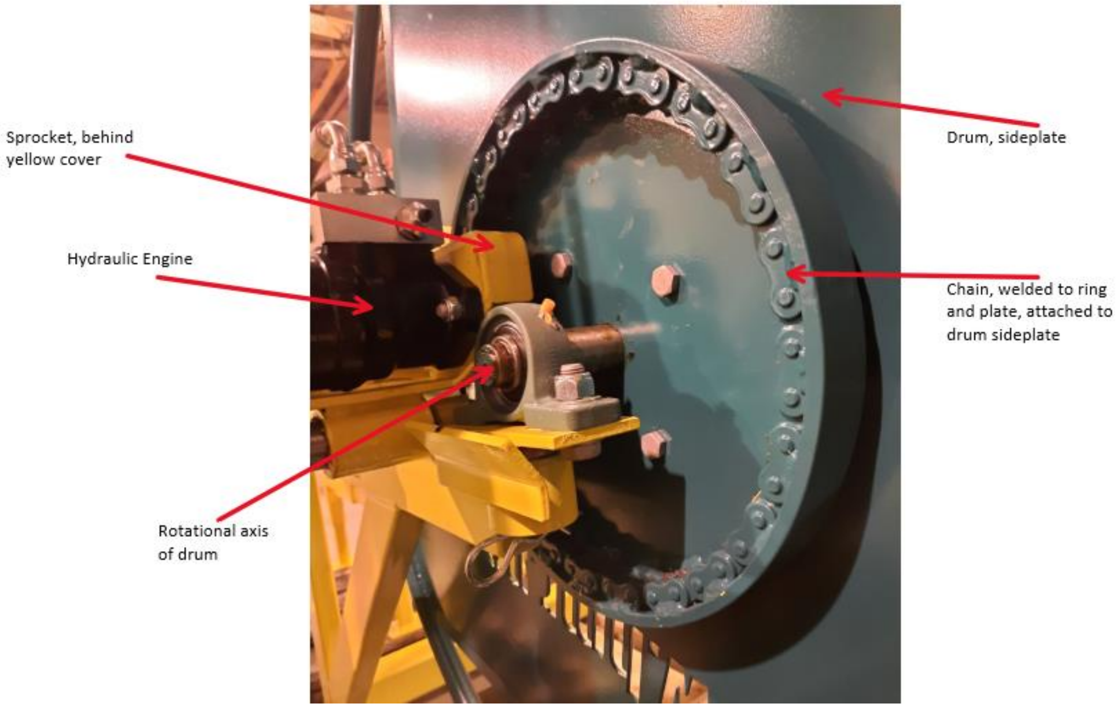


Figure 2-1 Existing product power transmission

After some time, the wear out on the sprocket will match the wear on the chain. Changing the sprocket may intensify the wear on the system since the new sprocket and old chain won't fit perfectly. Simultaneously, changing the whole chain assembly is not as easy as the customer may want it to be, as the drum needs to be disassembled from the frame.

It's uncertain how precise the chain and sprocket mesh are when mounted. An inconsistent tooth mesh influences the lifespan of the power transfer parts and relies on the chain being circular and centred with high precision. If a more precise mesh can be obtained, it contributes to lowering the wear of these components and increasing their lifespan. In addition, lubrication of the chain and sprocket has a direct impact on the wear. Performed lubrication intervals depend on the owner, where the "follow up" on lubrication (and general maintenance) varies from person to person.

As mentioned above, making the chain laying in a perfectly circular pattern has been a challenge. All the welding can create an unwanted movement of the chain as it undergoes expansion and contraction due to heat. In addition, it's time-consuming. Possibilities of having a power transmission system with a simpler way of manufacturing will be considered.

While the current power transmission design has no cover, it's exposed. Having an open system with moving parts doesn't follow basic EHS (Environment, health, and safety). Also, a cover provides keeping dirt, water etc. away from the moving parts. By having a cover that can be easily detached, both inspection and changing of parts will be more accessible.

A closer look at the existing system will be covered in the design study and the design work

2.2 Design limitations

Positioning of the frame and drum will most likely stay the same. Frame will be connected to the tractor and the hydraulic motor will stay on the frame. The mechanical power transmission will be linked between drum and motor, also interacting with the frame. Positioning of the motor and transmission will most likely change place. This depends on which type of power transmission is chosen. Overall, it will have many of the same characteristics. In terms of space, the thesis will aim at keeping the system simple and compact.

2.3 Financial limitations

Finding a sweet spot between price per unit and the customers desire to buy the product, without being too expensive to manufacture. E.g., if the price increases relative to the existing, can it be compensated by creating a more reliable system? A profit is of course necessary for the company making the product. The customers buying this product won't be the ones with the highest salary, so aiming for a reasonable price is critical.

3. Theory

3.1 Fatigue

The theory about fatigue is taken from the book *"Callister's Materials Science and Engineering"* [3].

Of all metallic failures, fatigue is estimated to cover about 90% and stands for the single largest failure cause in metals. Fatigue failure happens in structures subjected to dynamic and fluctuating stresses, which makes it possible for failure to occur at low-stress levels, considerably lower than the yield and tensile strength for static loads. The failure is brittle-like and occurs due to the initiation and propagation of cracks, typically creating a fracture surface perpendicular to the applied tensile stress direction. Several variables are making the fatigue behaviour for engineering metals sensitive. These could be surface effects, geometric design, mean stress level and metallurgical variables, as well as the environment.

Two important parameters that characterize a material's fatigue behaviour are fatigue strength and fatigue life. Fatigue strength relates to a specific stress level, where failure occurs, read of at a specific number of cycles. Fatigue life is the number of cycles that causes failure at a specific stress level.

A higher mean stress affects the fatigue life, i.e., a higher mean stress decreases fatigue life. In addition, several factors affect fatigue life, especially surface and geometric factors. Usually, maximum stresses occur at the surface, where most cracks leading to fatigue failure emerges from the surface. Adjustments in both design and treatment can avoid these cracks to occur (or slow down the process). Considering the design adjustments, locations that acts as stress raisers (holes, keyways, grooves, threads etc.), can be tuned by e.g., avoiding sharp corners or by increasing the radii of curvature in rounded fillets, usually at dimension changes. Surface treatment is also a common practice, making sure that the surface resists crack initiation. Treatments could be polishing, shot peening, case hardening etc. These treatments won't be discussed any further but can be an option for further development.

3.2 S-N diagram

S-N-diagram, also called strength-life diagram, provides the material fatigue strength in MPa, S_f, versus cycle life, N. Ultimate tensile stress is defined as S_u. For iron and steel, the S-N curve becomes horizontal at some point, which represents the endurance limit S_e, often called the fatigue limit. The endurance limit occurs somewhere between 10^6 and 10^7 cycles. Any applied amplitude stress below the endurance limit won't result in fatigue failure, no matter how many cycles [4]. An example of a S-N diagram is shown in Figure 3-1.

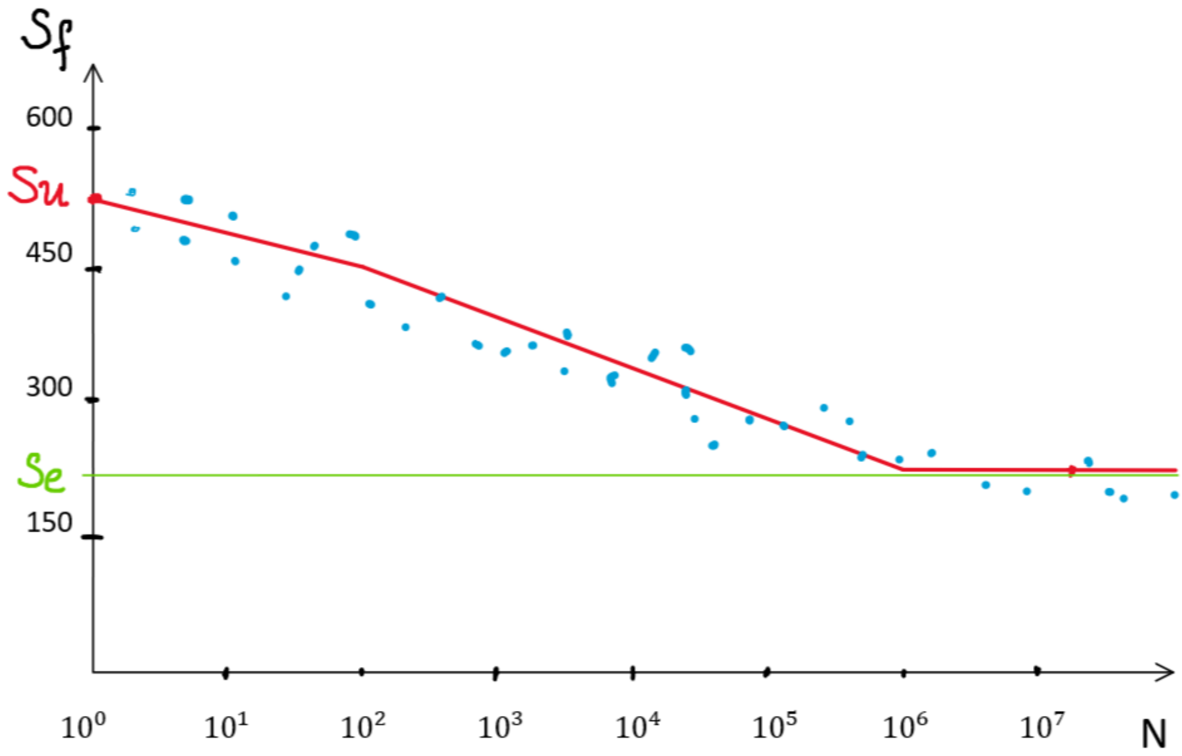


Figure 3-1 Example on a S-N diagram, inspiration from [4]

Most S-N diagrams are read using the amplitude stress. Such diagrams are often for reversed cycle situations i.e., mean stress equals zero. For a reversed cycle, the stress ratio R = -1. S-N curves can display several fatigue curves for different mean stresses [3]. It is then crucial having the correct information about the S-N diagram and its used data values.

3.3 Goodman rule

When using the S-N diagram, the amplitude stresses that reads the fatigue life are used when the mean stress is zero. The project encounters cyclic stresses with non-zero mean stress. By using the Goodman rule, equivalent amplitude stress can be found, that simulates a mean stress equal to zero. The following criteria decide whether a mean stress correction should be made, where the stress ratio R [5] should be

$$R = \left|\frac{\sigma_{min}}{\sigma_{max}}\right| = 1 \tag{3.1}$$

If $|R| \neq 1$, a mean stress correction should be made. That implies the use of the Goodman rule to obtain an equivalent amplitude stress, simulated with a mean stress equal to zero.

$$\frac{\sigma_a}{\sigma_{eq}} + \frac{\sigma_m}{S_u} = 1 \tag{3.2}$$

From Equation 3.2, σ_a, σ_m, σ_{eq} and S_u are the amplitude, mean, equivalent amplitude and the ultimate tensile stress, respectively. By solving for the equivalent amplitude, the S-N diagram can be used to obtain the life cycle of a material, based on a simulated zero mean stress [5].

3.4 Synchronous belt

Synchronous belts are used for a wide range of applications from low to high speed, low to high torque, and for timing purposes. Main belt is made of a rubberized fabric which is coated with nylon fabric. Inside the belt, along with the pitch, there is wire(s) taking the tension load. Synchronous belts stretch insignificantly and has the "no-slip"-condition. The efficiency can range between 97 to 97% with the correct setup and installation method [4].

3.4.1 Geometry

There are several types of tooth geometry. The classic timing belt has a typical trapezoidal geometry [6], represented in Figure 3-2. Trapezoidal profiles are especially suitable for conveying applications and linear positioning. The engagement between the belt and pulley results in good capability of transmitting force and low backlash. Unfortunately, the geometry results in high-stress concentrations at the root, which may result in high wear rates as the transmitted speed or torque increases [7].

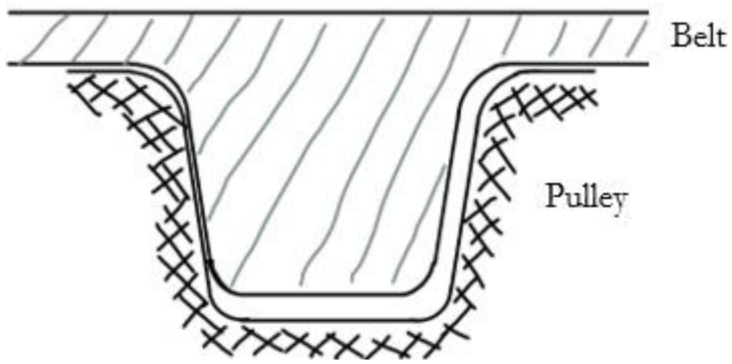


Figure 3-2 Trapezoidal tooth geometry, pulled to the left, inspiration from [7]

The curvilinear geometry is a more rounded geometry which is applied in the HTD belts used for the manure drum design, shown in Figure 3-3. In addition, it has a deeper tooth depth than the trapezoidal design. The mesh experiences a smoother transition than the trapezoidal, which also makes it quieter under operation. Nonetheless, it will have a greater backlash than the trapezoidal, making it less suitable for applications requiring high position accuracy [7]. The curvilinear geometry provides a greater contact area and better stress distribution (located in the middle of the tooth, rather than at the root), allowing a higher overall loading on the belt [8].

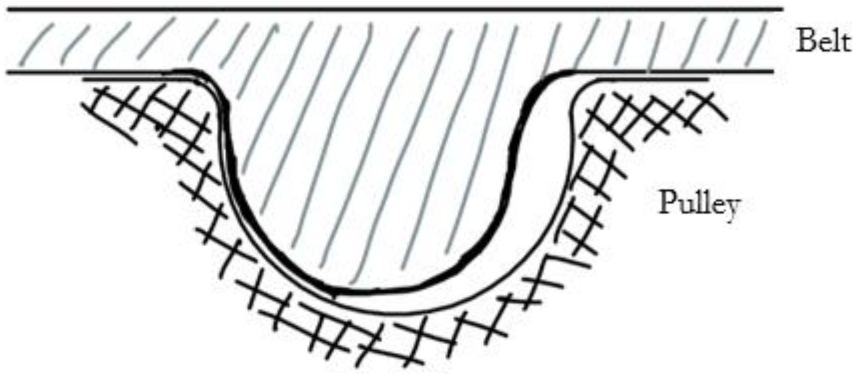


Figure 3-3 Curvilinear tooth geometry, pulled to the left, inspiration from [7]

Figure 3-4 shows the PowerGrip HTD belt components. The backing is durable and protects against environmental pollution. It also prevents (at least slows down) frictional wear if the back of the belt were to transmit power [6]. Backing material that requires a tougher high friction outer layer is represented by e.g., foam, where Neoprene is a common type [9]. The tensile member located at the pitch line is represented with chords made from fibreglass. They contribute with necessary strength, as well as high resistance against elongation. In addition, the fibreglass tensile chords provide flexibility for the belt. The nylon facing is protecting the tooth surface under operation [6].

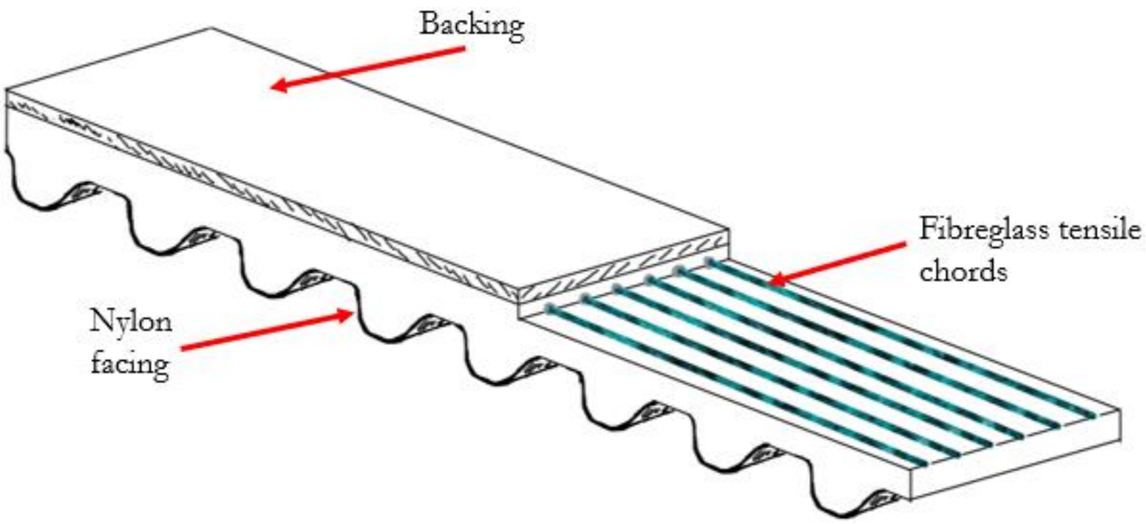


Figure 3-4 PowerGrip HTD belt components, inspiration from [6]

Pulley that is supposed to engage with the curvilinear toothed belt is partially illustrated in Figure 3-5. Pitch diameter represents the pitch circle, which the pitch of the belt coincides with, i.e., the fibreglass tensile chords [6].

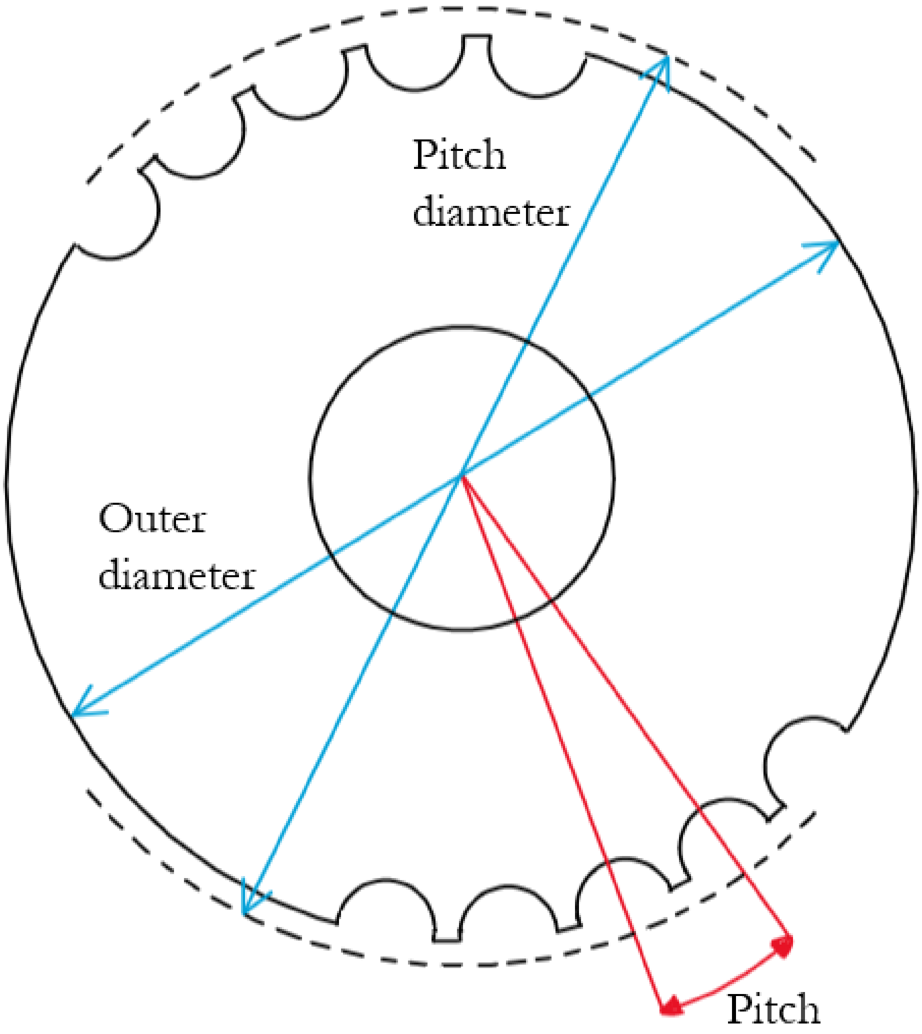


Figure 3-5 Belt pulley with curvilinear geometry, inspiration from [6]

3.4.2 Failure of synchronous belts

Most of the content about failure is taken from the article "*Synchronous belt failures: Six ways they can occur*" [10].

Even though the synchronous belt has many advantages, the installation is as important as anything else when it comes to having a fully functional belt setup. Even if the installation has been done correctly, belt performance can still be influenced by unexpected application conditions, as well as using components unsuitable for its operating requirements. As a result, several types of belt failures may occur.

When installing the belt and pulleys, there are three important factors. Angular misalignment, parallel misalignment, and installation tension. Misalignments, illustrated in Figure 3-6 and 3-7, can cause wear on both pulley and belt. The pulley flange often fails due to an angular or parallel misalignment, where the belt exerts a force against the pulley flange. In addition, this will also result in edge wear of the belt. It's then important to use correct belt and pulley sizes, as well as correct alignment from the start.

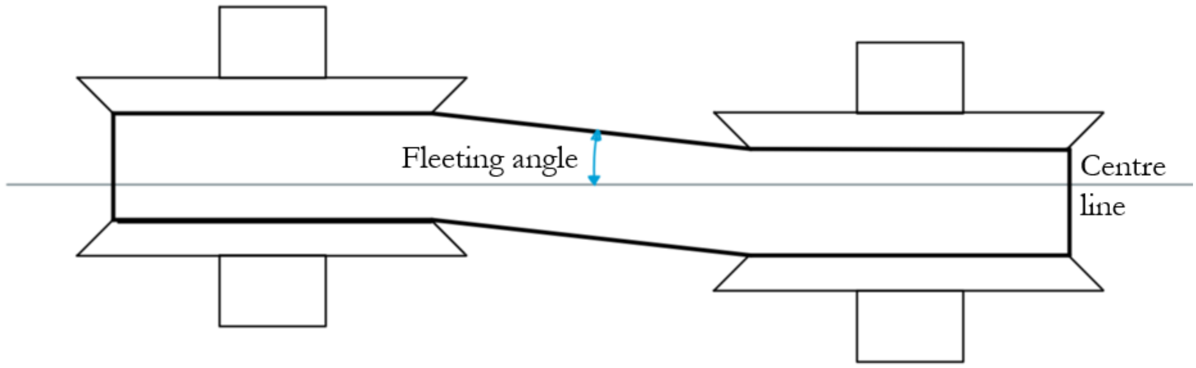


Figure 3-6 Parallel misalignment, inspiration from [6]

Misalignment can also cause problems due to belt tracking. Anyhow, belt tracking is normal to some extent, and won't influence performance. Misalignment should not exceed 1/4 ° or 5 mm per metre centre distance [6].

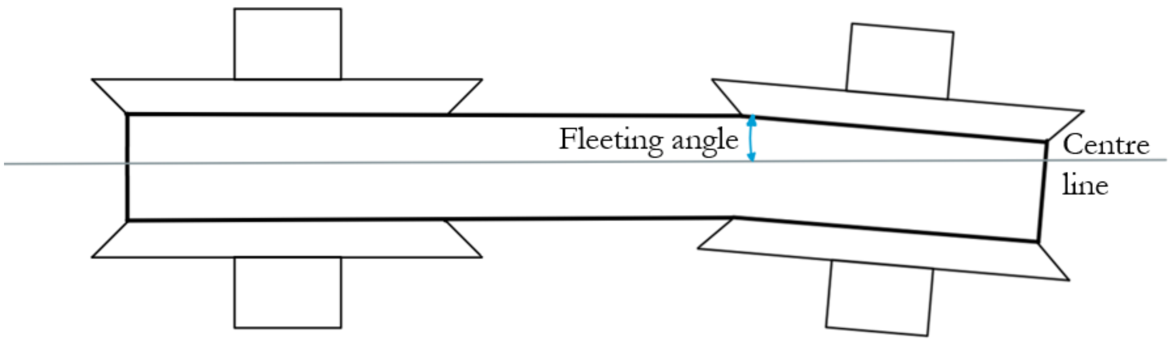


Figure 3-7 Angular misalignment, inspiration from [6]

Assuming the alignment is within the required settings, belt tension needs to be set properly. Inadequate belt tension can cause e.g., tensile break, excessive tooth wear, tooth shear, and ratcheting of belt teeth. Tensile break is more common to happen due to severe shock loads, mishandling of the belt, debris, or a too small pulley diameter. Same conditions can cause excessive tooth wear, but also conditions like a damaged pulley, or a pulley with insufficient hardness. Tooth wear does generally not affect the life span under normal operating conditions.

Tooth shear can be a result of both shock loads and misalignment, which results in shearing between the belt body and tooth root, leading to a belt missing its teeth. In addition, insufficient tension can cause the belt to "self-tension". Self-tensioning happens when the teeth forces themselves out of the pulley grooves, but not to the point of ratcheting. As the tooth is no longer experiencing a distributed load, but more of a concentrated load near the tooth flank, bending occurs about the root. The bending induces rotation of the tooth, trying to separate it from the base of the belt.

If the belt setup is suitable for its application of use and installation is carried out correctly, synchronous belts can perform at a very high level with low maintenance.

3.5 Welding

Welding theory is taken from the book *"Callister's Materials Science and Engineering"* [3].

Welding is used to form a single piece from two or more metal parts. In contrast to mechanical bonding like bolting and riveting, the weld bonds are metallurgical, where the filler material forms a fusion joint between the workpiece(s). Welding is a common technique when a one-part fabrication is inconvenient or/and expensive, simultaneously allowing both similar and dissimilar metals to be welded. Arc and gas welding, brazing, and soldering are some of the different welding methods, where arc welding will be used for this project.

During the arc weld, the workpieces and the filler material gets heated up (through high current) to a sufficiently high temperature which causes both to melt. Through the solidification process, the filler material creates a fusion joint between the workpieces, resulting in a solid single-piece part. The process is effective but involves a great amount of heat, which creates a heat-affected zone (HAZ), as shown in Figure 3-8. HAZ is the zone adjacent to the weld and may experience microstructural and property changes. For cold worked workpieces, recrystallization and grain growth can occur, decreasing hardness, strength, and toughness. For general workpieces, residual stress can occur upon cooling, which weakens the joint. From this, the cooling rate (and alloy compositions) also decides on the structure being created after the welding process. Normally, a material too brittle is not desirable, but neither too ductile.

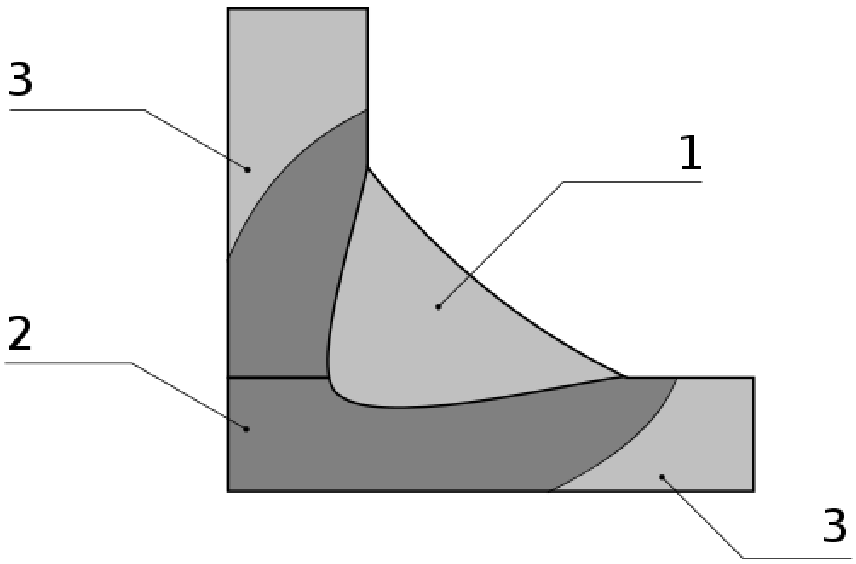


Figure 3-8 1. Fusion zone, 2. HAZ, 3. Base metal of a T-weld joint [11]

3.6 Hydraulic engine

The two following sections are taken from [12].

Normally, fluid gets pumped from a driven hydraulic pump (or any other pressure energy source), that transports fluid to a hydraulic engine. The hydraulic engine then extracts the fluid energy and transforms it to e.g., mechanical energy, typically to rotate the engine output shaft. For the output shaft, a specific volume of fluid is required to turn the shaft one revolution, also called engine displacement. Engines are made with either fixed or variable engine displacement. A fixed engine delivers constant torque, while a variable one can vary the torque and speed. Variable engines can obtain torque-speed ratios to obtain different load requirements.

Torque output is a function of engine displacement and system pressure. Torque ratings are regularly given for a distinct pressure drop across the engine. Normally, the torque available at the engine shaft is theoretical, with the assumption of no mechanical loss. Some datasheets provide mechanical efficiency for specific torque vs. pressure vs. speed at specific load cases.

MR315C/4 has the same looks as a Geroler engine [13]. A Geroler engine uses something called "the orbit principle", having the advantage of generating high power density at a compact size. Figure 3-9 shows the inside cross-section of the gear assembly. Fluid from the pressure pump enters the inlet, creating a pressure side, coloured in red. The blue area describes the low-pressure side. Pressure difference makes an imbalance of the forces, causing the "star" to orbit. For this geometry, the star (which is connected to the output shaft) rotates seven times for one single revolution within the stationary outer ring, resulting in a 7:1 speed reduction. The rollers reduce wear, friction, and improves the low-speed performance, resulting in a longer service life [14].

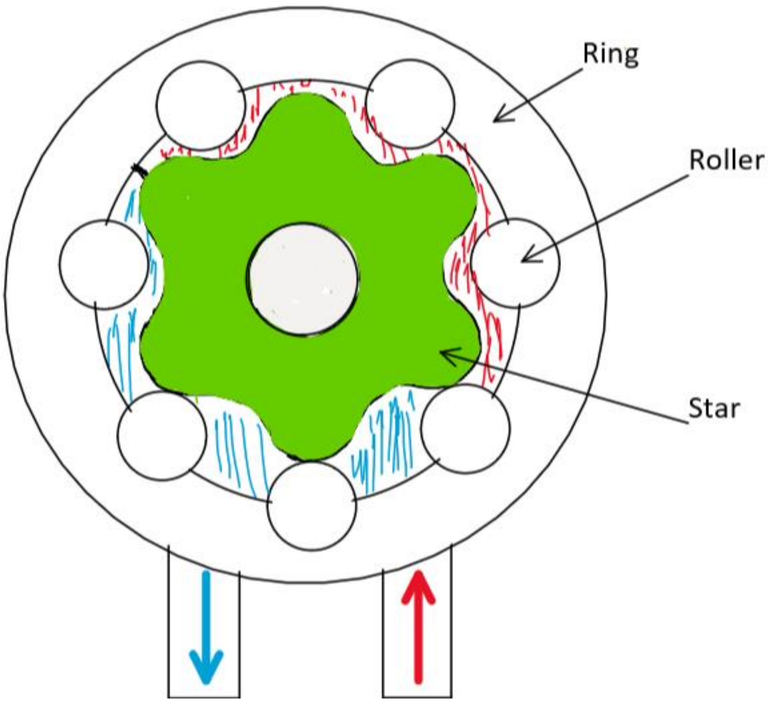


Figure 3-9 Geroler gear mechanism, Inspiration from [13] and [14]

3.7 Normal stresses due to bending and axial loads

The following section has been taken from [15].

In this thesis, drum components will experience bending, torsional shear, direct shear, and axial stresses. Figure 3-10 shows a circular shaft under both bending and axial load. Both loads will generate normal stress, where axial stress is equally distributed across the cross-section, while bending stress is zero at the neutral axis and increases towards the outer perimeter. Bending stress will have a maximum at the outer perimeter (distance c on Figure 3-10), where one side is in tension (top) and the other in compression (bottom). The highest normal stress occurs where both bending and axial stresses acts in the same direction, i.e., both generates tensile stresses.

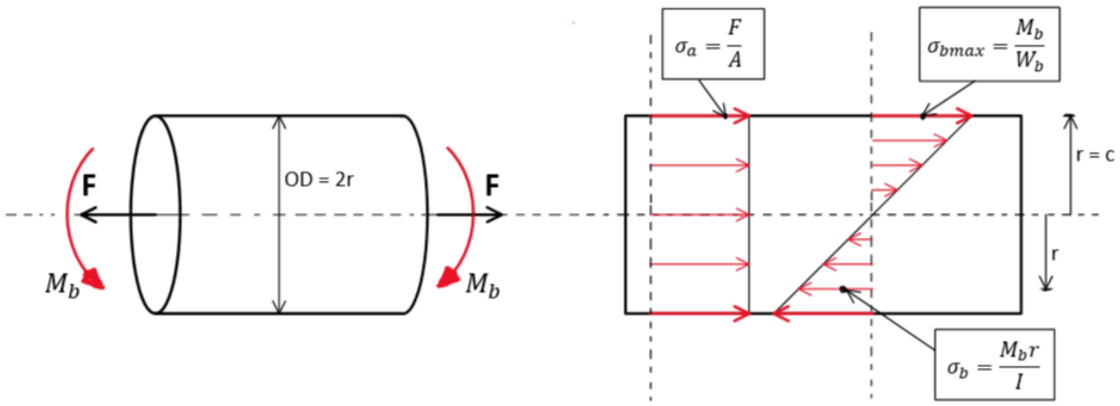


Figure 3-10 Bending and axial stresses from normal and bending loads, inspiration from [15]

Bending stresses can be defined as

$$\sigma_b = \frac{M_b}{W_b} = \frac{M_b \times r}{I} \tag{3.3}$$

Where max occurs at r = c. W_b is called the section modulus, which can be written as I/r, where I is the second moment of area about the bending axis. Axial stress can be written as

$$\sigma_a = \frac{F}{A} \tag{3.4}$$

F is either the compression or tensile force subjected to the body, and A is the cross-sectional area. [4]

3.8 Transverse shear stress due to direct shear force

In addition to the forces acting parallel with the neutral axis, perpendicular forces also create material stresses, called transverse shear stress.

$$\tau_{avg} = \frac{V}{A} \tag{3.5}$$

Like the axial stress, a shear force V is divided by the area A, generating an average transverse shear stress. V/A is not the actual max transverse shear stress. Max stresses occur at the neutral axis, while the outer perimeter experiences zero shear stress, illustrated in Figure 3-11. Max transverse shear stress for a circular cross-section, located at the neutral axis, can be defined as [4]

$$\tau_{max} = \frac{4V}{3A} \tag{3.6}$$

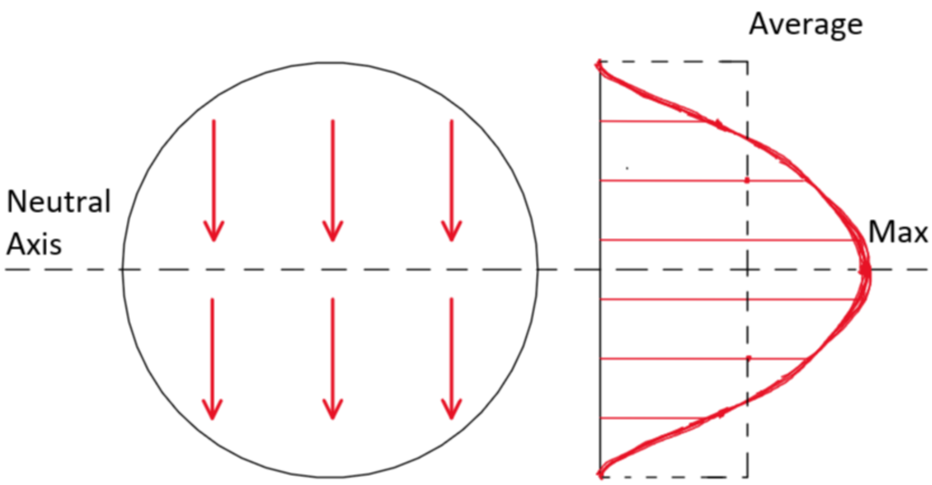


Figure 3-11 Average and max shear stress due to vertical shear force. inspiration from [4]

3.9 Torsional shear stress due to torsional moment

A bar subjected to a torsional moment is said to be in torsion. Figure 3-12 shows a circular solid shaft under torsional moment, M_T. The angle of twist, θ, can be found (with the right-hand rule in x-direction) with

$$\theta = \frac{M_T \times l}{G \times J} \tag{3.7}$$

Where G, l and J are the modulus of rigidity, length to the measured point and second polar moment of area, respectively. As a round bar is subjected to torsion, shear stresses occur (in a rotational pattern) throughout the cross-section. Stresses are constant at a specific distance/radius, ρ, from the neutral axis. These stresses can be written as [4]

$$\tau_{torsion} = \frac{M_T \times \rho}{J} \tag{3.8}$$

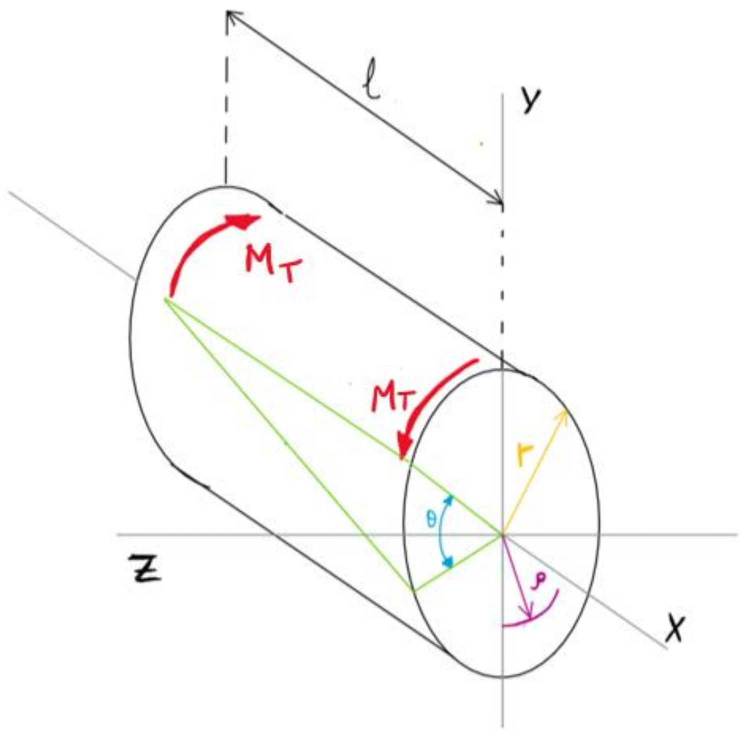


Figure 3-12 Torsion on a circular shaft. Inspiration from [4]

3.10 Circular weld and weld throat

Figure 3-13 shows a simulated load scenario that is relevant for the later weld calculations, which also show the applied forces and moments. Each one contributes to weld stresses.

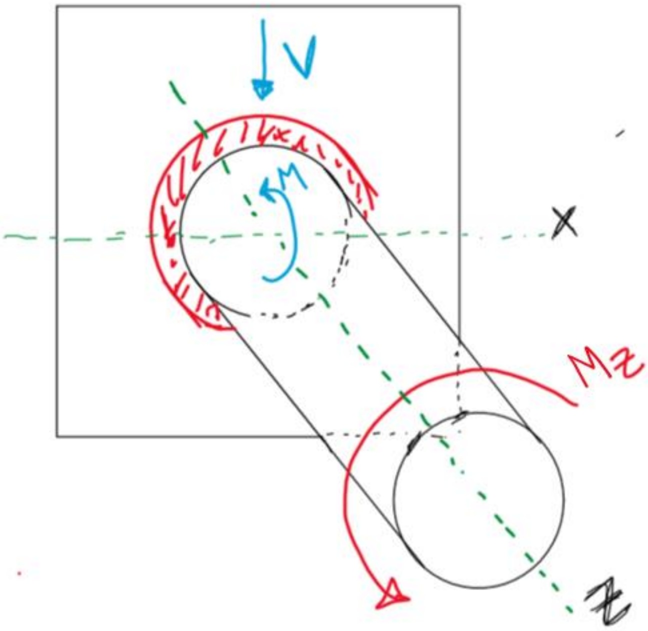


Figure 3-13 Scenario of a welded shaft under several loads

Shear force creates shear stresses and is defined as [4].

$$\tau_{shear} = \frac{V}{A} = \frac{V}{2\pi ra} \tag{3.9}$$

Area A can be defined in Equation 3.9 and refers to the throat area [4], based on the weld throat width, illustrated in Figure 3-14, and length. Looking at the unfolded weld throat plane in Figure 3-15, the shortest circumference is used as the length.

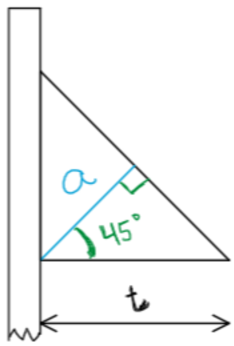


Figure 3-14 Throat width, a, and weld leg, t

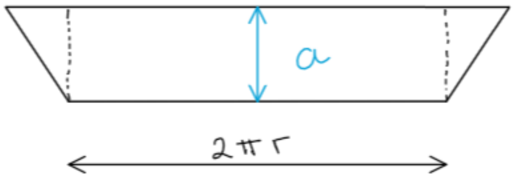


Figure 3-15 Unfolded weld throat plane

Throat width, a, can be found with

$$a = t \times \cos(45) \tag{3.10}$$

Torsional stress can be defined by

$$\tau_{torsion} = \frac{M_T \times r}{J} \tag{3.11}$$

Where M_T, r and J are the torsional moment, distance from centroid and second polar moment of area [4], respectively. Together with Figure 3-16, the second polar moment of area can be written as

$$J = \int r^2 dA \tag{3.12}$$

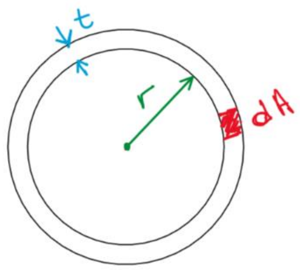


Figure 3-16 Cross section of a circular tube

With aid of the Pythagorean theorem and Figure 3-17, J can be expressed as

$$J = \int r^2 dA = \int (x^2 + y^2) dA = \int x^2 dA + \int y^2 dA \tag{3.13}$$

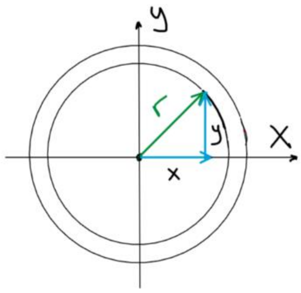


Figure 3-17 Cross section of a circular tube

If radius (constant) and area are the same as the ones defined for the throat area, the second polar moment of area will be

$$J = r^2 \int (ra) d\theta = 2\pi r^3 a \tag{3.14}$$

Bending stress can be defined as [4]

$$\sigma_b = \frac{M \times y}{I} \tag{3.15}$$

Where M, y and I are bending moment, distance from neutral axis and 2nd moment of area, respectively. For a circular geometry, the 2nd moment of area is

$$I_x = I_y = \int x^2\, dA = \int y^2\, dA \tag{3.16}$$

Due to Equation 3.16, J can be written as

$$J = 2 \times I_x = 2 \times I_y \tag{3.17}$$

Implying that

$$I_x = I_y = \frac{J}{2} = \pi r^3 a \tag{3.18}$$

3.11 S355 (St52) and 34CrNiMo6

Table 3.1 shows the material specs for both S355 and some quenched and tempered steels. S355 is a common construction steel that can be welded. By quenching and tempering, the material can be more weldable, tougher, and ductile than ordinary carbon steel. Some can be further treated, e.g., with surface hardening, obtaining a hard surface, but with a tough core [15].

Table 3-1 Material specifications for S355 and some quenched tempered steels, information from [15] and [16]

Material	**Dimensions** [mm]	**Yield Strength** [MPa]	**Tensile Strength** [MPa]
S355 (ST52)	≤ 40 40 ≤ 80	355 335	510 470
NS 13335-05 **34CrMo4**	16 ≤ 40 40 ≤ 100	670 560	880-1080 780-930
NS 13343-05 **42CrMo4**	16 ≤ 40 40 ≤ 100	760 640	980-1180 880-1080
NS 13411-05 **34CrNiMo6**	16 ≤ 40 40 ≤ 100	880 780	1080-1270 980-1180

Note: Specifications for S355 are listed in EN-NS 10025, cited by [15] and the quenched and tempered steels can be found in NS 13201, cited by [16].

3.12 Design workflow

Some key aspects will be mentioned in the workflow, giving boundaries to work within. The chart has been inspired by [4]. Note that the workflow has a general "direction", where steps need to be re-done or revised, making the workflow shift path. There can be many iteration steps without thinking of it. The steps are all different but still similar. It often pays off to revise and take a step back if needed.

Identify need: Can be set off by a particular action or several actions, generating e.g., a need for an improvement or similar. Needs can be identified by doing tests of an existing product, to consult with the mechanics and engineers responsible, or contact the users of the product.

Problem definition: Having a clear thought of the existing challenges that must be met. Overview of specifications, which can be influenced by expected life span, cost, reliability, operating conditions, manufacturing process etc. Design characteristics are also important, like variation in speeds, loading capabilities, stability etc.

Concept and concept design: Generating several design concepts with e.g., schemes and matrices, making up advantages and disadvantages. Designs can be picked out through an iterative selection process. Internal design concept reviews, where concepts that does not survive the design selection, can be revised, improved, or eliminated. Non-eliminated concepts can either be narrowed down to a single concept or analysed further in the next step(s).

Calculations and analyses: From the problem definition and concept designs, analyses and mathematical models for the simulation of a real physical system can be generated. Manual calculations and computer-aided analyses are often used for refining the design and the looks of it. It is important to discover whether the design concept(s) is feasible, reliable, functional, and sometimes good looking. In addition, identifying parts that needs to be checked out before anything else can be done, can save a lot of time. That is all a part of the structure of it.

Evaluation and production: The final test and physical evaluation, which is often led out in the workshop and test facility. Is it feasible to assemble and manufacture? Does it meet its needs? How will it stand in the market? Is the cost satisfying? These are some of the key points, which can lead the process back to the start, or any of the other steps. New products will most likely need to be revised and optimized.

Presenting the design: Selling a product is not always straight forward. When the product is ready, the information about the product should be advertised to the consumers. This is a vital step, which is the seller's responsibility, i.e., a seller's job.

4. Choosing a power transmission system

4.1 Considerations

Before working on a new design, some considerations will be mentioned, which will contribute to "shape" the new design into something feasible, for both customer and the manufacturer. Some of the considerations taken are:

- Two-way drum rotation
- Accessibility of the system. Drum staying in its place when doing repairs with enough space for tools and inspection
- Owner of the drum should not do a lot of work to prepare the maintenance work
- Dynamic, shock and/or static loads
- Change of loads due to variable mass on the drum (hose) i.e., change of frictional surface between hose and ground
- Rotational friction for the drum (bearings etc.)
- Types of connections i.e., bolts, welds, press-fit
- Off-set and misalignment
- Low RPM and high torque

4.2 Specifications

The new design needs some specifications and assumptions to work with, so that the shop-made parts, as well as the 3rd party parts are suitable for its use. Obtaining the following specifications will aid in further development of a feasible design:

- Max torque
- Max RPM
- Max power
- Dimensions

4.3 Possible candidates for the transmission system

Three types of transmission systems have been picked out for further investigation. First off, a comparative look at each type and some of their advantages and disadvantages. Afterwards, they will be compared in a comparison matrix, which gives a better overview of important design factors. Later, a system will be chosen, based on the matrix, but also on some simple calculations based on the tractor's output, hydraulic engine and the required power.

4.3.1 Belt drive

Absorbing shock loads, damping out and isolating effects of vibrations are many of the advantages of flexible elements (belts, ropes, chains etc.). In the industry, roller chains are often replaced by belt drives for many types of applications. As a replacement, they often simplify the design of the machine, which (mostly) includes a reduction in cost [4]. Efficiency and energy loss are affected by torque and speed loss. Torque loss due to required energy to bend the belt around the sprocket or sheave, and speed loss due to belt slip and creep. Lifespan can be increased by adding an idler, which maintains the belt tension. Most common idlers are fixed, manually adjusted, or spring-loaded/automatically adjusted. The type of idler depends upon the application (peak loads, static loads, maintenance, belt and sheave wear, elongation). Idler adjustment is not the only solution to maintain belt tension, but also direct adjustment on e.g., the hydraulic engine. Speed, power, efficiency, and cost varies between the different type of belt setups and determines which is the most suitable [17]. Here are some examples of different belt types:

i. **Synchronous**
Synchronous belt has a wide range of operating speed, high load capacity, both long and short belt applications, and high efficiency [17]. Because of the toothed pattern on the inside of the belt, it's suitable for high-power transmissions and timing applications [18]. They are often used where precision is an important factor. Synchronous belt experiences minimal friction as the belt tooth enters and exits the grooves on the pulley [17].

ii. **V belt**
V belts uses more energy by bending around the sheave than the synchronous, due to a thicker cross-section. Friction caused by the wedging action between the belt and sheave results in heat loss, generally more than the synchronous belt. But, the V belt provides an increased frictional contact effect, providing higher speed and load capability [19]. If poorly maintained, slipping may occur. Slipping is useful if torque overload is accounted for. V belts are also often used in agricultural machinery [17].

iii. **Flat belt**
Drive efficiency of about 98 %. Flat belts produce little noise and absorb more torsional vibration than V belts. Flat belts have no upper limit for centre distance limits [4].

Belts can handle large power, whereas V belts can handle at least 75 kW [4]. It is assumed that cost will increase with increased belt power capacity.

4.3.2 Chain drive

Like the belt drive, the chain drive is also categorised as a flexible element, with long service life. The difference will be the "no-slip"-condition, resulting in no creep. Roller chain needs regular maintenance and lubrication, or else it won't perform at its peak level. With less effective lubrication, horsepower needs to be reduced to avoid shortened lifetime. Without lubrication, a system with greater wear capacity is necessary. Poor ineffective lubrication usually results in an unsatisfactory chain life [17]. Chain speed is the basis for choosing the correct lubrication method [20].

Chordal speed variation/effect is based on the sprocket's number of teeth, where two radii appear at the sprocket. When a chain is engaged around the sprocket, it alternates between the two, creating a speed variation [19]. Such variations can cause vibration within the system and is the main cause of noise. Speed variation can be decreased by increasing number of teeth [20].

For higher loads and lower speeds, a chain with a large pitch diameter is suitable. As the chordal effect creates speed variation, the number of teeth on the smaller sprocket needs to be determined. Slow speed means fewer teeth required, and high speed means more teeth required. The cost will increase when the number of teeth increases. A sprocket with larger number of teeth is more expensive to manufacture. The large sprocket is normally limited to 120 teeth [20].

Many hours of run time are the most common cause of failure. Actual failure, wear of the rollers on the pins or fatigue of the surface of the rollers [4]. After some runtime, the wear on the chain "fits" the wear on the sprocket. Based on experience, if the chain then gets replaced, the sprocket should also be changed. If not, the wear on either part won't mesh as before. This causes more rapid wear.

4.3.3 Gear drive

Gear trains have the purpose of transmitting motion from one axis to another. It requires at least two meshed gears [20]. Driving gear (input) as number 1, the driven gear (output) as number 2, and the rest as idlers. In addition, the gear drive handles higher loads than the chain and belt. The use of multiple gears makes it possible to change torque, directions, speed, and gear ratio. Too many gears may reduce the mechanical efficiency [18].

Backlash may occur over time. The backlash creates a gap between two meshing teeth. At high power outputs, this can result in a shock throughout the whole gear train. This may result in gear teeth damage [18]. Alignment and correct instalment are crucial.

A simple gear train has one gear for each axis, shown in Figure 4-1. Compound train has several gears on one axis [20]. Spur gears are the simplest of all types. Helical gears have teeth inclined to the axis of rotation which is less noisy because of the gradual engagement of the teeth during meshing [4].

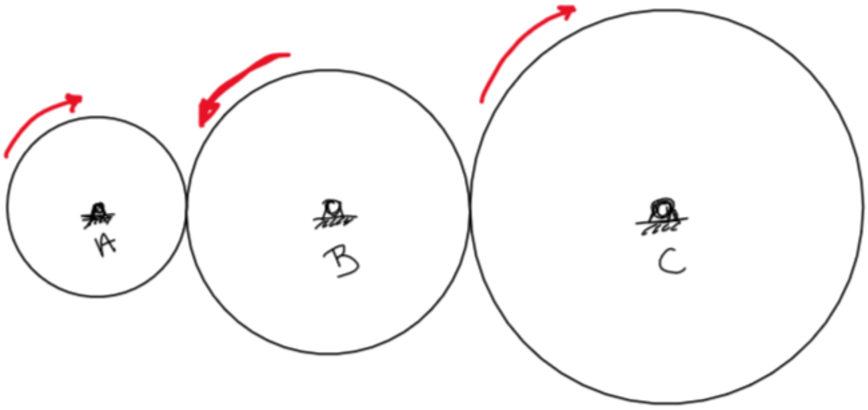


Figure 4-1 Simple gear train, inspiration from [4]

Planetary gear trains refer to one or more gears orbiting about trains' central axis, as shown in Figure 4-2. The main difference from an ordinary gear train; they have moving axis or axes [20]. Changing between input, output and fixed axis/gears, different speed and torque can be obtained. They can provide a remarkably high-speed reduction, even in small spaces. Planetary gears can be found as compounds, not just simple and the range of types are many [17].

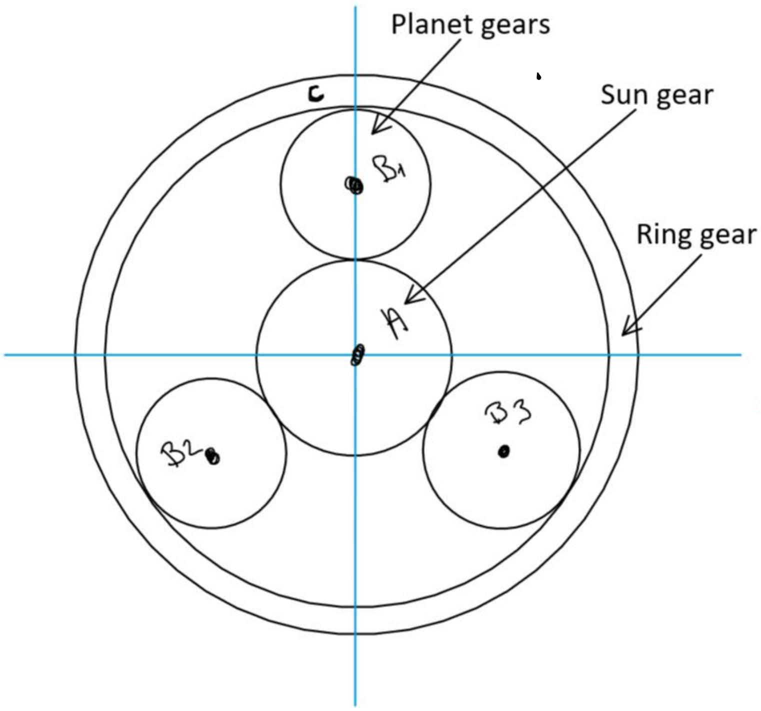


Figure 4-2 Simple planetary gear train, inspiration from [17]

By looking at today's solution, the drum shaft lays on the frame, rotating, held in place by a bearing on each side. If a simple planetary gear were to be used, the sun gear (middle gear) would coincide with the drum's rotational axis. It should then be figured out a way to easily assemble the planetary gear.

4.4 Comparison matrices

Based on the transmission system candidates, a comparison between the three can aid in choosing the most suitable system for this thesis. First off, Table 4-1 contains a matrix comparing the general advantages and disadvantages made independent of the manure cable drum. Afterwards, the matrix in Table 4-2 compares the advantages and disadvantages the three power transmissions have towards selecting a new power transmission.

In Table 4-2, text highlighted in green indicates the most important advantages, while the text highlighted in red shows the most important disadvantages.

Table 4-1 General advantages and disadvantages

	Belt drive	**Chain drive**	**Gear drive**
General Advantages [18]	- Low component cost and high efficiency (varies on the type of belt) makes it affordable. - Power can be transmitted over long distances. - Smoother and quieter than chain drives. - Vibration and shock absorbent. - Overload protection - Relatively durable - Flexibility - Lightweight - Low maintenance cost	- No-slip (and creep) and used for timing applications. - Affordable - Power can be transmitted over long distances. - Can power multiple shafts - Little friction and high mechanical efficiency - All kinds of environments and high temperature - Can be put in tight places - Flexibility	- Long service lives - Sturdy - High mechanical power applications - Compact - High efficiency and no-slip - Can be reversed
General Disadvantages [18]	- Velocity ratio varies due to belt slippage/creep - Heavy load on bearings/shaft - Needs tensioning (sag) - Misalignment can cause slip off (depends more on the type of belt) - Enclosure - Both V belts and synchronous have a centre distance limit [4] - Limited power transmission capacity [20]	- Noisy (chordal effect) - Can cause vibrations - Some require constant lubrication - Misalignment can cause slip off - Needs chain tensioning (sag) - Enclosure	- Short centre distances. Need direct contact - Too many gears can reduce the mechanical efficiency - Expensive repair, and generally more expensive than others - Heavy - Little flexibility - Lubrication - Not suitable for shock loads. Damage on gears - Requires precise alignment for meshing

Table 4-2 Project (dis)advantages. Green indicates important advantages, and red important disadvantages

	Belt drive	Chain drive	Gear drive
Project Advantages	- Hopefully no big design impact on either drum or frame - Accessibility of parts - Simple and effective, both construction and principle - May decrease the moment arm on the frame, created by the hydraulic engine - Flexible - "Built-in" torque overload (not synchronous) - Efficiency (depends on type) [4] - Cost (depends on type) - Vibration and shock absorber (more than a chain)	- Hopefully no big design impact on either drum or frame - Accessibility of parts - Simple and effective, both construction and principle - May decrease the moment arm created by the hydraulic engine - Little friction and high mechanical efficiency - Flexible - Cost	- Compact and saves space - Can be hidden inside the drum (planetary) - Robust and durable under the right conditions - No need for tensioning. - High efficiency [4]
Project Disadvantages	- Takes up more space (length) - Belt tensioner - Safety cover takes up even more space (trade-off) - V belt should be running in-between 5-25 m/s [4]	- Requires lubrication - Takes up more space (length) - Chain tensioner - Doesn't prevent torque overload - Noise - Safety cover takes up even more space (trade-off) Larger sprockets/more teeth i.e., manufacturing cost increases	- Requires lubrication - Doesn't prevent torque overload - Expensive - Think that it requires more change in design to fit right. Depends on the type of gear. - Weight (frame needs more strength. Extra weight doesn't help) - Shock loads? - Repair (depends on which type)

4.5 Loads acting on the drum

If the loads can be estimated, a decision can be made upon one of the three transmission systems mentioned above.

4.5.1 Drum setup

By having the correct drum setup, a decision on whether max load from the hose, the tractor, or max power from the hydraulic engine should be done. As for this project, HMR provided information about the bestselling drum setup:

System name: GS2			
Frame name:	GS110	**Weight:** 105 kg	
Drum name:	GS400	**Weight:** 100 kg	

Manure hose types:

Name:	4" Ultraman	**Weight:** 1.16 kg/m	**Length:** 500 m
Name:	4" STD Tilførsel	**Weight:** 1.10 kg/m	**Length:** 700 m

500 m of "Ultraman" weights 580 kg. 700 m of "STD" weights 770 kg. The last mentioned will be used for the calculations.

When rotating the drum with a counteracting load (the manure hose), power from e.g., a hydraulic engine is needed. The power depends on the drum's RPM, assuming the torque remains constant under operation (actually just the initial phase). RPM interval is not known but can be assumed to operate between 5 and 15.

4.5.2 Tension force from the hose

Assuming the whole manure hose is spooled out, the smallest radius, R1, acts as the moment arm. The hose lifts off the ground at an angle θ, wrapping around the drum, illustrated in Figure 4-3. As the drum rotates, the amount of hose on the field decreases. This results in less tension force. Meanwhile, the amount wrapped on the drum increases. Hose build-up on the drum increases its mass and radius, contributing to a higher drum moment of inertia. Since the drum is reasonably wide, it's assumed that the tension force decreases faster than the moment of inertia increases (as a function of the amount of hose wrapped around the drum).

A higher moment of inertia requires higher torque to change the drum's rate of rotation (i.e., acceleration) [21]. In the end, it has the characteristics of a flywheel [22], which may ease the rotation (if constant RPM is assumed) rather than slowing it down.

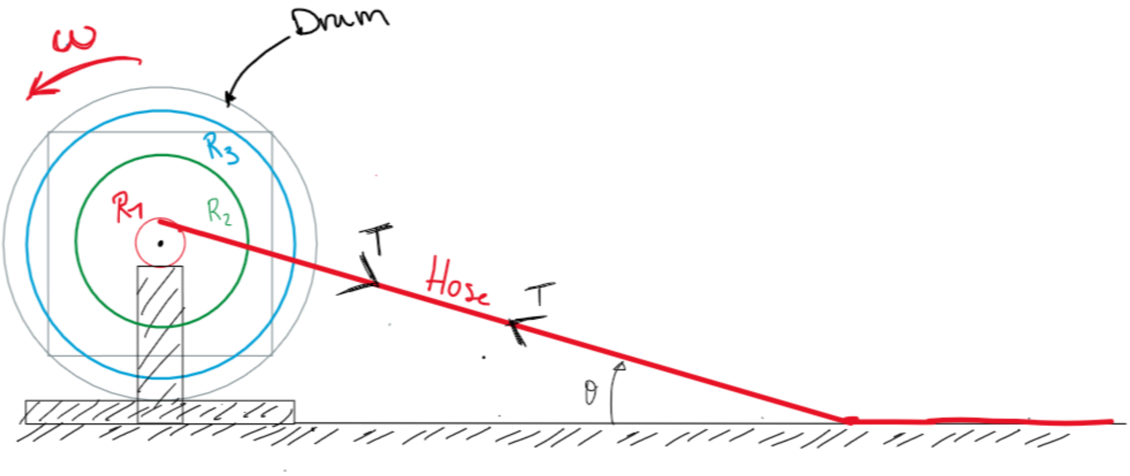


Figure 4-3 Illustrated view of the manure hose pulling on the cable drum

Based on the frictionless rotation of the drum, the tension in the manure hose acts as the main force with radius R1 as the moment arm.

4.5.3 Power created by the load

For the calculations of force, torque, and power, some assumptions and factors are needed:

- Constant drum RPM, neglected acceleration
- $5 \leq RPM \leq 15$
- Tension force as the whole length of hose lays on the field
- Torque based on the inner radius
- Coefficient of friction, $0.35 \leq \mu_s \leq 0.45$ for tire on grass [23]
- Angle between the inclined part of hose and ground is set to $\theta = 20$
- Gravitational contribution on the hose is neglected (as it lifts off the ground)
- Frictionless rotation/bearings
- Manure hose contains excessive manure. Added 10% of the hose weight

Tension force is obtained by the force dragging the hose across the field, Fx, Figure 4-4.

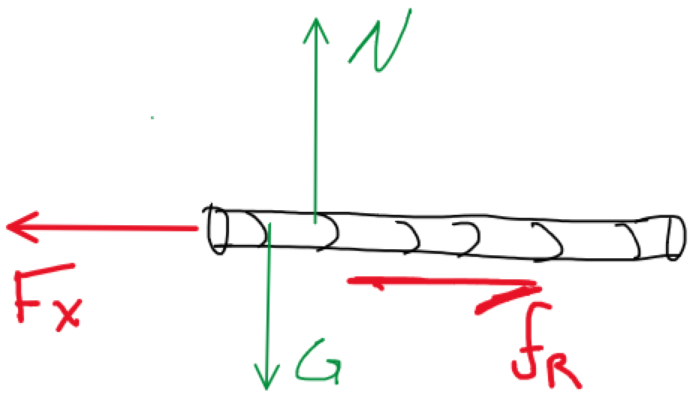


Figure 4-4 Forces acting on the manure hose

It's assumed that there is an increase of 10% in extra weight due to excessive manure. Fx is then defined as

$$F_x = \mu_s \times N \times 1.10 = \mu_s \times m \times g \times 1.10 \tag{4.1}$$

As the hose leaves the ground, the tensional force won't be in the horizontal direction anymore. Figure 4-5 shows T as the tension force, and Tx as the horizontal component.

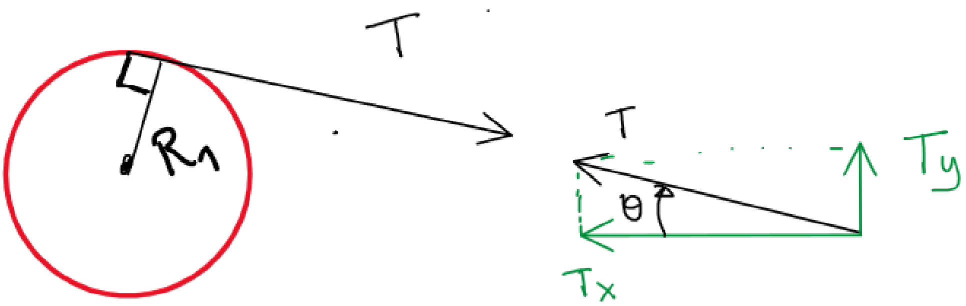


Figure 4-5 Vector components as the manure hose lifts off the ground

In this case, Ty is neglected. Knowing that Tx equals Fx, tensional force can be obtained by

$$T = \frac{Tx}{\cos(\theta)} = \frac{Fx}{\cos(\theta)} \tag{4.2}$$

The tension T describes the resistive force acting on the drum as the hydraulic engine starts turning. T will then create a torque about the drum's axis, with the moment arm R1. Torque is defined by

$$M_T = T \times R_1 \tag{4.3}$$

Power will then be

$$p = \omega \times M_T \tag{4.4}$$

Writing the RPM in terms of angular velocity as

$$\omega = RPM \times \frac{2\pi}{60} = RPM \times \frac{\pi}{30} \tag{4.5}$$

By combining Equation 4.1 to 4.5, a general formula for the required power, as a function of the drum's RPM, can be written as

$$p(RPM) = \frac{\mu \times m \times g \times 1.10 \times R_1}{\cos(\theta)} \times RPM \times \frac{\pi}{30} \tag{4.6}$$

Table 4-3 shows the parameters used to obtain the loads and powers.

Table 4-3 Parameters needed to calculate loads and power

μ_s	m	g	R_1	θ	RPM_{LOW}	$RPM_{HIGH\ drum}$
0.45	770 kg	9.81 m/s^2	0.108 m	20	5	15

Table 4-4 sums up results based on Equation 4.1, 4.2, 4.3 and 4.6, based on parameters from Table 4-3.

Table 4-4 Results from Equation 4.1, 4.2, 4.3 and 4.6

Eq.1	Eq.2	Eq.3	Eq.6	Eq.6
Fx = 3739 N	T = 3979 N	M_T = 430 Nm	$p(5)$ = 225 W	$p(15)$ = 675 W

Results from the applied tension force are low, compared to the hydraulic engine's power capacity, a max of 5 kW. There should not be any problem turning the drum (from a power point of view), even a manure hose at double length.

4.5.4 Power from the tractor

For a regular and semi-modern tractor, the max working pressure and flow can be found, provided by its hydraulic system. HMR suggested using 180 Bar and 100-140 L/min. Equation 4.7 can then be used to find the power.

$$Power\ [kW] = \frac{Pressure\ [Bar] \times Flow\ \left[\frac{L}{min}\right]}{600} \tag{4.7}$$

Using 180 Bar and 100 L/min, the power output from the tractor will be 30 kW. This is far above the max continuous hydraulic engine output, 5 kW, which implies that using the tractor output as a power reference isn't suitable.

4.5.5 Output from the hydraulic engine

Looking at the max output in the MRC315CD/4 engine datasheet [13], the continuous drive condition has a max output of 5 kW. Using the continuous drive as a boundary condition, the engine capacity can be further investigated.

The continuous drive implies that max torque output is set as 360 Nm. Datasheet reads of some outer boundaries, where 130 RPM reads ~360 Nm and 180 RPM reads 260 Nm. For further calculations, 5 kW will be used to decide on the belt drive systems capacity. Comparing the power with the power requirements from the drum load, the hydraulic engine is more than capable. Anyhow, the engine may require more power to compensate for the efficieny loss at such low engine RPMs.

The hydraulic engine's minimum RPM is set to 10. This information should be taken into account when defining the ratio between the driving (engine) and driven (drum) axes, i.e., if the RPM of the drum is 5, the engine should not go below 10.

4.6 Summary and conclusion

4.6.1 Power capacity

From the basic calculations and assumptions, the power needed can be met by each system (if the belt drive is considered the "weakest"). Efficiency in real life will not be 100%, but close, depending on the system. If a safety factor of 10 should be used, on e.g., the torque, a system can still be developed with that range, based on the assumption of a belt drive.

4.6.2 Space limitations

Looking at Figure 4-6, the distance from the bearing mount to the front of the frame equals ~880 mm. The tractor has its mounting device (hitch) hooked up at the front of the frame (down low in Figure 4-6). There are some space limitations for mounting of e.g., an engine in front of the frame since the tractor will be placed there. Space limitations also depend on the type of tractor.

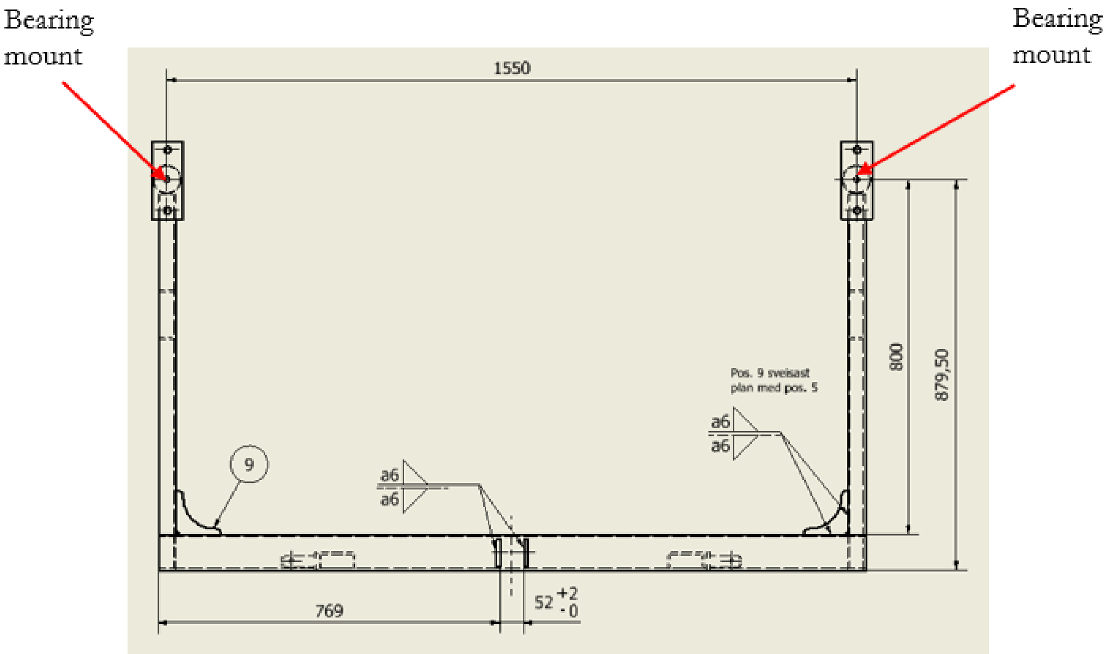


Figure 4-6 Top view of frame with bearing locations

Figure 4-7 shows the width of the drum, from the bearing's inner surfaces. This indicates a limited space between the drum and frame, making it inconvenient to mount things there. The space will most likely be benefited outside the frame, either in width, length, or both. The length of 880 mm gives many possibilities for gear, chain, and belt drive.

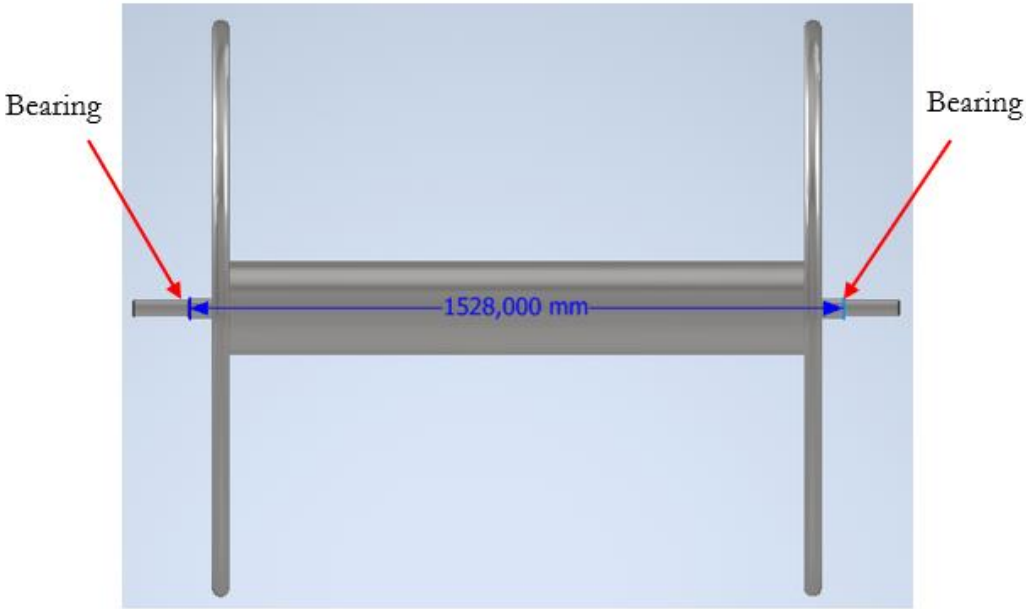


Figure 4-7 Top view of drum with bearing locations

With a direct drive or planetary gear, the engine output shaft would coincide with the drum axis. If a planetary gear were to be used, some alternative mounting setup should be sorted out. A driven pulley or sprocket for a belt or chain drive can be mounted directly on the drum shaft, either in between the bearing and drum or at the outside of the bearing.

4.6.3 Additional

For each system, there should be additional "accessories", like tensioner, idler, overload preventer, and lubricater. Both chain and gear are recommended to be lubricated. Both chain and belt are recommended to have an idler for tensioning (or tensioner on the engine). All of them should have a solution for torque overload, whereas the belt may have the opportunity to slip. A torque overload device can be implemented for both gear and chain.

Maintenance by e.g., changing belts, is an important factor. If the driven pulley is located between the bearing and drum, the drum needs to be disassembled from the frame, before changing the belt. The chain can be split in half, making it easy to change. Gear change depends on the setup, but most likely by lifting it off from the frame.

4.6.4 System conclusion

The design of a new transmission system should be easy to use, maintain, understand and to manufacture and produce. A fancy solution with a crazy setup isn't necessary. After some thought, belt and chain drive are the most suitable ones. Gear is more costly (maybe not for a simple two-axis gear train) and damage can occur on all gears, even if one fails. Having a precise setup and alignment is important, which is far from impossible, but requires more when assembling the drum. It should then be built with no complications of removing and placing the drum (owner should not require special skill for this). In addition, a torque overload device is needed (can be as easy as a keyway on the input shaft).

Chain and belt are very similar. Belt drive can provide slip for overload purposes, which chain can't. The chain can be split for easy replacement while the belt can't. Chain works best with lubrication while belt has no need for lubrication. Working from this, belt seems more suitable than chain.

The greatest advantages with belt drive are the slip condition, no required lubrication, cost-effective and may not wreck itself if there is some misalignment. The disadvantages are high bearing loads and that a low speed is not optimal (especially for V belts). This may also be the tipping point for deciding between the chain and belt, or between different belt types

Further design study will be focused on V belts, finding the most suitable size for the required specifications.

5. Solution and design methodology

Design and calculations are closely connected. Without the correct calculations, the design can be pretty, but not functional for its tasks. Without a design study, calculations have no foundation to work on. Design work and methodology will aim towards linking the general requirements, making sure they meet the superior system need. By doing this, constraints can be set, which also gives a lead on a feasible design. In the end, design and functionality are the most important, seen from the user's perspective. The user doesn't care about the calculations. That's the engineer's concern.

5.1 Considerations

Some boundary conditions need to be set. The boundaries will keep the progress within a specific path, knowing that these must be met with the following solution.

5.1.1 Belt and pulley

The system will strive against the use of a two-axis setup. Pulley number 1 with input from the engine and pulley number 2 as output to the drum, with the belt in between. As mentioned earlier, V belts are suitable for transmitting high torque. Only downside with a pulley system, especially V belts: belt speed should not be lower than 5 m/s. Below this, slip and creep may occur. Even though this is a limit, it does not imply that it's impossible. Why not find a way around it? A thought could be by adding several belts, increasing the wedging action, or by having a higher tension, where all of them can influence the belt wear. Can higher wear based on a shorter life span be accepted? If the runtime is relatively low for each year, designing based on a shorter life span can be an option

5.1.2 Tensioner

After some runtime, the belt will most likely stretch, creating slack in the system. By having correct belt tension, the belt will last longer. If the calculations are based on a specific value for belt tension, then it should be maintained. This can be done either manually, where the user adjusts at some time interval, or by having an automatically adjustable system. By having a spring-based idler, the correct tension can be held over time. If not an idler, adjustments can be done on e.g., the driven pulley, or the hydraulic engine. Required pre-tension will most likely change based on the belt type and system setup.

5.1.3 Safety cover

An open system is not preferable. Adding a safety cover has its advantages, especially two: to prevent dirt from reaching the system (increasing lifespan) and personal injury. When working with moving parts, there must be precautions. Even though a cover is necessary, it may take up a lot of space. The following question should then be asked: What's more important? Take up more space, increase design work, parts, and money, or to prevent injuries on its surroundings? By covering basic EHS, injuries can be prevented. Anyhow, the cover should be robust and lightweight, with an easy detachment function.

5.2 Investigating the existing system

Figure 5-1 shows the chain solution on the existing system. The sprocket is attached to the engine, transmitting rotational power to the chain, having the characteristics of an internal gear [4]. The drum itself is mounted on the frame with bearings.

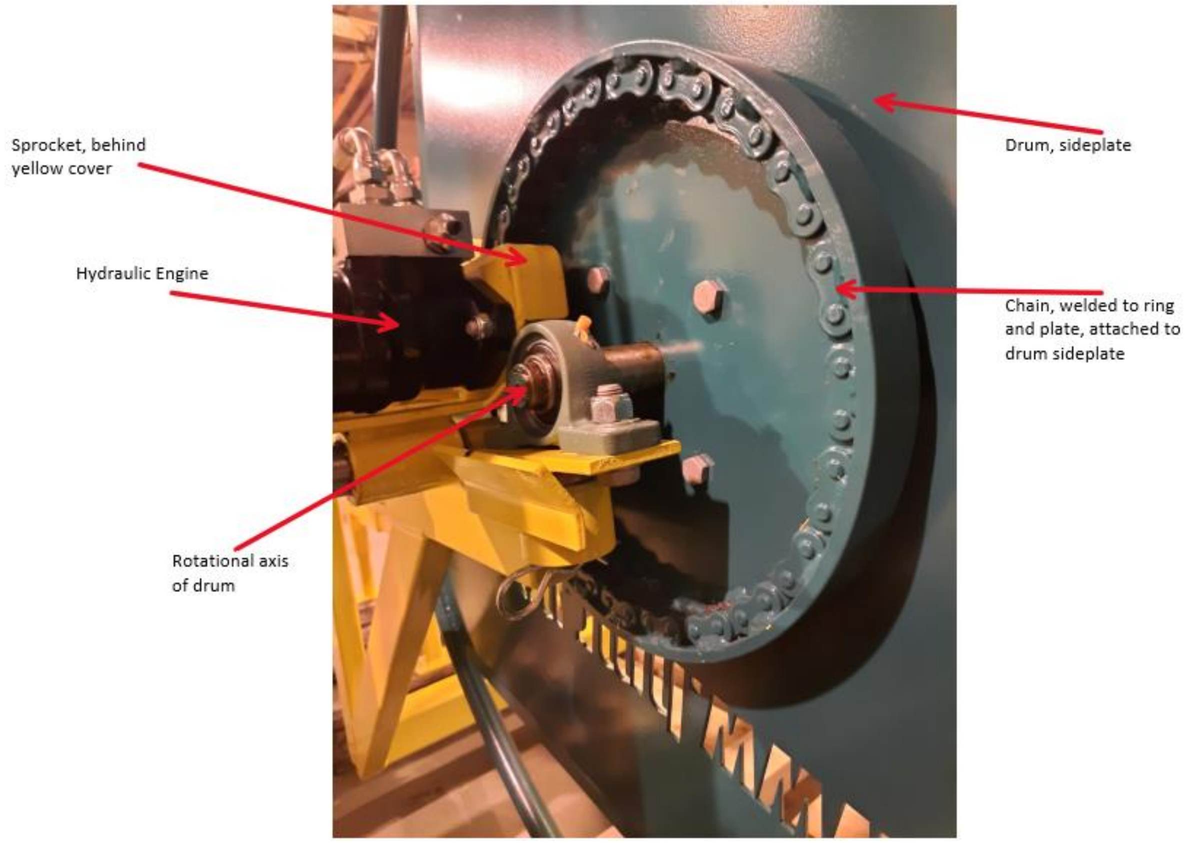


Figure 5-1 Today's power transmission solution

By using the assumptions from the engine output RPMs, it can be investigated at what RPM the drum will turn at, and the torque transmitted. The capability of the new system should be equal to, or greater than the existing one. It's important that the new system has at least the same capability to transmit power.

When assuming a no-slip condition between the sprocket and the chain, tangential velocities at their pitch circle are equal. Figure 5-2 indicates the sprocket as number 1 and the chain ring as number 2.

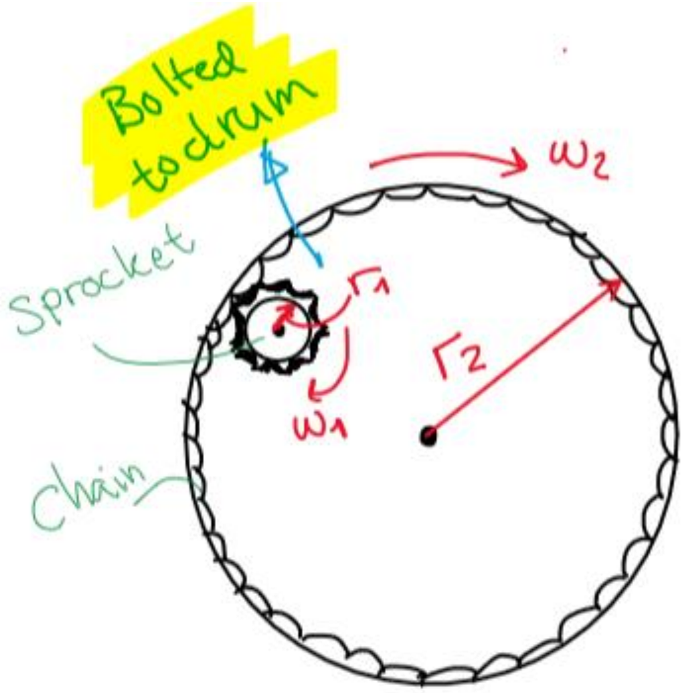


Figure 5-2 Theoretical illustration of the existing power transmission

Velocities can then be written as

$$v_1 = v_2 \tag{5.1}$$

By using the relations from Figure 5-2

$$\omega_1 r_1 = \omega_2 r_2 \tag{5.2}$$

Solving for the angular velocity of the drum

$$\omega_2 = \omega_1 \frac{r_1}{r_2} = \omega_1 \frac{D_1}{D_2} \tag{5.3}$$

As before, RPM will be the working unit, obtaining

$$RPM_2 = RPM_1 \frac{D_1}{D_2} \tag{5.4}$$

Since the sprocket is attached to the engine, they will have the same angular velocity (and torque). Recalling that the engine power was set to 5 kW and 130 RPM occurred at 360 Nm and 180 RPM at 260 Nm, the

Pitch diameter of the sprocket D_1 ~ 0.109m

and

Pitch diameter of the chain ring D_2 ~ 0.309m

Using Equation 5.4, drum RPM ~36 RPM at the lowest, and ~50 RPM at the highest. This is way above the assumption of 5 to 15 RPM. The torque transmitted and the ratio are useful to know since the new design should have an equal or higher capability of transmitting torque. Assuming negligible power loss, transmitted power can be written as

$$p_1 = p_2 \tag{5.5}$$

Using the relations between power, torque, and angular velocity

$$p = M_T \times \omega \tag{5.6}$$

Combining Equation 5.5 and 5.6 into

$$M_2 = M_1 \times \frac{\omega_1}{\omega_2} = M_1 \times \frac{RPM_1}{RPM_2} = M_1 \times 3.61 \tag{5.7}$$

3.61 describes the torque ratio. By Equation 5.7, the 360 Nm (max) torque transmitted by the sprocket (from the engine), gives an output of 1300 Nm. The internal gear setup provides low speed and high torque on the drum. The new system needs to surpass the torque capability, as well as the lowest RPM output from the engine.

5.3 Design solutions for the belt drive system

There are several types of belts to use for the transmission system which may lead to different types of design. This Chapter will go through three design proposals. Figure 5-3 shows simply how the existing design looks like today. The chain ring is not modelled in Figure 5-3 and is supposed to be bolted onto the drum (four threaded holes). It is a compact build, with easy access.

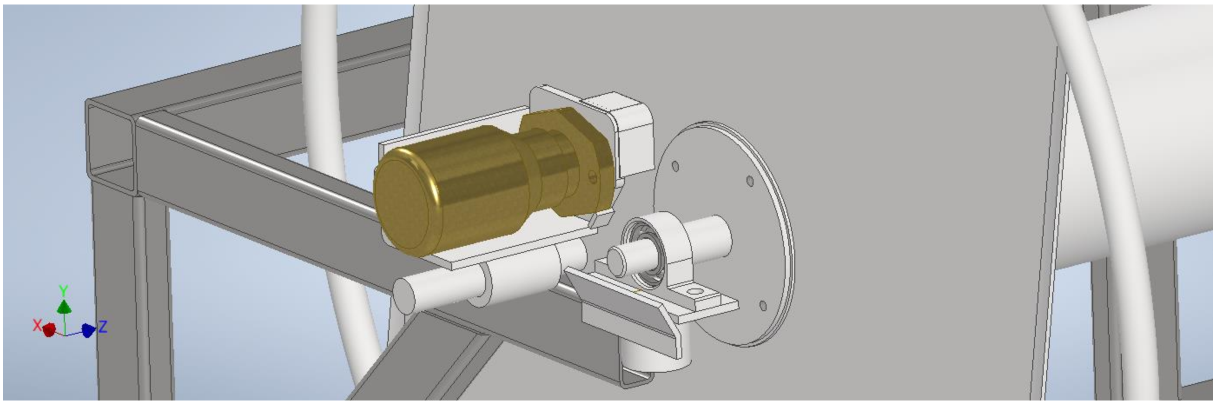


Figure 5-3 Illustration of the existing setup, without chain ring

5.3.1 Design proposal 1

Figure 5-4 shows the engine, bolted onto a triangular plate located between the beam members. The triangular plate acts both as a engine bracket and frame reinforcement. Anyhow, the placement of the engine is unlucky. Imagine the process of attaching the drum onto the frame. Tractor starts backing up, with its frame mounted on the rear hitch. With a bit of unluck, the drum might hit the engine. Even though hydraulic engines are robust, it's not convenient. This might not be the most suitable design.

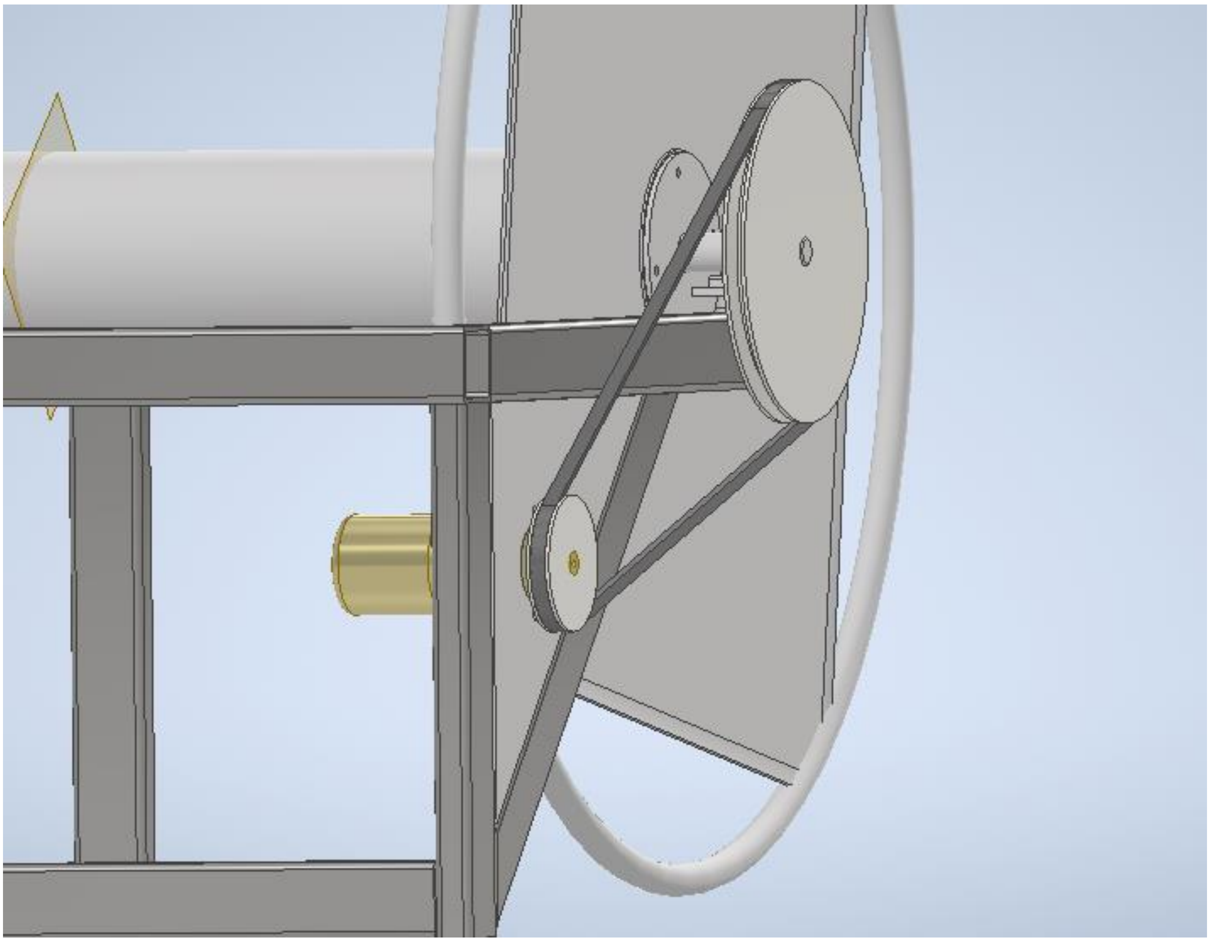


Figure 5-4 Design proposal 1, engine mounted internally on the frame

5.3.2 Design proposal 2

Figure 5-5 is inspired by the existing solution. The engine is mounted on the outside of the frame, increasing the width of the frame, but remains compact. A tensioner has been mentioned, and this is one way to solve it. The engine is mounted on a bracket, which slides back and forth on a base plate. On the base plate, there is an adjusting bolt. When tightening the bolt, the baseplate stands still while the engine gets pushed away, increasing the belt tension. Top view is shown in Figure 5-6.

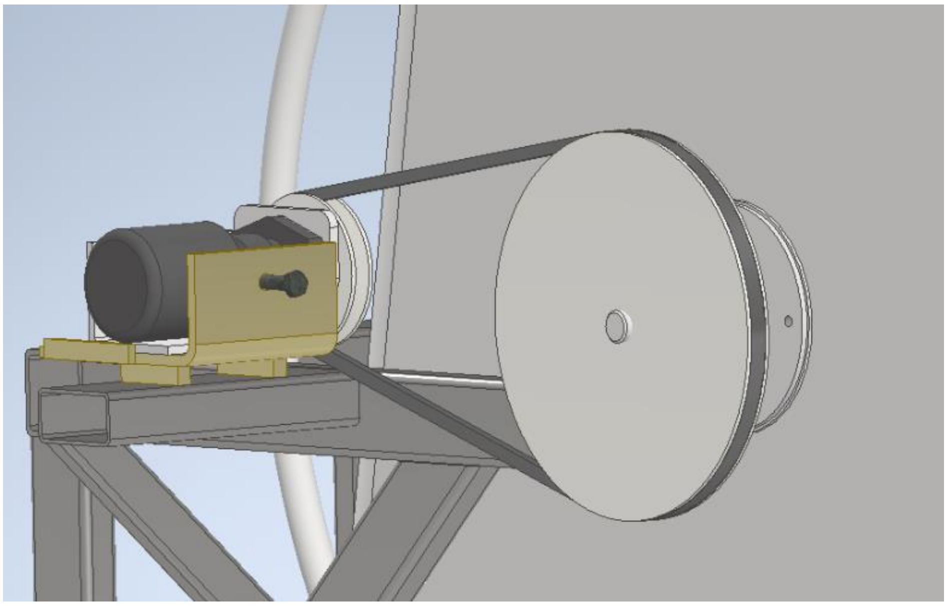


Figure 5-5 Design proposal 2, engine mounted close to the drum, outside of the frame

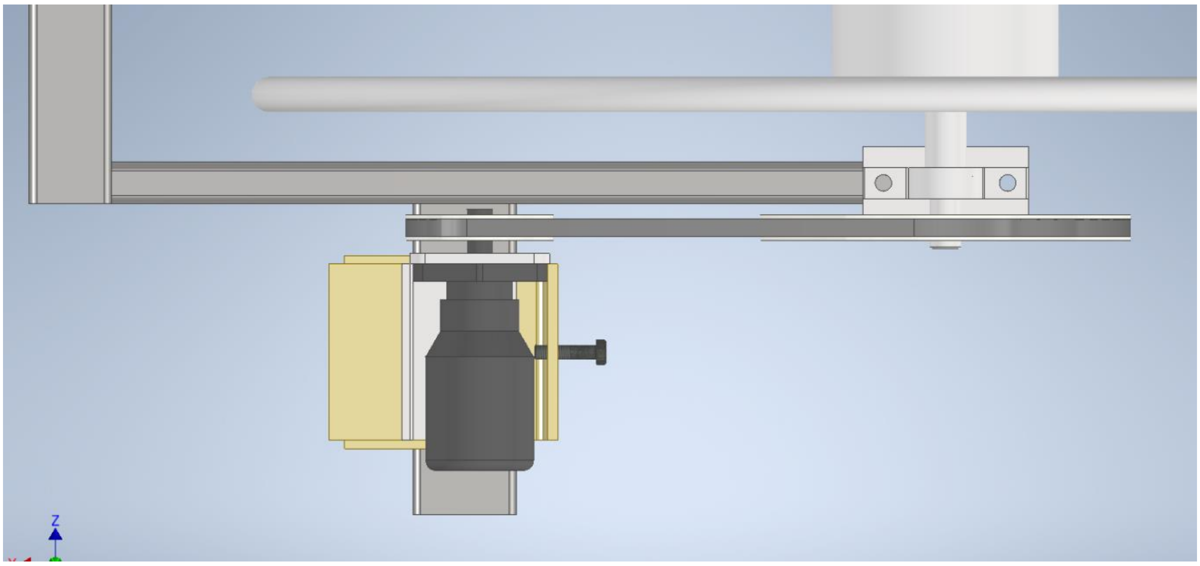


Figure 5-6 Design proposal 2, top view

5.3.3 Design proposal 3

As shown in Figure 5-7, the engine is mounted on the top front of the frame. The transmission system is not compact anymore, and the C-C (centre to centre) distance between the pulleys is greater. V belts are commonly used for shorter C-C distances. Whether the C-C distance is too long, needs to be investigated. If so, another belt type can be used.

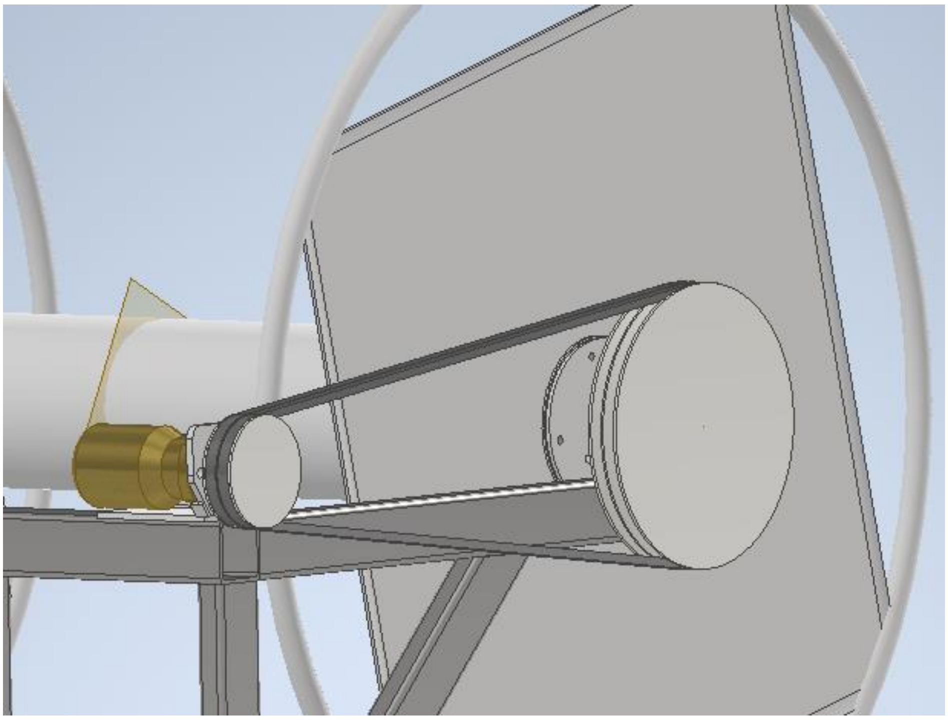


Figure 5-7 Design proposal 3, engine mounted on top of frame, longest distance

5.4 Preferrable design solution

All the designs have its driven pulley placed on the outside of the bearing. Placing the pulley on the inside would make it more compact whereas belt change would then be more stressful. If the belt should be replaced (with a pulley on the inside), the drum needs to be detached from the frame. Placing the pulley on the outside will eliminate the drum detachment.

Design number 1 has its biggest disadvantage, engine location. The user should not be worried about hitting the engine when attaching the drum to the frame. Design number 1 won't hold.

Both design number 2 and number 3 are feasible and have both advantages and disadvantages. Design number 2 provides a compact and short C-C distance. It's also easier to make changes to this parameter since there is more space to work on. In addition, extra members for the engine bracket is necessary. Simultaneously, the driven pulley should not get in contact with these members. This can be avoided by placing the engine at a higher point.

Design number 3 is mounted on the top front of the frame. C-C length increases, and it's not that compact anymore. Tensioning can be done with e.g., a baseplate, as introduced in design number 2. There is also a possibility to attach an idler on the side.

HMR has mentioned design number 3 as the most convenient design. Design number 3 allows HMR to further develop a frame using a (different) engine with two output shafts, extended to both sides. This allows mounting an extra pulley system on the other side, with the same dimensions. Since this is for the futuristic development plan, it will not be covered in this thesis.

5.5 Design study of V belts

Since the V belt was decided on, it should be investigated if it is feasible for this project. Some limitations about the belt speed will be mentioned before the calculations and selection of a proper V belt will be made.

5.5.1 Biggest limitation

The biggest limitation is the belt speed. As mentioned before, the centrifugal force limits the use of V belts with speeds lower than 5 m/s [24]. Several books mention this, but without reasoning. From personal experience, a V belt with low speed has been observed. Why is it so difficult to find literature on this?

A theory can be the energy loss when forcing the belt out of the pulley since it relies strongly on friction. The belt wraps around the sheave and wedges itself down the groove. The centrifugal force forces the belt out of the groove, lowering the contact between the sheave and belt. At optimal belt speeds, typically 20-25 m/s, the belt has enough friction to transmit motion and power, and little enough to exit the groove with low energy loss. At high belt speeds, the centrifugal force forces the belt outwards, reducing the necessary friction needed to transmit the power and motion, resulting in slip. For the manure drum, the belt speed is below 5 m/s, and the centrifugal belt force is low, implying that the belt requires more energy to exit (due to friction).

Question is: is there too much energy loss, unwedging the belt from the groove, making the efficiency of the system even lower? Are there any other belt types that does not rely on friction, which satisfy the design needs?

Further calculations will be done, based on the design of V belts. It's hard to determine if V belt is functional in real life, or if the formulas used has large deviations at low belt speeds.

5.6 Pulley diameters

If the ratio of the old system (3.61) should be obtained, the same ratio between the driven and driving pulley should be equal. Setting the driven pulley with pitch diameter D_P = 350 mm, the dimension of the driving pulley can be obtained by

$$d_p = \frac{D_p}{3.16} \tag{5.8}$$

Equation 5.8 results in a pitch diameter of 96.95 mm. A diameter of 95 mm obtains the ratio. Even though the ratio is obtained, it does not mean it will transfer it. Power loss occurs between the two pulleys. Some thought solutions would be to add a second pulley. The second pulley won't increase the power (since it's limited by the engine) but helps if the single belt slips. Adding a second pulley means adding a second belt, which also needs energy to unwedge from the groove. A wider angle on the groove could be considered, which may decrease the wedging friction. Further on, the torque ratio may be adjusted if allowable.

5.7 Belt speed

As mentioned before, the belt speed should be as high as possible since the lower limit of 5 m/s is not yet reached. With 5 kW output at 130 RPM, the belt velocity can be defined as

$$V_{belt} = \frac{\pi}{30} \times RPM \times \frac{d_p}{2} \tag{5.9}$$

With 130 RPM and diameter of 95 mm, a belt speed of 0.656 m/s is obtained. With a diameter of e.g., 135 mm, belt speed equals 0.918 m/s. In reality, the RPM of the drum is assumed to be 15 at max. With a ratio of 3.16, the RPM of the engine will then be ~47. This is not even close to 130 RPM. The belt speed will be low either way.

5.8 V Belt selection

V belt selection process has been taken from *"Shigley's Mechanical Engineering Design"* [4], and is based on the cross section of a V belt, shown in Figure 5-8.

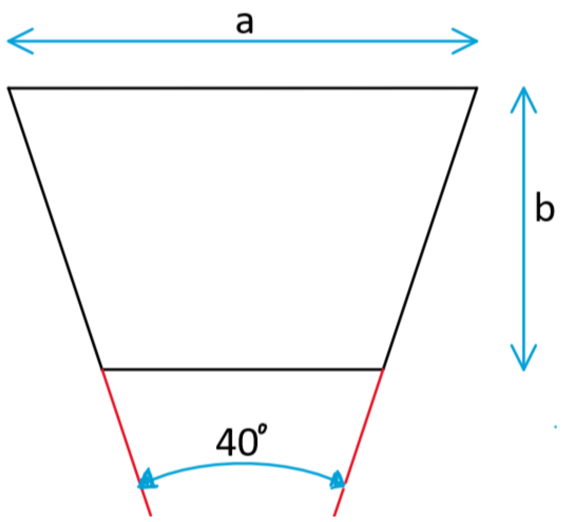


Figure 5-8 Cross-section of a V belt, according to [4]

Belt type is divided into several sections depending on the kW range for one belt. Section A from [4] uses the range of 0.2 to 7.5 kW, which is suitable for the 225 W at 5 RPM on the drum, obtained earlier. Section A belt specifications are presented in Table 5-1.

Table 5-1 Specifications for belt type A [4]

Belt Section	**Width a** [mm]	**Thickness b** [mm]	**Min. pulley OD** [mm]	**kW range**
A	12	8.5	75	0.2-7.5

5.8.1 Belt length and C-C distance

Belt length can be determined from C-C distance and pulley diameters. Pitch length of belt is defined by

$$L_p = 2(C - C) + \frac{\pi(D_p + d_p)}{2} + \frac{(D_p - d_p)^2}{4C} \tag{5.10}$$

Where D_P is the big pulleys pitch diameter and d_P is the pitch diameter of the small pulley. The inside circumference of the belt can be picked out from Table 5-2, which equals to L. Since the pitch length and inside circumference is unequal, a length conversion will be made, obtaining the inside circumference

$$L = L_p - L_{correction} \tag{5.11}$$

$L_{CORRECTION}$ gave 32 mm for section A belts [4]. The different inside circumferences are listed in Table 5-2, which will be used when ordering a belt.

Table 5-2 Inside circumference and added length for section A belts [4]

Section	Inside circumference [mm]	$L_{CORRECTION}$ [mm]
A	650, 775,825, 875, 950, 1050, 1150, 1200, 1275, 1325, 1375, 1425, 1500, 1550, 1600, 1650, 1700, 1775, 1875, 1950, 2000, 2125, 2250, 2400, 2625, 2800, 3000, 3200	32

If the belt length changed when subtracting $L_{CORRECTION}$, finding the new C-C distance would be necessary. Equation 5.10 finds the new C-C based on the new pitch length.

$$New\ C - C = 0.25 \times \left\{ \left[L_p - \frac{\pi(D_p + d_p)}{2} \right] + \sqrt{\left[L_p - \frac{\pi(D_p + d_p)}{2} \right]^2 - 2(D_p - d_p)^2} \right\} \tag{5.10}$$

Table 5-3 shows the belt specifications for design number 3.

Table 5-3 Specifications with belt length of 2447 mm and corrected 2400 mm

	Belt section	**C-C** [mm]	**Belt length, circumference** [mm]	**From Table 5-2** [mm]	**New C-C** [mm]
D = 350 mm **d = 95 mm**	A	881	2447	2400	~857

The C-C distance should not be greater than 3 times the sum of the sheave diameters[4], 1335 mm, which is approved.

5.9 V Belt power ratings

Following calculations is aided based on belt section type, correction factors on the transmittable power per belt, and the power to be aimed for. This will give the amount of belts needed to transmit the power [4].

Power ratings of standard A section V belts are shown in Table 5-4. It's observed that the belt speed stops at 5 m/s. Power ratings is based on the smallest sheave diameter, 95 mm in this case. Looking at the 95 mm row in Table 5-4, the highest power capacity for a single belt occurs at 20 m/s. At 25 m/s it drops, making 20 m/s look like the optimum belt speed.

Table 5-4 Belt power ratings for several pulley pitch diameters, based on section A belt [4]

Belt section	**Pulley pitch dia.** [mm]	**Belt speed** [m/s]				
		5	**10**	**15**	**20**	**25**
A	85	0.60	0.98	1.17	1.64	0.84
	95	0.69	1.16	1.43	1.49	1.28
	105	0.77	1.30	1.64	1.78	1.63
	115	0.83	1.41	1.82	2.01	1.93

Using dCodes "Function Equation Finder" [25], a function based on the 95 mm pulley can be approximated, assumed concave parabolic. Datapoints to be evaluated is listed in Equation 5.11.

$$\begin{Bmatrix} Belt\ speed\ \frac{m}{s} \\ Power\ ratings\ kW \end{Bmatrix} = \begin{Bmatrix} X \\ Y \end{Bmatrix} = \begin{bmatrix} 0 & 5 & 10 & 15 & 20 & 25 \\ 0 & 0.69 & 1.16 & 1.43 & 1.49 & 1.28 \end{bmatrix} \tag{5.11}$$

Since the power is a function of the circumferential force of the belt and the belt speed, it can be written as

$$p = F_c \times v \tag{5.12}$$

An approximated power function from based on datapoints in Equation 5.11 can be written as

$$p(v) \approx -0.00438571x^2 + 0.161471x - 0.005 \tag{5.13}$$

Equation 5.13 is then plotted with Matlab [26], Figure 5-9.

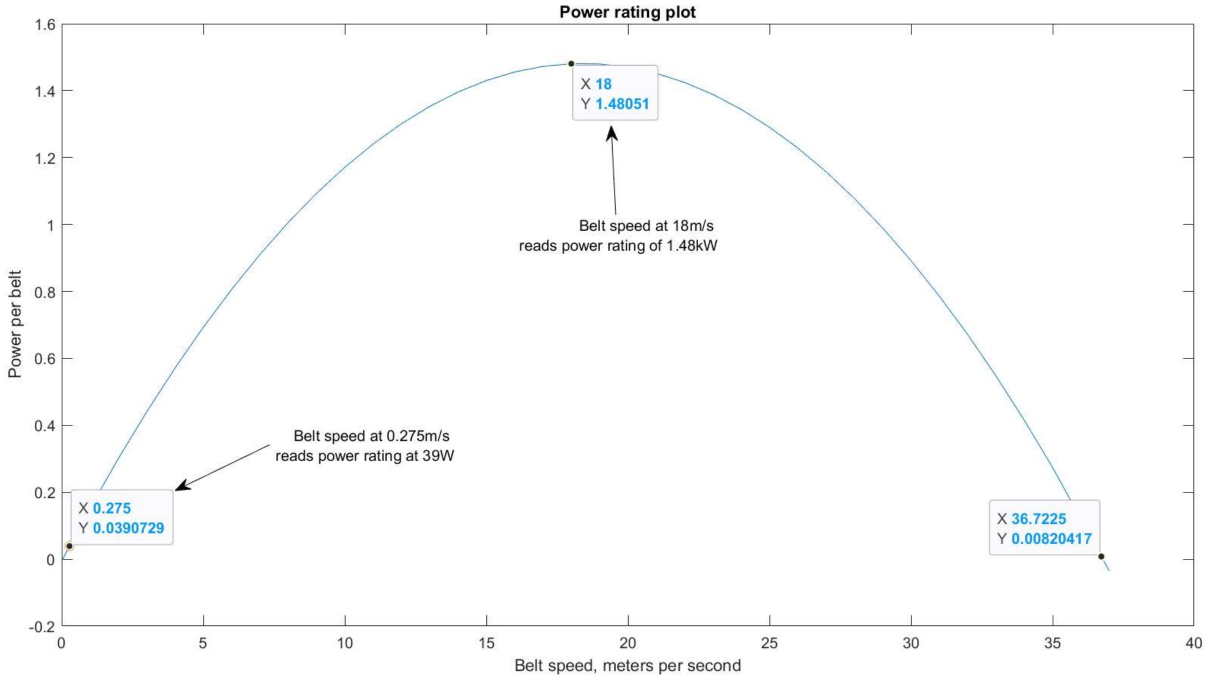


Figure 5-9 Plot of power rating per belt at different belt speeds

The plot shows the approximated parabola with its optimum belt speed at 18 m/s and power rating of 1.48 kW. Comparing it to Table 5-4, the values have a small deviation, about +/- 30 W. Max drum RPM was assumed to be 15 RPM, which converts to a belts speed of 0.275 m/s. Looking at the belt speed at 0.275 m/s, the transmittable power per belt is 39 W (which will be the tabulated power). Since the required power for 15 RPM is 675 W, it is not feasible. Number of belts can be decided by Equation 5.14.

$$N_b \geq \frac{H_d}{H_a} \tag{5.14}$$

In Equation 5.15, N_b, H_d, and H_a is number of belts, design power and allowable power per belt, respectively. Allowable power per belt can be obtained from the tabulated power ratings (from the approximated power function). As for a belt setup with equal pulley diameters (the angle of contact, θ, equals 180 for both pulleys), a moderate length and a steady load, the tabulated power ratings can be used directly. If this is not the case, the allowable power is a corrected version of the tabulated power ratings.

$$H_a = K_1 \times K_2 \times H_{tab} \tag{5.15}$$

K_1 – angle of contact, θ, correction factor

K_2 – belt length correction factor

H_{tab} – tabulated power rating

K_1 can be found from Table 5-5 based on different parameters. Since the V belt has both pulleys designed with grooves, correction factor is listed under section VV.

Table 5-5 K1 correction factor based on different parameters [4]

$\frac{\boldsymbol{D-d}}{\boldsymbol{C}}$	**θ, degrees**	**K1 correction factor VV**
0.00	180	1.00
0.10	174.3	0.99
0.20	166.3	0.97
0.30	162.5	0.96
0.40	156.9	0.94
0.50	151.0	0.93
0.60	145.1	0.91
0.70	139.0	0.89
0.80	132.8	0.87

To define the correction factor, some parameters need to be defined. First, (D-d)/C

$$\frac{D_p - d_p}{C} = \frac{350 - 95}{451} = 0.565 \tag{5.16}$$

And angle of contact, θ

$$\theta = 180 - 2\alpha \tag{5.17}$$

Where α is shown in Figure 5-10, and obtained by

$$\sin(\alpha) = \frac{D_{p-}d_p}{2 \times C} \tag{5.18}$$

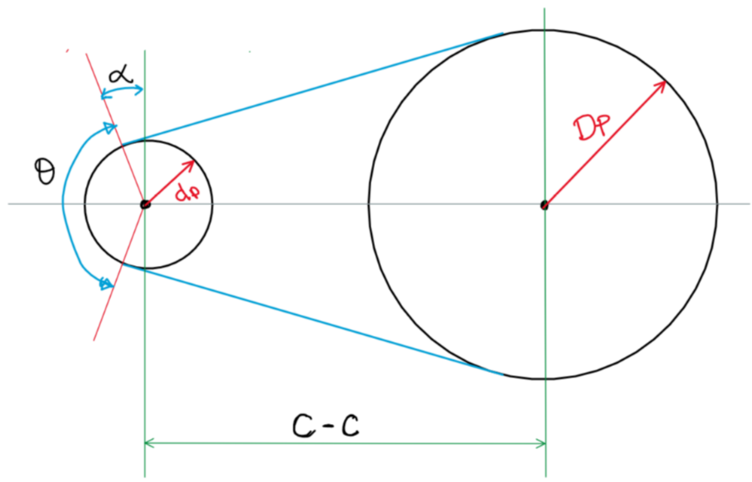


Figure 5-10 Belt and pulley parameters

Following Equation 5.16, 5.17, 5.18, Table 5-5 reads (with interpolation) K_1 ~ 0.92.

For A-section belt at 1600 mm in nominal belt length, K_2 equals 1.00 [4]. From Equation 5.15, the allowable power per belt gives H_a = 35.88 W. Equation 5.20 defines the number of belts to transmit the required (nominal) power. First, the nominal power H_{nom} will be corrected.

Equation 5.19 corrects H_{nom} into the following design power, H_d. Nominal power is the power to be delivered (i.e., the resistive power the manure hose generates), service factor K_s and design factor n_d. Assuming light shock on the driven machinery and a normal torque characteristic from the source of power, the service factor K_s is set to 1.2. Design factor is typically assumed to be 1.0 to 1.2. For this system, it can be set to 1.1 [4].

$$H_d = H_{nom} \times K_s \times n_d \tag{5.19}$$

From the assumption of 15 RPM on the drum and the calculations done on the required power (nominal power, 675 W), Equation 5.19 sets H_d to be 891 W. Recommended number of belts to transmit the required power can be found by Equation 5.20.

$$N_b \geq \frac{891\ W}{35.88\ W} = 24.83\ belts \tag{5.20}$$

5.10 Conclusion on V belts

~25 belts are required, which is not the best option. This shows that the belt speed is an important factor. A similar calculation with a belt speed of 5 m/s (and the same parameters) has shown that two belts are enough to transmit and meet 675 W on the drum. This concludes V belt as unfeasible for this design. This also yields for design number 3, since the only difference is the factors K_1 and K_2, which won't provide a suitable number of belts.

Since increasing the belt speed, the centrifugal forces will separate the belt from the sheave, decreasing the frictional contact force. As the belt speed increases past the optimum speed (in this case ~20 m/s), the transmitted power will decrease. As the speed decrease, the transmissible circumferential force increases (since the centrifugal force attribution decreases). If the belt speed is then reduced to such an extent that there is hardly any movement in the belt, the increased transmissible circumferential force won't help [27]. That's where the optimum belt speed comes into picture. A real-life test is a way to debunk or confirm the belt speed "problem".

Since the theoretical calculations didn't turn out very well, synchronous belts will be investigated and considered for the next step in the design work.

5.11 Design study of synchronous belts

HMR provided contact info to Rubix Group [28], selling industrial products, such as gears, bearings, tools etc. Rubix shared their design manual (found in Appendix C) [6] for different types of synchronous belt setups, in this case, Gates [29] belt drives. The following belt type, *Power Grip HTD* (High Torque Drive), are ideal for high power transmissions in low speed and high torque applications, where durability and low maintenance is required (Appendix C) [30].

5.12 Design manual for synchronous belts

The design manual uses 5 steps for the design of a synchronous belt drive. Before step 1, following requirements need to be determined:

- Power requirement and type of driveN machine
  a. **225 W for 5 RPM on drum**
  b. **675 W for 15 RPM on drum**
- The RPM of the driveR machine **(with ratio of 3.6)**
  a. **18 to 54 RPM**
- The RPM of the driveN machine
  a. **5 to 15 RPM**
- The approximate C-C distances for the drive
  a. **~ 881 mm**
- Hours per day operation
  a. **50 hours pr year, 10 years, stated by HMR**

5.12.1 Step 1 – Determine the service factor(s)

Manual uses two service factors. One for "Speed-up ratio range" and "Service factor". Speed-up ratio would be if the drum where to rotate faster than the engine. This is not the case. Table 5-6 shows a portion of the service factor chart from the manual.

Table 5-6 Service factors for timing belts, provided by Rubix [6]

SERVICE FACTOR CHART						
DRIVE N MACHINE	DRIVE R					
The driveN machines listed below are representative samples only. Select a driveN machine whose load characteristics most closely approximate those of the machine being considered.	AC motors: normal torque, squirrel cage, synchronous, split phase, inverter controlled. DC motors: shunt wound, stepper motors. Engines: multiple cylinder internal combustion.			AC motors: high torque, high slip, repulsion induction, single phase, series wound, slip ring. DC motors: series wound, compound wound, servo motors. Engines: single cylinder internal combustion. Line shafts. Clutches.		
	Intermittent service	Normal service	Continuous service	Intermittent service	Normal service	Continuous service
	3-8 hours daily or seasonal	8-16 hours daily	16-24 hours daily	3-8 hours daily or seasonal	8-16 hours daily	16-24 hours daily
Display equipment. Dispensing equipment. Instrumentation. Measuring equipment. Medical equipment. Office equipment. Projection equipment.	1.0	1.2	1.4	1.2	1.4	1.6
Appliances. Sweepers. Sewing machines. Screens: oven, drum, conical. Woodworking equipment (light): band saws, drills, lathes.	1.1	1.3	1.5	1.3	1.5	1.7
Agitators for liquids. Conveyors: belt, light package. Drill presses. Lathes. Saws. Laundry machinery. Woodworking equipment (heavy): circular saws, jointers, planers.	1.2	1.4	1.6	1.6	1.8	2.0

HMR suggested a service factor of S_f = 1.1-1.2, based on the low run time and acting loads per year. S_f = 1.2 will be used.

5.12.2 Step 2 – Calculate the design power

Design power will be obtained from the power requirement and service factor and can be defined as

$$p_{design} = p_{required} \times S_f \tag{5.21}$$

Design power is represented in Table 5-7 at different drum RPMs.

Table 5-7 Design power at different drum RPM

Drum RPM →	**5 RPM and 225 W**	**15 RPM and 675 W**
Design Power	270 W	810 W

5.12.3 Step 3 – Belt pitch

Using the belt pitch selection guides in Figure 5-11, correct belt type within PowerGrip HTD can be chosen. "Faster shaft" represents the engine shaft.

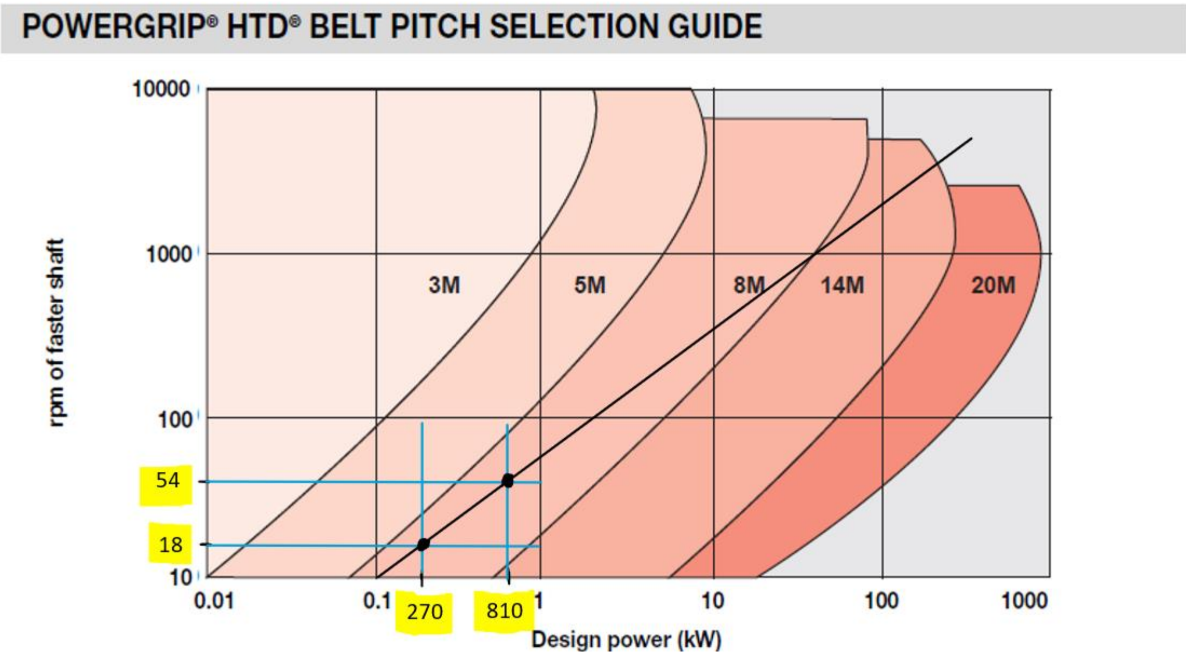


Figure 5-11 PowerGrip HTD belt pitch selection guide [6], modified

Reading of the design powers and RPMs, the points will be within 8M (8 mm belt pitch). Since the powers and RPMs are proportional to each other, they will act linearly on the selection guide, presented by the **black line** in Figure 5-11. 8M belt will be used for the next steps.

5.12.4 Step 4 – Select the pulley combination, belt length and centre distance

First off, speed ratio is obtained by dividing the faster shafts RPM on the slower shafts RPM.

$$r_v = \frac{RPM_{faster}}{RPM_{slower}} \tag{5.22}$$

Current setup gives a 3.61 speed ratio. Further on, a centre distance table will aid to obtain the correct pulleys (with number of grooves), belt length and C-C distance. By using Table 5-8, the tabulated C-C distance closest to the desired distance can be found.

Table 5-8 Centre distances tables 8M and 8MGT [6], modified

CENTRE DISTANCE TABLES 8M & 8MGT

Theoretical centre distance in mm									Number of grooves		Speed ratio	
Belt length code designation in mm												
1200	1280	1440	1600	1760	1800	2000	2400	2800	DriveN	DriveR		
		324.6	410.4	494.1	514.9	617.8	821.3	1023.4	144	40	3.60	1
389.0	429.6	510.7	591.4	671.9	692.1	792.6	993.3	1193.7	80	22	3.64	2
297.5	339.8	423.1	505.2	586.7	607.0	708.3	910.0	1111.1	112	30	3.73	3
362.2	403.2	484.7	565.7	646.5	666.7	767.4	968.4	1169.0	90	24	3.75	
		327.8	413.8	497.6	518.3	621.3	824.9	1027.1	144	38	3.79	
300.8	343.2	426.5	508.7	590.3	610.6	712.0	913.7	1114.9	112	28	4.00	

Table 5-8 shows the speed ratios 3.60, 3.64 and 3.73 as number 1, 2 and 3, respectively. Yellow markings represent the closest C-C distances. As the pulley grooves increases, the pulley diameter also increases, which leads to increased weight. Having smaller pulleys with lower weight would be preferable. This needs to be discussed further on, depending on the influence of the system.

5.12.5 Step 5 – Select the belt width

8M power ratings (green marker) with different size of pulleys are listed in Table 5-9. They are nowhere near the design power required, so an increase in belt width should be considered. The 40 grooved pulley with 3.60 speed ratio looks like the most suitable one.

Table 5-9 8M Power ratings in kW [6], modified

8M POWER RATINGS - KILOWATTS

rpm of faster shaft	Number of grooves in small pulley															
	22	24	26	28	30	32	34	36	38	40	44	48	56	64	72	80
	Pulley pitch diameter in mm															
	56.02	61.12	66.21	71.30	76.39	81.49	86.58	91.67	96.77	101.86	112.05	122.23	142.60	162.97	183.35	203.72
10	0.02	0.02	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.06	0.06	0.07	0.08	0.09	0.10	0.11
20	0.03	0.04	0.04	0.05	0.06	0.07	0.08	0.09	0.11	0.11	0.12	0.14	0.16	0.18	0.20	0.23
50	0.08	0.09	0.11	0.13	0.16	0.18	0.21	0.23	0.27	0.28	0.31	0.34	0.40	0.45	0.51	0.56
100	0.16	0.19	0.22	0.27	0.31	0.36	0.41	0.47	0.54	0.56	0.62	0.68	0.79	0.90	1.02	1.13
200	0.33	0.37	0.45	0.53	0.62	0.72	0.82	0.93	1.05	1.13	1.24	1.34	1.54	1.73	1.93	2.12
300	0.49	0.53	0.65	0.77	0.90	1.04	1.19	1.34	1.51	1.64	1.78	1.93	2.21	2.50	2.77	3.05
400	0.65	0.71	0.84	0.99	1.16	1.34	1.54	1.74	1.96	2.12	2.31	2.50	2.87	3.23	3.59	3.94

These power ratings have a "belt width factor" K_b of 1.0, which is the same as 20 mm belt width [6]. By increasing the belt width, the factor increases. The corrected power rating can be found as

$$p_{corrected} = K_b \times p_{tabulated} \tag{5.23}$$

For a speed ratio of ~3.60, number of grooves for the driveR pulley can be obtained from Table 5-8, as 40 grooves. In Table 5-9, 280 W is found at 50 RPM. The correct power rating from the column of "40 Number of grooves(…)" is then found by interpolation

$$\frac{100-50}{54-50} = \frac{0.56-0.28}{X-0.28} \tag{5.24}$$

Solving Equation 5.24, X equals 302.5 W. For a belt width of 50 mm and factor K_b of 2.74, $p_{corrected}$ equals 828.6 W. If Equation 5.25 is true, the belt is theoretically capable.

$$p_{corrected} > p_{design} \tag{5.25}$$

It's OK. In addition, a belt width of 85 mm and belt factor of 4.76, the power rating would be 1440 watts per belt(!). Main concern will be the increased weight and size as the belt width increases. If 22 or 30 grooves were to be used, it would not make the required power ratings, even with a belt width of 85 mm.

5.13 14M belt review

8M barely met the requirements. Further investigation of a stronger belt will be carried through. Since the hose in this project isn't the longest and heaviest, it would be an advantage if it could manage some of the larger hoses. First off, the initial values are the foundation. Afterwards, a comparison between 8M and 14M decides whom is the most suitable. Iteration will be carried out from step 4, selecting pulley combination, belt length and C-C.

From Table 5-10, yellow marker represents the two closest C-C, 884.1 mm and 901.0 mm. The advantage of C-C of 884.1 mm is a shorter belt (cost wise), higher torque ratio, and less number of grooves. More grooves give greater diameter, leading to increased weight.

Table 5-10 Centre distances for 14M and 14MGT, modified

CENTRE DISTANCE TABLES 14M & 14MGT

Theoretical centre distance in mm								Number of grooves		Speed ratio	
Belt length code designation in mm											
2450	2590	2800	3150	3500	3850	4326	4578	DriveN	DriveR		
518.3	594.8	706.6	888.9	1068.7	1247.0	1488.3	1615.6	144	44	3.27	
692.1	764.1	871.6	1049.6	1226.7	1403.2	1642.8	1769.5	112	34	3.29	
			633.0	825.7	1011.2	1258.3	1387.8	192	56	3.43	
698.1	770.3	877.8	1055.9	1233.1	1409.7	1649.4	1776.0	112	32	3.50	
529.4	606.1	718.3	901.0	1081.1	1259.6	1501.1	1628.5	144	40	3.60	1
704.2	776.4	884.1	1062.2	1239.5	1416.2	1655.9	1782.6	112	30	3.73	2

From Table 5-11 (same principle as for 8M), 30 groove pulley has the power ratings marked in yellow. By interpolating, the power rating can be obtained. For 15 RPM on the drum, interpolation gives

$$\frac{60-40}{55.95-40}=\frac{1.3-0.8}{x-0.8} \tag{5.26}$$

Which reads to X ~1199 W. This is with a belt width factor of 1.0, referring to 40 mm belt width. With a belt width of 55 mm and belt factor of 1.50, Equation 5.26 gives X~1799 W.

Table 5-11 14M Power Ratings in kilowatts, modified

14M POWER RATINGS - KILOWATTS

rpm of faster shaft	Number of grooves in small pulley																
	28	29	30	32	34	36	38	40	44	48	52	56	60	64	68	72	80
	Pulley pitch diameter in mm																
	124.78	129.23	133.69	142.60	151.52	160.43	169.34	178.25	196.08	213.90	231.73	249.55	267.38	285.21	303.03	320.86	356.51
10	0.2	0.2	0.2	0.2	0.3	0.3	0.3	0.4	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.7	0.8
20	0.4	0.4	0.4	0.5	0.6	0.6	0.7	0.7	0.8	0.9	1.0	1.1	1.1	1.2	1.3	1.4	1.5
40	0.7	0.8	0.8	1.0	1.1	1.2	1.4	1.4	1.6	1.8	1.9	2.1	2.3	2.4	2.6	2.7	3.0
60	1.1	1.2	1.3	1.5	1.7	1.9	2.0	2.2	2.4	2.7	2.9	3.2	3.4	3.6	3.8	4.1	4.5

By comparing the 8M and 14M belts, not only power ratings, but weights and diameters, obtained from *"Tyma.eu"* [31] , a clearer picture is made. Table 5-12 shows a comparison between 8M and 14M, with different number of grooves, which lists up the pitch diameter and weights.

Table 5-12 Comparison of belt and pulley specifications

Specifications	8M	8M	14M	14M	14M	14M
Belt width [mm]	50	85	40	55	40	55
5 RPM on drum	18 on engine	18 on engine	18.65 on engine	18.65 on engine	18 on engine	18 on engine
15 RPM on drum	54 on engine	54 on engine	56 on engine	56 on engine	54 on engine	54 on engine
Power rating (corrected) [W]	828.6	1140	1199	1799	Not calculated	Not calculated
Grooves smallest	40	40	30	30	40	40
Pitch dia. Smallest [mm]	101.86	101.86	133.69	133.69	178.25	178.25
Weight smallest [kg]	3.5	5.2	5.45	6.6	10.6	11.2
Grooves largest	144	144	112	112	144	144
Pitch diameter largest [mm]	366.69	366.69	499.11	499.11	641.71	641.71
Weight largest [mm]	13.8	21.5	26.7	29.5	35	39
Speed ratio	3.60	3.60	3.73	3.73	3.60	3.60

It's worth noticing the difference in pitch diameter and weight for 8M and 14M, with the same speed ratio (3.60). 14M transmit enough power per belt but has a significantly high pitch diameter and weight on the largest pulleys. 8M also transmit enough power, but with almost half the weight, and a decrease in pitch diameter. The pitch diameter isn't far off compared to the existing (chain) pitch diameter. On the other hand, power is more limited, and may not be feasible for a longer and heavier manure hose (if this system should be implemented further on). It's hard to say if it's doable with 8M for heavier hoses, since the calculations are based on assumptions. E.g., the actual coefficient of friction in real life may be 0.2, and not 0.45, which is assumed in the previous calculations.

5.14 Change in speed ratio

The ratio between the engine and drum are currently set as 3.61. Higher ratio gives a higher torque output and decreased RPM output. This is optimal since high torque and low speed is preferable for this system. Downside is the size of the drum pulleys. Greater ratio, larger pulleys, leading to increased weight. More material also impacts the cost. If weight is to be decreased, the ratio can be decreased. In addition, the "torque lost" must be compensated for, due to decreased ratio.

One crucial consequence will be lower engine speed, avoiding going **below** 10 RPM. If the rotations on the engine and the torque required from the engine is met, it should theoretically be fine. Table 5-13 shows different speed ratios with engine torque and RPMs.

Table 5-13 Overview of torque, RPM and engine power, based on 5 and 15 RPM on drum

5 RPM on drum	2	2.5	2.8	3	3.3	3.61	3.73
Torque [Nm]	215	172	154	143	130	120	116
RPM	10	12.5	14	15	16.5	~18	~18.7
Power [W]	225	-	-	-	-	-	-
15 RPM on drum							
Torque [Nm]	215	172	154	143	130	120	116
RPM	30	37.5	42	45	49.5	~54	~56
Power [W]	675	-	-	-	-	-	-

5.15 Picking suitable belt and pulley dimensions

Table 5-14 shows power ratings for different speed ratios, not included 2.0, 3.61 and 3.73. 5 RPM has been used for this table. It also shows that the power rating is a function of RPM and number of grooves for the smaller pulley. Table 5-14 has ratio 2.8 and 3.0 marked with green and blue, respectively. These has the highest power ratings, making them the most interesting ones.

Not to be confused, the faster pulley (usually the smallest) decides the power rating, and the largest will (obviously) have the largest size, weight, and cost. Also, an increased belt width results in a greater power rating. A suitable ratio between the largest and the smallest pulley should be investigated and obtained.

Table 5-14 8M belt with 5RPM on drum, where yellow marking > 270 W

	2.55	**2.8**	**2.95**	**3.0**	**3.29**
Z_1	44	40	38	48	34
Z_2	112	112	112	144	112
RPM_{ENGINE}	12.5	14	14.75	15	16.45
C-C [mm]	883.8	891.3	895.0	806.7	902.5
Power rating 20mm belt [W]	75	80	78.5	105	65.8
Power rating 30mm belt [W]	118.5	126.4	124.03	165.9	104
Power rating 50mm belt [W]	205.5	219.2	215.09	287.7	180.3
Power rating 85mm belt [W]	357	380.8	373.66	499.8	313.21

Table 5-15 shows a weight and dimension comparison between the ratio 2.8 and 3.0, 8M belt.

Table 5-15 8M pulley weights and sizes [31] for ratio 2.8 and 3.0

Number of teeth	**Weight 50 mm** [kg]	**Material 50 mm**	**Weight 85 mm** [kg]	**Material 85 mm**	**Pitch diameter** [mm]
Z = 40			**1.92**	**steel**	**101.86**
Z = 48	**2.54**	**Steel**	**2.96**	**steel**	**122.23**
Z = 112			**15**	**steel**	**285.21**
Z = 144	**15.2**	**Steel**	**20**	**steel**	**366.69**

The same comparisons for 8M will be done on 14M, covered in Table 5-16 and 5-17.

Table 5-16 14M belt with 5RPM on drum with yellow marks > 270 W

	2.48	**2.76**	**3.0**	**3.29**
$\mathbf{Z_1}$	29	29	30	34
$\mathbf{Z_2}$	72	80	90	112
RPM$_{\mathbf{ENGINE}}$	12.4	13.8	15	16.45
C-C [mm]	866.2 or 936.6	906.4	864.6	871.6
Power rating 40mm belt [W]	248	276	300	493.5
Power rating 55mm belt [W]	372	414	450	740.25

Note: Ratio 2.48 has the lowest pulley weight (for the largest ones), with 72 grooves and a weight of 16.08 kg.

Table 5-17 14M pulley weights and sizes based yellow marks in Table 5-16

number **of teeth**	**Weight 40mm** [kg]	**Material 40mm belt***	**Weight 55mm** [kg]	**Material 55mm belt***	**Pitch diameter** [mm]
Z = 29	**2.58**	**steel**	**2.54**	**steel**	**129.23**
Z = 30	**2.89**	**steel**	**3.14**	**steel**	**133.69**
Z = 34	**4.11**	**steel**	**4.91**	**steel**	**151.52**
Z = 72	**16.08**	**Gray cast iron****	**18.04**	**Gray cast iron**	**320.86**
Z = 80	**16.02**	**Gray cast iron**	**19.96**	**Gray cast iron**	**356.51**
Z = 90	**18.520**	**Gray cast iron**	**21.94**	**Gray cast iron**	**401.07**
Z = 112	**18.94**	**Gray cast Iron**	**30.46**	**Gray cast Iron**	**499.11**

*** Material is obtained from Tyma** [31].

**** Gray cast iron needs to be balanced with v < 30m/s** [31]**.**

5.16 Two temporary and feasible belt dimensions

Several types of 8M belts are doable, providing smaller and lighter pulleys than the 14M. 14M delivers more power with a smaller belt width but has the downside of increased weight and size. Conclusion has come to two systems, one with 8M and one with 14M, specified in Table 5-18.

Table 5-18 Two temporary and feasible belt dimensions

	8M	14M
Ratio	2.8	2.48
Z1	40	29
Z2	112	72
RPM engine	14	12.4
Pitch smallest	101.86	129.23
Weight smallest	1.92	2.54
Pitch Diameter largest [mm]	285.21	320.86
C-C [mm]	891.3	866.2 or 936.6
Weight largest [kg]	15	18.04
Belt width [mm]	85	55
Power rating [W] 5RPM	380.8	372
Material	Steel	Steel and cast iron

5.17 Mounting of the belt system

When assembling the belt system, the belt needs to be pre-tensioned. The pre-tension will act as a force on both the engine and drum shaft. Design manual for gates system, provided by Rubix [6], includes pre tensioning forces, installation tension, mid-belt deflection and belt span. Figure 5-12 shows a sketch of the different factors mentioned.

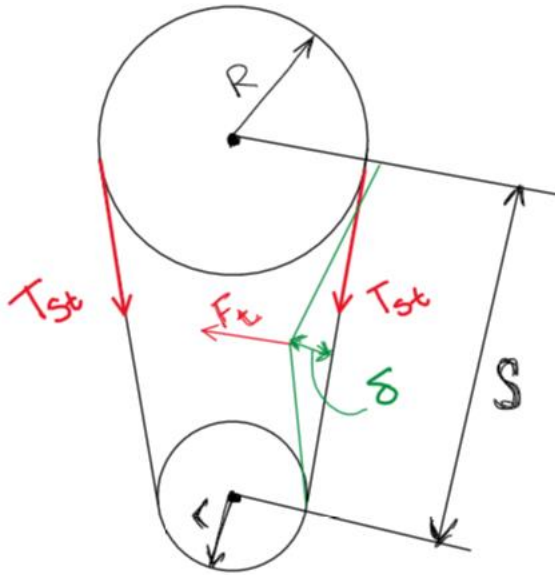


Figure 5-12 Belt setup and parameters

S – Belt span as following

$$S = \sqrt{(C)^2 - (R - r)^2} \tag{5.27}$$

δ – Mid-belt deflection, described as

$$\delta = \frac{S}{50} \tag{5.28}$$

F_t – Installation tension, with following interval

$$F_t = [F_{t\,min}, F_{t\,max}] = \left[\frac{p \times 25}{v}, \frac{p \times 60}{v}\right] \tag{5.29}$$

T_{st} – Static tension pr. belt defined as (Where C is centre distance)

$$T_{st} = 600 \times \frac{p}{v} \tag{5.30}$$

R – Radius of largest pulley
r – Radius of smallest pulley

Results from Equations 5.27 to 5.30 are listed in Table 5-19.

Table 5-19 Different belt configurations for installation purposes

	8M-belt configuration	14M-belt configuration
S – Belt span [mm]	890.57	865.9
δ – Mid-belt deflection [mm]	17.81	17.32
F_t – Installation tension [N]	[75, 180]	[54, 67]
T_{st} – Static tension pr. belt [N]	1800	1607
R – Largest pulley radius [mm]	142.605	160.43
r – Smallest pulley radius [mm]	50.93	64.615

5.18 Verify that existing parts can withstand the pulley system

As the pulley system replaces the chain and sprocket, the drum experiences a different load scenario. As a designer and product developer, these parts should be identified and verified if they can withstand the new load scenario(s). Dimension and design change will most likely occur since the load scenario is different.

There has been no decision upon a safety factor, but a safety of 4 against yielding stress can be a reasonable number. As long as the different parts has sufficient safety factors, they can be thoroughly investigated later on if the belt system goes to production.

5.19 Driven drum shaft

Since the existing shaft wasn't used as a torque converting shaft, it should be checked if it's strong enough to withstand the "added" loads. Figure 5-13 shows the drum shaft, aligned with the circular plate (welded on the non-visible drum), through the bearing and with the pulley attached at the other end.

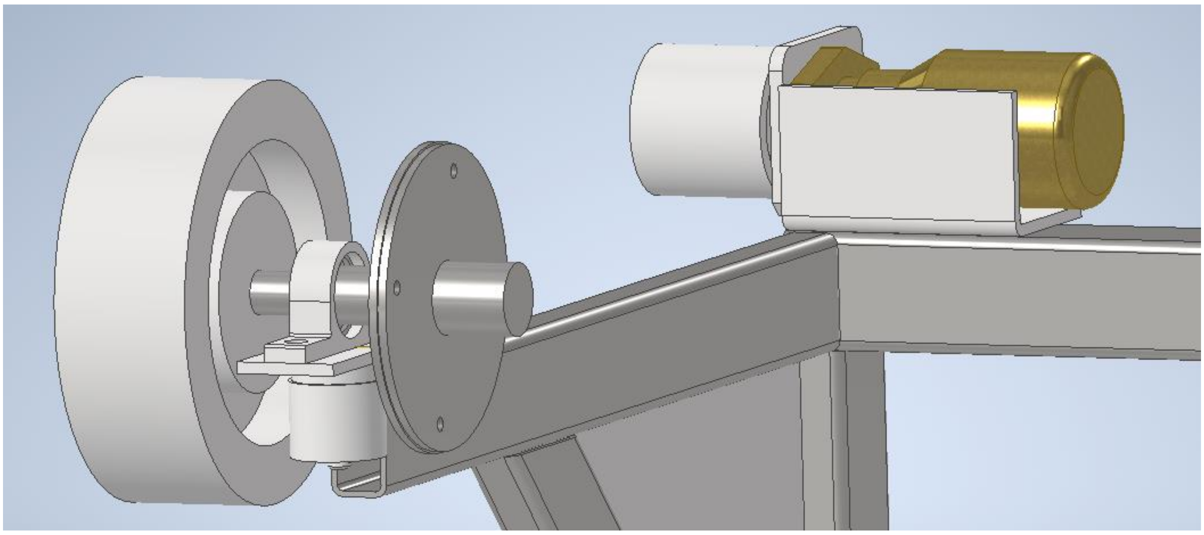


Figure 5-13 Base layer setup for future calculations

Existing drum shaft has a diameter of OD 30 mm at its smallest. That's also where the bearing is mounted. Shaft is extended for pulley installation purposes.

5.19.1 Simulation of drum shaft, OD 30, 35, 38 and 40 mm

Simulations will be worked out in Inventor Autodesk's FEA.

General simulation criterions:

- Shaft and plate are bonded. In reality, there are welds keeping them together
- Shaft can rotate/slide within the bearing, simulating the rolling motion
- Bearing is fixed at its outer perimeter, simulating the bearing being fixed
- Shaft can't slide in Z-direction (horizontally) but can be displaced in the vertical (together with the plate)
- Vertical weight force from the heaviest pulley, 158 N, at a 177 mm distance from the plate face
- Horizontal tension force from the total (resultant) tension of the belt, 3600 N, at a 177 mm distance from the plate face
- Torque at 430 Nm
- Additional counteracting torque on the backside, simulating the counter torque induced by the tensional force of the manure hose.
- S355 material will be used, where it has it's yielding point at 355 MPa for OD ≤ 40 mm and 335 MPa for OD > 40 mm
- Shaft is extended, as mentioned earlier

Simulation 1 – OD 30 mm

Figure 5-14, existing shaft with an OD of 30 mm. 165 MPa is about half the yield strength. This gives a yield safety factor of 2.15, which may seem too low. The overall stress is relatively high.

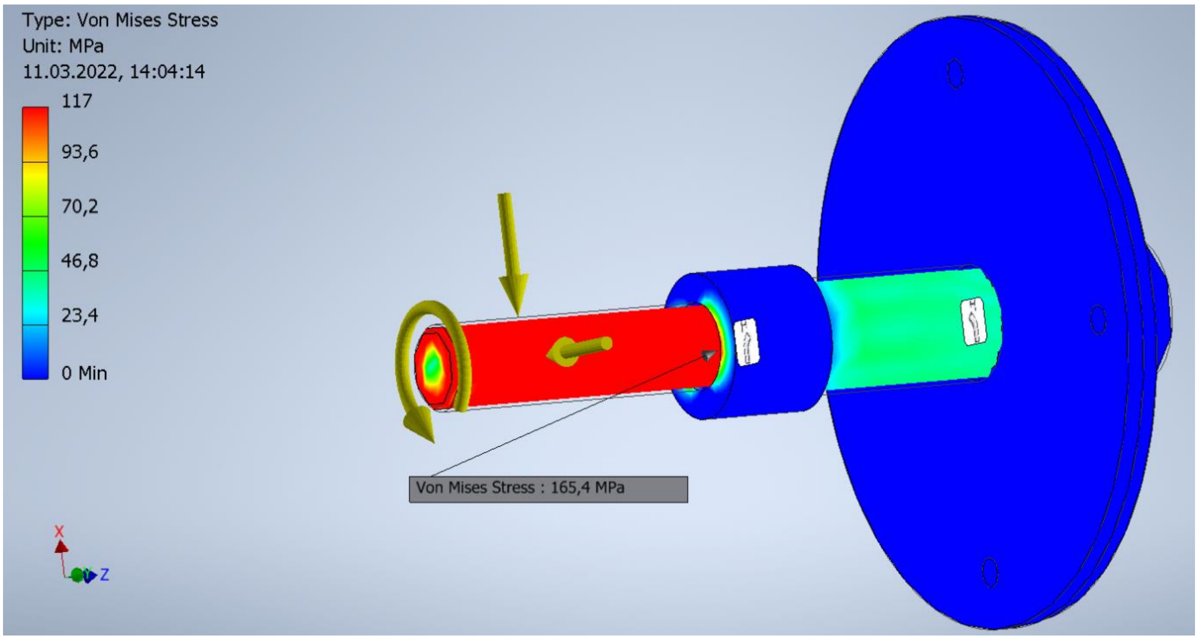


Figure 5-14 FEA of drum shaft with OD 30 mm

Note: Weight force from the drum was not added. Some tweaking was done with the forces and contact conditions. Tweaking did not give satisfying results.

Simulation 2 – OD 35mm

Figure 5-15 shows 105.8 MPa at the dimension change, with a safety factor of 3.38, which is more promising. The stress between the shaft and plate, 179.8 MPa, occurs because they are bonded. In reality, shaft is welded onto the back (and front if necessary).

Note: Simulated bearing component is non-visible in the following analysis.

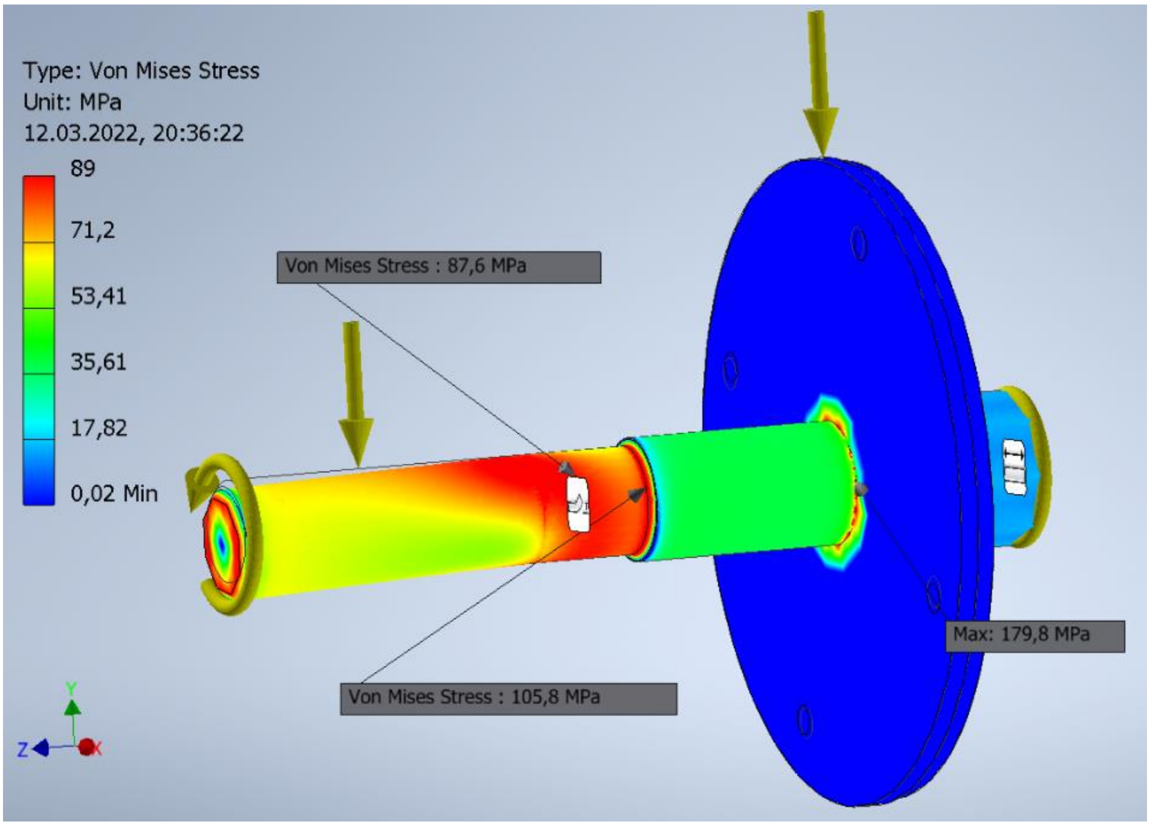


Figure 5-15 FEA of drum shaft with OD 35 mm

Simulation 3 – OD 38 mm

Figure 5-16 shows analysis of 38 mm OD shaft. Mesh area at the dimension change was adjusted to 2 mm. 364.9 MPa peak stresses look like the contact stresses between the shaft and bearing. Or else, it is random peak stresses, i.e., not "real" stresses. Otherwise, 79.9 MPa gives a safety factor of 4.44. That's feasible. Unclear whether the 364.9 MPa is realistic.

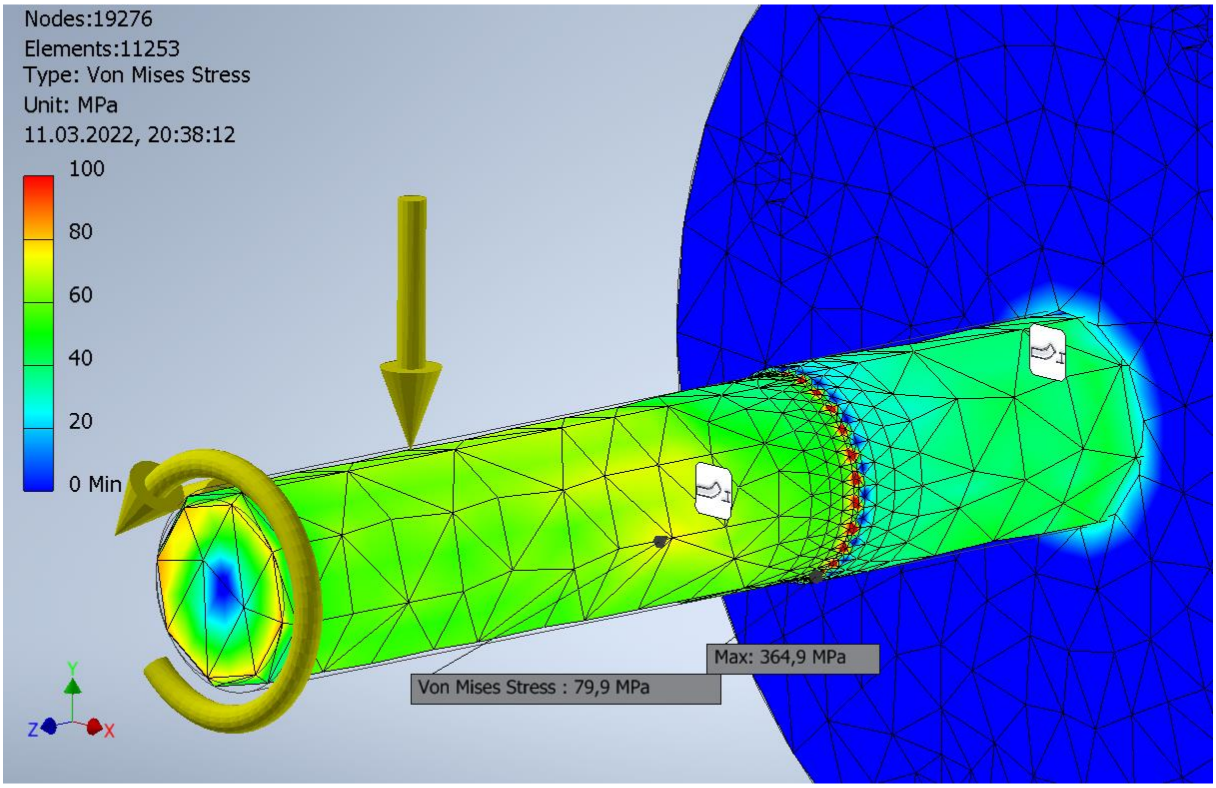


Figure 5-16 FEA of drum shaft with OD 38 mm

Simulation 4 – OD 40 mm

Next shaft has an OD 40 mm, with dimension change to OD 50 mm, with OD 60 mm at the back of the plate. For this, the hole through the plate was adjusted to 50 mm. Reason for the shaft dimension change, is to "make it stronger" by changing it to a greater dimension, rather than tweaking fillets and other parameters. Figure 5-17 shows a more satisfying result, with a max peak stress of 78.71 MPa. Safety factor will then be ~4.26. At this stage, the shaft, bearing and taper bushes for the heaviest and longest hose, needs to be investigated. It seems the shaft should make it based on the safety factor obtained, but that is out of the scope for this study.

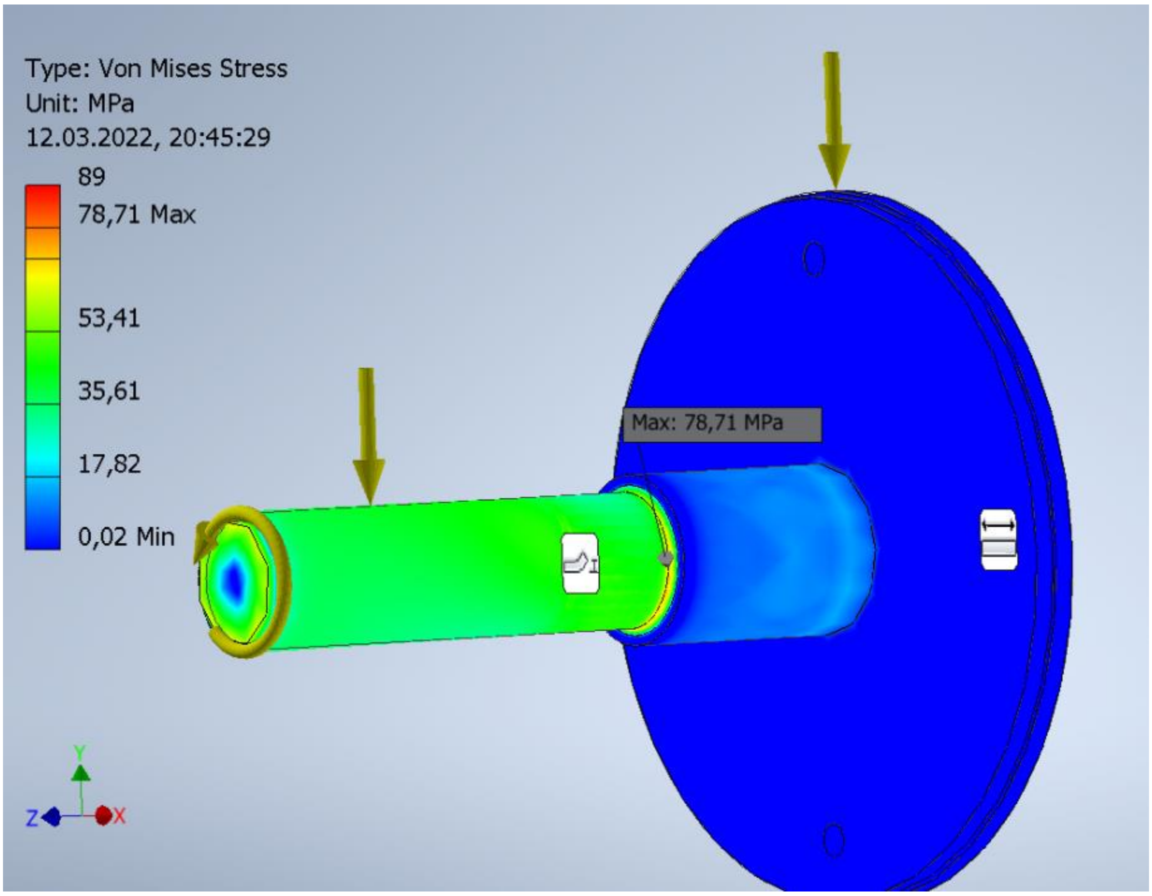


Figure 5-17 FEA of drum shaft with OD 40 mm

5.19.2 New analysis on simulation 4 due to load errors

It was discovered that the tensional forces from the hose was not implemented in the analysis. A new analysis was made, with more parts included, making it easier to analysis. Analysis conditions are equal to that for test 4, with some additional points. The tube where the hose wraps around is "fixed" (not allowed to move or rotate), acting as a counter torque as the hydraulic engine turns the drum. Tensional force was also added in the following analysis, represented in Figure 5-18.

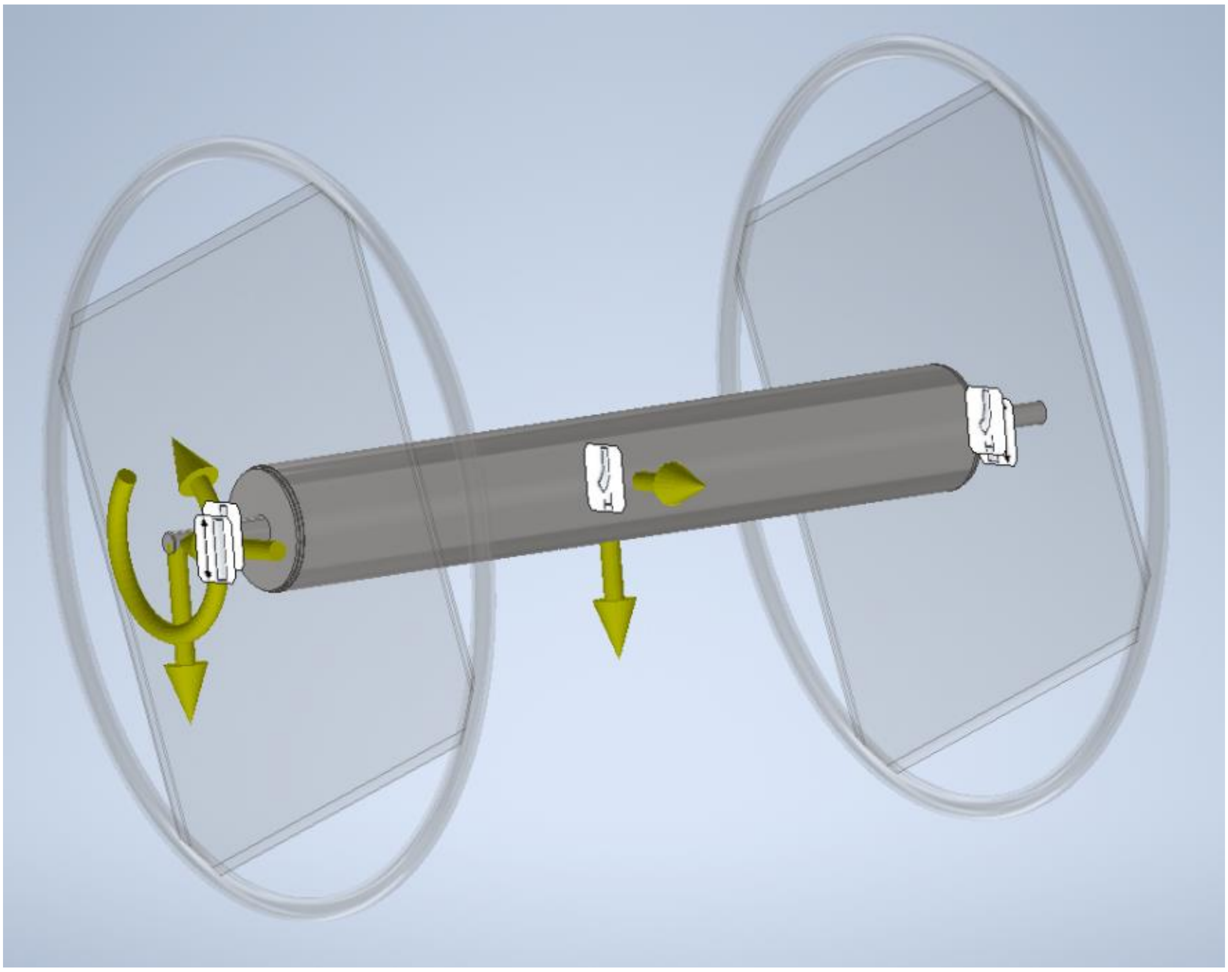


Figure 5-18 Representation of the different loads acting on a simplified model, simulation 4

Equivalent von Mises stresses came to 99.91 MPa, as shown in Figure 5-19. For S355 steel, a safety factor of 3.55 is obtained. It's more than enough for the system and can be feasible for a larger hose type. If there are any concern of the material properties being too "weak", materials like 34CrNiMo6 etc. can be used, which has a greater yield point, and is more durable against fatigue. Even though a safety factor of 3.55 is reasonably good, it's necessary to look at fatigue due to cyclic stresses. Fatigue will be covered later in the design process.

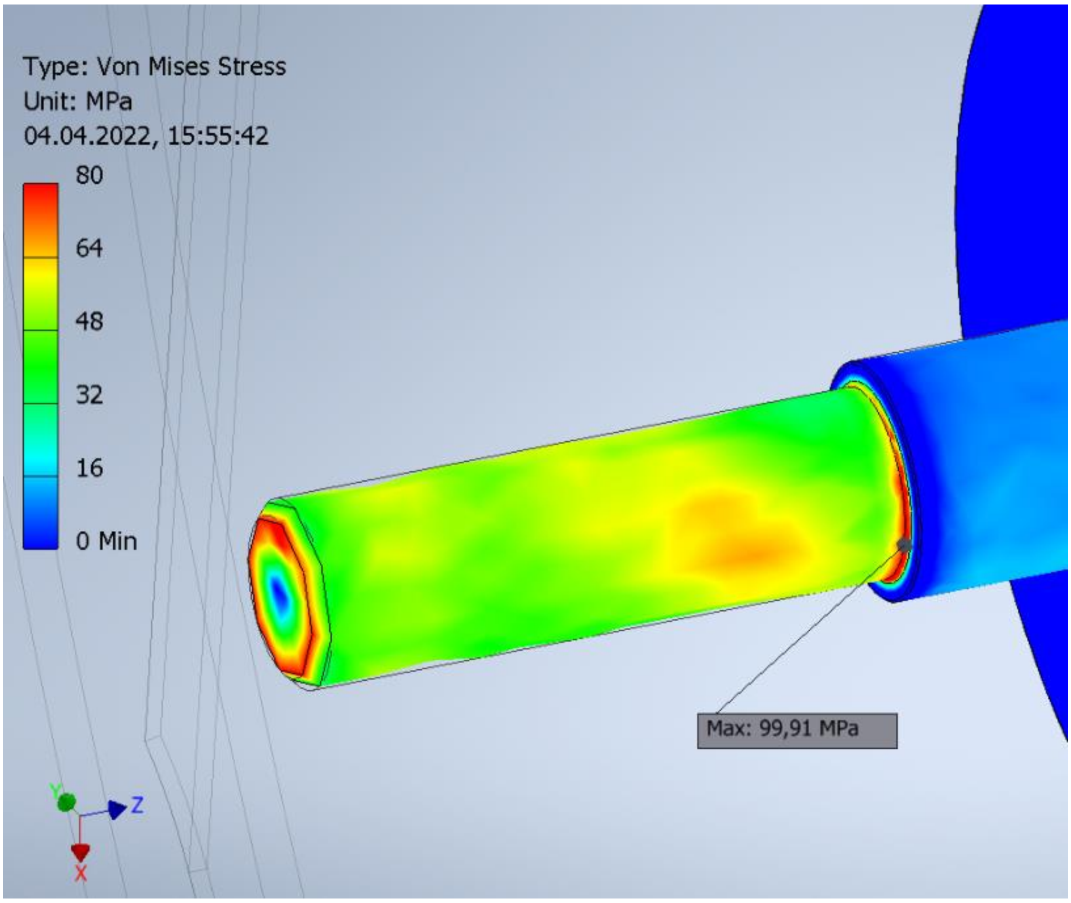


Figure 5-19 New simulation with drum shaft OD 40 mm, max von Mises 99.91 MPa

Shaft used in simulation 4 is shown in Figure 5-20.

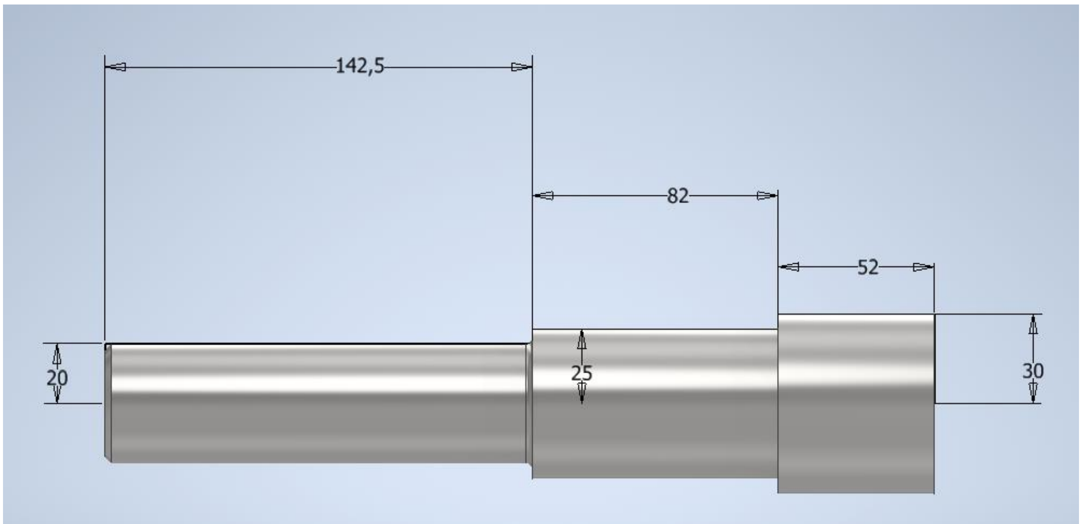


Figure 5-20 New shaft generated from simulation and required mounting for pulleys

The plate will have the same attachment method (welding) to the drum and shaft. Difference will be the bore diameter, increased to ID 50 mm, Figure 5-21.

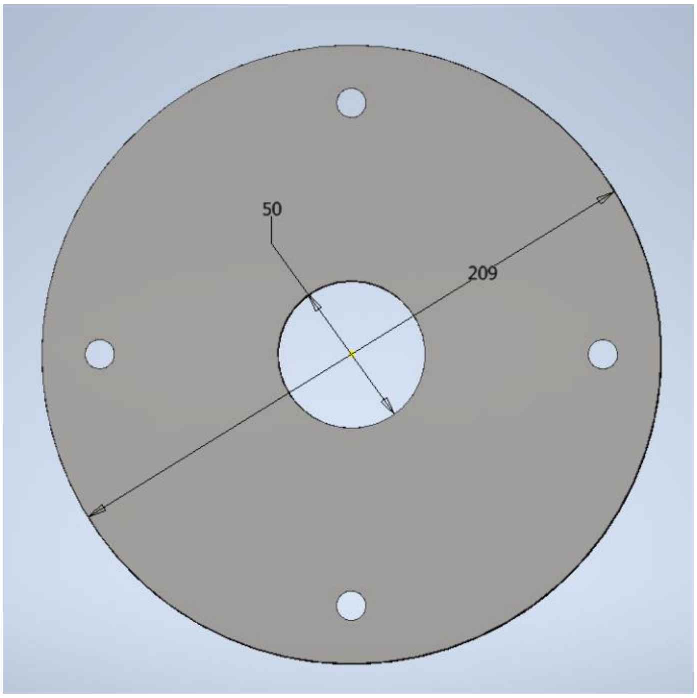


Figure 5-21 Plate with new ID 50 mm due to change in shaft dimensions

5.20 Hand calculations

Hand calculations can be a way to verify the FE analysis. Sometimes, the FEA has been run with the wrong boundary conditions, creating deviations. Not all calculations can be run with a single FEA, like bearing calculations. Hand calculations gives more understanding to the theory, but also provides an overview of the different variables and its consequences on the model.

5.20.1 Scenario 1 – Spooling in the manure hose

First off, a FBD (free body diagram) can identify the bearing reaction forces, but also the different values for bending moment and shear forces. Figure 5-22 shows the drum in a three-dimensional perspective, defining the XYZ-directions.

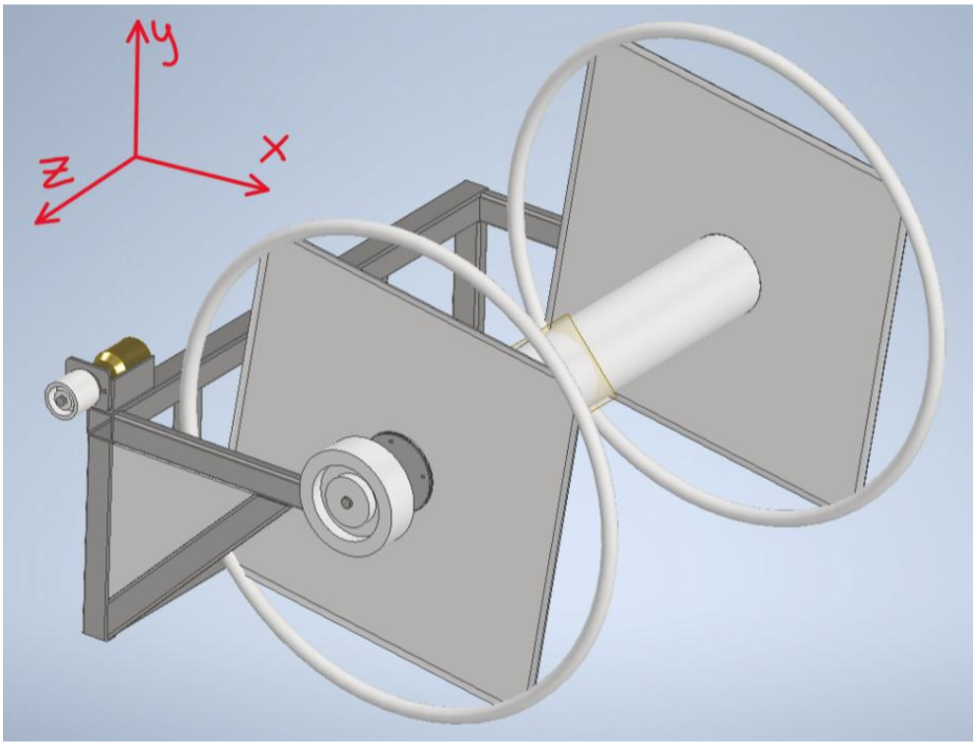


Figure 5-22 Assembly of drum, frame, and belt setup in a three-dimensional perspective

As the manure hose gets spooled in (at the very start), there's a tensional force on the drum, but no weight force from the hose, only the drum itself. FDB for XZ plane is represented in Figure 5-23.

Note: T_{st} has been set to maximum, 1800 N, and the weight forces represents the heaviest pulley. Tension force from the hose is assumed to act in the middle.

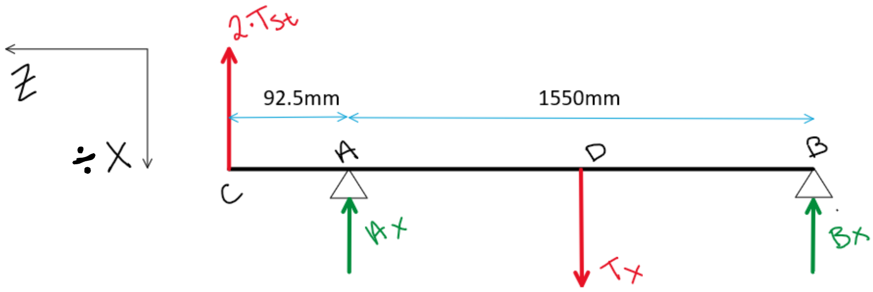


Figure 5-23 FBD of XZ plane

From statics, reaction forces can be defined with the equilibrium equations. The sum of moments of point A

$$\circlearrowleft \sum M_A = 2 \times T_{st} \times 92.5 + T_X \times \frac{1550}{2} - B_X \times 1550 = 0 \tag{5.31}$$

Solving reaction force at point B, B_X equals 2204 N.

Sum of forces in X-direction are defined by

$$\sum F_X = 2 \times T_{st} + A_X + B_X - T_X = 0 \tag{5.32}$$

Equation 5.32 can be used to solve for A_X, giving 1825 N.

YZ plane contains only forces generated by the gravitational acceleration on the drum and largest pulley, as shown in Figure 5-24. With same procedure as XZ, A_Y and B_Y equals 517 N and 489 N, respectively.

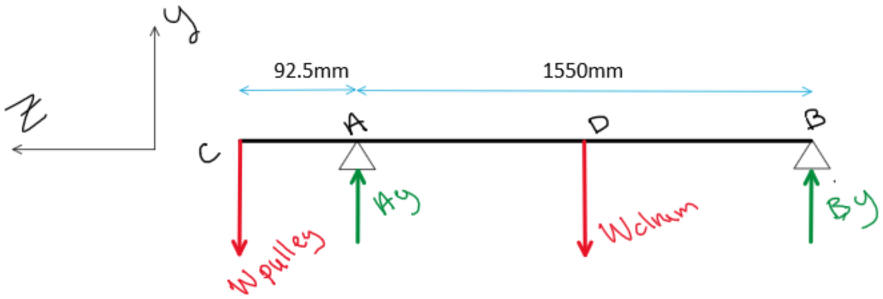


Figure 5-24 FBD diagram of YZ plane

The third dimensional perspective is shown in Figure 5-25.

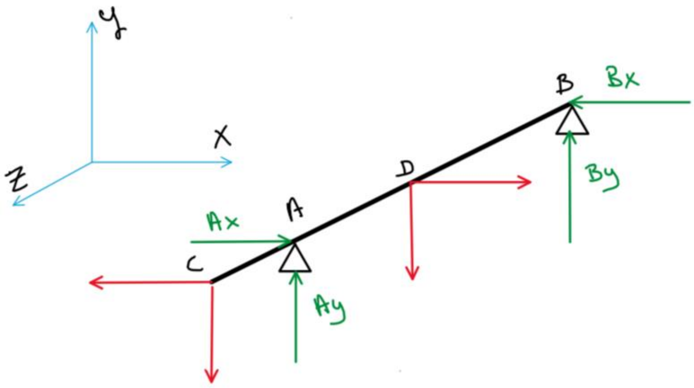


Figure 5-25 FBD in XYZ dimension

Since there are forces in several planes, a (bearing) resultant can be found by the Pythagoras Theorem

$$a^2 = b^2 + c^2 \tag{5.33}$$

Where the resultant bearing forces, F_A and F_B, shown in Figure 5-26.

Figure 5-26 Resultant forces on bearing A and B, due to reaction forces in A and B

Then, the bearing resultant forces at scenario 1 can be written as

$$F_A = \sqrt{A_X^2 + A_Y^2} = 1897\ N \tag{5.34}$$

and

$$F_B = \sqrt{B_X^2 + B_Y^2} = 2258\ N \tag{5.35}$$

5.20.2 Scenario 2 – Standing still with fully loaded drum

In consideration, the resultant forces should be investigated for scenario 2, where the whole manure hose is winded in, adding 770 kg on the drum, setting the tension force to zero. Going through the same process with FBD for both XZ and YZ planes, and looking at the resultant force, the largest radial forces can be identified. Scenario 2 yields

$$F_A = \sqrt{(-3814)^2 + 4455^2} = 5865\ N \tag{5.36}$$

And

$$F_B = \sqrt{214.8^2 + 4256.8^2} = 4262\ N \tag{5.37}$$

5.20.3 Bending moment diagram (BMD)

From the FBDs, equations can be expressed for the BMDs. Both BMD and SFD can be defined visually, at least its shapes. Having a theoretical foundation, like an equation, it's easy to find the bending or shear for a specific location, several times. For BMD in XZ and YZ, following procedure will be followed.

Note: An example of a resultant bending moment will be done at the weld location of the shaft.

Sketching up imaginary cuts on the element, between the acting forces, with bending moment for each cut. XZ plane can be sketched like Figure 5-27 shows.

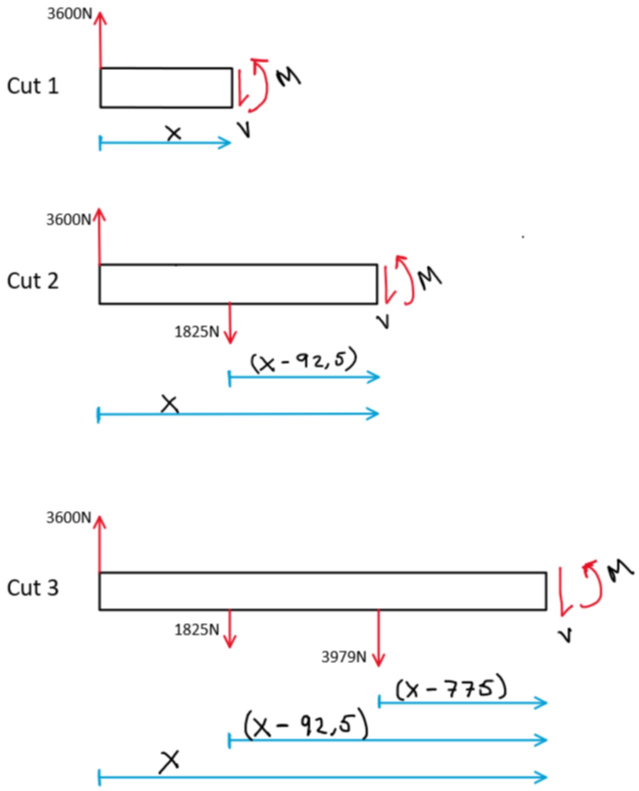


Figure 5-27 Representation of imaginary cuts in XZ to aid with moment diagram and loads

Each cut has its own equation. Using sum of moments, by equilibrium conditions, where the moment "M" represents the positive direction

$$\circlearrowleft \sum M = 0 \tag{5.38}$$

Equilibrium equations is defined as, in Nm

$$Cut\ 1\ : M_1 = 3600X, \quad for\ \ X < 92.5\ mm \tag{5.39}$$

$$Cut\ 2 : M_2 = 1775X + 168.8, \quad for\ \ 92.5mm < X < 868\ mm \tag{5.40}$$

$$Cut\ 3 : M_3 = \ -2204X + 3253.5, \quad for\ \ X > 868\ mm \tag{5.41}$$

BMD for XZ is represented in Figure 5-28.

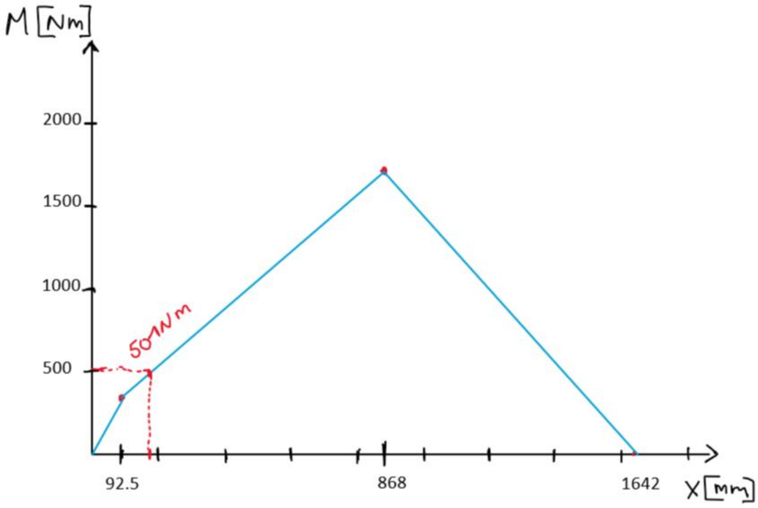


Figure 5-28 BMD for XZ plane

Weld is located at X ~ 187.5 mm from the left, where M_Y equals 501 Nm, obtained from the BMD. With the same procedure for YZ plane, BMD can be defined, as shown in Figure 5-29.

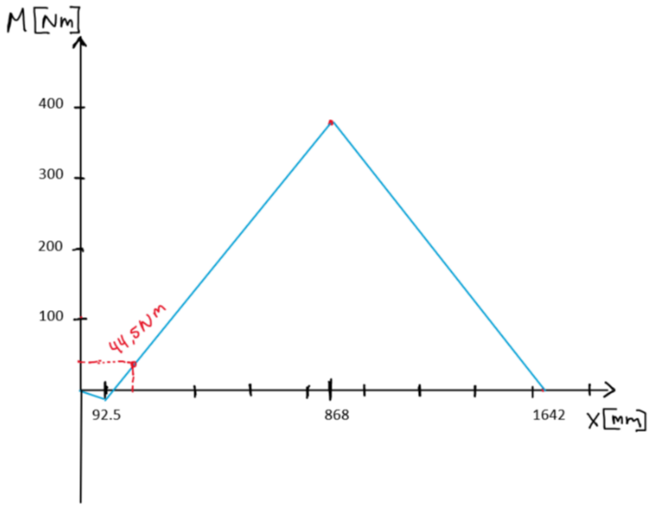


Figure 5-29 BMD for YZ plane

At 187.5 mm, M_X equals 44.5 Nm in the YZ plane. Like the radial resultant forces on the bearings, there is also a resultant bending moment, i.e., max bending in the weld. Equation 5.42 defines the resultant bending moment.

$$M_{res} = \sqrt{M_x^2 + M_Y^2} \cong 503\ Nm \tag{5.42}$$

Note: X-axis in the diagrams does not refer to the X-axis in the representation of the three-dimensional drum model.

5.20.4 Shear force diagram (SFD)

Taking advantage of the bending moment formulas, the shear force formulas can be found by taking the derivative of the bending moment formulas. Worth noting, the shear force diagram can be easily sketched by looking at the acting and reaction forces. In addition, since the shear forces are the derivative, they play the role as the "slope" in the moment functions. When having several loads, distributed loads etc., the derivatives (or the integrals) can be a clever way to define the bending and shear forces, as well as the deflections.

Note: An example of a resultant shear force will be done at the weld location of the shaft.

With shear forces defined as

$$V(x) = \frac{dM(x)}{dx} \tag{5.43}$$

The shear force formulas for the XZ-plane can be written as

$$Cut\ 1\ : V_1 = 3600, \qquad for\ \ X > 92.5\ mm \tag{5.44}$$

$$Cut\ 2 : V_2 = 1775, \qquad for\ \ 92.5mm < X < 868\ mm \tag{5.45}$$

$$Cut\ 3 : V_3 = \ -2204, \qquad for\ \ X > 868\ mm \tag{5.46}$$

Equation 5.44, 5.45, and 4.46 can then be used to sketch the SFD for XZ-plane, represented in Figure 5-30.

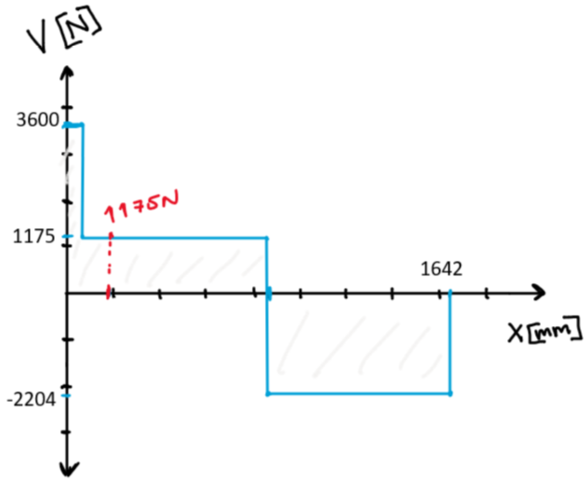


Figure 5-30 SFD for XZ plane

And the YZ-plane in Figure 5-31.

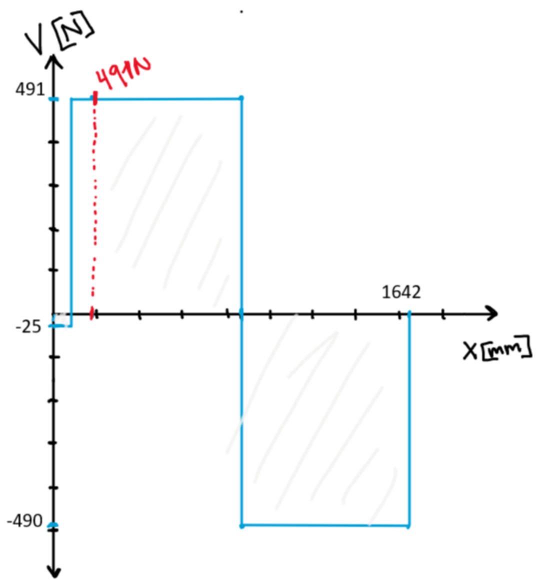


Figure 5-31 SFD for YZ plane

As before, resultant shear force at the weld can be written as

$$V_{res} = \sqrt{V_{XZ}^2 + V_{YZ}^2} = 1273N \tag{5.47}$$

5.20.5 Stresses at dimension change

An imaginary cut can identify the materials reaction forces. BMD and SFD can aid to evaluate the loads at distance 113 mm from the driven pulley. The max moment was read to 370.3 Nm. Torsional moment will be the same as always, 430 Nm. The transverse shear force is neglected, due to its low shear stress contribution. Reaction forces can be represented with a small element at the maximum bending stress (equal absolute value for bending and compression), at the outer perimeter, shown in Figure 5-32.

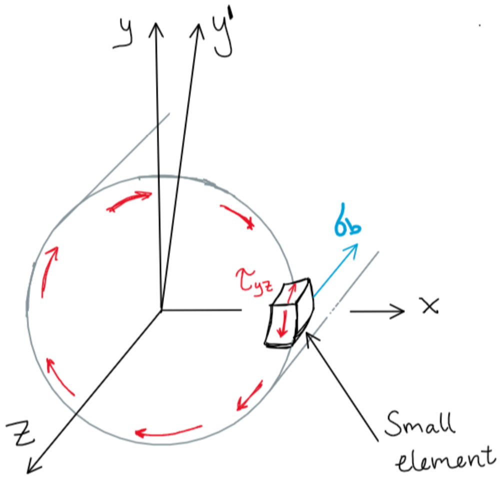


Figure 5-32 Max stress point due to torsional and max bending stress, represented with a small element

y´-axis represents the max bending moment axis. For a shaft with change in dimensions, correction factor will be used to compensate for stress concentrations at the dimension change. For bending load, a stress concentration factor, $K_{bending}$, is set to 2.05 [15]. Max bending stress can be written as

$$\sigma_{bmax} = \frac{K_{bending} \times M \times r}{I_{y'}} = \frac{2.05 \times 503000 \times 20}{\pi \times \frac{20^4}{4}} = 164.11\ MPa \tag{5.48}$$

And torsional shear force, with stress concentration factor $K_{torsion}$ of 1.55 [15], gives a corrected torsional shear stress

$$\tau_{torsion} = \frac{K_{torsion} \times T \times r}{J} = \frac{1.55 \times 430000 \times 20}{\pi \times \frac{20^4}{2}} = 53.01\ MPa \tag{5.49}$$

From the obtained stresses, von Mises yields

$$\sigma_{max}^{vM} = \sqrt{164.11^2 + 3 \times 53.01^2} = 188\ MPa \tag{5.50}$$

Which results in a safety of 355/188 = 1.88. This gives more reason to apply a stronger material, such as 34CrNiMo6 for the shaft. It's noted that there is a deviation between simulations and hand calculations. Matter will be discussed later.

5.21 Fatigue and safety factor of drum shaft

Fatigue has not been discussed thoroughly since the operation time is not that high for the manure hose system. HMR recommended a service life of 50 hours per year, for 10 years. For 15 RPM, rotations will be

$$10 years \times 50 \frac{hours}{year} \times 60 \frac{minutes}{hour} \times 15 \frac{rotations}{minute} = 450000\ rotations \tag{5.51}$$

Initially, a specific point on the shaft will experience the highest von Mises stress. After a quarter turn, it will only experience the torsional shear forces i.e., minimum von Mises stress, illustrated in Figure 5-33.

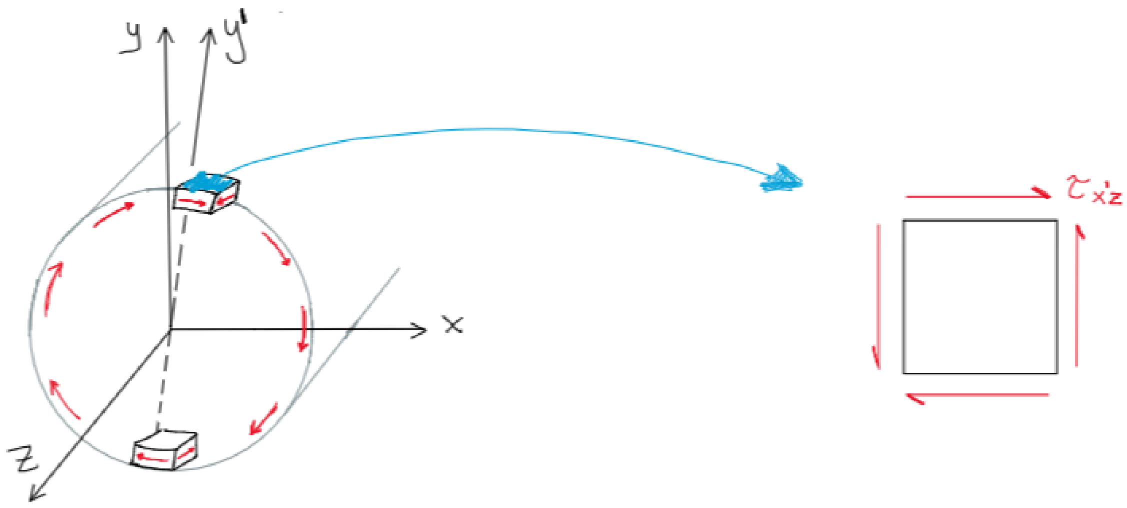


Figure 5-33 Minimum von Mises stress with only torsional stress

After another quarter turn, it will again experience max von Mises stress. That gives enough parameters to calculate the cyclic stresses. From the last calculation, max von Mises stress came to 188 MPa. Minimum von Mises stress can then be defined as

$$\sigma_{min}^{vM} = \sqrt{0^2 + 3 \times 53.01^2} = 91.81\ MPa \tag{5.52}$$

Cyclic stresses will be

$$\sigma = \sigma_m \pm \sigma_a \tag{5.53}$$

Where σ, σ_m and σ_a is stress, midrange/mean stress, and amplitude stress component, respectively, all based on von Mises. Midrange stress can be described as

$$\sigma_m = \frac{\sigma_{max}^{vM} + \sigma_{min}^{vM}}{2} \tag{5.54}$$

And the amplitude stress component as

$$\sigma_a = \left|\frac{\sigma_{max}^{vM} - \sigma_{min}^{vM}}{2}\right| \tag{5.55}$$

Combining Equation 5.52 to 5.55, cyclic stresses equal

$$\sigma_{vM} = \sigma_m \pm \sigma_a = \frac{188 + 91.81}{2} \pm \left|\frac{188 - 91.81}{2}\right| \tag{5.56}$$

$$\sigma_{vM} = 140 \pm 48.1 \tag{5.57}$$

For S355 material, a S-N diagram can decide on the shaft's lifespan. The S-N curve is based on cyclic stresses i.e., amplitude stresses, where the mean stress equals zero. In this specific case, there's no mean stress of zero.

Using the Goodman rule, an equivalent amplitude stress can be obtained, that can be used for mean stress equal to zero. Goodman rule is defined as

$$\frac{\sigma_a}{\sigma_{eq}} + \frac{\sigma_m}{S_u} = 1 \tag{5.58}$$

Where σ_a, σ_m, σ_{eq}, and S_u are the amplitude, midrange, equivalent amplitude and ultimate tensile stress, respectively. Solving for the equivalent amplitude stress

$$\sigma_{eq} = \frac{\sigma_a}{\left(1 - \frac{\sigma_m}{S_u}\right)} = \frac{48.1}{\left(1 - \frac{140}{510}\right)} = 66.3\ MPa \tag{5.59}$$

The equivalent amplitude stress can then be plotted into the S-N curve., reading off the shaft's life cycle. New simulated cyclic stresses are visualised in Figure 5-34, compared to the actual stresses.

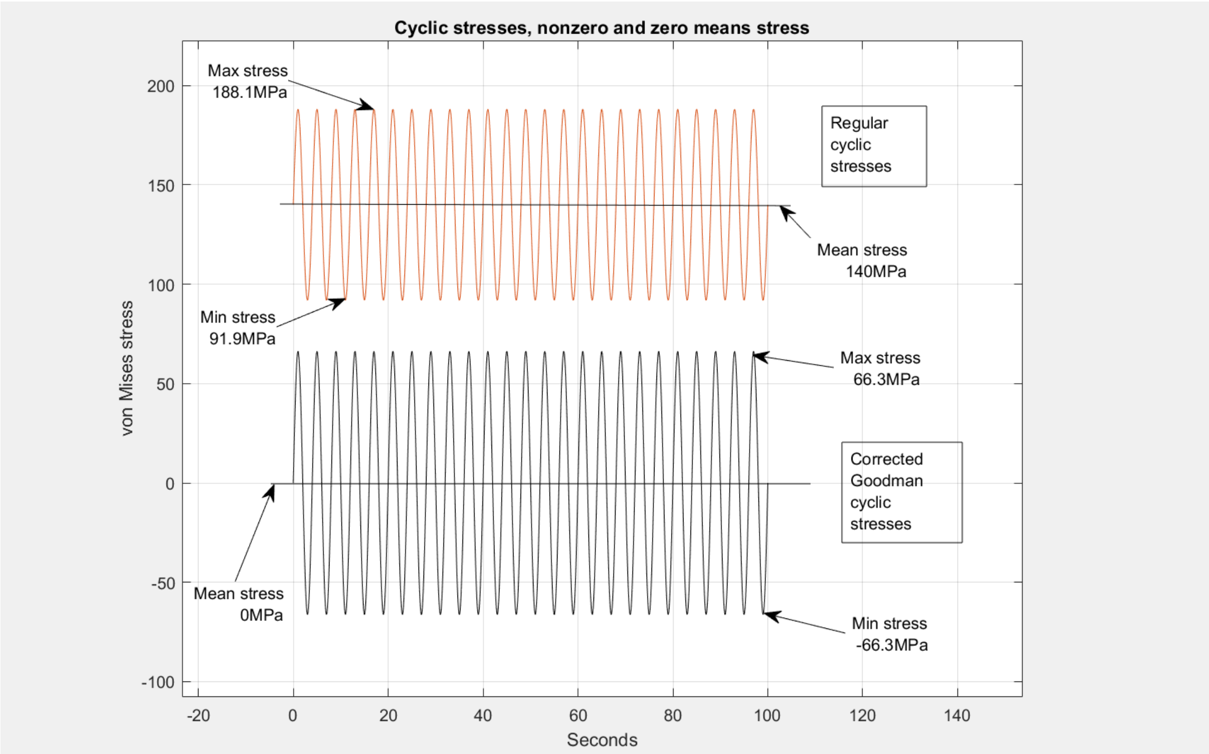


Figure 5-34 Cyclic stresses for a non-zero mean stress vs. zero mean stress, corrected with Goodman rule

The equivalent amplitude stress should be lower than the fatigue stress limit. From a study at the Department of Materials Science Engineering, Ankara, a S-N curve for ST52 (similar specifications as S355) [32], was defined through experimental work. The fatigue stress limit came to ~204 MPa. This indicates infinite life, with a safety against fatigue at 204/66.3 = 3.08 for the shaft.

5.22 Check engine shaft for loads.

As the system experiences different load scenarios, this affects the hydraulic engine. A simple FEA will be done of a simulated engine model with an extended shaft.

5.22.1 General analysis conditions

- Engine bracket is fixed
- Engine and bracket are bonded together. Assuming bolts and the flange will withstand the loads
- Extended shaft and engine shaft are bonded together
- Vertical weight force represents the gravitational force from the heaviest pulley, 25 N, acting at distance of 62.5 mm from the engine flange
- Horizontal tension force represents the tensional force from the belt tensions, 3600 N, acting at 62.5 mm from the engine flange
- Torque from ratio of 2.5, 172 Nm

5.22.2 MR315C engine – OD 25 mm

Shaft and extender are set to the existing OD of 25 mm. Extended shaft is necessary since the pulleys needs a longer shaft to be properly fastened. Figure 5-35 shows that it experiences 247 MPa, which is relatively high. As well as the loads acting on the shaft, the engine has internal parts like bearing(s), maybe some seals, and engine unit, that will experience reaction forces. Since belt tension force acts as the largest load, it should be considered lowering the belt tension force, especially if the plan is to keep the setup as it is. These types of pumps also have an optional shaft dimension of OD 32 mm, which lowers the stresses on the shaft, but not the reaction forces in the engine.

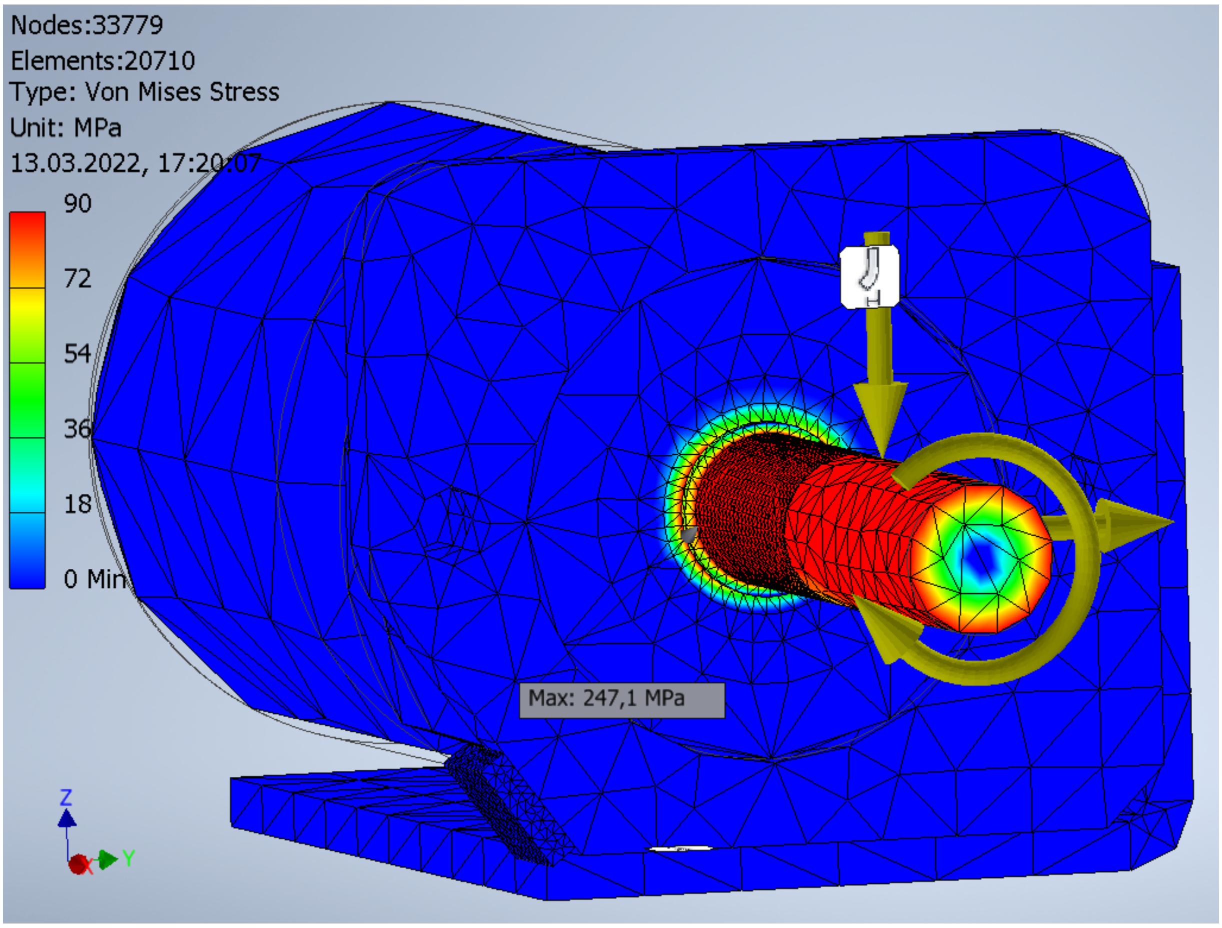


Figure 5-35 FEA of hydraulic engine with OD 25 mm and belt tension of 1800 N per belt

5.22.3 MR315CB – OD 32 mm

Same engine can be delivered with a shaft OD of 32 mm [13]. It's unknown if the shaft can be changed, or if a new motor must be invested in. If a new motor should be invested in, MR400CB [13] could be a better engine when turning at low speeds.

The torque is the only load that will increase when increasing the load (or RPM). Further analysis, where the torque was increased by a factor of 4, the max von Mises stress resulted in 131.9 MPa, illustrated in Figure 3-36. Torque shown in Figure 5-36 has been placed at the same location as the other acting forces.

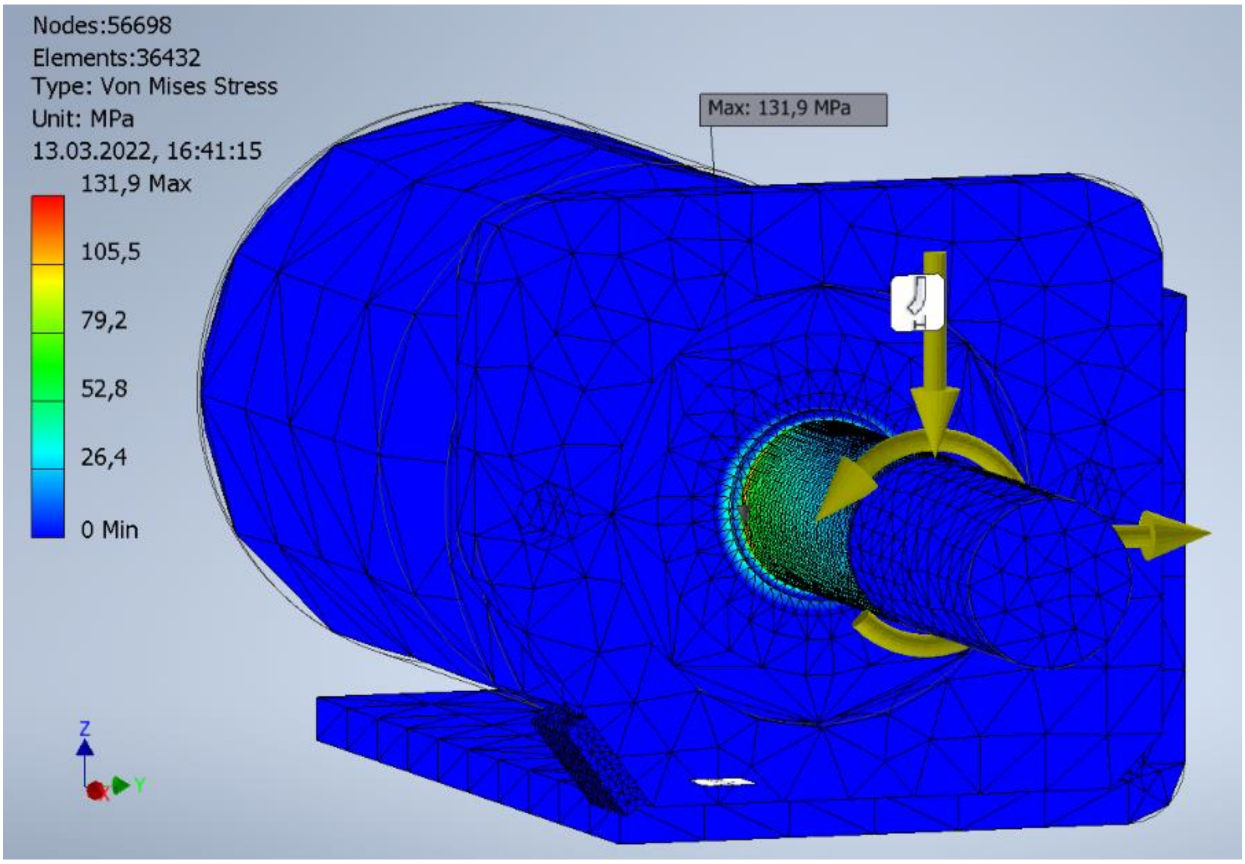


Figure 5-36 FEA engine with OD 32 mm, belt tension of 1800 N per belt, and four times 430 Nm torque

Still, the reaction forces in the engine hub or internal parts, is a concern that needs to be addressed. In addition, the shaft extender also needs to be mounted to the motor shaft. Either way, a design for a (simulated) shaft extension is necessary.

5.23 Bearing loads

General assumptions

- Bearing has both radial and possible moment loads I.e., needs to withstand moments
- Pillow block bearing, suitable for its housing accessories

According to SKF, there are two regular ways of dimensioning and choosing the right bearing type. Size selection based on rating life or based on static loads. Rating life represents the dynamic loads of the bearing. Dynamic loads results in rolling contact fatigue, leading to a correct rating life based on the expected operating conditions. On the other hand, fatigue is not the only concern. Slow rotations and static loads can cause plastic/permanent deformations, and requires static safety factor calculations, s_0, for the bearing [33].

Since the bearings will rotate under load at low speeds, the static load factors will be in focus. SKF states that RPM < 10 should be dimensioned by static load assumptions. If the bearing is design based on rating life (with low RPM), the dynamic load rating, C, would be very low. That would result in seriously overload in the slowly rotating bearing [34]. Bearings will be picked based on static loads.

5.23.1 Bearing calculation for static load

Table 5-20 summarizes and shows that the largest radial forces occur during scenario 2, where bearing A experiences the largest load.

Table 5-20 Forces acting on bearing A and B for scenario 1 and 2

	Radial X-forces [N]	Radial Y-forces [N]	Resultant force [N]
Scenario 1 Bearing A	-1825	516.92	1897
Scenario 1 Bearing B	2204	489	2258
Scenario 2 Bearing A	-3814	4455	5865
Scenario 2 Bearing B	214.8	4256.8	4262

Further calculations will use the largest load, 5865 N, as a cornerstone for selecting the bearings.

Choosing the correct bearing can be done with the aid of SKFs static load size selection [34]. For a static load scenario, selected bearing requires a basic static load rating, C_0, that is larger than the equivalent static load, P_0, multiplied with the static safety factor, S_0.

$$P_0 \times S_0 \leq C_0 \tag{5.60}$$

Where the equivalent static load can be written as

$$P_0 = X_0 \times F_r + Y_0 \times F_a \tag{5.61}$$

X_0 and Y_0 are the radial and axial load factors, respectively. In this case, the axial force F_a is zero. The radial force F_r stands. Since X_0 is a factor less than 1, X_0 will be neglected (or set to 1), so that.

$$P_0 = F_r = 5.87\ kN \tag{5.62}$$

Static safety factor is assumed $S_0 = 1$ for this, based on some continuous loads, low vibrations and gravitational forces [34]. From Equation 5.62, the required static load rating for the selected bearing, will be

$$C_0 \geq 5.87 \times 1 = 5.87kN \tag{5.63}$$

Existing pillow bearing has C_0 of 1150 kg, which equals to 11.28 kN. 11.28 kN is within the requirements, with the highest S_0 to be 11.28/5.87 = 1.92. Since the shaft analysis indicated generated a dimension change, a pillow block bearing with ID 40 mm is selected.
SKFs SY 40 FM with C_0 = 19 kN gives a reasonable factor of

$$S_0 = \frac{19}{5.87} = 3.27 \tag{5.64}$$

5.23.2 Bearing load at two-dimensional tension force

Recalling that the tensional hose direction has been set perpendicular to the drum, all axial bearing loads has been neglected. Figure 5-37 shows the worst-case scenario at an initial start, having an added axial load.

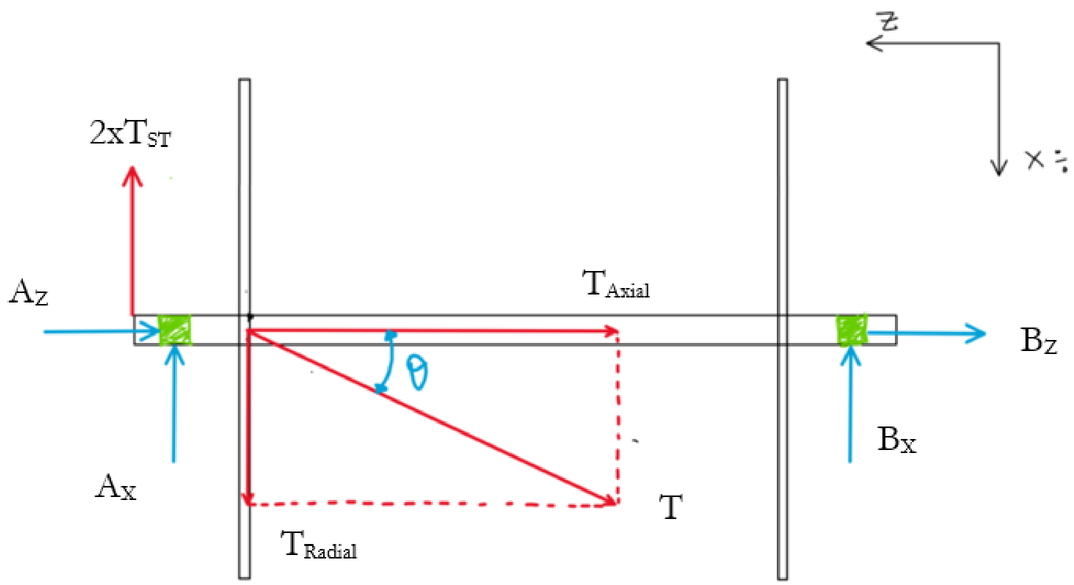


Figure 5-37 Bearing loads if the hose where to pull at an angle

The angle θ can be found from trigonometry based on the dimension of the drum

$$\theta = \tan^{-1}\left(\frac{525\ mm}{1379\ mm}\right) = 20.8\ \circ \tag{5.65}$$

Since T = 3979 N, radial and axial component is divided into

$$T_{Axial} = 3979 \times \cos(20.8) = 3719.7\ N \tag{5.66}$$

$$T_{Radial} = 3979 \times \sin(20.8) = 1413\ N \tag{5.67}$$

From this, the bearing radial forces can be identified. With the same procedure as before, a FBD for XZ gives the following parameters

$$A_X = -2480\ N\ and\ A_Z = -1860\ N \tag{5.68}$$

$$B_X = 293\ N\ and\ B_Z = -1860\ N \tag{5.69}$$

The radial forces in XZ and YZ plane are highest at bearing A, resulting in a resultant radial force of

$$F_A = \sqrt{(-2480)^2 + 517^2} = 2533.3\ N \tag{5.70}$$

From Equation 5.61, the equivalent static load [35] can be written as

$$P_0 = X_0 \times F_A + Y_0 \times A_Z = 0.6 \times 2533.3 + 0.5 \times |-1860| = 2450\ N \tag{5.71}$$

Where $X_0 = 0.6$ and $Y_0 = 0.5$ for single row ball bearing [35]. Two conditions must be met

$$S_0 P_0 \leq C_0 \tag{5.72}$$

$$1 \times 2.450 \leq 19 \rightarrow OK$$

And

$$A_Z \leq 0.5 C_0 \tag{5.73}$$

$$|-1860| \leq 0.5 \times 19000$$

$$1860 \leq 9500 \rightarrow OK$$

The new pillow block bearing meets the requirements and can be used.

5.24 Weld on the driveN shaft

The shaft mounted through the drum plate (Figure 5-38) experiences loads that has not been a concern in the existing design. Consequence of this new design is the additional loads acting on the weld. It should be addressed whether it's ok or not.

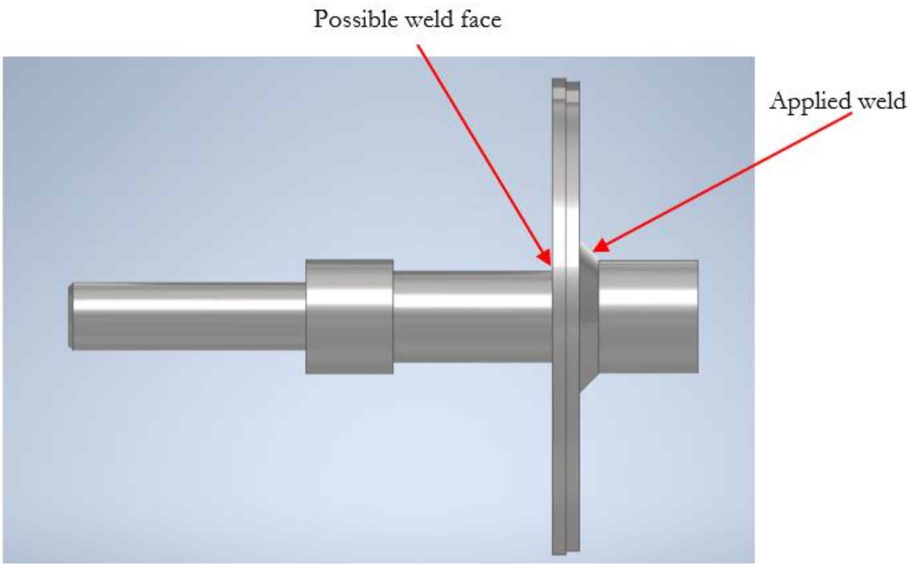


Figure 5-38 Drum shaft with location of applied weld as of today vs. possible weld face

General load consideration represented in Figure 5-39

- Shear stress due to torque
- Shear and tensile stress due to drum weight
- Bending stress due to bending moment in the whole drum

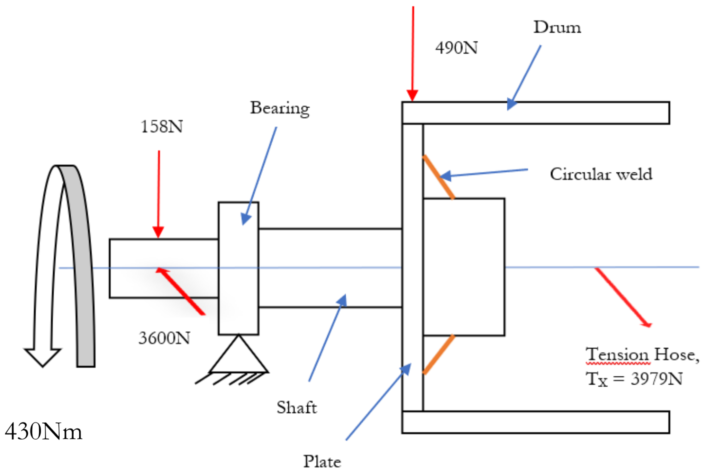


Figure 5-39 Representation of the drum shaft with its components

Weld is assumed to experience bending moment, torsional moment and shear stresses. By locating the shear and bending at the weld location, they can be combined with the torsional stress, expressing the equivalent stresses acting at the weld.

5.25 Weld stresses

Weld stresses will be based of the torsional moment, bending moment and the shear force at the point 187.5 mm, and shaft diameter of 60 mm. Weld leg will be set to 12 mm, which is the (regularly smallest) width of the plate. Weld throat will then be

$$a = t \times 12 = 8.5\ mm \tag{5.74}$$

Note: weld is applied on the back side but has a possibility to be welded on the front.

5.25.1 Max shear stress

Average transverse shear stress due to shear force can be defined as

$$\tau_{shear} = \frac{V}{A} = \frac{V}{2\pi ra} \tag{5.75}$$

From the SFD, V equals 1273 N. Shear stress yields, with a radius r of 30 mm

$$\tau_{shear} = \frac{V}{2\pi ra} = \frac{1273}{2\pi \times 30 \times 8.5} = 0.80\ MPa \tag{5.76}$$

Shear stresses due to torsion is

$$\tau_{torsion} = \frac{T \times r}{2\pi r^3 a} = \frac{430 \times 10^3}{2\pi \times 20^2 \times 8.5} = 9.0\ MPa \tag{5.77}$$

Bending stresses due to bending will be

$$\tau_{bending} = \sigma_b = \frac{M \times y}{I} = \frac{503 \times 10^3}{\pi \times 20^2 \times 8.5} = 21\ MPa \tag{5.78}$$

The max shear stress can be found by a vector resultant, treating shear and bending stresses as vectors. In 2D, Pythagorean theorem can be used to find the resultant (hypotenus) when the vectors act at a right angle (e.g., X-Y directions). Same principle can be used finding the resultant in a 3-dimensional state, where vector A_{res} can be written as

$$A_{res} = \sqrt{A_x^2 + A_y^2 + A_z^2} \tag{5.79}$$

Figure 5-40 visualizes the stresses acting.

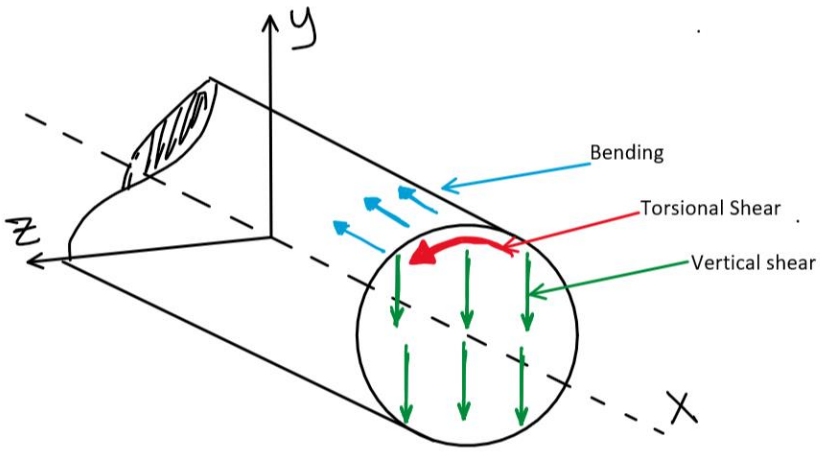


Figure 5-40 Stresses in the shaft due to bending, torsion and shear force

Threating every vector perpendicular to each other, as well as using max values, max shear stress can be written as

$$\tau_{\max weld} = \sqrt{\tau_{shear}^2 + \tau_{torsion}^2 + \tau_{bending}^2} = \sqrt{0.80^2 + 9.0^2 + 21^2} = 22.86\ MPa \tag{5.80}$$

As the maximum shear stress occurs at an angle of 45 degrees on the tensile surface (from simple tensile stress σ), the maximums shear stress at yield can be written as Re/2 [4]. From this, the safety factor is 177.5/22.86 = 7.33**.** There is no concern of material yield with these stresses.

5.25.2 Equivalent weld stresses

From Härkegård [36], the equivalent stress can be found by looking at the throat plane of the weld. The same procedure of finding stresses due to bending and torsion are the same as before but has another way of finding the equivalent stress.

The weld will experience bending, torsion and shear. Following equations is similar to the max shear stress method equations. Mean shear stress from calculations above are very low and will be neglected from this calculation.

Stresses due to bending moment

$$t_x = \frac{M}{\pi r^2 a} = 21\ MPa \tag{5.81}$$

Axial stress vector, t_X, comes from a simulated force acting on the throat plane in the horizontal direction of the element, shown in Figure 5-41. For a<<r, t_X is assumed constant throughout the whole throat cross-section.

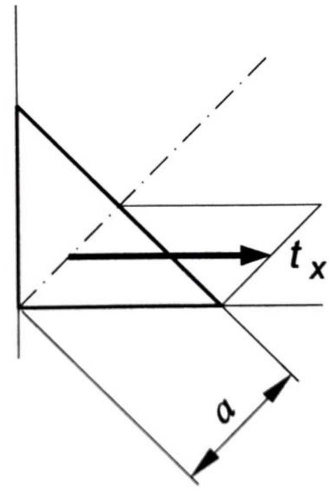


Figure 5-41 Axial stress vector on the throat width [32]

Stresses due to torsional moment

$$\tau_{\parallel} = \frac{T}{2\pi r^2} = 9\ MPa \tag{5.82}$$

$\tau_{\parallel}$ represents the shear stress acting along the throat plane. The perpendicular weld plane stresses are

$$\sigma_{\perp} = \tau_{\perp} = \frac{t_x}{\sqrt{2}} = 4.58\ MPa \tag{5.83}$$

Calculated stresses can be represented like Figure 5-42.

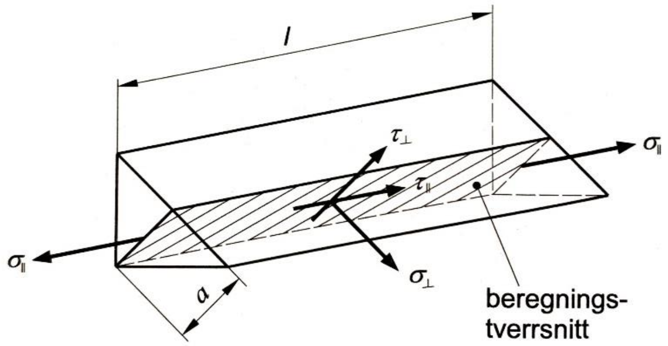


Figure 5-42 Different stresses acting on the throat plane [32]

The applied stress can be written as

$$\sigma_{equivalent} = \sqrt{\sigma_{\perp}^2 + 3 \times \tau_{\perp}^2 + 3 \times \tau_{\parallel}^2} \tag{5.84}$$

Giving us

$$\sigma_{equivalent} = 18.1\ MPa \tag{5.85}$$

Allowable stress is defined as

$$\sigma_{allowed} = \frac{R_e}{\gamma_m} = \frac{355}{1.15} = 309\ MPa \tag{5.86}$$

Where γ_m is a material coefficient, typically 1.0-1.15 for steel. Continuing on, the safety factor can be found by

$$n = \frac{309}{18.1} = 17 \tag{5.87}$$

Which is within safety.

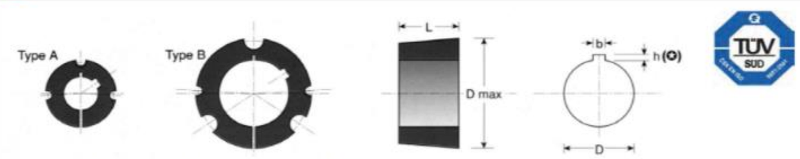
5.26 Taper lock selection

Table 5-21 shows specifications on some specific taper locks. Taper locks 3020-9W, with inner diameter of 40 mm and 2012-4F with inner diameter of 25 mm, are used for the drum and engine shaft, respectively. Looking at the drum taper lock, it can transfer 2712 Nm with a keyway. Without a keyway, only relying on clamping force, it can transfer between 480-600 Nm. By interpolating, a transmissible torque of 504 Nm for an OD of 40 mm. With 430 Nm load, safety factor will be 1.17. This seems a bit low but can act as a torque overload preventer. Safety factor with a keyway gives a safety factor of 6.3, which is higher than the shaft safety factor.

Table 5-21 Specifications on taper locks [37]

Taper pouzdra
Taper Bushes

TAPER



Příklad objednávky: Obj.číslo/ Part.No. **= TB 1610 - 25**

Upinací pouzdra Taper / Taper Bushes

Typ-Obj.č. Part.No.	Rozměry / Dimensions D od-do [mm]	D max [mm]	L [mm]	Typ Type	Up.šrouby/Clamp.Screws Typ/ Type	Klíč Wrench	Moment/ Torq.Nm	Moment/Torque D= [mm]	bez pera [Nm]	s perem [Nm]	Svěrná síla Clamp.force [N]	Hmotnost Weight [kg]
2012	**14-50**	70,0	31,8		7/16x7/8	6	31 (25)*	24	165	808	11500	0,75
2017	**14-50**	70,0	44,5	A	7/16x7/8	6	31 (25)*	38	310	808	14400	0,90
	D = 14, 15, 16, 18, 19, 20, 22, 24, 25, 28, 30, 32, 35, 38, 40, 42, 45, 48, 50							50	420		16700	
2517	**16-60**	85,5	44,5		1/2x1	6	49 (35)*	38	380	1311	17000	1,10
2525	**16-60**	85,5	63,5	A	1/2x1	6	49 (35)*	48	510	1311	19200	2,25
	D = 19, 20, 22, 24, 25, 28, 30, 32, 35, 38, 40, 42, 45, 48, 50, 55, 60, 65							60	690		22300	
								38	480		23900	
3020	**25-75**	108,0	50,8		5/8x1 1/4	8	92 (70)*	48	600	2712	26100	2,61
3030	**25-75**	108,0	76,2	A	5/8x1 1/4	8	92 (70)*	60	900	2712	31500	3,80
	D = 25, 28, 30, 32, 35, 38, 40, 42, 45, 48, 50, 55, 60, 65, 70, 75							75	1300		34500	

5.27 Final conclusion on the belt system

After a discussion with HMR, a conclusion on belt and pulley sizes has been made. In many cases, theory can approximate the behaviour of a system in real life. As for this system, a lot of approximations have been made (especially in the early stage), which can cause a deviation from the actual behaviour. At this stage, the design will carry on in the real world, using Figure 5-43 as a foundation. A prototype will be made and tested. For testing, requirements should be carried out for a successful test.

Rubix could not deliver the 14M pulleys, due to lack of material in the global market. Even though the 8M with 85 mm belt width could be ordered, HMR thought it was "too chunky". 8M belt with 50 mm belt width was then decided upon and ordered, even though it has a lower power rating than required. Decision has been made, and will work as an experiment, hoping that the low belt speed won't be such a big "catch" as experienced through the theoretical work.

To be noted: recommended maximum belt tension for static condition was calculated to be 1800 N per belt. As Equation 5.30 is a function of belt speed (in the numerator), the low belt speed will give a high pre tensional force. The engine may not handle this. Engine won't necessarily break due to high loads, but rather fatigue. A solution to this will be to start at a low belt tension and increase it for testing purposes, which is also an argument to proceed with the belt solution.

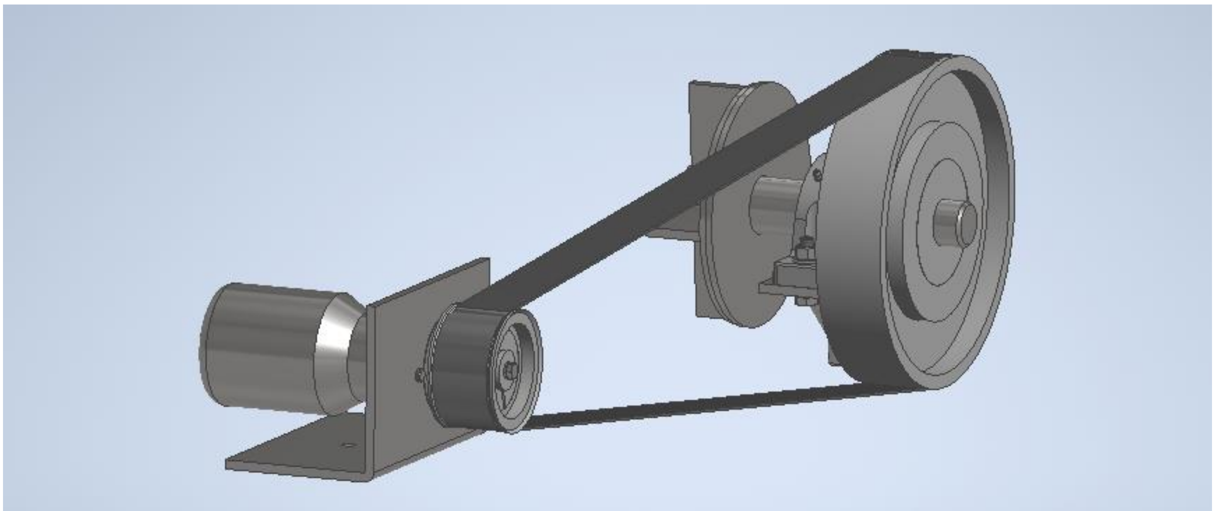


Figure 5-43 Concluded belt setup

6. Design of belt tensioner

6.1 Belt tensioner

For this system, the two thought solutions for a belt tensioner was either with a tension idler, or by moving the engine. Together with HMR, adjustment by moving the engine was decided on. As for this, the belt mounting consists of two phases. First when C-C distance is small enough to mount a new belt, and second when the C-C distance is increased, meeting the desired belt tension. For this to happen, there should be a mechanism that increases and decreases the C-C distances, as well as holding the C-C distance under loads. Before starting the design, several challenges and needs should be met.

- C-C distance interval, providing easy mounting and sufficient tension
- Ability to change C-C distance in a linear direction, maintaining a straight belt track
    - Correct alignment between engine bracket and drum frame
- Ability to fasten the engine bracket so it doesn't slide (or lift itself) while under load
- Ability to adjust the angle of the belt track, if necessary
- Having sufficient distance between the drum itself and engine mount
- Simple in use and to manufacture

Belt tensioner will be divided into two main components, sliding plate and engine bracket. Sliding plate is stationary, responsible for the parallelism in the linear motion of the engine bracket relative to the frame. Engine bracket will hold the engine, having the opportunity to slide on the sliding plate, changing the C-C distance.

With some consulting from HMR based on experience and knowledge, a design was worked out through some iteration and sketching. In addition, the design is not meant to be used in a mass production, but rather for testing purposes. Chapter 6.2 shows a quick simulation verifying that the design should hold. A more thorough work with analysis and calculations would be necessary if this was the final design on a commercial product.

Note: Yellow markings on the following figures represents the welds.

6.1.1 Sliding plate

Sliding plate is built up by three components. Main base with slotted holes, flat bar with threads, and an angle bar. Main base provides a sliding surface (as well as friction surface) between the base and engine bracket and its slotted holes as a subordinate guide for the engine bracket. Base plate also has a small bend, resulting in a vertical plate surface, where the engine bracket will rest against. That will be the main guide for the engine bracket. Flat bar with threads will be responsible for the adjustment screw, separating the engine rotational axis from the drum's rotational axis. Angle bar acts as a sideways adjuster for the engine bracket, as well as steering when mounting the engine bracket. The horizontal part of the angle bar also offers stability to the angle bar itself, because of the weld attaching it to the flat bar. Figure 6-1 shows the final design of the sliding plate.

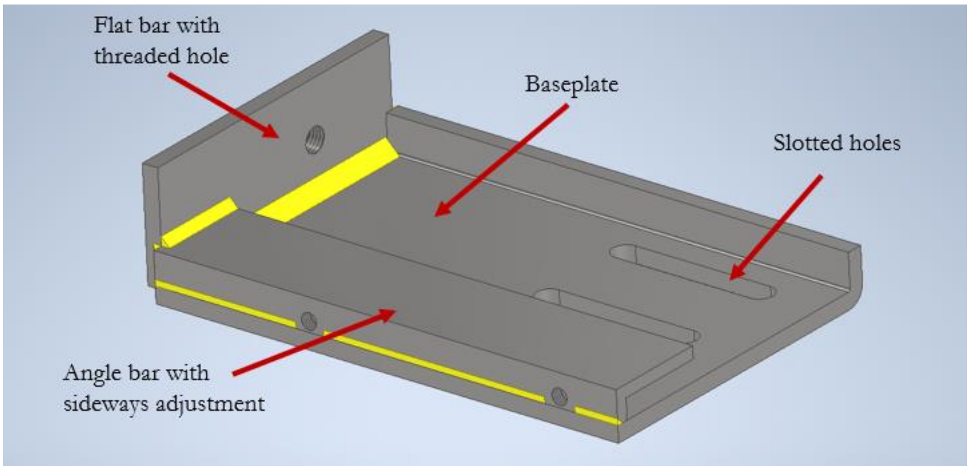


Figure 6-1 Sliding plate for the belt tensioner mechanism

6.1.2 Engine bracket

Bracket has the engine mounted on it (vertical plane), sliding on the sliding plate. The two holes on the horizontal plane will be responsible for the location of the fastening bolts and nuts. These are located mostly for easy access purposes. Engine will be fastened with two bolts, threaded into the plate. The resistance plate will transfer the force from the adjustment screw into sliding motion on the engine bracket. Figure 6-2 shows the final design of the engine backet.

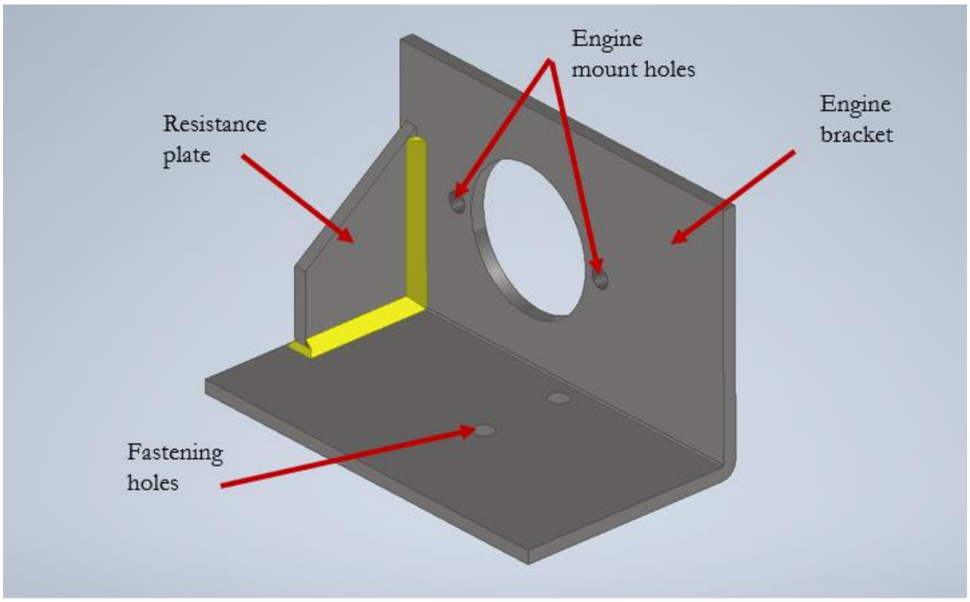


Figure 6-2 Engine bracket for the belt tensioner mechanism

6.1.3 Belt tensioner final design

When the sliding plate has been welded onto the frame, the engine bracket can then be slided into it. The implementation of the tensioner mechanism is through welding the sliding plate onto the frame. Mounting procedure is important since the sliding plate needs to be parallel with the frame, where larger angle deviations can cause unnecessary installation work and/or belt wear. Details about positioning is described in the technical drawings, located in Appendix B.
Figure 6-3 shows the thought sliding plate and engine bracket, engaged, where the sliding plate has been welded onto the frame.

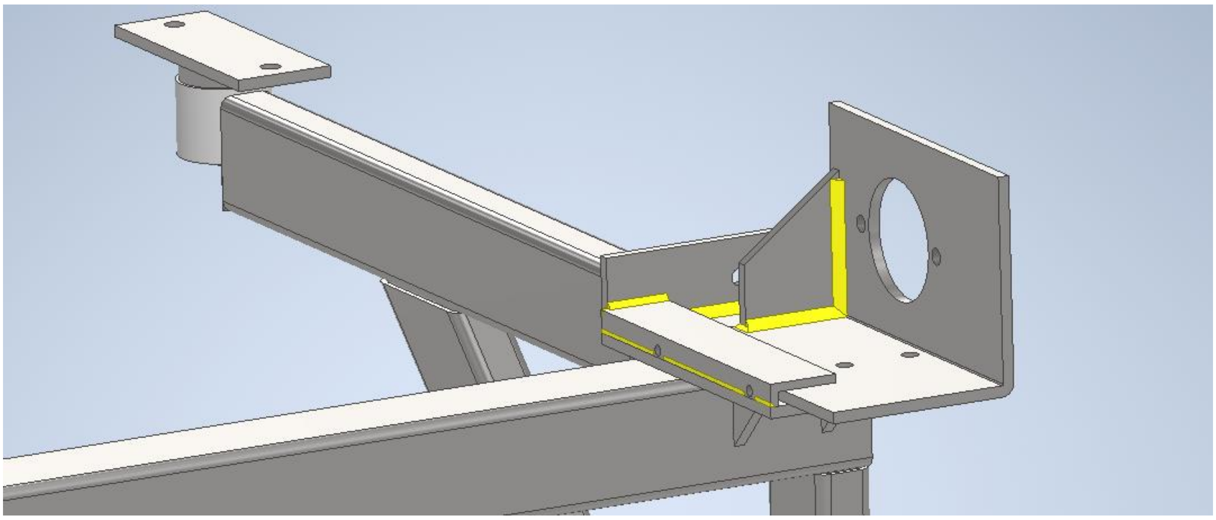


Figure 6-3 Tensioning mechanism located on the frame

Following figures, 6-4, 6-5 and 6-6 shows the implementation of the tensioner mechanism into the main drum and frame assembly.

Note: Weld is not visible on following figures. See Appendix B for technical drawings.

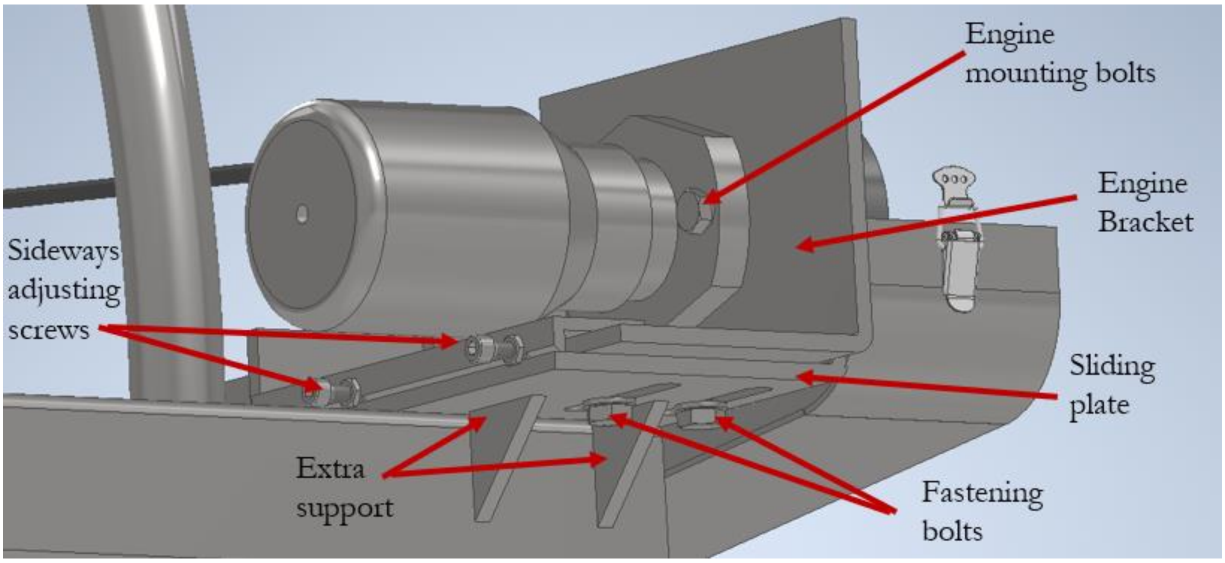


Figure 6-4 Implementation of tensioner mechanism

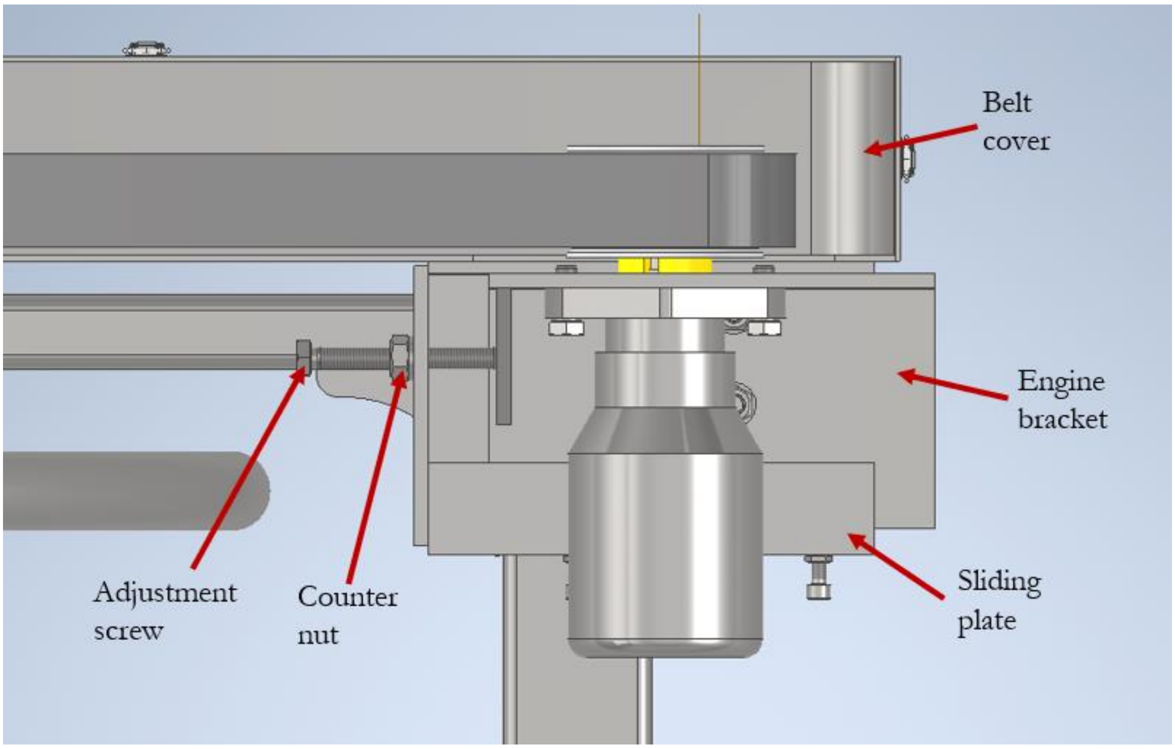


Figure 6-5 Implementation of tensioner mechanism

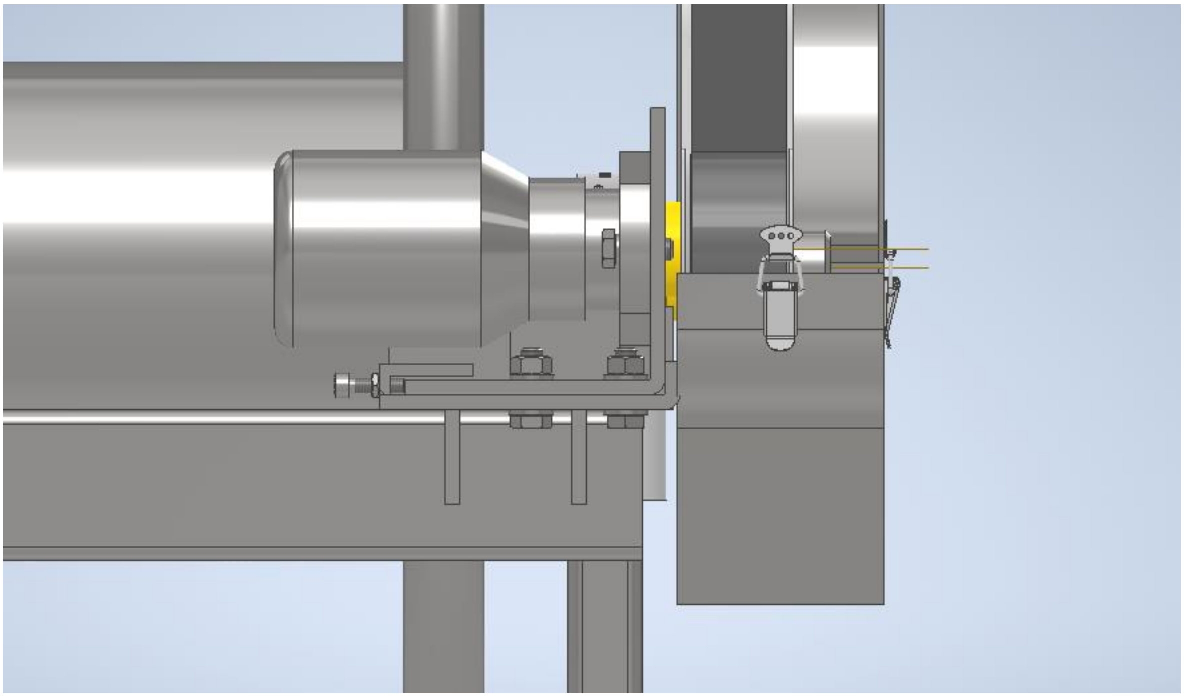


Figure 6-6 Implementation of tensioner mechanism

6.2 Belt tensioner analysis

After finishing the belt tensioner mechanism design, a simple simulation was executed. Same load scenario for the engine analysis, with the following boundary conditions:

- Engine bracket and sliding plate could separate, meaning the bolt and nut where simulated to keep the bracket and sliding plate together
- Engine bracket and engine could separate, meaning the engine mount bolts where simulated to keep the bracket and engine together
- Sliding plate had its bottom plane fixed to any movement
- Engine bracket could lift off from, but not slide relative to the sliding plate
- Clear red area defines stresses above 80 MPa

Figure 6-7, 6-8, and 6-9 shows the loads acting and some of the stresses that where more interesting than others.

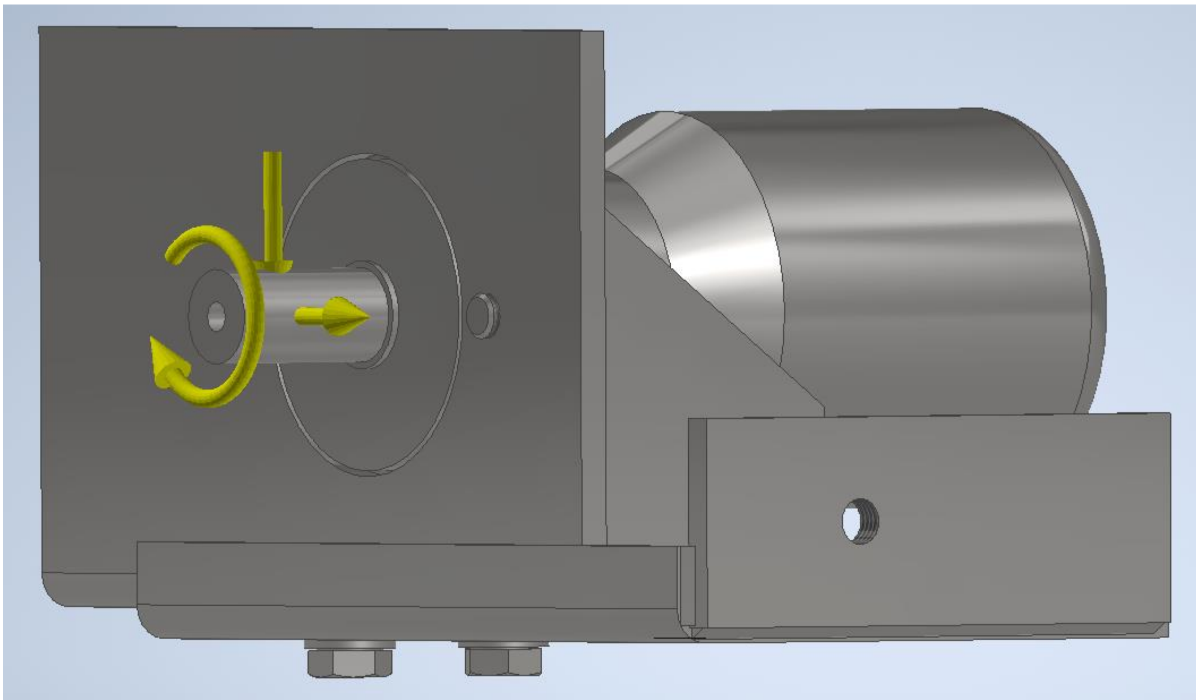


Figure 6-7 Loads acting on the shaft

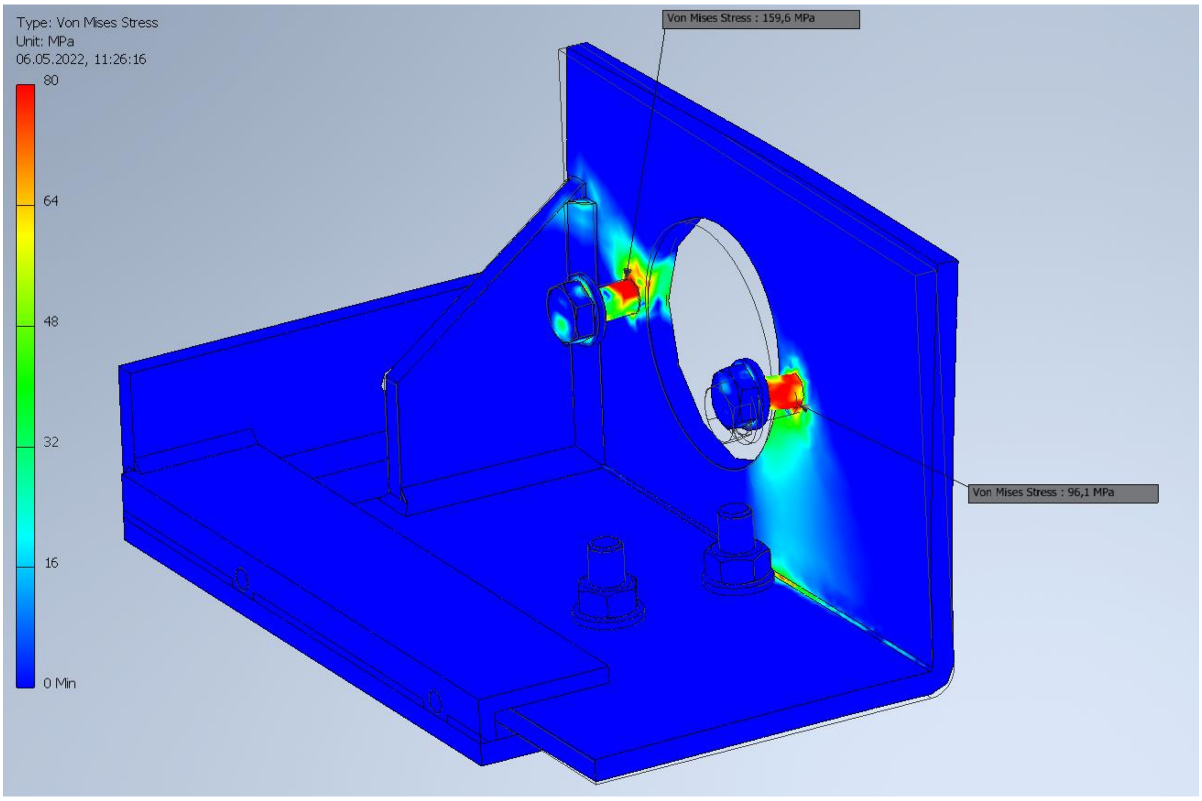


Figure 6-8 FEA of engine bolts

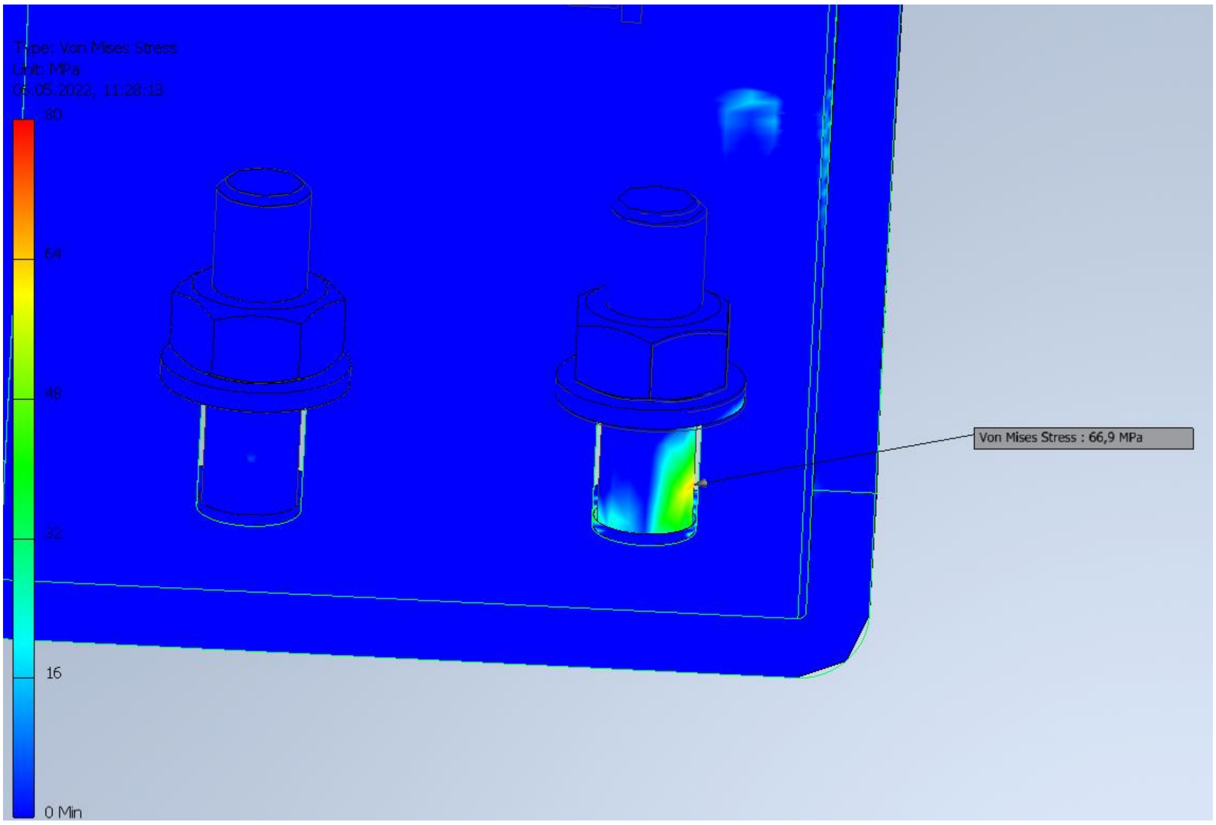


Figure 6-9 FEA of engine bracket fastening bolts

6.3 Final belt transmission design with belt cover

After a review on the taper lock and pulley mounting for the engine shaft, where the engine shaft needed an extension, a new design was worked out. Figure 6-10 illustrates the transparency of the pulley and taper lock, showing the spacer fitted to the keyway. An additional flange was made, tightened to the engine shaft, ensuring the taper lock and pulley being adequately fastened to the shaft. This safety measure also compensates for a deviation in the engine shaft length (shorter shaft) or an extended spacer.

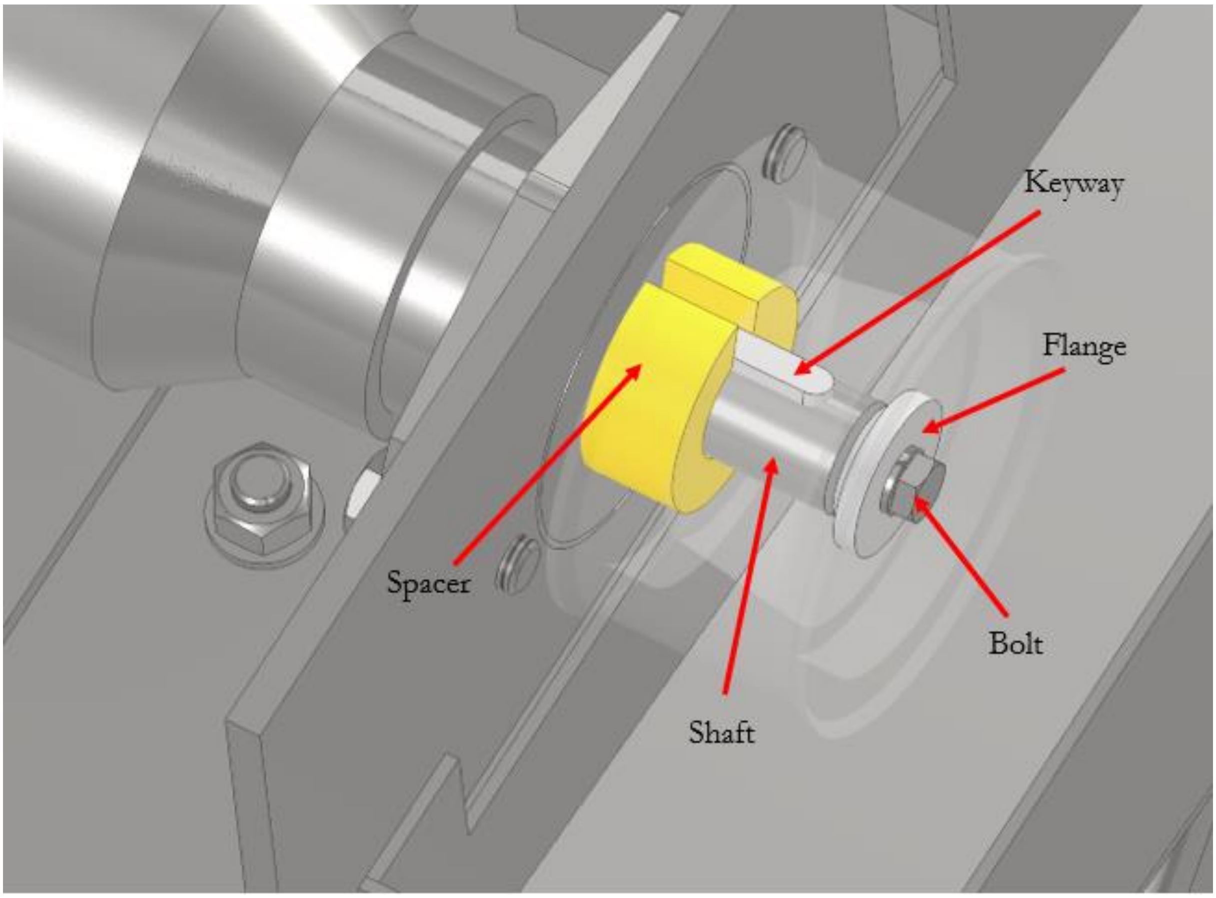


Figure 6-10 Taper lock mounting for smaller pulley and engine shaft

After finalizing the tensioner mechanism, the whole assembly was put together, making up the final design. Belt cover was designed as a simple cover with hinges. Figure 6-11 shows an example of how the EHS-scenario could be solved, by adding belt covers. Spacers was a necessity to obtain correct mounting of the pulleys and taper locks. Figure 6-12 shows the foundation for the prototype going into manufacturing.

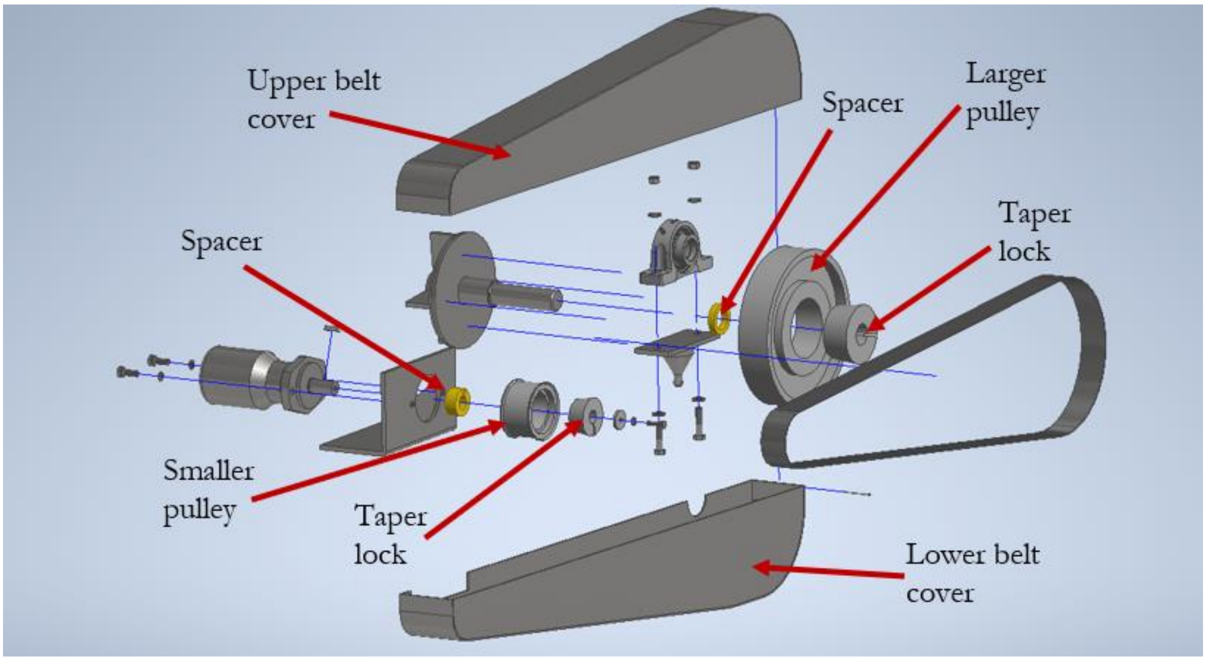


Figure 6-11 Final belt and pulley assembly

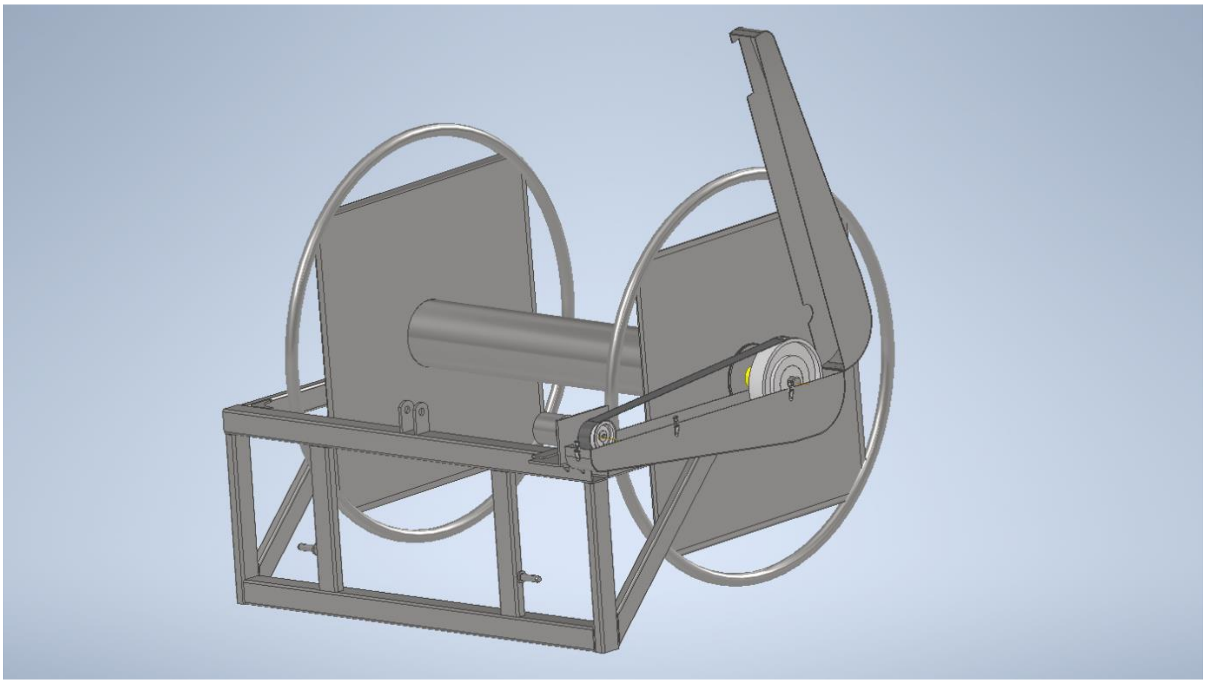


Figure 6-12 Full assembly and final design, foundation for the prototype

7. Production and testing

Table 7-1 shows the specifications of the belt system ordered and used for the prototype.

Table 7-1 Overview of the chosen system specifications

Specs			**8M belt specs**
Ratio			2.8
Z1			40
Z2			112
RPM engine			14
Pitch Diameter smallest			101.86 (with flange)
Weight smallest [kg]			1.34
Pitch Diameter largest [mm]			285.21 (without flange)
Weight largest [kg]			13.07
C-C [mm]			891.3
Material			Steel
Belt width [mm]			50
Power rating [W] 5RPM			219.2
Installation specs			**8M belt configuration**
S – Belt span [mm]			890.57
δ – Mid-belt deflection [mm]			17.81
F_t – Installation tension [N]			[75, 180]
T_{st} – Max Static tension pr. belt [N]			1800
Belt type			**Length** [mm]
HTD-2400-8M-50			2400
Pulley type	**Taper bushing**	**Type**	**ID** [mm]
40-8M-50	2012	4F	25
112-8M-50	3020	9W	40

Pictures from the production, prototype and testing, are all found in the Appendix A.

7.1 Flow of technical drawings

Appendix B shows an overview of the technical drawings to be used for prototyping. HMR has their own way to arrange the different assembly drawings and single-part drawings. Every assembly is named AX00, where A stands for "assembly", and X is the assembly number. Every part is named PX00, PX01, PX03 and so on. P stands for "part", X for the assembly number, and 01, 02, 03 etc. for part number within the specific assembly. After every assembly drawing, the associated part drawings are listed. Standardized parts are not drawn. Some assemblies can have other assemblies within it, as well as parts that originates from other assemblies. In light of using the same part names and descriptions, it's less chance for confusion. The goal will be to have a complete overview which is easy to follow, not just for the engineer, but also for the mechanic making the parts, as well as assembling the prototype.

7.2 Prototype testing

Since the thesis is written in Stavanger and production takes place in Voss, frequent communication and meetings makes sure there are no misinterpretations regarding the parts, assemblies, and the prototype itself. Daily updates from HMR have provided insight, with constructive opinions and reviews on the design work.

A running test was done and successful, where the behaviour of the belt (tracking, slack, alignment etc.), engine and engine mount, and the whole prototype was investigated. Testing was executed without any external loads, looking at how well the prototype was acting. For future testing, where external loads are added, testing criterions from Chapter 7.4 can be implemented.

For the prototype testing, drum rotated with a constant of ~8.5 RPM, which is within the anticipated RPM and indicated to be close to the actual RPM. Belt tension was based on the mid-belt deflection, which came to ~10 mm, compared to the theoretical max deflection of 17.81 mm. Adjustment screw came to less than 20 Nm when tightening. Belt had little to no tracking but was located more to the one side than the other at both directions. Adjustments were made so that the belt would run with minimum tracking.

The engaging of the belt drive went smooth and without any lag or abrupt motion. That is an improvement from the old chain and sprocket system. As the engine starts turning, the belt will tension up before transferring movement, making the acceleration smooth. In addition, most of the noise arose from the engine, where the belt ran very quiet. In the end both the engine and belt were running fine. This may not be the case under actual loads.

7.3 Testing Criteria

As the prototype is tested, criterions can be a way to determine if a test has been successful or not, or at least avoid scenarios that can lead to an unsuccessful test. Matrix in Table 7-2 can be used as a check list when running a test, either with or without loads. After all, a test with loads acting is more interesting than without.

Table 7-2 Criterions and check list for testing

Area of interest	OK	Not OK	Comments	If not OK
Belt is not slipping or ratcheting				Check tension and alignment
Belt should not be self-tensioning				Increase tension if possible
Angular and parallel alignment				Adjust or define possible design changes
Signs of wear				Adjust or define possible design changes
Motor condition				
Tensioning mechanism				Define problem and possible solutions
Any mechanical issues like vibration, deflection etc.				Define problem and possible solutions
Other…				

Data can be gathered to improve and get an estimate towards a more optimal system installation. There are two parameters that should be obtained from the test, which is the actual rotations of the drum, and the torque applied to the tension adjustment screw. Table 7-3 can be used to track different tests with different belt adjustment scenarios, as well as RPMs.

Table 7-3 Gathering data

	RPM	**Torque on adjusting screw** [Nm]
Test 1		
Comment		
Test 2		
Comment		
Test 3		
Comment		
Test 4		
Comment		
Test 5		
Comment		
Test 6		
Comment		

8. Discussion and conclusion

8.1 Deviations in required power

As the thesis has a very theoretical approach, especially the loads, it is uncertain how accurate the load estimate is. Even though they can be very approximate, several small errors can result in a larger deviation in the result. Coefficient of friction can vary depending on the field surface. Rotations of the drum has also been an estimate which is based on common sense, and not on investigation. The way the hose is laying, either in a straight path or e.g., a snake formation, may influence the required power to wind it onto the drum. Some fields are elevated relative to the horizontal, which may increase the requirements, if the hose where to be dragged uphill. All these factors may not have a big influence on its own, but rather resulting in a bigger deviation when they are combined.

Since the manure hose is the main resistive force that the systems need to overcome, it's hard to tell if the highest tensional load occur at the initial spooling phase. It makes sense that the tension decreases as the hose gets winded up, but at the same time, the radius of the drum increases. An increased radius leads to a larger moment arm. It could be further investigated whether the torque has a peak or not after some amount of drum rotations. For this thesis, it has been considered highest in the initial phase.

To solve these uncertainties, the estimated hand calculations could be replaced with actual field tests, particularly by measuring hose tension. By implementing a measuring device (e.g., a load cell, spring device), which reads axial loads in either compression or tension, an accurate reading of the acting tensional load would be obtained. Several field tests could be completed, with different elevations, hose layout and so on. Adding a torque cell on the engine output shaft, would further simplify, but requires a bit more work implementing it. It's hard to tell how close the theoretical approach is, compared to the real-life behaviour.

8.2 Deviations in drum shaft hand calculations

When comparing the hand calculations up against the FEA done with Inventor, it was observed a deviation of ~90 MPa, which is a lot. If the hand calculations would have been done without the correction factors, the deviation would equal 0.4 MPa, which is incredibly close to the results from the FEA. More thorough research on understanding the FEA stresses is crucial. There are two thought scenarios for this to happen. First will be the main setup of the forces and boundary conditions for the simulation. Having even a single boundary condition wrong can lead to terribly wrong values. Hopefully, this is not the case. Second could be the mesh setup. When simulating, the mesh is set to a specific value, which makes the simulation comfortable in terms of waiting time. Mesh is typically increased (by decreasing the size of the partial elements) at more interesting and crucial locations, like a fillet. It could be that the mesh should have been finer with smaller elements, resulting in a more correct result.

After all, a more thorough calculation review could have been followed up with. It is not said that the hand calculations could have had some errors in it.

8.3 Safety factor

Safety factor was set to be between 3 to 4. Originally, the safety factor should be decided over in the very start, which would be followed through the whole design process. When deciding upon a decent safety factor, an investigation/survey should be led out, defining the majority of who's buying the product. It could make sense that the safety factors for private use is higher than for industrial use, since in private, some (or many) customers doesn't read the manual, or maybe exposes the product for higher loads than it can handle. To avoid consequences like injuries and accidents, a higher safety factor can be implemented. In the industry, there are more regulations and strict rules for tools and their purposes, which may result in a safety factor lower than the private. A good an easily readable "Safety and use"-manual is a good way to ensure that the customer is within the boundaries of the product and should be implemented in the future.

8.4 Better research on existing solutions on the market

More research on existing drum setups and systems on the market would be preferrable before starting the project. Having an overview over existing solutions, either a drum setup or a similar mechanical principle, could have given more inspiration on how to solve and propose a new design. Knowing how others solve problems and maybe their product weakness, a new product can be made which eliminates that weakness, while keeping up the quality. A product market will most likely have some issues, weaknesses, or any other challenge, that can be identified and worked out in a new prototype.

8.5 Engine analysis and belt tension force

Generally, the system has been analysed with the theoretical loads, but also max recommended static belt tension of 1800 N per belt. If the max tension where to be used, 3600 N would act on the engine shaft, which most likely would lead to fatigue in either the shaft, or internal parts. It's hard to determine how the forces will influence the engine house and internal parts. For that, complementary bearings on either side of the smaller pulley could be practiced, lowering the reaction forces in the engine. As the system is designed with an adjustable engine bracket located as close to the smaller pulley as possible, tension force can be adjusted to a sufficient value and simultaneously decreasing the bending moment due to horizontal pulley reaction forces.

8.6 Low belt speed and formula range

A challenge in the theoretical aspect of the belt selection has been the low belt speed. For both V belt and synchronous belt, the low belt speed has given some challenges, especially on the power ratings. As recalled, a synchronous belt was picked, but with a lower power rating than required. Further on, physical testing would rather determine if synchronous was a wrong decision, or if the belt chosen is feasible for its intended use. In addition, the required belt life span is long in terms of years, but not in terms of working hours, assuming that it will last out the entire 10 years.

8.7 Torque overload mechanism

In the proposed design, keyway is used on the smaller pulley, ensuring that it will transfer the torque required. Simultaneously, the larger pulley is mounted without a keyway, where the torque transmitted only relies on the clamping force from the taper lock. A solution like that is a simple way to implement a torque overload mechanism to the belt system, without introducing more components. Recalling the safety factor of 1.17 between the permissible torque relative to the torque acting, which is lower than the other safety factors, the taper lock will slip before anything breaks (theoretically). Overload scenario is not something that is meant to happen but must be considered. Or at least having a fail-safe design, which causes rest of the equipment (or even the environment and people around) minimal to no harm. Better to have a slipping belt pulley than an unattached drum due to shaft failure.

8.8 Cost and product value

Cost comparison is based on the power transmission only, looking at the old and new design separately. Table 8-1 shows a comparison between the old and new design total cost, where the new design is ~5000,- more expensive. Yellow markings indicate the total costs for production, components and the total power transmission cost. Unfortunately, there was no specific plan on separating into specific types of cost (for each part produced etc.), so it will be the sum of parts that will be compared. Looking at the new part, the production hours contain all the hours spent on manufacturing and assembling, but with no material cost. Components are all purchased parts and the estimated material cost is based on today's prices. Looking at the old system, some of the parts may not be included, e.g., drum shaft. The drum shaft is most likely enclosed to the main drum cost, which is not represented in Table 8-1 (left side). In addition, the old system is based on a mean part cost, where there is more production volume hence making the produced parts cheaper than producing a single one.

Table 8-1 Cost comparison between the new and old design

Production (including material cost)	**Old system [NOK]**		**Production (excluding material cost)**	**New system [NOK]**
Engine sprocket	557.8,-		Production hours	4485,-
Engine bracket	1277.7,-			
Bearing mount	367.0			
Engine bracket shaft	328.1			
Total production and material cost	2530.6,-		Total production cost	4485,-
Components	**Old system [NOK]**		**Components**	**New system [NOK]**
Hydraulic engine	2140		Hydraulic engine	2140,-
Bearings	1140		Belt drive components	2592.5,-
			Bearings	1323.5,-
Total component cost	3280		Total component cost	6056,-
			Estimated material cost	200,-
Total power transmission cost	~5811,-		Total power transmission cost	10 741,-

Rather than looking at the direct unit cost, it may be more relevant to look at the bigger picture of it. That would be how purchased parts affect the production and assembly line at HMR, does the cost change as the production volume increase, time reduced with increased production volume, time reduced/increased with new design, and does the new design make up for the extra cost etc.

By increasing the production volume, the cost per product will decrease. Some reasons can be that ordering bigger batches is often cheaper, both production and assembly speed up due to more experience, and some parts can be welded with the aid of a welding jig. After some time, the cost will decrease to a certain value. HMR also mentioned that the assembly of the new prototype took less time than before, as well as parts to produce. This is time (and money) saved, which can be spent at something or somewhere else. It's also worth mentioning that the first time producing a new part or product will always take more time than the next time producing it.

In addition, the belt cover hasn't been taken into account, making the total cost of the new design a bit higher. Hopefully, the extra cost observed in the comparison (assuming it's close to right) will decrease as the volume increases. If it stays more expensive, it's rather a matter of looking at the value of the new design. Has it eliminated previous challenges in both production and usability? Does the customer appreciate the new design more than the old design? To determine whether the design is worth it comes down to several factors than just cost, which is something HMR needs to consider if they plan to switch designs.

8.9 Too much workload for one person?

Reviewing the past couple of months, errors has been identified due to some lack of knowledge within certain subjects, or solutions has been found quicker than preferred. After all, that is the process of design work. It should not be perfect at the first try. Sometimes decisions must be made, even if it doesn't feel like the best decision. Learning the hard way can often be the best way to learn, as long as it doesn't occur at every work phase.

Throughout the thesis, it was realized that there has been more work than originally thought. Working hours has not been exaggerated in specific periods, but evenly distributed over five months. Having continuous work for five months, where it's mostly theoretical work, the thesis and its content expands at a fast pace. Simultaneously, it was important that the whole process was thought through and documented. Especially for the design work. It has been hard to balance between quantity and quality. A lot of sketching and calculations has been excluded from the thesis, as a thought of keeping the most relevant content towards a new design. Even though the workload has given a huge quantity of information, it has been done with the mindset of thinking quality, or at least common sense.

All in all, having several people on the project vs. having a single person is more a give and take scenario. It may result in better quality, but less overall knowledge per person. You win some, you lose some.

8.10 Further Development

If the prototype looks promising and has the potential to be further developed, a revision of it can identify further possible improvements. Following sub-chapters would identify some key points to look at if the prototype got potential.

8.10.1 Belt Cover

For prototyping, belt cover has not been in focus, due to lack of time. If HMR decides to use a similar setup, a belt cover is necessary to cover the basic EHS of mechanical (moving) components. It's also preferable to know that the system is working and having identified possible improvements, before any cover design is worked out.

8.10.2 Testing with realistic loads

HMR impressed with how quickly they built the prototype, even in a hectic period. Afterwards, a test was executed, driving it in both directions. The test showed a functional power transmission system, with very little further adjustments. As the time ran out, no actual load testing was completed. For further testing, it would be exciting to see how the prototype would act under realistic loads. Both Table 7-3 and 7-4 can aid in executing a successful test, with data regarding actual RPM and belt tensioning. Knowing that a product is functional before selling it, is alpha omega.

8.10.3 Long term prototype testing

Having physical tests are an effective way to determine how well the design is working. Depending on the test, it may not serve actual feedback on its life span and useability for the customer. A way to actually get relevant feedback, would be to deliver or sell the prototype at a lower price than the original. In that way, an external unbiased view of the prototype can give feedback along its service life, as well as using it for all kinds of scenarios. It is also important to know at what extent it may fail, and why.

8.10.4 Challenges within the production stage

As an engineer, knowing how the production process takes place is a vital knowledge. Having an ongoing communication with the production crew is an easy way to avoid complications of producing non-feasible parts and products. It's essential that each part is revisited and looked at, deciding whether it leads to a good manufacturing practice or should be changed.

As an example, welding makes it easy to join the several parts, as for the engine bracket and sliding plate. As a consequence, the assembly experiences a lot of heat which may change the structure of the metal, but also the geometry and required tolerances between working parts. It should then be investigated if the setup works as it is, or if the welding has an unwanted effect on the performance. If the setup works fine, the production of it should be looked into. Are the parts hard to manufacture without error? Is the preparational work very time consuming for welding? Can a weld jig decrease the time consumed at the welding process? Or even, is there another solution besides welding?

Parts should be easy to understand, make, assemble and replace. That is important for the engineer and the manufacturing worker, but also the customer using the product. It should be aimed towards the customers (farmer in this case) opportunity to troubleshoot and maintain.

8.10.5 Investigate other belt types

Through the design process, more knowledge about belt types has been obtained. As the obtained design manual wasn't up to date, PowerGrip HTD belt was picked in front of e.g., PowerGrip GT3. It has been observed that PowerGrip GT3 can replace the HTD, with up to two times the rating. It is also said to be suited for performance upgrades on existing drives [38]. As the design process had many factors within it, there was less time than expected when it came to figuring out each step. If synchronous belt seems like a feasible solution, more investigation should be done regarding an optimum belt type. Generally, more knowledge about belt drives should be obtained before making decisions.

It should also be mentioned that V belt could be investigated more thoroughly than it was but seemed unlikely to be used when the design process encountered the low belt speed condition.

8.10.6 Pillow bearing and moment loads

As the drum shaft increased in diameter, the pillow bearings were replaced with new ones suitable for the additional load and size change. Originally, the pillow bearings aren't meant for moment loads. The moment loads may not be a determining factor on the life span of the bearings but has not been researched thoroughly either. The original design used pillow bearings and have had that design for many years without negative feedback. It was therefore feasible to scale it up and add new ones. If the prototype testing shows that there is a need for bearing replacement, new bearings should be picked out with care, taking the moment loads into considerations. It also depends on the reason for replacement. It may be other factors.

8.10.7 Belt tensioner

Since the belt tensioner mechanism was designed for testing purposes, a review and more thorough work should be done on it. At first, the engine bracket was thought to have more machined surfaces with steering pins and slots, making the linear movement as straight as possible. Downside would be having less adjustment opportunity in e.g., sideways adjustment. It should be considered having more machined surfaces if the rest of the system is straight. If there are some misalignments between the pulleys, it should be eliminated in a new design, or stay with a design that gives the opportunity to adjust it manually (as of now).

The tensioning mechanism served its purpose, and the tensioning screw worked as it should. An improvement regarding the screw, would be to add a ball and socket base which the screw would push on, minimizing the friction and wear on both screw and resistance plate. With less friction, the torque induced to the adjusting screw is more accurate towards the belt tension.

The design may be good enough as it is, even for an actual product. Downsides of the worked-out design would be all the material used to make a tensioning mechanism when a tensioning idler could have been used. In addition, a great amount of weld could be reduced by adding a tensioning idler. If the principle of separating the two rotation axes is maintained, other designs and solutions (like bolts instead of welds) could be investigated, or just making an improved version.

8.11 Conclusion on the belt design

The synchronous belt design turned out to be feasible and easy to work with. More or less, the whole power transmission setup was redesigned, where a belt cover was added in the design to keep up with the basic EHS. The belt had minimal signs of tracking and ran fine. Belt tensioner mechanism worked for its intended use. Even though the tensioner design seemed close to a working solution, it should be revised and improved. The largest pulley was mounted with clamping force only, having the opportunity to slip on the drum shaft for overload protection. In that way, an actual torque overload device was eliminated, which also acts as a safety measure for other components in the system.

Ratio between the engine and drum was reduced, but within the engine's capacity. How the engine will act under load has not been identified. Even though the engine meets the power capacity, internal reaction forces can be a factor that generates new challenges. Complementary bearings have been mentioned to reduce the reaction forces which may intensify the design work. Engine was not replaced for this design and should work as it is. For further suggestions, the engine can be replaced with minimal change in design.

All the hours on the design process paid off as there was little complications making the prototype in physical form. It was also observed that purchased parts decreased both the assembly time and the number of produced parts. In addition, a lot of welding was eliminated as a result of replacing the chain and sprocket setup. Cost comparison showed a more expensive solution for the new design, which is assumed to decrease as the production volume increases.

Did it eliminate challenges from the old design? Yes, it did. The new design came with many advantages. Need of lubrication was eliminated, slipping was eliminated, smooth engaging was obtained, easier to replace parts and more cost effective towards the user, as well as a belt cover was added. The design had no exaggerated cost and was made simple, with an aim towards low maintenance. Speaking of maintenance, it was a big improvement knowing that the belt can be changed while keeping the pulleys, which eliminates meshing problems. In addition, changing belt does not require the owner to detach the drum, as it would require if the chain where to be replaced.

Future work will come down to testing and improving the current design. That will also give an indication on how well it may work. As the thesis is very theoretical, it is important to figure out how comparable the theory and the real-life are. To declare its functionalities and further development, it needs to be tested for validating its use. The goal would be to get a proper prototype ready for sale.

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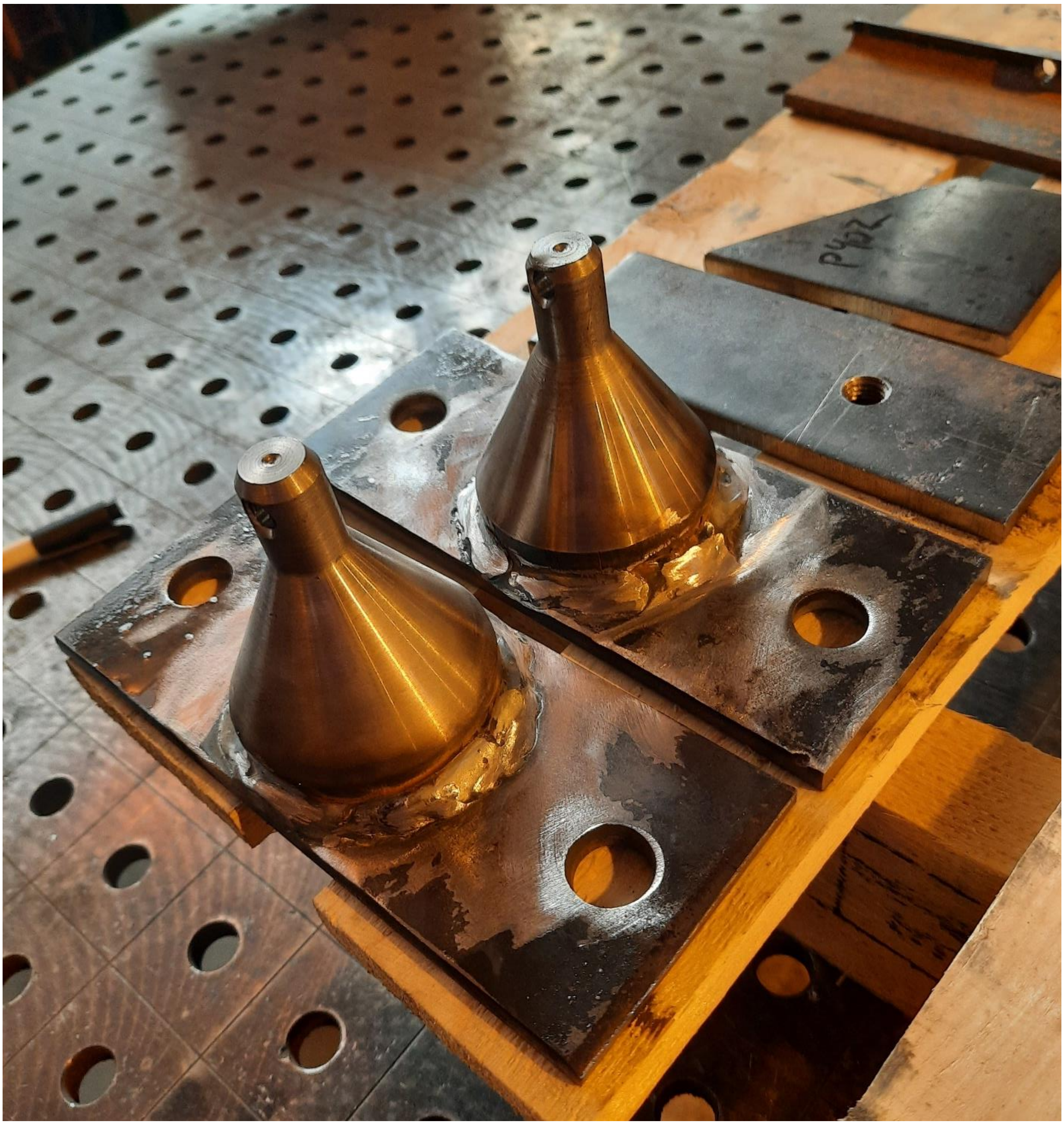
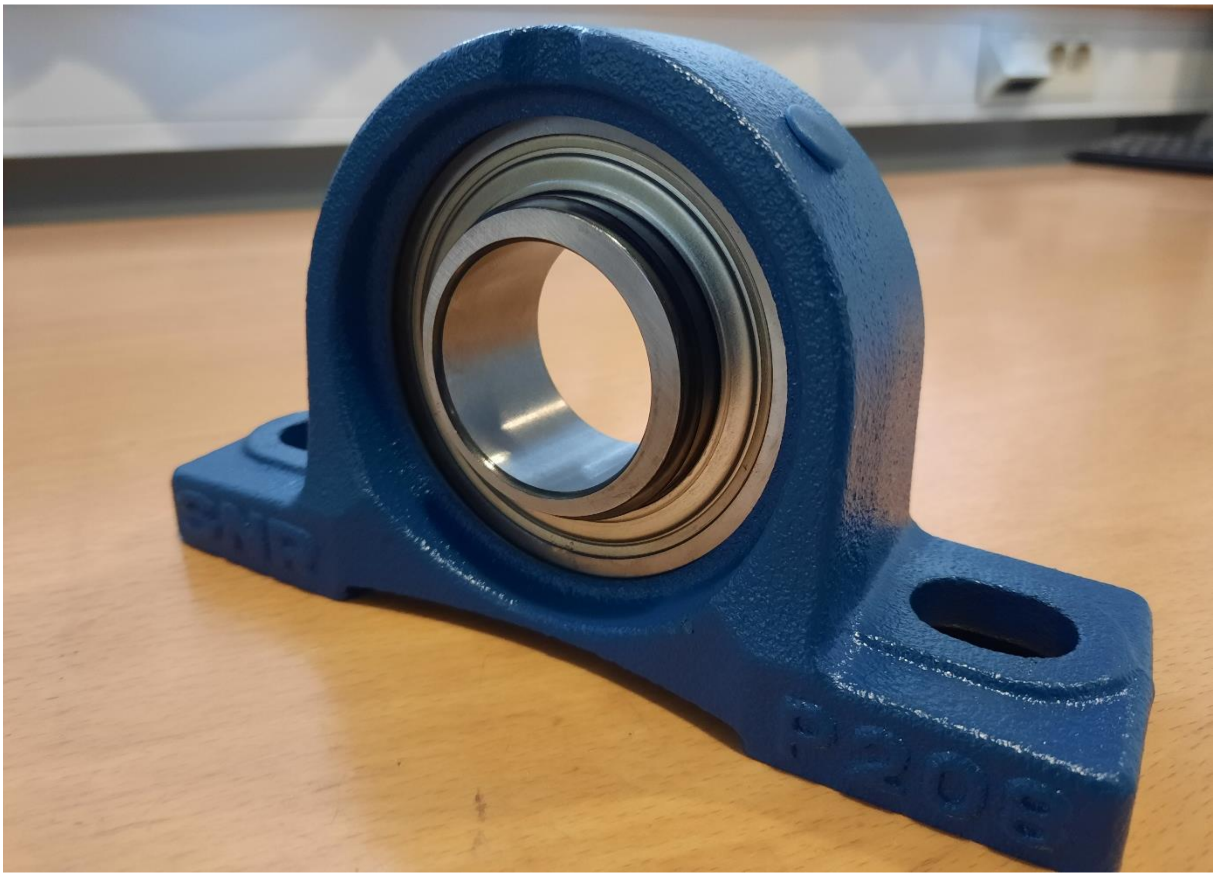
Appendix A – Pictures from the production

Appendix A includes an extended number of pictures from the production, as well as some of the prototype.

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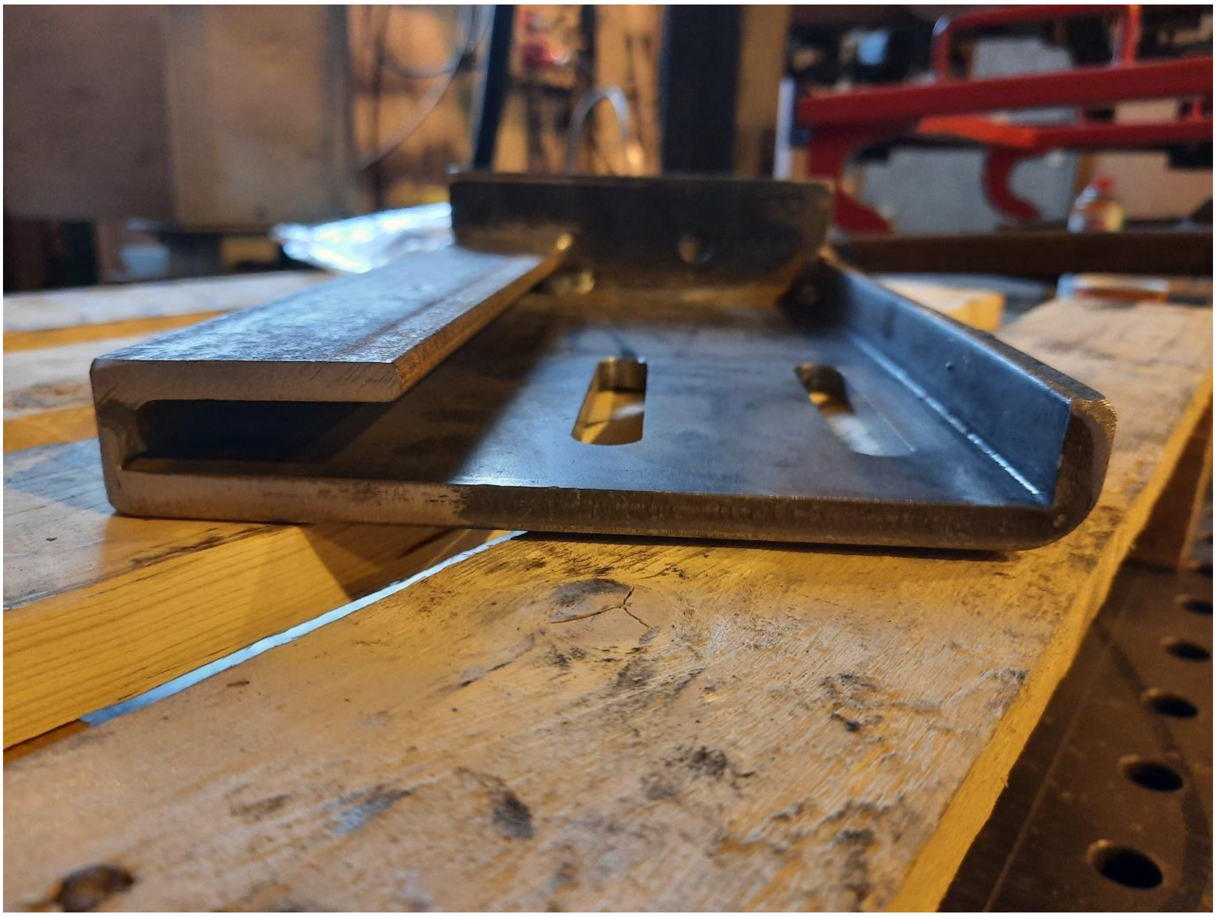
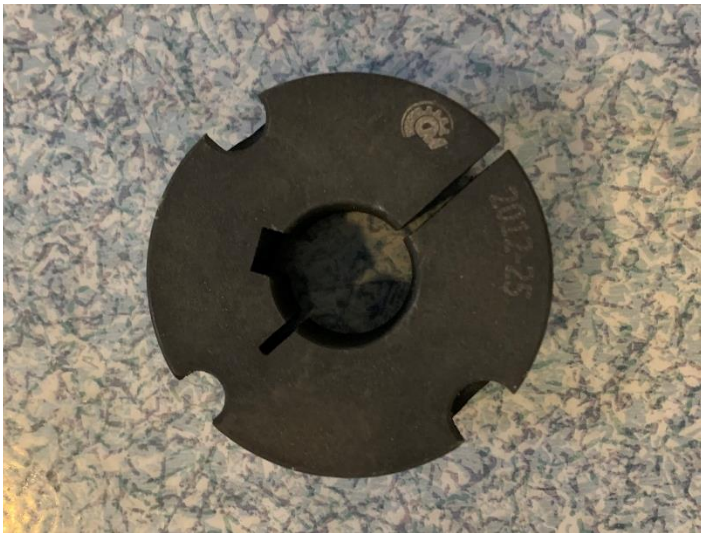
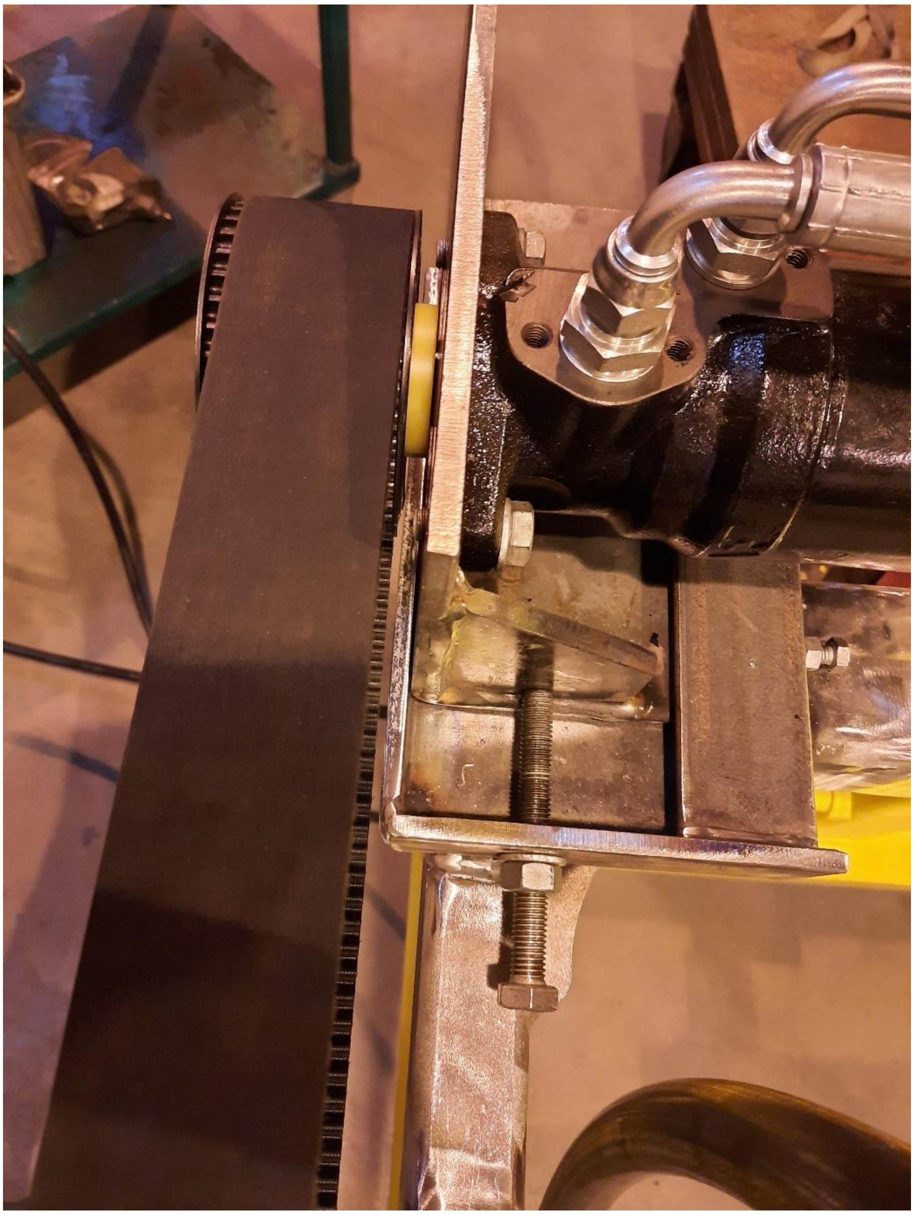
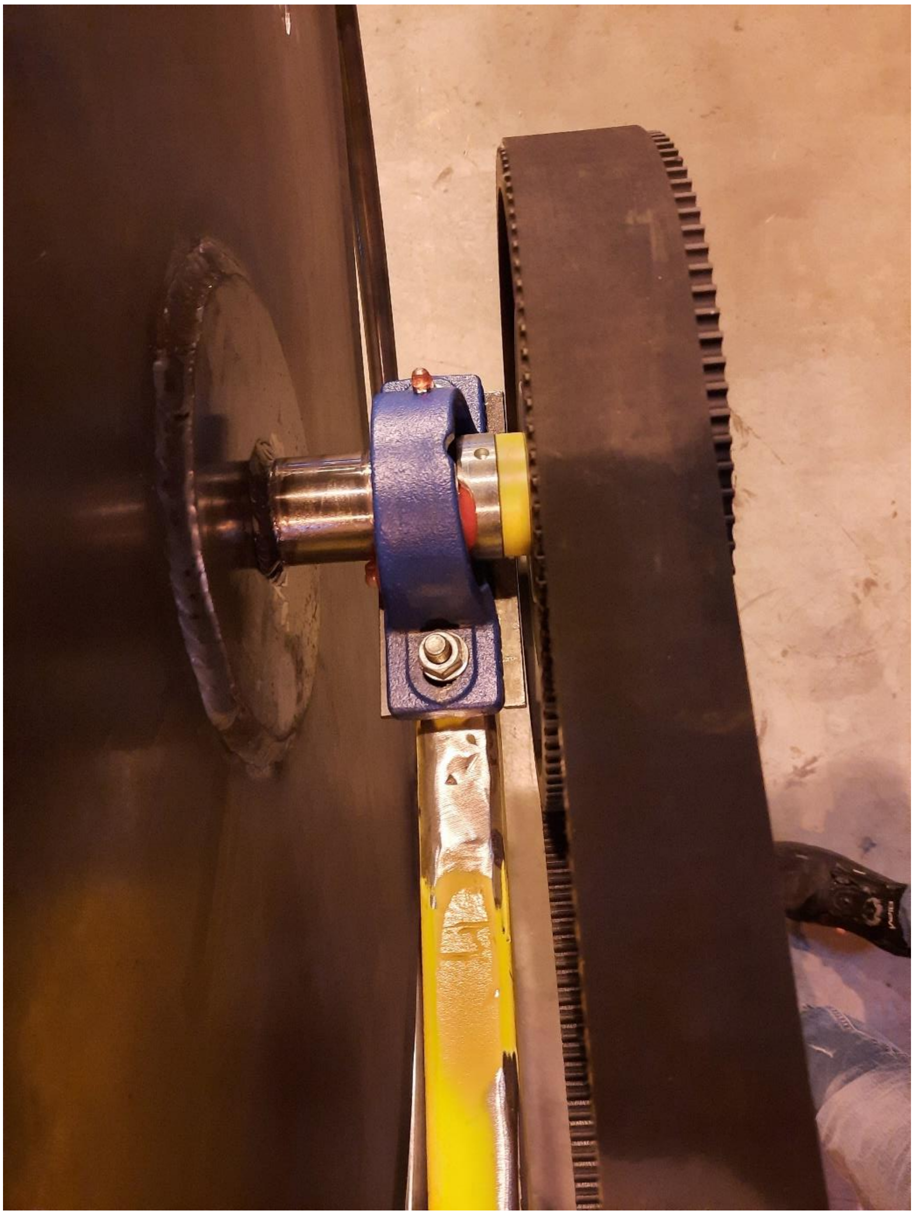


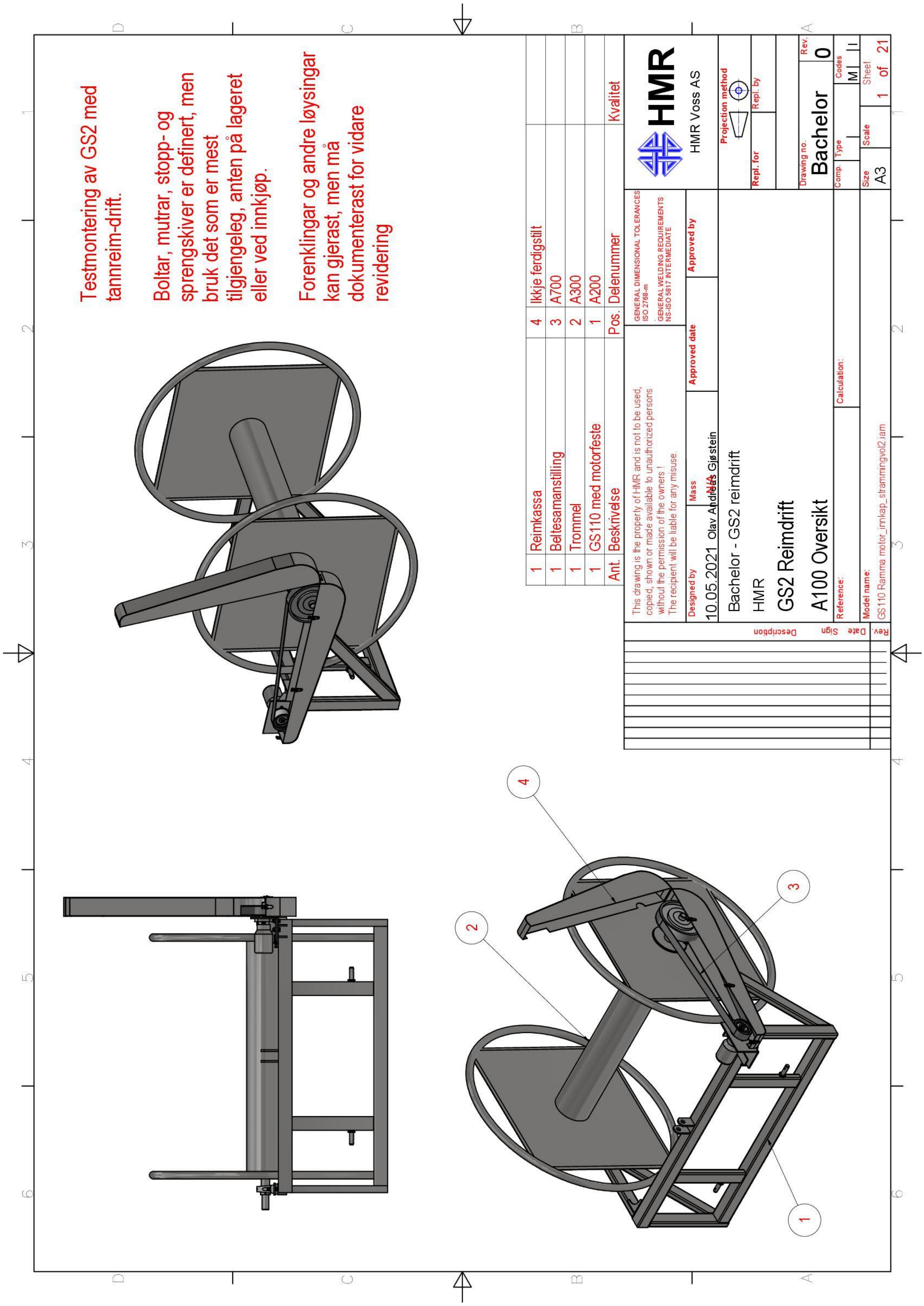






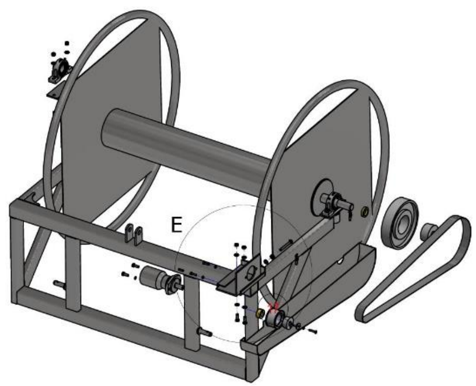
Appendix B – Technical drawings

Appendix B shows technical drawings that was made to manufacture the prototype.

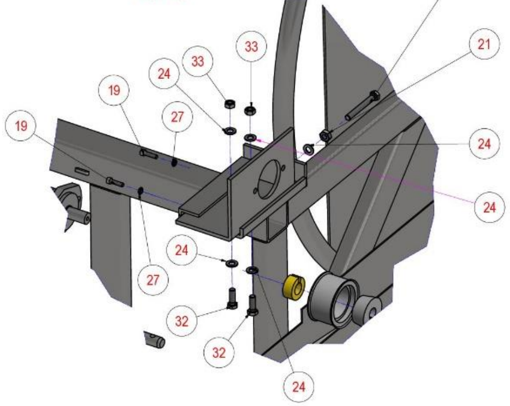


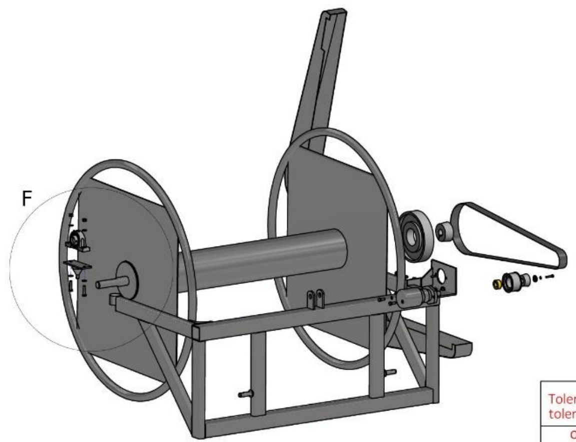
10.9 fastheitsklasse er å foretrekka
Bruk momenttabell for boltar og mutrar. Er ikkje spesifisert noko.

Forenklingar og andre løysingar kan gjerast, men må dokumenterast for vidare revidering

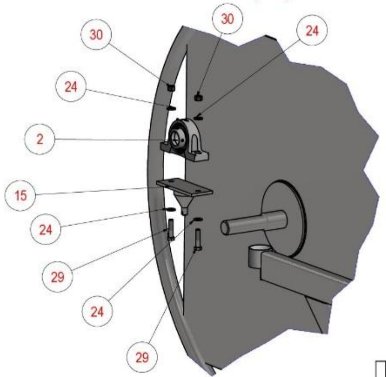


E (1 : 6)





F (1 : 8)



Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
2	Hex Nut	33	DIN 934 - M12		0 kg
2	Hex-Head Bolt	32	DIN 933 - M12 x 35		0 kg
2	Split Pin	31	DIN EN ISO 1234 - 4x28		0 kg
2	Hex Nut	30	ISO 4034 - M12		0 kg
4	Hex-Head Bolt	29	ISO 4014 - M12 x 50		0,1 kg
2	Hex Nut	28	ISO 4032 - M12		0 kg
2	Hex Nut	27	DIN 439 - M8		0 kg
2	Washer	26	DIN 125 - A 8,4		0 kg
2	Hex-Head Bolt	25	ISO 4017 - M12 x 30		0 kg
13	Washer	24	DIN 125 - A 13		0 kg
1	Spring Washer	23	DIN 128 - A8		0 kg
1	Hex-Head Bolt	22	DIN 933 - M8 x 40		0 kg
1	Hex Nut	21	DIN 6915 - M12		0 kg
1	Hex-Head Bolt	20	DIN 933 - M12 x 100		0,1 kg
2	Cylinder Head Cap Screw	19	DIN 912 - M8 x 30		0 kg
1		18	Keyway	S355	0 kg
1		17	Extender	S355	0 kg
2		16	Akseltapp m plate og styreplater	34CrNiMo6	7,6 kg
1		15	GS110 Ramma og trommel		82,5 kg

Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR
HMR Voss AS

Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	N/A		

Description: Bachelor - GS2 reimdrift
HMR
Splitteikning
A100 split

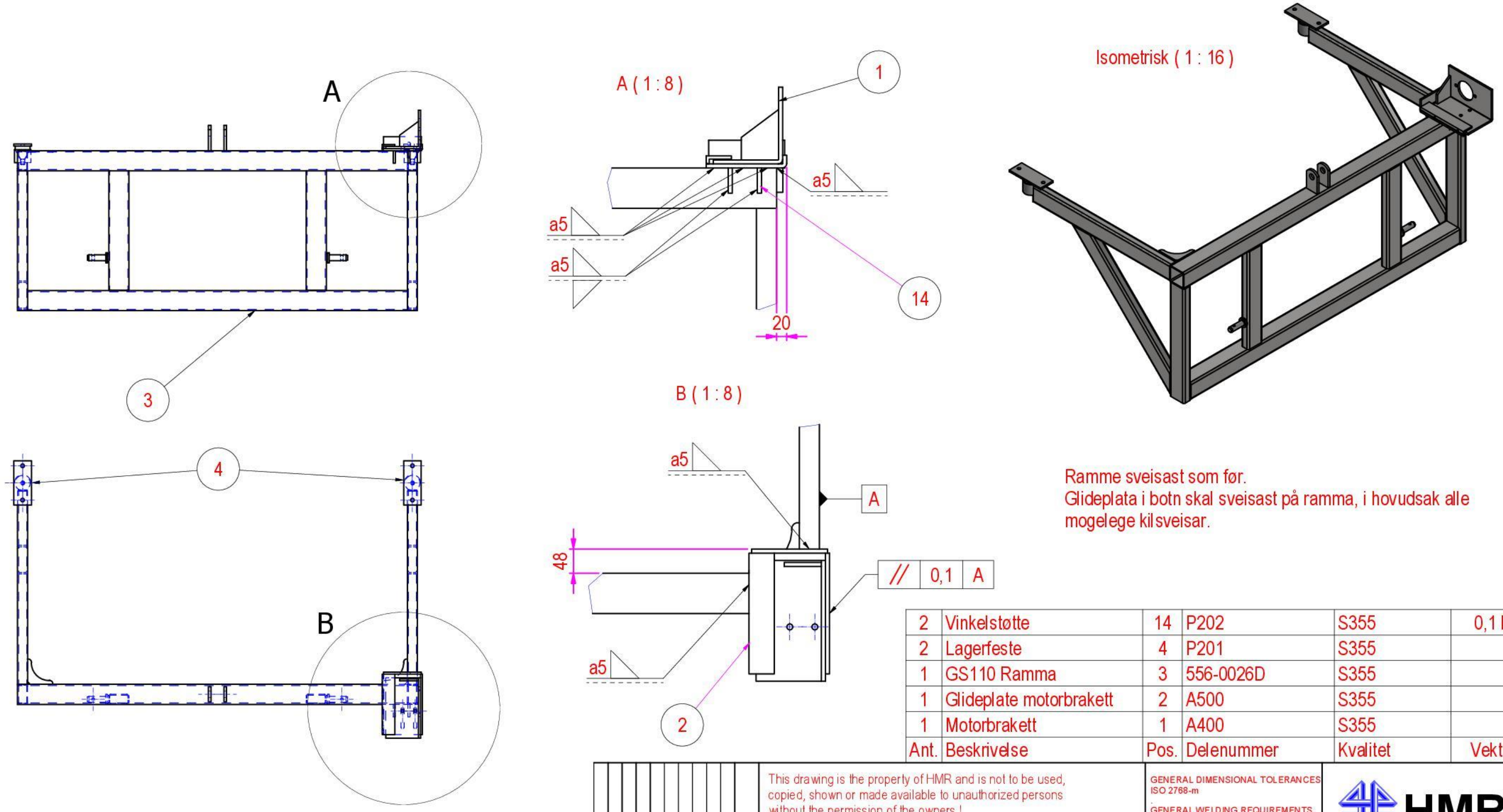
Projection method
Repl. for | Repl. by

Drawing no.: Bachelor | Rev.: 0

Reference: | Calculation: | Comp. | Type | Codes: M | I

Model name: GS110 Ramma motor_innkap_strammingvol2.iam | Size: A2 | Scale | Sheet: 2 of 21

Rev | Date | Sign | Description



Ramme sveisast som før.
Glideplata i botn skal sveisast på ramma, i hovudsak alle mogelege kilsveisar.

Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
2	Vinkelstøtte	14	P202	S355	0,1 kg
2	Lagerfeste	4	P201	S355	
1	GS110 Ramma	3	556-0026D	S355	
1	Glideplate motorbrakett	2	A500	S355	
1	Motorbrakett	1	A400	S355	

Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

Rev. Date Sign Description

GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

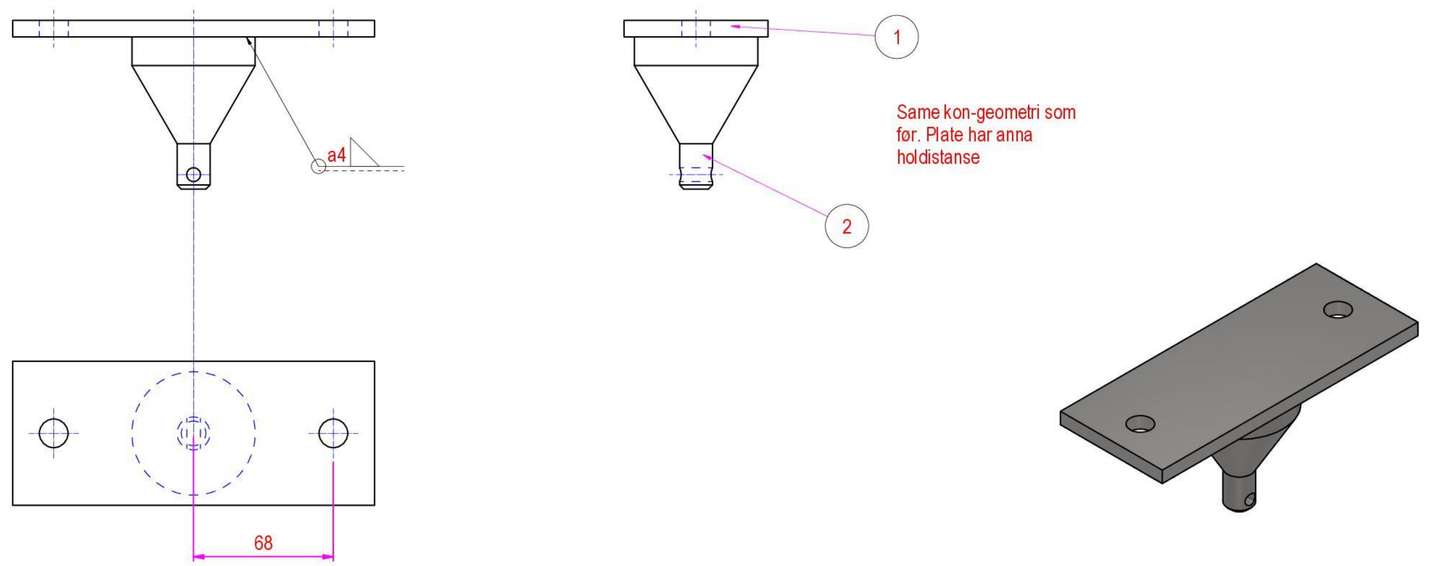
HMR
HMR Voss AS

Designed by: 10.05.2021 Olav Andreas Gjøstein
Mass: NA
Approved date
Approved by

Projection method
Repl. for
Repl. by

Bachelor - GS2 reimdrift
HMR
Sveis og plassering av motorfeste
A200

Drawing no.: Bachelor
Rev.: 0
Reference:
Calculation:
Comp. Type Codes: M | I
Model name: GS110 ramma med motorfeste.iam
Size: A3
Scale
Sheet: 3 of 21



Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Konfeste for lagerfeste - rundstål Ø60-74	2	556-0016 - 10512	S355	
1	Fotlagerfeste	1	Flattstål 176x70x8mm	S355	

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GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR
HMR Voss AS

Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	1,47 kg		

Projection method

Repl. for | Repl. by

Description | Sign | Date | Rev.

Bachelor - GS2 reimdrift
HMR
Lagerfeste
P201

Drawing no.: Bachelor | Rev.: 0

Reference: | Calculation: | Comp. | Type | Codes: M | I

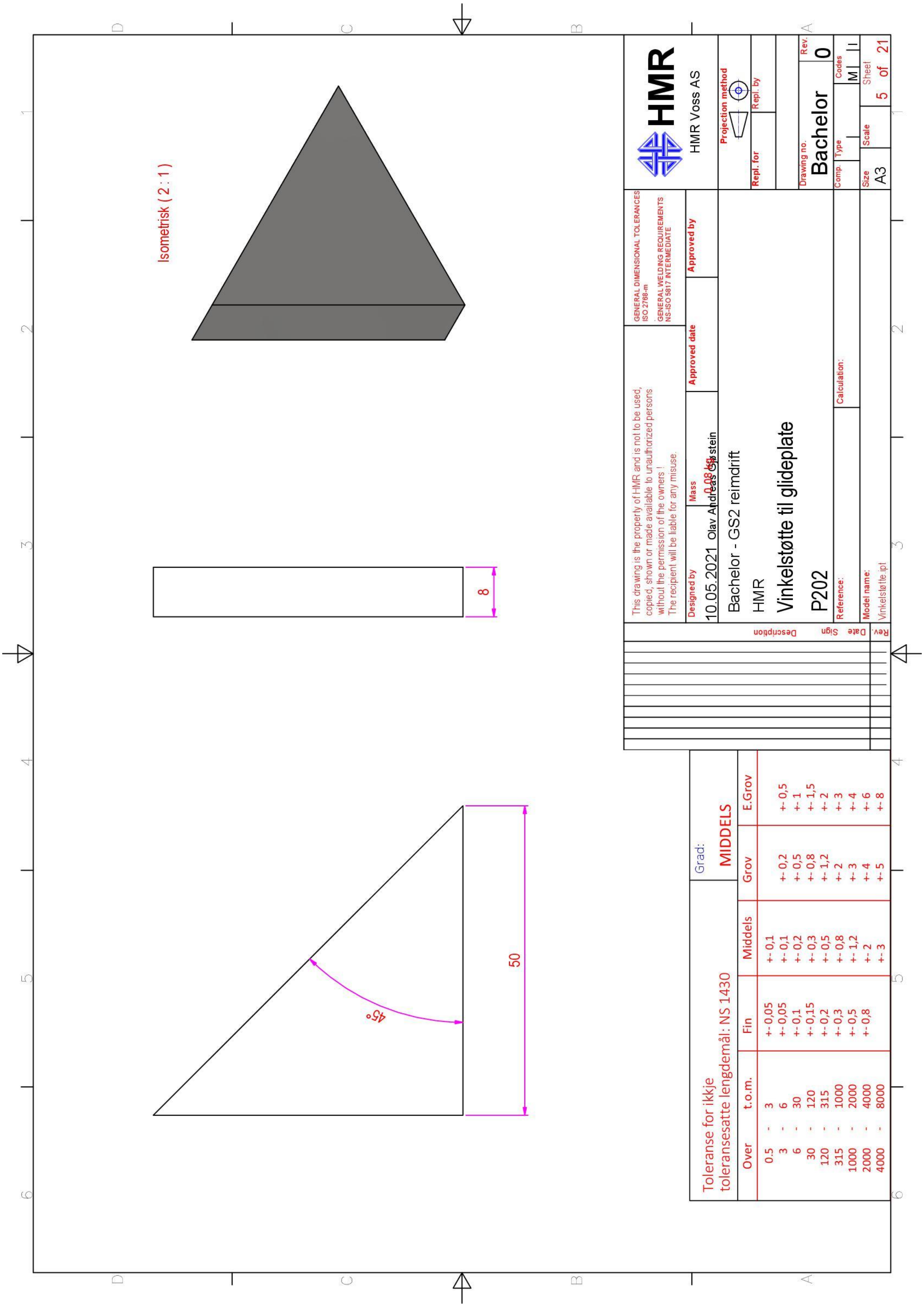
Model name: 10039 - Flattstål til lager SKF SY40FM.ipt

Size: A3 | Scale | Sheet: 4 of 21

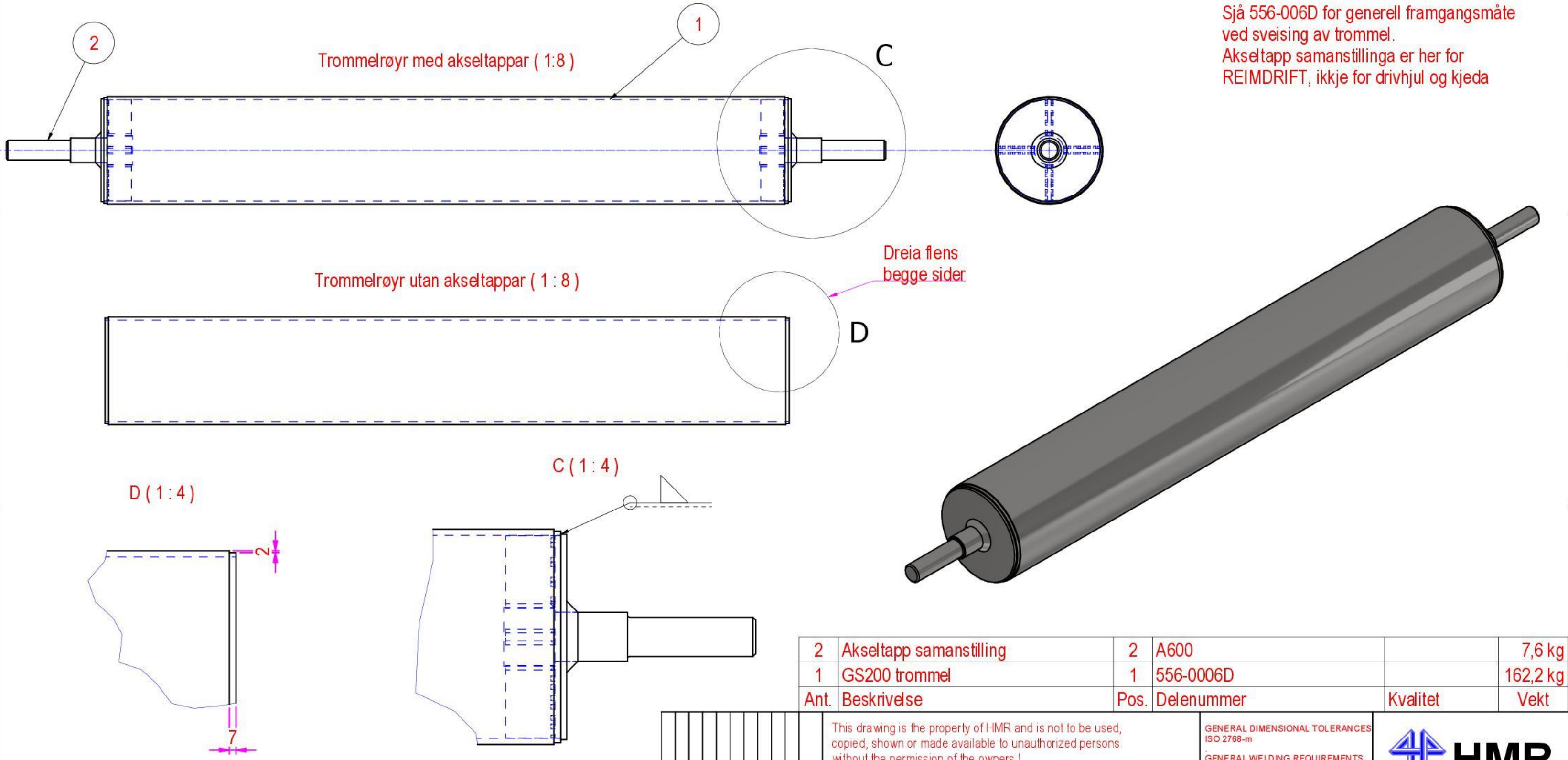
Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8



Over	t.o.m.	Fin	Middels	Grov	E.Grov
0.5	3	+- 0,05	+- 0,1		
3	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	4000	+- 0,8	+- 2	+- 4	+- 6
4000	8000		+- 3	+- 5	+- 8



Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
2	Akseltapp samanstilling	2	A600		7,6 kg
1	GS200 trommel	1	556-0006D		162,2 kg

Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

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GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR
HMR Voss AS

Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	NA		

Projection method

Repl. for | Repl. by

Rev. | Date | Sign | Description

Bachelor - GS2 reimdrift

HMR

Trommelrøyr utan plater og midtdelar

A300 samanstilling

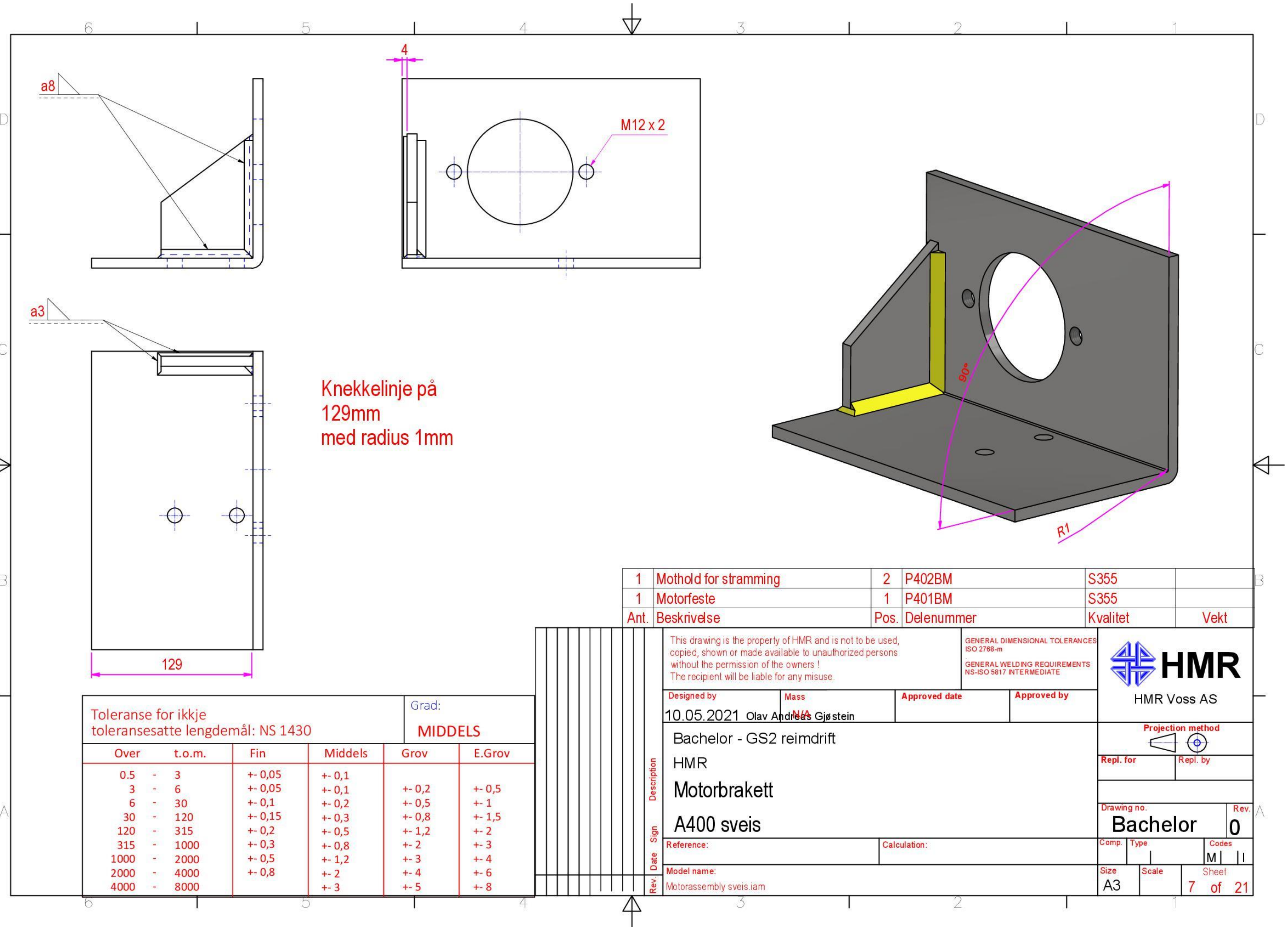
Drawing no. Bachelor | Rev. 0

Reference: | Calculation:

Comp. | Type | Codes M | I

Model name: Trommel med akseltapp uten røyr.iam

Size A3 | Scale | Sheet 6 of 21


a8
a3
4
M12 x 2
129
90°
R1
Knekkelinje på
129mm
med radius 1mm
Toleranse for ikkje
toleransesatte lengdemål: NS 1430
Grad:
MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Mothold for stramming	2	P402BM	S355	
1	Motorfeste	1	P401BM	S355	

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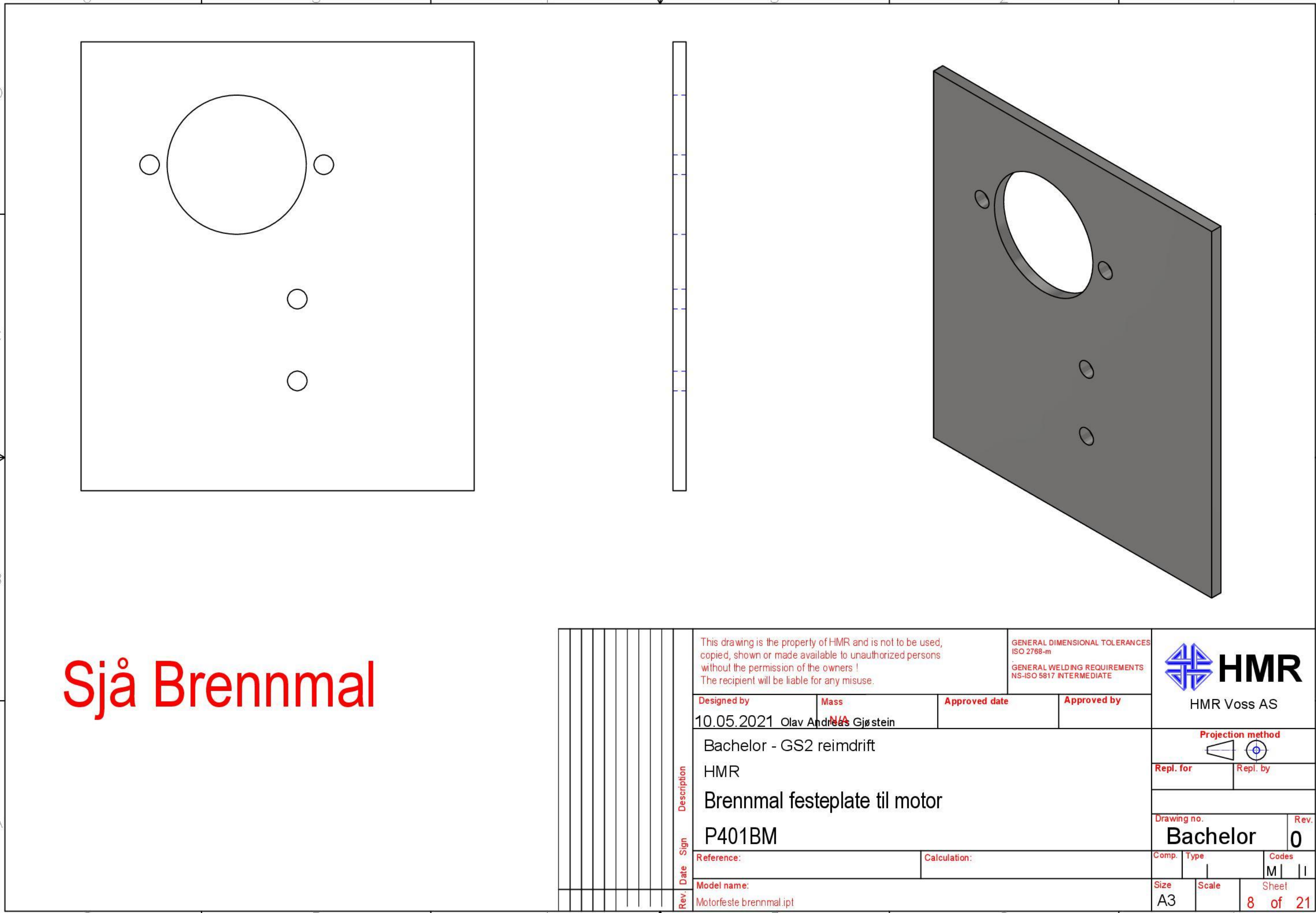
GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE
HMR
HMR Voss AS
Designed by
10.05.2021 Olav Andreas Gjøstein
Mass
N/A
Approved date
Approved by
Projection method
Repl. for
Repl. by
Bachelor - GS2 reimdrift
HMR
Motorbrakett
A400 sveis
Description
Sign
Date
Rev.
Drawing no.
Bachelor
Rev.
0
Reference:
Calculation:
Comp.
Type
Codes
M I
Model name:
Motorassembly sveis.iam
Size
A3
Scale
Sheet
7 of 21



Sjå Brennmal
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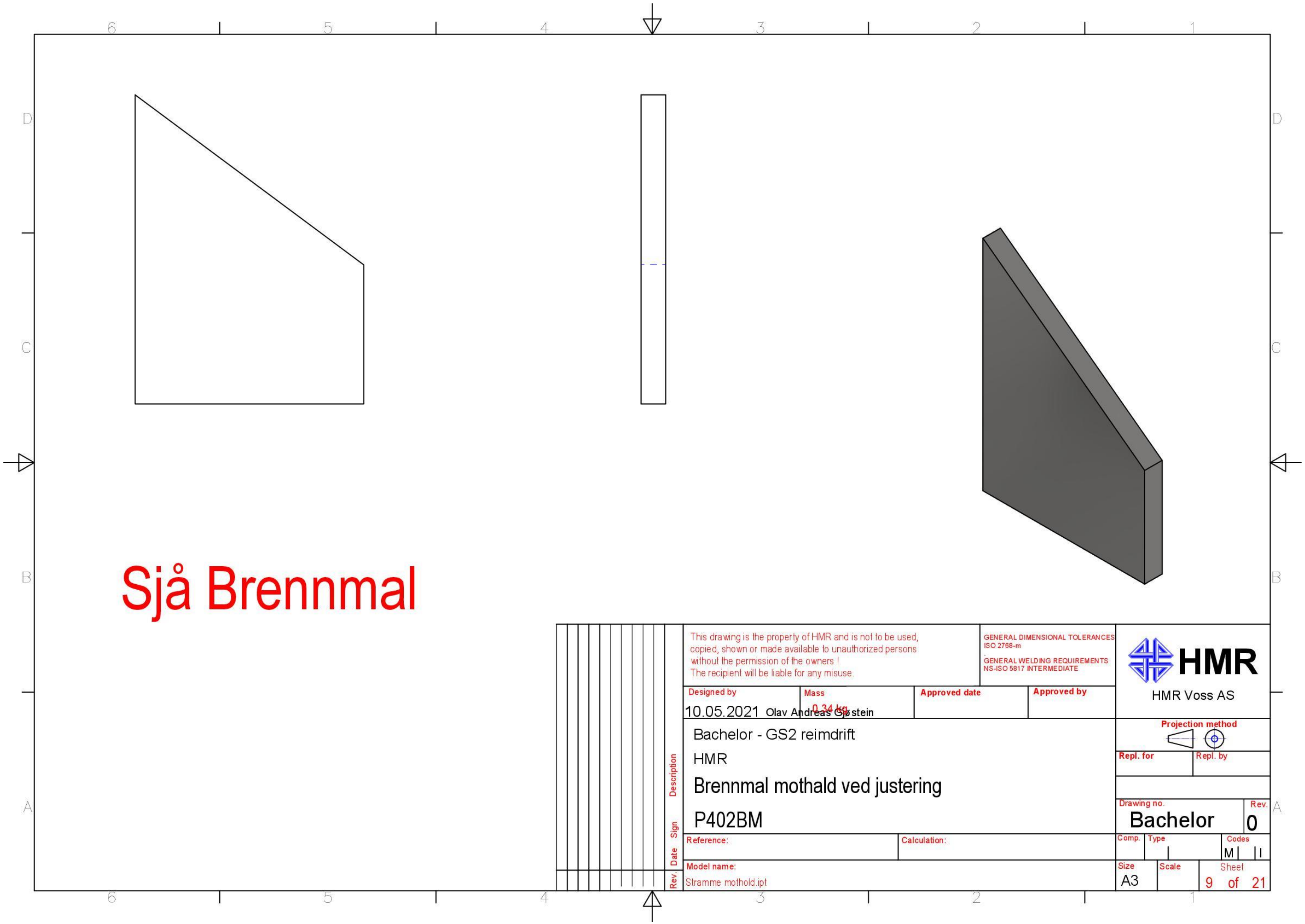
GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE
HMR
HMR Voss AS
Designed by
10.05.2021 Olav Andreas Gjøstein
Mass
N/A
Approved date
Approved by
Projection method
Repl. for
Repl. by
Bachelor - GS2 reimdrift
HMR
Brennmal festeplate til motor
P401BM
Drawing no.
Bachelor
Rev.
0
Reference:
Calculation:
Comp.
Type
Codes
M I
Model name:
Motorfeste brennmal.ipt
Size
A3
Scale
Sheet
8 of 21
Rev.
Date
Sign
Description



Sjå Brennmal
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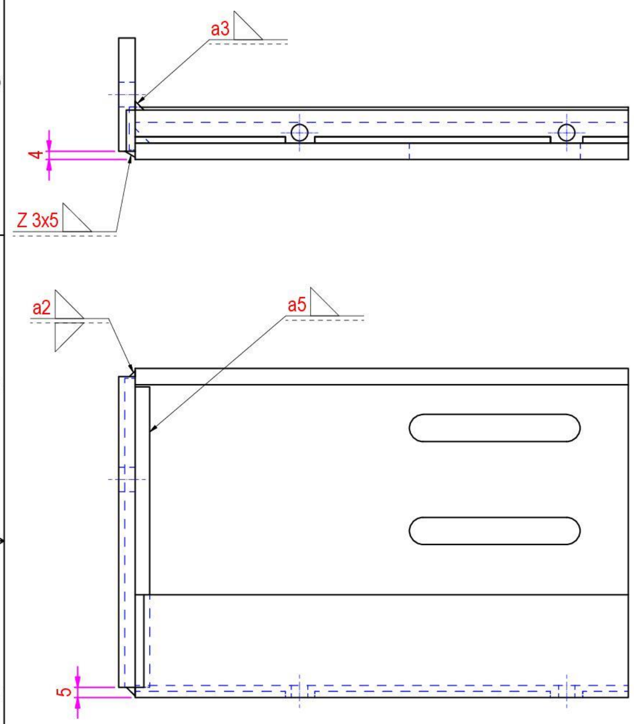
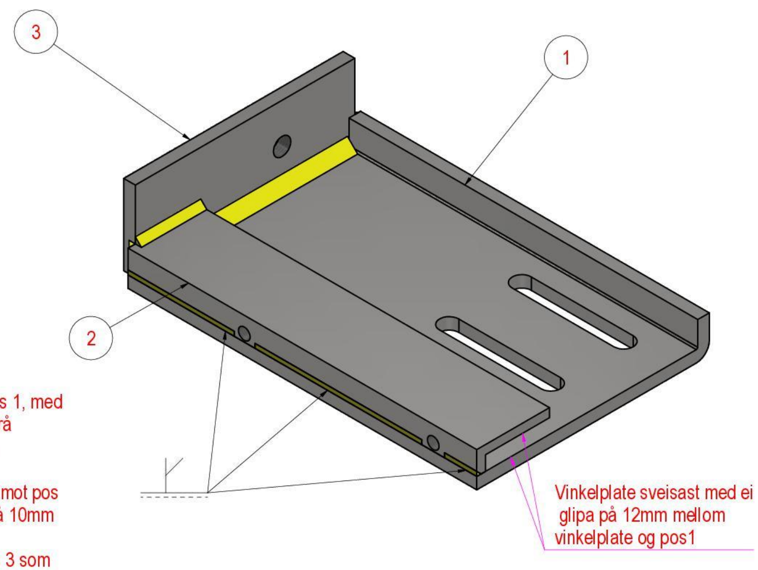
GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE
HMR
HMR Voss AS
Designed by
Mass
0,34 kg
Approved date
Approved by
10.05.2021
Olav Andreas Gjøstein
Projection method
Repl. for
Repl. by
Bachelor - GS2 reimdrift
HMR
Brennmal mothald ved justering
P402BM
Drawing no.
Bachelor
Rev.
0
Description
Sign
Date
Rev.
Reference:
Calculation:
Comp.
Type
Codes
M
I
Model name:
Stramme mothold.ipt
Size
A3
Scale
Sheet
9 of 21





1. Pos 3 sveisast på pos 1, med 4mm vertikal distanse frå botnplate si underside

2. Pos 2 klemmast fast mot pos 1, med distanseplate på 10mm

3. Pos 2 sveisast til pos 3 som anvist

Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Gjenga plate for stramming av motor	3	P503	S355	0,5 kg
1	Kappa vinkelstål til styring	2	P502	S355	0,7 kg
1	Botnplate	1	P501	S355	2,5 kg

Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

GENERAL DIMENSIONAL TOLERANCES ISO 2768-m

GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR

HMR Voss AS

Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	NA		

Projection method

Repl. for | Repl. by

Description

Bachelor - GS2 reimdrift

HMR

Glideplate for motorbrakett

A500

Drawing no. Bachelor

Rev. 0

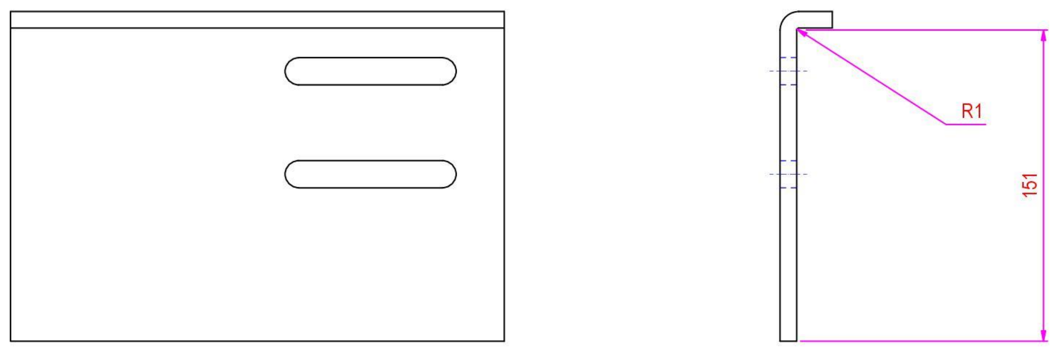
Sign | Date | Rev.

Reference: | Calculation:

Comp. | Type | Codes M | I

Model name: Motorjustering vol 2 sveis.iam

Size A3 | Scale | Sheet 10 of 21



Knekkast 90 grader, radius 1,
med 151mm distanse til knekk-kant

Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Botnplate	1	P501BM	S355	2,5 kg

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GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR
HMR Voss AS

Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	2,51 kg		

Projection method

Repl. for | Repl. by

Description:
Bachelor - GS2 reimdrift
HMR
Botnplate
P501

Drawing no.: Bachelor
Rev.: 0

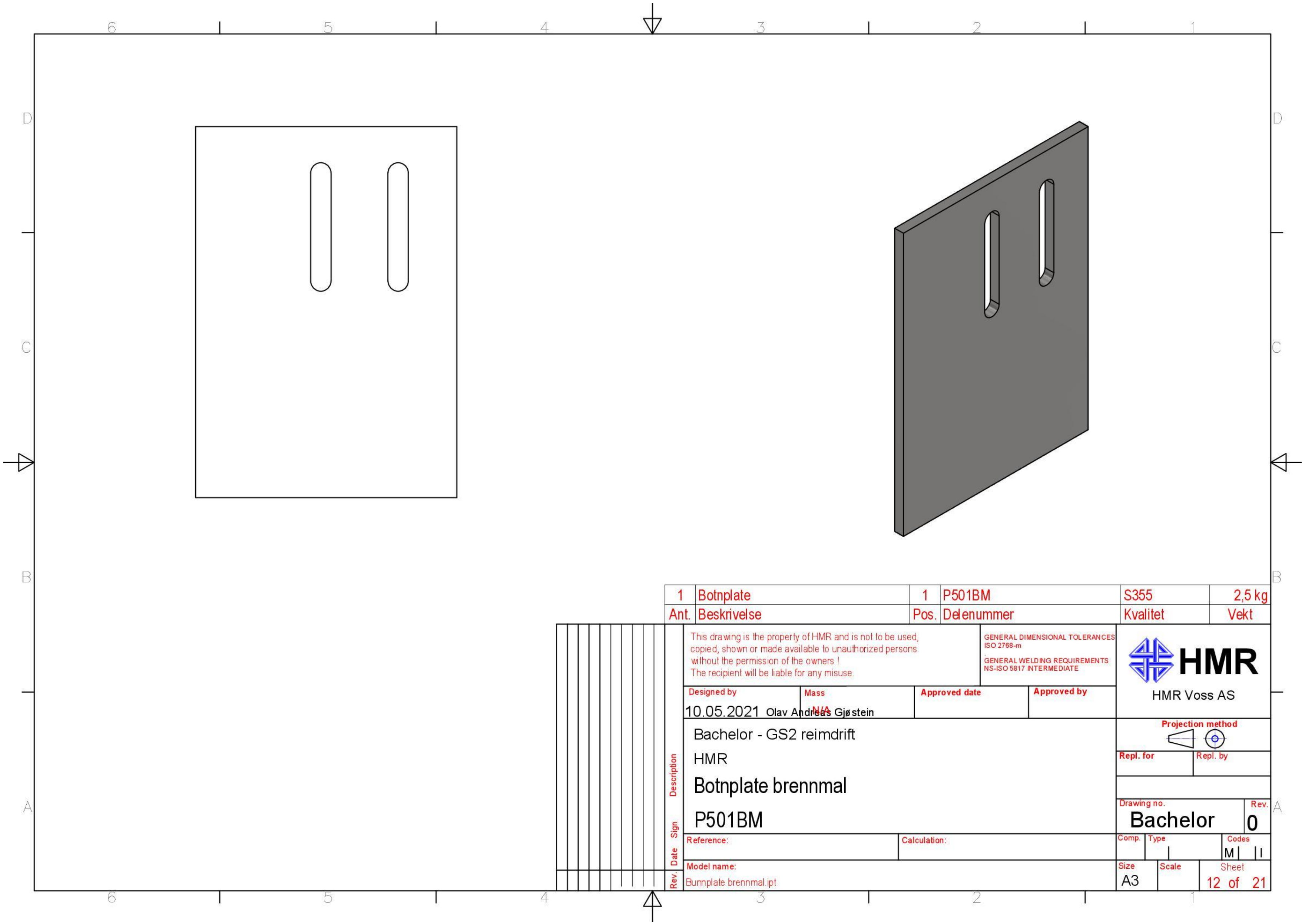
Reference: | Calculation:

Comp. | Type | Codes: M | I

Model name: Bunnplate.ipt

Size: A3 | Scale | Sheet: 11 of 21

Rev. | Date | Sign | Description


1 Botnplate 1 P501BM S355 2,5 kg
Ant. Beskrivelse Pos. Delenummer Kvalitet Vekt
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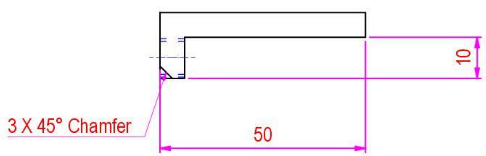
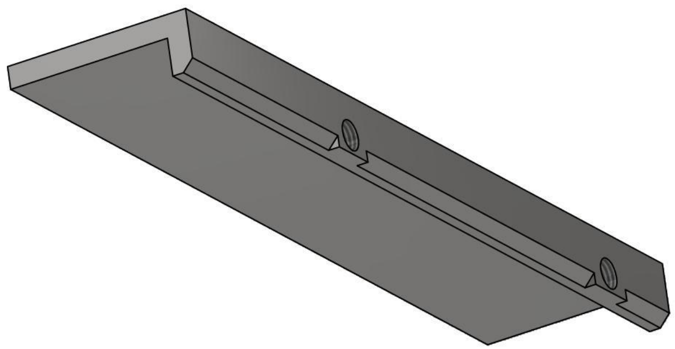
The recipient will be liable for any misuse.
GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE
HMR
HMR Voss AS
Designed by
Mass
Approved date
Approved by
10.05.2021 Olav Andreas Gjøstein
N/A
Projection method
Repl. for
Repl. by
Bachelor - GS2 reimdrift
HMR
Botnplate brennmal
P501BM
Drawing no.
Bachelor
Rev.
0
Reference:
Calculation:
Comp.
Type
Codes
M I
Model name:
Bunnplate brennmal.ipt
Size
A3
Scale
Sheet
12 of 21
Rev.
Date
Sign
Description



1. Vinkeljern med 6mm tjukkelse, 50x50mm.
2. Bora og gjenga opp hol
3. Kappa eine flensen som vist på teikning
4. Slipa til fuga der ein har kappa.

Fugelengda treng ikkje væra 100% nøyaktig.
Skal ikkje sveisast ved posisjonen til gjengehola.



Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Vinkeljern til sidejustering av brakett	1	Vinkeljern 50x50x6	S355	0,7 kg

GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR
HMR Voss AS

Designed by: 10.05.2021 Olav Andreas Gjøstein
Mass: NA
Approved date:
Approved by:

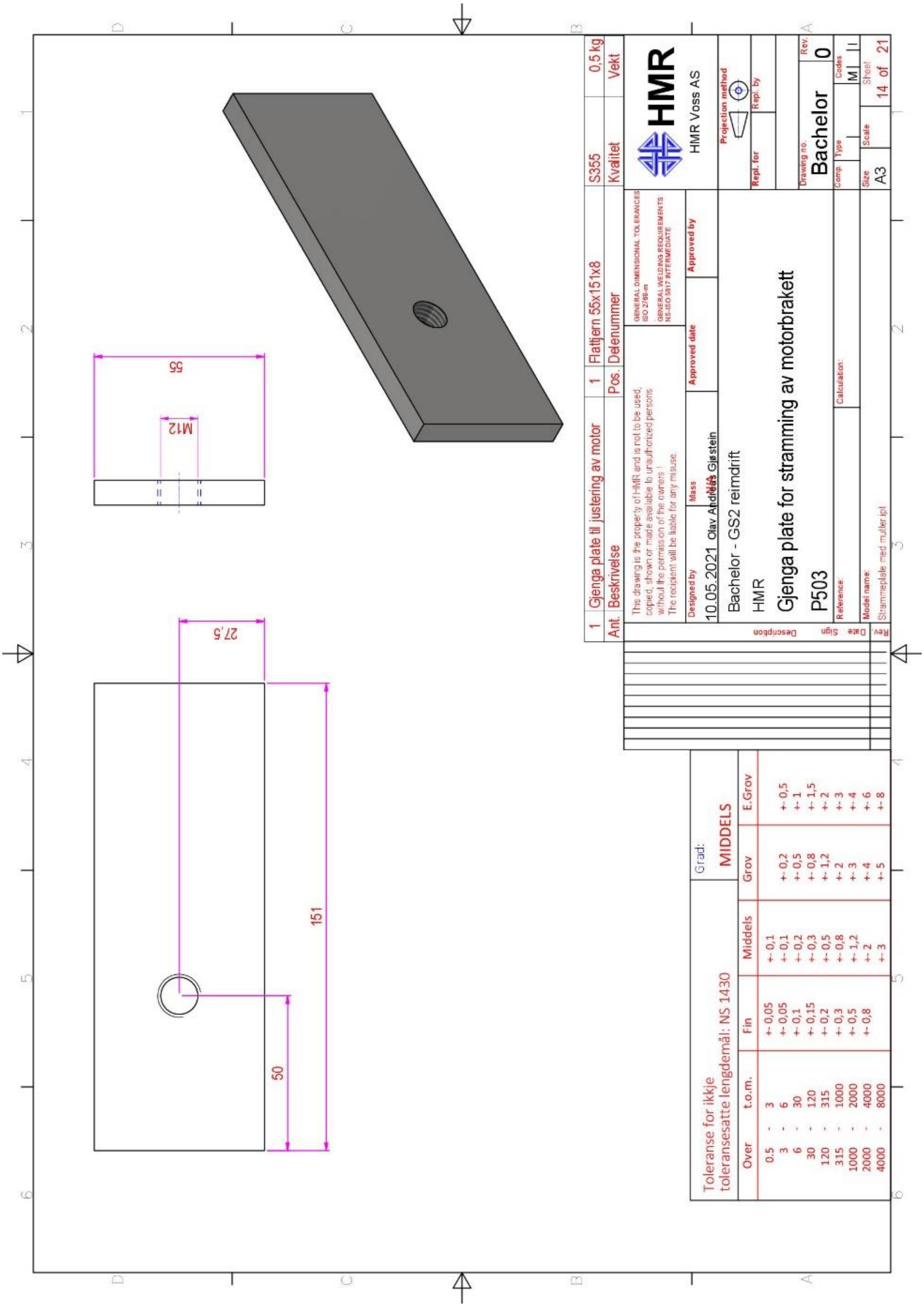
Projection method

Repl. for:
Repl. by:

Description / Sign / Date / Rev.

Bachelor - GS2 reimdrift
HMR
Vinkel for sidejustering av motorbrakett
P502

Drawing no.: Bachelor
Rev.: 0
Reference:
Calculation:
Comp.:
Type:
Codes: M | | I
Model name: Vinkel.ipt
Size: A3
Scale:
Sheet: 13 of 21



55
M12
27,5
151
50
1 | Gjenga plate til justering av motor | 1 | Flatjern 55x151x8 | S355 | 0,5 kg
Ant. | Beskrivelse | Pos. | Delenummer | Kvalitet | Vekt
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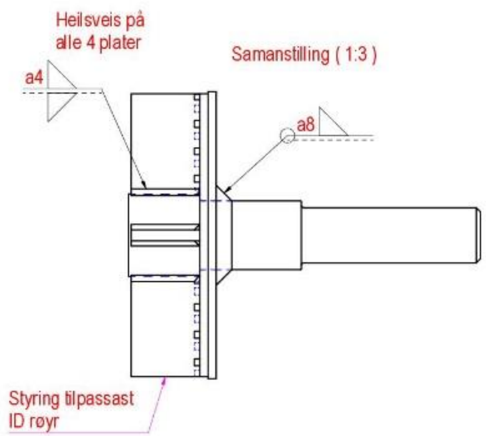
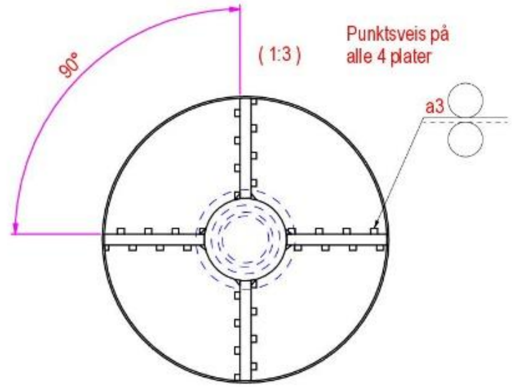
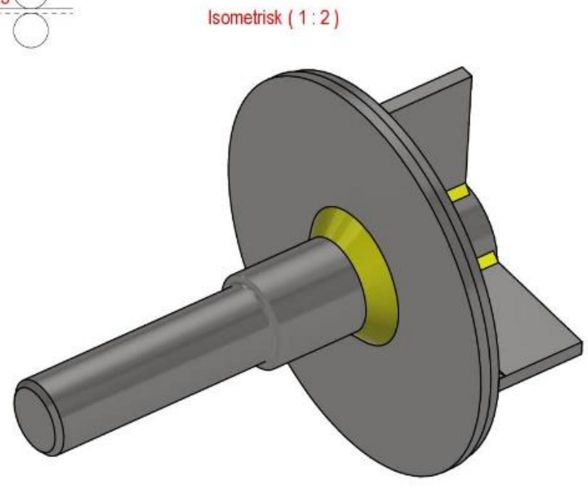
GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE
HMR
HMR Voss AS
Designed by
10.05.2021 Olav Andreas Gjøstein
Mass
Approved date
Approved by
Projection method
Bachelor - GS2 reimdrift
HMR
Gjenga plate for stramming av motorbrakett
P503
Repl. for
Repl. by
Drawing no.
Bachelor
Rev.
0
Reference:
Calculation:
Comp.
Type
Codes
M
Model name:
Stammeplate med mutter.ipt
Size
A3
Scale
Sheet
14 of 21
Description
Sign
Date
Rev.
Toleranse for ikkje toleransesatte lengdemål: NS 1430
Grad:
MIDDELS
Over | t.o.m. | Fin | Middels | Grov | E.Grov
0.5 - 3 | +- 0,05 | +- 0,1 | +- 0,2 | +- 0,5
3 - 6 | +- 0,05 | +- 0,1 | +- 0,5 | +- 1
6 - 30 | +- 0,1 | +- 0,2 | +- 0,8 | +- 1,5
30 - 120 | +- 0,15 | +- 0,3 | +- 1,2 | +- 2
120 - 315 | +- 0,2 | +- 0,5 | +- 2 | +- 3
315 - 1000 | +- 0,3 | +- 0,8 | +- 3 | +- 4
1000 - 2000 | +- 0,5 | +- 1,2 | +- 4 | +- 6
2000 - 4000 | +- 0,8 | +- 2 | +- 5 | +- 8
4000 - 8000 | | +- 3 | |







Same montasje-framgangsmåte som
før, men med andre mål og
dimensjonar.

Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Opplagring -og drivaksling	3	P603	34CrNiMo6	3,6 kg
4	Flatstål 8x50-80, 10036	2	P602	S355	0,2 kg
1	Plate 12xØ209, 10681 med ID50	1	P601	S355	3 kg

This drawing is the property of HMR and is not to be used, copied, shown or made available to unauthorized persons without the permission of the owners ! The recipient will be liable for any misuse.

GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR
HMR Voss AS

Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	7,62 kg		

Bachelor - GS2 reimdrift
HMR
Akseltapp samanstilling
A600

Projection method

Repl. for	Repl. by

Drawing no.	Rev.
Bachelor	0

Reference:	Calculation:	Comp.	Type	Codes
				M I

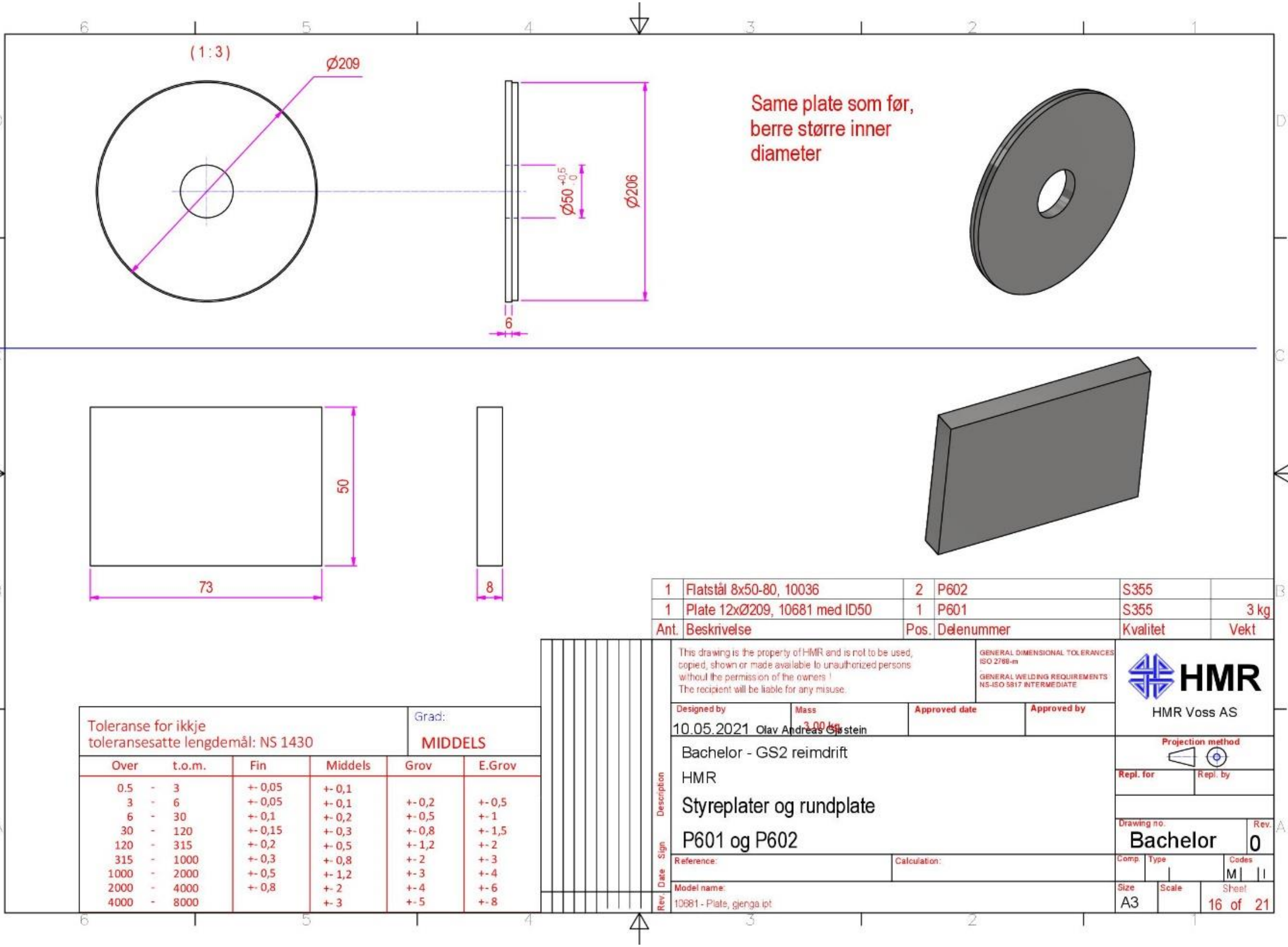
Model name:	Size	Scale	Sheet
Akseltapp m plate og styreplater.iam	A3		15 of 21

Description
Rev. Date Sign

Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

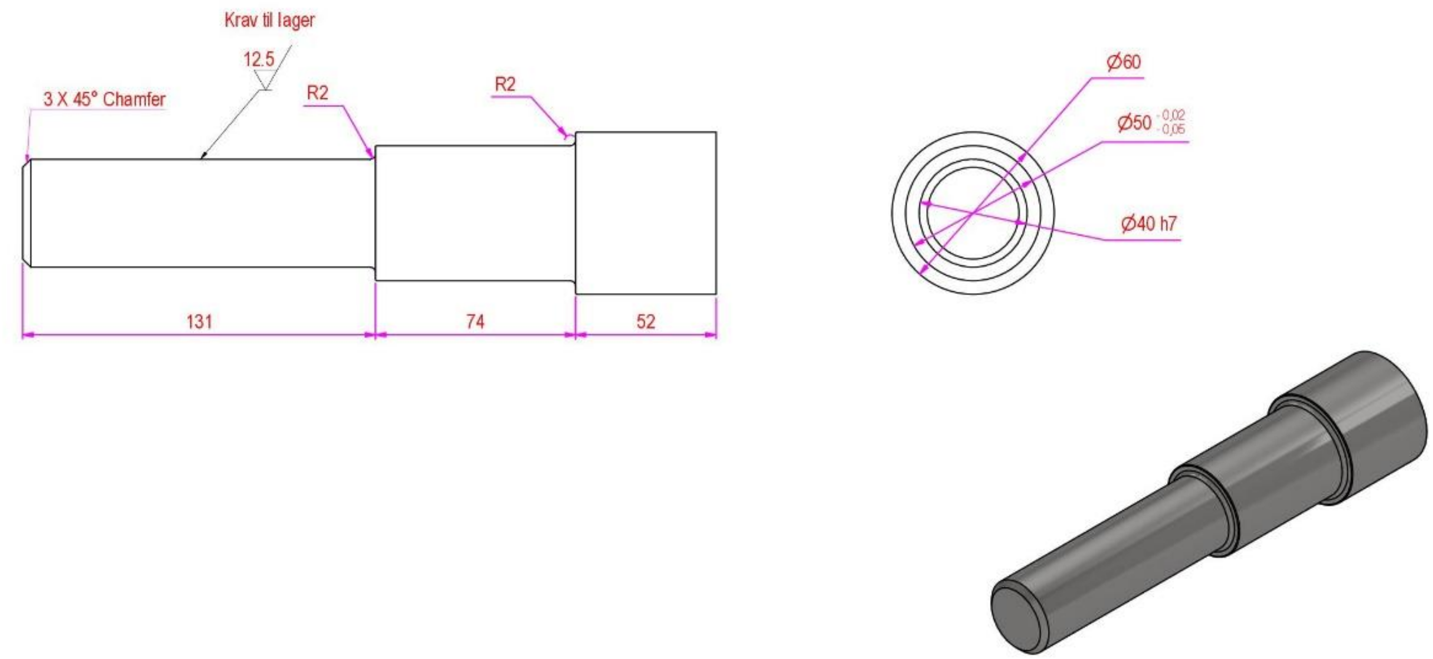


Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Flatstål 8x50-80, 10036	2	P602	S355	
1	Plate 12xØ209, 10681 med ID50	1	P601	S355	3 kg

Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8



Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

Rev. | Date | Sign | Description

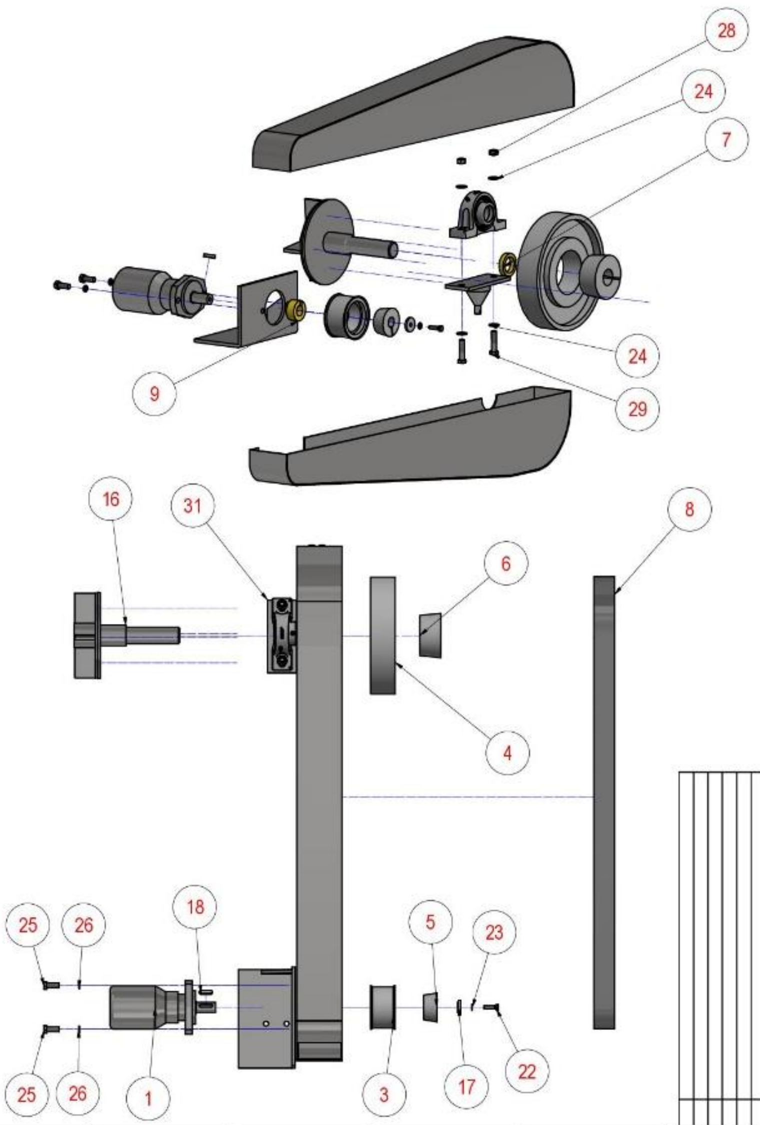
GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

HMR
HMR Voss AS

Designed by: 10.05.2021 Olav Andreas Gjøstein
Mass: 3.58 kg
Approved date:
Approved by:

Bachelor - GS2 reimdrift
HMR
Drivaksel
P603

Projection method
Repl. for
Repl. by
Drawing no.: Bachelor
Rev.: 0
Reference:
Calculation:
Comp.
Type
Codes: M | I
Model name: F.1 aksel drivside - Ø40 til Ø60 plate.ipt
Size: A3
Scale
Sheet: 17 of 21



Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Lagerfeste	31	P201	S355	1,5 kg
1	Motorbrakett	30	A400		4,2 kg
2	Hex-Head Bolt	29	ISO 4014 - M12 x 50		0,1 kg
2	Hex Nut	28	ISO 4032 - M12		0 kg
2	Washer	26	DIN 125 - A 8,4		0 kg
2	Hex-Head Bolt	25	ISO 4017 - M12 x 30		0 kg
4	Washer	24	DIN 125 - A 13		0 kg
1	Spring Washer	23	DIN 128 - A8		0 kg
1	Hex-Head Bolt	22	DIN 933 - M8 x 30		0 kg
1	Kile til kilspor, motor	18	A8x7x32 DIN 6885		0 kg
1	Styretapp til taper lock	17	P703		0 kg
1	Akseltapp med plate	16	A600		7,6 kg
1	Distansehylse motor	9	P702	Nylon	0 kg
1	Tannreim	8	HTD-2400-8M-50	Rubber	0,2 kg
1	Distansehylse trommel	7	P701	Nylon	0 kg
1	Taper lock trommel	6	3020 ID40	Steel	0,4 kg
1	Taper lock motor	5	2012 ID25	Steel	0,1 kg
1	Reimhjul trommel	4	112-8M-50	Steel	2 kg
1	Reimhjul motor	3	40-8M-50	Steel	0,2 kg
1	Fotlager	2	SKF SY 40 FM		0,8 kg
1	Hydraulikkmotor	1	MR314C/4		11,9 kg

GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

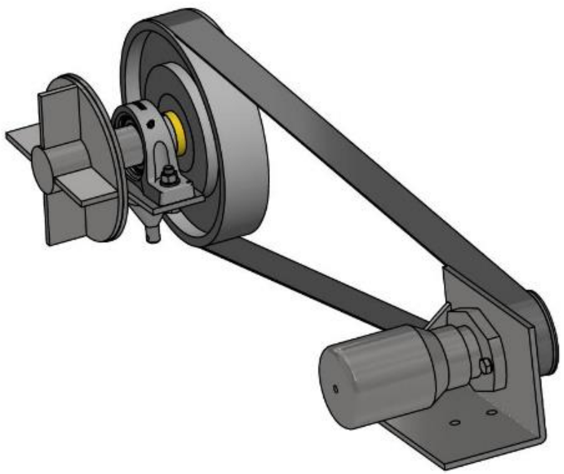
HMR
HMR Voss AS

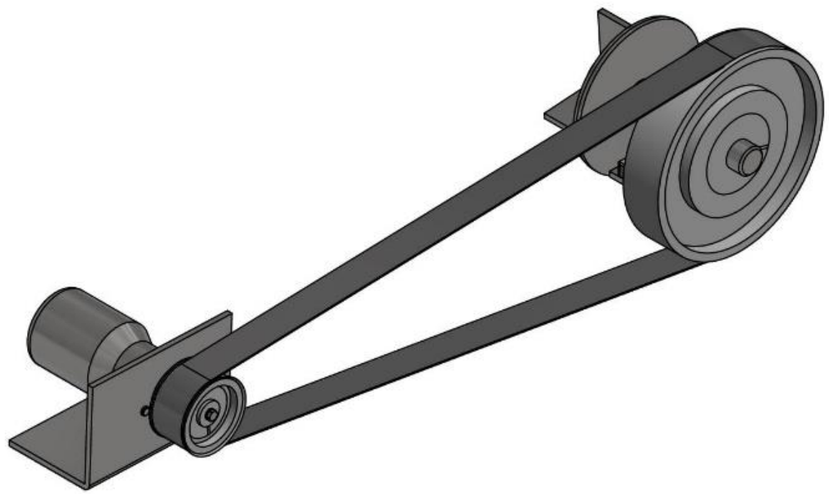
Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	N/A		

Description: Bachelor - GS2 reimdrift
HMR
Montasje av reimsystem- splitteikning
A700 Splitt

Projection method
Repl. for | Repl. by
Drawing no.: Bachelor | Rev.: 0
Reference: | Calculation:
Comp. | Type | Codes: M | SS | I
Model name: GS110 Beltemontering.iam | s
Size: A3 | Scale: S | Sheet: 18 of 21

Rev. | Date | Sign





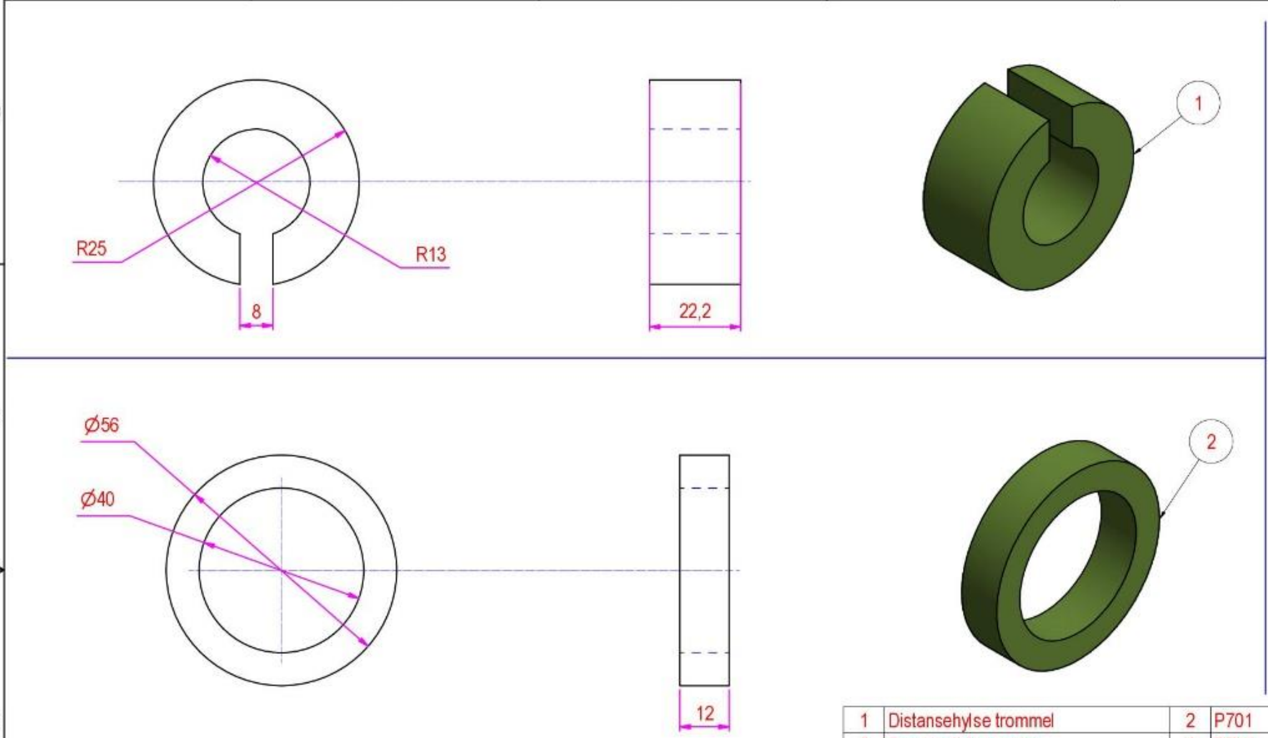
1. Trommel med aksel er montert i lager, og festast på lagerbrakett. Hugs hårnålssplint. Settskrue på lager skal snørast med 16.5Nm
2. Motor boltast fast i motorbrakett med loctite xxx. Om naudsynt, bruk sprengskive
3. Juster på justeringsskruene på glideplata (A500), slik at motorbraketten er omtrent låst til bevegelse i belteretning. Justeringsskruene låsast med kontramutter når OK
4. Set inn distansehylser
5. Set inn Taper lock på motor, med kile i kilsporet
6. Set inn Taper lock på trommel, utan kile i kilsporet
7. Set reimskiver på taper lock
8. Stram til set-skruer. 25Nm på motor, 92Nm på trommel.
9. Sjekk forskyvning og beinheita til reima. Juster på motorbrakett ved behov. Shims må vurderast om nødvendigt.
10. Styretapp for taper lock med bolt og skive vert skrudd fast i motoraksel. Bruk Loctite på gjengene. Her må ein finna rett gjengetype

Belte skal mindre enn 5mm i forskyvning per meter senterdistanse.
Bruk momenttabell for boltar og muttrar. 10.9 Fastheit er å foretrekka.

Forenklingar og andre løysingar kan gjerast, men må dokumenterast for vidare revidering

This drawing is the property of HMR and is not to be used, copied, shown or made available to unauthorized persons without the permission of the owners ! The recipient will be liable for any misuse.		GENERAL DIMENSIONAL TOLERANCES ISO 2768-m GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE		HMR HMR Voss AS
Designed by 10.05.2021	Mass Olav Andreas Gjøstein	Approved date	Approved by	
Description: Bachelor - GS2 reimdrift HMR Montasjeprosedyre A700 Montasje				Projection method
				Repl. for / Repl. by
				Drawing no. Bachelor / Rev. 0
Reference:	Calculation:			Comp. / Type / Codes M I
Model name: GS110 Beltemontering.iam				Size A3 / Scale / Sheet 19 of 21

Rev. | Date | Sign



Ant.	Beskrivelse	Pos.	Delenummer	Kvalitet	Vekt
1	Distansehylse trommel	2	P701	Nylon	
1	Distansehylse motor	1	P702	Nylon	0 kg

Toleranse for ikkje toleransesatte lengdemål: NS 1430

Grad: MIDDELS

Over	t.o.m.	Fin	Middels	Grov	E.Grov
0.5 -	3	+- 0,05	+- 0,1		
3 -	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6 -	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30 -	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120 -	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315 -	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000 -	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000 -	4000	+- 0,8	+- 2	+- 4	+- 6
4000 -	8000		+- 3	+- 5	+- 8

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GENERAL DIMENSIONAL TOLERANCES ISO 2768-m
GENERAL WELDING REQUIREMENTS NS-ISO 5817 INTERMEDIATE

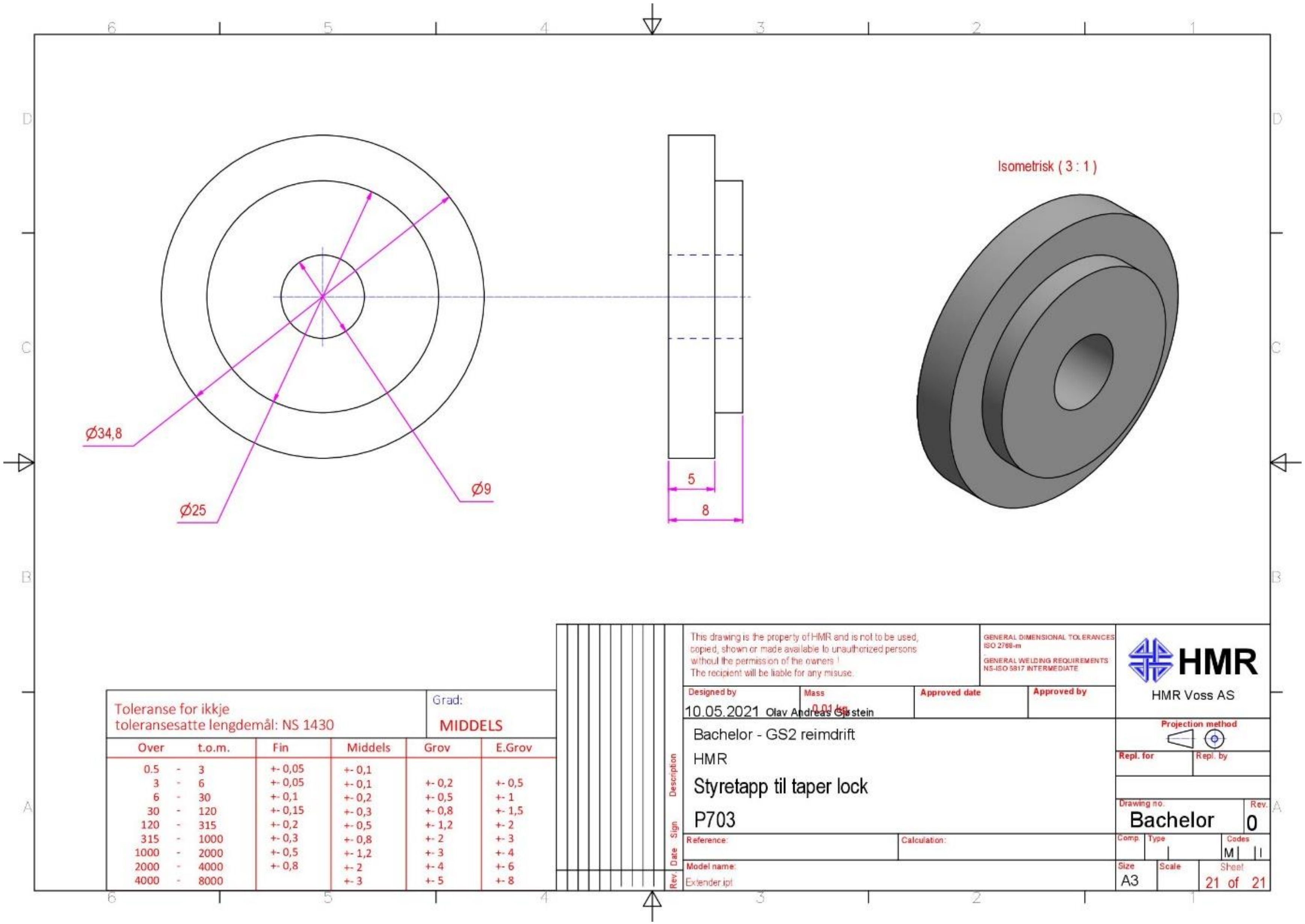
HMR
HMR Voss AS

Designed by	Mass	Approved date	Approved by
10.05.2021 Olav Andreas Gjøstein	0,03 kg		

Bachelor - GS2 reimdrift
HMR
Distansehylser til reimhjula
P701 og P702

Projection method
Repl. for
Repl. by
Drawing no. Bachelor
Rev. 0
Reference:
Calculation:
Comp.
Type
Codes M I I
Model name: Distansehylse motor.ipt
Size A3
Scale
Sheet 20 of 21

Rev. Date Sign Description



Toleranse for ikkje toleransesatte lengdemål: NS 1430

Over		t.o.m.	Fin	Middels	Grov	E.Grov
0.5	-	3	+- 0,05	+- 0,1		
3	-	6	+- 0,05	+- 0,1	+- 0,2	+- 0,5
6	-	30	+- 0,1	+- 0,2	+- 0,5	+- 1
30	-	120	+- 0,15	+- 0,3	+- 0,8	+- 1,5
120	-	315	+- 0,2	+- 0,5	+- 1,2	+- 2
315	-	1000	+- 0,3	+- 0,8	+- 2	+- 3
1000	-	2000	+- 0,5	+- 1,2	+- 3	+- 4
2000	-	4000	+- 0,8	+- 2	+- 4	+- 6
4000	-	8000		+- 3	+- 5	+- 8

Appendix C – Design Manual

First page has some general information regarding the PowerGrip HTD [30]. The rest contains the most relevant pages from the design manual for synchronous belts used in this thesis [6].

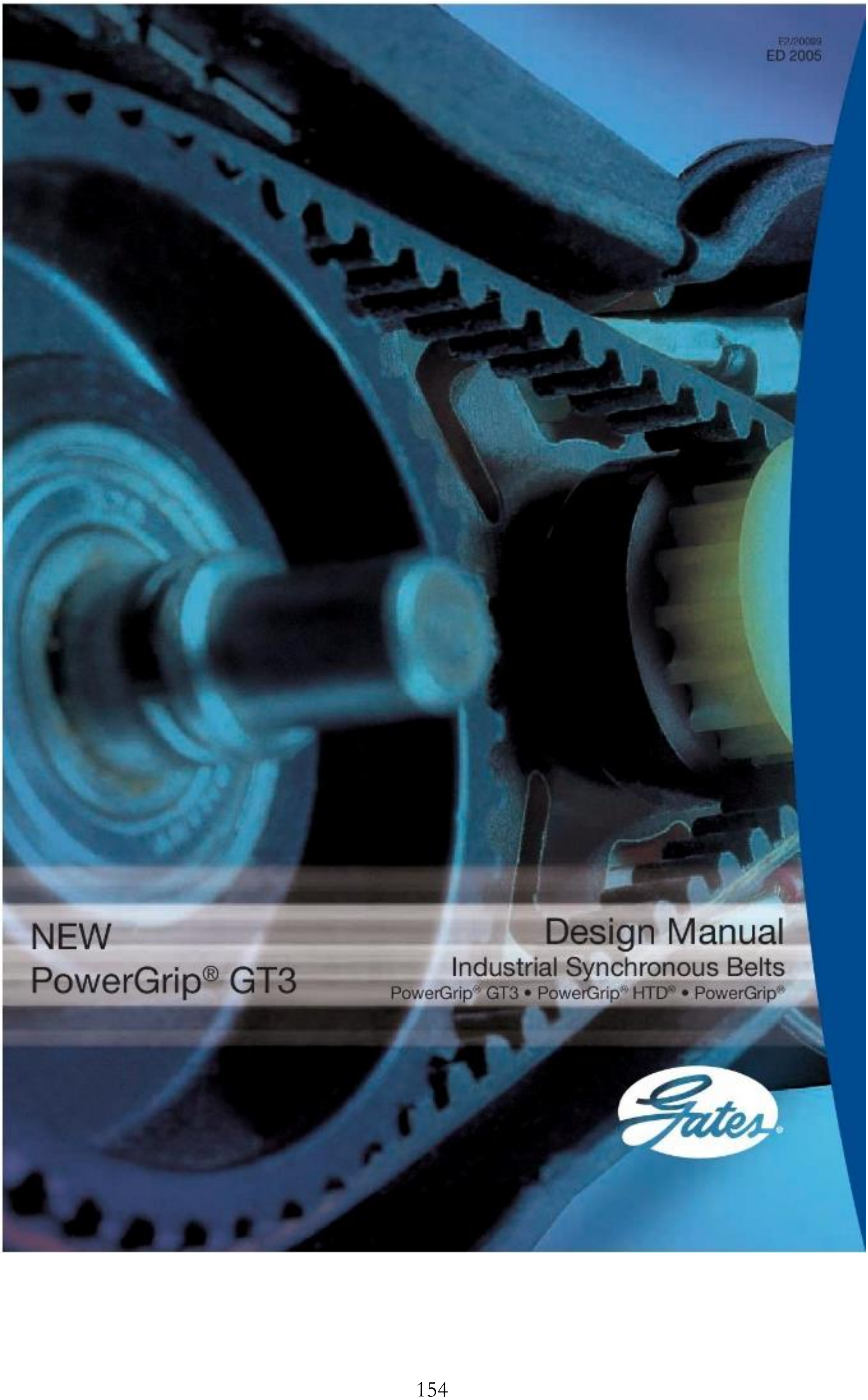
PowerGrip® HTD®

Rubber synchronous belts for high torque drives

PowerGrip® HTD® belts are ideal for high power transmission in low speed and high torque applications. PowerGrip® HTD® belts are used in high performance drives where durability and low maintenance are required.

Advantages

- Special curvilinear tooth form improves stress distribution and allows higher overall loading.
- No slippage. PowerGrip® HTD® belt teeth mesh smoothly with pulley grooves, reducing speed variations.
- Economical operation. No lubrication needed, no need for adjustment due to stretch and wear.
- Long trouble-free service life (because of excellent abrasion resistance) in many applications where metal components like chains and gears wear out in a matter of months.
- PowerPaint™ version available on request.


ED 2005
NEW
PowerGrip® GT3
Design Manual
Industrial Synchronous Belts
PowerGrip® GT3 • PowerGrip® HTD® • PowerGrip®
Gates

DRIVE DESIGN MANUAL FOR POWERGRIP® GT3, POWERGRIP® HTD® & POWERGRIP® SYNCHRONOUS BELTS

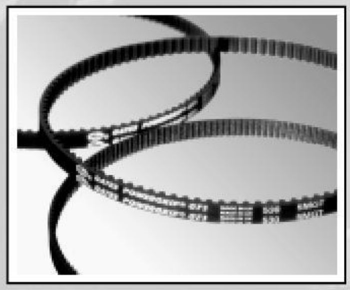
CONTENTS

INTRODUCTION

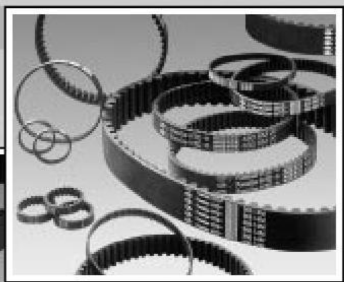
DRIVE DESIGN MANUAL FOR POWERGRIP® GT3, POWERGRIP® HTD® AND POWERGRIP® BELTS - ALL IN ONE

This combined PowerGrip® design manual provides engineers and designers with information on the range and scope of PowerGrip® GT3, PowerGrip® HTD® and classical PowerGrip® belt drives, together with full details of belt lengths, centre distances, power ratings and pulley ranges.
All information necessary to design the most appropriate synchronous belt drive to power your machines is provided in this manual. No need to consult various design manuals: this manual guides you through the complete drive selection procedure.

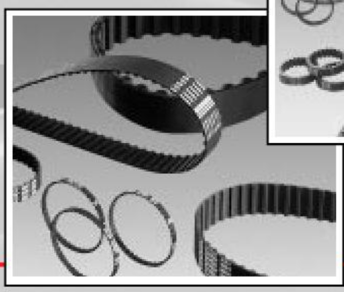
POWERGRIP® GT3



POWERGRIP® HTD®



POWERGRIP®



1

GATES SYNCHRONOUS BELTS: THE DESIGNER'S CHOICE

In 1946, Gates developed the first synchronous belt to synchronise the needle and bobbin movement of the Singer sewing machine. Through a programme of continuous innovation, research and development of high quality products Gates has acquired and maintained a leadership position in power transmission technology ever since. Gates offers designers and engineers a premium range of synchronous belts meeting industry's requirements.

Today Gates PowerGrip® conventional belt drives take their place in industry as a highly efficient proven medium for mechanical power transmission. PowerGrip® belts with classical trapezoidal teeth have been adopted as standard equipment for a wide range of industrial applications.

Improvements in materials and tooth design technology lead to the development of the PowerGrip® HTD® belt (High Torque Drive). The curvilinear HTD® tooth geometry eliminates stress concentration at tooth roots and allows higher power capacity and longer life compared to the classical timing belt.

Gates' latest development in synchronous rubber belts is PowerGrip® GT3.

The PowerGrip® GT3 product range is a major leap in synchronous rubber belt technology. Through the use of a highly advanced combination of materials, this new synchronous belt transmits up to 30% more power than previous generation belts.

PowerGrip® GT3 is available in 2MGT, 3MGT, 5MGT, 8MGT and 14MGT pitches.

The small 2MGT, 3MGT and 5MGT pitches are ideal for compact drives on hand tools, business machines, domestic appliances, high precision servomotor drives and multiaxis applications.

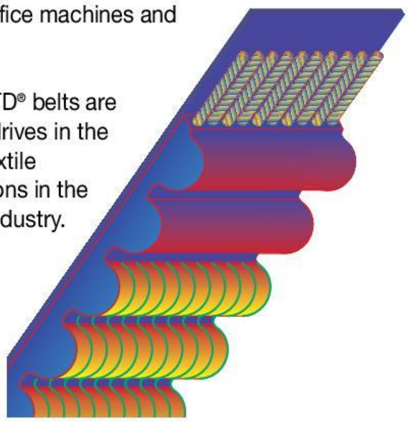
The larger 8MGT and 14MGT pitches are the optimum choice for high performance drives in the machine tool, paper and textile industries where durability and low maintenance are required.

POWERGRIP® HTD® BELT COMPONENTS AND BENEFITS

PowerGrip® HTD® drives provide positive power transmission for a wide range of industrial applications, and offer many advantages over conventional chain and gear drives.

3M and 5M pitch HTD® belts are especially suited for domestic appliances, office machines and electric hand tools.

8M, 14M and 20M pitch HTD® belts are used in high performance drives in the machine tool, paper and textile industries and for applications in the processing and chemical industry.



FEATURES

- Special curvilinear tooth design substantially improves stress distribution and allows higher overall loading.
- Precisely formed and accurately spaced elastomeric teeth ensure smooth engagement with the pulley grooves.
- Fibreglass tensile cords provide necessary strength, excellent flex life plus high resistance to elongation.
- The durable backing protects against environmental pollution. It also protects against frictional wear if power is transmitted from the back of the belt.
- Tough nylon facing protects the tooth surface.

BENEFITS

- 3M and 5M pitch belts: for speeds up to 20000 rpm and capacities up to 10 kW.
- 8M, 14M and 20M pitch belts: capacities up to 1000 kW.
- Positive slip-proof engagement.
- Wide speed range.
- Constant driven speeds.
- Efficiencies up to 99%.
- Compact design. High flexibility allows the use of very small pulleys (outside pulley diameters from 8.79 mm).
- Long trouble-free service life.

POWERGRIP® HTD® SYSTEM SPECIFICATIONS

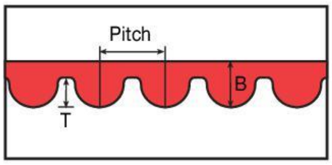
POWERGRIP® HTD® BELT DIMENSIONS

The three principal dimensions of a PowerGrip® HTD® belt are
- pitch;
- pitch length;
- width.

Belt pitch is the distance in millimetres between two adjacent tooth centres as measured on the pitch line of the belt. Belt pitch length is the total length (circumference) in millimetres as measured along the pitch line. The theoretical pitch line of a PowerGrip® HTD® belt lies within the tensile member.
Gates PowerGrip® HTD® belts are made in five stock pitches.

REFERENCE DIMENSIONS

	Pitch mm	T mm	B mm
3M	3.0	1.17	2.41
5M	5.0	2.08	3.81
8M	8.0	3.40	5.60
14M	14.0	6.00	10.00
20M	20.0	8.40	13.20



Gates PowerGrip® HTD® belt sizes are listed on pages 15-17. These tables list the belt lengths & pitch codes, pitch lengths and number of teeth. On these pages you will also find the standard widths. Using these tables, you will have all the information to complete the PowerGrip® HTD® ordering code.

Example: HTD 1040 8M 30
HTD .... PowerGrip® HTD®
1040 .... Pitch length (mm)
8M .... Pitch 8 mm
30 .... Belt width (mm)

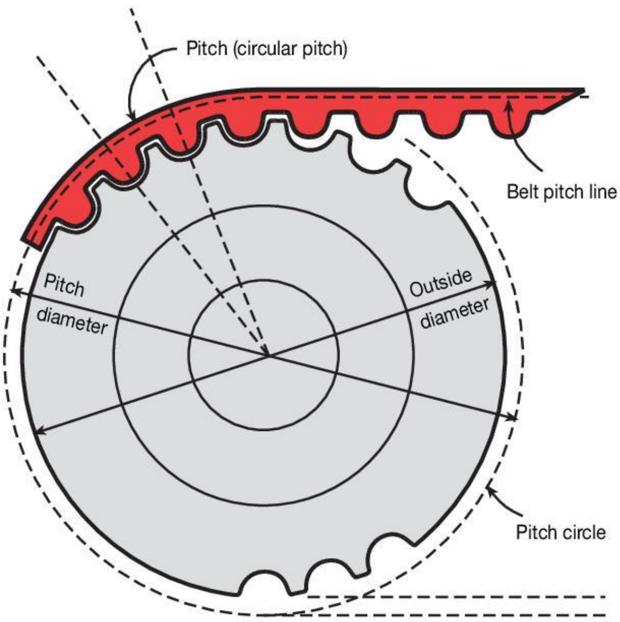
POWERGRIP® HTD® PULLEY DIMENSIONS

The three principal dimensions of a pulley are
- pitch;
- number of grooves;
- belt width.

On the pulley, pitch is the distance between groove centres and is measured on the pulley's pitch circle. The pitch circle of the pulley coincides with the pitch line of the belt engaging with it. The pulley's pitch diameter is always greater than its outside diameter.

A given PowerGrip® HTD® belt must be run on pulleys of the same pitch. Pulleys for PowerGrip® HTD® belts are made in 3, 5, 8, 14 and 20 mm pitches.
Standard pulley diameters are listed on pages 149-151. These tables list the number of grooves, the flange diameter and the outside diameter. On these pages you will also find the belt and pulley widths. Using these tables, you will have all the information to complete the pulley ordering code.

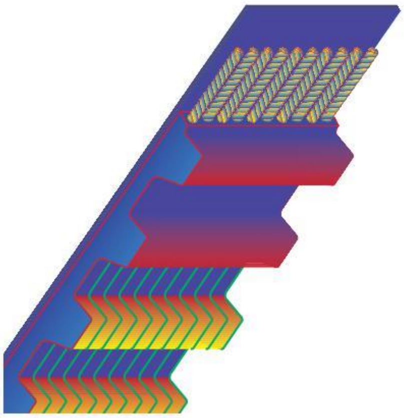
Example: P48-8M-50
P48 .... Pulley designation (P) and number of grooves (48)
8M .... Pitch 8 mm
50 .... Belt width (mm)



1

POWERGRIP® CLASSICAL SYNCHRONOUS BELT COMPONENTS AND BENEFITS

Gates classical synchronous PowerGrip® belts offer a maintenance-free and economical alternative to conventional drives like chains and gears. Applications range from minimum drives (printers) to heavy-duty machinery (oil pumps, etc).



FEATURES

- Trapezoidal tooth profile.
- Accurately spaced elastomeric teeth ensure smooth engagement with the pulley grooves.
- Fibreglass tensile cords provide strength, excellent flex life and high resistance to elongation.
- Durable backing protects against environmental pollution. It also protects against frictional wear if power is transmitted from the back of the belt.
- Tough nylon facing protects the tooth surface. This facing, after long service, becomes highly polished.

BENEFITS

- Power transmission of up to 150 kW and speeds of up to 10000 rpm (up to 20000 rpm for MXL pitch).
- Positive slip-proof engagement.
- Constant angular velocity.
- Low bearing load because of freedom of high tension.
- Maintenance-free continuity of operation.
- Wide range of load capacities and speed ratios.
- Economical operation.

POWERGRIP® SYSTEM SPECIFICATIONS

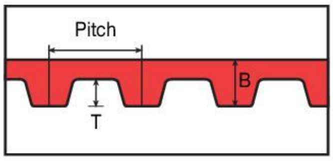
POWERGRIP® BELT DIMENSIONS

The three principal dimensions of a PowerGrip® belt are
- pitch;
- pitch length;
- width.

Belt pitch is the distance in inches between two adjacent tooth centres as measured on the pitch line of the belt. Belt pitch length is the total length (circumference) as measured along the pitch line. The theoretical pitch line of a PowerGrip® belt lies within the tensile member.
Gates PowerGrip® classical belts are made in six pitches according to ISO 5296: MXL, XL, L, H, XH and XXH.

REFERENCE DIMENSIONS

	Pitch inch	T mm	B mm
MXL	0.08	0.51	1.14
XL	1/5	1.27	2.3
L	3/8	1.91	3.5
H	1/2	2.29	4.0
XH	7/8	6.35	11.4
XXH	1 1/4	9.53	15.2



Gates PowerGrip® timing belt sizes are listed on pages 18-20. These tables list the belt lengths & pitch designation, pitch lengths and number of teeth. On these pages you will also find the standard widths. Using these tables, you will have all the information to complete the PowerGrip® timing belt ordering code.

Example: 600 H 200
600 .... Pitch length 60" (1524.0 mm)
H............. Pitch 1/2" (12.7 mm)
200 .... Belt width 2.0" (50.8 mm)

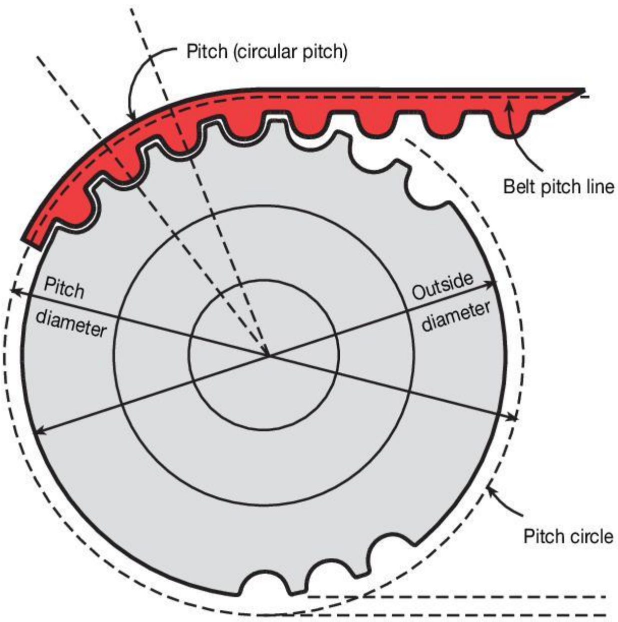
POWERGRIP® PULLEY DIMENSIONS

The three principal dimensions of a pulley are
- pitch;
- number of grooves;
- belt width.

On the pulley, pitch is the distance between groove centres and is measured on the pulley's pitch circle. The pitch circle of the pulley coincides with the pitch line of the belt engaging with it. The pulley's pitch diameter is always greater than its outside diameter.

A given PowerGrip® timing belt must be run on pulleys of the same pitch, so pulleys for PowerGrip® belts are made in MXL, XL, L, H, XH and XXH pitches. Standard pulley diameters are listed on pages 152-155. These tables list the number of grooves, the flange diameter and the outside diameter. On these pages you will also find the belt and pulley widths. Using these tables, you will have all the information to complete the pulley ordering code.

Example: P12-XL-050
P12......... Pulley designation (P) and number of grooves (12)
XL .... Pitch 1/5"
050 .... Belt width 1/2"



1

POWERGRIP® HTD® BELT SIZES

5M — Pitch: 5 mm

Length and pitch designation	Pitch length mm	Number of teeth
535-5M	**535**	**107**
550-5M	**550**	**110**
560-5M	**560**	**112**
565-5M	565	113
575-5M	**575**	**115**
580-5M	**580**	**116**
600-5M	**600**	**120**
610-5M	**610**	**122**
615-5M	**615**	**123**
635-5M	**635**	**127**
640-5M	**640**	**128**
645-5M	**645**	**129**
665-5M	**665**	**133**
670-5M	**670**	**134**
695-5M	695	139
700-5M	**700**	**140**
710-5M	710	142
720-5M	720	144
740-5M	**740**	**148**
750-5M	**750**	**150**
755-5M	755	151
770-5M	770	154
775-5M	775	155
800-5M	**800**	**160**
825-5M	825	165
835-5M	**835**	**167**
860-5M	**860**	**172**
870-5M	870	174
890-5M	890	178
900-5M	**900**	**180**
925-5M	925	185
935-5M	935	187
940-5M	940	188
950-5M	**950**	**190**
965-5M	965	193
980-5M	**980**	**196**
1000-5M	1000	200
1025-5M	1025	205
1035-5M	1035	207
1050-5M	**1050**	**210**
1100-5M	**1100**	**220**
1125-5M	**1125**	**225**
1135-5M	1135	227
1175-5M	1175	235
1200-5M	**1200**	**240**
1225-5M	1225	245
1270-5M	1270	254
1350-5M	**1350**	**270**
1380-5M	1380	276
1420-5M	1420	284

5M — Pitch: 5 mm

Length and pitch designation	Pitch length mm	Number of teeth
1595-5M	1595	319
1690-5M	**1690**	**338**
1790-5M	**1790**	**358**
1870-5M	1870	374
2100-5M	**2100**	**420**
2350-5M	**2350**	**470**

Available in widths of 9 mm, 15 mm and 25 mm.

8M — Pitch: 8 mm

Length and pitch designation	Pitch length mm	Number of teeth
264-8M	264	33
320-8M	320	40
376-8M	376	47
384-8M	384	48
424-8M	**424**	**53**
480-8M	**480**	**60**
512-8M	512	64
520-8M	520	65
560-8M	**560**	**70**
576-8M	576	72
600-8M	**600**	**75**
608-8M	608	76
624-8M	624	78
640-8M	**640**	**80**
656-8M	656	82
720-8M	**720**	**90**
760-8M	**760**	**95**
776-8M	776	97
800-8M	**800**	**100**
856-8M	856	107
880-8M	**880**	**110**
912-8M	912	114
920-8M	**920**	**115**
960-8M	**960**	**120**
968-8M	968	121
976-8M	**976**	**122**
1000-8M	**1000**	**125**
1040-8M	1040	130
1064-8M	**1064**	**133**
1080-8M	**1080**	**135**
1120-8M	**1120**	**140**
1128-8M	1128	141
1160-8M	**1160**	**145**
1176-8M	1176	147
1200-8M	**1200**	**150**
1216-8M	1216	152

8M — Pitch: 8 mm

Length and pitch designation	Pitch length mm	Number of teeth
1224-8M	**1224**	**153**
1256-8M	**1256**	**157**
1264-8M	**1264**	**158**
1280-8M	**1280**	**160**
1304-8M	1304	163
1360-8M	**1360**	**170**
1424-8M	1424	178
1432-8M	1432	179
1440-8M	**1440**	**180**
1512-8M	1512	189
1520-8M	**1520**	**190**
1552-8M	**1552**	**194**
1584-8M	1584	198
1600-8M	**1600**	**200**
1696-8M	1696	212
1728-8M	1728	216
1760-8M	**1760**	**220**
1800-8M	**1800**	**225**
1896-8M	1896	237
1904-8M	1904	238
2000-8M	**2000**	**250**
2080-8M	**2080**	**260**
2200-8M	2200	275
2240-8M	**2240**	**280**
2272-8M	**2272**	**284**
2400-8M	**2400**	**300**
2504-8M	2504	313
2600-8M	**2600**	**325**
2800-8M	**2800**	**350**

Available in widths of 20 mm, 30 mm, 50 mm and 85 mm.

Preferred sizes are printed in bold.

POWERGRIP® HTD® BELT SIZES

1

14M **Pitch: 14 mm**

Length and pitch designation	Pitch length mm	Number of teeth
784-14M	784	56
826-14M	826	59
924-14M	924	66
966-14M	**966**	**69**
1092-14M	1092	78
1190-14M	**1190**	**85**
1400-14M	**1400**	**100**
1610-14M	**1610**	**115**
1778-14M	**1778**	**127**
1890-14M	**1890**	**135**
2100-14M	**2100**	**150**
2310-14M	**2310**	**165**
2450-14M	**2450**	**175**
2590-14M	**2590**	**185**
2800-14M	**2800**	**200**
3150-14M	**3150**	**225**
3500-14M	**3500**	**250**
3850-14M	**3850**	**275**
4004-14M	4004	286
4326-14M	4326	309
4578-14M	4578	327

Available in widths of 40 mm, 55 mm, 85 mm, 115 mm and 170 mm.

20M **Pitch: 20 mm**

Length and pitch designation	Pitch length mm	Number of teeth
2000-20M	2000	100
2500-20M	2500	125
3400-20M	3400	170
3800-20M	3800	190
4200-20M	4200	210
4600-20M	4600	230
5000-20M	5000	250
5200-20M	5200	260
5400-20M	5400	270
5600-20M	5600	280
5800-20M	5800	290
6000-20M	6000	300
6200-20M	6200	310
6400-20M	6400	320
6600-20M	6600	330

Available in widths of 115 mm, 170 mm, 230 mm, 290 mm and 340 mm.

Preferred sizes are printed in bold.

BELT DRIVE SELECTION PROCEDURE

Before designing a synchronous belt drive, you need to determine and tabulate the following drive requirements:

1. Power requirement and type of driveN machine
2. The rpm of the driveR machine
3. The rpm of the driveN machine
4. The approximate centre distance for the drive
5. Hours per day operation.

To select a Gates PowerGrip® GT3, PowerGrip® HTD® or PowerGrip® belt drive, you need to complete the following steps:

STEP 1

DETERMINE THE SERVICE FACTOR

Service life of a belt drive depends on the specific use and function. By selecting the appropriate service life for a drive and designing it accordingly, you will obtain the most economical drive for your specific application. If the drive conditions are unknown, then the following classification guide will assist in the selection of the appropriate service factor.
For an idler, add 0.2 to the basic service factor. For intermittent or seasonal operation, deduct 0.2 from the basic service factor.
For speed-up drives, add to the basic service factor an additional factor as given in the table.

Speed-up ratio range	Additional factor
1 to 1.24	none
1.25 to 1.74	0.1
1.75 to 2.49	0.2
2.50 to 3.49	0.3
3.50 and over	0.4

Additional service factors are required for unusual conditions such as load reversal, heavy shock, plugged motor stop, electric brake. These should be determined by a Gates transmission specialist.
Any change in the service factor affects the entire calculation. For the majority of drive applications, the service factors here are adequate. It must be recognised, however, that these factors are not a substitute for judgement. You may find it practical to adjust the service factor to conform with your knowledge of the drive conditions and their severity.

STEP 2

CALCULATE THE DESIGN POWER

Design power = service factor x power requirement

A. To calculate the design power it is necessary to determine the service factor for the drive. Using the service factor chart on page 22, determine the type of driveR machine.

B. Using the service factor chart, determine the service factor for the driveN machine and the type of operational service.

C. Multiply the power requirement of the drive by the service factor you have selected. This gives you the design power for use in designing the drive.

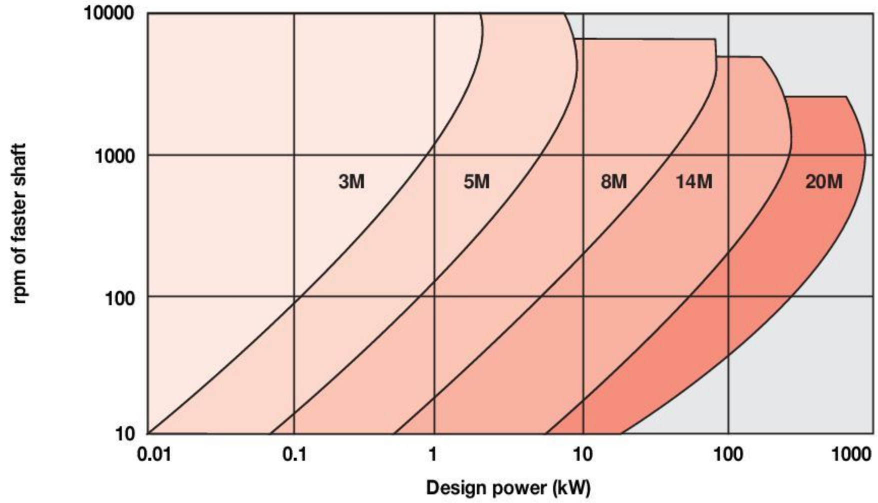
2

SERVICE FACTOR CHART

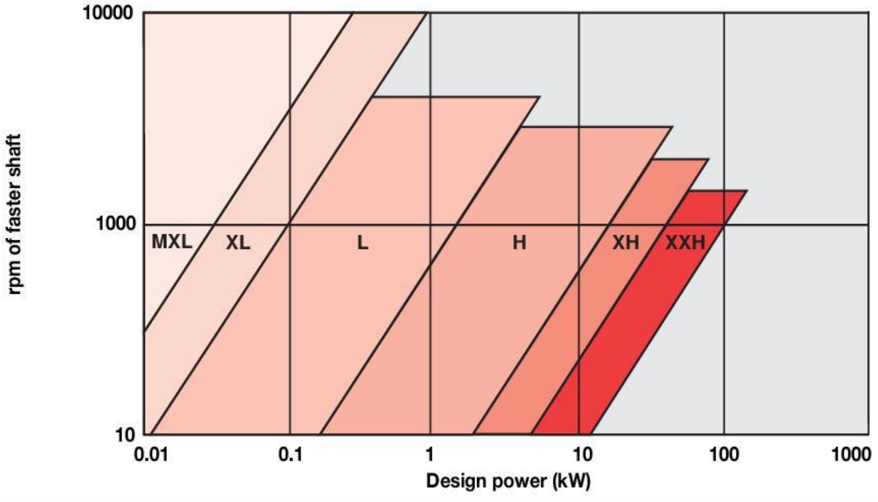
DRIVE N MACHINE	DRIVE R					
The driveN machines listed below are representative samples only. Select a driveN machine whose load characteristics most closely approximate those of the machine being considered.	AC motors: normal torque, squirrel cage, synchronous, split phase, inverter controlled. DC motors: shunt wound, stepper motors. Engines: multiple cylinder internal combustion.			AC motors: high torque, high slip, repulsion induction, single phase, series wound, slip ring. DC motors: series wound, compound wound, servo motors. Engines: single cylinder internal combustion. Line shafts. Clutches.		
	Intermittent service	Normal service	Continuous service	Intermittent service	Normal service	Continuous service
	3-8 hours daily or seasonal	8-16 hours daily	16-24 hours daily	3-8 hours daily or seasonal	8-16 hours daily	16-24 hours daily
Display equipment. Dispensing equipment. Instrumentation. Measuring equipment. Medical equipment. Office equipment. Projection equipment.	1.0	1.2	1.4	1.2	1.4	1.6
Appliances. Sweepers. Sewing machines. Screens: oven, drum, conical. Woodworking equipment (light): band saws, drills, lathes.	1.1	1.3	1.5	1.3	1.5	1.7
Agitators for liquids. Conveyors: belt, light package. Drill presses. Lathes. Saws. Laundry machinery. Woodworking equipment (heavy): circular saws, jointers, planers.	1.2	1.4	1.6	1.6	1.8	2.0
Agitators for semi-liquids. Centrifugal compressors. Conveyor belt: ore, coal, sand. Dough mixers. Line shafts. Machine tools: grinders, shapers, boring mills, milling machines. Paper machinery (except pulpers): presses, punches, shears. Printing machinery. Pumps: centrifugal, gear. Screens: revolving, vibratory.	1.3	1.5	1.7	1.6	1.8	2.0
Brick machinery (except pug mills). Conveyors: apron, pan, bucket, elevator. Extractors. Washers. Fans. Centrifugal blowers. Generators and exciters. Hoists. Rubber calender. Mills. Extruders.	1.4	1.6	1.8	1.8	2.0	2.2
Centrifuges. Screw conveyors. Hammer mills. Paper pulpers. Textile machinery.	1.5	1.7	1.9	1.9	2.1	2.3
Blowers: positive displacement. Mine fans. Pulverisers.	1.6	1.8	2.0	2.0	2.2	2.4
Reciprocating compressors. Crushers: gyratory, jaw, roll. Mills: ball, rod, pebble, etc. Pumps: reciprocating. Saw mill equipment.	1.7	1.9	2.1	2.1	2.3	2.5

These service factors are adequate for most belt drive applications. Note that service factors cannot be substituted for good engineering judgement. Service factors may be adjusted based upon an understanding of the severity of actual drive operating conditions.

POWERGRIP® HTD® BELT PITCH SELECTION GUIDE



POWERGRIP® CLASSICAL TIMING BELT PITCH SELECTION GUIDE



STEP 4

SELECT THE PULLEY COMBINATION, BELT LENGTH AND CENTRE DISTANCE

Locate the correct centre distance table for the belt you selected (pages 28-127).

For standard and non-standard motor speeds:

A. Calculate the speed ratio by dividing the rpm of the faster shaft by the rpm of the slower shaft. In the centre distance tables, refer to the column headed speed ratio. Locate the speed ratio nearest to your requirements.
B. For the speed ratio selected, note the number of grooves and the pitch diameter of each pulley. If there are several combinations close to your requirements, you may want to consider more than one combination in your drive selection.
C. Reading further to the right on the same line, locate and record the centre distance nearest to your requirements. The belt length code is given at the top of that column in mm. Note these values.

Alternative method to establish the belt length/centre distance

The nomograph on page 26 provides a quick, effective method for determining the nominal centre distance and belt length of a drive and converting these nominal values into design values. The values of belt length and centre distance obtained using this nomograph are approximate and only intended for use in applications where reasonable centre distance adjustment is possible.
The nomograph is based on the number of pitches rather than on actual diameters and lengths.

Hence:

Pulley size

N = number of grooves in large pulley
n = number of grooves in small pulley

Belt length

$$Nb = \frac{\text{belt length}}{\text{pitch}} \text{ (number of pitches)}$$

Centre distance

$$Nc = \frac{\text{centre distance}}{\text{pitch}} \text{ (number of pitches)}$$

To establish required belt length

a. Calculate the values N + n and Nc.
b. Place a straight edge across the nomograph connecting these two points.
c. Read off the Nb value and multiply it by the pitch to give the nominal belt length in mm.
d. Select the nearest suitable belt length using the size listings on pages 13-20 .
e. Convert this belt length to pitches and re-apply this value to the nomograph to obtain the actual centre distance (Nc).

This method will give sufficient accuracy for drives having a speed ratio of 3:1 or less. If the ratio is greater than 3:1 then a correction will be necessary.

$$\text{Corrected centre distance} = Nc - \frac{N - n}{1.114 \times Nc}$$

If there is limited room for centre distance adjustment

Establish the belt length in millimetres as previously outlined. Calculate the centre distance using the following formula:

Pulley centre distance

$$C = \frac{K + \sqrt{K^2 - 32\,(D - d)^2}}{16}$$

Where $K = 4L - 6.28\,(D + d)$
D = pitch circle diameter large pulley (mm)
d = pitch circle diameter small pulley (mm)
L = belt length (mm)

Fixed centre applications

For applications where no centre distance adjustment is possible, contact Gates' application engineers.

Exact values may be calculated from the following:

1. Belt length (L)

$$L = 2C \sin\frac{\beta}{2} + \frac{\pi}{2}\left[(D + d) + \left(1 - \frac{\beta}{180}\right)(D - d)\right]$$

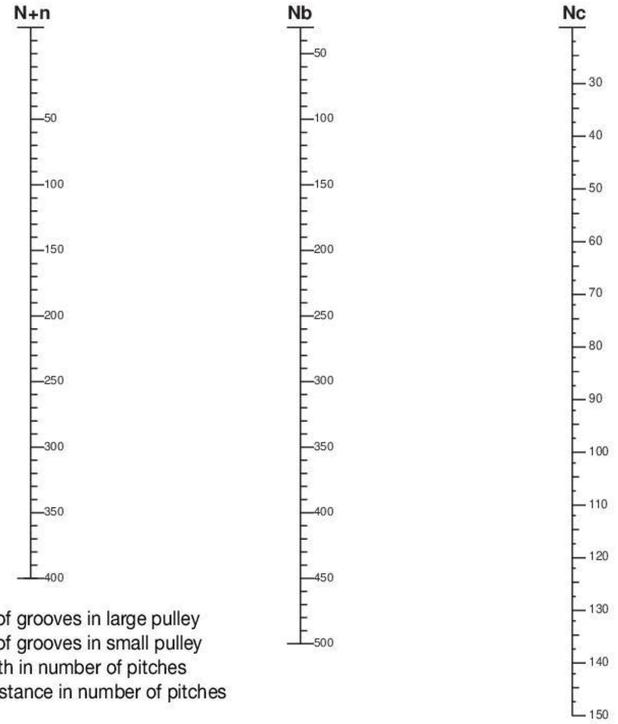
$$\text{Where } \beta = 2 \cos^{-1}\left(\frac{D - d}{2C}\right)$$

2. Centre distance (C)

$$C = \frac{1}{2 \sin\left(\frac{\beta}{2}\right)} \left\{L - \frac{\pi}{2}\left[(D - d) + \left(1 - \frac{\beta}{180}\right)(D - d)\right]\right\}$$

BELT DRIVE SELECTION PROCEDURE

DESIGN NOMOGRAPH



Where N = number of grooves in large pulley
n = number of grooves in small pulley
Nb = belt length in number of pitches
Nc = centre distance in number of pitches

STEP 5

SELECT THE BELT WIDTH

A. The tables on pages 128-147 show the power ratings for each belt which, when combined with the width factors, will give the rating for each belt width. The left hand column lists the rpm of the smaller pulley. The stock pulleys are listed across the top of the columns and are designated by the number of grooves and the pitch diameter. By reading down the first column to the speed of your faster shaft and across the line to the column headed by your smaller pulley, the power rating can be determined for any stock belt width.

IMPORTANT
The tables on pages 128-147 provide power ratings that are based on a minimum of six teeth in mesh. If less than six teeth are in mesh the power rating should be multiplied by the approximate teeth in mesh factor from the following table.

Use the following formula to establish the number of teeth in mesh:

$$\text{Teeth in mesh (T.I.M.)} = n \left[0.5 - \frac{(N - n)}{18.85 \times Nc}\right]$$

Teeth in mesh factor

Teeth in mesh	≥6	5	4	3	2
Factor	1	0.8	0.6	0.4	0.2

B. Select a stock belt width and determine the power rating as outlined in Step 5A. If the power rating is equal to or exceeds the design power found in Step 2, that belt width can be used. If not, move on to the next stock belt width and repeat this step. If the widest stock belt width for the pitch selected is still not acceptable, you may want to consider larger pulley diameters or a larger pitch belt if possible.

C. Where there are several pulley combinations which meet your drive requirements, the following rules of thumb may influence your choice.
a. The larger the pulley diameter, the narrower the belt.
b. Larger diameter pulleys typically reduce bearing and shaft loads.

BELT DRIVE SELECTION EXAMPLE

A centrifugal blower is to be driven by an AC Motor.
Drive requirements and characteristics are as follows:

Driver machine
Type: AC motor- normal torque
Power: 740 Watts
Speed: 2850 rpm
Shaft diameter: 19 mm

Driven machine
Type: Centrifugal blower
Power: 600 Watts (absorbed)
Speed: 6800 rpm
Shaft diameter: 12 mm

Drive conditions
Smooth uniform load
Operating 8 hrs/day, 5 days/week

Drive design limitation
Maximum driving pulley diameter = 75 mm
Shaft centres = 70 mm ± 5 mm
Idler: not requested

STEP 1

DETERMINE THE SERVICE FACTOR

From the service factor chart select the service factors which are applicable to the drive.
Basic service factor = 1.5
In this case additional factors must be added:
Speed up factor:
it is a speed increasing drive ratio: $\frac{6800}{2850}$

Additional factor = 0.2
Resultant service factor = 1.5 + 0.2
= 1.7

STEP 2

CALCULATE THE DESIGN POWER

a) Determine speed ratio
Driver speed = 2850 rpm
Driven speed = 6800 rpm
Speed ratio = 2.39 (speed increase)

b) Design power
Multiply the drive absorbed power by the service factor:
600W x 1.7 = 1020W

STEP 3

DETERMINE THE BELT PITCH

Refer to the belt pitch selection guides on pages 23-24. Use the design power of 1020W and the small pulley speed of 6800 rpm. The chart will show that these conditions give an intercept inside the 3MGT power envelope. Therefore a 3MGT drive is required.

STEP 4

SELECT THE PULLEY COMBINATION, BELT LENGTH AND CENTRE DISTANCE

a) Select pulleys
Check size limitation (see page 148).
Driven pulley max. dia. = 75 mm hence max.
Stock pulley = 3MR - 72S

Driven pulley shaft dia. = 12 mm hence min.
Driven pulley = 3MR - 30S

Bearing these limitations in mind, the stock pulley combination to give the speed ratio of 2.4 : 1 is 3MR - 72S : 3MR - 30S

b) Select belt length
Required centres = 70 ± 5 mm
Referring to centre distance table page 48, the most suitable will be the belt 300 - 3MGT which will give centres of 70.63 mm when combined with the above pulley selection.

Hence the pulley/belt combinations required will be:
pulleys: 3MR - 72S, 3MR - 30S
belt: 300 - 3MGT

STEP 5

SELECT BELT WIDTH

Selection is always based on the smallest pulley, i.e. 3MR - 30S running at 6840 rpm.

Refer to the 3MGT power ratings table on page 129 and note the ratings for the 30 groove pulley for 6000 and 8000 rpm.

Interpolate these ratings for a speed of 6840 rpm (i.e. 1920W).

This value is for a width of 6 mm.
Multiply by the width factor:

Width	Factor	Watts
6 mm	0.62	1190
9 mm	1	1920
15 mm	1.89	3629

Teeth in mesh factor
See page 26. Calculated value is 14 teeth in mesh. As this figure is greater than 5, the factor is 1. Hence the power rating is not changed.

Our design power requirement is 1020W, hence a belt width of 6 mm will be required.

The selected drive will therefore be:

Driver pulley: **3MR - 72S - 6**
Driven pulley: **3MR - 30S - 6**
Belt: **300 - 3MGT - 6**

CENTRE DISTANCE TABLES 8M & 8MGT

3

8 mm

Speed ratio	Number of grooves		Theoretical centre distance in mm									
			Belt length code designation in mm									
	DriveR	DriveN	480	560	600	640	720	800	880	960	1040	1120
1.00	22	22	152.0	192.0	212.0	232.0	272.0	312.0	352.0	392.0	432.0	472.0
1.00	24	24	144.0	184.0	204.0	224.0	264.0	304.0	344.0	384.0	424.0	464.0
1.00	26	26	136.0	176.0	196.0	216.0	256.0	296.0	336.0	376.0	416.0	456.0
1.00	28	28	128.0	168.0	188.0	208.0	248.0	288.0	328.0	368.0	408.0	448.0
1.00	30	30	120.0	160.0	180.0	200.0	240.0	280.0	320.0	360.0	400.0	440.0
1.00	32	32	112.0	152.0	172.0	192.0	232.0	272.0	312.0	352.0	392.0	432.0
1.00	36	36		136.0	156.0	176.0	216.0	256.0	296.0	336.0	376.0	416.0
1.00	40	40		120.0	140.0	160.0	200.0	240.0	280.0	320.0	360.0	400.0
1.00	44	44				144.0	184.0	224.0	264.0	304.0	344.0	384.0
1.00	48	48					168.0	208.0	248.0	288.0	328.0	368.0
1.00	56	56						176.0	216.0	256.0	296.0	336.0
1.00	64	64							184.0	224.0	264.0	304.0
1.05	38	40		124.0	144.0	164.0	204.0	244.0	284.0	324.0	364.0	404.0
1.06	32	34	108.0	148.0	168.0	188.0	228.0	268.0	308.0	348.0	388.0	428.0
1.06	34	36		140.0	160.0	180.0	220.0	260.0	300.0	340.0	380.0	420.0
1.06	36	38		132.0	152.0	172.0	212.0	252.0	292.0	332.0	372.0	412.0
1.07	28	30	124.0	164.0	184.0	204.0	244.0	284.0	324.0	364.0	404.0	444.0
1.08	24	26	140.0	180.0	200.0	220.0	260.0	300.0	340.0	380.0	420.0	460.0
1.08	26	28	132.0	172.0	192.0	212.0	252.0	292.0	332.0	372.0	412.0	452.0
1.09	22	24	148.0	188.0	208.0	228.0	268.0	308.0	348.0	388.0	428.0	468.0
1.10	40	44			131.9	151.9	191.9	231.9	272.0	312.0	352.0	392.0
1.11	36	40		127.9	147.9	167.9	207.9	247.9	288.0	328.0	368.0	408.0
1.12	34	38		135.9	155.9	175.9	215.9	255.9	296.0	336.0	376.0	416.0
1.13	30	34	111.9	151.9	171.9	191.9	231.9	272.0	312.0	352.0	392.0	432.0
1.13	64	72								207.8	247.8	287.8
1.13	32	36	103.9	143.9	163.9	183.9	223.9	264.0	304.0	344.0	384.0	424.0
1.14	28	32	119.9	159.9	179.9	199.9	239.9	280.0	320.0	360.0	400.0	440.0
1.14	56	64							199.7	239.8	279.8	319.8
1.15	26	30	127.9	167.9	187.9	207.9	247.9	288.0	328.0	368.0	408.0	448.0
1.16	38	44			135.8	155.8	195.9	235.9	275.9	315.9	355.9	395.9
1.17	24	28	135.9	175.9	195.9	215.9	255.9	296.0	336.0	376.0	416.0	456.0
1.18	22	26	143.9	183.9	203.9	223.9	264.0	304.0	344.0	384.0	424.0	464.0
1.18	34	40		131.8	151.8	171.8	211.9	251.9	291.9	331.9	371.9	411.9
1.19	32	38	99.7	139.8	159.8	179.8	219.9	259.9	299.9	339.9	379.9	419.9
1.20	30	36	107.7	147.8	167.8	187.8	227.9	267.9	307.9	347.9	387.9	427.9
1.20	40	48			123.6	143.6	183.7	223.8	263.8	303.8	343.8	383.9
1.21	28	34	115.7	155.8	175.8	195.9	235.9	275.9	315.9	355.9	395.9	435.9
1.22	36	44		119.6	139.6	159.7	199.7	239.8	279.8	319.8	359.9	399.9
1.23	26	32	123.8	163.8	183.8	203.9	243.9	283.9	323.9	363.9	403.9	443.9
1.25	24	30	131.8	171.8	191.8	211.9	251.9	291.9	331.9	371.9	411.9	451.9
1.25	32	40		135.6	155.7	175.7	215.8	255.8	295.8	335.8	375.9	415.9
1.25	64	80									231.1	271.2
1.26	38	48			127.4	147.4	187.6	227.6	267.7	307.7	347.8	387.8
1.27	22	28	139.8	179.8	199.9	219.9	259.9	299.9	339.9	379.9	419.9	459.9
1.27	30	38	103.5	143.6	163.7	183.7	223.8	263.8	303.8	343.8	383.9	423.9
1.27	44	56					159.3	199.4	239.5	279.6	319.6	359.7
1.29	28	36	111.5	151.7	171.7	191.7	231.8	271.8	311.8	351.9	391.9	431.9
1.29	34	44		123.3	143.4	163.5	203.6	243.7	283.7	323.7	363.8	403.8
1.31	26	34	119.6	159.7	179.7	199.7	239.8	279.8	319.8	359.9	399.9	439.9
1.33	24	32	127.6	167.7	187.7	207.8	247.8	287.8	327.8	367.9	407.9	447.9

Theoretical centre distance in mm									Number of grooves		Speed ratio
Belt length code designation in mm											
1200	1280	1440	1600	1760	1800	2000	2400	2800	DriveN	DriveR	
512.0	552.0	632.0	712.0	792.0	812.0	912.0	1112.0	1312.0	22	22	1.00
504.0	544.0	624.0	704.0	784.0	804.0	904.0	1104.0	1304.0	24	24	1.00
496.0	536.0	616.0	696.0	776.0	796.0	896.0	1096.0	1296.0	26	26	1.00
488.0	528.0	608.0	688.0	768.0	788.0	888.0	1088.0	1288.0	28	28	1.00
480.0	520.0	600.0	680.0	760.0	780.0	880.0	1080.0	1280.0	30	30	1.00
472.0	512.0	592.0	672.0	752.0	772.0	872.0	1072.0	1272.0	32	32	1.00
456.0	496.0	576.0	656.0	736.0	756.0	856.0	1056.0	1256.0	36	36	1.00
440.0	480.0	560.0	640.0	720.0	740.0	840.0	1040.0	1240.0	40	40	1.00
424.0	464.0	544.0	624.0	704.0	724.0	824.0	1024.0	1224.0	44	44	1.00
408.0	448.0	528.0	608.0	688.0	708.0	808.0	1008.0	1208.0	48	48	1.00
376.0	416.0	496.0	576.0	656.0	676.0	776.0	976.0	1176.0	56	56	1.00
344.0	384.0	464.0	544.0	624.0	644.0	744.0	944.0	1144.0	64	64	1.00
444.0	484.0	564.0	644.0	724.0	744.0	844.0	1044.0	1244.0	40	38	1.05
468.0	508.0	588.0	668.0	748.0	768.0	868.0	1068.0	1268.0	34	32	1.06
460.0	500.0	580.0	660.0	740.0	760.0	860.0	1060.0	1260.0	36	34	1.06
452.0	492.0	572.0	652.0	732.0	752.0	852.0	1052.0	1252.0	38	36	1.06
484.0	524.0	604.0	684.0	764.0	784.0	884.0	1084.0	1284.0	30	28	1.07
500.0	540.0	620.0	700.0	780.0	800.0	900.0	1100.0	1300.0	26	24	1.08
492.0	532.0	612.0	692.0	772.0	792.0	892.0	1092.0	1292.0	28	26	1.08
508.0	548.0	628.0	708.0	788.0	808.0	908.0	1108.0	1308.0	24	22	1.09
432.0	472.0	552.0	632.0	712.0	732.0	832.0	1032.0	1232.0	44	40	1.10
448.0	488.0	568.0	648.0	728.0	748.0	848.0	1048.0	1248.0	40	36	1.11
456.0	496.0	576.0	656.0	736.0	756.0	856.0	1056.0	1256.0	38	34	1.12
472.0	512.0	592.0	672.0	752.0	772.0	872.0	1072.0	1272.0	34	30	1.13
327.8	367.9	447.9	527.9	607.9	627.9	727.9	927.9	1128.0	72	64	1.13
464.0	504.0	584.0	664.0	744.0	764.0	864.0	1064.0	1264.0	36	32	1.13
480.0	520.0	600.0	680.0	760.0	780.0	880.0	1080.0	1280.0	32	28	1.14
359.9	399.9	479.9	559.9	639.9	659.9	759.9	959.9	1160.0	64	56	1.14
488.0	528.0	608.0	688.0	768.0	788.0	888.0	1088.0	1288.0	30	26	1.15
435.9	475.9	555.9	636.0	716.0	736.0	836.0	1036.0	1236.0	44	38	1.16
496.0	536.0	616.0	696.0	776.0	796.0	896.0	1096.0	1296.0	28	24	1.17
504.0	544.0	624.0	704.0	784.0	804.0	904.0	1104.0	1304.0	26	22	1.18
451.9	491.9	571.9	652.0	732.0	752.0	852.0	1052.0	1252.0	40	34	1.18
459.9	499.9	579.9	660.0	740.0	760.0	860.0	1060.0	1260.0	38	32	1.19
467.9	507.9	588.0	668.0	748.0	768.0	868.0	1068.0	1268.0	36	30	1.20
423.9	463.9	543.9	623.9	703.9	723.9	823.9	1023.9	1224.0	48	40	1.20
475.9	515.9	596.0	676.0	756.0	776.0	876.0	1076.0	1276.0	34	28	1.21
439.9	479.9	559.9	639.9	719.9	739.9	839.9	1040.0	1240.0	44	36	1.22
483.9	523.9	604.0	684.0	764.0	784.0	884.0	1084.0	1284.0	32	26	1.23
491.9	531.9	612.0	692.0	772.0	792.0	892.0	1092.0	1292.0	30	24	1.25
455.9	495.9	575.9	655.9	735.9	755.9	855.9	1056.0	1256.0	40	32	1.25
311.3	351.4	431.5	511.6	591.6	611.7	711.7	911.8	1111.8	80	64	1.25
427.8	467.8	547.9	627.9	707.9	727.9	827.9	1027.9	1227.9	48	38	1.26
499.9	539.9	620.0	700.0	780.0	800.0	900.0	1100.0	1300.0	28	22	1.27
463.9	503.9	583.9	663.9	743.9	763.9	863.9	1064.0	1264.0	38	30	1.27
399.7	439.7	519.8	599.8	679.8	699.8	799.9	999.9	1199.9	56	44	1.27
471.9	511.9	591.9	671.9	751.9	771.9	871.9	1072.0	1272.0	36	28	1.29
443.8	483.8	563.9	643.9	723.9	743.9	843.9	1043.9	1243.9	44	34	1.29
479.9	519.9	599.9	679.9	759.9	779.9	879.9	1080.0	1280.0	34	26	1.31
487.9	527.9	607.9	687.9	767.9	787.9	887.9	1088.0	1288.0	32	24	1.33

CENTRE DISTANCE TABLES 8M & 8MGT

Speed ratio	Number of grooves		Theoretical centre distance in mm									
			Belt length code designation in mm									
	DriveR	DriveN	480	560	600	640	720	800	880	960	1040	1120
1.33	30	40	99.2	139.4	159.5	179.5	219.6	259.7	299.7	339.8	379.8	419.8
1.33	36	48			131.1	151.2	191.4	231.5	271.6	311.6	351.7	391.7
1.33	48	64						174.8	215.0	255.2	295.3	335.4
1.36	22	30	135.6	175.7	195.7	215.8	255.8	295.8	335.8	375.9	415.9	455.9
1.36	28	38	107.2	147.4	167.5	187.6	227.6	267.7	307.7	347.8	387.8	427.8
1.38	26	36	115.3	155.5	175.5	195.6	235.7	275.7	315.7	355.8	395.8	435.8
1.38	32	44		127.1	147.2	167.3	207.4	247.5	287.6	327.6	367.7	407.7
1.40	40	56					166.8	207.0	247.2	287.3	327.4	367.4
1.41	34	48		114.6	134.8	155.0	195.2	235.3	275.4	315.5	355.6	395.6
1.41	64	90									209.4	249.8
1.42	24	34	123.3	163.5	183.6	203.6	243.7	283.7	323.7	363.8	403.8	443.8
1.43	28	40	102.9	143.2	163.3	183.4	223.5	263.6	303.6	343.7	383.7	423.7
1.45	22	32	131.4	171.5	191.6	211.6	251.7	291.7	331.8	371.8	411.8	451.8
1.46	26	38	110.9	151.2	171.3	191.4	231.5	271.6	311.6	351.7	391.7	431.7
1.47	30	44		130.8	150.9	171.1	211.2	251.4	291.5	331.5	371.6	411.6
1.47	38	56				130.0	170.5	210.8	251.0	291.1	331.2	371.3
1.50	24	36	119.0	159.3	179.3	199.4	239.5	279.6	319.6	359.7	399.7	439.7
1.50	32	48		118.2	138.5	158.7	199.0	239.1	279.3	319.4	359.4	399.5
1.50	48	72							197.6	238.0	278.3	318.5
1.54	26	40	106.5	146.9	167.0	187.2	227.3	267.4	307.5	347.5	387.6	427.6
1.55	22	34	127.1	167.3	187.4	207.4	247.5	287.6	327.6	367.7	407.7	447.7
1.56	36	56				133.6	174.1	214.5	254.7	294.9	335.0	375.1
1.57	28	44		134.5	154.7	174.8	215.0	255.2	295.3	335.4	375.4	415.5
1.58	24	38	114.6	155.0	175.1	195.2	235.3	275.4	315.5	355.6	395.6	435.6
1.60	30	48		121.8	142.1	162.4	202.7	242.9	283.1	323.2	363.3	403.3
1.60	40	64					148.9	189.5	230.0	270.3	310.5	350.7
1.61	56	90									223.8	264.4
1.64	22	36	122.7	163.0	183.1	203.2	243.3	283.4	323.5	363.6	403.6	443.6
1.64	44	72						164.1	204.9	245.4	285.8	326.0
1.65	34	56				137.1	177.8	218.2	258.5	298.7	338.8	379.0
1.67	24	40	110.1	150.6	170.8	190.9	231.1	271.2	311.3	351.4	391.5	431.5
1.67	48	80							179.4	220.2	260.8	301.2
1.68	38	64					152.4	193.2	233.7	274.0	314.3	354.5
1.69	26	44		138.1	158.3	178.5	218.8	259.0	299.1	339.2	379.3	419.4
1.71	28	48		125.4	145.8	166.0	206.4	246.7	286.9	327.0	367.1	407.2
1.73	22	38	118.2	158.7	178.8	199.0	239.1	279.3	319.4	359.4	399.5	439.5
1.75	32	56				140.7	181.4	221.9	262.2	302.5	342.6	382.8
1.75	64	112										
1.78	36	64					155.9	196.8	237.3	277.7	318.0	358.2
1.80	40	72						171.1	212.1	252.7	293.2	333.5
1.82	22	40	113.7	154.3	174.5	194.6	234.9	275.0	315.2	355.3	395.3	435.4
1.82	44	80							186.3	227.4	268.1	308.6
1.83	24	44	100.8	141.7	162.0	182.2	222.5	262.8	302.9	343.1	383.2	423.2
1.85	26	48		128.9	149.4	169.7	210.1	250.4	290.6	330.8	370.9	411.0
1.87	30	56			123.5	144.2	185.0	225.6	265.9	306.2	346.4	386.6
1.88	48	90								196.7	238.0	278.9
1.88	34	64					159.4	200.3	241.0	281.4	321.7	362.0
1.89	38	72						174.6	215.6	256.3	296.8	337.2
2.00	22	44	104.2	145.3	165.6	185.9	226.3	266.5	306.7	346.9	387.0	427.1
2.00	24	48		132.5	152.9	173.3	213.8	254.2	294.4	334.6	374.8	414.9

CENTRE DISTANCE TABLES 8M & 8MGT

Theoretical centre distance in mm									Number of grooves		Speed ratio
Belt length code designation in mm											
1200	1280	1440	1600	1760	1800	2000	2400	2800	DriveN	DriveR	
459.8	499.8	579.9	659.9	739.9	759.9	859.9	1059.9	1259.9	40	30	1.33
431.7	471.8	551.8	631.8	711.8	731.8	831.9	1031.9	1231.9	48	36	1.33
375.4	415.5	495.6	575.6	655.7	675.7	775.7	975.8	1175.8	64	48	1.33
495.9	535.9	615.9	695.9	775.9	795.9	895.9	1096.0	1296.0	30	22	1.36
467.8	507.8	587.9	667.9	747.9	767.9	867.9	1067.9	1267.9	38	28	1.36
475.8	515.8	595.9	675.9	755.9	775.9	875.9	1075.9	1275.9	36	26	1.38
447.7	487.8	567.8	647.8	727.8	747.8	847.9	1047.9	1247.9	44	32	1.38
407.5	447.5	527.6	607.7	687.7	707.7	807.7	1007.8	1207.8	56	40	1.40
435.6	475.7	555.7	635.8	715.8	735.8	835.8	1035.8	1235.9	48	34	1.41
290.1	330.3	410.7	490.9	571.0	591.1	691.2	891.4	1091.5	90	64	1.41
483.8	523.8	603.9	683.9	763.9	783.9	883.9	1083.9	1283.9	34	24	1.42
463.7	503.8	583.8	663.8	743.8	763.8	863.9	1063.9	1263.9	40	28	1.43
491.8	531.8	611.9	691.9	771.9	791.9	891.9	1091.9	1291.9	32	22	1.45
471.8	511.8	591.8	671.8	751.8	771.8	871.9	1071.9	1271.9	38	26	1.46
451.6	491.7	571.7	651.8	731.8	751.8	851.8	1051.8	1251.9	44	30	1.47
411.4	451.4	531.5	611.6	691.6	711.6	811.7	1011.7	1211.8	56	38	1.47
479.8	519.8	599.8	679.8	759.8	779.9	879.9	1079.9	1279.9	36	24	1.50
439.5	479.6	559.6	639.7	719.7	739.7	839.8	1039.8	1239.8	48	32	1.50
358.7	398.8	479.0	559.2	639.3	659.3	759.4	959.5	1159.6	72	48	1.50
467.7	507.7	587.7	667.8	747.8	767.8	867.8	1067.9	1267.9	40	26	1.54
487.8	527.8	607.8	687.8	767.8	787.9	887.9	1087.9	1287.9	34	22	1.55
415.2	455.3	535.4	615.5	695.5	715.5	815.6	1015.7	1215.7	56	36	1.56
455.5	495.6	575.6	655.7	735.7	755.7	855.8	1055.8	1255.8	44	28	1.57
475.7	515.7	595.7	675.8	755.8	775.8	875.8	1075.9	1275.9	38	24	1.58
443.4	483.5	563.5	643.6	723.6	743.6	843.7	1043.7	1243.8	48	30	1.60
390.8	430.9	511.1	591.2	671.3	691.3	791.4	991.5	1191.6	64	40	1.60
304.9	345.3	425.8	506.1	586.4	606.5	706.7	907.0	1107.2	90	56	1.61
483.7	523.7	603.7	683.8	763.8	783.8	883.8	1083.9	1283.9	36	22	1.64
366.3	406.4	486.7	566.9	647.0	667.0	767.2	967.3	1167.5	72	44	1.64
419.1	459.1	539.3	619.4	699.4	719.5	819.5	1019.6	1219.7	56	34	1.65
471.6	511.6	591.6	671.7	751.7	771.7	871.8	1071.8	1271.8	40	24	1.67
341.6	381.8	462.2	542.5	622.7	642.7	742.9	943.1	1143.3	80	48	1.67
394.6	434.7	514.9	595.1	675.2	695.2	795.3	995.4	1195.5	64	38	1.68
459.4	499.5	579.5	659.6	739.6	759.7	859.7	1059.8	1259.8	44	26	1.69
447.3	487.3	567.4	647.5	727.6	747.6	847.6	1047.7	1247.7	48	28	1.71
479.6	519.6	599.7	679.7	759.7	779.7	879.8	1079.8	1279.8	38	22	1.73
422.9	463.0	543.1	623.3	703.3	723.4	823.4	1023.5	1223.6	56	32	1.75
240.2	281.3	362.8	443.8	524.4	544.6	645.1	845.8	1046.2	112	64	1.75
398.4	438.6	518.8	598.9	679.1	699.1	799.2	999.4	1199.5	64	36	1.78
373.8	414.0	494.3	574.6	654.7	674.8	774.9	975.1	1175.3	72	40	1.80
475.4	515.5	595.6	675.6	755.7	775.7	875.7	1075.8	1275.8	40	22	1.82
349.0	389.3	469.8	550.1	630.3	650.4	750.6	950.9	1151.1	80	44	1.82
463.3	503.4	583.4	663.5	743.6	763.6	863.6	1063.7	1263.7	44	24	1.83
451.1	491.2	571.3	651.4	731.5	751.5	851.5	1051.6	1251.7	48	26	1.85
426.7	466.8	547.0	627.1	707.2	727.2	827.3	1027.5	1227.6	56	30	1.87
319.5	360.0	440.8	521.3	601.6	621.7	722.0	922.4	1122.7	90	48	1.88
402.2	442.3	522.6	602.8	682.9	703.0	803.1	1003.3	1203.4	64	34	1.88
377.5	417.8	498.1	578.4	658.6	678.6	778.8	979.0	1179.2	72	38	1.89
467.2	507.2	587.3	667.4	747.5	767.5	867.5	1067.6	1267.7	44	22	2.00
455.0	495.1	575.2	655.3	735.4	755.4	855.5	1055.6	1255.6	48	24	2.00

3

8 mm

CENTRE DISTANCE TABLES 8M & 8MGT

3

8 mm

Speed ratio	Number of grooves		Theoretical centre distance in mm									
			Belt length code designation in mm									
	DriveR	DriveN	480	560	600	640	720	800	880	960	1040	1120
2.00	28	56			127.0	147.7	188.6	229.2	269.6	309.9	350.2	390.4
2.00	32	64					162.9	203.9	244.6	285.1	325.4	365.7
2.00	36	72						178.1	219.2	259.9	300.5	340.9
2.00	40	80							193.2	234.4	275.3	315.9
2.00	56	112										
2.05	44	90								203.5	245.0	286.0
2.11	38	80							196.7	238.0	278.9	319.5
2.12	34	72						181.5	222.7	263.5	304.1	344.6
2.13	30	64					166.3	207.5	248.2	288.7	329.1	369.5
2.15	26	56			130.4	151.1	192.2	232.9	273.3	313.7	353.9	394.1
2.18	22	48		135.9	156.5	176.9	217.5	257.9	298.2	338.4	378.6	418.7
2.22	36	80						158.0	200.1	241.5	282.4	323.1
2.25	32	72					142.8	184.9	226.2	267.1	307.8	348.3
2.25	40	90								210.3	251.9	293.1
2.25	64	144										
2.29	28	64				127.7	169.8	211.0	251.8	292.4	332.8	373.2
2.33	24	56		112.5	133.7	154.6	195.7	236.5	277.0	317.4	357.7	397.9
2.33	48	112										225.1
2.35	34	80						161.2	203.5	245.0	286.0	326.7
2.37	38	90							171.0	213.7	255.4	296.6
2.40	30	72					146.1	188.4	229.7	270.7	311.4	351.9
2.46	26	64				131.0	173.2	214.5	255.4	296.0	336.5	376.9
2.50	32	80						164.5	206.9	248.4	289.5	330.3
2.50	36	90							174.3	217.0	258.8	300.1
2.55	22	56		115.8	137.1	158.0	199.3	240.1	280.7	321.1	361.4	401.7
2.55	44	112										231.6
2.57	28	72					149.4	191.8	233.2	274.3	315.0	355.6
2.65	34	90							177.5	220.4	262.2	303.6
2.67	24	64				134.2	176.6	218.0	259.0	299.7	340.2	380.6
2.67	30	80						167.8	210.3	251.9	293.1	333.9
2.77	26	72					152.6	195.1	236.7	277.8	318.6	359.2
2.80	40	112										238.1
2.81	32	90							180.7	223.7	265.7	307.1
2.86	28	80						171.0	213.7	255.4	296.6	337.5
2.91	22	64				137.5	180.0	221.5	262.5	303.3	343.8	384.3
2.95	38	112										241.4
3.00	24	72					155.9	198.5	240.2	281.3	322.2	362.8
3.00	30	90							183.9	227.0	269.1	310.6
3.00	48	144										
3.00	64	192										
3.08	26	80						174.3	217.0	258.8	300.1	341.0
3.11	36	112										244.6
3.21	28	90							187.1	230.3	272.5	314.0
3.27	44	144										
3.27	22	72					159.1	201.9	243.6	284.9	325.8	366.5
3.29	34	112									203.2	247.8
3.33	24	80						177.5	220.4	262.2	303.6	344.6
3.43	56	192										
3.46	26	90							190.3	233.6	275.9	317.5
3.50	32	112									206.3	251.0

CENTRE DISTANCE TABLES 8M & 8MGT

Theoretical centre distance in mm									Number of grooves		Speed ratio
Belt length code designation in mm											
1200	1280	1440	1600	1760	1800	2000	2400	2800	DriveN	DriveR	
430.5	470.6	550.8	631.0	711.1	731.1	831.2	1031.4	1231.5	56	28	2.00
406.0	446.1	526.4	606.6	686.8	706.8	807.0	1007.2	1207.3	64	32	2.00
381.2	421.5	501.9	582.2	662.4	682.5	782.7	982.9	1183.1	72	36	2.00
356.4	396.7	477.3	557.7	638.0	658.0	758.3	958.6	1158.9	80	40	2.00
253.9	295.4	377.2	458.4	539.3	559.5	660.1	861.0	1061.6	112	56	2.00
326.7	367.3	448.2	528.8	609.2	629.3	729.6	930.2	1130.5	90	44	2.05
360.0	400.4	481.0	561.5	641.8	661.8	762.1	962.5	1162.8	80	38	2.11
385.0	425.2	505.7	586.0	666.2	686.3	786.5	986.8	1187.0	72	34	2.12
409.7	449.9	530.2	610.5	690.6	710.7	810.8	1011.1	1211.2	64	30	2.13
434.3	474.5	554.7	634.9	715.0	735.0	835.1	1035.3	1235.4	56	26	2.15
458.8	498.9	579.1	659.2	739.3	759.3	859.4	1059.5	1259.6	48	22	2.18
363.7	404.1	484.8	565.2	645.6	665.6	766.0	966.4	1166.7	80	36	2.22
388.7	429.0	509.5	589.8	670.1	690.1	790.4	990.7	1190.9	72	32	2.25
333.9	374.6	455.5	536.2	616.7	636.8	737.2	937.8	1138.2	90	40	2.25
		285.6	369.9	452.5	473.0	575.0	777.3	978.7	144	64	2.25
413.5	453.7	534.0	614.3	694.5	714.5	814.7	1015.0	1215.1	64	28	2.29
438.1	478.3	558.5	638.7	718.8	738.9	839.0	1039.2	1239.3	56	24	2.33
267.5	309.2	391.5	473.0	554.0	574.2	675.1	876.2	1076.9	112	48	2.33
367.3	407.8	488.5	569.0	649.4	669.4	769.8	970.2	1170.5	80	34	2.35
337.5	378.2	459.2	539.9	620.5	640.6	741.0	941.7	1142.1	90	38	2.37
392.4	432.7	513.2	593.6	673.9	693.9	794.2	994.6	1194.8	72	30	2.40
417.2	457.4	537.8	618.1	698.3	718.4	818.6	1018.9	1219.0	64	26	2.46
371.0	411.5	492.2	572.7	653.1	673.2	773.6	974.1	1174.4	80	32	2.50
341.0	381.8	462.9	543.6	624.2	644.3	744.8	945.5	1145.9	90	36	2.50
441.9	482.1	562.3	642.5	722.7	742.7	842.9	1043.1	1243.2	56	22	2.55
274.2	316.1	398.6	480.2	561.3	581.5	682.5	883.8	1084.5	112	44	2.55
396.0	436.4	517.0	597.4	677.7	697.7	798.0	998.4	1198.7	72	28	2.57
344.6	385.4	466.5	547.3	627.9	648.1	748.6	949.3	1149.8	90	34	2.65
420.9	461.2	541.6	621.9	702.2	722.2	822.4	1022.7	1222.9	64	24	2.67
374.6	415.1	495.9	576.5	656.9	677.0	777.4	977.9	1178.3	80	30	2.67
399.7	440.1	520.7	601.1	681.5	701.6	801.9	1002.3	1202.6	72	26	2.77
280.9	322.9	405.6	487.4	568.6	588.8	689.9	891.3	1092.2	112	40	2.80
348.1	389.0	470.2	551.0	631.7	651.8	752.4	953.1	1153.6	90	32	2.81
378.2	418.8	499.6	580.2	660.7	680.8	781.2	981.8	1182.1	80	28	2.86
424.6	464.9	545.4	625.7	706.0	726.0	826.3	1026.6	1226.8	64	22	2.91
284.2	326.3	409.1	490.9	572.2	592.5	693.6	895.0	1095.9	112	38	2.95
403.4	443.8	524.4	604.9	685.3	705.4	805.7	1006.1	1206.5	72	24	3.00
351.7	392.5	473.8	554.7	635.4	655.5	756.1	956.9	1157.5	90	30	3.00
		311.7	397.0	480.4	501.0	603.6	806.7	1008.6	144	48	3.00
					349.2	458.7	668.0	872.7	192	64	3.00
381.8	422.4	503.3	583.9	664.4	684.5	785.0	985.6	1186.0	80	26	3.08
287.6	329.7	412.6	494.5	575.9	596.1	697.3	898.8	1099.7	112	36	3.11
355.2	396.1	477.5	558.4	639.1	659.3	759.9	960.8	1161.3	90	28	3.21
		318.2	403.8	487.3	508.0	610.7	814.0	1016.0	144	44	3.27
407.0	447.5	528.2	608.7	689.1	709.1	809.5	1010.0	1210.3	72	22	3.27
290.9	333.1	416.1	498.1	579.5	599.8	701.0	902.5	1103.5	112	34	3.29
385.4	426.0	507.0	587.7	668.2	688.3	788.8	989.4	1189.9	80	24	3.33
				338.7	361.7	471.9	681.9	887.0	192	56	3.43
358.7	399.7	481.1	562.1	642.8	663.0	763.6	964.6	1165.1	90	26	3.46
294.2	336.5	419.6	501.6	583.1	603.4	704.6	906.3	1107.3	112	32	3.50

CENTRE DISTANCE TABLES 8M & 8MGT

3

8 mm

Speed ratio	Number of grooves		Theoretical centre distance in mm									
			Belt length code designation in mm									
	DriveR	DriveN	480	560	600	640	720	800	880	960	1040	1120
3.60	40	144										
3.64	22	80						180.7	223.7	265.7	307.1	348.1
3.73	30	112									209.4	254.2
3.75	24	90							193.4	236.9	279.3	320.9
3.79	38	144										
4.00	28	112									212.5	257.4
4.00	36	144										
4.00	48	192										
4.09	22	90						150.3	196.6	240.2	282.6	324.4
4.24	34	144										
4.31	26	112									215.5	260.6
4.36	44	192										
4.50	32	144										
4.67	24	112									218.6	263.8
4.80	30	144										
4.80	40	192										
5.05	38	192										
5.09	22	112									221.7	267.0
5.14	28	144										
5.33	36	192										
5.54	26	144										
5.65	34	192										
6.00	24	144										
6.00	32	192										
6.40	30	192										
6.55	22	144										
6.86	28	192										
7.38	26	192										
8.00	24	192										
8.73	22	192										

Theoretical centre distance in mm									Number of grooves		Speed ratio
Belt length code designation in mm											
1200	1280	1440	1600	1760	1800	2000	2400	2800	DriveN	DriveR	
		324.6	410.4	494.1	514.9	617.8	821.3	1023.4	144	40	3.60
389.0	429.6	510.7	591.4	671.9	692.1	792.6	993.3	1193.7	80	22	3.64
297.5	339.8	423.1	505.2	586.7	607.0	708.3	910.0	1111.1	112	30	3.73
362.2	403.2	484.7	565.7	646.5	666.7	767.4	968.4	1169.0	90	24	3.75
		327.8	413.8	497.6	518.3	621.3	824.9	1027.1	144	38	3.79
300.8	343.2	426.5	508.7	590.3	610.6	712.0	913.7	1114.9	112	28	4.00
	239.3	331.0	417.1	501.0	521.8	624.8	828.6	1030.8	144	36	4.00
				350.9	374.1	484.9	695.7	901.3	192	48	4.00
365.7	406.8	488.3	569.4	650.2	670.4	771.1	972.1	1172.8	90	22	4.09
	242.3	334.2	420.4	504.4	525.2	628.3	832.2	1034.5	144	34	4.24
304.1	346.6	430.0	512.3	593.9	614.2	715.6	917.5	1118.6	112	26	4.31
				357.0	380.3	491.4	702.6	908.4	192	44	4.36
	245.2	337.4	423.8	507.8	528.6	631.8	835.8	1038.2	144	32	4.50
307.3	349.9	433.4	515.8	597.5	617.8	719.3	921.2	1122.4	112	24	4.67
	248.2	340.6	427.1	511.3	532.1	635.3	839.4	1041.9	144	30	4.80
				363.1	386.4	497.9	709.4	915.5	192	40	4.80
				366.1	389.5	501.1	712.9	919.0	192	38	5.05
310.6	353.2	436.9	519.3	601.0	621.4	722.9	924.9	1126.2	112	22	5.09
	251.2	343.8	430.4	514.7	535.5	638.8	843.0	1045.6	144	28	5.14
				369.1	392.6	504.4	716.3	922.5	192	36	5.33
	254.1	346.9	433.7	518.1	538.9	642.3	846.6	1049.2	144	26	5.54
				372.2	395.7	507.6	719.7	926.1	192	34	5.65
	257.1	350.1	437.0	521.5	542.3	645.8	850.2	1052.9	144	24	6.00
				375.2	398.7	510.8	723.1	929.6	192	32	6.00
				378.2	401.8	514.0	726.5	933.1	192	30	6.40
	260.1	353.3	440.3	524.8	545.7	649.3	853.8	1056.6	144	22	6.55
				381.2	404.8	517.2	729.9	936.6	192	28	6.86
				384.3	407.9	520.5	733.3	940.1	192	26	7.38
			282.2	387.3	411.0	523.7	736.7	943.7	192	24	8.00
			285.0	390.3	414.0	526.9	740.1	947.2	192	22	8.73

CENTRE DISTANCE TABLES 14M & 14MGT

3

14 mm

Speed ratio	Number of grooves		Theoretical centre distance in mm								
			Belt length code designation in mm								
	DriveR	DriveN	966	1190	1400	1610	1750	1778	1890	2100	2310
1.00	28	28	287.0	399.0	504.0	609.0	679.0	693.0	749.0	854.0	959.0
1.00	29	29	280.0	392.0	497.0	602.0	672.0	686.0	742.0	847.0	952.0
1.00	30	30	273.0	385.0	490.0	595.0	665.0	679.0	735.0	840.0	945.0
1.00	32	32	259.0	371.0	476.0	581.0	651.0	665.0	721.0	826.0	931.0
1.00	34	34	245.0	357.0	462.0	567.0	637.0	651.0	707.0	812.0	917.0
1.00	36	36	231.0	343.0	448.0	553.0	623.0	637.0	693.0	798.0	903.0
1.00	38	38	217.0	329.0	434.0	539.0	609.0	623.0	679.0	784.0	889.0
1.00	40	40	203.0	315.0	420.0	525.0	595.0	609.0	665.0	770.0	875.0
1.00	44	44		287.0	392.0	497.0	567.0	581.0	637.0	742.0	847.0
1.00	48	48		259.0	364.0	469.0	539.0	553.0	609.0	714.0	819.0
1.00	56	56			308.0	413.0	483.0	497.0	553.0	658.0	763.0
1.00	64	64				357.0	427.0	441.0	497.0	602.0	707.0
1.03	29	30	276.5	388.5	493.5	598.5	668.5	682.5	738.5	843.5	948.5
1.04	28	29	283.5	395.5	500.5	605.5	675.5	689.5	745.5	850.5	955.5
1.05	38	40	210.0	322.0	427.0	532.0	602.0	616.0	672.0	777.0	882.0
1.06	32	34	252.0	364.0	469.0	574.0	644.0	658.0	714.0	819.0	924.0
1.06	34	36	238.0	350.0	455.0	560.0	630.0	644.0	700.0	805.0	910.0
1.06	36	38	224.0	336.0	441.0	546.0	616.0	630.0	686.0	791.0	896.0
1.07	28	30	280.0	392.0	497.0	602.0	672.0	686.0	742.0	847.0	952.0
1.07	30	32	266.0	378.0	483.0	588.0	658.0	672.0	728.0	833.0	938.0
1.09	44	48		272.9	377.9	482.9	552.9	566.9	622.9	727.9	833.0
1.10	29	32	269.4	381.4	486.5	591.5	661.5	675.5	731.5	836.5	941.5
1.10	40	44		300.9	405.9	510.9	580.9	594.9	650.9	755.9	861.0
1.11	36	40	216.8	328.9	433.9	538.9	608.9	622.9	678.9	783.9	889.0
1.12	34	38	230.8	342.9	447.9	552.9	622.9	636.9	692.9	798.0	903.0
1.13	30	34	258.8	370.9	475.9	580.9	650.9	664.9	720.9	826.0	931.0
1.13	32	36	244.8	356.9	461.9	566.9	636.9	650.9	706.9	812.0	917.0
1.13	64	72				328.5	398.6	412.6	468.7	573.7	678.8
1.14	28	32	272.9	384.9	489.9	594.9	664.9	678.9	734.9	840.0	945.0
1.14	56	64				384.6	454.7	468.7	524.7	629.7	734.8
1.16	38	44		307.7	412.8	517.8	587.8	601.9	657.9	762.9	867.9
1.17	29	34	262.3	374.3	479.4	584.4	654.4	668.4	724.4	829.4	934.4
1.17	48	56			335.5	440.6	510.7	524.7	580.7	685.8	790.8
1.18	34	40	223.6	335.7	440.8	545.8	615.9	629.9	685.9	790.9	895.9
1.19	32	38	237.6	349.7	454.8	559.8	629.9	643.9	699.9	804.9	909.9
1.20	30	36	251.6	363.8	468.8	573.8	643.9	657.9	713.9	818.9	923.9
1.20	40	48		286.4	391.6	496.7	566.7	580.7	636.8	741.8	846.8
1.21	28	34	265.7	377.8	482.8	587.8	657.9	671.9	727.9	832.9	937.9
1.22	36	44	202.2	314.5	419.6	524.7	594.7	608.7	664.8	769.8	874.8
1.24	29	36	255.0	367.2	472.2	577.3	647.3	661.3	717.3	822.4	927.4
1.25	32	40	230.3	342.5	447.6	552.7	622.7	636.8	692.8	797.8	902.8
1.25	64	80					369.3	383.3	439.6	544.8	650.0
1.26	38	48		293.2	398.4	503.5	573.6	587.6	643.6	748.7	853.7
1.27	30	38	244.3	356.6	461.7	566.7	636.8	650.8	706.8	811.8	916.8
1.27	44	56		243.5	349.0	454.2	524.3	538.3	594.4	699.5	804.6
1.29	28	36	258.4	370.6	475.7	580.7	650.8	664.8	720.8	825.8	930.8
1.29	34	44	208.8	321.2	426.4	531.5	601.6	615.6	671.6	776.7	881.7
1.29	56	72				355.2	425.5	439.6	495.7	600.9	706.1
1.31	29	38	247.7	359.9	465.1	570.1	640.2	654.2	710.2	815.3	920.3
1.33	30	40	237.0	349.3	454.5	559.6	629.6	643.6	699.6	804.7	909.7
1.33	36	48		299.8	405.1	510.3	580.4	594.4	650.5	755.5	860.6

Theoretical centre distance in mm								Number of grooves		Speed ratio
Belt length code designation in mm										
2450	2590	2800	3150	3500	3850	4326	4578	DriveN	DriveR	
1029.0	1099.0	1204.0	1379.0	1554.0	1729.0	1967.0	2093.0	28	28	1.00
1022.0	1092.0	1197.0	1372.0	1547.0	1722.0	1960.0	2086.0	29	29	1.00
1015.0	1085.0	1190.0	1365.0	1540.0	1715.0	1953.0	2079.0	30	30	1.00
1001.0	1071.0	1176.0	1351.0	1526.0	1701.0	1939.0	2065.0	32	32	1.00
987.0	1057.0	1162.0	1337.0	1512.0	1687.0	1925.0	2051.0	34	34	1.00
973.0	1043.0	1148.0	1323.0	1498.0	1673.0	1911.0	2037.0	36	36	1.00
959.0	1029.0	1134.0	1309.0	1484.0	1659.0	1897.0	2023.0	38	38	1.00
945.0	1015.0	1120.0	1295.0	1470.0	1645.0	1883.0	2009.0	40	40	1.00
917.0	987.0	1092.0	1267.0	1442.0	1617.0	1855.0	1981.0	44	44	1.00
889.0	959.0	1064.0	1239.0	1414.0	1589.0	1827.0	1953.0	48	48	1.00
833.0	903.0	1008.0	1183.0	1358.0	1533.0	1771.0	1897.0	56	56	1.00
777.0	847.0	952.0	1127.0	1302.0	1477.0	1715.0	1841.0	64	64	1.00
1018.5	1088.5	1193.5	1368.5	1543.5	1718.5	1956.5	2082.5	30	29	1.03
1025.5	1095.5	1200.5	1375.5	1550.5	1725.5	1963.5	2089.5	29	28	1.04
952.0	1022.0	1127.0	1302.0	1477.0	1652.0	1890.0	2016.0	40	38	1.05
994.0	1064.0	1169.0	1344.0	1519.0	1694.0	1932.0	2058.0	34	32	1.06
980.0	1050.0	1155.0	1330.0	1505.0	1680.0	1918.0	2044.0	36	34	1.06
966.0	1036.0	1141.0	1316.0	1491.0	1666.0	1904.0	2030.0	38	36	1.06
1022.0	1092.0	1197.0	1372.0	1547.0	1722.0	1960.0	2086.0	30	28	1.07
1008.0	1078.0	1183.0	1358.0	1533.0	1708.0	1946.0	2072.0	32	30	1.07
903.0	973.0	1078.0	1253.0	1428.0	1603.0	1841.0	1967.0	48	44	1.09
1011.5	1081.5	1186.5	1361.5	1536.5	1711.5	1949.5	2075.5	32	29	1.10
931.0	1001.0	1106.0	1281.0	1456.0	1631.0	1869.0	1995.0	44	40	1.10
959.0	1029.0	1134.0	1309.0	1484.0	1659.0	1897.0	2023.0	40	36	1.11
973.0	1043.0	1148.0	1323.0	1498.0	1673.0	1911.0	2037.0	38	34	1.13
1001.0	1071.0	1176.0	1351.0	1526.0	1701.0	1939.0	2065.0	34	30	1.13
987.0	1057.0	1162.0	1337.0	1512.0	1687.0	1925.0	2051.0	36	32	1.13
748.8	818.8	923.8	1098.9	1273.9	1448.9	1686.9	1812.9	72	64	1.13
1015.0	1085.0	1190.0	1365.0	1540.0	1715.0	1953.0	2079.0	32	28	1.14
804.8	874.8	979.8	1154.9	1329.9	1504.9	1742.9	1868.9	64	56	1.14
937.9	1007.9	1112.9	1287.9	1462.9	1637.9	1876.0	2002.0	44	38	1.16
1004.4	1074.4	1179.4	1354.5	1529.5	1704.5	1942.5	2068.5	34	29	1.17
860.8	930.8	1035.8	1210.9	1385.9	1560.9	1798.9	1924.9	56	48	1.17
965.9	1035.9	1140.9	1315.9	1490.9	1665.9	1904.0	2030.0	40	34	1.18
979.9	1049.9	1154.9	1329.9	1504.9	1679.9	1918.0	2044.0	38	32	1.19
993.9	1063.9	1168.9	1343.9	1518.9	1693.9	1932.0	2058.0	36	30	1.20
916.8	986.8	1091.9	1266.9	1441.9	1616.9	1854.9	1980.9	48	40	1.20
1007.9	1077.9	1182.9	1357.9	1532.9	1707.9	1946.0	2072.0	34	28	1.21
944.8	1014.8	1119.9	1294.9	1469.9	1644.9	1882.9	2008.9	44	36	1.22
997.4	1067.4	1172.4	1347.4	1522.4	1697.4	1935.4	2061.4	36	29	1.24
972.8	1042.8	1147.9	1322.9	1497.9	1672.9	1910.9	2036.9	40	32	1.25
720.1	790.2	895.3	1070.4	1245.5	1420.6	1658.6	1784.6	80	64	1.25
923.7	993.8	1098.8	1273.8	1448.8	1623.8	1861.9	1987.9	48	38	1.26
986.8	1056.8	1161.9	1336.9	1511.9	1686.9	1924.9	2050.9	38	30	1.27
874.6	944.6	1049.7	1224.7	1399.7	1574.8	1812.8	1938.8	56	44	1.27
1000.8	1070.9	1175.9	1350.9	1525.9	1700.9	1938.9	2064.9	36	28	1.29
951.7	1021.8	1126.8	1301.8	1476.8	1651.8	1889.9	2015.9	44	34	1.29
776.2	846.2	951.3	1126.4	1301.5	1476.6	1714.6	1840.7	72	56	1.29
990.3	1060.3	1165.3	1340.3	1515.4	1690.4	1928.4	2054.4	38	29	1.31
979.7	1049.8	1154.8	1329.8	1504.8	1679.9	1917.9	2043.9	40	30	1.33
930.6	1000.6	1105.7	1280.7	1455.8	1630.8	1868.8	1994.8	48	36	1.33

CENTRE DISTANCE TABLES 14M & 14MGT

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14 mm

Speed ratio	Number of grooves		Theoretical centre distance in mm								
			Belt length code designation in mm								
	DriveR	DriveN	966	1190	1400	1610	1750	1778	1890	2100	2310
1.33	48	64			305.9	411.5	481.7	495.7	551.8	657.0	762.2
1.36	28	38	251.0	363.3	468.5	573.6	643.6	657.6	713.7	818.7	923.7
1.38	29	40	240.2	352.6	457.8	563.0	633.0	647.0	703.1	808.1	913.2
1.38	32	44	215.3	327.9	433.2	538.3	608.4	622.4	678.5	783.5	888.6
1.40	40	56		256.5	362.2	467.6	537.8	551.8	608.0	713.1	818.2
1.41	34	48		306.4	411.8	517.1	587.2	601.2	657.3	762.4	867.4
1.41	64	90							401.8	507.7	613.3
1.43	28	40	243.5	356.0	461.2	566.4	636.4	650.5	706.5	811.6	916.6
1.43	56	80				324.6	395.4	409.5	465.9	571.5	676.9
1.45	44	64			318.9	424.7	495.0	509.0	565.2	670.5	775.7
1.47	30	44	221.8	334.5	439.9	545.1	629.6	629.2	685.3	790.4	895.5
1.47	38	56		262.9	368.8	474.3	544.5	558.6	614.7	719.9	825.0
1.50	32	48	199.8	313.0	418.5	523.8	593.9	608.0	664.0	769.2	874.3
1.50	48	72				381.2	451.8	465.9	522.3	627.7	733.0
1.52	29	44	225.0	337.8	443.2	548.5	618.6	632.6	688.7	793.8	898.9
1.56	36	56		269.3	375.4	480.9	551.2	565.2	621.4	726.6	831.8
1.57	28	44	228.2	341.1	446.6	551.8	622.0	636.0	692.1	797.2	902.3
1.60	30	48	206.1	319.5	425.1	530.5	600.7	614.7	670.8	776.0	881.1
1.60	40	64			331.7	437.7	508.2	522.3	578.5	683.9	789.2
1.61	56	90						370.2	427.3	533.6	639.5
1.64	44	72			287.2	394.1	464.8	478.9	535.4	641.0	746.4
1.65	34	56		275.6	381.8	487.5	557.8	571.9	628.1	733.4	838.6
1.66	29	48	209.2	322.7	428.4	533.8	604.0	618.0	674.2	779.3	884.5
1.67	48	80				349.7	420.9	435.5	491.2	597.7	703.4
1.68	38	64		230.7	338.0	444.2	514.7	528.8	585.1	690.6	795.9
1.71	28	48	212.3	325.9	431.7	537.2	607.4	621.4	677.5	782.7	887.9
1.75	32	56		281.9	388.3	494.1	564.5	578.5	634.7	740.1	845.3
1.75	64	112								420.3	528.1
1.78	36	64			344.3	450.7	521.3	535.4	591.7	697.2	802.6
1.80	40	72			299.5	406.7	477.7	491.8	548.4	654.1	759.7
1.82	44	80				362.1	433.6	447.8	504.6	610.7	716.5
1.87	30	56		288.2	394.7	500.6	571.1	585.1	641.4	746.8	852.0
1.88	34	64		242.7	350.6	457.1	527.8	541.9	598.3	703.8	809.2
1.88	48	90					380.4	394.9	452.3	559.2	665.4
1.89	38	72			305.6	413.0	484.1	498.2	554.8	660.7	766.3
1.93	29	56		291.3	397.9	503.9	574.3	588.4	644.7	750.1	855.4
2.00	28	56		294.4	401.1	507.2	577.6	591.7	648.0	753.4	858.7
2.00	32	64		248.7	356.9	463.5	534.2	548.4	604.8	710.4	815.9
2.00	36	72			311.6	419.3	490.4	504.6	561.3	667.2	772.8
2.00	40	80				374.3	446.1	460.3	517.3	623.6	729.5
2.00	56	112								444.4	552.9
2.05	44	90				319.4	392.5	407.0	464.6	571.8	678.2
2.11	38	80			270.7	380.4	452.3	466.6	523.6	630.0	736.0
2.12	34	72			317.6	425.5	496.8	511.0	567.7	673.7	779.4
2.13	30	64		254.6	363.1	469.9	540.7	554.8	611.3	717.0	822.5
2.21	29	64		257.6	366.2	473.1	543.9	558.0	614.5	720.3	825.8
2.22	36	80			276.4	386.5	458.5	472.8	529.9	636.4	742.5
2.25	32	72			323.6	431.8	503.1	517.3	574.1	680.2	785.9
2.25	40	90				331.1	404.6	419.1	476.9	584.3	691.0
2.25	64	144									
2.29	28	64		260.6	369.3	476.2	547.1	561.3	617.8	723.5	829.1

Theoretical centre distance in mm								Number of grooves		Speed ratio
Belt length code designation in mm										
2450	2590	2800	3150	3500	3850	4326	4578	DriveN	DriveR	
832.2	902.3	1007.4	1182.5	1357.5	1532.6	1770.6	1896.7	64	48	1.33
993.8	1063.8	1168.8	1343.8	1518.8	1693.9	1931.9	2057.9	38	28	1.36
983.2	1053.2	1158.2	1333.3	1508.3	1683.3	1921.3	2047.4	40	29	1.38
958.6	1028.7	1133.7	1308.7	1483.8	1658.8	1896.8	2022.8	44	32	1.38
888.3	958.3	1063.4	1238.5	1413.6	1588.6	1826.7	1952.7	56	40	1.40
937.5	1007.5	1112.6	1287.6	1462.7	1637.7	1875.7	2001.8	48	34	1.41
683.5	753.8	859.0	1034.4	1209.6	1384.8	1623.0	1749.0	90	64	1.41
986.6	1056.7	1161.7	1336.7	1511.8	1686.8	1924.8	2050.8	40	28	1.43
747.1	817.2	922.4	1097.7	1272.9	1448.0	1686.2	1812.2	80	56	1.43
845.8	915.9	1021.0	1196.2	1371.3	1546.4	1784.4	1910.5	64	44	1.45
965.5	1035.5	1140.6	1315.6	1490.7	1665.7	1903.7	2029.8	44	30	1.47
895.1	965.2	1070.2	1245.4	1420.4	1595.5	1833.6	1959.6	56	38	1.47
944.3	1014.4	1119.4	1294.5	1469.6	1644.6	1882.7	2008.7	48	32	1.50
803.2	873.4	978.5	1153.8	1328.9	1504.0	1742.2	1868.2	72	48	1.50
968.9	1039.0	1144.0	1319.1	1494.1	1669.2	1907.2	2033.2	44	29	1.52
901.9	972.0	1077.1	1252.2	1427.3	1602.4	1840.5	1966.5	56	36	1.56
972.3	1042.4	1147.4	1322.5	1497.6	1672.6	1910.7	2036.7	44	28	1.57
951.2	1021.2	1126.3	1301.4	1476.5	1651.5	1889.6	2015.6	48	30	1.60
859.3	929.5	1034.6	1209.8	1385.0	1560.1	1798.2	1924.3	64	40	1.60
710.0	780.3	885.8	1061.3	1236.7	1412.0	1650.3	1776.4	90	56	1.61
816.6	886.8	992.0	1167.3	1342.6	1517.7	1755.9	1882.0	72	44	1.64
908.7	978.8	1083.9	1259.0	1434.2	1609.3	1847.3	1973.4	56	34	1.65
954.6	1024.6	1129.7	1304.8	1479.9	1655.0	1893.0	2019.1	48	29	1.66
773.7	844.0	949.3	1124.7	1300.0	1475.3	1713.5	1839.6	80	48	1.67
866.1	936.2	1041.4	1216.6	1391.8	1566.9	1805.1	1931.1	64	38	1.68
958.0	1028.0	1133.1	1308.2	1483.3	1658.4	1896.5	2022.5	48	28	1.71
915.4	985.5	1090.7	1265.9	1441.0	1616.1	1854.2	1980.3	56	32	1.75
599.4	670.5	776.6	953.0	1128.9	1304.6	1543.3	1669.6	112	64	1.75
872.8	942.9	1048.1	1223.4	1398.6	1573.8	1811.9	1938.0	64	36	1.78
829.9	900.2	1005.5	1180.8	1356.1	1531.3	1769.6	1895.7	72	40	1.80
786.9	857.2	962.7	1138.2	1313.6	1488.8	1727.1	1853.3	80	44	1.82
922.2	992.3	1097.5	1272.7	1447.8	1623.0	1861.1	1987.2	56	30	1.87
879.5	949.6	1054.9	1230.2	1405.4	1580.6	1818.8	1944.9	64	34	1.88
736.0	806.6	912.2	1088.0	1263.5	1439.0	1677.4	1803.6	90	48	1.88
836.6	906.8	1012.2	1187.6	1362.9	1538.1	1776.4	1902.5	72	38	1.89
925.5	995.7	1100.9	1276.1	1451.3	1626.4	1864.5	1990.6	56	29	1.93
928.9	999.1	1104.2	1279.5	1454.7	1629.8	1868.0	1994.0	56	28	2.00
886.1	956.3	1061.6	1236.9	1412.2	1587.4	1825.6	1951.7	64	32	2.00
843.2	913.5	1018.8	1194.3	1369.7	1544.9	1783.2	1909.3	72	36	2.00
800.0	870.4	975.9	1151.5	1327.0	1502.4	1740.7	1866.9	80	40	2.00
624.5	695.8	802.3	979.0	1155.3	1331.1	1570.0	1696.4	112	56	2.00
749.0	819.6	925.3	1101.2	1276.9	1452.4	1690.9	1817.1	90	44	2.05
806.6	877.0	982.5	1158.2	1333.7	1509.1	1747.5	1873.7	80	38	2.11
849.8	920.1	1025.5	1201.0	1376.4	1551.7	1790.0	1916.1	72	34	2.12
892.8	963.0	1068.3	1243.7	1419.0	1594.2	1832.4	1958.5	64	30	2.13
896.1	966.4	1071.7	1247.1	1422.4	1597.6	1835.8	1961.9	64	29	2.21
813.1	883.6	989.1	1164.9	1340.4	1515.8	1754.3	1880.4	80	36	2.22
856.4	926.7	1032.1	1207.7	1383.1	1558.5	1796.8	1922.9	72	32	2.25
761.8	832.5	938.4	1114.4	1290.2	1465.8	1704.4	1830.6	90	40	2.25
	537.1	647.3	827.7	1006.2	1183.6	1423.8	1550.7	144	64	2.25
899.4	969.7	1075.0	1250.4	1425.7	1601.0	1839.3	1965.4	64	28	2.29

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14 mm

CENTRE DISTANCE TABLES 14M & 14MGT

Speed ratio	Number of grooves		Theoretical centre distance in mm								
			Belt length code designation in mm								
	DriveR	DriveN	966	1190	1400	1610	1750	1778	1890	2100	2310
2.33	48	112								468.1	577.3
2.35	34	80			282.2	392.5	464.6	479.0	536.2	642.8	749.0
2.37	38	90				336.9	410.5	425.1	483.0	590.6	697.4
2.40	30	72			329.6	438.0	509.4	523.6	580.4	686.6	792.5
2.48	29	72			332.6	441.1	512.5	526.8	583.6	689.8	795.7
2.50	32	80			287.9	398.6	470.8	485.2	542.4	649.2	755.4
2.50	36	90				342.6	416`.5	431.1	489.1	596.8	703.7
2.55	44	112							367.3	479.9	589.4
2.57	28	72			335.6	444.1	515.7	529.9	586.8	693.1	799.0
2.57	56	144									
2.65	34	90				348.4	422.4	437.1	495.2	603.0	710.0
2.67	30	80			293.6	404.6	476.9	491.3	548.6	655.5	761.8
2.76	29	80			296.4	407.6	480.0	494.4	551.8	658.7	765.0
2.80	40	112							378.5	491.6	601.5
2.81	32	90				354.1	428.4	443.0	501.2	609.2	716.3
2.86	28	80			299.3	410.5	483.0	497.4	554.9	661.8	768.2
2.95	38	112							384.0	497.4	607.5
3.00	30	90				359.9	434.3	448.9	507.3	615.4	722.6
3.00	48	144									
3.00	64	192									
3.10	29	90				362.7	437.2	451.9	510.3	618.5	725.7
3.11	36	112							389.6	503.2	613.5
3.21	28	90				365.6	440.1	454.9	513.3	621.6	728.9
3.27	44	144									439.2
3.29	34	112						331.3	395.1	509.0	619.5
3.43	56	192									
3.50	32	112						336.6	400.6	514.8	625.4
3.60	40	144									449.9
3.73	30	112						341.9	406.2	520.6	631.4
3.79	38	144									455.2
3.86	29	112						344.6	408.9	523.5	634.3
4.00	28	112						347.2	411.7	526.4	637.3
4.00	36	144									460.6
4.00	48	192									
4.24	34	144									465.9
4.36	44	192									
4.50	32	144									471.2
4.80	40	192									
4.80	30	144									476.5
4.97	29	144									479.2
5.05	38	192									
5.14	28	144									481.9
5.33	36	192									
5.65	34	192									
6.00	32	192									
6.40	30	192									
6.62	29	192									
6.86	28	192									

Theoretical centre distance in mm								Number of grooves		Speed ratio
Belt length code designation in mm										
2450	2590	2800	3150	3500	3850	4326	4578	DriveN	DriveR	
649.3	720.8	827.7	1004.9	1181.4	1357.5	1596.6	1723.1	112	48	2.12
819.6	890.1	995.7	1171.5	1347.1	1522.5	1761.0	1887.2	80	34	2.35
768.2	839.0	944.9	1121.0	1296.8	1472.4	1711.1	1837.3	90	38	2.37
862.9	933.3	1038.8	1214.4	1389.8	1565.2	1803.6	1929.7	72	30	2.40
866.2	936.6	1042.1	1217.7	1393.2	1568.6	1807.0	1933.1	72	29	2.48
826.1	896.6	1002.3	1178.1	1353.8	1529.3	1767.8	1894.0	80	32	2.50
774.6	845.4	951.4	1127.6	1303.4	1479.1	1717.8	1844.1	90	36	2.50
661.6	733.3	840.3	1017.7	1194.4	1370.6	1609.9	1736.4	112	44	2.55
869.5	939.9	1045.4	1221.1	1396.6	1571.9	1810.3	1936.5	72	28	2.57
484.8	560.3	671.1	852.3	1031.3	1209.1	1449.7	1576.8	144	56	2.57
781.0	851.8	957.9	1134.1	1310.1	1485.8	1724.5	1850.8	90	34	2.65
832.5	903.1	1008.8	1184.8	1360.4	1536.0	1774.5	1900.7	80	30	2.67
835.8	906.4	1012.1	1188.1	1363.8	1539.3	1777.9	1904.1	80	29	2.76
673.8	745.7	852.9	1030.5	1207.3	1383.7	1623.1	1749.6	112	40	2.80
787.4	858.3	964.3	1140.7	1316.7	1492.4	1731.2	1857.5	90	32	2.81
839.0	909.6	1015.4	1191.4	1367.1	1542.6	1781.2	1907.5	80	28	2.86
679.9	751.8	859.1	1036.9	1213.8	1390.2	1629.7	1756.3	112	38	2.95
793.7	864.6	970.8	1147.2	1323.2	1499.0	1737.9	1864.2	90	30	3.00
507.2	583.3	694.8	876.8	1056.3	1234.4	1475.5	1602.7	144	48	3.00
			611.2	802.8	987.5	1233.9	1363.1	192	64	3.00
796.9	867.8	974.0	1150.5	1326.5	1502.3	1741.2	1867.6	90	29	3.10
686.0	758.0	865.4	1043.2	1220.2	1396.7	1636.2	1762.9	112	36	3.11
800.0	871.0	977.2	1153.7	1329.8	1505.7	1744.5	1870.9	90	28	3.21
518.3	594.8	706.6	888.9	1068.7	1247.0	1488.3	1615.6	144	44	3.27
692.1	764.1	871.6	1049.6	1226.7	1403.2	1642.8	1769.5	112	34	3.29
			633.0	825.7	1011.2	1258.3	1387.8	192	56	3.43
698.1	770.3	877.8	1055.9	1233.1	1409.7	1649.4	1776.0	112	32	3.50
529.4	606.1	718.3	901.0	1081.1	1259.6	1501.1	1628.5	144	40	3.60
704.2	776.4	884.1	1062.2	1239.5	1416.2	1655.9	1782.6	112	30	3.73
535.0	611.8	724.1	907.1	1087.2	1265.9	1507.5	1634.9	144	38	3.79
707.2	779.5	887.2	1065.4	1242.7	1419.4	1659.2	1785.9	112	29	3.86
			1068.6	1245.9	1422.7	1662.5	1789.2	112	28	4.00
540.5	617.5	730.0	913.1	1093.4	1272.2	1513.8	1641.3	144	36	4.00
			654.7	848.6	1034.8	1282.7	1412.4	192	48	4.00
546.0	623.1	735.8	919.1	1099.6	1278.4	1520.2	1647.7	144	34	4.24
			665.5	860.0	1046.6	1294.8	1424.7	192	44	4.36
551.5	628.8	741.6	925.1	1105.7	1284.7	1526.6	1654.1	144	32	4.50
			676.3	871.3	1058.3	1306.9	1436.9	192	40	4.80
557.0	634.4	747.4	931.1	1111.9	1290.9	1532.9	1660.5	144	30	4.80
559.8	637.3	750.3	934.1	1114.9	1294.0	1536.1	1663.7	144	29	4.97
			681.7	877.0	1064.2	1312.9	1443.0	192	38	5.05
562.5	640.1	753.2	937.1	1118.0	1297.2	1539.2	1666.9	144	28	5.14
			687.0	882.6	1070.0	1318.9	1449.1	192	36	5.33
			692.4	888.3	1075.9	1324.9	1455.2	192	34	5.65
			697.8	893.9	1081.7	1331.0	1461.3	192	32	6.00
			703.1	899.6	1087.5	1337.0	1467.4	192	30	6.40
			705.8	902.4	1090.4	1340.0	1470.4	192	29	6.62
			708.5	905.2	1093.3	1343.0	1473.5	192	28	6.86

8M POWER RATINGS - KILOWATTS

rpm of faster shaft	Number of grooves in small pulley															
	22	24	26	28	30	32	34	36	38	40	44	48	56	64	72	80
	Pulley pitch diameter in mm															
	56.02	61.12	66.21	71.30	76.39	81.49	86.58	91.67	96.77	101.86	112.05	122.23	142.60	162.97	183.35	203.72
10	0.02	0.02	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.06	0.06	0.07	0.08	0.09	0.10	0.11
20	0.03	0.04	0.04	0.05	0.06	0.07	0.08	0.09	0.11	0.11	0.12	0.14	0.16	0.18	0.20	0.23
50	0.08	0.09	0.11	0.13	0.16	0.18	0.21	0.23	0.27	0.28	0.31	0.34	0.40	0.45	0.51	0.56
100	0.16	0.19	0.22	0.27	0.31	0.36	0.41	0.47	0.54	0.56	0.62	0.68	0.79	0.90	1.02	1.13
200	0.33	0.37	0.45	0.53	0.62	0.72	0.82	0.93	1.05	1.13	1.24	1.34	1.54	1.73	1.93	2.12
300	0.49	0.53	0.65	0.77	0.90	1.04	1.19	1.34	1.51	1.64	1.78	1.93	2.21	2.50	2.77	3.05
400	0.65	0.71	0.84	0.99	1.16	1.34	1.54	1.74	1.96	2.12	2.31	2.50	2.87	3.23	3.59	3.94
500	0.81	0.89	1.02	1.21	1.42	1.64	1.88	2.13	2.40	2.59	2.82	3.05	3.50	3.94	4.37	4.80
600	0.98	1.07	1.21	1.43	1.68	1.94	2.21	2.51	2.82	3.05	3.32	3.59	4.11	4.63	5.13	5.63
730	1.19	1.30	1.44	1.71	2.00	2.31	2.64	2.98	3.36	3.63	3.95	4.27	4.89	5.50	6.09	6.68
800	1.30	1.42	1.56	1.85	2.17	2.50	2.86	3.24	3.64	3.94	4.28	4.63	5.30	5.95	6.60	7.23
870	1.42	1.54	1.68	1.99	2.34	2.70	3.08	3.49	3.93	4.24	4.61	4.98	5.70	6.41	7.09	7.76
970	1.58	1.72	1.86	2.20	2.57	2.97	3.39	3.84	4.32	4.67	5.08	5.48	6.27	7.04	7.79	8.52
1000	1.63	1.77	1.92	2.26	2.64	3.05	3.49	3.95	4.44	4.80	5.22	5.63	6.44	7.23	7.99	8.74
1170	1.90	2.07	2.25	2.59	3.04	3.51	4.00	4.53	5.10	5.51	5.98	6.45	7.37	8.26	9.13	9.96
1200	1.95	2.13	2.30	2.65	3.11	3.59	4.09	4.63	5.21	5.63	6.12	6.60	7.53	8.44	9.32	10.17
1460	2.37	2.58	2.80	3.15	3.69	4.26	4.86	5.50	6.19	6.68	7.25	7.81	8.90	9.95	10.95	11.92
1600	2.60	2.83	3.06	3.41	4.00	4.61	5.26	5.95	6.70	7.23	7.84	8.44	9.61	10.72	11.79	12.80
1750	2.84	3.09	3.34	3.69	4.32	4.98	5.69	6.43	7.23	7.80	8.46	9.10	10.35	11.53	12.64	13.70
2000	3.24	3.52	3.81	4.18	4.85	5.59	6.37	7.21	8.11	8.74	9.47	10.17	11.53	12.80	13.99	15.08
2500	4.03	4.38	4.74	5.19	5.86	6.75	7.69	8.69	9.77	10.52	11.36	12.17	13.70	15.08	16.32	17.40
2920	4.68	5.09	5.50	6.02	6.66	7.66	8.73	9.86	11.08	11.92	12.84	13.71	15.31	16.71	17.89	18.83
3500					7.71	8.85	10.07	11.36	12.75	13.70	14.68	15.60	17.20	18.47		
4000						9.79	11.13	12.55	14.07	15.08	16.09	16.99	18.47			
4500							12.10	13.62	15.26	16.32	17.30	18.14				
5000								14.57	16.30	17.40	18.31	19.04				
5500									17.20	18.31	19.10					
6000									17.95	19.04	19.65					

Belt width correction factors				
Belt width (mm)	20	30	50	85
Width factors	1.00	1.58	2.74	4.76

Bold figures refer to standard widths.

Belt length correction factors					
Belt length (mm)	480-608	640-912	960-1216	1280-1760	1800-2800
Length factors	0.8	0.9	1.0	1.1	1.2

Power ratings are based on a minimum of six teeth in mesh.
If you have less than this, you have to make an adjustment - see page 26.

14M POWER RATINGS - KILOWATTS

rpm of faster shaft	Number of grooves in small pulley																
	28	29	30	32	34	36	38	40	44	48	52	56	60	64	68	72	80
	Pulley pitch diameter in mm																
	124.78	129.23	133.69	142.60	151.52	160.43	169.34	178.25	196.08	213.90	231.73	249.55	267.38	285.21	303.03	320.86	356.51
10	0.2	0.2	0.2	0.2	0.3	0.3	0.3	0.4	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.7	0.8
20	0.4	0.4	0.4	0.5	0.6	0.6	0.7	0.7	0.8	0.9	1.0	1.1	1.1	1.2	1.3	1.4	1.5
40	0.7	0.8	0.8	1.0	1.1	1.2	1.4	1.4	1.6	1.8	1.9	2.1	2.3	2.4	2.6	2.7	3.0
60	1.1	1.2	1.3	1.5	1.7	1.9	2.0	2.2	2.4	2.7	2.9	3.2	3.4	3.6	3.8	4.1	4.5
100	1.8	1.9	2.1	2.4	2.8	3.1	3.4	3.6	4.0	4.4	4.9	5.2	5.6	6.0	6.4	6.7	7.5
200	3.6	3.9	4.2	4.8	5.5	6.2	6.8	7.2	8.0	8.9	9.7	10.5	11.2	12.0	12.7	13.5	15.0
300	4.9	5.3	5.7	6.6	7.5	8.5	9.2	9.7	10.8	12.0	13.1	14.2	15.3	16.5	17.7	18.9	21.3
400	6.1	6.6	7.1	8.2	9.3	10.5	11.3	12.0	13.3	14.7	16.1	17.4	18.7	20.1	21.5	22.9	25.8
500	7.2	7.8	8.4	9.6	11.0	12.3	13.3	14.1	15.6	17.2	18.7	20.2	21.7	23.3	24.8	26.4	29.6
600	8.2	8.9	9.5	11.0	12.5	14.0	15.1	15.9	17.6	19.4	21.1	22.7	24.4	26.1	27.8	29.5	32.9
730	9.4	10.2	10.9	12.6	14.2	16.0	17.2	18.2	20.0	22.0	23.8	25.6	27.4	29.3	31.1	32.9	36.5
800	10.0	10.8	11.6	13.4	15.1	17.0	18.3	19.3	21.2	23.2	25.2	27.0	28.9	30.8	32.6	34.5	38.2
870	10.6	11.4	12.3	14.1	16.0	17.9	19.3	20.3	22.4	24.4	26.4	28.3	30.2	32.2	34.0	36.0	39.7
970	11.4	12.3	13.2	15.1	17.1	19.2	20.6	21.7	23.8	26.0	28.0	30.0	32.0	33.9	35.8	37.7	41.4
1000	11.6	12.5	13.5	15.4	17.5	19.6	21.0	22.1	24.3	26.5	28.5	30.5	32.5	34.4	36.3	38.2	41.9
1160	12.8	13.8	14.8	16.9	19.1	21.4	22.9	24.1	26.3	28.6	30.7	32.7	34.7	36.7	38.5	40.3	43.7
1200	13.1	14.1	15.1	17.3	19.5	21.8	23.4	24.5	26.8	29.1	31.2	33.2	35.2	37.1	38.9	40.7	44.1
1460	14.7	15.8	16.9	19.3	21.8	24.3	25.9	27.1	29.5	31.8	33.8	35.7	37.5	39.3	40.8	42.3	44.7
1600	15.4	16.6	17.8	20.3	22.8	25.4	27.1	28.3	30.6	32.9	34.8	36.6	38.3	39.8	41.1	42.3	44.0
1750	16.2	17.4	18.6	21.2	23.8	26.5	28.2	29.4	31.6	33.8	35.6	37.2	38.6	39.9	40.8	41.6	42.5
2000	17.3	18.5	19.8	22.5	25.2	28.0	29.6	30.8	32.8	34.7	36.2	37.3	38.2	38.9	39.1		
2500	20.8	21.4	22.0	24.2	26.9	29.7	31.2	32.0	33.4	34.4	34.7	34.4					
2920	23.6	24.2	24.8	26.0	27.4	30.0	31.1	31.6	31.9	31.7							
3500			28.1	29.1	30.0	30.7	31.2	31.6									
4000				30.9	31.4												

Belt width correction factors					
Belt width (mm)	**40**	**55**	**85**	**115**	**170**
Width factors	**1.00**	**1.50**	**2.50**	**3.48**	**5.29**

Bold figures refer to standard widths.

Belt length correction factors						
Belt length (mm)	**966-1190**	**1400-1610**	**1778-1904**	**2100-2450**	**2590-3150**	**3500-4578**
Length factors	0.8	0.9	0.95	1.0	1.05	1.1

Power ratings are based on a minimum of six teeth in mesh.
If you have less than this, you have to make an adjustment - see page 26.

PREFERRED PULLEY RANGES

POWERGRIP® HTD®

8M

Number of grooves	Outside diameter (mm)	Flange diameter (mm)	Maximum bore - (mm) Standard belt width (mm)				System widths - (mm) Standard belt width (mm)			
			20	30	50	85	20	30	50	85
22	54.65	60	25.0	25.0	28.0	28.0	38.0	48.0	70.0	105.0
24	59.75	66	28.0	28.0	28.0	30.0	38.0	48.0	70.0	105.0
26	64.84	71	30.0	30.0	30.0	32.0	38.0	48.0	70.0	105.0
28	70.08	75	30.0	30.0	30.0	32.0	38.0	48.0	70.0	105.0
30	75.13	83	32.0	32.0	32.0	32.0	38.0	48.0	70.0	105.0
32	80.16	87	35.0	35.0	35.0	35.0	38.0	48.0	70.0	105.0
34	85.22	91	42.0	42.0	42.0	42.0	38.0	48.0	70.0	105.0
36	90.30	98.5	42.0	42.0	42.0	42.0	38.0	48.0	70.0	105.0
38	95.39	103	45.0	45.0	45.0	45.0	38.0	48.0	70.0	105.0
40	100.49	106	45.0	45.0	45.0	45.0	38.0	48.0	70.0	105.0
44	110.67	119	45.0	45.0	45.0	45.0	38.0	48.0	70.0	105.0
48	120.86	127	45.0	45.0	45.0	45.0	38.0	48.0	70.0	105.0
56	141.23	148	45.0	50.0	50.0	45.0	38.0	48.0	60.0	105.0
64	161.60	168	45.0	50.0	60.0	60.0	38.0	48.0	60.0	95.0
72	181.97	192	45.0	55.0	60.0	65.0	38.0	48.0	60.0	95.0
80	202.35		50.0	60.0	65.0	65.0	38.0	48.0	60.0	95.0
90	227.81		50.0	60.0	65.0	65.0	38.0	48.0	60.0	95.0
112	283.83		50.0	60.0	65.0	65.0	38.0	48.0	60.0	95.0
144	365.32		50.0	60.0	65.0	70.0	38.0	48.0	60.0	95.0
168	426.44		60.0	60.0	70.0	70.0	38.0	48.0	60.0	95.0
192	487.55		60.0	60.0	70.0	70.0	38.0	48.0	60.0	95.0

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NOTE: PowerGrip® GT3 8MGT and 14MGT belts are designed to run in standard PowerGrip® HTD® pulleys.

PREFERRED PULLEY RANGES

POWERGRIP® HTD®

14M

Number of grooves	Outside diameter (mm)	Flange diameter (mm)	Maximum bore - (mm) Standard belt width (mm)					System widths - (mm) Standard belt width (mm)				
			40	55	85	115	170	40	55	85	115	170
28	122.12	127	60.0	60.0	60.0	60.0	60	69.0	85.0	117.0	148.0	202
29	126.57	138	60.0	60.0	60.0	60.0	60	69.0	85.0	117.0	148.0	202
30	130.99	138	60.0	60.0	60.0	60.0	60	69.0	85.0	117.0	148.0	202
32	139.88	154	60.0	60.0	60.0	60.0	60	69.0	85.0	117.0	148.0	202
34	148.79	160	60.0	60.0	60.0	60.0	60	69.0	85.0	117.0	148.0	202
36	157.68	168	60.0	60.0	60.0	70.0	70	69.0	85.0	117.0	148.0	202
38	166.60	183	70.0	70.0	70.0	70.0	75	69.0	85.0	117.0	148.0	202
40	175.49	188	70.0	70.0	75.0	75.0	80	69.0	85.0	117.0	148.0	202
44	193.28	211	70.0	70.0	75.0	80.0	90	69.0	85.0	117.0	148.0	202
48	211.11	226	75.0	75.0	85.0	80.0	90	69.0	70.0	117.0	148.0	202
56	246.76	256	75.0	75.0	85.0	85.0	90	69.0	70.0	102.0	148.0	202
64	282.41	296	75.0	75.0	85.0	85.0	100	69.0	70.0	102.0	133.0	202
72	318.06		75.0	75.0	85.0	85.0	100	69.0	70.0	102.0	133.0	187
80	353.71		75.0	75.0	85.0	85.0	100	69.0	70.0	102.0	133.0	187
90	398.28		75.0	75.0	85.0	85.0	100	69.0	70.0	102.0	133.0	187
112	496.32		75.0	75.0	85.0	85.0	110	69.0	70.0	102.0	133.0	187
144	638.92		75.0	75.0	85.0	95.0	120	69.0	70.0	102.0	133.0	187
168	745.87		75.0	75.0	85.0	95.0	120	69.0	70.0	102.0	133.0	187
192	852.82		75.0	75.0	95.0	95.0	120	69.0	70.0	102.0	133.0	187
216	959.76		85.0	85.0	95.0	95.0	120	69.0	70.0	102.0	133.0	187

20M

Number of grooves	Outside diameter (mm)
34	*212.13*
36	*224.87*
38	*237.60*
40	*250.33*
44	*275.79*
48	*301.26*
52	*326.72*
56	*352.19*
60	*377.65*
64	*403.12*
68	*428.58*
72	*454.05*
80	*504.98*
90	*568.64*
112	*708.70*
144	*912.41*
168	*1065.20*
192	*1217.99*
216	*1370.79*

Pulley references in italics are mainly "made-to-order" designs.

1. PULLEY DIAMETER - SPEED

Blanks in the lower right-hand portions of the power rating tables occur because pulley rim speed exceeds 40 m/s. Centrifugal forces developed beyond this speed may prohibit the use of stock grey cast iron pulleys. For rim speeds exceeding 40 m/s, contact your Gates sales representative for recommendations.

2. USE OF FLANGED PULLEYS

Flanges are needed in order to keep the belt on the pulley. Due to tracking characteristics, even on the best aligned drives, belts will ride off the edge of the pulleys. Flanges will prevent this belt ride-off.

On all drives using stock or made-to-order pulleys, the following conditions should be considered when selecting flanged pulleys:

1. On all two-pulley drives, the minimum flanging requirements are two flanges on one pulley or one flange on each pulley on opposite sides.
2. On drives where the centre distance is more than eight times the diameter of the small pulley, special care has to be taken when setting up the drive. Always make sure the belt runs correctly on both pulleys. In some cases it might be necessary that both pulleys are flanged on both sides. (See point 7 "Belt installation and drive alignment" on page 158).
3. On drives with more than two pulleys, the minimum flanging requirements are two flanges on every other pulley or one flange on every pulley — alternating sides around the system.

On made-to-order pulleys, flanges must be securely fastened, by using mechanical fasteners, welding, shrink-fit or other equivalent methods.

3. FIXED (NON-ADJUSTABLE) CENTRES

Consult Gates' application engineers.

4. IDLERS

Use of idlers should be restricted to those cases in which they are functionally necessary. Idlers usually are used to apply tension when centres are not adjustable.

Idlers should be located on the slack side of the belt drive. For inside idlers, grooved pulleys are recommended up to 40 grooves. On larger diameters, flat, uncrowned idlers may be used. Inside idler diameters should not be smaller than the smallest loaded pulley in the system.

Outside or backside idlers should be flat and uncrowned; flanges are not recommended. Diameters should generally not be smaller than the smallest loaded pulley in the system.

Slack side spring loaded idlers can be used, as long as care is taken to avoid resonant vibration conditions and load reversals.

5. OPERATING ENVIRONMENT

Temperature

Gates PowerGrip® GT3, PowerGrip® HTD® and PowerGrip® belt performance is generally unaffected in ambient temperature environments between -25°C and +100°C. In cases where belts are constantly running at or above these temperature extremes, contact Gates' application engineers.

Aircraft drives

Gates belts should not be used on aircraft or aircraft related applications.

6. INSTALLATION AND TENSIONING ALLOWANCES

The information on centre distance allows for the installation of the belt without damage and then to tension it correctly. The standard installation allowance is the minimum decrease in centre distance required to install a belt when flanged pulleys are removed from their shafts for belt installation. Standard installation allowances are shown in the table below. This table also lists the minimum increase in centre distance required to ensure that a belt can be properly tensioned.

If a belt is to be installed over flanged pulleys without removing them, the additional centre distance allowance for installation shown in the second table must be added to the first table data.

Table No. 1
Centre distance allowance for installation and tensioning (mm)

Belt	Standard installation allowance (flanged pulleys removed for installation)	Tensioning allowance (any drive)
1000 mm and under	1.8	0.8
over 1000 mm to 1780 mm	2.8	0.8
over 1780 mm to 2540 mm	3.3	1.0
over 2540 mm to 3300 mm	4.1	1.0
over 3300 mm to 4600 mm	5.3	1.3

Table No. 2
Additional centre distance allowance for installation over flanged pulleys

Belt type	One pulley flanged (mm)	Both pulleys flanged (mm)
3MGT, 3M, XL	8	14
5MGT, 5M, L	14	19
8MGT, 8M, H	22	33
14MGT, 14M, XH	36	58
20M, XXH	47	78

7. BELT INSTALLATION AND DRIVE ALIGNMENT

If you cannot adjust the centre distance to install the belt according to the two tables on page 157, you need to change the idler position so that the belt can be slipped easily onto the drive. When installing a belt, never force it over the flange. This will damage the belt tensile member.

Synchronous belt performance may be affected by misalignment, which can result in inconsistent belt wear and premature failure.

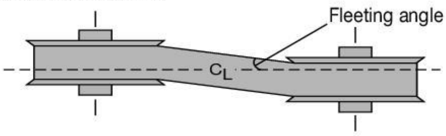
There are two types of misalignment: parallel and angular. Parallel misalignment is where the driveR and driveN shafts are parallel, but the two pulleys lie in different planes. When the two shafts are not parallel, the drive is angularly misaligned.

A fleeting angle is the angle at which the belt enters and exits the pulley, and equals the sum of the parallel and angular misalignments.

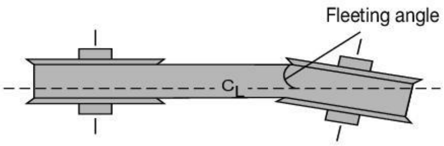
Misalignment of all positive belt drives should not exceed 1/4° or 5 mm per metre of centre distance. Misalignment should be checked with a good straight edge.
The straight edge should be applied from driveR to driveN and from driveN to driveR so that the effect of parallel and angular misalignment is taken into account.

Drive misalignment can also cause belt tracking problems. However, some degree of belt tracking is normal and won't affect performance.

Parallel misalignment



Angular misalignment



8. BELT STORAGE AND HANDLING

For storage, the belt should be protected from moisture, oil, temperature extremes, direct sunlight and high ozone environments. The belt should be stored in its original package, avoiding any sharp bends or crimping which will damage the belt.

9. EFFICIENCY

When properly designed and applied, Gates synchronous belts are up to 98% efficient and above, thanks to the positive, no-slip characteristics.

Synchronous belt drive efficiency can be calculated as follows:

$$\% \text{ efficiency} = \frac{\text{DN rpm x DN Torque}}{\text{DR rpm x DR Torque}} \times 100$$

When examining the loss of energy, it is necessary to consider belt losses in terms of shaft torque and shaft speed. Torque losses are created due to bending stress and friction.

Chain drives running unlubricated generate significant heat build-up due to increased friction in the roller joints. Even properly lubricated chains running at higher speeds tend to throw off the oil due to centrifugal forces, making it difficult to maintain proper lubrication at the load-bearing surfaces. Consequently, chain drives are typically only 92-98% efficient.

The belt drive is only part of the total system. Motors should be properly sized for the application. They must have sufficient capacity to meet the power needs, yet overdesigned motors will lead to electrical insufficiencies. DriveN machines also may have inherent inefficiencies which are not a factor in evaluating drive efficiency.

6

10. INSTALLATION TENSION

Gates' synchronous belts operate by positive meshing and do not require high installation tension.

However, if optimum belt performance is to be achieved, belts should be installed at an installation tension level suitable for the particular duty envisaged.

The required tension level will be between the maximum and minimum values (see formulae below). As a general guide, the lower level will be applicable to lightly loaded, smooth running drives, whereas drives subjected to high shock loads and/or frequent starts will be tensioned at the higher level.

A. Recommended maximum installation tension

$$T_{st} = 600\ \frac{P}{v}$$

Where: T_{st} = static tension (N)
P = power (kW)

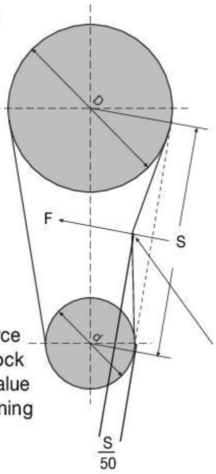
B. Recommended deflecting forces

1. For maximum installation tension

$$F = \frac{P \times 60}{v}\ (N)$$

2. For minimum installation tension

$$F = \frac{P \times 25}{v}\ (N)$$



The higher level of deflecting force should be applied where shock loads are expected. The lower value may be used for smooth running drives.

3. Belt deflection

$$\text{belt deflection} = \frac{S}{50}\ (mm)$$

Note:

$$v = \frac{\text{Pitch} \times N \times rpm}{60\,000}$$

Where
P = transmitted power (kW)
F = deflecting force
v = belt speed (m/s)
S = belt span length (mm)
N = number of grooves in driveR
rpm = rpm of driveR

11. BELT TOLERANCES

Synchronous belt width tolerances

Belt width mm	Belt width tolerances in mm		
	Belt lengths 0-838 mm	Belt lengths 839-1676 mm	Belt lengths 1677+ mm
3-11	+0.4	+0.4	
	-0.8	-0.8	
12-38	+0.8	+0.8	+0.8
	-0.8	-1.2	-1.2
39-51	+0.8	+1.2	+1.2
	-1.2	-1.2	-1.6
52-64	+1.2	+1.2	+1.6
	-1.2	-1.6	-1.6
65-76	+1.2	+1.6	+1.6
	-1.6	-1.6	-2.0
77-102	+1.6	+1.6	+2.0
	-1.6	-2.0	-2.0
102-178	+2.4	+2.4	+2.4
	-2.4	-2.8	-3.2
178+			+4.8
			-6.4

Synchronous belt centre distance tolerances

Belt length mm	Belt centre distance tolerances in mm	
	PowerGrip®/ PowerGrip® HTD®	PowerGrip® GT3
127-254	±0.20	±0.20
255-381	±0.23	±0.23
382-508	±0.25	±0.23
509-762	±0.30	±0.27
763-1016	±0.33	±0.30
1017-1270	±0.38	±0.32
1271-1524	±0.41	±0.36
1525-1778	±0.43	±0.39
1779	(±0.43)	±0.42
	(±0.025 mm per 254 mm)	(±0.025 mm per 250 mm)

Important
If belts need to be removed and replaced, the tension prior to removal has to be measured and applied for re-installation.

USEFUL INFORMATION

1. FORMULAE

PITCH DIAMETER

$$d = \frac{N \times p}{\pi}$$

SPEED RATIO

$$i = \frac{r}{R} = \frac{N}{n} = \frac{D}{d}$$

WRAP ANGLE

$$\beta = 2 \cos^{-1}\left[\frac{D - d}{2C}\right]$$

TEETH IN MESH

$$TIM = n \frac{\beta}{360}$$

$$\text{or } TIM = n \left[0.5 - \frac{(N - n)}{18.85 \times Nc}\right]$$

BELT LENGTH

$$L = 2C \sin \frac{\beta}{2} + \frac{\pi}{2}\left[(D + d) + \left(1 - \frac{\beta}{180}\right)(D - d)\right]$$

for $i = 1 \rightarrow \beta = 180°$, $\sin\left(\frac{\beta}{2}\right) = 1$, for $D = d \rightarrow L = 2C + \pi D$

CENTRE DISTANCE

$$C = \frac{1}{2 \sin \frac{\beta}{2}} \left\{ L - \frac{\pi}{2}\left[(D + d) + \left(1 - \frac{\beta}{180}\right) \times (D - d)\right]\right\}$$

for $i = 1 \rightarrow \beta = 180°$, $\sin\left(\frac{\beta}{2}\right) = 1$, for $D = d \rightarrow C = \frac{1}{2}(L - \pi D)$

APPROXIMATE BELT LENGTH

$$L = 2C + \frac{\pi}{2}(D + d) + \frac{(D - d)^2}{4C}$$

2. UNITS OF MEASUREMENT

kW	=	kilowatts
Nm	=	newton metre
N	=	newton
J	=	joule
s	=	second
mm	=	millimetre
m/s	=	metre/second
kg	=	kilogramme
g/m	=	gramme/metre

3. ABBREVIATION TABLE

ß	=	wrap angle
C	=	centre distance (mm)
D	=	pitch circle diameter of large pulley (mm)
d	=	pitch circle diameter of small pulley (mm)
DN	=	driven pulley
DR	=	driver pulley
F	=	force (N)
i	=	speed ratio
L	=	belt length (mm)
N	=	number of grooves of large pulley
n	=	number of grooves of small pulley
Nb	=	belt length in number of pitches
Nc	=	centre distance in number of pitches
p	=	pitch
P	=	transmitted power (kW)
R	=	speed of large pulley (rpm)
r	=	speed of small pulley (rpm)
S	=	belt span length (mm)
T	=	torque (Nm)
T.I.M.	=	teeth in mesh
T.I.R.	=	total indicator reading
U.R.D.	=	upper reference depth
v	=	belt speed (m/s)

4. CONVERSION TABLE

1 lbf	=	0.454 kgf
1 lbf	=	4.448 N
1 kgf	=	9.807 N
1 lbf in	=	0.113 Nm
1 ft	=	0.3048 m
1 in	=	25.4 mm
1 ft^2	=	0.093 m^2
1 in^2	=	645.16 mm^2
1 ft^3	=	0.028 m^3
1 in^3	=	16.387 mm^3
1 oz	=	28.35 g
1 lb	=	0.454 kg
1 Imp. ton	=	1.016 tonne
1 Imp. gal	=	4.546 litres
1 Imp. pint	=	0.568 litre
1 radian	=	57.296 degrees
1 degree	=	0.0175 radian
1 horsepower	=	0.746 kW