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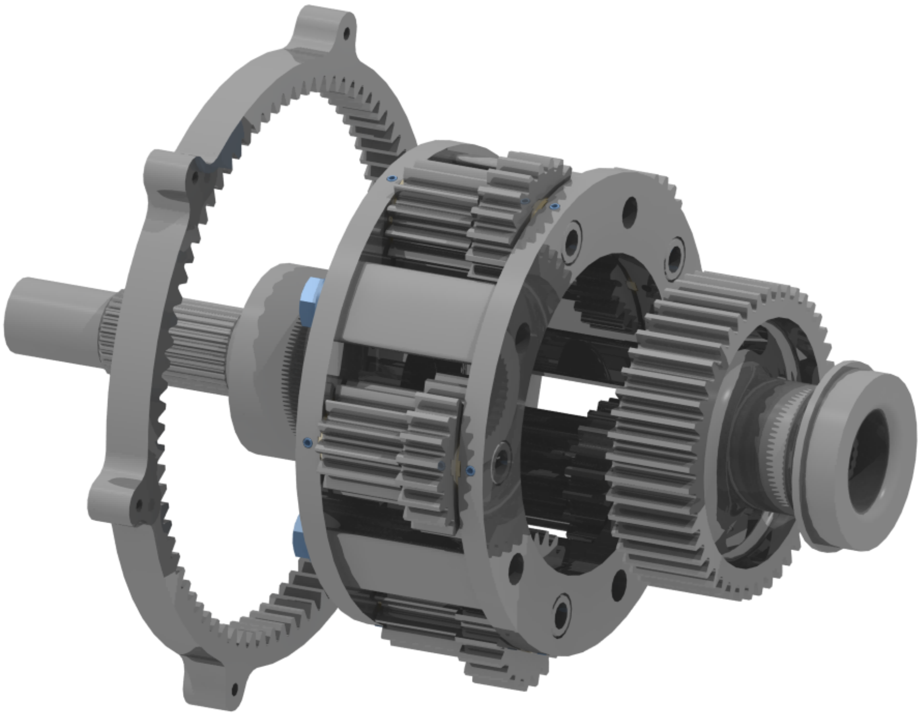

Concept design of hub reduction gear for off-road modified Toyota Hilux

Master's thesis in Automotive Engineering

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Gothenburg, Sweden 2017

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Cover: CATIA 3D-model of Hub planetary gear set for off-road modified Toyota Hilux.

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Concept design of hub reduction gear for off-road modified Toyota Hilux

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Abstract

The task is to develop a proposal for a hub reduction gear for a Toyota Hilux for the company Arctic Trucks. The hub reduction should relieve stress on the power-train, increase traction force and reduce speed from the increase in tire size. The gears has been developed, with analysis to give the expected life and low weight. Requirements on size and packaging has been considered. A full CAD model has been created with all necessary parts included to make a first prototype for testing. A 3D-printed version in scale 1:1 has been created for easy understanding of the final proposal. A spur gear compound planetary design has been chosen because of its simplicity and no axial load. The gear combination that fulfilled all set geometrical and stress constraints and was assessed to be the best option has the gear ratio of **2.19:1**. The gear casing has been designed to act also as the upright with same mounting points as original upright and the original wheal bearing and break caliper can still be used. The added weight due to the hub reduction gear is **20.197** kg at each front wheel and is assumed to be slightly less than that in the rear due to smaller casing. The hub reduction has moved the hub **146**mm outwards and therefore changed the ET offset of the rims from **-115** to **+31**. A different ABS sensor is needed since the original one will not fit. The next step would be to do a prototype and tests the design. The gear manufacturing cost for 1 car is **56620 SEK** and for 10 cars it is **255300 SEK**.

Keywords: Gear ratio, Helical gear, Module, Off-road vehicle, Pitch diameter, Planetary gear, Spur gear, Pressure angle, Tooth stress.

Acknowledgements

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In the end we want to thank our families and friends for there support during our many years of studies.

Joakim Gullstrand, Sigurdur Indridason, Gothenburg, June 2017

Acronyms

Acronym	Description
CAD	Computational Aided Design
CAE	Computational Aided Engineering
CTIS	Central Tire Inflation System
FEM	Finite Element Method
HPSTC	Highest Point of Single Tooth Contact
LOA	Line of action
LPSTC	Lowest Point of Single Tooth Contact
OEM	Original Equipment Manufacturer
R&D	Research and Development

Nomenclature

Greek symbol	Description	Unit	First used
η	Driveline efficiency	-	Eq.3.4
θ	Hill slope	$^\circ$	Eq.3.4
ρ	Density	Kg/m^3	Eq.3.5
ρ_{air}	Air density	Kg/m^3	Eq.3.4
ρ_{a0}	Radius of the circular tip of the tool	mm	Eq.2.23
σ_B	Tensile strength	MPa	Eq.3.2
σ_F	Tooth bending stress	MPa	Eq.2.35
σ_{FP}	Allowable bending stress	MPa	Eq.2.37
σ_{FP1}	Permissible bending stress, Pinion.	MPa	Eq.2.25
σ_{FP2}	Permissible bending stress, Gear.	MPa	Eq.2.25
σ_H	Hertzian contact stress	MPa	Eq.2.27
σ_{HP}	Permissible contact stress	MPa	Eq.2.33
σ_L	Lewis bending stress	MPa	Eq.2.26
σ_y	Yield strength	MPa	Eq.3.2
ϕ	Standard transverse pressure angle	rad	Eq.2.6
ϕ_n	Normal pressure angle	rad	Eq.2.6
ϕ_r	Operating transverse pressure angle	rad	Eq.2.11
ψ	Helix angle	rad	Eq.2.4

Roman symbol	**Description**	**Unit**	**First Used**
$a1$	Life adjustment factor	-	Eq.2.60
a_{skf}	SKF mofification factor	-	Eq.2.60
A	Frontal area	m^2	Eq.3.4
b	Face width	mm	Eq.2.26
BH	Surface hardness	-	Tab.3.9
c	Clearence	mm	-
C	Basic dynamic loading	kN	Eq.2.58
C_{1-6}	Distance along the LOA	mm	Eq.2.13-2.18
C_d	Drag coefficient	-	Eq.3.4
C_r	Operating centre distance	mm	Eq.2.11
d	Pitch diameter	mm	Eq.2.1
d_1	Pitch diameter sun gear	mm	Eq.3.1
d_2	Pitch diameter planet stage 1	mm	Eq.3.1
d_3	Pitch diameter planet stage 2	mm	Eq.3.1
d_4	Pitch diameter ring gear	mm	Eq.3.8
D_{life}	Distance during Lifetime	Km	Tab.1.1
d_{w1}	Operating pitch diameter of pinion	mm	Tab.2.28
E	Young's module	GPa	Eq.2.27
f_r	Rolling resistance coefficient	-	Eq.3.4
F_{1-2}	Tang. force between sun and planet, stage 1	N	Eq.2.44
F_{3-4}	Tang. force between ring and planet, stage 2	N	Eq.2.45
$F_{lateral}$	Maximum lateral force on one tire	N	Eq.3.17
F_{normal}	Maximum normal force on one tire	N	Eq.3.16
F_{out}	Force on planet shaft	N	Eq.2.46
g	Gravitational constant	$\mathrm{m/s}^2$	Eq.3.4
G_1	Ratio first gear	-	Tab.1.1
G_2	Ratio second gear	-	Tab.1.1
G_3	Ratio third gear	-	Tab.1.1
G_4	Ratio fourth gear	-	Tab.1.1
G_5	Ratio fifth gear	-	Tab.1.1
G_6	Ratio sixth gear	-	Tab.1.1
h_a	Addendum	mm	Eq.2.9
h_{a0}	Addendum of the tool	mm	Eq.2.23
h_{aP0}	-	mm	Eq.2.22
h_f	Dedendum	mm	-
h_t	Whole depth	mm	-
H_R	High range ratio	-	Tab.1.1
HRC	Surface hardness	-	Tab.3.2

K_B	Rim thickness factor	-	Eq.2.35
K_H	Load distribution factor	-	Eq.2.28
K_o	Overload factor	-	Eq.2.28
K_s	Size factor	-	Eq.2.28
K_v	Dynamic factor	-	Eq.2.28
L_{10}	Basic rating life	rev	Eq.2.58
L_{10h}	Basic rating life	Hours	Eq.2.59
L_{nm}	SKF rating life	rev	Eq.2.60
L_{nmh}	SKF rating life	Hours	Eq.2.61
L_h	Lifetime	Hours	Eq.2.34
L_R	Low range ratio	-	Tab.1.1
m	Curb weight vehicle	kg	Tab.1.1
m_1	Estimated mass sun gear	kg	Eq.3.5
m_2	Estimated mass of stage 1 planet gear	kg	Eq.3.6
m_3	Estimated mass of stage 2 planet gear	kg	Eq.3.7
m_4	Estimated mass of ring gear	kg	Eq.3.8
m_a	Maximum rear axle weight	kg	Tab.1.1
m_G	Gear ratio	-	Eq.2.3
m_{Gp}	Ratio between sun and planet	-	Eq.2.57
m_n	Module or normal module	mm	Eq.2.1
m_p	Transverse contact ratio	-	Eq.2.20
m_{tot}	Estimated total mass of planetary gears	Kg	Eq.3.9
m_t	Transverse metric module	mm	Eq.2.35
n_L	Stress cycles	-	Eq.2.34
$n_{planets}$	Number of planets	-	Eq.2.44
N	Number of cycles	-	Eq.2.3
N_f	Cycles to failure	-	Eq.2.3
p	Circular pitch	mm	Eq.2.2
p_l	Exponent of life equation	-	Eq.2.58
P	Diametrical pitch	mm	Eq2.26
P_b	Transverse base pitch	mm	Eq2.12
P_{load}	Equivalent dynamic bearing load	kN	Eq.2.58
P_{res}	Resistance power	W	Eq.3.4
q	Number of contacts per revolution	-	Eq.2.34
r_1	Pitch radius of sun gear	m	Eq.2.42
r_2	Pitch radius of planet gear stage 1	m	Eq.2.42
r_3	Pitch radius of planet gear stage 2	m	Eq.2.42
r_4	Pitch radius of ring gear	m	Eq.2.42
r_{eg}	Equivalent radius of gear	m	Eq.2.27
r_{ep}	Equivalent radius of pinion	m	Eq.2.27
r_{ps}	Radius of planetshaft	m	Eq.3.5
r_{ss}	Radius of sun gear shaft	m	Eq.3.6
R_1	Pitch radius for pinion	m	Eq.2.4
R_2	Pitch radius for gear	m	Eq.2.5

R_{b1}	Base radius for pinion	mm	Eq.2.7
R_{b2}	Base radius for pinion	mm	Eq.2.8
R_{fd}	Final drive ratio	-	Tab.1.1
R_{o1}	Addendum radius for pinion	mm	Eq.2.9
R_{o2}	Addendum radius for pinion	mm	Eq.2.10
$R_{w,orig}$	Radius of original wheel	m	Tab.1.1
$R_{wheel,44}$	Radius of 44inch wheel	m	Tab.1.1
S_F	Safety factor for bending	-	Eq.2.37
S_H	Safety factor for pitting	-	Eq.2.33
t_r	Thickness of ring gear	m	Eq.3.8
$T_{20\%}$	Torque used for lifetime calculations	Nm	Tab.1.1
T_{in}	Input torque to sun gear	Nm	Eq.2.44
T_{Max}	Maximum traction torque at one wheel	Nm	Tab.1.1
T_{out}	Output torque	Nm	Eq.2.48
T_{ring}	Reaction torque on the ring	Nm	Eq.2.47
v	Poisson's ratio	-	Tab.3.2
v_t	Pitch line velocity	m/s	Eq.2.30
V	Velocity	m/s	Tab.1.1
V_{1-2}	Velocity between sun and planet, stage 1	m/s	Eq.2.52
V_{out}	Velocity carrier	m/s	Eq.2.53
w	Rotational speed	rpm	Eq.2.34
w_e	Engine speed	rpm	Eq.3.2
w_{in}	Input rotational speed	rad/s	Eq.2.52
w_{out}	Output rotationalspeed	rad/s	Eq.2.55
w_p	rotational speed planet	rad/s	Eq.2.54
$x_{1\ min}$	Minimum profile shift coefficient for undercut	-	Eq.2.21
x_{stage1}	Profile shift coefficient stage 1	-	Tab.4.2
x_{stage2}	Profile shift coefficient stage 2	-	Tab.4.2
$y_{1\ min}$	Minimum profile shift for undercut	mm	Eq.2.22
Y	Lewis form factor		Eq.2.26
Y_θ	Temperature factor	-	Eq.2.33
Y_J	Bending geometry factor	-	Eq.2.35
Y_{J1}	Bending geometry factor,pinion	-	Eq.2.25
Y_{J2}	Bending geometry factor,gear	-	Eq.2.25
Y_N	Stress cycle factor	-	Eq.2.37
Y_Z	Reliability factor	-	Eq.2.33
z	Tooth number	-	Eq.2.1
z_1	Pinion or sun gear tooth number	-	Eq.2.3
z_2	Wheel or first stage planet tooth number	-	Eq.2.3
z_3	Second stage planet tooth number	-	Eq.2.41
z_4	Ring gear tooth number	-	Eq.2.39
Z	Active length of line	-	Eq.2.19
Z_E	Elastic coefficient	-	Eq.2.28
Z_I	Geometry factor for pitting resistance	-	Eq.2.28
Z_N	Stress cycle factor	-	Eq.2.33
Z_R	Surface condition factor	-	Eq.2.28
Z_W	Hardness ratio factor	-	Eq.2.33

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1 Introduction

1.1 Arctic Trucks

Arctic Trucks is an Icelandic company which was founded 1990. They are also located in Norway, UAE, UK, Russia, Poland and Finland. They do custom modification on 4x4 vehicles so they can be driven in very harsh environment spanning from deserts to the North and South Pole, and also on normal roads. Arctic Trucks is leading in the world in their fields and are working in collaboration with OEM manufactures like Toyota, Nissan, Mercedes-Benz and many more to get the best solutions. The levels of modification can be divided in 3 levels: Sport & Utility, Professional and Exploration. Sport & Utility vehicles are slightly re-engineered, aimed for normal people. Professional vehicles are re-engineered for professionals like military, police forces, rescue forces and others that require excellent mobility. Exploration vehicles are heavily re-engineered for the most extreme environments like the North and South Pole (see Figure 1.1). Since 1997 Arctic Trucks offers guided expeditions in Iceland and Antarctica. Expeditions helps Arctic Trucks increase their knowledge and quality of their cars. Top Gear used vehicles from Arctic Trucks when they drove to the magnetic North Pole in 2007 [16].

(a) Toyota Hilux AT44 4x4 [41]

(b) Toyota Hilux 6x6 exploration vehicle [42]

Figure 1.1: Example of vehicles that Arctic Trucks have modified.

1.2 Background

Arctic Trucks just finished developing new 44” tires in collaboration with NOKIAN tires. Large wheels makes it easier for the vehicles to drive in rough terrain, unfortunately there is a drawback with larger wheels, they give greater torque which leads to high stresses on the drive-train as well as higher vehicle speed. As can be observed in Figure 1.2 there is a risk that for example drive axles or differentials break when they get exposed to such high stresses.

(a) Broken axle

(b) Broken final drive

Figure 1.2: Example of damage that can occur in modified 4x4 vehicles[26]

So to prevent this from happening a gear reduction out on each hub could be implemented together with a stronger suspension setup, to lower the stress on the drive axles and other parts of the drive-train.
There are already vehicles that uses hub reductions. It is common that trucks and heavy machinery vehicles uses planetary hub reduction to relief the drive shaft from stresses but in these vehicles, size and weight is usually not a concern [39]. Terrain vehicle such as the AM General Hummer H1 and Mercedes-Benz Unimog uses portal hub reduction or (portal axle) to relief torque from their drive trains [37]. A portal axle is when the drive axle is above the center of the wheel hub due to a gear reduction at the hub. A portal axle has the advantages that the ground clearance can be increased or decreased depending on how the hub reduction is installed [37]. There are also many companies that specializes in manufacturing of Portal hub reductions for commercial vehicles such as Jeep, Land rover, Toyota and others [26], [31].
All vehicles have different gear ratios and final drive ratio, and the hub reduction is dependent on what characteristics that vehicle should have, such as speed, wheel torque, fuel consumption etc.
Arctic Trucks would like to investigate if it is possible to design a planetary hub reduction gear for a 2016 model Toyota Hilux. Possible changes to wheel bearings, uprights and brakes have to be considered when designing the Planetary hub reduction.

1.3 Problem statement

The major questions are if it is feasible to use a planetary hub reduction gear in a vehicle like this and how many of the original parts has to be replaced to make it work? Is it possible to make the hub reduction durable enough for these harsh conditions and at the same time make it small enough so the hub reduction will fit inside the wheel rim? The aim with the project is therefore to make a feasibility-study to answer these questions and come up with a conceptual design of a hub reduction setup that meets all initial requirements which can be taken further into the prototype stage.

1.4 Scope and limitations

The scope of this thesis will be on the gear design itself and to give basic ideas on how it can be implemented as a hub reduction. The possible influences on the vehicle dynamics is not part of the scope of this thesis. Due to limited time it will not be possible to do optimization's and detailed simulations and the focus is therefore to deliver a complete package that gives a good idea of a possible final outcome and provide a base for further improvements. The installation design will be focused on the front axle due to limited time. There is not time for any physical test of the components in this thesis due to the limited time but that is a very important step of the product development.

1.5 Deliverables

- Gear specifications and calculations.
- Give possible gear solutions.
- A complete concept design in CAD of a planetary hub reduction that should fit on a Toyota Hilux 2016 year model with NOKIAN Hakkepelitta AT44 (LT475/70R17).
- Make drawings.
- Comparison with other solutions already on the market.
- Investigation of possibility of a Central Tire Inflation System (CTIS) together with the planetary hub reduction gear set.
- A 3D-printed model in scale 1:1.

1.6 Prerequisites

In Table 1.1 all major input values that have been used in this project are presented.

Table 1.1: Vehicle data for Toyota Hilux AT44 [1].

m	Curb weight		3650 [Kg]
m_a	Maximum weight on rear axle		2000 [Kg]
T_{Max}	Maximum traction torque at one wheel		6950 [Nm]
$T_{20\%}$	Torque used for life time calculation		1390 [Nm]
D_{life}	Distance for life time calculation		300000 [Km]
V	Average speed		60 [Km/h]
$R_{wheel,44}$	Radius of the 44inch wheel		0.55 [m]
$R_{w,orig}$	Radius of the original wheel		0.39 [m]
L_R	Low range ratio	Aut	2.566
H_R	High range ratio	Aut	1
G_1	Ratio first gear	Aut	3.6
G_2	Ratio second gear	Aut	2.09
G_3	Ratio third gear	Aut	1.488
G_4	Ratio fourth gear	Aut	1
G_5	Ratio fifth gear	Aut	0.687
G_6	Ratio sixth gear	Aut	0.58
R_{fd}	Final drive ratio	(Aut/Man)	4.1/3.6

To make it cheaper for Arctic Trucks to implement the hub reduction it is preferred to use as much of the OEM parts as possible, but some custom solutions have to be made. Below are all the major parts that are desired to be carried over:

- Brake caliper front
- Axles
- Suspension and steering linkages
- Final drive ratio
- Front wheel bearing
- Brake disk

Other preferred requirements from Arctic Trucks which have been considered in the design are:

- Concept design of planetary reduction.
- Reduction ratio of 1.5:1 - 2.5:1.
- Easy to service.

1.7 Work process

The intention of this section is to shortly describe the work process trough out the project. The first thing was to get a master thesis and find examiner and supervisor which was challenging since the thesis is within both suspension and power-train. The thesis work started in beginning of January with two weeks of pre-study. That time was used to read about the subject, gather useful information and study the solutions that already exist and assess there pros and cons.
The next step was to go to Arctic Trucks in Iceland for roughly three weeks to see and understand the vehicles that are going to be the foundation of the project. The requirements and deliverables were determined together with Arctic Trucks R&D department and vehicle information's were gathered.
The main task in this thesis project is the gear design and therefore a lot of time was spent to understand and implement the gear theory which can be tricky to understand. Calculation scripts were made in Matlab based on gear calculation standards to find all possible gear solutions that fulfilled all the constraints and then the solutions were compared to a gear simulation software called KISSsoft.
Next step was to make a CAD design of the gears and planet carrier and perform stress analysis of the gear components such as planet shafts and the sun gear. Parallel to the gear CAD design the bearings for the planets and the sun were selected based on lifetime calculations.
When the gears had been designed in CAD the next thing was to design the casing for the gears and to select appropriate seals. The design took some time since it was hard to make it work together with the OEM brake system but after some brainstorming a solution was found.
In the final phase of the work the focus was on report writing, which actually started during mid CAD design, and presentation preparation.

1.8 Software

In this section all major software's that are used in in this thesis are listed:

- **Matlab:** Matlab has been used for gear calculations and to iterate possible gear combinations that are within the set constraints as well as for any other calculations in this project.
- **KISSsoft:** The software kISSsoft is a specialized gear simulation and analysis software and is used to validate the gear calculations made in Matlab and to generate CAD models of the gears tooth profile.
- **Catia V5:** The CAD software Catia V5 is used to create 3D models of Hub gear casing, Planet carrier, Planet shafts etc. and to make drawings of the components.
- **ANSYS Workbench:** ANSYS was used to analyze stresses in components.

2

Theory

In this chapter all major definitions and equations used in this project are presented and explained.

2.1 Gear basics

Gears transfer movement and work trough the tooth mesh of gear wheel pair. When talking about gear pairs it is common that the smaller gear is called *the pinion* and the bigger one *the gear* and those names will be used in this thesis. Some of the advantages of using gears are[2]:

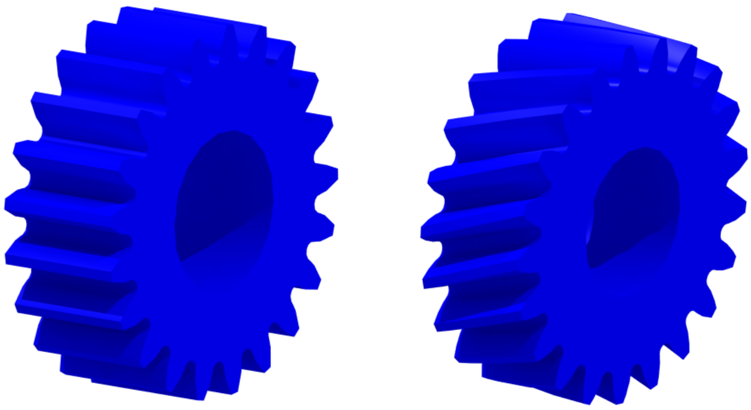
- High efficiency (97-98%)
- Relatively good load capacity considering mass and dimensions of gear wheels.
- Wide range in applications and velocity.
- Small bearing and shaft loads.
- Easy maintenance.

The major disadvantages are:
- High manufacturing and mounting accuracy.
- Noise at high operating velocities.

2.2 Gear teeth concepts

Gear wheels are defined by the orientation of the teeth lines. Teeth lines are lines that follow the teeth surface from one side of the wheel to the other. If the teeth line lies parallel to the gear rotation axis it is called spur or straight cut gear. If the lines are at an angle and have constant pitch the gear is called helical gear[2]. The helical angle is usually $15 - 35°$[7]. These two concepts are the most common for Automotive gears today[2].
The main advantage of the **spur gear** is that it is simple and therefore simpler to design and manufacture compared to a helical design. It is also easier to assemble due to the straight cut gears and it doesn't produce axial load and therefore it is possible to design lighter casing and use smaller bearings. The major disadvantage for spur gears is noise especially for high rotational velocity[30][20].
Helical gears have higher load capability compered to spur gear for the same module and width since it has higher contact ratio(more tooth's in contact). Helical gear engagement is smooth and therefore produce far less noise than the spur gear and

therefore the most common choice for automotive gearboxes. The major disadvantage for helical gear is that due to the angled tooth contact it produce axial force which means that the bearings and the casings need to be dimensioned for greater load which leads to heavier solutions. That is the main reason most racing gearboxes have spur gears. It is often claimed that spur gears are more efficient which is in theory correct due to less contact ratio but the difference compared to helical gears are usually negligible and both solutions have around $97 - 98\%$ efficiency[30][20].



(a) Spur gear wheel **(b)** Helical gear wheel

Figure 2.1: Gear teeth concepts considered in this project.

2.3 Spur gear nomenclature

The spur-gear terminology is based on few main parameters which are illustrated in Figure 2.2. These parameters are [12]:

- **Pitch circle**: Theoretical circle that founds the base for almost all gear calculations by using the **pitch diameter**. The pitch circle of two meshing gears are always tangential to each other.
- **Circular pitch** p: A distance measured on the pitch circle from a point on a tooth to a corresponding point on the next tooth. Can be thought of as the sum of the **tooth thickness** and the **width of space**.
- **Module** m_n: Index of tooth size in SI units, usually expressed in mm. It is the ratio of the pitch diameter to the number of teeth.
- **Addendum** h_a: Radial distance between the **top land** and the pitch circle.
- **Dedendum** h_f: Radial distance between the **bottom land** and the pitch circle. The tooth's **whole depth** h_t is the sum of the addendum and dedendum.
- **Clearance circle** : Tangent to the addendum circle of the mating gear. The **clearance** c is the radial distance from the dedendum circle of a gear to the addendum circle in mating gear.
- **Backlash**: The difference in gears width of space and mating gears tooth thickness at the pitch circle.

The module and circular pitch is expressed as

$$m_n = \frac{d}{z} \tag{2.1}$$

$$p = \frac{\pi d}{z} = \pi m_n \tag{2.2}$$

where m_n is the module in mm , d is the pitch diameter in mm, z is the number of teeth and p is the circular pitch.

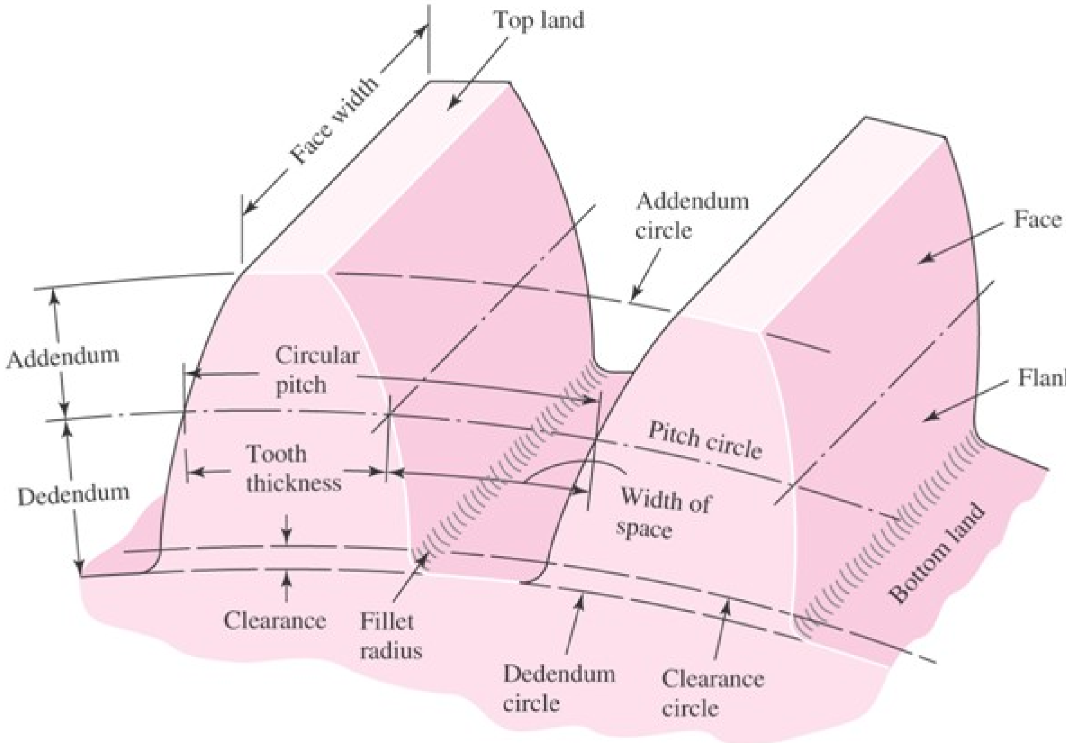


Figure 2.2: Nomenclature of spur-gear teeth[12]

2.4 Basic Gear Geometry

Following equations are according to AGMA-908-B89 [5] and are valid for helical and spur gears. The equations are often the same for both helical and spur gears where the difference is the helix angle, which is 0 for spur gears and will make the calculations simpler for spur gears. These equations applies both for external and internal gears and for those equations that have double sign (*e.g.*$\pm$), the $+$ is for external and $-$ for internal. These equations are made "dimensionless" by letting the normal module be equal to 1 ($m_n = 1$).

The gear ratio between two meshing gears is

$$m_G = \frac{z_2}{z_1} \tag{2.3}$$

where z_2 is the gear(bigger gear wheel) tooth number and z_1 is the pinion(smaller gear wheel) tooth number. This ratio can never be less than 1.

The standard (reference) pitch radius for the pinion, R_1

$$R_1 = \frac{z_1}{2\ cos(\psi)} \tag{2.4}$$

where ψ is the helix/helical angle. This angle is often also presented as β in other literature but since ψ is used in AGMA-908-B89 and will be used in this thesis for consistency. The standard pitch radius for the gear is

$$R_2 = R_1\ m_G \tag{2.5}$$

The standard transverse pressure angle is dependent on the helix angle ψ and the standard normal pressure angle ϕ_n

$$\phi = tan^{-1}\left(\frac{tan(\phi_n)}{cos(\psi)}\right) \tag{2.6}$$

The base radius for the pinion and gear is defined as

$$R_{b1} = R_1\ cos(\phi) \tag{2.7}$$

$$R_{b2} = R_{b1}\ m_G \tag{2.8}$$

the addendum radius as

$$R_{o1} = \frac{z1}{2} + h_a \tag{2.9}$$

$$R_{o2} = \frac{z2}{2} \pm h_a \tag{2.10}$$

since this equation are dimensionless then $m_n = h_a = 1$ [21].

The operating transverse pressure angle ϕ_r is based on the base radius for the pinion and wheel and the operating center distance C_r

$$\phi_r = cos^{-1}\left(\frac{R_{b2} \pm R_{b1}}{C_r}\right) \tag{2.11}$$

The transverse base pitch is expressed as

$$P_b = \frac{2\pi\ R_{b1}}{z_1} \tag{2.12}$$

The pitch point is the point where the pitch circles of two meshing gears intersect (see Figure 2.3). All contact points of two meshing teeth follows a line called **line of action**(LOA) or **pressure line**. The LOA is tangent to the base circles of the pinion and gear and goes trough the pitch point. The normal pressure angle ϕ_n is defined as the angle between the the LOA and a line that is tangent to the pitch circle and goes trough the pitch point(see Figure 2.3 and 2.4). The normal pressure angle can vary depending on the application but the most common angles are $20 - 25°$ [7].

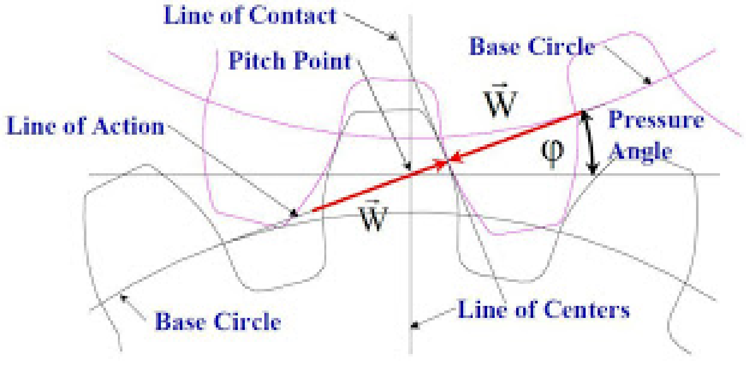
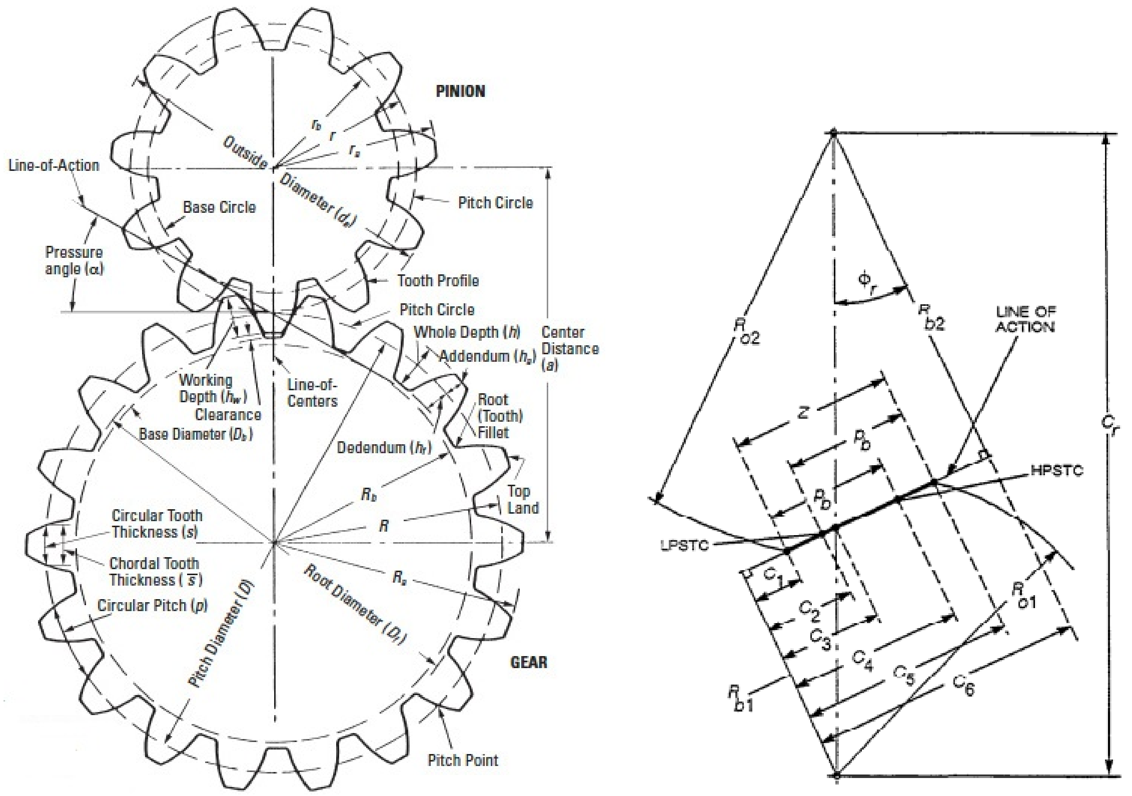


Figure 2.3: Pitch point and Line of action[24]



(a) Pinion and Gear meshing[45]

(b) Definition of the LOA or pressure line[5]

Figure 2.4: Basic gear geometry definitions

It is possible to determine the active length of line of contact Z by calculating the lengths C_1 trough C_6 using the basic gear equations defined in above text(see Figure 2.4).

$$C_6 = C_r\ sin(\phi_r) \tag{2.13}$$

$$C_1 = \pm\left[C_6 - \left(R_{o2}^2 - R_{b2}^2\right)^{0.5}\right] \tag{2.14}$$

$$C_3 = \frac{C_6}{m_G \pm 1} \tag{2.15}$$

$$C_4 = C_1 + P_b \tag{2.16}$$

$$C_5 = \left(R_{o1}^2 - R_{b1}^2\right)^{0.5} \tag{2.17}$$

$$C_2 = C_5 - P_b \tag{2.18}$$

The active length of line of contact is dependent on the lengths C_1 and C_5

$$Z = C_5 - C_1 \tag{2.19}$$

The lowest point of single tooth contact (LPSTC) and the highest point of single tooth contact (HPSTC) are located by C_2 and C_4 respectively.
The transverse contact ratio between a pinion and a gear is determined by the active length of line of action(Z) and the transverse base pitch (P_b).

$$m_p = \frac{Z}{P_b} \tag{2.20}$$

2.5 Failure Modes in Gears

When designing gears it is important to understand the failure modes so the gear can be dimensioned to withstand the loads that are expected to act on the gear teeth and still be within reasonable size and weight. This section will point out the most common one.
Gear failure modes have been named and categorized in a different way based different perspectives. Following are listed few examples of how they have been categorized[8]:

Here it is the lubrication that defines the major categories.

- **Non lubrication** related failure
- **Lubrication related** failure

This classification is more detailed than the above one and is listed in the order of frequency where the most frequent failure mode is at the top.

- **Fatigue**
- **Impact**
- **Wear**
- **Stress rupture**

The third classification focus on where on the tooth the failure occurs and is the one that will be used in this thesis with main focus on the pitting and bending fatigue (see Figure 2.5).

- **Failure modes on gear tooth flanks**, including pitting, scuffing, and wear
- **Failure modes on gear root fillets**, including bending fatigue and impact

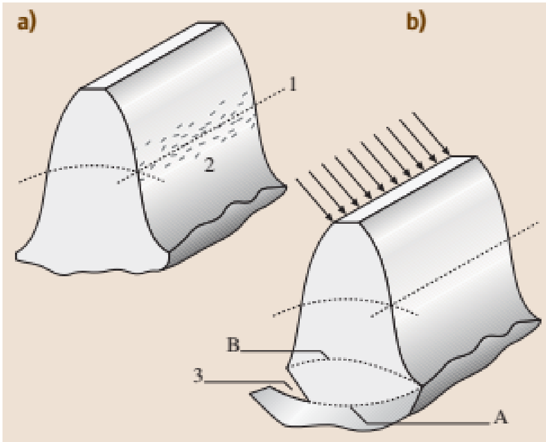


Figure 2.5: Location of the gear failure modes, a) the flank and b) the root radius [2]

2.5.1 Failure modes on gear tooth flanks

The failure modes acting on the tooth flank are pitting ,scuffing and wear as has been mentioned before. These failure modes can be broken down even further but for the purpose of this project only the basic explanation for each mode will be presented.

2.5.1.1 Pitting

Pitting (macro-pitting) or fatigue flaking as it is also sometimes called is a fatigue surface damage and is considered one of the most common gear failure modes. The surface damage is caused by cyclic contact stress also known as Hertzian stress. It usually starts with localized plastic deformation that evolves into small crack on the tooth surface(see **a)** in Figure 2.6). Lubrication oil gets into the crack(see b) in Figure 2.6) and under load the pressure in the lubricant arise(see **c)** in Figure 2.6) and the crack propagates until metallic particle chips of the tooth surface and forms so called pit(see **d)** in Figure 2.6)[2] [8].

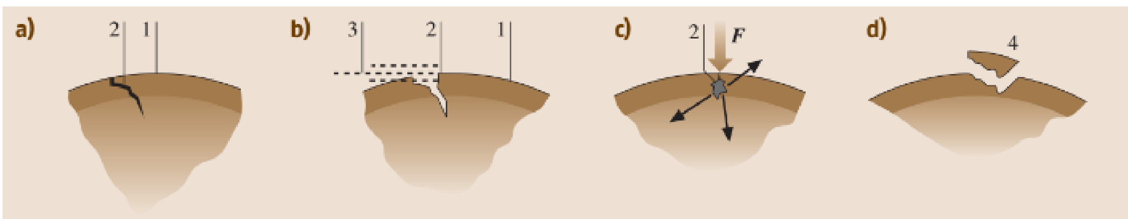


Figure 2.6: Pitting formation due to cyclic contact stress. [2]

When a pit is formed it acts as a stress concentration and spreads out to close areas until the hole flank is covered with pits which can cause fracture of the tooth. How severe the pitting becomes is depended on how high the contact stress is compared to the load cycles[25]. Figure 2.7 illustrates gear tooth flank that has suffered to high cyclic load.

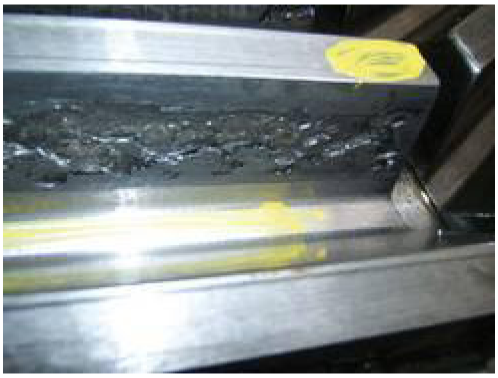


Figure 2.7: Example of fatigue pitting. [49]

2.5.1.2 Scuffing

Scuffing also known as scoring happens when the lubrication film between two meshing teeth brakes down often because of high operating temperature. The lack of lubrication film introduce metal to metal contact at high spots(asperities) and if the load and temperature is high enough the asperities welds together and when the gears continue to rotate some material is removed from the tooth flank surface(see Figure 2.8)[8].

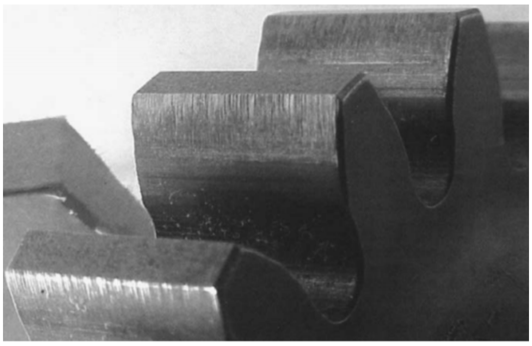


Figure 2.8: Typical example of scuffing. [8]

2.5.1.3 Wear

Wear is deterioration of the loaded tooth flank. This can happen in two ways, either by abrasive or adhesive wear. Abrasive wear occurs when small abrasive particles in the lubrication grinds down the teeth and happens only when two meshing flank surfaces are in sliding contact(see Figure 2.9). Adhesive wear occurs when the lubrication film thickness is not sufficient and material is removed from the mating gear due to local plastic deformation and adhesion caused by pressure between the tooth flanks.

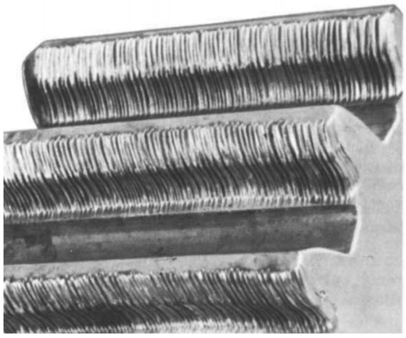


Figure 2.9: Typical example of Abrasive wear. [46]

2.5.2 Failure modes on gear root fillets

The failure modes on the gear root are caused by tooth bending either from fatigue or overload/impact. This failure mode is not common but has the most impact on the gear operation and often causes complete operating failure and damage to other machine components such as bearings and shafts.

2.5.2.1 Fatigue bending

Fatigue bending failure is something that takes place over long time period. The crack starts at the weakest point of the tooth or the root fillet where the bending tensile stress is high as well as the stress concentration. The crack propagates slowly the majority of the gear life but in the end it propagates much faster which leads eventually to tooth fracture(see Figure 2.10)[25].

2.5.2.2 Overload/impact

The bending overload fracture occurs when the momentary load exceeds the tensile strength of the gear material causing fracture at the root[25].

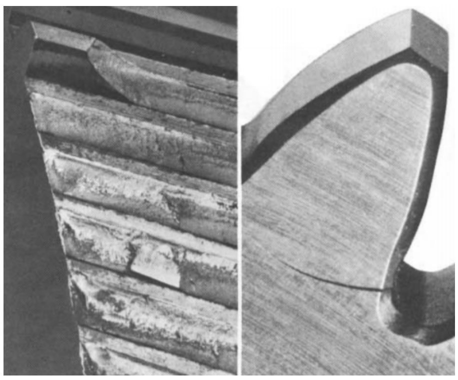


Figure 2.10: Crack propagation (left)and complete tooth fracture due to fatigue bending(right). [46]

2.6 Profile shift

Profile shift is used to improve the performance of meshing gears by moving the tooth either out or inwards so the contact area is moved but the involute profile is the same. The addendum and dedendum circles are moved according to the profile shift which is often + for the pinion and − for the gear but the pitch and base circles are the same. Profile shift was originally used to avoid undercut on gears with less than 17 teeth but later it was realized that it could be used to manipulate and enhance the gear performance.The profile shift does not require special tools if straight sided hobs are used and therefore no extra cost, milling is the only gear manufacturing method that requires special tools[27] .Examples of the things that profile shift can have affect on is:

- Avoiding undercut
- Avoiding narrow top land
- Balanced specific sliding
- Balanced flash temperature
- Balanced bending failure

Balancing the specific sliding is used when improved wear and Hertzian pressure resistance is desired, balanced flash temperature is used to maximize the scuffing resistance and balancing the bending failure maximizes the bending fatigue failure resistance. The profile shift needed to achieve the balanced specific sliding, flash temperature and bending failure are usually different and therefore should the profile shift value be based on the criteria that is considered to be the most important for the given application[10].
Gears wit few teeth are more sensitive to profile shift and therefore it is usually the pinion that decides the amount of profile shift. The affect on tooth shape from profile shift can be seen on Figure 2.11

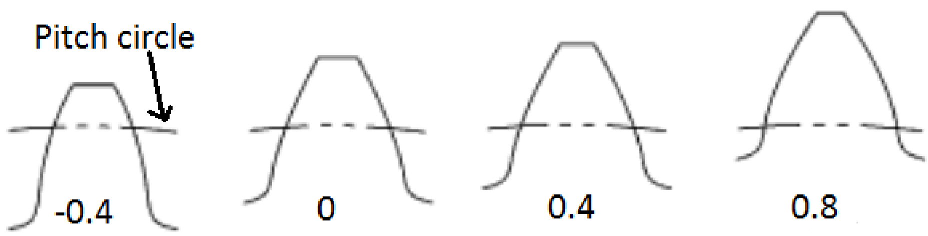


Figure 2.11: The affect of profile shift on gear tooth for 30 teeth gear wheel. [10]

The use of profile shift in this thesis project will be aimed to maximize the load capacity of the gears and therefore only equations for balancing the specific sliding and bending fatigue will be presented as well as for undercut avoidance.Following equation are based on the AGMA 913 a98 standard[10].

The minimum profile shift coefficient x_1 to avoid **undercut** for the pinion is

$$x_{1\ min} = \frac{y_{1\ min}}{m_n} \tag{2.21}$$

where $y_{1\ min}$ is the profile shift and is defined as

$$y_{1\ min} = h_{aP0} - R_1\ sin^2(\psi) \tag{2.22}$$

where

$$h_{aP0} = h_{a0} - \rho_{a0} + \rho_{a0}\ sin(\phi_n) \tag{2.23}$$

where
h_{a0} is the addendum of the tool (in this thesis $h_{a0} = 1.25$)
ρ_{a0} is the radius of the circular tip of the tool (in this thesis $\rho_{a0} = 0.38$)

Specific sliding is defined as the ratio between tooth's rolling and siding velocity at specific point on the line of action. The balanced specific sliding is obtained by iteratively varying the profile shift coefficient for two meshing gears until following equation is satisfied

$$\left(\frac{C_6}{C_1} - 1\right)\left(\frac{C_6}{C_5} - 1\right) = m_G^2 \tag{2.24}$$

The balanced bending fatigue is obtained in similar way as for specific sliding or by iterative varying the profile shift coefficient until the ratio for bending strength geometry factor is equal as the ratio for allowable bending stress.

$$\frac{Y_{J1}}{Y_{J2}} = \frac{\sigma_{FP2}}{\sigma_{FP1}} \tag{2.25}$$

2.7 Stress calculation

The two main failures in gears are tooth breakage and surface pitting /wear as has been mentioned before and are caused by to much bending or contact stress. It is therefore vital that the stresses are calculated correctly. The gear tooth stress equations are based on basic stress equations for bending(Lewis equation) and contact(Hertzian equation) stress. These equations are the foundation for more detailed tooth stress calculations used in different standards.

2.7.1 Lewis bending stress

This was the first equation introduced for bending stress in 1982 where the full load is assumed to be applied at the gears tip as a single cantilever beam as can been seen in Figure 2.12 [12]. The equation is defined as

$$\sigma_L = \frac{F_t \cdot P}{b \cdot Y} \tag{2.26}$$

where

F_t is the force acting on the tooth
P is the Diametrical pitch
b is the gears face width
Y is the Lewis form factor which accounts for the geometry of the tooth and is function of number of teeth, pressure angle and involute depth of the gear.

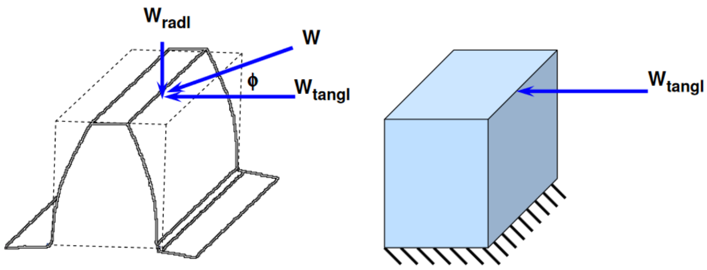


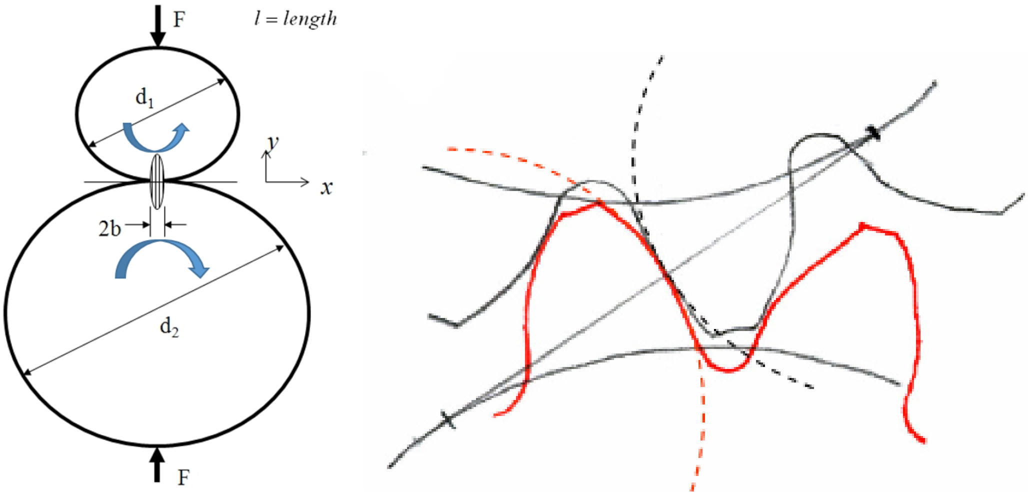
Figure 2.12: Definition of Lewis bending stress [36]

2.7.2 Hertzian stress

The contact stress is highest close to the pressure line and in the pitch point the teeth suffer pure rolling contact and zero sliding. Due to this it is possible to model this situation as Hertzian contact pressure[12]. The Hertzian contact stress for spur gears is defined as

$$\sigma_H = \sqrt{\frac{E \cdot F_t}{2\pi \cdot b}\left(\frac{1}{r_{eg}} + \frac{1}{r_{ep}}\right)} \quad (2.27)$$

where E is the effective modulus of elasticity
r_{eg}, r_{ep} is the equivalent radius of cylinders, equal to the pitch radius times $sin(\phi_n)$



(a) Hertzian contact pressure[50]

(b) Hertzian contact stress method implemented on gears[36]

Figure 2.13: Hertzian stress

2.7.3 Stress standards

More detailed and accurate equations have been derived from the two basic stress equations mentioned above and the main difference is that they include different kinds of factors that should account for multiple things that can affect the stress acting on the gear teeth. There are many standards out there that specifies in details these equations such as ISO,SS, DIN and AGMA. They are in many ways similar since they are often based on the same equations but all of them have their own version of the stress calculations and often different definitions on the factors. For the stress calculations in this thesis the ANSI/AGMA standard will be used or more thoroughly ANSI/AGMA 2101-C95[4]. Following are the main equations used from the AGMA standard with some explanations but for further details it is necessary to read trough the ANSI/AGMA 2101-C95 standard[4].

2.7.3.1 Contact stress calculation for pitting resistance

The equation for the tooth contact stress is as follows

$$\sigma_H = Z_E \sqrt{F_t K_o K_v K_s \frac{K_H Z_R}{d_{w1} b Z_I}} \tag{2.28}$$

where

Z_E is the **Elastic Coefficient**:
The elastic coefficient is based on the pinions and gears Modulus of elasticity E_1, E_2 and the Poisson's ratio v_1, v_2. For steel the Z_E is around 190 N/mm^2 for the pinion and gear .

$$Z_E = \sqrt{\frac{1}{\pi\left[\left(\frac{1-v_1^2}{E_1}\right)+\left(\frac{1-v_2^2}{E_2}\right)\right]}} \tag{2.29}$$

K_o is the **Overload factor**:
The overload factor is meant to take into considerations externally applied load that is higher than the nominal load for short period of time. It is only possible to determine the overload factor accurately by performing many field experiments for particular application. There have been tables made that should give the designer a good estimation depending on the driven and the drive unit. In Figure 2.14 is a example of such a table taken from KISSsoft[9].

Application factor according to DIN 3990/ISO 6336

Working characteristic of the driving machine	Working characteristic of the driven machine			
	uniform	light shocks	moderate shocks	heavy shocks
uniform	1.00	1.25	1.50	1.75
light shocks	1.10	1.35	1.60	1.85
moderate shocks	1.25	1.50	1.75	2.00
heavy shocks	1.5	1.75	2.00	2.25 or higher

Figure 2.14: Overload factor estimations[9]

K_v is the **Dynamic factor**:
The dynamic factor is used to account for internally generated gear tooth dynamic forces due to vibration of the gear masses caused by inaccuracy in manufacturing and meshing of the gears even if the input torque and rotational velocity is constant. The dynamic factor can be estimated by knowing the gear accuracy number,Q_v and the pitch line velocity, v_t. The accuracy number varies from $Q_v = 3$ to $Q_v = 12$ and is dependent on the manufacturing quality, Q_v between 3-7 is for commercial quality gears and Q_v between 8-12 is for precision quality gears.

$$K_v = \left(\frac{A+\sqrt{200 \cdot v_t}}{A}\right)^2 \tag{2.30}$$

where
$A = 50 + 56(1 - B)$ for $5 \leq Q_v \leq 11$
$B = 0.25(12 - Q_v)^{0.667}$

K_s is the **Size factor**:
The size factor is aimed to account for non uniformity of material properties. No standard size factors have been established yet and therefore AGMA suggests to use $K_s = 1$ for most gears if a proper steel, heat treatment and hardening process is

chosen.

K_H is the **Load distribution factor**:
The load distribution factor is used to account for the non-uniform distribution of the load along the line of contact. The load distribution is depended upon

- Manufacturing variation of gears.
- Assembly variations of installed gears.
- Deflection due to applied loads.
- Distortions due to thermal centrifugal effect.

The load distribution factor can be divided in two parts, the face load distribution factor $K_{H\beta}$ and the transverse load distribution factor $K_{H\alpha}$. Standard procedures have not yet been established to determine $K_{H\alpha}$ and therefore is the load distribution factor only dependent on the face load distribution factor $K_{H\beta}$ which can be approximated with following equation

$$K_H = K_{H\beta} = 1 + K_{Hmc}(K_{Hpf}K_{Hpm} + K_{Hma}K_{He}) \tag{2.31}$$

where
K_{Hmc} = lead correction factor
K_{Hpf} = pinion proportion factor
K_{Hpm} = pinion proportion modifier
K_{Hma} = mesh alignment factor
K_{He} = mesh alignment correction factor

For more information's on how the above load distribution factors are defined and calculated see Appendix G or section 15 in ANSI/AGMA 2101-C95.

Z_R is the **Surface condition factor**:
The surface condition factor has not been established but should account for the surface finish residual stress and work hardening. For gears with appropriate surface finish this factor can be set as 1.

Z_I is the **Geometry factor for pitting resistance**:
The geometry factor for pitting should account for the instantaneous radius of curvature based on the tooth's geometry which are used to evaluate the Hertzian contact stress acting on the tooth surface. Z_I is dependent on to many equation to list up in this section so for more information on the calculations see ANSI/AGMA 908-B89[5].

d_{w1} is **operating pitch diameter of the pinion** in mm and is as follows

$$d_{w1} = \frac{2C_r}{m_G \pm 1} \tag{2.32}$$

where $+$ is for external gears and $-$ for internal gears.
The **allowable tooth contact stress** is calculated as follows

$$\sigma_H \leq \frac{\sigma_{HP} Z_N Z_W}{S_H Y_\theta Y_Z} \tag{2.33}$$

where
σ_{HP} is the **allowable contact stress** number which is based on the material properties.

Z_N is the **stress cycle factor** for pitting resistance and is calculated using the number of stress cycle

$$n_L = 60\ L_H\ \omega\ q \tag{2.34}$$

and figure 17 in ANSI/AGMA 2101-C95 where L_H is the lifetime in hours, ω the rotational speed in RPM and q number of contacts per rotation, for sun and ring q is equal to the number of planets but can be set as one for the planets.

Z_W is the **hardness ratio factor** for pitting resistance and depends upon
- Gear ratio
- Surface finish
- Hardness of pinion and gear

for gear and pinion with the same Brinnel values $Z_W = 1$ can be used.

S_H is the **safety factor for pitting**.

Y_θ is the **temperature factor** and for lubrication temperature less than $120°C$ it is set as 1.

Y_Z is the **reliability factor** and accounts for the normal statistic distribution of failure based on field tests. $Y_Z = 1$ equals that there should be fewer than one failure in 100 applications or 99% reliability. The reliability factor is then adjusted if higher or lower reliability is desired(see Table 2.1).

Table 2.1: Reliability factor

Requirements of application	Y_Z
Fewer than one failure in 10000	1.50
Fewer than one failure in 1000	1.25
Fewer than one failure in 100	1.00
Fewer than one failure in 10	0.85
Fewer than one failure in 2	0.70

2.7.3.2 Bending stress calculation

The equation for the tooth bending stress is as follows

$$\sigma_F = F_t K_o K_v K_s \frac{K_H K_B}{b m_t Y_J} \tag{2.35}$$

where F_t, K_o, K_v, K_s and K_H are the same as for the pitting resistance and K_B is the **rim thickness** factor which needs to be adjusted if the rim thickness is not sufficient and therefore does not provide full support and therefore causing increased stress. It is dependent on the Backup ratio m_B as shown on Figure 2.15

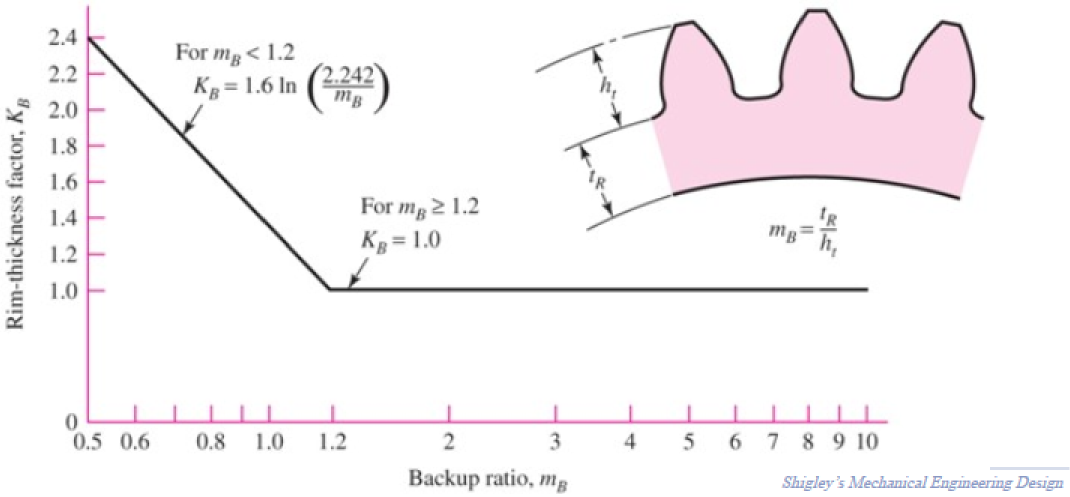


Figure 2.15: Rim factor calculation[12]

Y_J is the **geometry factor for bending** and is based on geometrical values like the geometry factor for pitting. It accounts for the shape of the tooth and the distance from the tooth root to the HPSTC which is the critical point since that is where the most damaging load is applied. To determine the geometry factor for bending aswell as pitting there are many equations that are used, so for further calculation information see ANSI/AGMA 908-B89[5]

m_t is the **transverse metric module**

$$m_t = \frac{m_n}{cos(\psi)} \tag{2.36}$$

The **allowable tooth bending stress** is defined as follows

$$\sigma_F \leq \frac{\sigma_{FP} Y_N}{S_F Y_\theta Y_Z} \tag{2.37}$$

where
σ_{FP} is the allowable bending stress number which is based on the material properties.

Y_N is the **stress cycle factor** for bending resistance and is calculated using Eq. 2.34 and Figure 18 in ANSI/AGMA2101-C95.

S_F is the **safety factor for bending**.

2.8 Gear concepts

In the following text are a few different gear concepts presented which can be implemented as hub reduction and their advantages and disadvantages explained.

Portal hub gear
The portal hub gear has a simple design, only two gears are needed, as can be seen in Figure 2.16. It is possible to increase the vehicles ground clearance with this setup. The input shaft and output shaft will rotate the opposite way and it is therefore necessary to flip the differential in order to make it work on a car that was not designed to have hub reduction[26]. Unfortunately this design has higher bending and contact stress on the gears compared to a planetary solution and therefore makes it a heavier and larger solution.

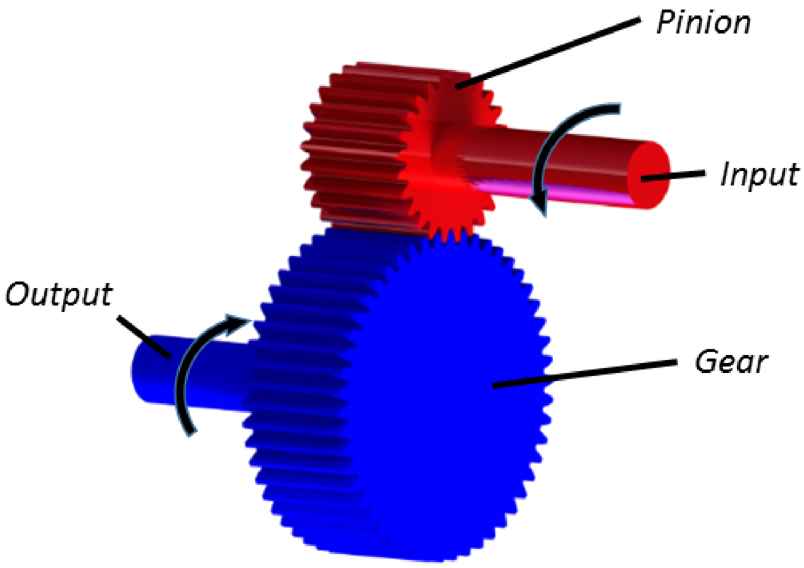


Figure 2.16: Illustration of a Portal hub gear

The gear ratio for the portal hub gear is very simple and is calculated with Eq. 2.38. z_1 is the number of teeth on the pinion and z_2 is the number of teeth on the Gear.

$$m_G = \frac{z_2}{z_1} \tag{2.38}$$

It is possible to relieve stresses from the teeth by having two or more idler gears as a stage between the pinion and the gear as illustrated in Figure 2.17 [23]. By doing this the total force that is transmitted is shared between more teeth and therefore less stress on each tooth, and therefore makes it possible to have a smaller and lighter Pinion and Gear. By adding the Idler gears the Pinion and Gear will now rotate at the same direction so no modifications on the differential is needed.

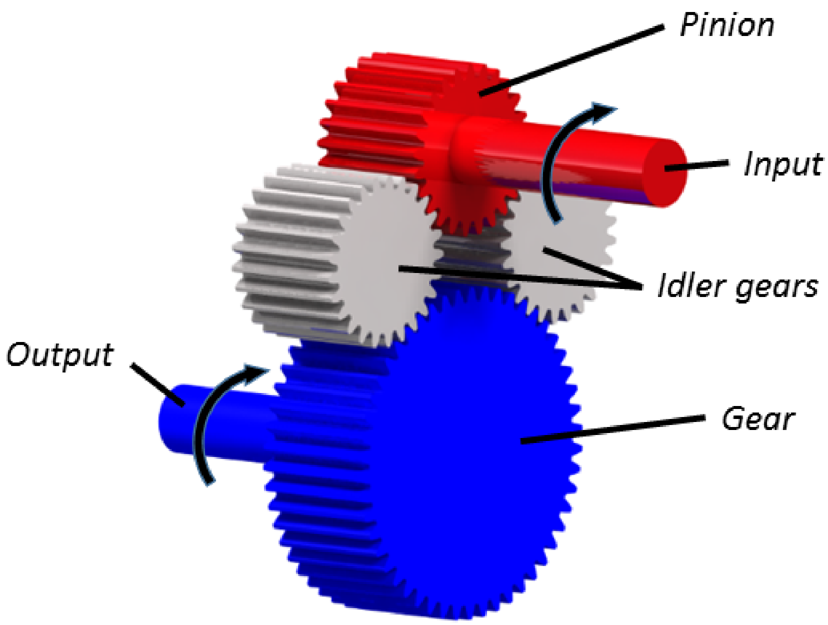


Figure 2.17: Illustration of a Portal hub gear with idler gears

Single stage planetary gear set

In This gear set the force is divided between several so called planet gears has and therefore the gears tooth stress gets lower than on a portal gear. It is possible to get a high reduction ratio if wanted. With the planetary gear it is possible to get different ratio depending of which gear wheel is the drive and which gear wheel is the driven. In Figure 2.18 are two different single stage planetary concept explained.

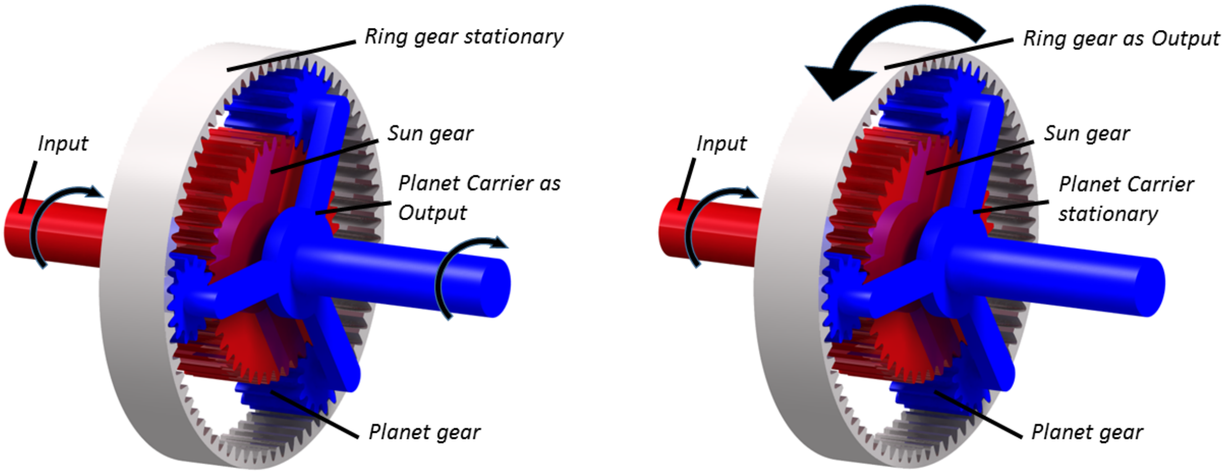


Figure 2.18: Illustration of a Single stage planetary gear set with ring gear stationary(left) and with planet carrier stationary(right).

- **Ring stationary**

With this solution it is not feasible to get a gear ratio higher than 2.5:1, which means that a number lower than 2.5 is not possible. This concept is therefore not interesting for us since we are aiming to be within a gear ratio of 1.5:1-2.5:1 [2]. The Gear ratio for the Single stage planetary gear set with Ring gear as stationary is calculated with Eq. 2.39, where z_1 is the number of teeth on the Sun gear and z_4 is the number of teeth on the Ring gear.

$$m_G = 1 + \frac{z_4}{z_1} \tag{2.39}$$

- **Carrier stationary**

It is feasible to get a reduction ratio of two and greater but to have the ring gear as output causes complications in the design and function of the hub reduction. If the ring gear should turn, it would mean that the gear casing would need to turn as well or the use of big bearing would be needed and that would make it to big and complicated. The ring will also turn in the opposite direction than the sun gear. [2]. The gear ratio for the Single stage planetary gear set with carrier as stationary is calculated with Eq. 2.40, where z_1 is the number of teeth on the Sun gear and z_4 is the number of teeth on the Ring gear.

$$m_G = -\frac{z_4}{z_1} \tag{2.40}$$

Compound stage planetary gear set

Based on the requirements, this solution is deemed to be the most suitable. It gives low stress on the gears and it is possible to get a reduction ratio higher than 2.5:1 within reasonable dimensions. The drawback of the compound planetary gear set is that it will be wider and therefore heavier than a Single stage and be more complex as can be seen in Figure 2.19 [2].

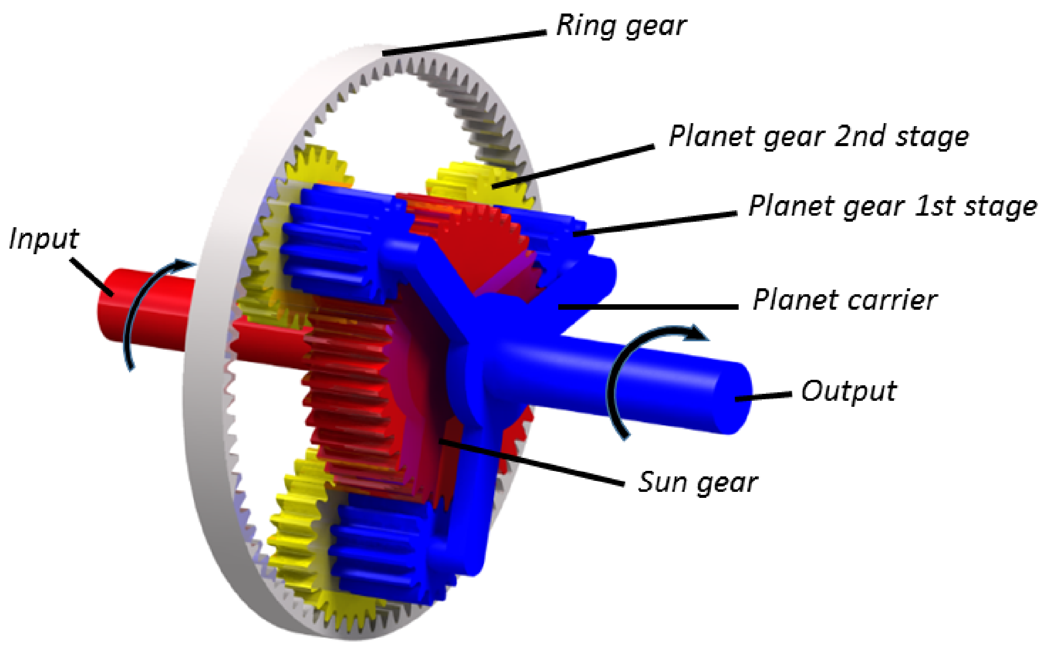


Figure 2.19: Illustration of a Compound stage planetary gear set

The gear ratio for a compound planetary gear set with carrier as stationary is calculated with Eq. 2.41. z_1 is the number of teeth on the Sun gear, z_2 number of teeth on planet gear 1st stage, z_3 number of teeth on planet gear 2nd stage and z_4 is the number of teeth on the Ring gear. How Eq. 2.41 is derived is explained in section 2.9.

$$m_G = 1 + \frac{z_4 \cdot z_2}{z_3 \cdot z_1} \tag{2.41}$$

2.9 Analysis of compound planetary gear

Understanding how planetary gears work can be hard and even harder when you have a compound planetary gear set. This section will focus on the equations behind the operation of compound planetary gear and tried to make it as easy as possible to understand. Figure 2.20 illustrates the layout of the compound planetary gear and the dimension definitions.

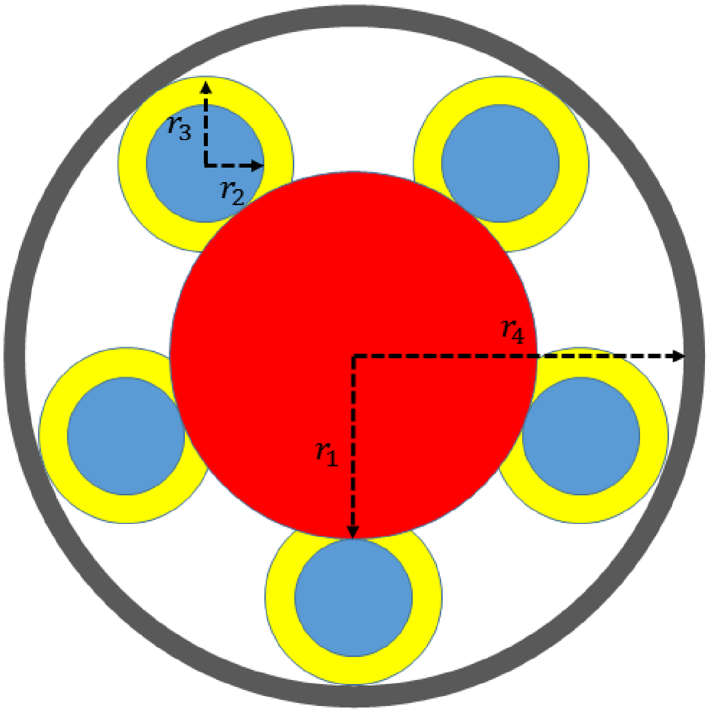


Figure 2.20: The layout of compound planetary gear.

Following equation shows the connections between the gear dimensions

$$r_4 = r_1 + r_2 + r_3 \tag{2.42}$$

and by using Eq. 2.1 we get the same connection between the gear teeth

$$z_4 = z_1 + z_2 + z_3 \tag{2.43}$$

To analyze the forces that are acting on the planetary gear box it is best to brake it up in 3 parts, sun, planets and ring and make free body diagram of each part(see Figure 2.21). Lets start on the sun where T_{in} is the torque on the drive axle(torque out of final drive). Force between the sun and the planets are equally big for all planets in this example for simplification but that is not the case in reality due to manufacturing tolerances.
The force acting on the sun and planet 1 is derived using simple force v.s torque relations.

$$F_{1-2} = \frac{T_{in}}{r_1 \cdot n_{planets}} \tag{2.44}$$

The force F_{3-4} that are acting on planet 2 and the ring can be determined in a similar way as for the planet 1 and the sun. When both the forces that are acting on the planet are known it is possible to calculate the output force F_{out} acting on the planet using force equilibrium.

$$F_{3-4} = F_{1-2}\,\frac{r_2}{r_3} \tag{2.45}$$

$$F_{out} = F_{1-2} + F_{3-4} = F_{1-2}\,\frac{r_2 + r_3}{r_3} \tag{2.46}$$

The reaction torque on the ring can then be derived as

$$T_{ring} = n_{planets} \cdot F_{3-4} \cdot (r_1 + r_2 + r_3) \tag{2.47}$$

which is the torque acting on the gearbox casing.

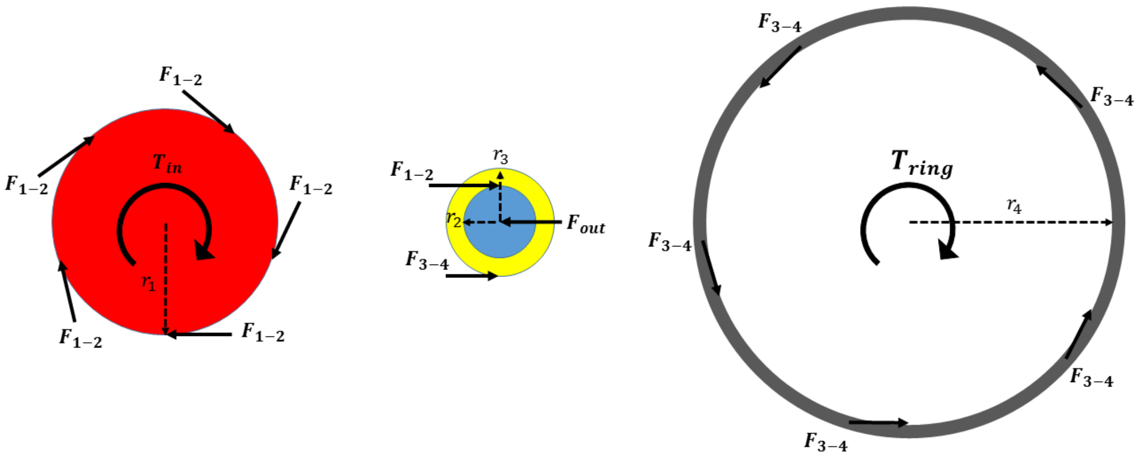


Figure 2.21: Free body diagram of the sun(left), planet(middle) and ring(right).

The output torque can be expressed as

$$T_{out} = n_{planets} \cdot F_{out} \cdot (r_1 + r_2) \tag{2.48}$$

And by combining Eq. 2.44, 2.46 and 2.48, T_{out} can be expressed as a function of the input torque and the sun and planets pitch radius.
The output torque can be expressed as

$$T_{out} = T_{in}\left(\frac{r_1 + r_2}{r_1}\right)\left(\frac{r_2 + r_3}{r_3}\right) \tag{2.49}$$

The over all reduction ratio can now be expressed as function of the sun and planets pitch radius

$$m_G = \frac{T_{out}}{T_{in}} = \left(\frac{r_1 + r_2}{r_1}\right)\left(\frac{r_2 + r_3}{r_3}\right) \tag{2.50}$$

or by number of gear teeth

$$m_G = 1 + \frac{z_4 \cdot z_2}{z_3 \cdot z_1} \tag{2.51}$$

It is also possible to derive the overall reduction ratio by using the angular velocity's. Following equations are derived using Figure 2.22. The first thing is to calculate the pitch line velocity V_{1-2}

$$V_{1-2} = \omega_{in} \cdot r_1 \tag{2.52}$$

by knowing that the pitch line velocity at the ring is 0 since the ring is stationary it is possible to derive the equation for V_{out}

$$V_{out} = V_{1-2} \cdot \frac{r_3}{r_2 + r_3} \tag{2.53}$$

By knowing V_{out} it is possible to calculate ω_p and ω_{out} using the following equations

$$\omega_p = \frac{V_{1-2} - V_{out}}{r_2} \tag{2.54}$$

$$\omega_{out} = \frac{V_{out}}{r_1 + r_2} \tag{2.55}$$

The equation for the final reduction ratio can now be determined by combining Eq. 2.52, 2.53 and 2.55

$$m_G = \frac{\omega_{in}}{\omega_{out}} = \left(\frac{r_1 + r_2}{r_1}\right)\left(\frac{r_2 + r_3}{r_3}\right) \tag{2.56}$$

it is also possible to determine the ratio between ω_{in} and ω_p

$$m_{Gp} = \frac{\omega_{in}}{\omega_p} = \left(\frac{r_2 + r_3}{r_1}\right) \tag{2.57}$$

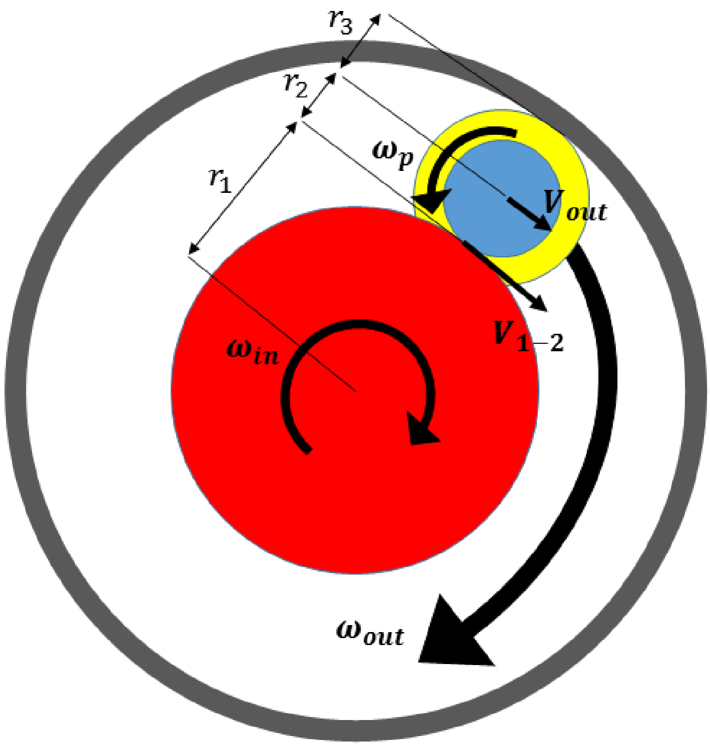


Figure 2.22: Illustration figure for the compound gear pitch line velocity and angular velocity.

2.10 Bearing calculations

The bearing calculations in this thesis where based on the equations presented in the SKF rolling bearings catalogue[14] and Shigley's Mechanical Engineering Design[12]. The basic bearing life equation is in accordance with ISO 281 and is as follow

$$L_{10} = \left(\frac{C}{P_{load}}\right)^{p_l} \tag{2.58}$$

where
L_{10} is the basic rating life at 90% reliability for million revolutions.
C is the basic dynamic loading
P_{load} is the equivalent dynamic bearing load
p_l is exponent of the life equation (3 for ball bearing and 10/3 for roller bearing)

the life can also be expressed in operating hours

$$L_{10h} = \frac{10^6}{60\ w} L_{10} \tag{2.59}$$

where
L_{10h} is the basic rating life at 90% reliability during its operating hours.
w is rotational speed

SKF have come up with their own life rating based on the basic rating life that should estimate more accurately the bearing life bases on the lubrication, degree of contamination, proper mounting and other environmental conditions. The equation is then as follows

$$L_{nm} = a_1 a_{skf} L_{10} = a_1 a_{skf} \left(\frac{C}{P_{load}} \right)^{p_l} \tag{2.60}$$

and for operating hours

$$L_{nmh} = \frac{10^6}{60\ w} L_{nm} \tag{2.61}$$

where

a_1 is the life adjustment factor for reliability(see Table 2.2)
a_{SKF} is the SKF life modification factor (see diagram 1-4 in SKF rolling bearing catalogue).

Table 2.2: Values for life adjustment factor a_1

Reliability	Failure probability	SKF rating life	Factor
	n	L_{nm}	a_1
%	%	million revolution	-
90	10	L_{10m}	1
95	5	L_{5m}	0.64
96	4	L_{4m}	0.55
97	3	L_{3m}	0.47
98	2	L_{2m}	0.37
99	1	L_{1m}	0.25

3
Methods

The first priority has been to calculate and provide a working concept of a spur compound planetary gear set that could work within the constraints given by Arctic Trucks. A spur portal gear set concept with two idle gear and a helical compound planetary gear set concept have also been analyzed and compared to the spur compound planetary gear set.
Planets shaft, planet carrier together with gearbox casing design are explained together with stress analysis and the calculations used for the Spur compound planetary gear set. 3D geometries and drawings of the gear have been produced using CAD tool and Stress simulations have been performed to make sure the components are strong enough for its purpose. Bearing calculations and selection are explained. Sealing selection and gear lubrication alternatives are looked into.

3.1 Gear development

Different hub reduction solutions have been compared. The constraints that have been used when excluding out alternative solutions is presented. How simulation program has been used to exclude possible solutions is explained. Gear lifetime calculations are explained as well as the hub reduction ratio choice and choice of gear material.

3.1.1 Gear geometrical constraints

The compound planetary gear has thousands of solutions that theoretically might work in our design but many of them might not be desired although they are possible. To sort out the solutions that are not desired a few constraints based on the gear geometry where used. Matlab has been used to get different gear combinations with different number of planetary gears, module and teeth numbers that are within the given constraints. The goal have been to get a solution that is as light, small and easy to assemble, that still can handle the loads that will occur during its use. These constraints are presented in following text.

The greatest common divisor for number of teeth for two meshing gears should be one so that the wear is spread over the whole gear (Hunting gears). This is to make sure that not the same gears have contact with each other in every turn and instead distribute the wear between all teeth. Another detail that is good with hunting gears are that if reassembly is needed there is no need to put the gears together in a certain way so that the same teeth is mated as before due to the equal wear on all teeth [6].

To make sure that the compound planetary gearbox can be assembled all the planets needs to be evenly distributed around the sun and oriented in such way that all planets engaging the next tooth at the same time. This is only the case when both the number of teeth on the **ring** and **sun** gear are **divisible by the number of planets**. For a single stage gear this would not be a problem. [38].

The teeth number of the pinion should not be lower than 17 with a pressure angle of 20° to avoid interference between pinion and gear. With undercut it is possible to go down to **14 teeth** when the pressure angle is 20° [11], [5].

A **aspect ratio** has been set to be within the range of 0.3 and 1. The aspect ratio is the ratio between the gears face width and pitch diameter. If the aspect ratio is too high it is likely to get high torsional twisting in the gears. On the other hand, if the ratio is low it is easier to get good manufacturing tolerances and alignment but the tooth gets weaker. [15].

The **maximum pitch diameter of the ring gear** is set to 230 mm to have good packaging. Another condition is that the **diameter of the sun** gear cannot be smaller than 32 mm because that is the diameter of the spindle shaft. The minimum planet diameter is set to 25 mm.

To make sure to not get out a gear combinations from Matlab where the planets are to large and to many so that they **clash** in to each other, a constraint is set so that there will always be a minimum distance of 20 $mm(D_{space})$ between the planets pitch diameters. The constraint is fulfilled with Eq.3.1.

$$\frac{2\pi}{n_{planets}}\frac{d_1 + d_2}{2} - d_3 > D_{space} \tag{3.1}$$

3.1.2 Ratio constraint

The ratio span that has been considered for the hub reduction is from 1.5:1 to 2.5:1 as mentioned in section 1.6. The reduction ratio will have influence on torque and speed for the rest of the power-train and with lower gear ratio(speed ratio), the more affect it will have. Placing bigger tires on the vehicle will result in a lower overall speed ratio from engine to wheels, but it can be compensated to some extend by lowering the final drive ratio. This has been done by Arctic Trucks for the new Hilux where the final drive ratio has been changed from 4.1:1 to 4.88:1. This change in ratio is not big enough to compensate fully for the larger tires. It has been calculated that the minimum additional ratio to compensate for the bigger tires is **1.413:1**.

One thing that was desired was that it would be possible to drive in 6th gear at 100 km/h and therefore be able to use all the gears with in normal operating range since these cars are not driven much faster than 100 km/h. The automatic transmission on the OEM Hilux rarely engages the 6th gear at that speed and with the bigger tires it will not engage at 100 km/h because the engine speed would be too low. Based on a test drive in the new Toyota Hilux where the main focus was to see at what engine speed the car wanted to shift to 6th gear, it was concluded that a ratio around 1.9:1 and higher numbers(lower ratio) would most likely be enough. Lower ratio will result in a stronger and lighter hub reduction but it will also increase the engine speed and lower the possible maximum velocity.

Based on calculations it was decided that ratios lower than 2.2:1 where not desired. All calculations where based on having the original differential with 4.1:1 in ratio but as an alternative option the differential from a Toyota Hilux with manual gearbox could be used as it has a ratio of 3.6:1, which would mean that it would be possible to either lower the engine RPM with the same reduction ratio or lower the ratio while maintaining the same engine speed. Using the 3.6:1 differential would of course introduce more work and cost. The influence from the hub reduction on the engine speed can been seen in Figure 3.1. Equation 3.2 is used to calculate the engine speed for the original Toyota Hilux and Equation 3.3 to calculate the engine speed for the modified Hilux with Hub reduction and 44inch tires. The parameters used in the two equations are found in Table 1.1.

$$w_e = \frac{V}{R_{w,orig}} \frac{60}{2\pi} H_R G_6 R_{fd} \quad [rpm] \tag{3.2}$$

$$w_e = \frac{V}{R_{wheel,44}} \frac{60}{2\pi} H_R G_6 R_{fd} m_G \quad [rpm] \tag{3.3}$$

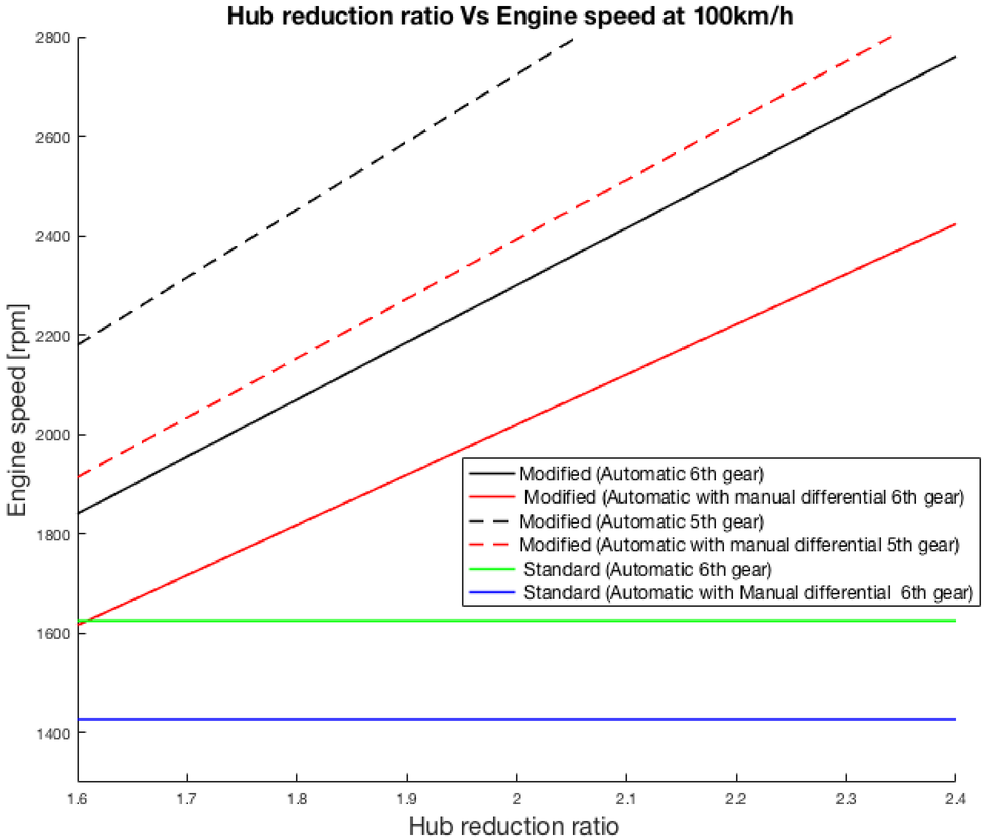


Figure 3.1: How the engine speed changes with respect to Hub reduction ratio at 100km/h

As mentioned before, lower hub reduction ratio means lower mechanical max velocity. The mechanical max velocity means that the power-train can't physically rotate faster due to limited engine speed. The maximum velocity can also be determent by the resistance forces acting an the vehicle which is usually the limiting factor. The resistance forces are from rolling resistance, aerodynamic resistance and hill resistance. When determine the maximum velocity it is manly the aerodynamic force acting when driving straight on flat road. Figure 3.2 illustrates the difference in maximum velocity between two vehicles where one has hub reduction of **2.19:1** and bigger tires and the other is without the hub reduction. For the vehicle without the additional reduction it is the resistance power that determines the max velocity at around 160-170 km/h but for a vehicle with hub reduction it is limited by the engine speed at around 150 km/h. The equation for the resistance power is as follows

$$P_{res} = \frac{(0.5 \cdot C_d \cdot \rho_{air} \cdot A \cdot V^2 + m \cdot g(sin(\theta) + cos(\theta) \cdot f_r)) \cdot V^2}{\eta} \tag{3.4}$$

where the values used can be seen in Table 1.1 and 3.1.

Table 3.1: Values used to determine the resistance power

C_d	Drag coefficient	0.48	-
ρ_{air}	Air density	1.199	kg/m^3
A	Frontal area	2.2	m^2
θ	Hill slope	0	$\circ$
f_r	rolling resistance coefficient	0.01	-
η	Drive line efficiency	0.9	-

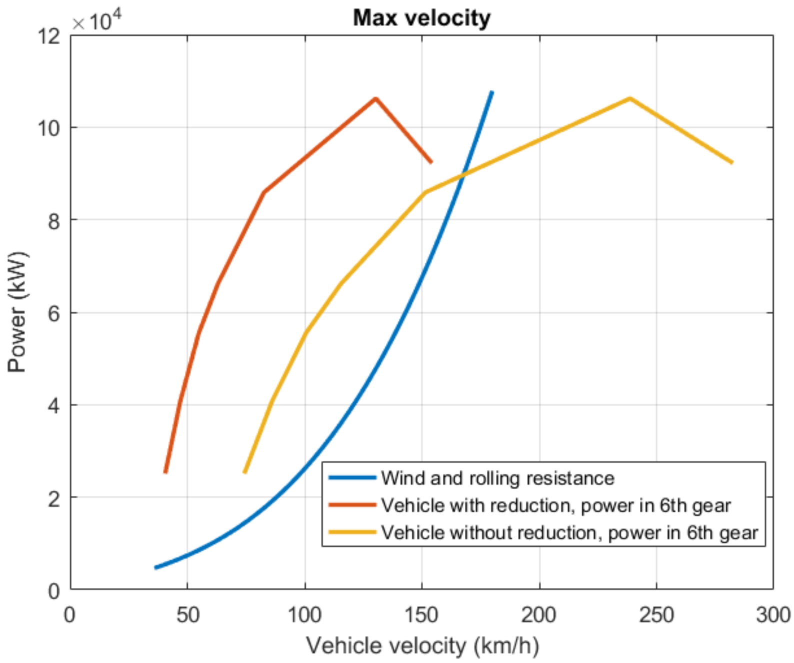


Figure 3.2: Maximum velocity for modified and OEM vehicle in 6th gear on flat road.

The influence that the hub reduction has on the overall vehicle traction force in each gear can be seen in Figure 3.3 where it is obvious that the traction force is significantly higher in the lower gears in the expense of smaller velocity span in each gear and higher engine speed at 100 km/h.

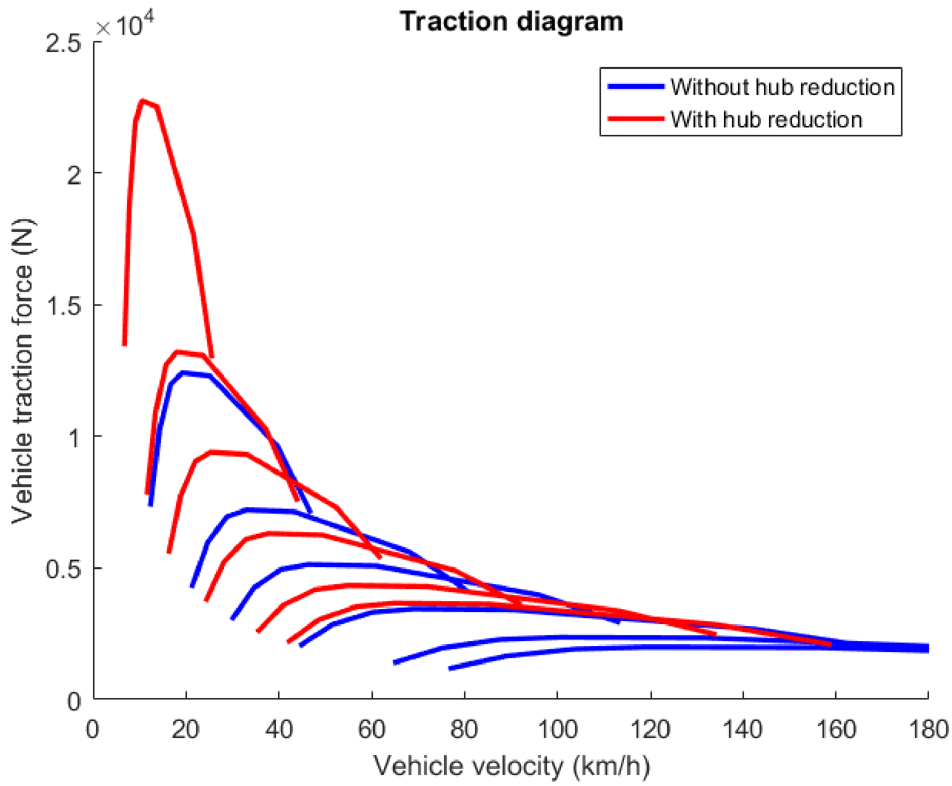


Figure 3.3: Comparison of the traction forces in each gear between a vehicle with hub reduction of 2.19:1 and one without

The Toyota Hilux is traction limited and the maximum torque that the car can be exposed to is calculated to be 6950Nm at one wheel. This number is provided by Arctic Trucks. Divide that by the chosen gear ratio and you get the maximum input torque to the sun gear. With this torque the tangential force to all gears wheels can be calculated with Eq.2.44 and 2.45.

3.1.3 Mass estimation

The mass of all the gears is calculated in Matlab. In the calculations the gears are assumed to be solid. The equation Eq.3.5 - 3.9 has been used for calculating and get a estimation of the total mass of the gears. Where r_{ps} is the radius of the planet shaft, r_{ss} is the radius of the spindle shaft and t_r is the rim thickness of the ring gear.

$$m_1 = \pi \rho b((\frac{d_1}{2})^2 - r_{ps}^2) \tag{3.5}$$

$$m_2 = \pi \rho b((\frac{d_2}{2})^2 - r_{ss}^2) \tag{3.6}$$

$$m_3 = \pi \rho b((\frac{d_3}{2})^2 - r_{ps}^2) \tag{3.7}$$

$$m_4 = \pi \rho b((\frac{d_4}{2})^2 - (\frac{d_4}{2} - t_r)^2) \tag{3.8}$$

$$m_{tot} = m_2 + (m_1 + m_3)n_{planets} + m_4 \tag{3.9}$$

Further weight reduction can be done by hollowing the gears. Most weight saving can be done on the sun gear because that is the largest gear.

Figure 3.4 shows the ratio vs. mass relationship for all solutions that fulfill the predetermined constraints. No solutions have a reduction ratio higher than 2.1:1 and ratios lower than 2.2:1 are not desired as mentioned in section 3.1.2 so only solutions to the left of the red line are considered and it can be seen that the lowest estimated weight is with the ratio of 2.185:1.

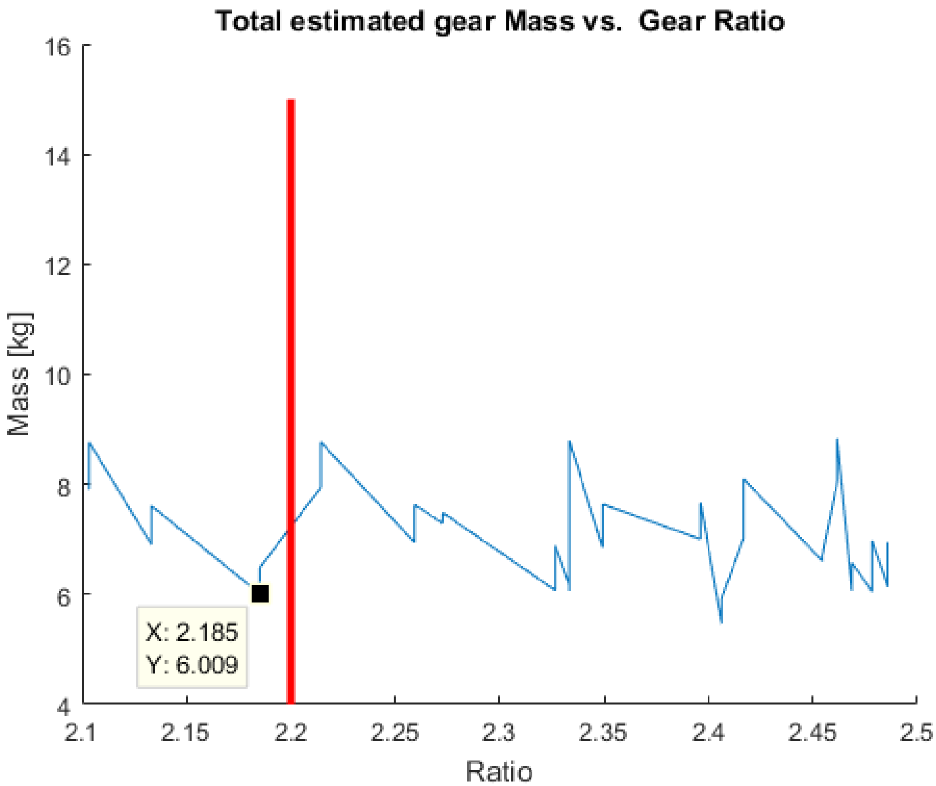


Figure 3.4: Gear mass with respect to Hub reduction ratio

3.1.4 Gear simulation

KISSsoft has been used to verify the matlab calculations. KISSsoft is a design software for mechanical engineering applications. Different kind of gear set types can be selected and analyzed. By filling gear parameters such as module, number of teeth, pressure angle, lifetime etc, KISSsoft will calculate the maximum bending stress, maximum contact stresses and lifetime for the gears. A 3D model of the gears are then created and can be exported from KISSsoft in a format that can be opened and edited in a CAD program.
A drawback with KISSsoft is that it is not possible to choose a compound gear set and analyze. Instead two single planetary gear set is created in KISSsoft where one simulates the contact between the sun and the small planets and the other the contact between the larger planets and the ring. To make this possible a virtual ring for stage one needs to be added and a virtual sun for stage two, see Figure 3.5.

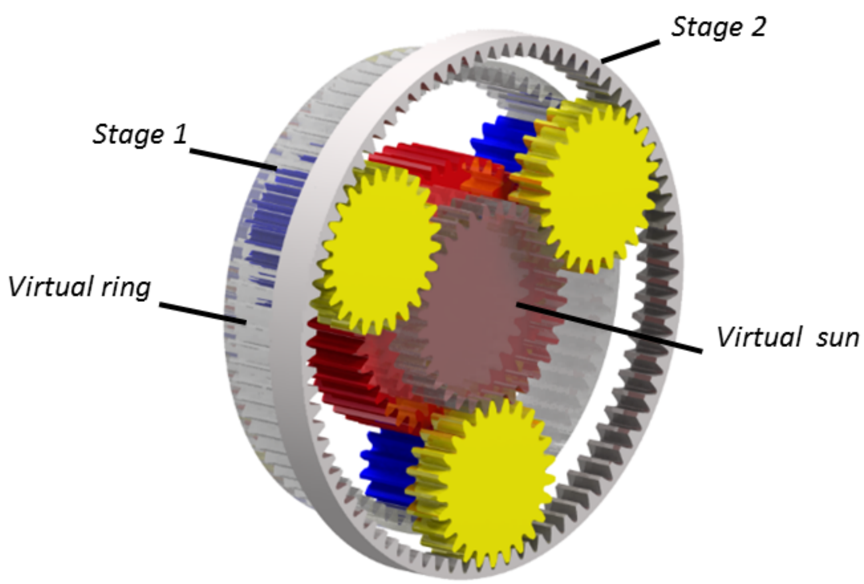


Figure 3.5: Virtual Sun and Ring gear from the software: KISSsoft

3.1.5 Gear Lifetime

The gears are calculated to withstand 20% of the max traction torque of 6950 Nm for 300 000 $Km(D_{life})$ with a average velocity of 60 $Km/h(V)$. This gives a lifetime L_h of:

$$L_h = \frac{D_{life}}{V} = 5000 \ hours \tag{3.10}$$

The Gears are dimensioned to handle the Max torque statically.
A safety factor of minimum **1.2** is set for both static and dynamic loading of the gears and for both bending and contact stress.

3.1.6 Gear Material

To be able to calculate the safety factors for the contact and bending stress it is necessary to choose material for the gears. The material chosen is the Steel Grade 2, Case carburized steel, case-hardened: HRC 58-64(AGMA). The material can withstand 1551.3 Mpa in contact stress and 448.2 Mpa bending stress.

Table 3.2: Material data for the gears and planet shafts [9].

ρ	Density	7950 [Kg/m^3]
σ_y	Yield strength	822 [Mpa]
σ_B	Tensile strength	966 [Mpa]
HRC	Surface hardness	58
σ_{HP}	Permissible contact stress	1551.3 [Mpa]
σ_{FP}	Permissible bending stress	448.2 [Mpa]
E	Young's modulus	206.8 [Gpa]
v	Poisson's ratio	0.3

Other materials with similar strength: 18CrNiMo 7-6 30HRC

3.1.7 Gear rim design

The gear rim design had the focus to minimize the weight but still maintain the enough strength to withstand the loads. The design process was as follows, first a rough dimensioning is done and then some material is removed, then the design is analyzed in ANSYS and adjusted accordingly and iterated until satisfying result obtained. More detailed description of the design of each gear is in the section 4.2

3.1.8 Gear Manufacturing

It is important that the gears are designed in such way that they are manufacturable and that the manufacturing method is in compliance to the requirements such as cost,accuracy, quantity and manufacturing time. The art of gear manufacturing is too broad to go deep into in this section and would probably require few books to give proper coverage so this section will only touch upon few main things related to this project. In Figure 3.6 are illustrated the methods of making gear wheels and in this section the focus will be on the metal removal method or more accurately **Hobbing**, **Shaping** and **Grinding**[3].

Figure 3.6: Hierarchy of gear manufacturing methods [3]

Hobbing is widely used gear manufacturing method and is used for precise gear tooth cutting. The cutting tool, Hob, is threaded with cutting teeth that rotate and cut several teeth at once(see Figure 3.7). The problem with the hobbing method is that it needs sufficient "run out" clearance and therefore impossible to cut two different size gears on the same shaft with small space in between like the compound planets, were there are only few mm between the gear wheels[3]. The hobbing can therefore only be used to cut the sun gear. There are some hobbing tools that can make internal gears but they are rare and expensive as it is relatively new technology and has only been existing for few years[18].

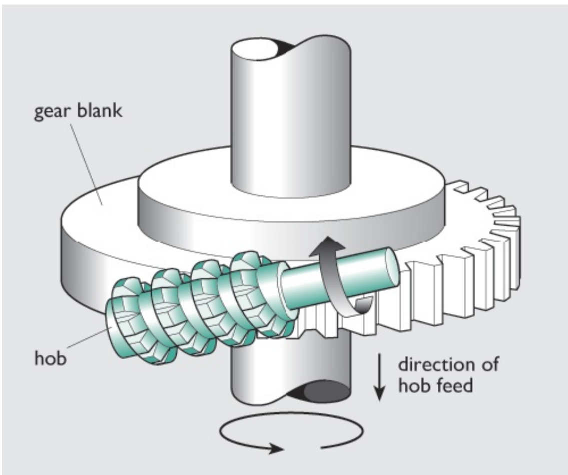


Figure 3.7: Hobbing method [43]

Gear shaping is also common method and is more suited for the planets and the ring gear as it only needs small "run out " clearance and can therefore make gears that are close to each other.The gear shaping cutting tool is shaped as a pinion and cuts with reciprocates movement while rotating according to the gear rotation(see Figure 3.8)[3].

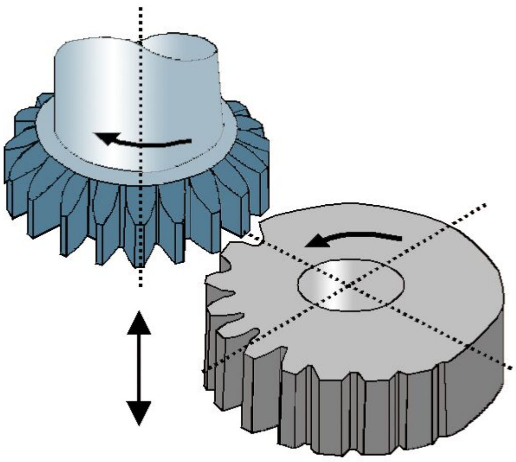


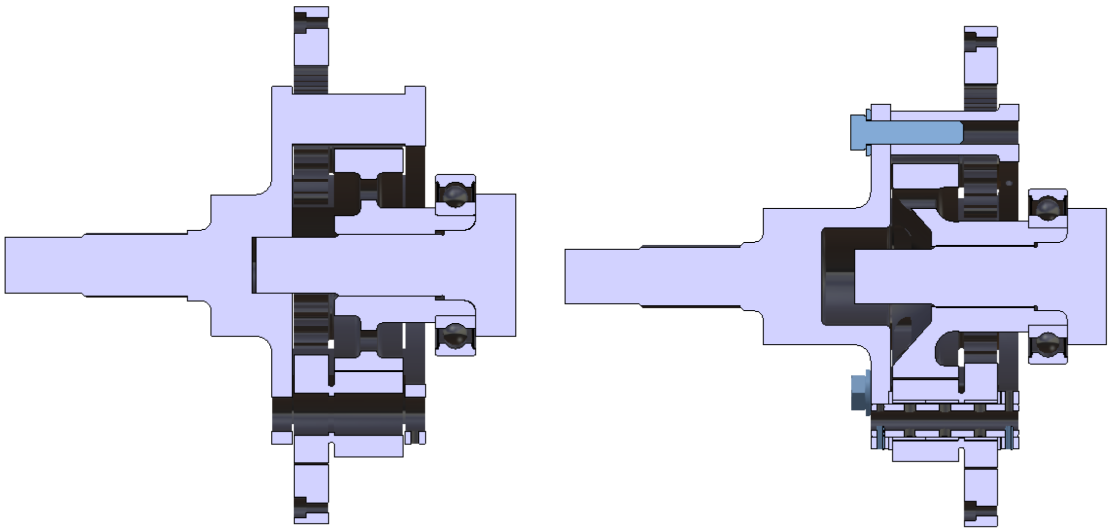
Figure 3.8: Gear shaping method[51]

Grinding is often used to resurface gears teeth after hardening as the teeth surfaces can get distorted due to the high heat. There are many types of grinding but they all have rotating abrasive wheel that grinds the tooth's[3].
There are also the method called Skiving that can both cut external and internal gears. Skiving is quick and suitable for mass production. It is relatively quick and inexpensive compared to other types of gear cutting methods like Gear Shaping[22], [29].

Gears are usually hardened by heat treatment. It is desired to have a stiff and hard surface but a ductile and softer core to make the gears as strong as possible. The sun and planets are case carburized for maximum pitting and bending resistance.
When manufacturing the ring gear, it is possible that the ring can get a oval shape after it has been heated up during hardening. This is because the ring is thin compared to its large diameter and when the ring gets soft when it gets warm and there is therefore a risk that the shape of the ring can change when the ring gear cools down again. To prevent that from happening it might be better to use nitride-hardening instead of case hardening because its done with lower temperature than case-hardening. [8].

3.2 Gear layout

Having a compound planetary gear it is possible to arrange the gears in two possible ways, either have the ring near the input or the output. It was first planed to have the ring near the output as it meant that it would be possible to take the planet carrier out of the gearbox in one piece but because of packaging reasons especially regarding the brake caliper it was decided to move the ring to the input side to make more space in the outer half of the gearbox. That layout has one major drawback and that is that it is not possible to take the planet carrier out in one piece and therefore needs to be disassembled in order to access the drive-shaft bolt but it was considered more important to fit the brake caliper. These two gear layouts can be seen in Figure 3.9



(a) Ring gear closer to output

(b) Ring gear closer to the input-chosen concept

Figure 3.9: Two different layouts of the compound gear.

3.3 Planet shaft

With the gears size and teeth profile done the spindle shaft hole profile could be created in CATIA as well as the planet shafts. The 3D-model of the shaft was simulated in ANSYS to figure out if the shaft had a dimension that was strong enough. To get the loading condition of the shaft, a free body diagram of the forces acting on the shaft was made. Figure 3.10 illustrate the forces acting on the shaft and in Table 3.3 the magnitude of the forces can be seen.

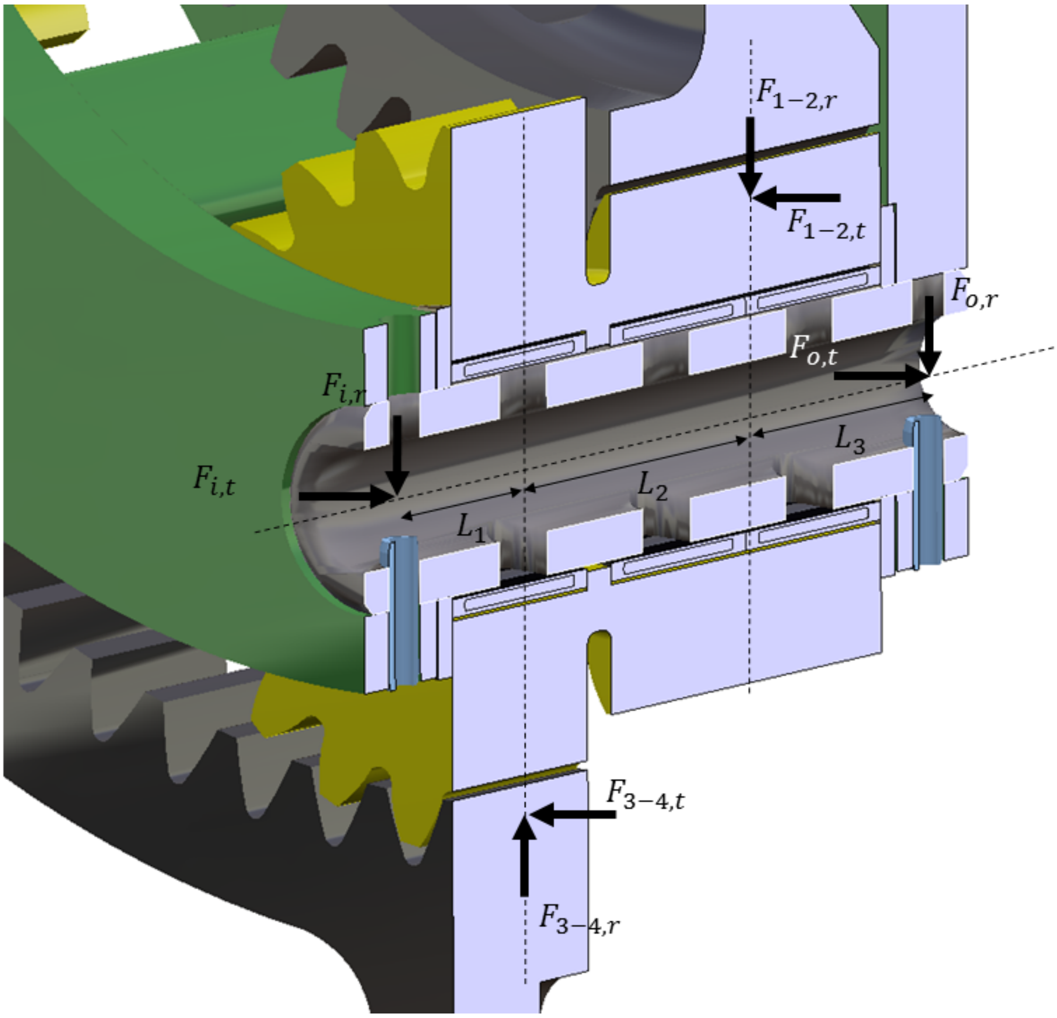


Figure 3.10: Forces acting on the planet shaft and bearings

Table 3.3: Name and magnitude of forces from Figure 3.10

$F_{1-2,t}$	Tangential force between sun and planet	11.3 [KN]
$F_{1-2,r}$	Radial force between sun and planet	4.12 [KN]
$F_{3-4,t}$	Tangential force between ring and planet	7.54 [KN]
$F_{3-4,r}$	Radial force between ring and planet	2.75 [KN]
$F_{i,t}$	Inner tangential reaction force	15.7 [KN]
$F_{i,r}$	Inner radial reaction force	5.7 [KN]
$F_{o,t}$	Outer tangential reaction force	3.2 [KN]
$F_{o,r}$	Outer radial reaction force	1.16 [KN]
L_1	Distance of axle segment 1	0.0095 [m]
L_2	Distance of axle segment 2	0.0285 [m]
L_3	Distance of axle segment 3	0.018 [m]

$F_{1-2,t}$ and $F_{3-4,t}$ are the same forces that are calculated with Eq. 2.44 and 2.45 in section 2.9. By knowing these tangential forces the radial forces can be calculated with Eq. 3.11 and 3.12.

$$F_{1-2,r} = F_{1-2,t} tan(\phi_n) \tag{3.11}$$

$$F_{3-4,r} = F_{3-4,t} tan(\phi_n) \tag{3.12}$$

To easier set up a simulation of the shaft and to make a extreme case of the shaft load, the two tangential forces: $F_{1-2,t}$ and $F_{3-4,t}$ were added together and so were the radial forces $F_{1-2,r}$ and $F_{3-4,r}$. The radial forces are acting opposite each other and therefore the negative sign, see Eq. 3.13 and 3.14.

$$F_{tot,t} = F_{1-2,t} + F_{3-4,t} \tag{3.13}$$

$$F_{tot,r} = F_{1-2,r} - F_{3-4,r} \tag{3.14}$$

$F_{tot,t}$ and $F_{tot,r}$ were brought in to ANSYS and was set to act in the middle of the shaft, to get the maximum stress and lifetime of the shaft. Magnitude of $F_{tot,t}$ and $F_{tot,r}$ can be seen in Table 3.4.

Table 3.4: Magnitude of planet forces in Ansys

$F_{tot,t}$	Total tangential force acting on planet shaft	18.85 [KN]
$F_{tot,r}$	Total radial force acting on planet shaft	1.38 [KN]

The same material as for the gears were chosen for the shaft as well. Figure 3.11 shows the the maximum stress on the final design of the planet shaft. In the figure it can be seen that the maximum stress is below the yield strength of the chosen material(822 Mpa). More stress analysis figures can be seen in Appendix D.

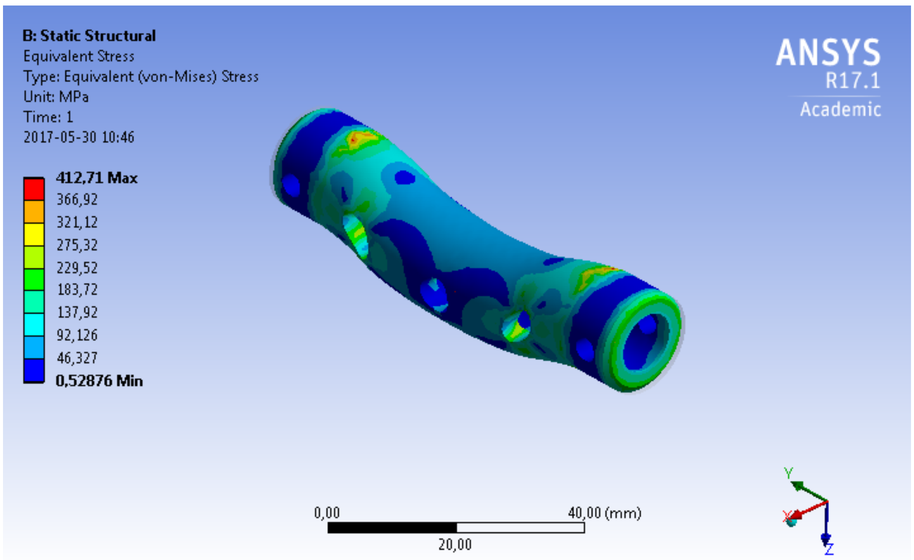


Figure 3.11: FEM analysis on planet shaft with maximum load

The total lifetime of the gearbox is assumed to be: D_{life}=300 000 Km, knowing that, the number of cycles of the shaft can be calculated with Eq. 3.15. Fatigue simulations in ANSYS gave that the shaft could survive 10^9 cycles when exposed to the loading explained in the Figure 3.10.

$$N = \frac{D_{life}}{2R_{wheel,44}\pi} = 8.68 \cdot 10^7 \ Cycles \tag{3.15}$$

Figure 3.12 illustrate the so called SN-curve for Steel. That the maximum possible stress that Steel can handle decreases with the number of load cycles it get exposed of, to a certain limit. After that limit the maximum stress the steel can handle will not decrease more.

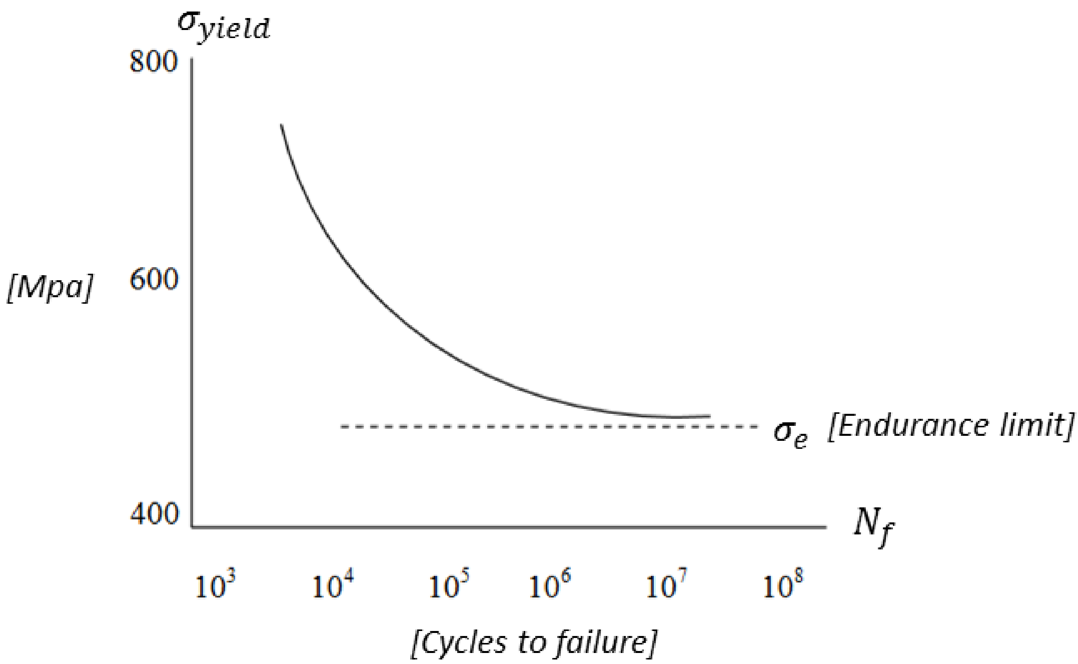
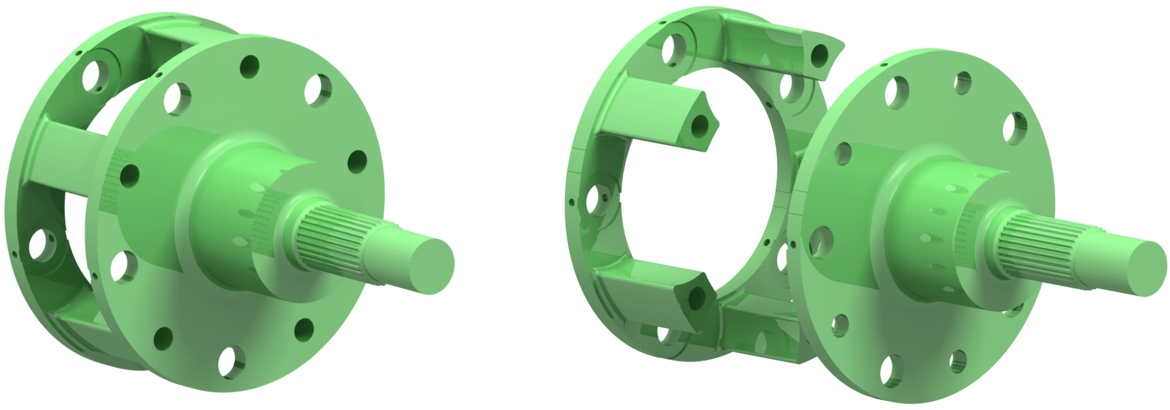


Figure 3.12: S-N curve

3.4 Planet carrier

The planet carrier has been designed in two parts just to make it possible to remove the gears from the upright as mentioned in section 3.2 (see Figure 3.13). Due to that the larger planet is located on the inside of the sun, it is therefore not possible to take out the carrier and planets without taking out the sun gear also. To make it possible to remove the planetary gears it is necessary to loosen the lock nut that locks the sun gear to the spindle shaft. To be able to reach the lock nut, first the casing needs to be separated, then the five bolts that holds the two carrier parts together has to be removed as well as the outer planet shafts spring pins. When this is done the lock nut can be removed and the axle can come out(see Figure 3.14). It was also considered to have the output shaft part of the carrier in two parts so it would be possible to only remove the center output axle and access the nut that way but it did not have much advantage and most likely weaken the carrier.



(a) Planet carrier

(b) Exploded view of the planet carrier

Figure 3.13: Final design of the planet carrier

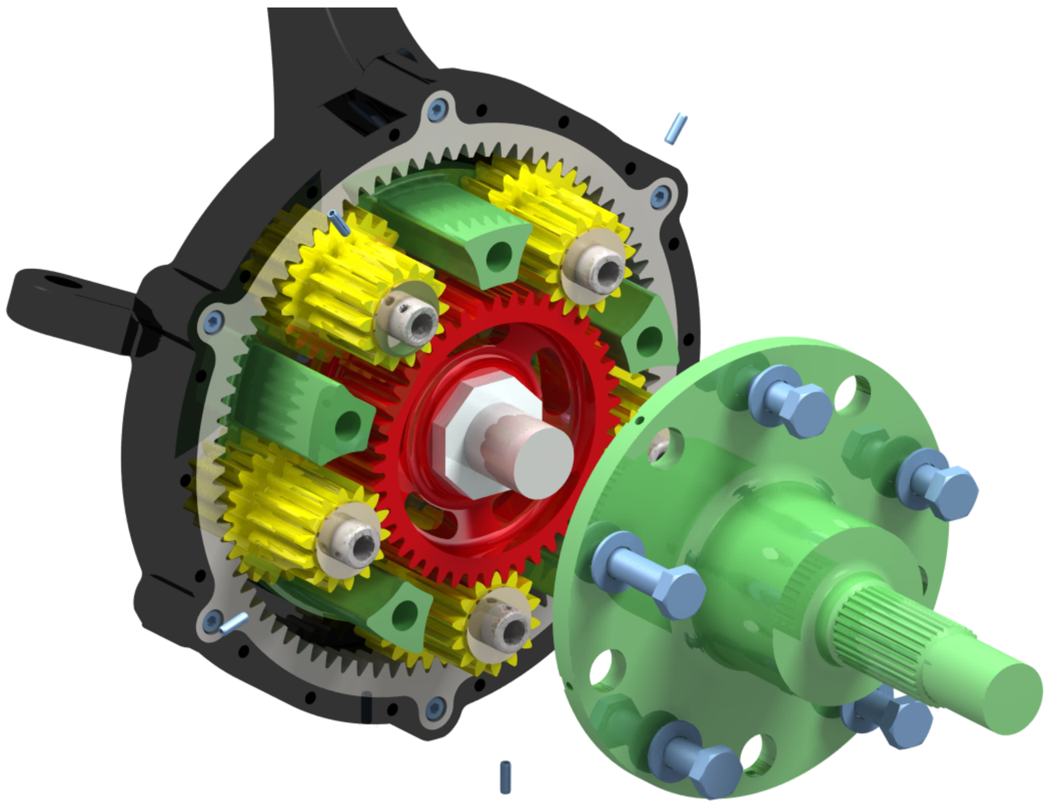


Figure 3.14: Dis-assembly of the carrier to access the axle nut

3.5 Bearings

For smooth operation of the gears, it is necessary to have bearings that allow the planets to rotate freely. There are two possible ways to do that for planetary gear, either the bearings are placed between the planets and the planet shaft or the planet shaft and the planet is in one piece and the bearing is placed between the carrier and the planet gear. The first option is much better for packaging where needle bearings are used between the planet gear and the planet shaft and that's why it is the choose for this project. The forces acting on the needle bearings can be derived by analyzing the forces acting on the planet gear. Illustration on how the gear forces translate to bearing force can be seen in Figure 3.10.
The forces used for dimension the bearings are the same that acts on the planet shaft. Calculation and magnitude of the forces can be seen in Eq. 3.13, 3.14 and Table 3.4.

Three needle bearings were chosen for each planet shaft. The selection of the bearings where based on the load capability and size availability. The width of the three bearings together are $3mm$ smaller than the width of the planet gear. Therefore a small protruding bearing seat has been made to keep the bearings from moving sideways along the planet shaft and is placed in such way that one bearing covers the bigger gear and two the smaller one since more force is acting on the smaller gear. The selected needle bearings is **SKF K 17x21x17** and more information about this bearing can be found in Appendix E.2. The basic life time of the bearings was calculated according to the equations in section 2.10 where the equivalent dynamic bearing load P_{load} is **4.129kN** , which is based on 20% of max torque and is divided between the three bearings. The resulting life time is presented in Table 3.5.

Table 3.5: Planet needle bearing life result for 3 bearings

Basic life	L_{10h}	25639 hours
SKF life	L_{nmh}	33331 hours

The life time for the 3 bearings can be seen from the results in Table 3.5 and is much more than the expected vehicle life and it would be enough to use only 2 bearings as results in Table 3.6 show but it was decided to go with 3 bearings as there was no drawbacks with that solution.

Table 3.6: Planet needle bearing life result for 2 bearings

Basic life	L_{10h}	6636 hours
SKF life	L_{nmh}	8627 hours

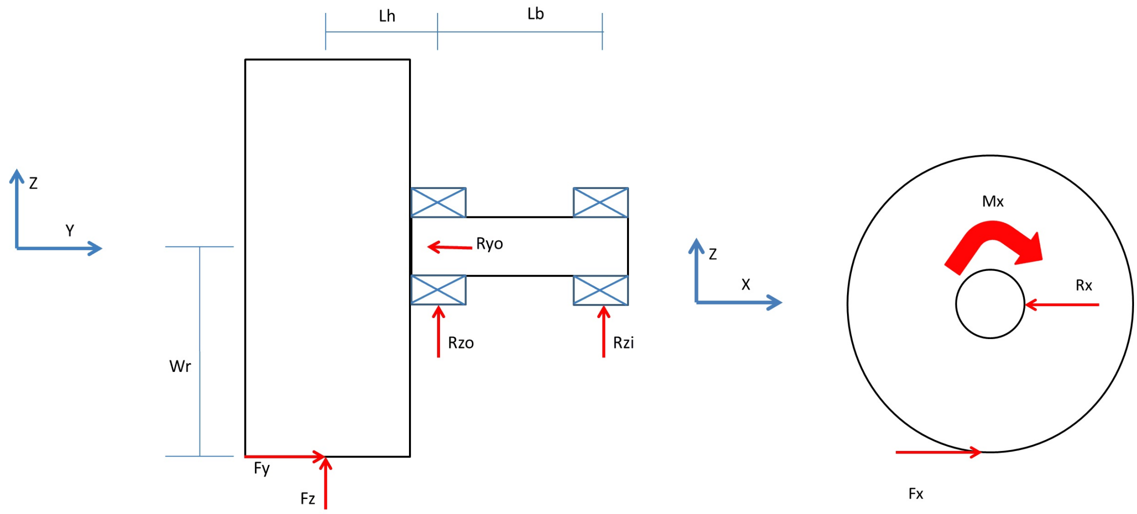
For the planetary gear it is necessary to have bearings both on the input shaft and output shaft. To select the right bearings, the lateral, normal and longitudinal loads acting on the tires has to be defined. To calculate maximum normal force a estimated maximum vertical acceleration of $3g$ is used. The acceleration is multiplied with the maximum axle mass m_a divided by two to get the mass on each tire.

$$F_z = 3g\frac{m_a}{2} \approx 30kN \tag{3.16}$$

For the maximum lateral force, a maximum lateral acceleration of $1g$ is used:

$$F_y = 1g\frac{m_a}{2} \approx 10kN \tag{3.17}$$

These values are based on information from Arctic Trucks. For the longitudinal force a 5000 N is used which corresponds to around 80% of the available traction force at 60 km/h. This values are only assumptions as there are no log data that can be used to derive accurate forces but should give some idea about the bearing life, which can vary a lot from car to car based on there loads, driving cycles and many other parameters. How these tire forces affect the bearing forces is explained with the help of the free body diagrams in Figure 3.15.



(a) Free body diagram in YZ plane

(b) Free body diagram in XZ plane

Figure 3.15: Free body diagrams for bearing calculation

The bearing reaction forces are calculated using momentum and force equilibrium equations and can be seen in Table 3.7. The axial load is assumed to only act on the outer bearings.

Table 3.7: Reaction forces acting on the wheel bearings

Radial bearing force for inner bearing	R_{zi}	3741 N
Radial bearing force for outer bearing	R_{zo}	34240 N
Axial force on outer bearing	R_{yo}	9810 N

The original hub and wheel-bearing from the Toyota Hilux is used as bearing for the output shaft. There are no load data available for the OEM wheel bearing but the size and type have been compared to a assumed similar SKF-bearing: **30210**

J2/QDF double tapered roller bearing. The bearing data has been used to confirm if the wheel bearing could be used also with the AT44 tire and the Hub reduction gear mounted and can be seen in Appendix E.2. Due to the hub reduction the OEM bearing is placed closer to the wheels center compered to a Hilux without the hub reduction on the new 44" tires. This means that the bearing suffer less load since the distance from the center of the tire to the center of the OEM bearing is less and therefore shorter leverage arm. From this result it is therefore assumed that the bearing life is at least not worse than for a vehicle without hub reduction.

The input shaft bearing is placed between the sun gear and the casing and to avoid that the bearing slides out of its seat a bearing backing plate is mounted to the casing(see Figure 3.16). The bearing is placed in such way that the inner surface is located exactly where the inner surface of the OEM bearing would be in the OEM upright.This is done to have the drive axle mounted in the same position as for the OEM upright. The Input shaft bearing is exposed to much lower loads and can be much smaller then the output shaft bearing. The bearing selected is **SKF 61913-2RZ** which is a deep grove ball bearing and more information's can be seen in Appendix E.2.

(a) Exploded view of inner wheel bearing **(b)** Assembled inner wheel bearing

Figure 3.16: Inner bearing mounting in casing

The bearing life for the inner bearing is presented in Table 3.8

Table 3.8: Inner wheel bearing life time

Basic life	L_{10h}	2553 hours
SKF life	L_{nmh}	10215 hours

As the SKF life rating is more accurate it can be assumed that the inner bearing will last the lifetime of the car.

3.6 Lubrication

The simulation done in KISSsoft has been run with an oil with viscosity ISO VG 220 (AGMA 5). Oil with viscosity VG 220 is common to use in automotive gearbox's today. When the first prototype is made it is necessary to make accurate tests on what oil that works best for this application. Important to find out the maximum and minimum temperature that will occur in the hub gearbox so a oil that can handle that temperature can be selected. The planet shaft has been designed with several holes so that the planet bearings can be lubricated by the oil. Through the large holes in each end of the shaft, oil will flow in when the shaft moves through the oil bath in the bottom of the gearbox and then due to centrifugal forces, the oil is forced through the holes to the needle bearings(see Figure 3.17 and 3.18).

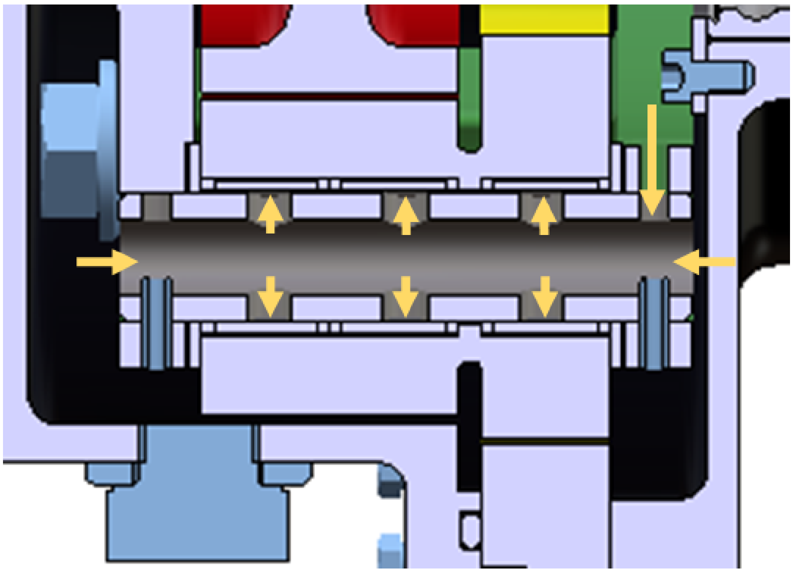


Figure 3.17: Oil lubrication of planet bearings

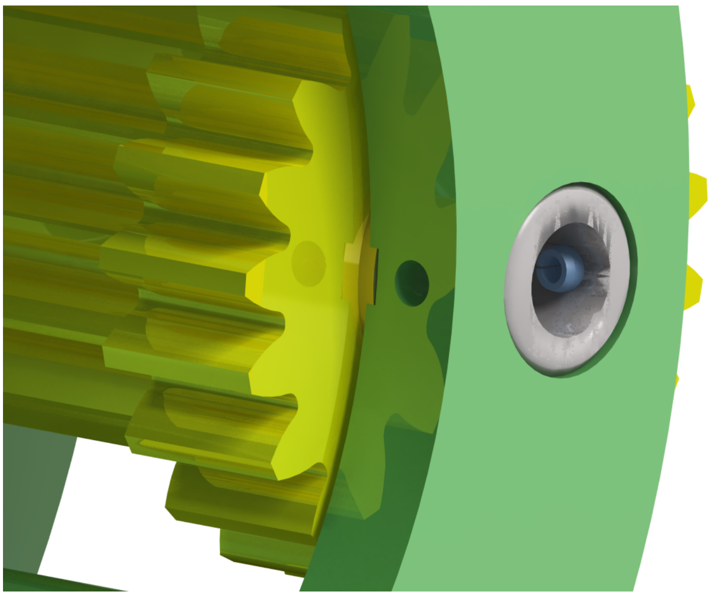


Figure 3.18: Inlet hole for oil lubrication of planet bearings

3.7 Casing for Planetary hub Gear

This section will focus on the installation of the planetary gear in the upright and general packaging.

3.7.1 Inner casing/upright

The Inner casing is designed in such a way that it is also the upright. It's design is based on the OEM upright and all OEM hard points are the same so the influence on the vehicle suspension should be minimum. First the casing around the ring gear was designed and then the upright part was designed using the OEM upright as a template. The idea was to have the design in such a way that no modification would be needed on the control arms/ball joints, steering tie rods or drive axle. The dimensioning factors where ring diameter, hard points and axle position. The new inner casing/upright design can be seen in Figure 3.19 and the OEM upright is presented in Figure 3.20 for comparison.

(a) Back side of the inner casing **(b)** front side of the inner casing

Figure 3.19: Inner casing/upright

Figure 3.20: 3D scanned OEM upright

3.7.2 Outer casing

The outer casing can be thought of as the lid on the casing. Its dimensioning factors are the ring, carrier, brake caliper and OEM wheel bearing. Two types of seals are installed in the outer casing as will be described in more detail in section 3.7.5. The outer casing has also the brake bracket connection points, the oil filler and drain hole and the abs sensor mounting so there are many things that needs to be taken into consideration to make everything fit. The design of the outer casing can be seen in Figure 3.21.

(a) Back side of the outer casing **(b)** front side of the Outer casing

Figure 3.21: Outer casing

3.7.3 Casing material

The material EN-GJS-600-3(European standard EN 1563) or GGG 60 (Germany standard DIN 1693) has been chosen for the planetary gear casing[17] but other materials with similar properties could also be used. It is a ductile cast iron which is commonly used for uprights/knuckles. It has good machinability and wear resistant. The stresses are not as high on the casing as on the gears and the planet shafts. Therefore a lower grade steel/iron can be chosen for the casing that has a lower yield and tensile strength, but also a lot cheaper. The material data for EN-GJS-600-3 is presented in Table 3.9.

Table 3.9: Material data for the planetary hub gear casing[17] .

ρ	Density	7200 $[Kg/m^3]$
σ_y	Yeild strength	370 $[Mpa]$
σ_B	Tensile strength	600 $[Mpa]$
BH	Surface hardness	$180 - 270$
E	Young's modulus	177 $[Gpa]$
v	Poisson's ratio	0.3

3.7.4 Brake caliper mounting

As mentioned before in section 1.6 it is desired to maintain the OEM brake caliper. The OEM brake caliper is mounted with two axial facing bolts as can be seen in Figure 3.22.

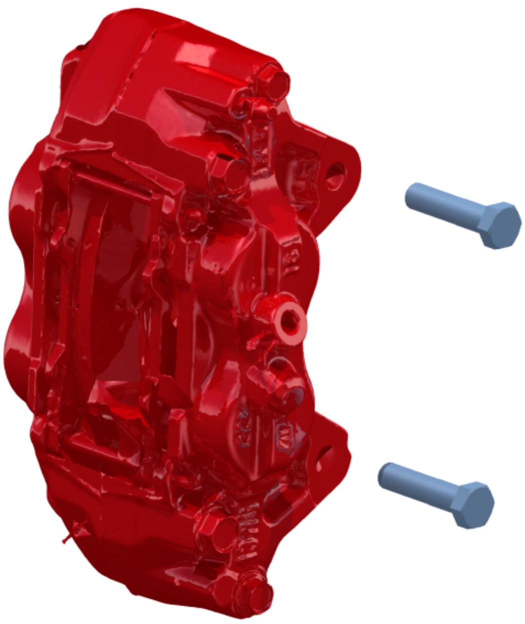
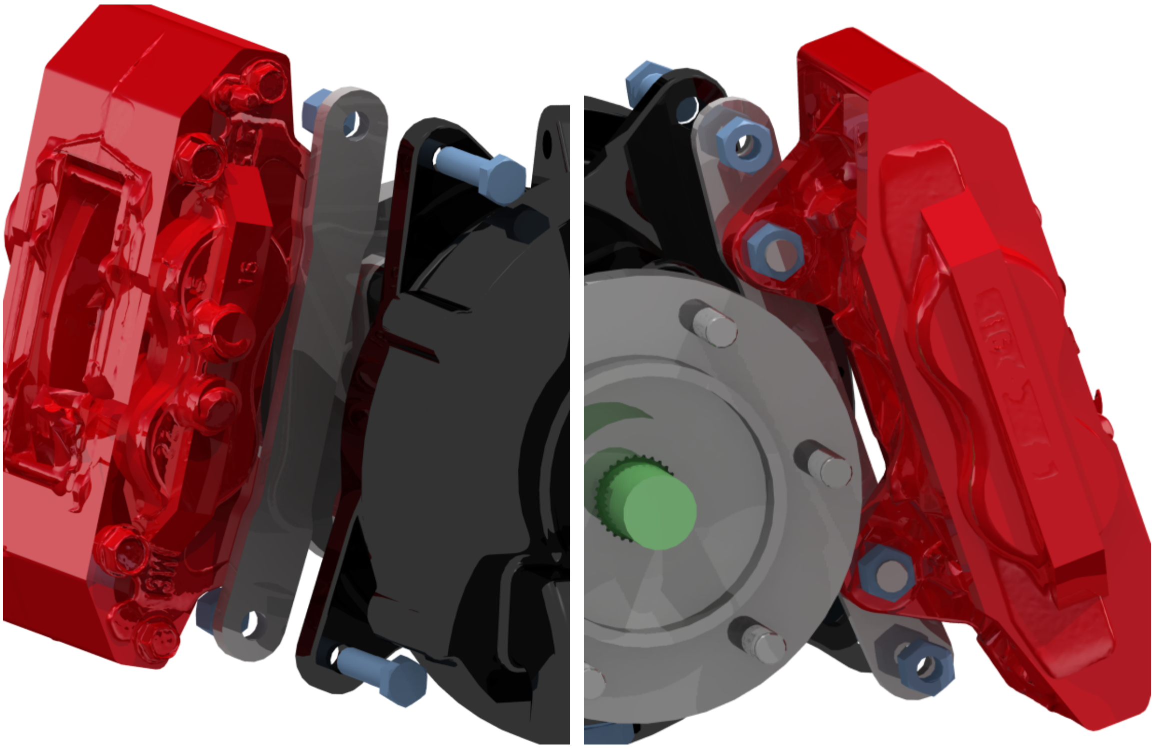


Figure 3.22: 3D scanned OEM caliper

Because the casing is much wider than the OEM upright it is not possible to mount the caliper directly to the casing since the bolts needs to have sufficient space behind them. It was found out that it is not possible to mount the OEM caliper if the gear ring is located near the output and still maintain good packaging as mentioned in section 3.2. With the ring moved back it is enough space to make a special brake caliper bracket which can be bolted to the casing since its mounting holes can be moved to the outside of the casings diameter. The bracket is bolted to the caliper first and then to the casing(see Figure 3.23).



(a) **(b)**

Figure 3.23: Brake caliper bracket mounting

The brake system is one of the most important systems of the vehicle and if any alterations are made it is important to make sure that it can still operate in same way as the OEM setup. Therefore it is vital that the design is verified by proper simulations and physical tests before it is implemented in sold vehicles. A preliminary stress simulation has been done fore the brake caliper bracket and can be seen in Appendix D. A 18000N force has been applied to each of the mounting studs. The force have been calculated with Eq. 3.18. In the simulation a support have been added to simulate the support from the caliper to get a shear stress rather then bending on the studs. A Chrome-Molly steel could be used for the bracket as it usually has high yield stress.

$$F_{disk} = \frac{m \cdot a \cdot R_{wheel,44}}{R_{disk} \cdot 4 \cdot 2} \tag{3.18}$$

m is the curb weight of the car divided by four to get an approximate load on each tire, then by two to get the load on each mounting stud, a is the maximum deceleration assumed to be $7.7 m/s^2$, $R_{wheel,44}$ is the tire radius for the 44inch tires and R_{disk} is the radius from the wheel center to the brake caliper mounting studs and is 0.104 meters.
Another solution would be to installing new brake calipers that can be mounted radially to the casing and therefore no bracket is needed. An example of such a brake caliper can be seen in Figure 3.24.

Figure 3.24: Brake caliper with radial mounted bolts[53]

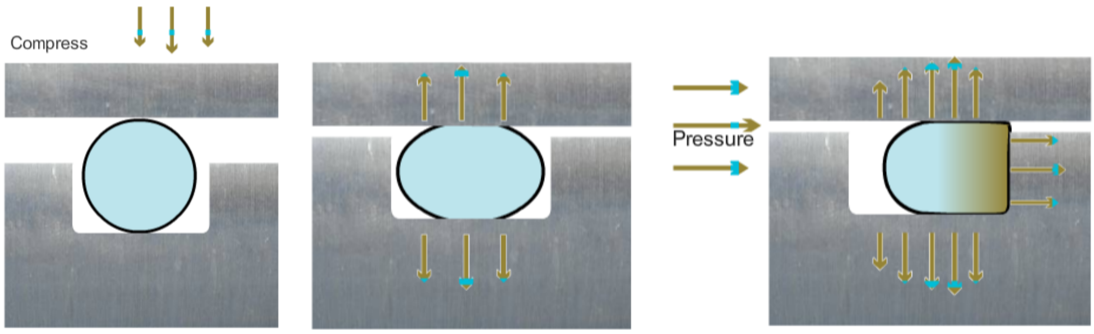
3.7.5 Seals

It is vital that the casing is sealed properly at all time and therefore it is important that the chosen seal can withstand appropriate temperatures and chemicals. Two seals have been chosen for the casings presented in section 3.7.1 and 3.7.2. To seal the two casings a o-ring type seal has been chosen (see Figure 3.25a) with the part number **ORAR00268-N7237** and for the output axle(carrier) a rotary oil seal is chosen (see Figure 3.25b) with the part number **TRE000700-N7MM**. Detailed information for this two seals can be found in Appendix E.1

(a) O-ring seal **(b)** Rotary oil seal

Figure 3.25: Seals chosen for the casing[34]

The O-ring is used as axial seal which means that the sealing surfaces is at the top and the bottom of the O-ring and is commonly used for flanges and cover plates. How the axial O-ring works can be seen in Figure 3.26 .The initial compression will result in the initial sealing force and allows the elastomeric O-ring to adapt to the two mating surfaces. The sealing of the O-ring increases when it is under system pressure which results in the total sealing force[35].

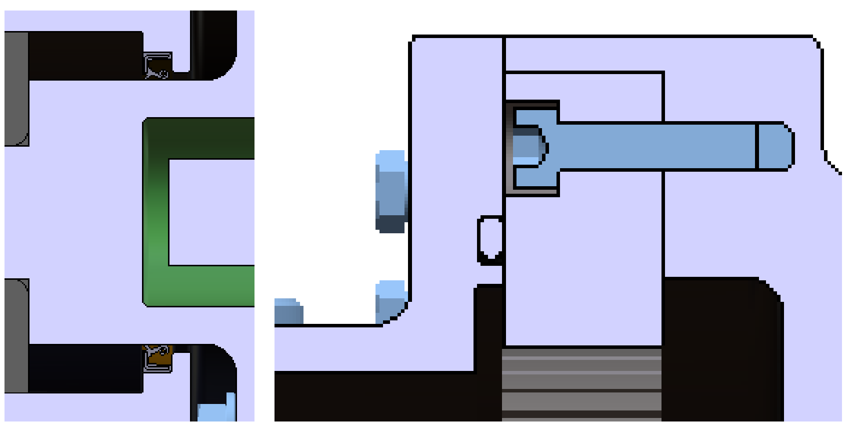


(a) Applying initial pressure **(b)** Initial sealing force **(c)** Total sealing force due to system pressure

Figure 3.26: Sealing function of axial placed O-ring [35]

The primary purpose of the rotary oil seal is to make it possible to change wheel bearing without oil leaking out and have the ABS sensor area free of oil. Both seals are made out of NBR material or Nitrile Rubber which is popular material for seals in the automotive industry since it has excellent oil resistance and has big operating temperature range from $-40°C$ to $120°C$ and can work in even lower temperature if some additives are used[33].

The seals installation in the casings can be seen in Figure 3.27. The rotary seal is placed in special seal seat in the outer half of the casing and the O-ring is placed right above the ring teeth in the outer casing flange.



(a) **(b)**

Figure 3.27: Installation of seals for the gearbox

3.7.6 ABS sensor

On the Toyota Hilux 2016 model the wheel speed sensor is mounted on the upright and is measuring the turn of the wheel bearing. With the planetary hub reduction unit, the wheel speed sensor has to be mounted on the hub gearbox casing, to be able to access the wheel bearing. In Figure 3.28 a example of a wheel speed sensor that would fit this concept can be seen. The wheel speed sensor on the 2016 Toyota Hilux will unfortunately not fit with the design that has been chosen. The electric contact on the sensor is angled in such away that it would clash with the brake disk or the hub gear casing itself. If the original wheel speed sensor is desired, the part of the casing where the sensor is in Figure 3.28 has to be extended. When extended, the lever arm from where the wheels is attached to the hard points will be longer and a increased moment will act on the upright.

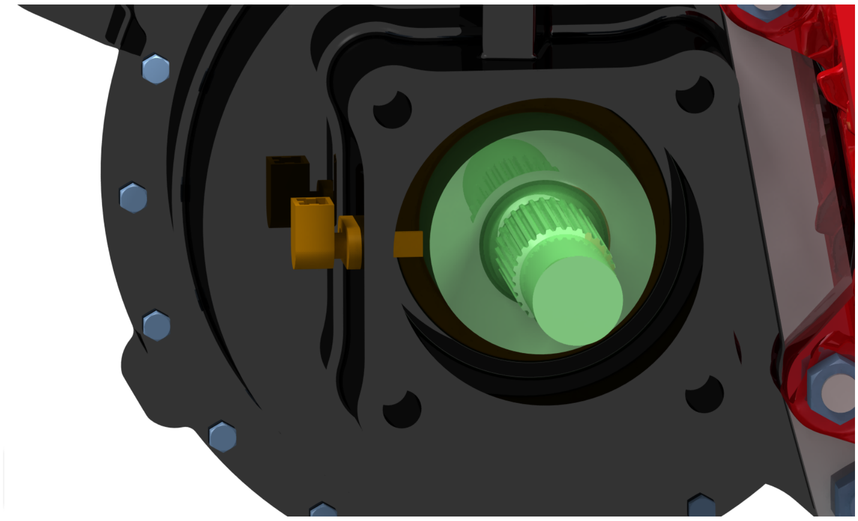
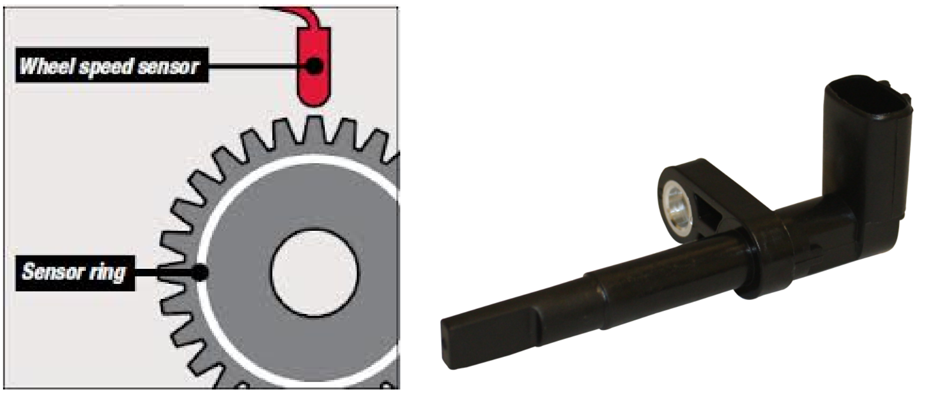


Figure 3.28: Position of wheel speed sensor

Two alternative solutions have been considered to solve this situation. The first one is to use the wheel speed sensor that was used in the earlier generation. That sensor has radial sensing where the wheel speed is recorded by using a wheel speed ring that is mounted on the output shaft(see Figure 3.29a). Or using a ABS sensor originally intended for other Toyota/Lexus vehicles. It is hard to find a suitable solution since it is needed to go manually through a lot of different kind of sensors. One possible sensor has been found from a Lexus is250 and has the part number **8954630070**. This sensor is longer than the original sensor, but the connection is oriented upward and could therefore fit properly. Further investigation would be needed to get the sensors exact dimensions and specifications(see Figure 3.29b).



(a) Radial sensing sensor and ABS ring [47]

(b) Lexus ABS sensor that might be used[40]

Figure 3.29: ABS solutions

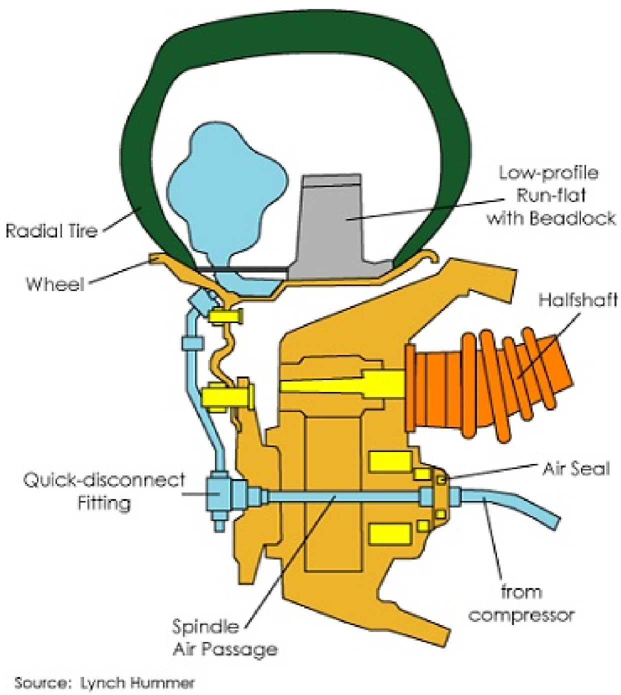
3.7.7 Central Tire Inflation System (CTIS)

CTIS is a system to control the air pressure in each tire. The driver has access to control the pressure, while he/she is driving, to improve the vehicles performance on different surfaces. If the the ground are very soft like when driving on snow, mud or sand the contact patch between the tire and ground will enlarge if air is let out from the tire which will lower the overall ground pressure. The tire will then not sink down as much and give the vehicle better traction(see Figure 3.30)[32]. When the vehicle is on tarmac or hard ground the pressure can be increased again to give better efficiency.

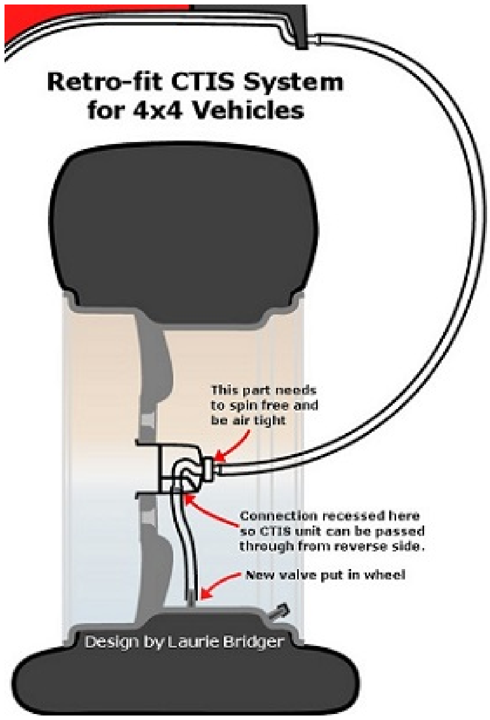
Figure 3.30: CTIS in action on Unimog [44]

The CTIS has been around since WW2 and was initially developed for military wheeled tactical vehicles to enhance there overall mobility[32]. Probably the most famous military vehicle that uses CTIS is the HUMMER. The design of the HUMMERS CTIS is rather simple due to the portal gear which has parallel input and output axles and therefore makes it possible to make air flow through the hub to the tire as can been seen on Figure 3.31a.

It is necessary that the CTIS comes out in the middle of the rim,otherwise it will tangle itself up when the wheel is rotating. For vehicles that don't have portal gear it is almost impossible to have CTIS that comes out trough the hub center so the only way is to have hose that lies outside of the tire and is connected to air tight mechanism that can rotate and is placed either on the hub or special made bracket that is attached on the rim (see Figure 3.31).



(a) CTIS on a vehicle with portal axle [48].



(b) Inflation system that connects from outside the wheel [52].

Figure 3.31: Common CTIS solutions

There are some other designs out there but these two are most common. The method Arctic Trucks have used for the vehicles they have equip with CTIS is the later one since it is the simplest solution to have the air hoes running outside of the tire. The only problem with this solution is that it is open to the environment and the hose could get caught in something and get damaged but that depends on the environment where the vehicle is driven(Forrest vs. Glacier).

Although the planetary gearbox has separated input and output axles like the portal gear they are inline which makes it challenging to implement CTIS where the air comes out of the hub center. The only solution we thought might be possible is illustrated in Figure 3.32. Two pressure seals are placed in such way that they form a high pressure section. The output shafts is partially hollow in the middle and has two small holes drilled radial to the shaft so the pressurized air can flow trough the casing into the pressure section and through the output shaft to the tire. Based on the Trelleborg sealing solutions on-line catalogue there are some radial oil seals that can handle up to 0.5MPa(5 bars) but that is dependent on the rotational speed, so a physical test is needed to evaluate the sealing and analysis of the hollow output shaft to see if it is still strong enough to withstand the load.

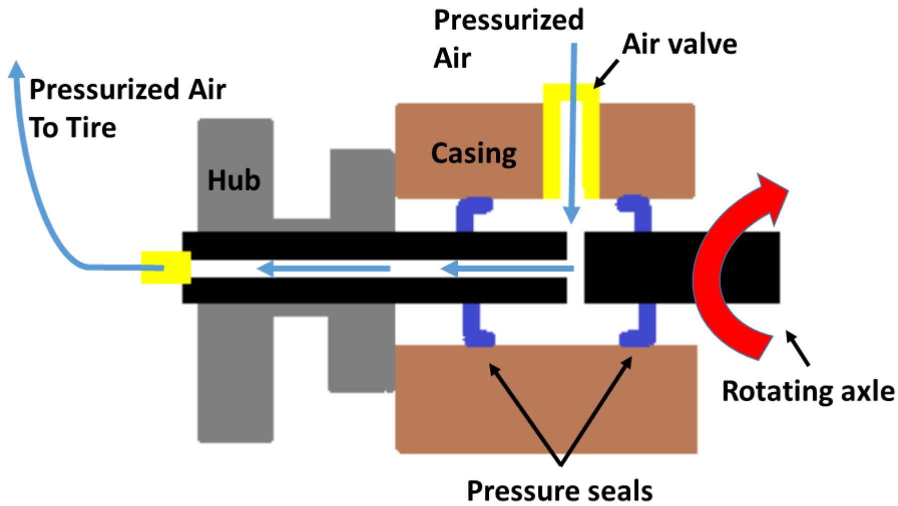


Figure 3.32: Possible solution for CTIS on planetary hub reduction gearbox.

4

Results

In this chapter the results of the following concepts are presented:

- Spur compound planetary gear set
- Helical compound planetary gear set
- Portal hub gear set

Spur compound planetary gear set is the selected gear concept and the design around it will be presented thoroughly in this chapter.

4.1 Final gear concepts

4.1.1 Spur compound planetary gear set

When all desired constraints mentioned in section 3.1.1 had been implemented in the Matlab script and all the stresses calculated it was only three gear combinations that were left. By gear combination it is meant gear teeth combinations and therefore there was more then three solutions since each gear combination could have different face widths for stage 1 and stage 2. The three smallest and lightest combination were picked. The major differences in these three combinations are illustrated in Table 4.1

Table 4.1: Possible solutions from the Matlab script

	Solution 1	Solution 2	Solution 3
Ratio	2.185:1	2.1333:1	2.1034:1
Ring pitch diameter	200 mm	212.5 mm	225 mm
Face with planet 1	34 mm	35mm	34 mm
Face with planet 2	17 mm	17mm	14 mm
Weight	6.0087 kg	7.0025	7.1616 kg
Number of planets	5	5	5

The main focus when deciding on the final solution was the ring diameter and the weight. When it comes to packaging every mm counts and bigger ring diameter also means more weight since the casing needs to be bigger. Solution 1 was therefore selected due to its low estimated weight and ring diameter. This solution has the highest ratio of those three solutions but it was considered that the ratio difference would not have much difference in the operation of the vehicle. Detailed results for solution 1 are presented in Tables 4.2 and 4.3. Detailed results from KISSsoft for solution 1 can be seen in Appendix H. The outlay of solution 1 can be seen in Figure 4.1

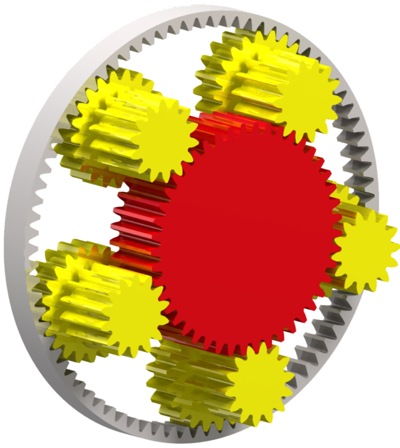


Figure 4.1: Size and outlay of Spur compound planetary gear set

Geometrical data for spur compound planetary gear concept:

Table 4.2: Geometrical values for final solution for spur compound planetary gear set

ϕ_n	Normal pressure angle	20 [°]
m_n	Module	2.5 [mm]
x_{stage1}	Profile shift coefficient stage 1	-0.4/+0.4
x_{stage2}	Profile shift coefficient stage 2	-0.5/+0.5
$n_{planets}$	Number of planets	5
z_s	Number of teeth, sun	45
z_{p1}	Number of teeth, planet stage 1	14
z_{p2}	Number of teeth, planet stage2	21
z_r	Number of teeth, ring	80
b_{stage1}	Face width stage 1	34 [mm]
b_{stage2}	Face width stage 2	17 [mm]
d_1	Pitch diameter sun	112.5 [mm]
d_2	Pitch diameter planet 1	35 [mm]
d_3	Pitch diameter planet 2	52.5 [mm]
d_4	Pitch diameter ring	200 [mm]
m_{tot_est}	Estimated total gear mass from matlab	6 [kg]
m_{4_est}	Estimated mass of Sun gear from matab	2.42 [kg]
m_G	Total reduction ratio	2.19:1

Gear stress's for spur compound planetary gear concept based on Matlab calculations:

Table 4.3: Stress values for final solution for spur compound planetary gear set

$\sigma_{s,stage1}$	Contact stress between sun and planet1	1123 [Mpa]
$S_{S,stage1}$	Safety factor stage 1	1.25
$\sigma_{s,stage2}$	Contact stress between ring and planet 2	811[Mpa]
$S_{S,stage2}$	Safety factor stage 2	1.6
$\sigma_{F,sun}$	bending stress sun (dynamic/static)	180/624[Mpa]
$S_{F,sun}$	Safety factor sun (dynamic/static)	2.3/1.3
$\sigma_{F,planet1}$	Bending stress planet 1 (dynamic/static)	134/466 [Mpa]
$S_{F,planet1}$	Safety factor planet 1 (dynamic/static)	2.3/1.4
$\sigma_{F,planet2}$	bending stress planet 2 (dynamic/static)	187/644 [Mpa]
$S_{F,planet2}$	Safety factor planet 2 (dynamic/static)	1.6/1.2
$\sigma_{F,ring}$	Bending stress ring (dynamic)	194 [Mpa]
$S_{F,ring}$	Safety factor ring (dynamic)	2.2

4.1.2 Helical compound planetary gear set

The same method as for spur gears was used to find possible solutions for helical gears and the final solution was picked in the same way. If using helical gears instead of spur gear the contact ratio on the teeth's gets higher due to that the teeth are angled. This gives a lower tangential force but a higher radial force than for spur gear. The helical gears could therefore be made smaller than for the spur gear, which gives a lighter solution. But the increased radial forces on the helical gears needs to be considered when selecting bearings and developing the gear casing. The helical gears are also more complex to manufacture and therefore often more expensive. In Figure 4.2 the gear layout of the concept for the chosen Helical compound planetary gear set can be seen. In Table 4.4 are the geometrical data presented and in Table 4.5 are the stress values for the gears presented.

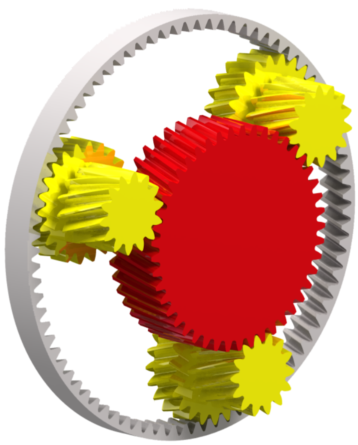


Figure 4.2: Size and outlay of Helical compound planetary gear set

Geometrical data for helical concept:

Table 4.4: Geometrical values for final solution for helical compound planetary gear set

ψ	Helix angle	15 [°]
ϕ_n	Normal pressure angle	20 [°]
m_n	Module	2.5 [mm]
x_{stage1}	Profile shift coefficient stage 1	-0.5/+0.5
x_{stage2}	Profile shift coefficient stage 2	-0.5/+0.5
$n_{planets}$	Number of planets	3
z_1	Number of teeth, sun	45
z_2	Number of teeth, stage 1	14
z_3	Number of teeth, stage 2	22
z_4	Number of teeth, ring	81
b_{stage1}	Face width stage 1	34 [mm]
b_{stage2}	Face width stage 2	18 [mm]
d_1	Pitch diameter sun	112.5 [mm]
d_2	Pitch diameter planet 1	35 [mm]
d_3	Pitch diameter planet 2	55 [mm]
d_4	Pitch diameter ring	202.5 [mm]
m_{tot_est}	Estimated total gear mass from matlab	5.42 [kg]
m_{4_est}	Estimated mass of sun gear from matlab	2.42 [kg]
m_G	Total reduction ratio	2.15:1

Gear stress's for helical concept:

Table 4.5: Stress values for final solution for helical compound planetary gear set

$\sigma_{s,stage1}$	Contact stress between sun and planet1	936 [Mpa]
$S_{S,stage1}$	Safety factor stage 1	1.5
$\sigma_{s,stage2}$	Contact stress between ring and planet 2	738 [Mpa]
$S_{S,stage2}$	Safety factor stage 2	1.9
$\sigma_{F,sun}$	bending stress sun (dynamic/static)	187/612 [Mpa]
$S_{F,sun}$	Safety factor sun (dynamic/static)	2.3/1.35
$\sigma_{F,planet1}$	Bending stress planet 1 (dynamic/static)	133/434 [Mpa]
$S_{F,planet1}$	Safety factor planet 1 (dynamic/static)	2.3/1.9
$\sigma_{F,planet2}$	bending stress planet 2 (dynamic/static)	183/623 [Mpa]
$S_{F,planet2}$	Safety factor planet 2 (dynamic/static)	1.7/1.3
$\sigma_{F,ring}$	Bending stress ring (dynamic)	152 [Mpa]
$S_{F,ring}$	Safety factor ring (dynamic)	2.8

4.1.3 Spur Portal hub gear set

The Portal hub gear concept has also been investigated. As have been mentioned earlier the portal concept is both larger and heavier then a compound planetary gear. When comparing the portal hub gear with the helical and spur compound gear with similar gear ratios of around 2.15 and similar face widths, the portal weighs roughly 2.5 times as much as the helical- and spur compound gear sets and takes up more space. In Figure 4.3 the gear layout of the concept for the chosen Spur portal hub gear set can be seen. In Table 4.6 are the geometrical data presented and in Table 4.7 are the stress values for the gears presented.

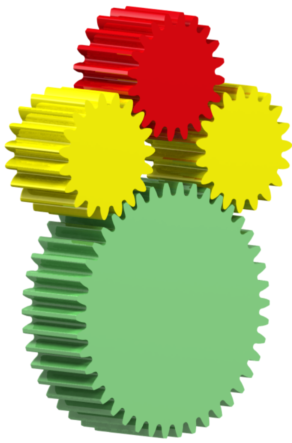


Figure 4.3: Size and outlay of Spur portal gear set

Geometrical data for portal concept:

Table 4.6: Geometrical values for final solution for spur portal gear set

ϕ_n	Normal pressure angle	20 [°]
m_n	Module	4 [mm]
x_1	Profile shift coefficient pinion/idle gear	-0.1/+0.1
x_2	Profile shift coefficient gear/idle gear	-0.1/+0.1
$n_{idlegear}$	Number of idle gears	2
z_p	Number of teeth, pinion	20
z_g	Number of teeth, gear	42
z_{id}	Number of teeth, idler gear	19
b	Face width	56 [mm]
d_p	Pitch diameter pinion	80 [mm]
d_g	Pitch diameter gear	168 [mm]
d_{id}	Pitch diameter idler gear	76 [mm]
m_{tot_est}	Estimated total gear mass from matlab	14.9 [kg]
m_{gear_est}	Estimated gear mass from matlab	9.1 [kg]
m_G	Total reduction ratio	2.1:1

Gear stress's for portal concept:

Table 4.7: Stress values for final solution for spur portal gear set

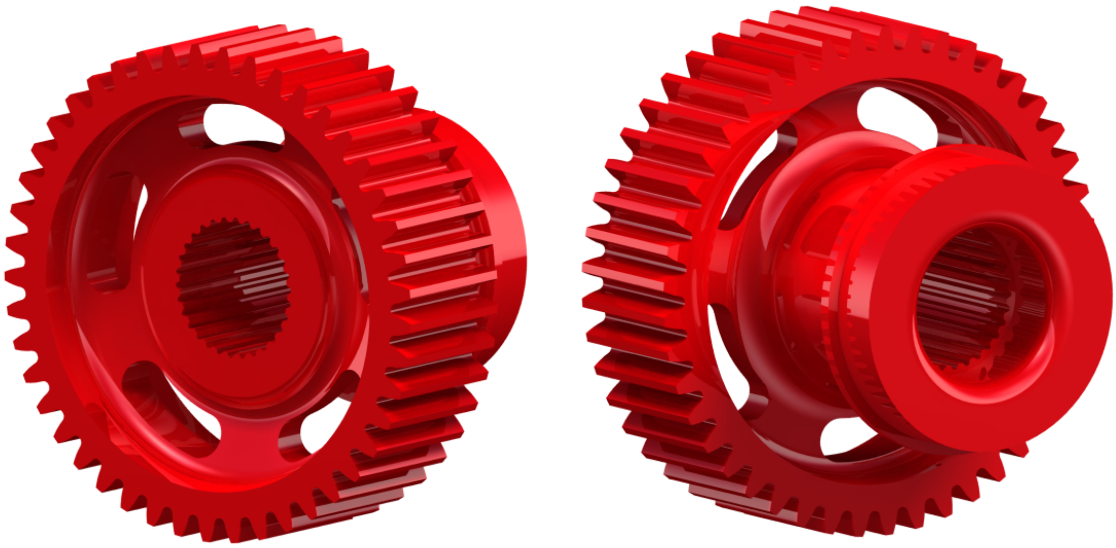
$\sigma_{S,p-id}$	Contact stress between pinion and idle gear	1227 [Mpa]
$S_{S,p-id}$	Safety factor between pinion and idle gear	1.2
$\sigma_{S,id-g}$	Contact stress between idle gear and gear	1074 [Mpa]
$S_{S,id-g}$	Safety factor between idle gear and gear	1.4
$\sigma_{F,pinion}$	bending stress sun (dynamic/static)	192/675[Mpa]
$S_{F,pinion}$	Safety factor pinion (dynamic/static)	2.3/1.2
$\sigma_{F,gear}$	Bending stress gear (dynamic/static)	166/584 [Mpa]
$S_{F,gear}$	Safety factor gear (dynamic/static)	3/1.5
$\sigma_{F,idlegear}$	bending stress idle gear (dynamic/static)	184/653 [Mpa]
$S_{F,idlegear}$	Safety factor idle gear (dynamic/static)	1.8/1.4

4.2 Final CAD design of the spur compound planetary gearbox

After having decided on using the spur planetary gear solution a CAD design of all components needed for installation and prototype manufacture was done. This section will present the final design for the planetary gear set as well as the casing and parts related to the installation at the front.

4.2.1 Sun gear

Figure 4.4 shows the final design of the sun gear. The center has been hollowed to remove redundant material and save weight. A hole with illustrative splines that should match the drive shaft splines has been created. A bearing seat has been designed in the end of the suns axle where the Inner bearing will sit between the sun and the planetary hub gear box. It was necessary to extend the sun outwards since the ring was moved which makes it slightly heavier. A stress analysis has been made and can be seen in Appendix D. Manufacturing drawing of the sun can be found in Appendix C

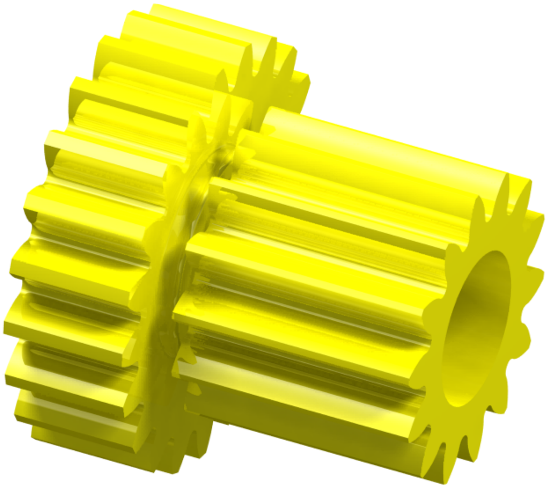


(a) Sun gear front view **(b)** Sun gear rear view

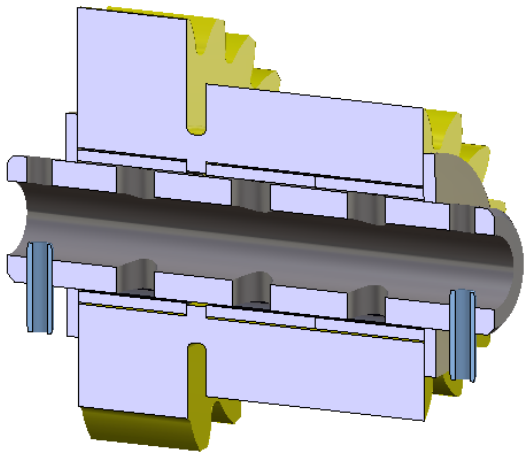
Figure 4.4: Final design of the sun gear

4.2.2 Planet gear and shaft

Figure 4.5 illustrate the final design of the planet gear. Each of the five planets have got a hole with 20mm in diameter that can fit the 17mm planet shaft and three needle bearings in between.The space between the to planet gears is 3mm and that is necessary so the gear cutting tool can cut the entire face width of the planet. The shaft has got five holes in it. The two holes in each end of the shaft has two functions: two let in oil for lubrication of the needle bearings and to fit a spring pin to prevent the shaft from rotating (see Figure 4.5b). On each side of the planet a washer is placed to take up possible wear that other wise would act on the planet carrier, the washer can then be changed if necessary. In the theory there should not be any forces acting along the shaft when using spur gears, but there will always be some small forces in that direction also. There is a 0.4mm space between the washer and the planet carrier to give the planets a chance to rotate smoothly and proper lubrication between the washer and planet.The stress analyses of the planet and shaft can be found in Appendix D and manufacturing drawing for the planet is presented in Appendix C.



(a) Final design of the planet gear



(b) Section cut of the planet gear

Figure 4.5: Final design of the planet gear

The exploded view of the planet assembly can be seen in Figure 4.6

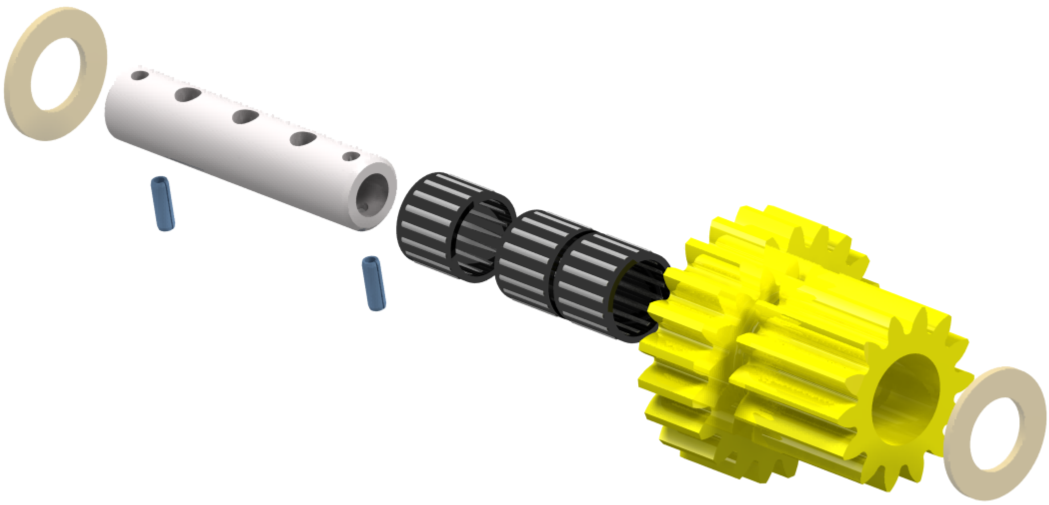


Figure 4.6: Planet gear assembly

4.2.3 Ring gear

From KISSsoft, only the tooth profile was imported to CATIA. To make the ring strong enough the rings rim thickness had to be increased. The ring gear thickness has been chosen to be 1.2 times the tooth height based on the AGMA standard, Figure 2.15. The rim thickness is defined as the distance between the tooth root diameter to the outer diameter. The ring has been fitted with six ears where six bolts can be inserted and fasten the ring to the gear housing. Figure 4.7 shows the ring with its six attachment points. Manufacturing drawing for the ring can be found in Appendix C. The Planetary gear assembly can be seen in Figure 4.8.

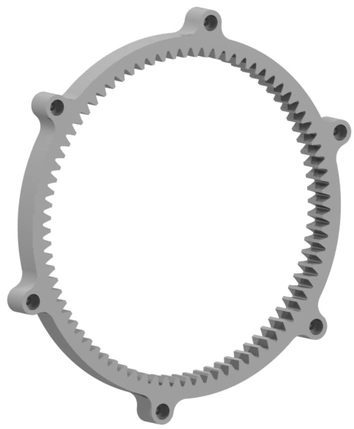


Figure 4.7: Final design of the ring gear

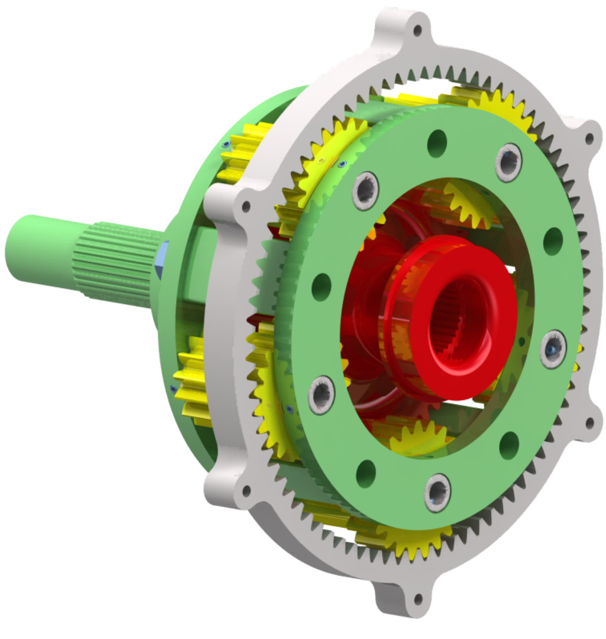


Figure 4.8: Gear assembly

4.2.4 Casing and installation

The final design of the casing can be seen in Figure 4.9 and 4.10. A extra supporting rod has been placed between the inner and outer casing to strengthen the casing. This additional support was implemented after stress analysis which is presented in Appendix D. The original wheel bearing has been used and is mounted to the outer casing and the carrier output shaft comes instead of the axle. The two casings are mounted together with 18 M5 bolts and the supporting rod. Oil filer plug is placed on top and drain in the bottom. The casing is mounted to the car using the same mounting points as the original upright and therefore no need of suspension changes. The hub reduction gearbox with brake disk and caliper is presented in Figure 4.11. The packaging inside the wheel rim can be seen in Figure 4.12 and 4.13.

Figure 4.9: The final design of the front hub reduction

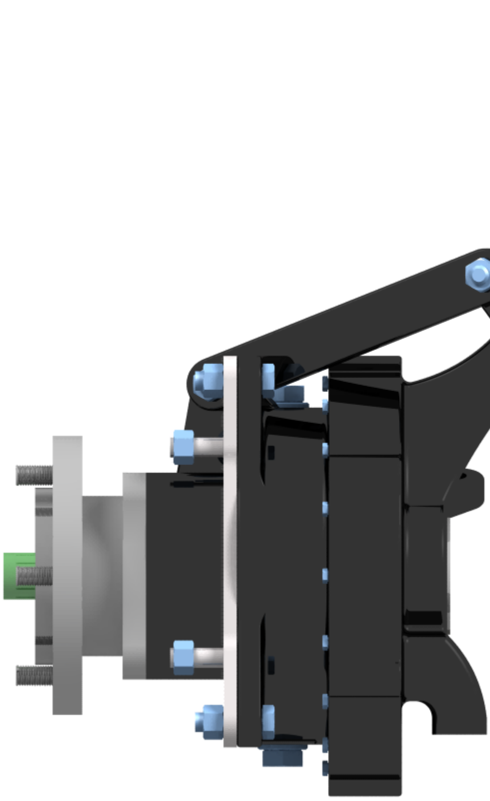
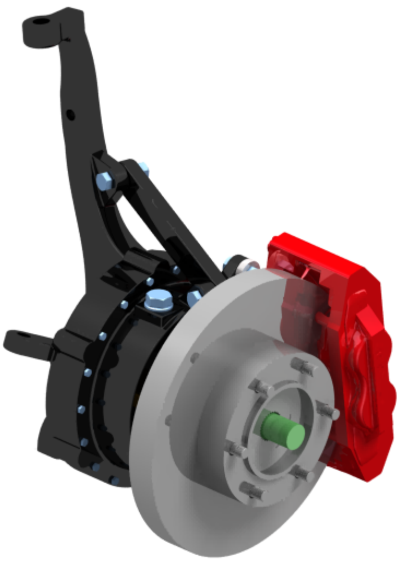
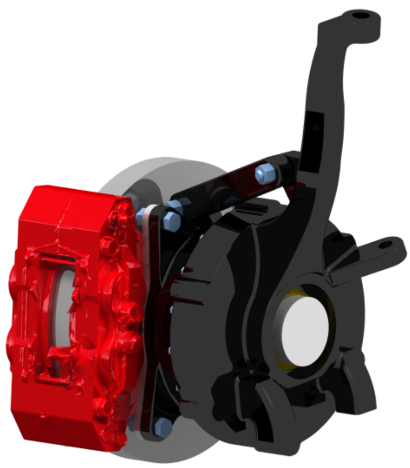


Figure 4.10: Complete design of planetary hub gear reduction



(a) Planetary hub reduction front view



(b) Planetary hub reduction rear view

Figure 4.11: Planetary hub reduction gearbox with brake disk and brake caliper

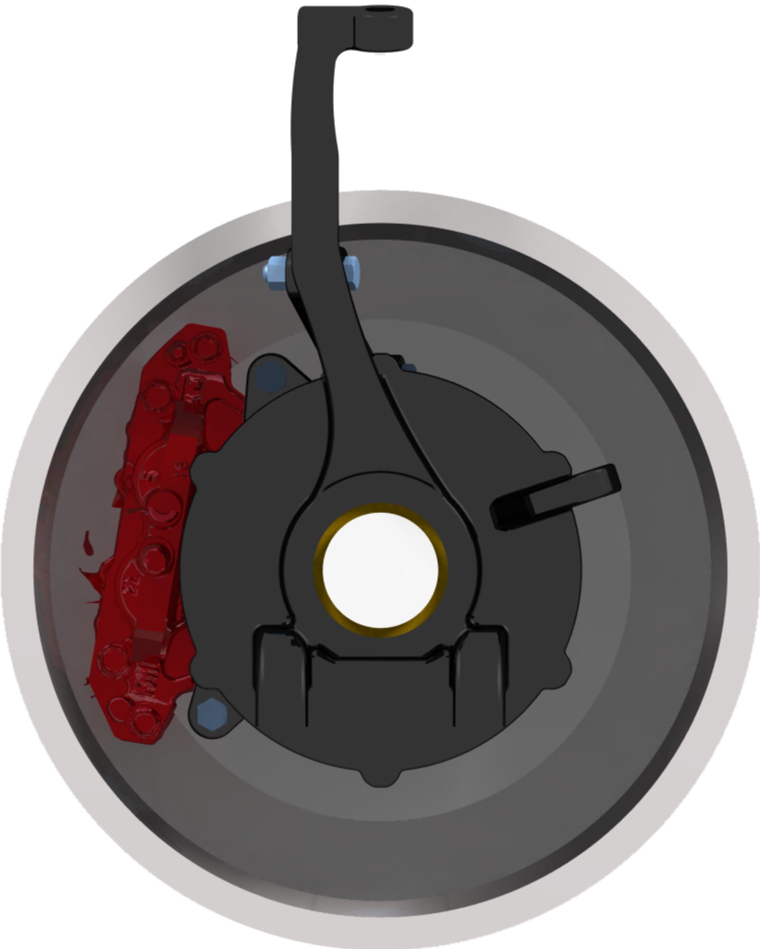


Figure 4.12: Planetary hub reduction, inside wheel rim

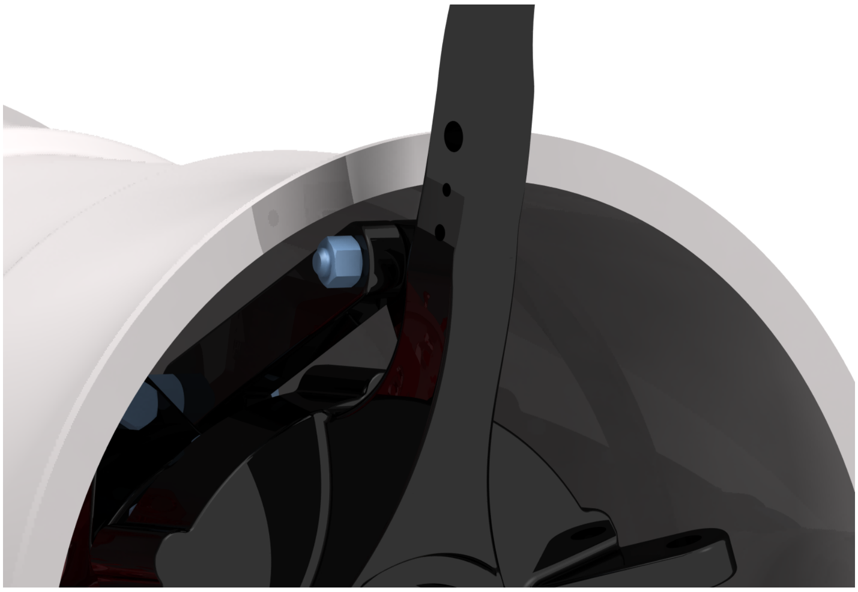


Figure 4.13: Support rod clearance

Figure 4.14 shows a section view of the hub reduction gearbox where all major components can be seen and there position. The hub has been moved **146** *mm* outward due to the hub reduction which means that the ET offset of the rims needs to change from **-115** to **+31**.

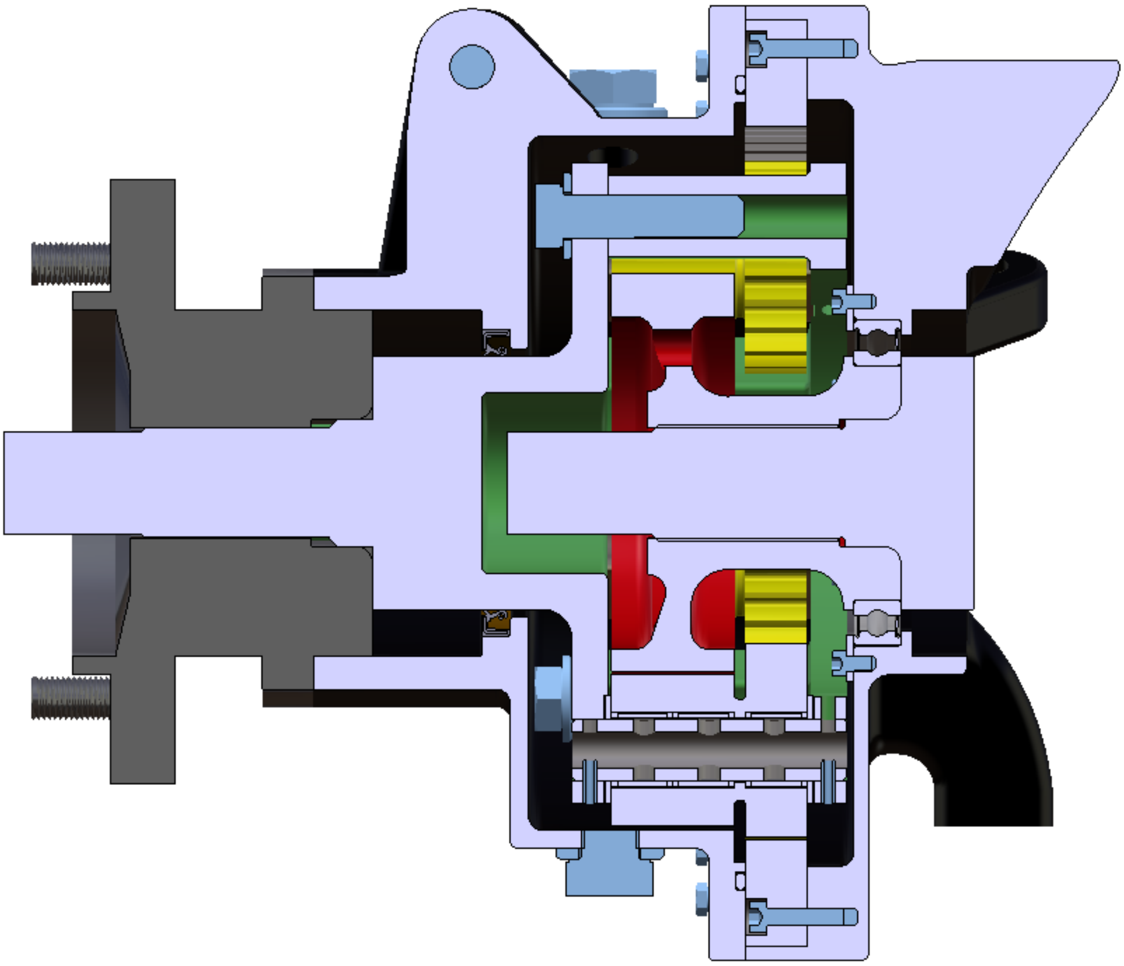


Figure 4.14: Section view of the hub reduction gearbox

4.3 CAD Design weight

The final CAD weight can be seen in Table 4.8 and the weight comparison between the original upright can be seen in Table 4.9.

Table 4.8: Final CAD weight of designed parts.

Part	**Weight**
Sun gear	1.951 kg
Planet gear	0.454 kg
Ring gear	1.12 kg
Planet shaft	0.082 kg
Planet carrier	5.346 kg
Inner casing/upright	8.309 kg
Outer casing	6.556 kg
Brake caliper bracket	0.435 kg
Bracing	0.838 kg
Total weight	25.091 kg

Table 4.9: Total added weight

Original upright	4.894 kg
Total weight minus the original upright	20.197 kg

The total weight of the hub reduction gear box is calculated by summarizing the parts mentioned in Table 4.9. By subtracting the total weight with the weight of the original upright a total added weight of 20.197 kg is calculated. This is the weight that has been added to the car when one hub reduction gear is mounted.

4.4 Cost

The Swedish gear manufacturer Eibers Edeby AB has been contacted to get a price estimation for the gears [54]. Eibers have provided a complete cost analysis for the gears where material cost, gear cutting, milling, lathing, hardening and grinding are included. In Table 4.10 the setup cost and the unit price for each gear are presented in Swedish Kronor[SEK].

Table 4.10: Setup cost and unit price for a Spur compound planetary gear set with 5 planet gears.

Gear	**Price** [SEK]
Setup cost Sun gear	10400
Unit price Sun gear	780
Setup cost Planet gear	11800
Unit price Planet gear	750
Setup cost Ring gear	7900
Unit price Ring gear	1100

Total price for ordering one spur compound planetary gear set with five planets is 35730 SEK. In Table 4.11 are some different Batch prices presented. These prices are only valid for Batch's up to 100 or for 25 cars, according to Eibers.

Table 4.11: Cost for a Spur compound planetary gear set with 5 planet gears.

Batch	**Price** [SEK]
1 set	35730
1 car	52620
5 cars	142700
10 cars	255300
25 cars	593100

A Icelandic casting company called Málmsteypa Þorgríms was contacted for rough price estimation for the inner and outer casing. The price was **100000 ISK** for a set of inner and outer casing and it is assumed that Arctic Trucks provides the plug and that the production quantity is dozens of units per year. There is no fine machining included in this price estimation. It is important to note that this prize is a very rough estimation since it is only based on conversation, CAD pictures, material and rough dimensions, but should give some idea of the order of magnitude.

4.5 Assembly

This section will focus on the assembly process and order of the hub reduction parts.

- The inner bearing is pressed in the inner casing.
- The bearing retainer plate it bolted to the inner casing.
- The ring gear is bolted in place.
- Planet shafts are inserted in the inner part of the carrier and locked in place with the inner spring pins.
- The planets, needle bearings and washers are slided on to the planet shaft.
- The inner part of the carrier with the planets is inserted in the casing.
- The sun is pressed in to the inner bearing.
- The outer part of the carrier is fixed to the planet shafts.
- The two carrier parts are bolted together and outer spring pins are placed in the planet shafts.
- The outer seal is placed in the outer casing.
- The O-ring is placed in the outer casing which is then bolted to the inner casing.
- The original wheel bearing is mounted to the outer casing.

A exploded view of the hub reduction can be seen in appendix A.

5 Conclusion

A Spur compound planetary gear set have been designed with help of CAE-tools that meet the requirements that has been set. All gears and the planet carrier have been 3D-printed in scale 1:1. Pictures of the 3D-printed model can be seen in Appendix B. The hub gear set, is designed to be mounted directly on to the suspension hard points. The only OEM parts that has to be changed to be able to use this design are the upright, that in the this design are merged together with the inner part of the gearbox casing, the ABS-sensor also have to be changed because of packaging. All other parts like the hub, wheel bearing, brake disk, brake caliper and A-arm can be kept as it is.
The final design has not been designed with a CTIS (Central Tire Inflation System) solution that comes out through the centre of the wheel because it was found to be to problematic and take to much time from the thesis work. If this is something that is desirable, further studies and testing has to be done to validate the possibilities for implementation of such concept. Otherwise the traditionally CTIS solution where the hose goes on the outside of the tire can still be used.

The hub reduction concept has been developed and considered for the front axle of the Toyota Hilux. If the same hub reduction concept should be mounted on the rear axle it is necessary that the original drum brakes is exchanged to disc brakes in order to make it possible to mount and fit the gearbox in a good way. The inner casing will have to be redesigned to make it possible to attach the hub gearbox on the rear axle but the outer one could most likely be the same or very similar.

As can be seen from table 4.8 it is the casing that weighs the most and therefore it might be room for some weight reduction. It might also be possible to make the casing out of Aluminum 7075 T6 which has higher yield strength and around 2.5 times lower density and therefore would make the weight of the casing drop by around 9 kg. The problem with the aluminum is that it is worse when it comes to fatigue and is more brittle. Aluminum 7075 T6 is also usually more expensive than steel or iron.

After contacting a gear manufacturer it was realized that it might not be possible to manufacture the gears according the accuracy grade used in the stress calculations, at least not for the planet gears. To get the a accuracy grade of 11(AGMA 2000) grinding of the gear is needed and that will result in a much higher cost. After discussions with the gear manufacturer it was decided that accuracy grade 9 would be the best choice since it does not require grinding and will still give good enough surface quality and therefore the stresses will not change that much.

Further investigation of what manufacturing method that should be used for gears and casing has to be performed to make sure a good result and to get the lowest production cost.

It is important to perform an extensive testing of the planetary hub gear box when a concept have been manufactured. First of all to make sure that the parts fit together as intended and make sure nothing is clashing. After that the Toyota Hilux should be equipped with the hub reduction gearbox and tested with the same load and velocities, in all environments that its intended to be used in.

Different gear oils need to be tested to validate that the oil will not be to hot or cold, when used in different environments like South pole or deserts. The oils viscosity changes with temperature and if the temperature differ much from its optimal temperature point, the lubrication can be to poor and damage the components. Components can also deform due to the high temperatures and start to wear each other out or break. There is a chance that the casing can expand due to too high temperatures and can lead to oil leakage. Sometimes pressure can be built up inside closed vessels due to temperature difference between the outside and the inside of the vessel. If this would be the case in the hub gearbox, it is recommended to have a pressure relief valve on the hub gearbox casing.
It is possible to use CAE-tools to simulate how the oil will move inside the gearbox. This can be used to see if their is enough oil and if the oil is reaching all parts inside that needs lubrication.

If the noise from the hub reduction should be deemed to be to high during testing the Helical compound planetary gear set concept could be considered and a casing could be designed around that concept instead. The size and weight of the helical planetary concept is very similar to the spur planetary concept and it would therefore not cause that much trouble to carry over the casing design to the helical gear concept. Planet shaft, bearings and sealings could be kept but planet carrier and the casing have to be redesigned.

To answer the questions presented in section 1.3 all pros and cons of the system needs to be considered. The pros are:

- Over 100% more output torque for a given engine torque
- The stresses on the power-train is decreased by more than half for a given desired traction force.
- Stronger upright
- All gears can be engaged for the usual driving cycle of the vehicle.
- Possibility of CTIS implementation

The cons are:

- Expensive
- Added weight
- Less efficient power-train
- More complex

From the pros and cons listed above it can be determined that using this hub reduction system, make a more capable off-road vehicle with stronger suspension and power-train where the cost and some added weight is not the major concern.

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Figures

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Email

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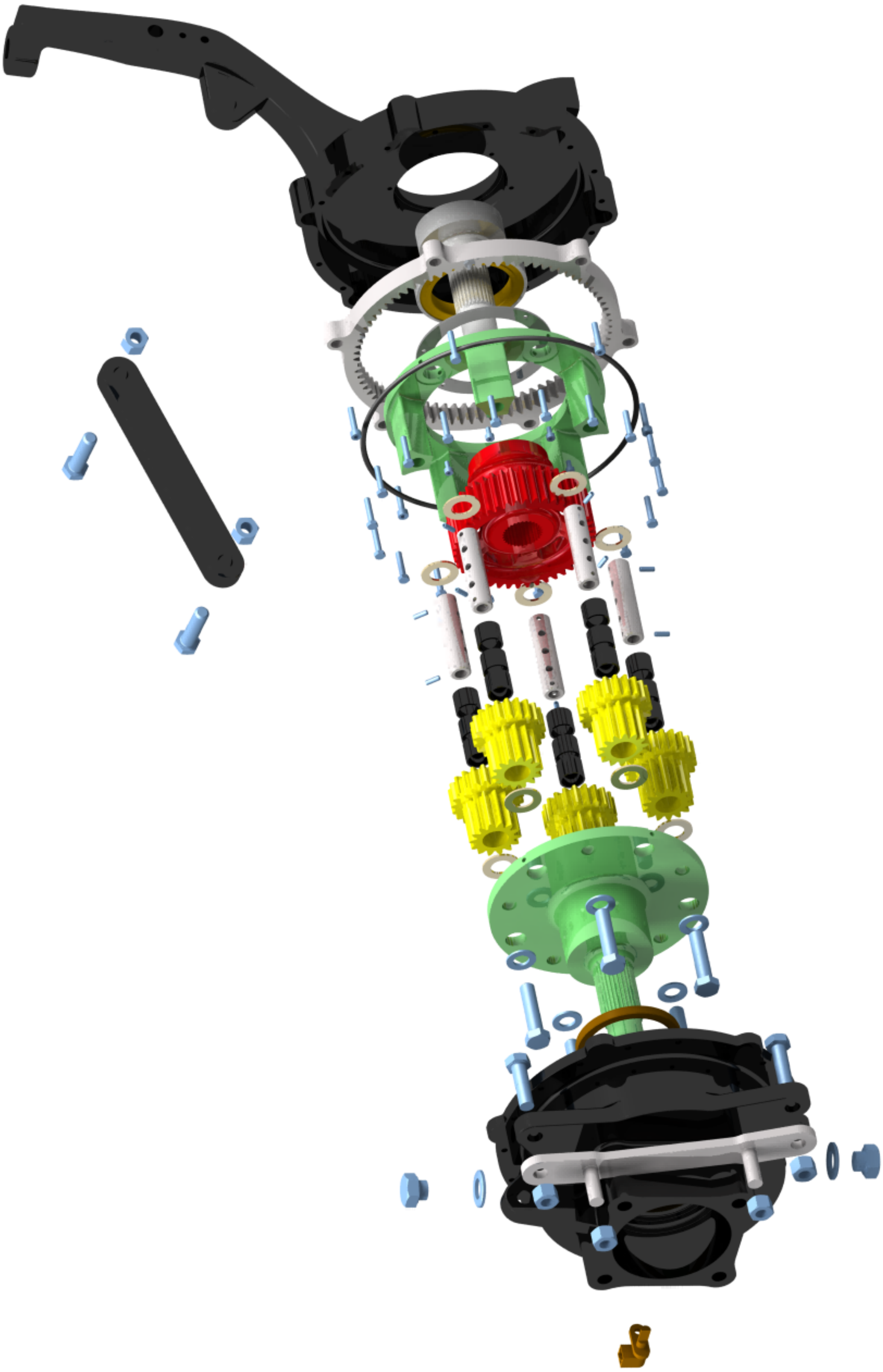
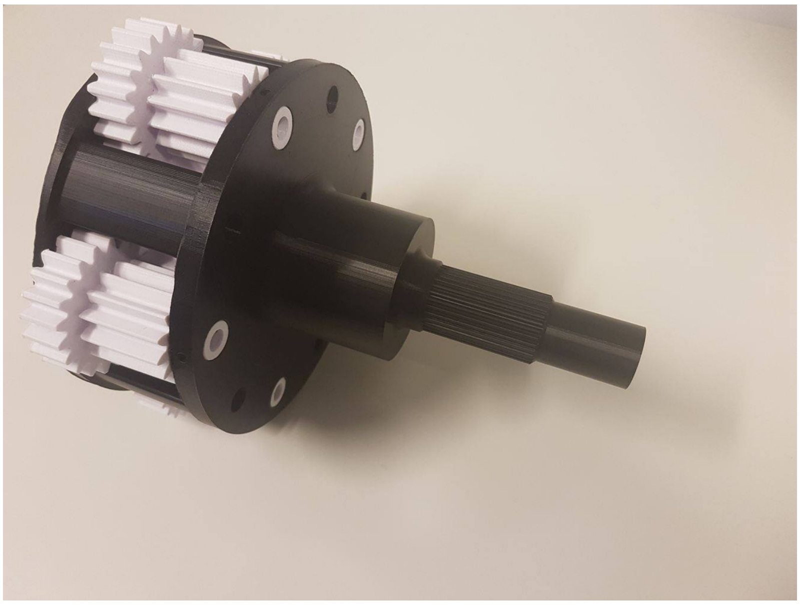
A

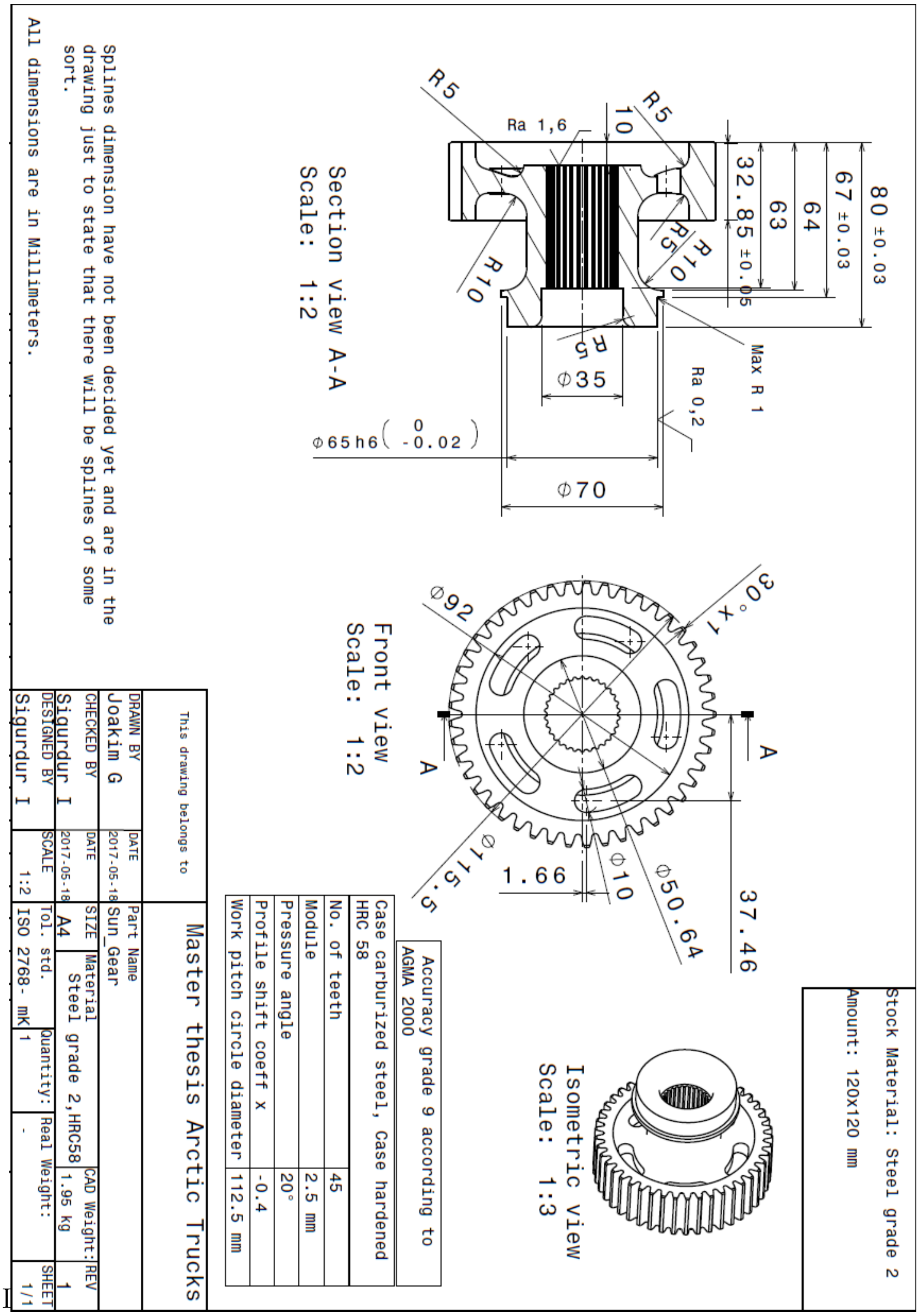
Exploded views

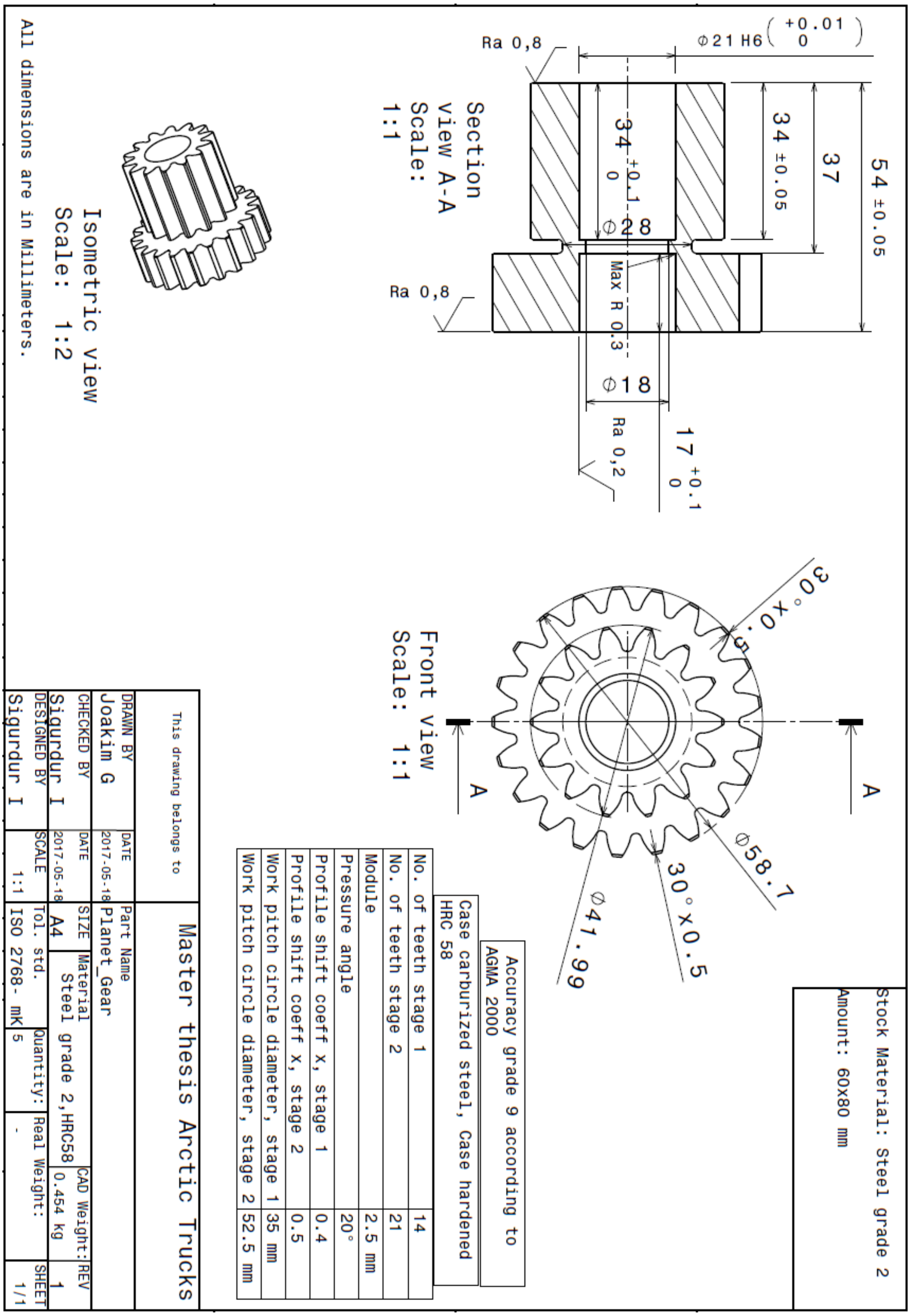


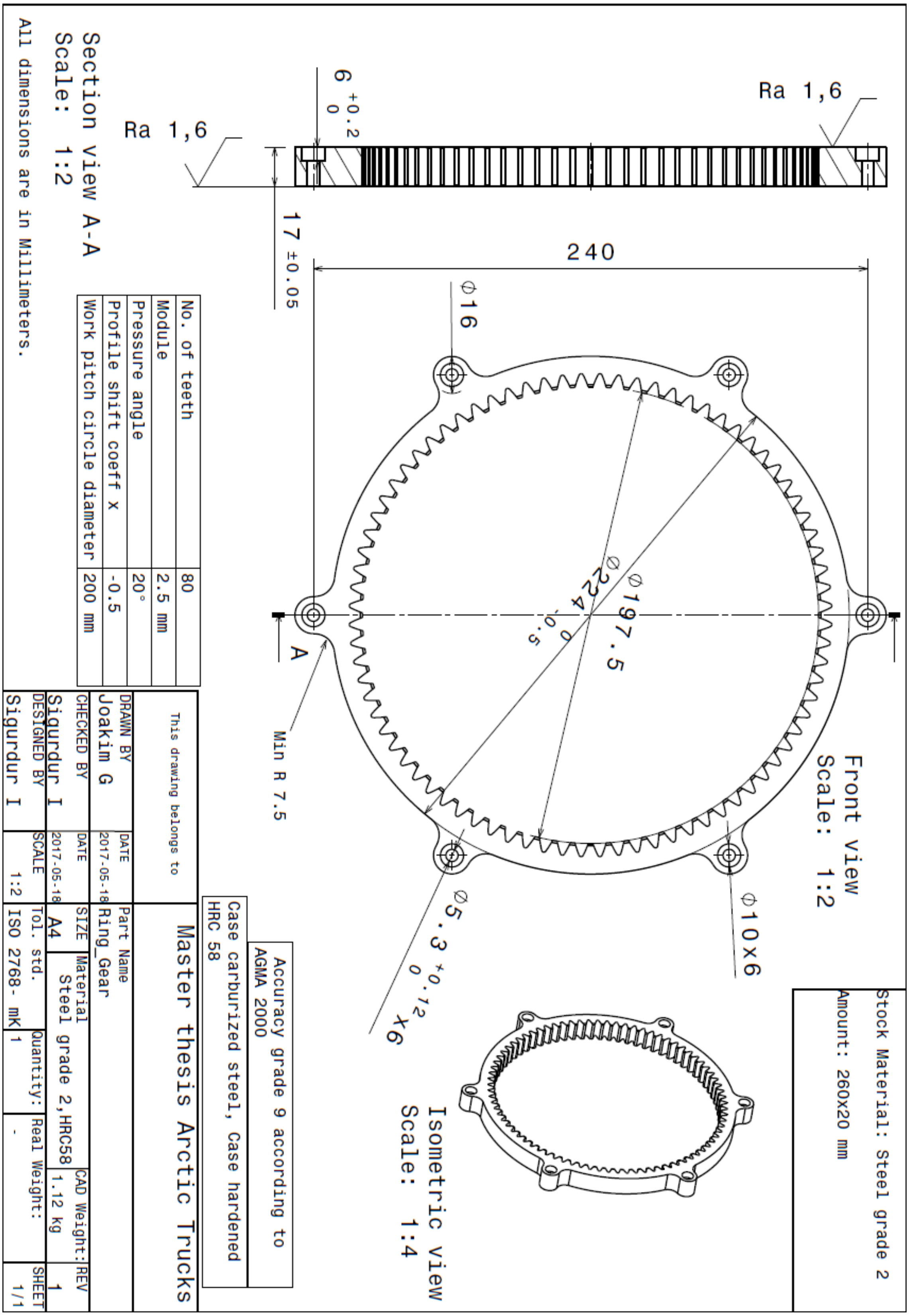
B
3D-printed Planetary gear model



C
Drawings


Section view A-A
Scale: 1:2
Front view
Scale: 1:2
Isometric view
Scale: 1:3
Splines dimension have not been decided yet and are in the drawing just to state that there will be splines of some sort.
All dimensions are in Millimeters.
Stock Material: Steel grade 2
Amount: 120x120 mm
Accuracy grade 9 according to AGMA 2000
Case carburized steel, Case hardened HRC 58
No. of teeth 45
Module 2.5 mm
Pressure angle 20°
Profile shift coeff x -0.4
Work pitch circle diameter 112.5 mm
This drawing belongs to
Master thesis Arctic Trucks
DRAWN BY Joakim G
DATE 2017-05-18
CHECKED BY Sigurdur I
DATE 2017-05-18
DESIGNED BY Sigurdur I
SCALE 1:2
Part Name Sun_Gear
SIZE A4
Material Steel grade 2,HRC58
CAD Weight: 1.95 kg
REV 1
Tol. std. ISO 2768- mK
Quantity: 1
Real Weight: -
SHEET 1/1


Stock Material: Steel grade 2
Amount: 60x80 mm
⌀21 H6 (+0.01 0)
Ra 0,8
54 ±0.05
37
34 ±0.05
34 +0.1 0
⌀28
Max R 0.3
Ra 0,8
⌀18
Ra 0,2
17 +0.1 0
Section view A-A
Scale: 1:1
30°x0.5
⌀58.7
30°x0.5
⌀41.99
A
A
Front view
Scale: 1:1
Isometric view
Scale: 1:2
All dimensions are in Millimeters.
Accuracy grade 9 according to AGMA 2000
Case carburized steel, Case hardened HRC 58
No. of teeth stage 1 | 14
No. of teeth stage 2 | 21
Module | 2.5 mm
Pressure angle | 20°
Profile shift coeff x, stage 1 | 0.4
Profile shift coeff x, stage 2 | 0.5
Work pitch circle diameter, stage 1 | 35 mm
Work pitch circle diameter, stage 2 | 52.5 mm
This drawing belongs to
Master thesis Arctic Trucks
DRAWN BY Joakim G
DATE 2017-05-18
Part Name Planet_Gear
CHECKED BY Sigurdur I
DATE 2017-05-18
SIZE A4
Material Steel grade 2,HRC58
CAD Weight: 0.454 kg
REV 1
DESIGNED BY Sigurdur I
SCALE 1:1
Tol. std. ISO 2768- mK
Quantity: 5
Real Weight: -
SHEET 1/1

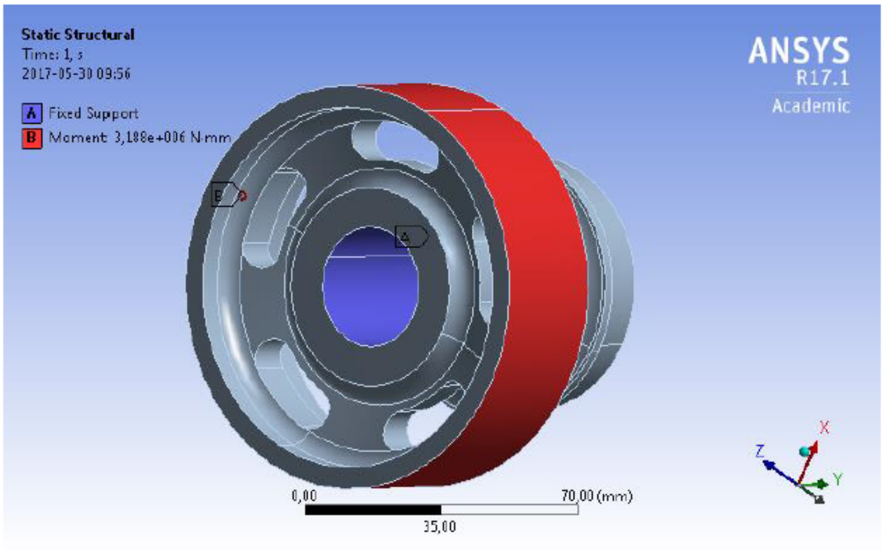

Section view A-A
Scale: 1:2
All dimensions are in Millimeters.
Ra 1,6
Ra 1,6
6 +0.2 0
17 ±0.05
240
Ø16
Ø197.5
Ø224 0 -0.5
A
Min R 7.5
Ø5.3 +0.12 0 x6
Ø10x6
Front view
Scale: 1:2
Isometric view
Scale: 1:4
Stock Material: Steel grade 2
Amount: 260x20 mm
No. of teeth 80
Module 2.5 mm
Pressure angle 20°
Profile shift coeff x -0.5
Work pitch circle diameter 200 mm
Accuracy grade 9 according to AGMA 2000
Case carburized steel, Case hardened HRC 58
This drawing belongs to
Master thesis Arctic Trucks
DRAWN BY Joakim G
DATE 2017-05-18
Part Name Ring_Gear
CHECKED BY Sigurdur I
DATE 2017-05-18
SIZE A4
Material Steel grade 2,HRC58
CAD Weight: 1.12 kg
REV 1
DESIGNED BY Sigurdur I
SCALE 1:2
Tol. std. ISO 2768- mK
Quantity: 1
Real Weight: -
SHEET 1/1

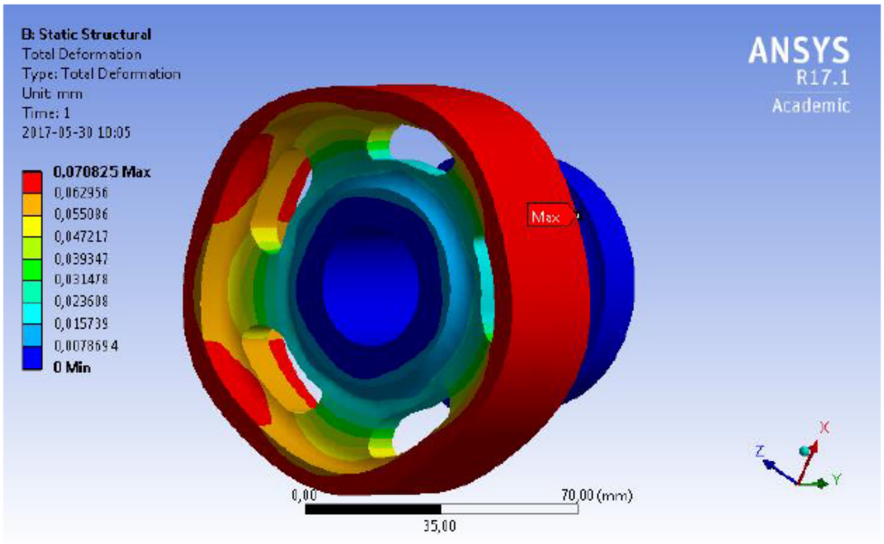
D

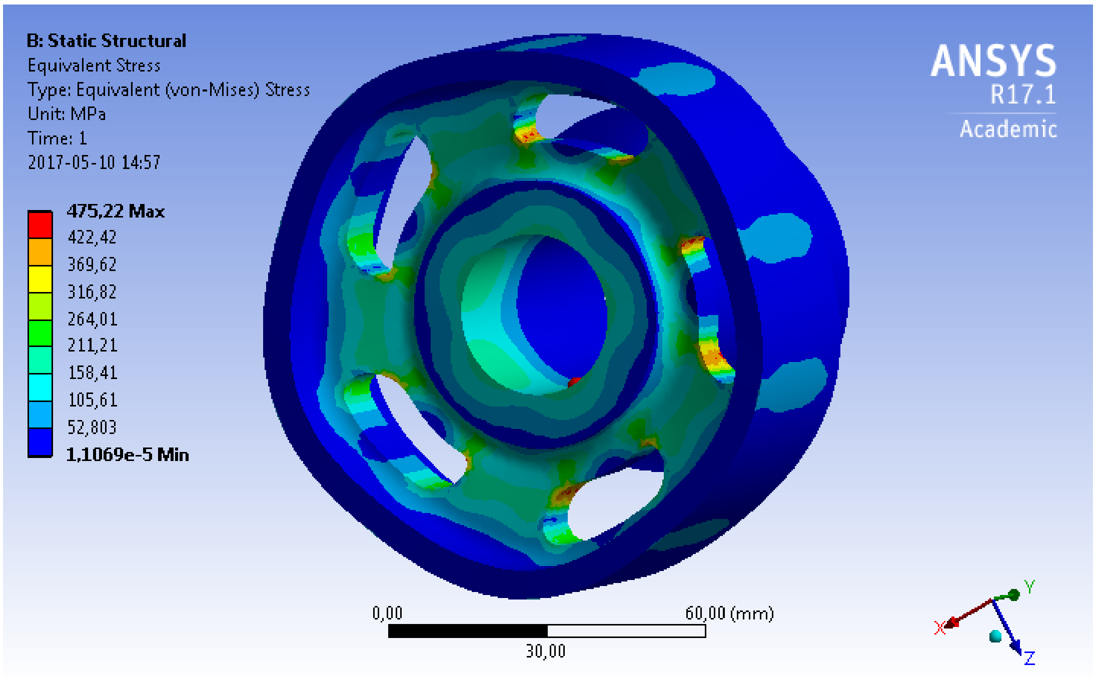
Stress analysis

Boundary conditions – Max moment acting on sun

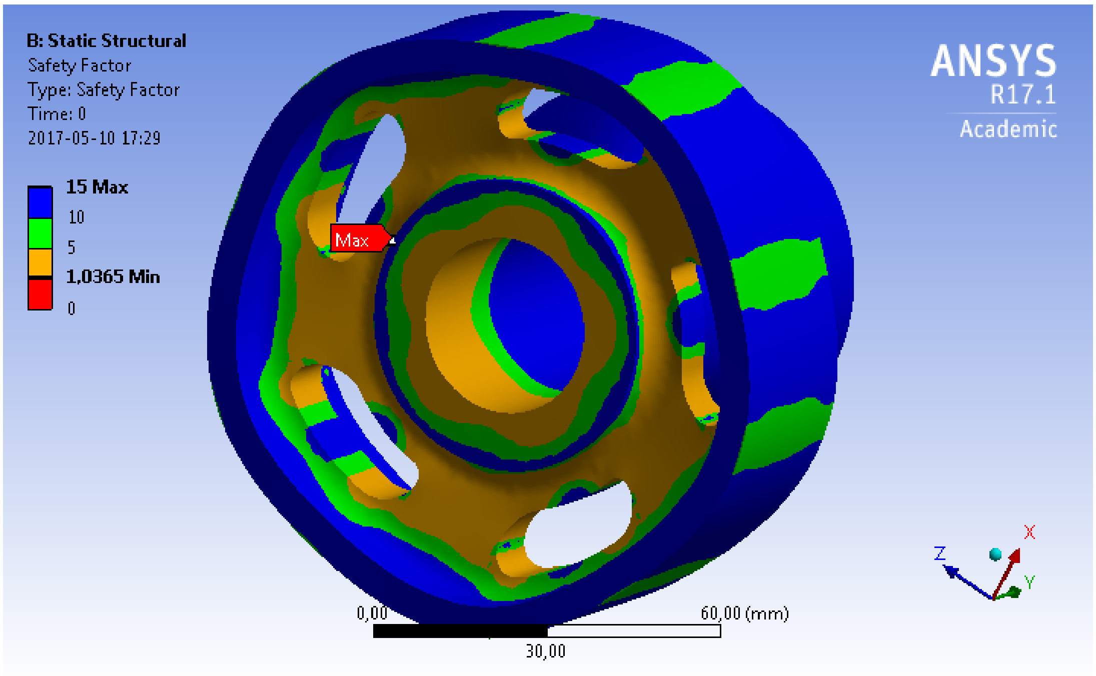
- Inside considered as fixed support
- 3188 Nm moment acting on the outer diameter of sun



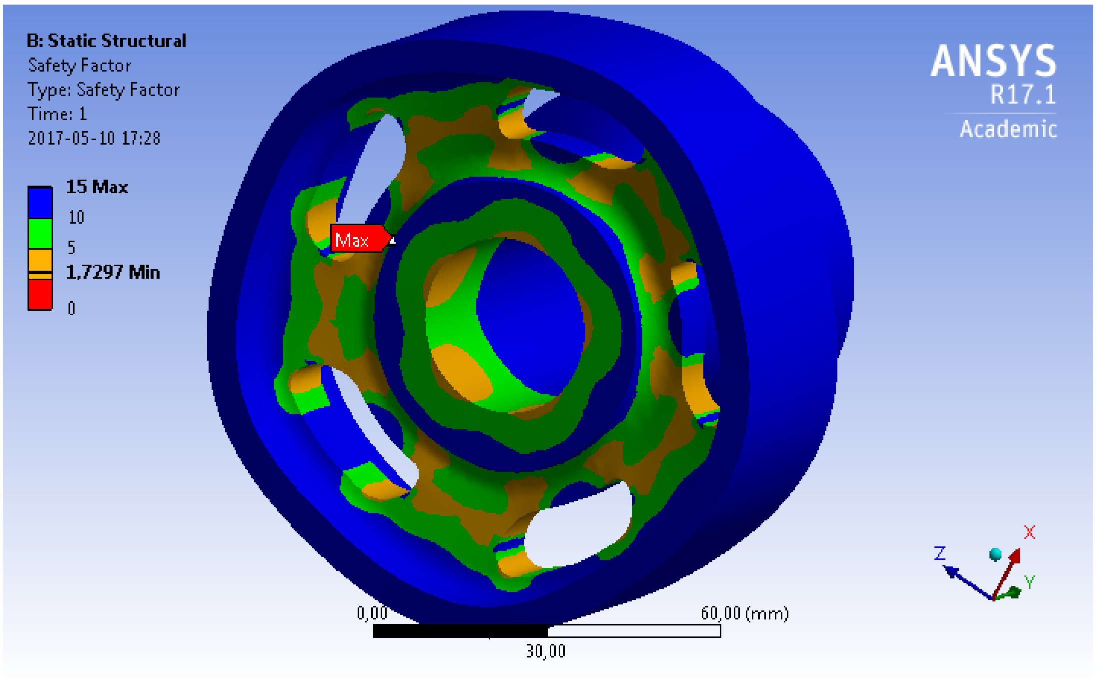


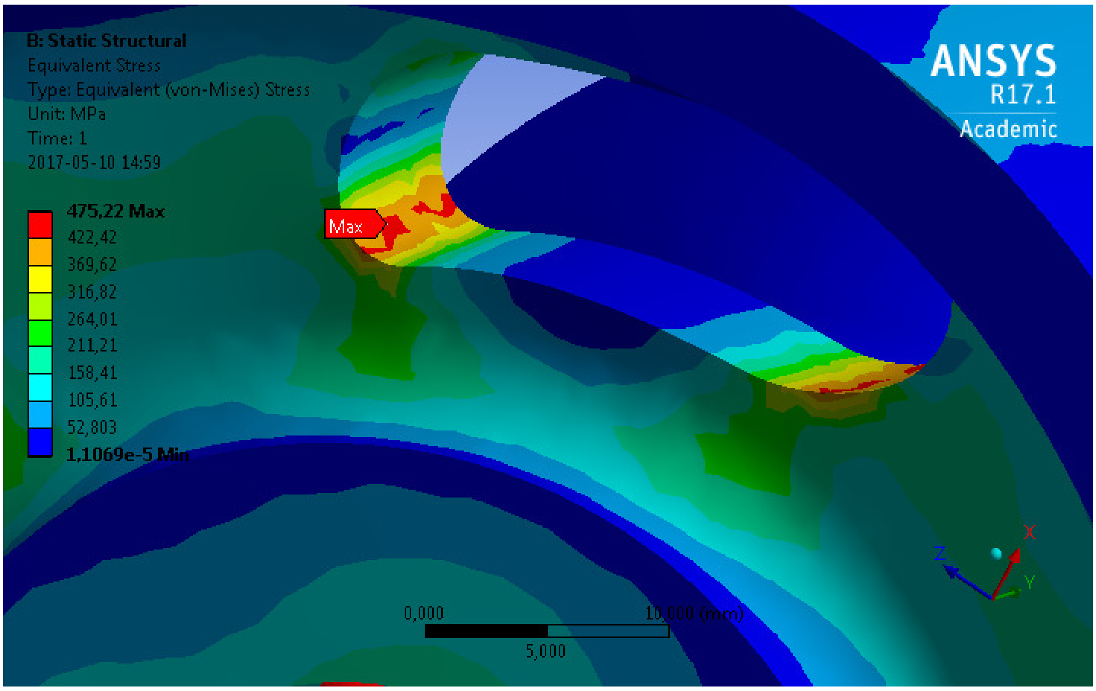


Fatigue safety factor for 10^9 cycles



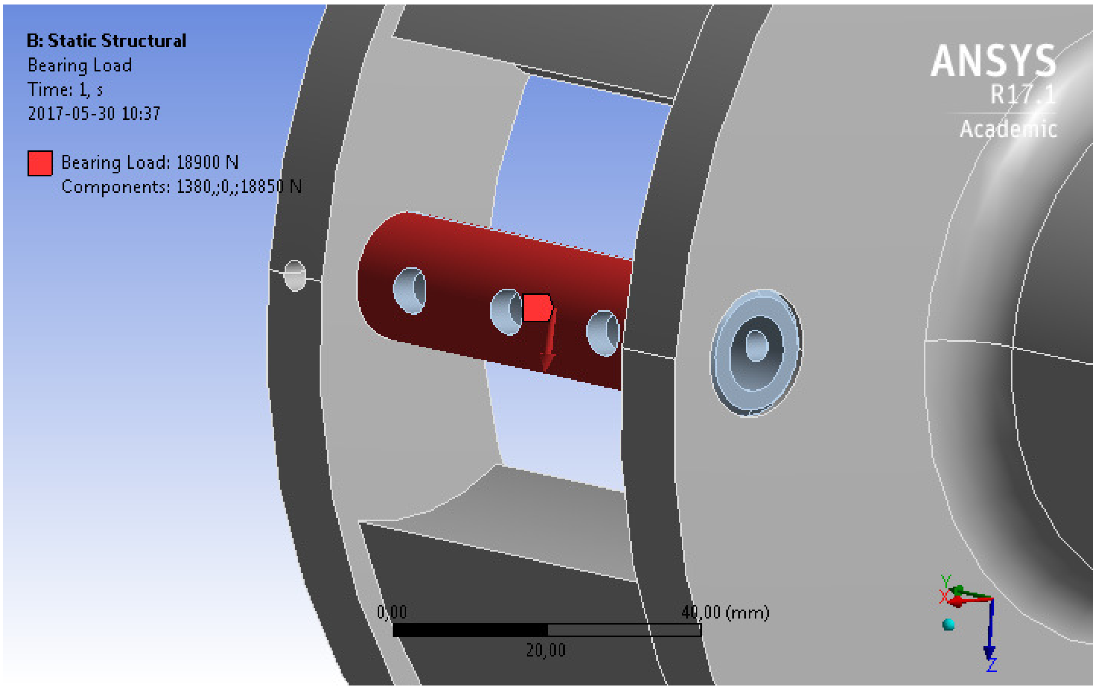
Static Safety factor

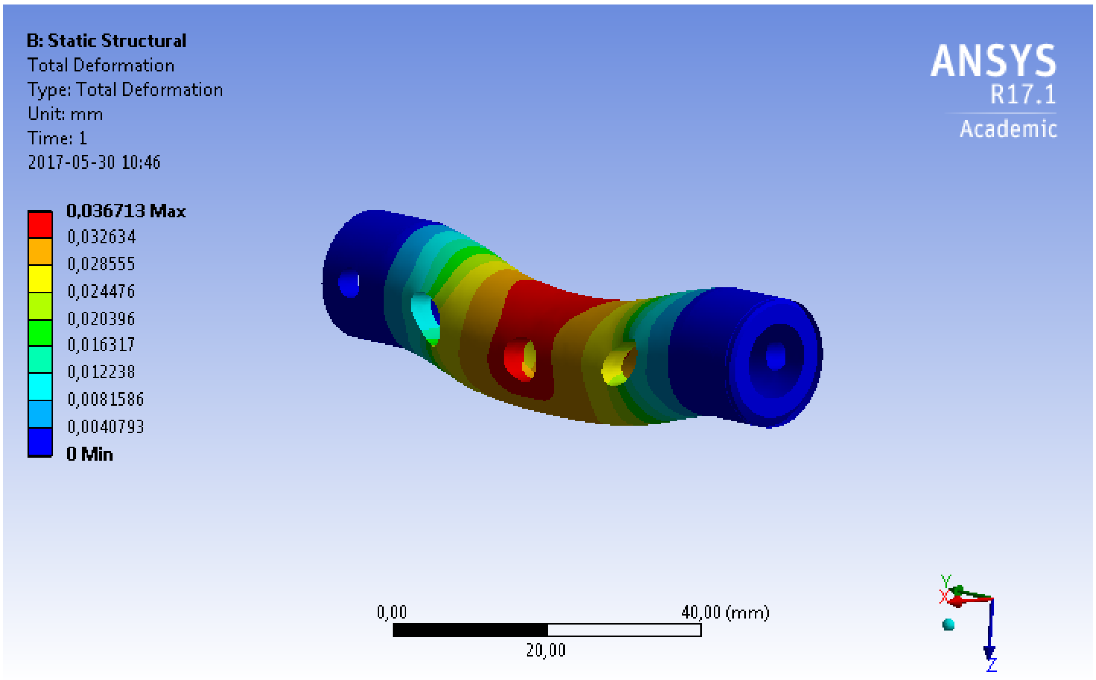


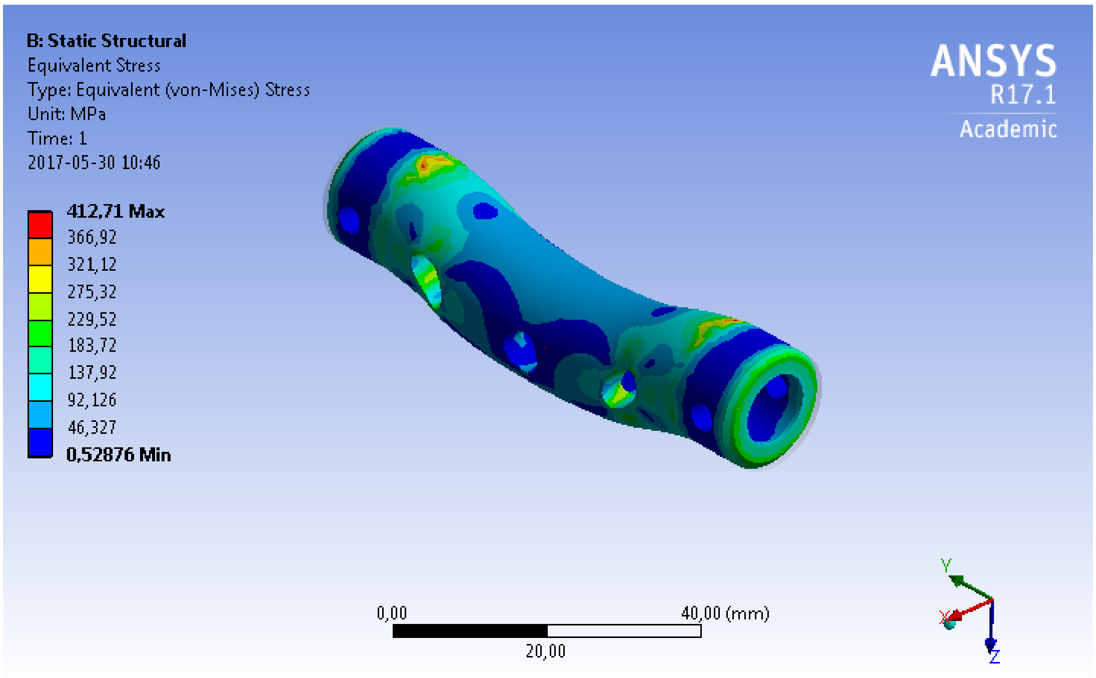


Boundary conditions- Planet shaft

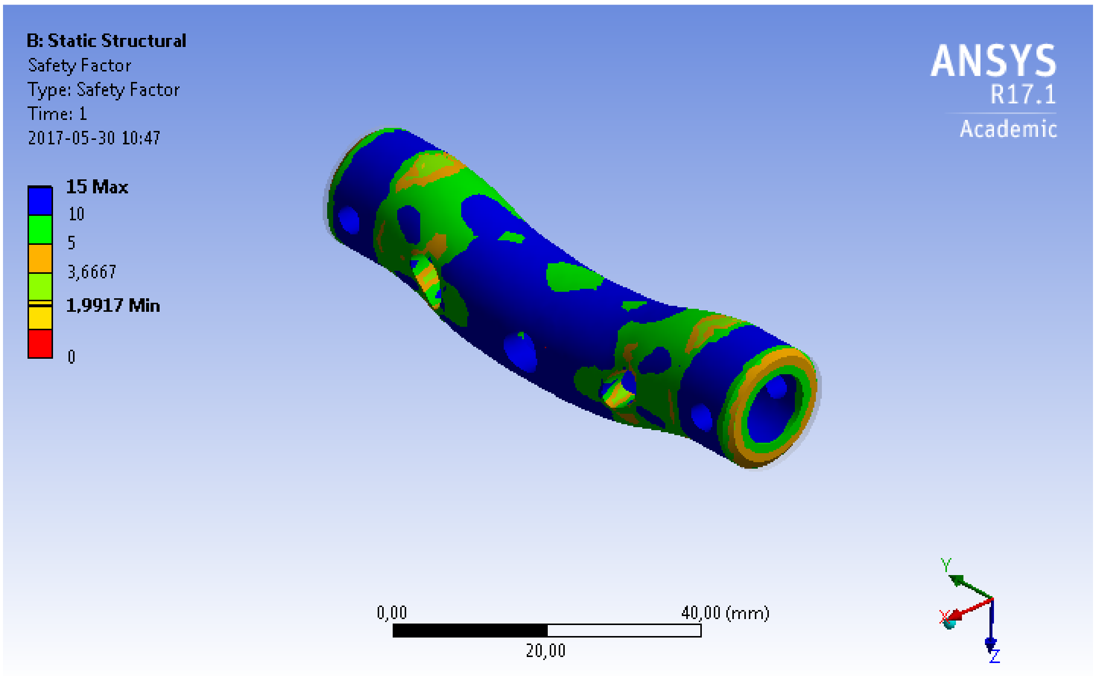
- Carrier is considered as rigid body
- The shaft is fixed in each end
- Load is applied as bearing load and is taken from Table.3.4



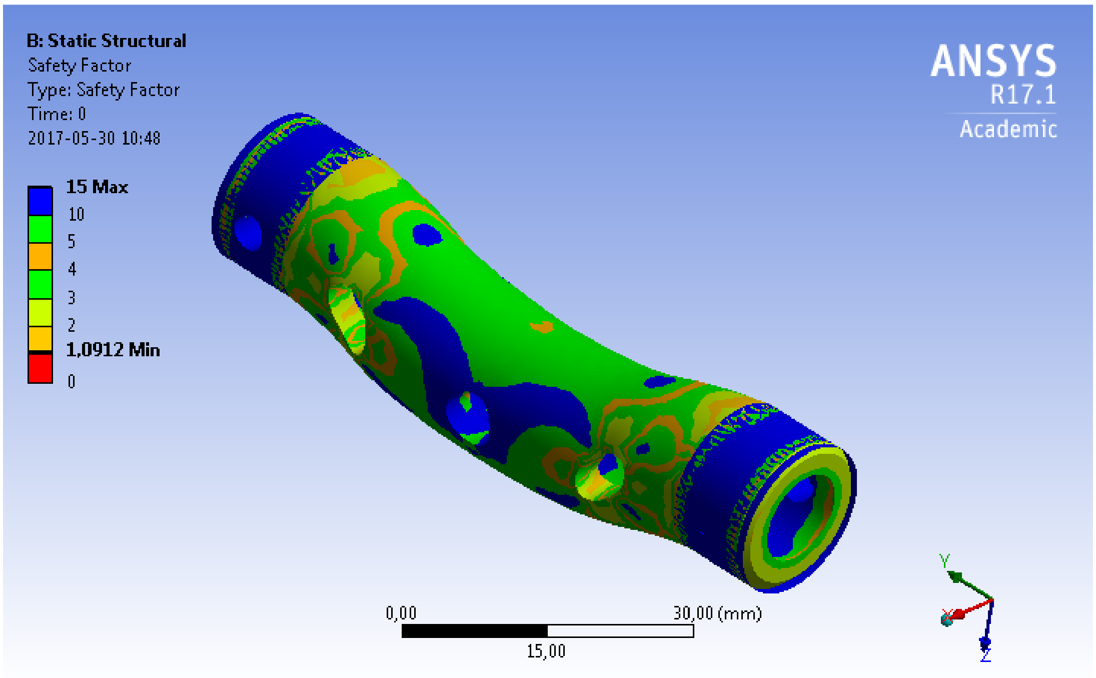




Static safety factor

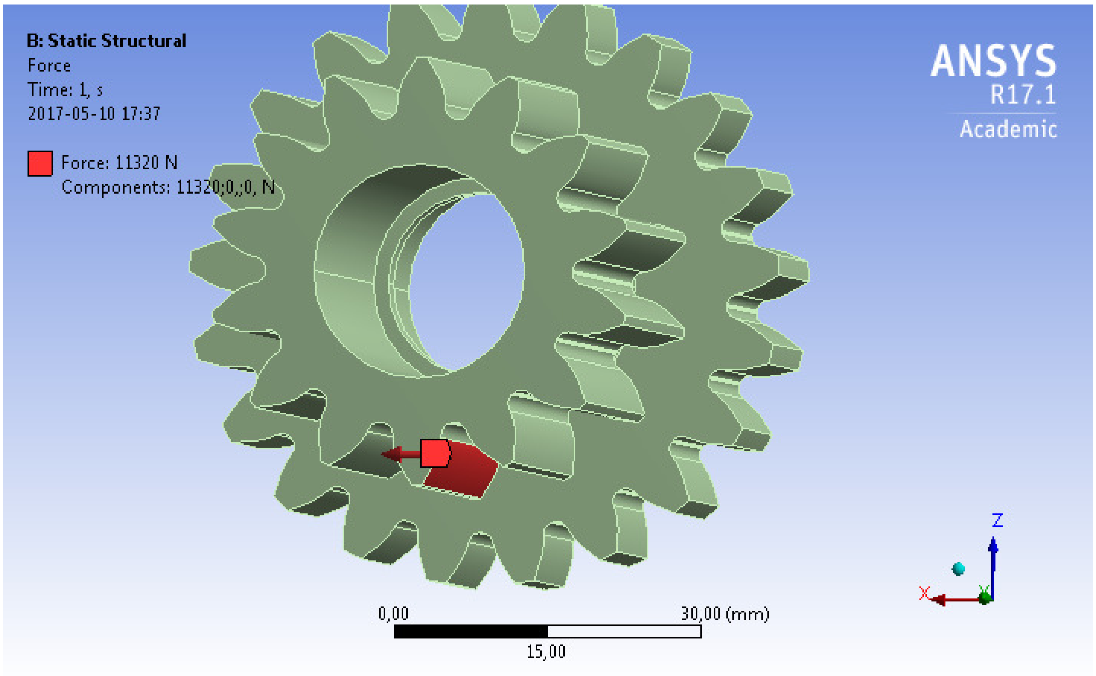


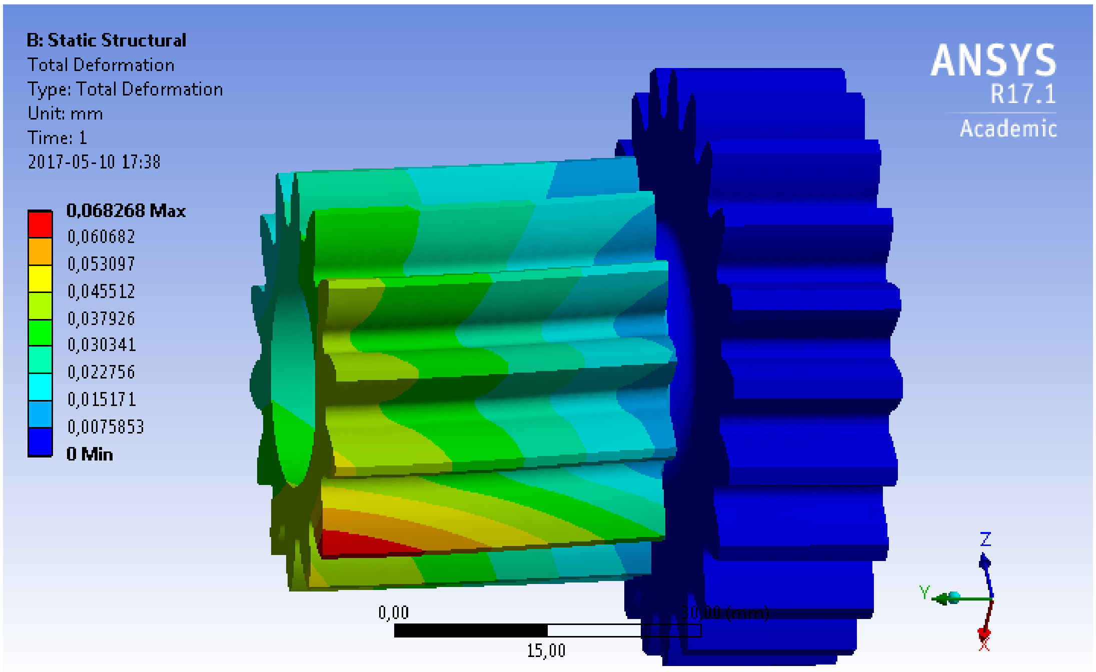
Fatigue safety factor for 10^9 cycles

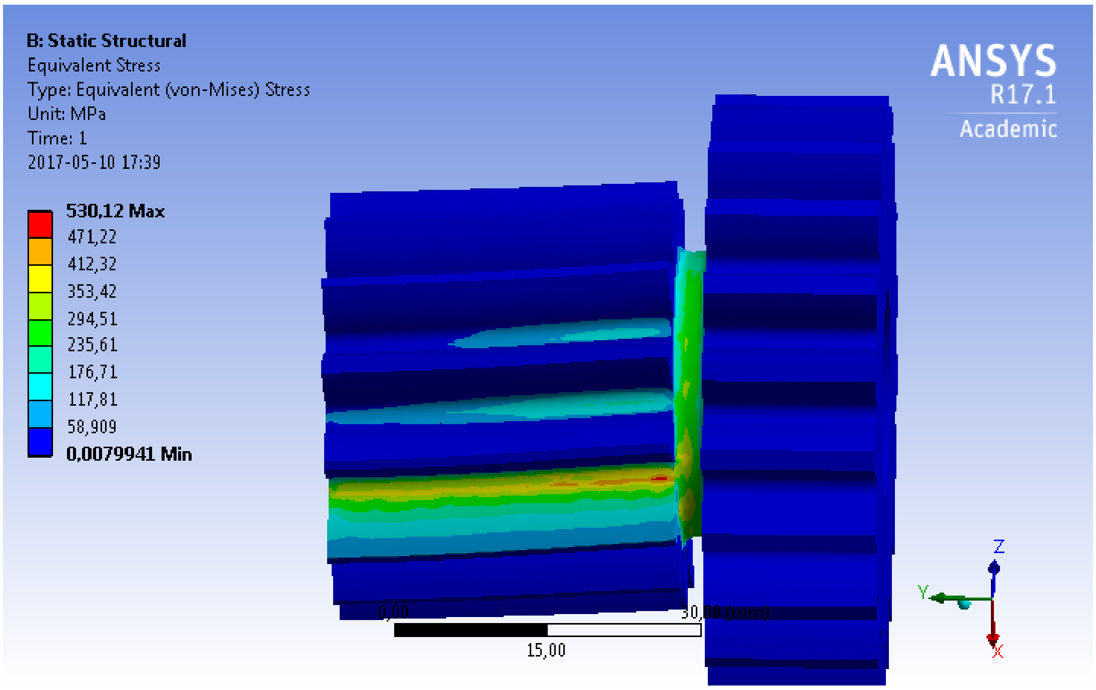


Boundary condition-Max moment on planet with focus on the section between the planets

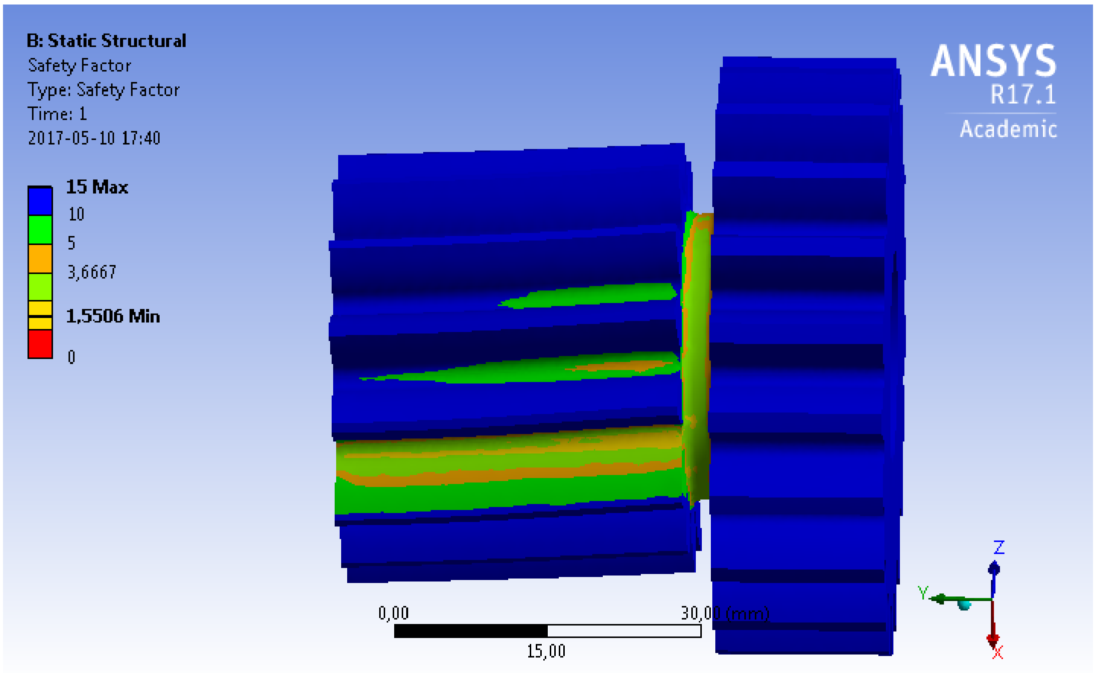
- Inside surface of big planet is fixed support
- 11320 N load acing on one tooth of the smaller planet

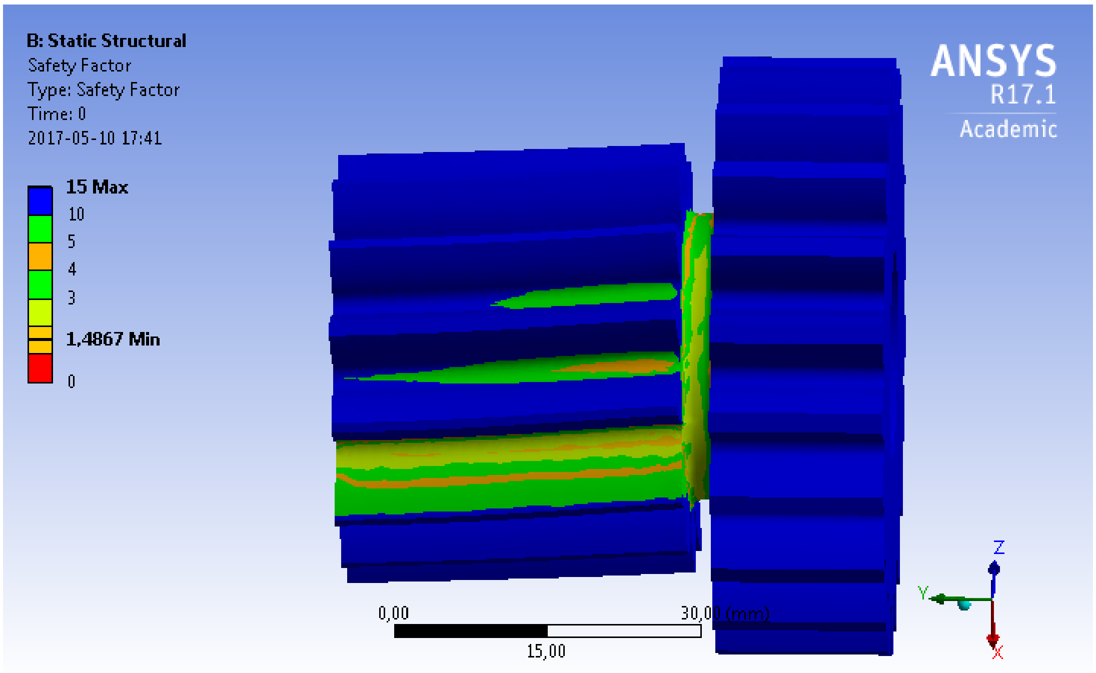






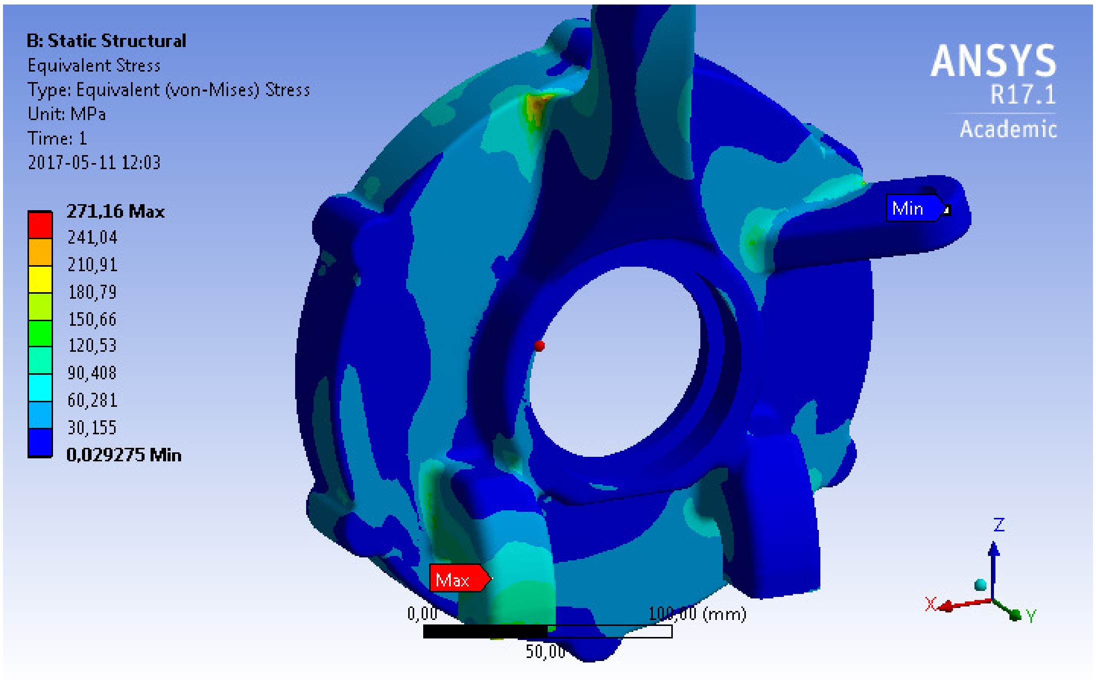
Static safety factor

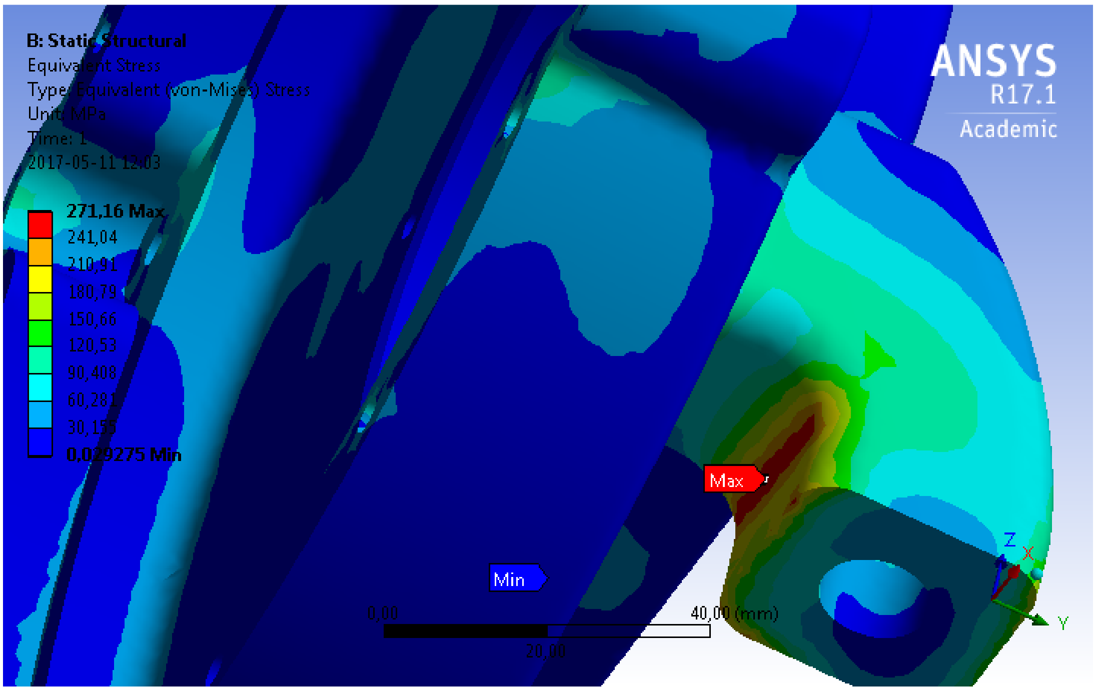


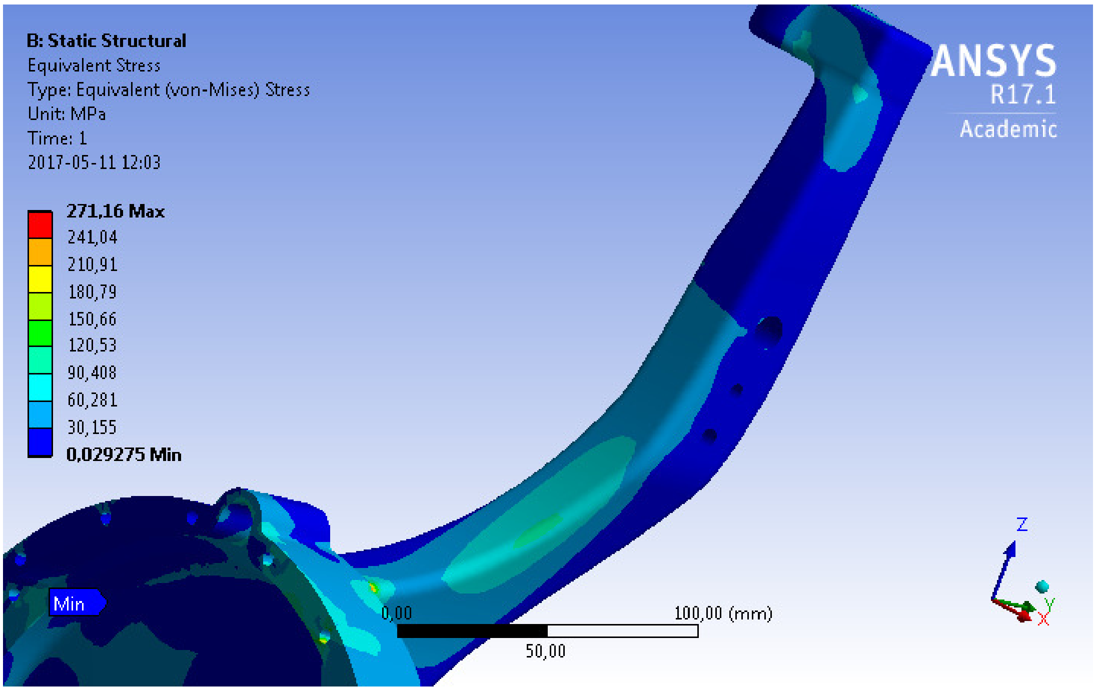

B: Static Structural
Safety Factor
Type: Safety Factor
Time: 0
2017-05-10 17:41
15 Max
10
5
4
3
1,4867 Min
0
ANSYS
R17.1
Academic
0,00
30,00 (mm)
15,00
Z
Y
X

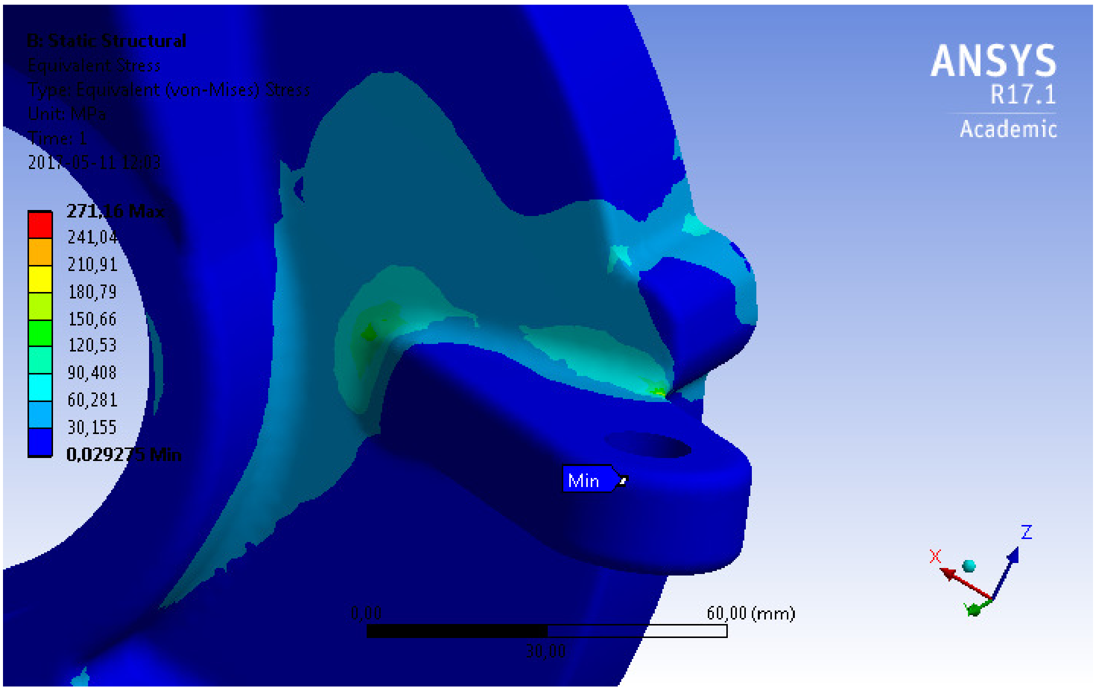
Boundary conditions - Casing/upright

- Remote force acting on tire contact patch and come from Table 3.7
- Upright is constrained at upper ball joint and steering hard point with spherical joint
- Lower ball joint mounting surface is fixed support
- Suspension linkages assumed to be infinitely stiff



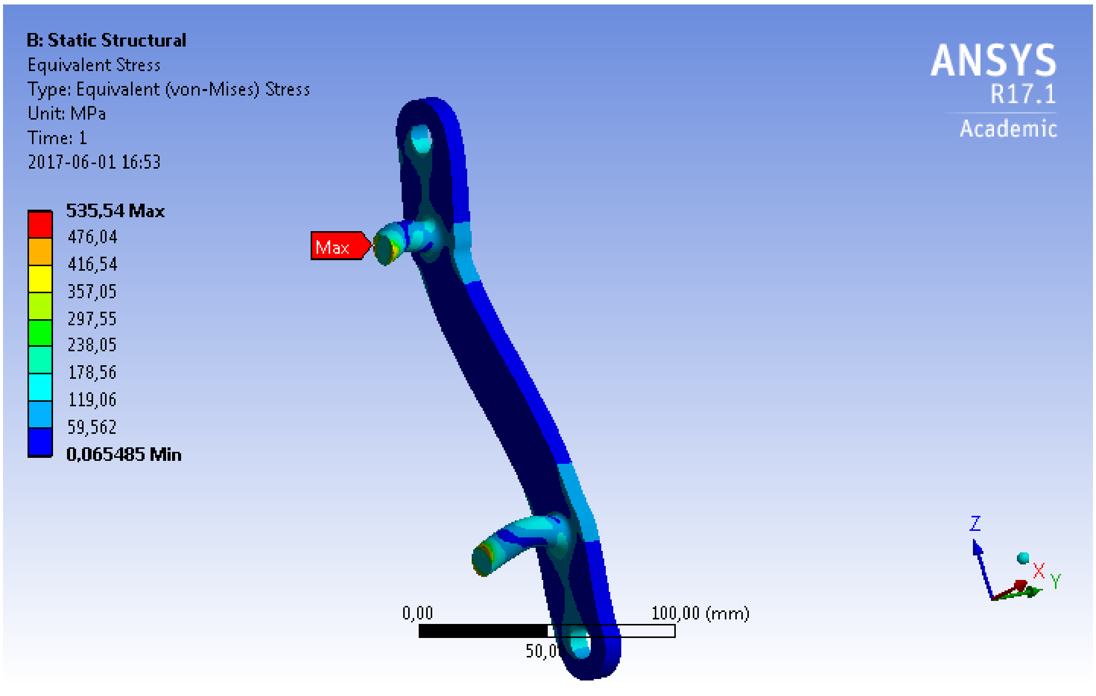


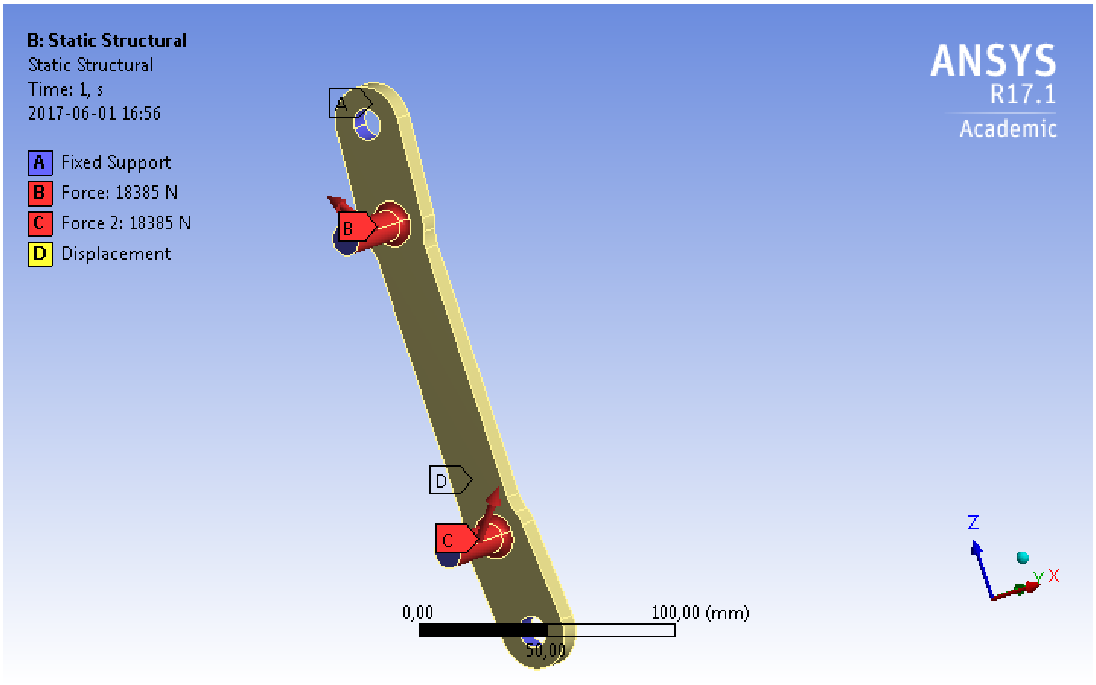

B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2017-05-11 12:03
271,16 Max
241,04
210,91
180,79
150,66
120,53
90,408
60,281
30,155
0,029275 Min
ANSYS
R17.1
Academic
Min
0,00
100,00 (mm)
50,00
Z
Y
X


B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2017-05-11 12:03
271,16 Max
241,04
210,91
180,79
150,66
120,53
90,408
60,281
30,155
0,029275 Min
ANSYS
R17.1
Academic
Min
0,00
60,00 (mm)
30,00
X
Z

Boundary conditions – Brake caliper bracket

- Fixed support in mounting holes and in the end of studs
- No deflection in Y direction for bracket
- 18385 N Force(assumption) acting on each mounting stud





E

Data sheets

E.1 Seals

E.1.1 Radial oil seal

Table E.1: Installation data for radial seal [34]

TSS item no.:	**TRE000700-N7MM2**
Installation Dimension	
Shaft diameter, d1 h11	70
Bore diameter, d2 H8	85
Width, b, d2 H8	8mm

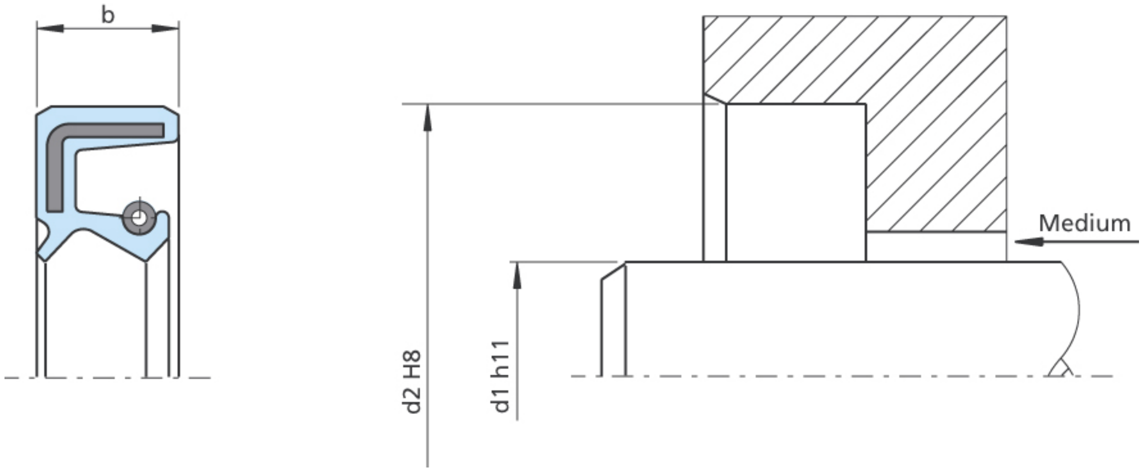


Figure E.1: Installation Dimension [34]

E.1.2 O-Ring

Table E.2: Installation Data for O-ring [34]

TSS item no.:	**ORAR00268-N7237**
Installation Dimension	
Inside diameter d_1	$215.49mm$
Cross section d_2	$3.53mm$
Marking	Without marking
Surface treatment	untreated
Material Properties	
Maximum Operating Temperature	$80°C$
Minimum Operating Temperature	$-50°C$
Hardness Nominal	70.0
Hardness Tolerance	$+/-5$
Hardness Unit	Shore A
Color	0 = Black
Specific Gravity	$1.2700g/cm^3$
Gravity Tolerance	$+/-0.02$
Specific Gravity Unit	g/cm^3
Modulus 100%	$6.9MPa\ or\ N/mm^2$
Tensile Strength	$12.5MPa\ or\ N/mm^2$
Elongation at break	180.0%
Compression Set	13.0000%
Compression Set Test Conditions	$24h/100°C$
TR Point	$-42.0°C$
TR 10 Point	$-42°C$

TRELLEBORG

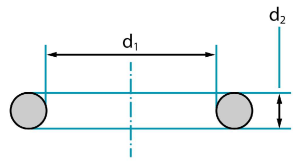
Trelleborg Sealing Solutions

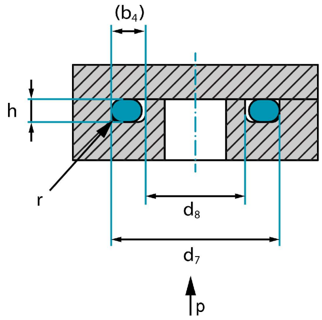
VERSION 5.1

LOAD
AXIAL SEALING (Internal pressure)

Input

	[mm]	Standard/Fit	Lower Tolerance Limit [mm]	Upper Tolerance Limit [mm]
Groove Outside-Ø (d_7)	225.000	H9[1]	0.000	0.115
Groove Inside-Ø (d_8)	215.000	h9[1]	-0.115	0.000
Seal housing height (h)	2.700	recom.	0.000	0.100
Groove Radius (r)	0.300	recom.	-0.100	0.100
O-Ring Inside-Ø (d_1)	215.49	ISO[2]	-1.59	1.59
Cross-Section-Ø (d_2)	3.53	ISO[2]	-0.10	0.10





O-Ring Material Group	
O-Ring Material Hardness [IRHD]	
Linear Thermal Expansion Coefficient 10^{-6} K^{-1}	0
Temperature [°C]	23

Note:

[1] in accordance with ISO 286-2 | [2] ISO = in accordance with ISO 3601-1, Tolerances Class B

Calculation Results

Compression incl. R	min.	=	**18.37**	[%]
	max.	=	**25.62**	[%]
Compression incl. R	min.	=	**0.63**	[mm]
	max.	=	**0.93**	[mm]
Groove Fill	min.	=	**63.81**	[%]
	max.	=	**77.30**	[%]
Stretch OR Inside-Ø	min.	=	**no**	[%]
	max.	=	**0.51**	[%]
Compression OR Outside-Ø	min.	=	**no**	[%]
	max.	=	**no**	[%]
Groove Width (b_4)	min.	=	**5.00**	[mm]
	max.	=	**5.12**	[mm]
Total Compression Force	min.	=		[N]
	max.	=		[N]

Selected O-Ring:	215.49 x 3.53
TSS Part No:	ORAR00268

Recommendations in accordance with ISO 3601-2

Cross-sectional Compression

radial static applications:	10 - 35%
radial dynamic applications:	6 - 27%
axial static applications:	13 - 36%

depending on the O-Ring cross-section-Ø and on the application. Detailed information available in ISO 3601-2.

Stretch at the OR Inside-Ø

radial static, outer sealing:	2 - 8%
radial dynamic, outer sealing:	2 - 5%

Circumferential Compression at the OR Outside-Ø, inner sealing

for OR Inside-Ø > 250 mm:	max. 3%
for OR Inside-Ø ≤ 250 mm:	max. 5%
Housing fill:	max. 85%

Recommendations for FFKM O-Rings

Cross-sectional Compression:	12 - 18%
Stretch at the Inside-Ø:	max. 3%

Trelleborg Sealing Solutions products have diverse application uses. The utilization, selection and designation of any specific Trelleborg Sealing Solutions product in any specific application is the sole responsibility of the O-Ring calculation program user, customer and/or purchaser. Trelleborg Sealing Solutions is not obliged to the user of the O-Ring calculation program, customer and/or purchaser to provide any consultation in conjunction with or as a result thereof. Any measurements, values and/or dimensions obtained through the O-Ring calculation program are guidelines only and as such are intended for orientation purposes only with the sole function to ease the selection of a possible Trelleborg Sealing Solutions product that may be suitable for the use in a particular application. Any Trelleborg Sealing Solutions product ascertained as a result of using data obtained through the O-Ring calculation program for use in a specific application must be assayed by the O-Ring calculation program end user, customer and/or purchaser through the appropriate test methods in order to ensure product suitability. Media contact, operating temperature and assembly conditions can result in significant divergences from the calculated values.
No warranty is therefore provided or implied by Trelleborg Sealing Solutions with regards to applicability of any of its products in conjunction with the specific application intended by the O-Ring calculation program user, customer and/or purchaser. In no event shall Trelleborg Sealing Solutions, or any agent or representative thereof, be liable for any damages whatsoever resulting from non-compliance with this advice.

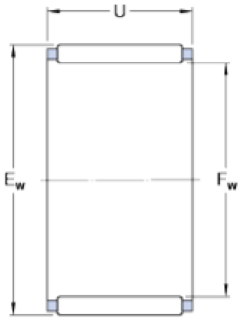
E.2 Bearings

These data sheets come from SKF Group home page [28]

SKF

K 17x21x17

Dimensions



F_w	17	mm
E_w	21	mm
U	17	mm

Calculation data

Basic dynamic load rating	C	11.7	kN
Basic static load rating	C_0	18.3	kN
Fatigue load limit	P_u	2.12	kN
Reference speed		22000	r/min
Limiting speed		26000	r/min

Mass

Mass needle roller and cage assembly	0.0095	kg

Tolerances: Single row, Stainless steel, Filling slots, Double row

Values: Normal (metric), P6, P5, Normal (inch)

Radial internal clearance: Single row, Stainless steel, Filling slots, Double row

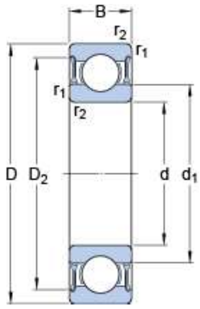
Values: Matched bearing pairs, Stainless steel d < 10 mm, Other bearings

Recommended fits

Shaft and housing tolerances and fits

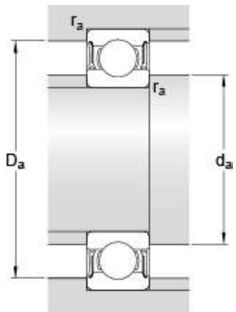
61913-2RZ

Dimensions



d		65	mm
D		90	mm
B		13	mm
d_1	≈	73	mm
D_2	≈	84.2	mm
$r_{1,2}$	min.	1	mm

Abutment dimensions



d_a	min.	69.6	mm
d_a	max.	73.1	mm
D_a	max.	85.4	mm
r_a	max.	1	mm

Calculation data

Basic dynamic load rating	C	17.4	kN
Basic static load rating	C_0	16	kN
Fatigue load limit	P_u	0.68	kN
Reference speed		15000	r/min
Limiting speed		7500	r/min
Calculation factor	k_r	0.02	
Calculation factor	f_0	16.6	

Mass

Mass bearing	0.22	kg

Menu

Select bearing
Select calculation
Enter input parameters
Results

› Feedback

Useful links

Ball bearings
Plain bearings
Roller bearings

Results

Bearing life : 30210 J2/QDF

Input parameters

F_r Radial load	34 kN
F_a Axial load	9.8 kN
n_i Rotational speed of the inner ring	656 r/min
Operating temperature Bearing outer ring	60 °C
η_c specification method	Simplified guidelines
η_c Factor for contamination level	0.55
Viscosity calculation input type	Viscosity input at 40 °C and 100 °C
Viscosity at 40 °C	100 mm^2/s
Viscosity at 100 °C	100 mm^2/s

Warning

κ=4 gives full surface separation, hence the SKF rating life has no additional lubrication benefit for κ>4. Operating at κ>4 is possible but depending on the speed and lubricant quantity, may lead to higher friction and temperature.

Result

L_{10mh} SKF rating life	1000 hour
a_{SKF} SKF life modification factor a_{SKF}	1.6
κ Viscosity ratio	4.95
P Equivalent dynamic bearing load	49.7 kN
η_c Factor for contamination level	0.55
ν_1 Required kinematic viscosity for κ=1	20.2 mm^2/s
L_{10h} Basic rating life	630 hour
C/P Load ratio	2.6

Unit system

Select unit system

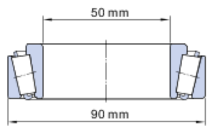
SI Imperial

Selected calculations

Bearing life, Equivalent dynamic bearing load

Remove

Bearing data



Designation	30210 J2/QDF
d	50 mm
D	90 mm
B	43.5 mm
C	130 kN
C_0	183 kN
Type	Tapered roller bearing

View bearing details

Note: The drawing displayed is only for general representation and may not be identical to the selected bearing variant.

Menu

Select bearing
Select calculation
Enter input parameters
Results

› Feedback

Useful links

Ball bearings
Plain bearings
Roller bearings

Results

Bearing life : 30210 J2/QDF

Input parameters

F_r Radial load	34 kN
F_a Axial load	9.8 kN
n_i Rotational speed of the inner ring	656 r/min
Operating temperature Bearing outer ring	60 °C
η_c specification method	Simplified guidelines
η_c Factor for contamination level	0.55
Viscosity calculation input type	Viscosity input at 40 °C and 100 °C
Viscosity at 40 °C	100 mm^2/s
Viscosity at 100 °C	100 mm^2/s

Warning

κ=4 gives full surface separation, hence the SKF rating life has no additional lubrication benefit for κ>4. Operating at κ>4 is possible but depending on the speed and lubricant quantity, may lead to higher friction and temperature.

Result

L_{10mh} SKF rating life	1000 hour
a_{SKF} SKF life modification factor a_{SKF}	1.6
κ Viscosity ratio	4.95
P Equivalent dynamic bearing load	49.7 kN
η_c Factor for contamination level	0.55
ν_1 Required kinematic viscosity for κ=1	20.2 mm^2/s
L_{10h} Basic rating life	630 hour
C/P Load ratio	2.6

Unit system

Select unit system

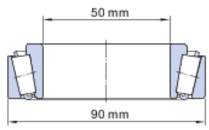
SI Imperial

Selected calculations

Bearing life, Equivalent dynamic bearing load

Remove

Bearing data



Designation	30210 J2/QDF
d	50 mm
D	90 mm
B	43.5 mm
C	130 kN
C_0	183 kN
Type	Tapered roller bearing

View bearing details

Note: The drawing displayed is only for general representation and may not be identical to the selected bearing variant.

F

Matlab code

F.1 Epeleptic-main script

```
clc
close all
clear all

%% inputs
%T=6950/2.4;                          % Input torque
L= 5000;                              % Life in hours
r_tyre=(475*0.7+17*25.4/2)/1000;      % Radius tyre
SpaceReservationPlanets=20;

rho=7800;                   % density of steel
t=0.02;                     % Thickness of ring gear m
r_h= 0.016;                 % radius of shaft hole sun m
r_h1=0.01;                  % radius of shaft hole planet 1 and 2

% Material properties (carburized and case hardened grade 2 AGMA)
s_HP =1550;                 % allowable contact stress (MPa)
s_FP=450;                   % allowable bending stress (MPa)
s_yield=822;                % Yield stress (MPa)

count=0;
for m=[1.5 2 2.5 3]         % Gear module (mm)
for b_stage1=30:35          % Width stage 1 (mm)
for b_stage2=17:20          % Width stage 2 (mm)
for n_planets=5:6           % Number of planets
for i= 40:90                % Number of sun teeths
for j=14:20                 % Number of planet 1 teeths
for k=20:40                 % Number of planet 2 teeths

z_s= i;                               % sun
z_p1= j;                              % planet 1
z_p2= k;                              % planet 2
z_r= z_s+z_p1+z_p2;                   % Number of ring teeths calculated
z_sun_virtual=z_r-2*z_p2;             % Virtual sun tooth number

d_s=m*z_s;                            % Pitch diameter of sun (mm)
d_r=m*z_r;                            % Pitch diameter of sun
d_p1=m*z_p1;                          % Pitch diameter of sun
d_p2=m*z_p2;                          % Pitch diameter of sun

r_s=d_s/2;                            % Radius of sun (mm)
r_r=d_r/2;                            % Radius of ring
r_p1=d_p1/2;                          % Radius of planet 1
r_p2=d_p2/2;                          % Radius of planet 2

r= 1+((z_r*z_p1)/(z_p2*z_s));         % Total gear ratio
T_max1=6950/r;                       % Max Torque (Nm)
T_20=6950/r*0.2;                     % 20% Max Torque

% Tangential tooth forces
F_t_max1=T_max1/(r_s/1000)/n_planets;    % Max static tooth force stage 1 (N)
F_t_max2=T_max1/(r_s/1000)/n_planets*(z_sun_virtual/z_s); % Max static tooth
force stage 2

F_t_stage1=T_20/(r_s/1000)/n_planets;    % Dynamic tooth force stage 1
```

```
F_t_stage2=T_20/(r_s/1000)/n_planets*(z_sun_virtual/z_s);  % Dynamic tooth
force stage 2

% Velocitys
% Sun
omega_sun=60/(3.6*r_tyre)*r;  % (rad/s)
rpm_sun=omega_sun/(2*pi)*60;  % (rpm)
v_t_sun=omega_sun*(r_s/1000); % (m/s)
% Carrier
omega_carrier=60/(3.6*r_tyre);
rpm_carrier=omega_carrier/(2*pi)*60;
v_carrier=omega_carrier*((r_s+r_p1)/1000);
% Planets
omega_planets=(v_t_sun-v_carrier)/(r_p1/1000);
rpm_planets=omega_planets/(2*pi)*60;
% Virtual sun in stage 2
omega_virtual_sun=((-omega_carrier)*(-z_r/z_sun_virtual))+omega_carrier;
rpm_virtual_sun= omega_virtual_sun/(2*pi)*60;

% Constraints
if r>1.5 && r<2.2 % ratio coverage
if d_s>40 && d_p1>25 && d_r<230
if rem(z_s,n_planets)==0 && rem(z_r,n_planets)==0
if ((360/n_planets)/360)*pi*2*((d_s+d_p1)/2) - d_p2 > SpaceReservationPlanets
if 0.3 < b_stage1/d_s && b_stage1/d_s<1
if 0.3 < b_stage1/d_p1 && b_stage1/d_p1<1
if 0.3 < b_stage1/d_p2 && b_stage1/d_p2<1

x_stage_1=0.4; % Profile shift coeff stage 1
x_stage_2=0.5; % Profile shift coeff stage 2

% Stress calculation for sun and planet 1 according to AGMA standards
% (Mpa)
s_F_max_stage1_planet =
bending_stress_static_AGMA(F_t_max1,z_s,z_p1,m,b_stage1,n_planets,x_stage_1,1)
;
S_F_max_stage1_planet = s_yield/s_F_max_stage1_planet;

s_F_max_stage1_sun =
bending_stress_static_AGMA(F_t_max1,z_s,z_p1,m,b_stage1,n_planets,x_stage_1,0)
;
S_F_max_stage1_sun = s_yield/s_F_max_stage1_sun;

s_F_stage1_sun =
bending_stress_AGMA(F_t_stage1,z_s,z_p1,m,b_stage1,n_planets,omega_sun,x_stage
_1,0);
S_F_stage1_sun =
allowable_bending_stress_AGMA(s_F_stage1_sun,L,rpm_sun,n_planets,s_FP,0);

s_F_stage1_planet =
bending_stress_AGMA(F_t_stage1,z_s,z_p1,m,b_stage1,n_planets,omega_sun,x_stage
_1,1);
S_F_stage1_planet =
allowable_bending_stress_AGMA(s_F_stage1_planet,L,rpm_planets,1,s_FP,1);
```

```
s_H_stage1_sun =
contact_stress_AGMA(F_t_stage1,z_p1,z_s,m,b_stage1,n_planets,omega_sun,1,x_sta
ge_1);
S_H_stage1_sun =
allowable_contact_stress_AGMA(s_H_stage1_sun,L,rpm_sun,n_planets,s_HP);
S_H_stage1_planet =
allowable_contact_stress_AGMA(s_H_stage1_sun,L,rpm_planets,1,s_HP);

% Stress calculation for ring and planet 2 according to AGMA standards
s_F_max_stage2_planet =
bending_stress_static_AGMA(F_t_max2,z_sun_virtual,z_p2,m,b_stage2,n_planets,x_
stage_2,1);
S_F_max_stage2_planet = s_yield/s_F_max_stage2_planet;

s_F_stage2_planet =
bending_stress_AGMA(F_t_stage2,z_sun_virtual,z_p2,m,b_stage2,n_planets,omega_v
irtual_sun,x_stage_2,1);
S_F_stage2_planet =
allowable_bending_stress_AGMA(s_F_stage2_planet,L,rpm_planets,1,s_FP,1);

s_H_stage2_planet =
contact_stress_AGMA(F_t_stage2,z_p2,z_r,m,b_stage2,n_planets,omega_carrier,0,x
_stage_2);
S_H_stage2_ring =
allowable_contact_stress_AGMA(s_H_stage2_planet,L,rpm_carrier,n_planets,s_HP);
S_H_stage2_planet =
allowable_contact_stress_AGMA(s_H_stage2_planet,L,rpm_planets,1,s_HP);

% Mass estimation (Kg)
m_s=rho*(b_stage1/1000)*pi*(((d_s/2000)^2)-r_h^2); % mass sun
m_p1=rho*(b_stage1/1000)*pi*(((d_p1/2000)^2)-r_h1^2); % mass planet 1
m_p2=rho*(b_stage2/1000)*pi*(((d_p2/2000)^2)-r_h1^2); % mass planet 2
m_r=rho*(b_stage2/1000)*pi*(((d_r/2000)^2)-(d_r/2000-t)^2); % mass ring
Tot_mass=m_s+(m_p1+m_p2)*n_planets+m_r;             % Total mass

% Safety factor
if S_F_max_stage1_planet<1.2
elseif S_F_max_stage1_sun<1.2
elseif S_F_stage1_sun<1.2
elseif S_F_stage1_planet<1.2
elseif S_H_stage1_sun<1.2
elseif S_H_stage1_planet<1.2
elseif S_F_max_stage2_planet<1.2
elseif S_F_stage2_planet<1.2
elseif S_H_stage2_ring<1.2
elseif S_H_stage2_planet<1.2
elseif Tot_mass>10
else
count=count+1;
x(1,count)=z_s;
x(2,count)=z_p1;
x(3,count)=z_p2;
x(4,count)=z_r;
x(5,count)=r;
x(6,count)=d_r;
x(7,count)=m;
```

```
x(8,count)=b_stage1;
x(9,count)=b_stage2;
x(10,count)=n_planets;
x(11,count)=Tot_mass;
x(12,count)=m_s;

x(13,count)=s_F_max_stage1_sun;
x(14,count)=S_F_max_stage1_sun;
x(15,count)=s_F_max_stage1_planet;
x(16,count)=S_F_max_stage1_planet;
x(17,count)=s_F_stage1_sun;
x(18,count)=S_F_stage1_sun;
x(19,count)=s_F_stage1_planet;
x(20,count)=S_F_stage1_planet;
x(21,count)=s_H_stage1_sun;
x(22,count)=S_H_stage1_sun;
x(23,count)=S_H_stage1_planet;
x(24,count)=s_F_max_stage2_planet;
x(25,count)=S_F_max_stage2_planet;
x(26,count)=s_F_stage2_planet;
x(27,count)=S_F_stage2_planet;
x(28,count)=s_H_stage2_planet;
x(29,count)=S_H_stage2_planet;
x(30,count)=x_stage_1;
x(31,count)=x_stage_2;

end
end
end
end
end
end
end
end
end
end
end
end
end
end
end

x_row=x';
x_sort=sortrows(x_row,11)';
```

F.2 Bending stress dynamic

```
function s_F =
bending_stress_AGMA(F_t,z_s,z_p1,m,b,n_planets,omega_in,x,index)

beta=0;
d=m*z_s;
r=d/(2*1000);
u=z_s/z_p1;
a=m*(z_s+z_p1)/2; % Center distance
dw_1=2*a/(u+1); % pinion diameter
% Transverse metric module,mm (same as normal metric module for spur gear)
m_t=m/cosd(beta);

% Overload factor (application factor)
K_o=1.35;

% Dynamic factor (see table in AGMA standards)
%  rpm_shaft=2000/(3.6*2.566*4.1);
% omega_in=50;%2*pi*rpm_shaft/60;
v_t=omega_in*r;
Qv=11;
B_v=0.25*(12-Qv)^(0.667);
A_v=50+56*(1-B_v);
K_v=((A_v+sqrt(200*v_t))/A_v)^B_v;
%  Size factor for m< 5mm K_s=1
K_s=1;
% load distribution factor
K_Hmc=1; % Lead corection factor, 1 for unmodified leads and 0.8 for gear
     % with leads properly modifiedby crowning

K_Hpf=  b/(10*dw_1)-0.03754+0.000492*b; % pinion proportion factor

K_Hpm=1; % pinion proportion modifier

A=0.0675;
B=0.504*10^(-3);
C=-1.44*10^(-7);
K_Hma= A+B*b+C*b^2;  % mesh alignment factor, A,B and C for precision enclosed
gear units

K_He= 1;   % mesh alignment correction factor

K_Hbeta=1+K_Hmc*(K_Hpf*K_Hpm+K_Hma*K_He);
K_H=K_Hbeta;

% Rim thickess factor (might be higher depending on gear wheel design)
K_B=1;

% Geometry factor for bending stress
if index==1
%Y_J=Geometry_factor_bending_table(z_p1,z_s);
Y_J=Geometry_factor_bending(z_p1,z_s,x);
elseif index==0
Y_J=Geometry_factor_bending_gear(z_p1,z_s,x);
end
% Mesh load factor
```

```
if n_planets<=3
K_y=1;
elseif n_planets==4
K_y=1.2;
elseif n_planets==5
K_y=1.3;
else
K_y=1.4;
end

% Bending stress
s_F=K_y*F_t*K_o*K_v*K_s*K_H*K_B/(b*m_t*Y_J);
end
```

F.3 Bending stress static

```
function s_F = bending_stress_static_AGMA(F_t,z_s,z_p1,m,b,n_planets,x,index)

beta=0;
d=m*z_s;
r=d/(2*1000);
u=z_s/z_p1;
a=m*(z_s+z_p1)/2; % Center distance
dw_1=2*a/(u+1); % pinion diameter

% Transverse metric module,mm (same as normal metric module for spur gear)
m_t=m/cosd(beta);
% Overload factor (application factor)
K_o=1;
K_v=1;  %((A_v+sqrt(200*v_t))/A_v)^B_v;
%  Size factor for m< 5mm K_s=1
K_s=1;
% load distribution factor
K_Hmc=1; % Lead corection factor, 1 for unmodified leads and 0.8 for gear
     % with leads properly modifiedby crowning
K_Hpf=  b/(10*dw_1)-0.03754+0.000492*b; % pinion proportion factor
K_Hpm=1; % pinion proportion modifier

A=0.0675;
B=0.504*10^(-3);
C=-1.44*10^(-7);
K_Hma= A+B*b+C*b^2;  % mesh alignment factor, A,B and C for precision enclosed
gear units
K_He= 1;   % mesh alignment correction factor

K_Hbeta=1+K_Hmc*(K_Hpf*K_Hpm+K_Hma*K_He);
K_H=K_Hbeta;
% Rim thickess factor (might be higher depending on gear wheel design)
K_B=1;

% Geometry factor for bending stress
if index==1
%Y_J=Geometry_factor_bending_table(z_p1,z_s);
Y_J=Geometry_factor_bending(z_p1,z_s,x);
elseif index==0
Y_J=Geometry_factor_bending_gear(z_p1,z_s,x);
end

if n_planets<=3
K_y=1;
elseif n_planets==4
K_y=1.2;
elseif n_planets==5
K_y=1.3;
else
K_y=1.4;
end

% Bending stress
s_F=K_y*F_t*K_o*K_v*K_s*K_H*K_B/(b*m_t*Y_J);
end
```

F.4 Contact stress

```
function s_H =
contact_stress_AGMA(F_t,n1,n2,m,b,n_planets,omega_in,gear_type,x)
%Pitting resistance according to AGMA

d=m*n2;
r=d/(2*1000);

if gear_type==1
a=m*(n2+n1)/2;
elseif gear_type==0
a=m*(n2-n1)/2;
end

u=n2/n1;
if gear_type==1
dw_1=2*a/(u+1);
elseif gear_type==0
dw_1=2*a/(u-1);
end

% Elastic coefficient
v_1= 0.3; %v_1 and v_2 is Poisson's ratio for pinion and gear
v_2= 0.3;
E_1= 2.06*10^5; %E_1 and E_2 is modulus of Elasticity for pinion and
gear(N/mm^2)
E_2= 2.06*10^5;
Z_E= sqrt(1/(pi*(((1-v_1^2)/E_1)+((1-v_2^2)/E_2))));

% Overload factor (application factor)
K_o=1.35;

% Dynamic factor (see table in AGMA standards)

v_t=omega_in*r;
Qv=11;
B_v=0.25*(12-Qv)^(0.667);
A_v=50+56*(1-B_v);
K_v=((A_v+sqrt(200*v_t))/A_v)^B_v;

%  Size factor for m< 5mm K_s=1
K_s=1;

% load distribution factor
K_Hmc=1; % Lead corection factor, 1 for unmodified leads and 0.8 for gear
     % with leads properly modifiedby crowning
if b/(10*dw_1)<0.05
K_Hpf= 0.05-0.025;
elseif b<=25
K_Hpf= b/(10*dw_1)-0.025;
elseif b>25 && b<432
K_Hpf= b/(10*dw_1)-0.03754+0.000492*b; % pinion proportion factor
end

K_Hpm=1; % pinion proportion modifier
```

```
A=0.0675;
B=0.504*10^(-3);
C=-1.44*10^(-7);
K_Hma= A+B*b+C*b^2;  % mesh alignment factor, A,B and C for precision enclosed
gear units

K_He= 1;   % mesh alignment correction factor

K_Hbeta=1+K_Hmc*(K_Hpf*K_Hpm+K_Hma*K_He);
K_H=K_Hbeta;

% Surface condition factor for pitting resistance
Z_R=1;

% Geometry factor for pitting resistance
if gear_type==1
Z_I=Geometry_factor_pitting(n1,n2,gear_type,x);
elseif gear_type==0
Z_I=Geometry_factor_pitting(n1,n2,gear_type,x);
end
%
% Mesh load factor

if n_planets<=3
K_y=1;
elseif n_planets==4
K_y=1.2;
elseif n_planets==5
K_y=1.3;
else
K_y=1.4;
end

% contact stress
s_H=Z_E *sqrt(K_y*F_t*K_o*K_v*K_s*K_H*Z_R/(dw_1*b*Z_I));
end
```

F.5 Geometry factor bending gear

```
function Y_J=Geometry_factor_bending_gear(n1,n2,x)
%AGMA 908-B89 Bending strength geometry factor J
%% Basic gear gometry (dimensionless)
mG=n2/n1; % Gear ratio
phi_n=20*pi/180; % standard normal pressure angle in rad
psi=0; % Standard helix angle

R1=n1/(2*cos(psi)); % Standard (reference) pitch radius  R1 and R2 (eq. 3.2
and 3.3 in AGMA 908-B89)
R2=R1*mG;

phi=atan(tan(phi_n)/cos(psi)); % Standard transverse pressure angle (eq. 3.4
in AGMA 908-B89)

Rb1=R1*cos(phi); % Pinion base radius (eq. 3.5 in AGMA 908-B89)
Rb2=Rb1*mG; % Gear base radius (eq. 3.6 in AGMA 908-B89)

Ro1=((n1+2)/2)+x;
Ro2=((n2+2)/2)-x;

Pb=2*pi*Rb1/n1; % Transverse base pitch (eq. 3.8 in AGMA 908-B89)
PN=pi*cos(phi_n); % Normal base pitch (eq. 3.9 in AGMA 908-B89)

Cr=(n1+n2)/2; % Operating center distance

phi_r= acos((Rb2+Rb1)/Cr); % Opperating transverse pressure angle (eq. 3.7 in
AGMA 908-B89)
psi_b= acos(PN/Pb); % Base helix angle (eq. 3.10 in AGMA 908-B89)

m_N=1;

psi_r=atan(tan(psi_b)/cos(phi_r)); % Opperating Helix angle (eq. 3.27 in AGMA
908-B89)
phi_nr=asin(cos(psi_b)*sin(phi_r)); % Operating Normal Pressure angle (eq.
3.28 in AGMA 908-B89)

C6=Cr*sin(phi_r); %10,6055 (eq. 3.11 in AGMA 908-B89)
C3=C6/(mG+1);
C5=(Ro1^2-Rb1^2)^0.5; % (eq. 3.15 in AGMA 908-B89)
C2=C5-Pb; % (eq. 3.16 in AGMA 908-B89)
C1=(C6-(Ro2^2-Rb2^2)^0.5); % (eq. 3.13 in AGMA 908-B89)
C4=C1+Pb;

T1_g=(C6-(Ro1^2-Rb1^2)^0.5);
T3_g=C6-C3;
T5_g=(Ro2^2-Rb2^2)^0.5;
T2_g=T5_g-Pb;
T4_g=T1_g+Pb;
Z=C5-C1; % (eq. 3.17 in AGMA 908-B89)

%% Virtual spur gear
n=n2/cos(psi)^3; % Virtual tooth number,n (eq. 5.2 in AGMA 908-B89)
r_n=n/2; % Standard (reference) pitch radius of virtual spur gear, (eq. 5.3 in
AGMA 908-B89)
r_nb=Rb2; % Virtual base radius
```

```
r_na= Ro2; % Virtual outside radius

%% Pressure angle at the Load application point
phi_nW=atan(T4_g/r_nb);
%% Generating Rack shift coefficient
P_nd=1; % Normal diametral pitch
x=-x; % Addendum modification factor
delta_sn=0.024/P_nd; % Amount gear tooth is thinned for backlash
x_g=x-(delta_sn)/(2*tan(phi_n)); % Generating Rack Shift Coefficient (eq. 5.19
in AGMA 908-B89)
sn=pi/2+2*x_g*tan(phi_n); % normal circular tooth thickness measured on the
Standard (reference) pitch cylinder (eq. 5.21 in AGMA 908-B89)

%% Load angle and load radius

inv_phi_n=tan(phi_n)-phi_n; % (eq. 5.24 in AGMA 908-B89)
inv_phi_np=inv_phi_n+sn/n ; % (eq. 5.23 in AGMA 908-B89)
phi_nL=tan(phi_nW)-inv_phi_np; % Load angle (eq. 5.22 in AGMA 908-B89)
r_nL=r_nb/cos(phi_nL); %(eq. 5.28 in AGMA 908-B89)

%% Tool Geometry
n_c=10000; % Tool tooth number
n_o=n_c/cos(psi)^3; % Virtual tooth number of tool (eq. 5.29 in AGMA 908-B89)
r_no=n_o/2;
r_nbo=r_no*cos(phi);
R_oc=r_no+1.25;

s_no=pi/2;
x_o=(s_no-pi/2)/(2*tan(phi_n));  %Addendum modification factor of the tool
x_o=0 when s_no=pi/2 (eq. 5.35 in AGMA 908-B89)
rho_ao=0.38;
h_ao=R_oc-r_no;
r_no_s=r_no+h_ao-rho_ao; % Radius to center "S" of tool tip radius, (eq. 5.39
in AGMA 908-B89)

phi_ns=acos(r_nbo/r_no_s); % Pressure angle at point "S" on tool (eq. 5.40 in
AGMA 908-B89)
inv_phi_ns=tan(phi_ns)-phi_ns; % (eq. 5.41 in AGMA 908-B89)
inv_phi_npo=inv_phi_n+s_no/(2*r_no);
lamda_ns=(inv_phi_npo-inv_phi_ns-rho_ao/r_nbo)*2;  % Angle to center "S" of
tool tip radius (eq. 5.47 in AGMA 908-B89)

%% Generating Pressure angle

inv_phi_n2dot=inv_phi_n+2*(x_g+x_o)*tan(phi_n)/(n+n_o);
phi_n2dot(1)=(3*inv_phi_n2dot)^0.33; % First trial value ,(eq. 5.49 in AGMA
908-B89)
for i=1:100

phi_n2dot(i+1)=phi_n2dot(i)+(inv_phi_n2dot+phi_n2dot(i)-
tan(phi_n2dot(i)))/tan(phi_n2dot(i))^2;

if  abs(phi_n2dot(i+1)-phi_n2dot(i))>1e-6

  phi_n2dot(i)=phi_n2dot(i+1);
```

```
else
    phi_n2dot_final=phi_n2dot(i+1);
end
end
r_2dot_n= r_n*cos(phi_n)/cos(phi_n2dot_final); % The generating pitch radius
of virtualspur gear, (eq. 5.51 in AGMA 908-B89)
r_2dot_no= r_no*cos(phi_n)/cos(phi_n2dot_final); % The generating pitch radius
of virtualspur tool, (eq. 5.52 in AGMA 908-B89)

%% Algorithm for determing the critical point

alfa_n=pi/4; % Angle of surface normal
for i=1:10
mu_no= acos(r_2dot_no*cos(alfa_n)/r_no_s)-alfa_n; % (eq. 5.53inv  in AGMA 908-
B89)

K_S=r_2dot_no*sin(alfa_n)-r_no_s*sin(alfa_n+mu_no);

K_F=K_S-rho_ao; %(eq. 5.53  in AGMA 908-B89)

theta_no=mu_no-lamda_ns/2+pi/n_o;
theta_n=n_o/n*theta_no;
beta_n=alfa_n-theta_n;

eps_nF=r_2dot_n*sin(theta_n)+K_F*cos(beta_n);
eta_nF=r_2dot_n*cos(theta_n)+K_F*sin(beta_n);
h_F=r_nL-eta_nF; % Height of Lewis parabola (eq. 5.62 in AGMA 908-B89)

y=2*h_F*tan(beta_n)-eps_nF;

k1=(2*h_F/cos(beta_n));
k2=K_F*sin(beta_n);
k3=n_o/n*((r_2dot_no*sin(alfa_n)/(r_no_s*sin(alfa_n+mu_no))-1));
k4=(2*eps_nF*tan(beta_n)-eta_nF-2*h_F/cos(beta_n)^2);
k5=r_2dot_no*(cos(alfa_n)-sin(alfa_n)/tan(alfa_n+mu_no));
k6=((1+sin(beta_n)^2)/cos(beta_n));
y_dott=k1-k2+k3*k4-k5*k6;

alfa_n=alfa_n-y/y_dott;
alfa_n1(i)=alfa_n;
end

sF=2*eps_nF;

%% Radius of curvature of root fillet

rho_F=rho_ao+(r_2dot_no-r_no_s)^2/((r_2dot_n*r_2dot_no/(r_2dot_n+r_2dot_no))-
(r_2dot_no-r_no_s));

%% Helical factor
Ch=1 ; % Helical factor, for spur =1 (eq. 5.69 in AGMA 908-B89)

%%  Helix angle factor
K_psi=cos(psi_r)*cos(psi);
```

```
%% Tooth form factor
Y=K_psi/((cos(phi_nL)/cos(phi_nr))*((6*h_F/(sF^2*Ch))-(tan(phi_nL)/sF)));

%% sress correction factor

H=0.331-0.436*phi_n; %(eq. 5.74 in AGMA 908-B89)
L=0.324-0.492*phi_n; %(eq. 5.75 in AGMA 908-B89)
M=0.261+0.545*phi_n; %(eq. 5.76 in AGMA 908-B89)
Kf=H+(sF/rho_F)^L*(sF/h_F)^M; % (eq. 5.72 in AGMA 908-B89)

%%  Helical overlap factor
C_psi=1;  % (eq. 4.11 in AGMA 908-B89)

%% Bending strength geometry factor
Y_J=Y*C_psi/(Kf*m_N);
end
```

F.6 Geometry factor bending pinion

```
function Y_J=Geometry_factor_bending_p(n1,n2,x)
%AGMA 908-B89 Bending strength geometry factor J

%% Basic gear gometry (dimensionless)
mG=n2/n1; % Gear ratio
phi_n=20*pi/180; % standard normal pressure angle in rad
psi=0; % Standard helix angle

R1=n1/(2*cos(psi)); % Standard (reference) pitch radius  R1 and R2 (eq. 3.2
and 3.3 in AGMA 908-B89)
R2=R1*mG;

phi=atan(tan(phi_n)/cos(psi)); % Standard transverse pressure angle (eq. 3.4
in AGMA 908-B89)

Rb1=R1*cos(phi); % Pinion base radius (eq. 3.5 in AGMA 908-B89)
Rb2=Rb1*mG; % Gear base radius (eq. 3.6 in AGMA 908-B89)

Ro1=((n1+2)/2)+x;
Ro2=((n2+2)/2)-x;

Pb=2*pi*Rb1/n1; % Transverse base pitch (eq. 3.8 in AGMA 908-B89)
PN=pi*cos(phi_n); % Normal base pitch (eq. 3.9 in AGMA 908-B89)

Cr=(n1+n2)/2; % Operating center distance

phi_r= acos((Rb2+Rb1)/Cr); % Opperating transverse pressure angle (eq. 3.7 in
AGMA 908-B89)
psi_b= acos(PN/Pb); % Base helix angle (eq. 3.10 in AGMA 908-B89)

m_N=1;

psi_r=atan(tan(psi_b)/cos(phi_r)); % Opperating Helix angle (eq. 3.27 in AGMA
908-B89)
phi_nr=asin(cos(psi_b)*sin(phi_r)); % Operating Normal Pressure angle (eq.
3.28 in AGMA 908-B89)

C6=Cr*sin(phi_r); %10,6055 (eq. 3.11 in AGMA 908-B89)
C3=C6/(mG+1);
C5=(Ro1^2-Rb1^2)^0.5; % (eq. 3.15 in AGMA 908-B89)
C2=C5-Pb; % (eq. 3.16 in AGMA 908-B89)
C1=(C6-(Ro2^2-Rb2^2)^0.5); % (eq. 3.13 in AGMA 908-B89)
C4=C1+Pb;

T1_g=(C6-(Ro1^2-Rb1^2)^0.5);
T3_g=C6-C3;
T5_g=(Ro2^2-Rb2^2)^0.5;
T2_g=T5_g-Pb;
T4_g=T1_g+Pb;
Z=C5-C1; % (eq. 3.17 in AGMA 908-B89)

%% Virtual spur gear
n=n1/cos(psi)^3; % Virtual tooth number,n (eq. 5.2 in AGMA 908-B89)
```

```
r_n=n/2; % Standard (reference) pitch radius of virtual spur gear, (eq. 5.3 in
AGMA 908-B89)
r_nb=Rb1; % Virtual base radius
r_na= Ro1; % Virtual outside radius

%% Pressure angle at the Load application point
phi_nW=atan(C4/r_nb);
%% Generating Rack shift coefficient
P_nd=1; % Normal diametral pitch
x=x; % Addendum modification factor
delta_sn=0.024/P_nd; % Amount gear tooth is thinned for backlash
x_g=x-(delta_sn)/(2*tan(phi_n)); % Generating Rack Shift Coefficient (eq. 5.19
in AGMA 908-B89)
sn=pi/2+2*x_g*tan(phi_n); % normal circular tooth thickness measured on the
Standard (reference) pitch cylinder (eq. 5.21 in AGMA 908-B89)

%% Load angle and load radius

inv_phi_n=tan(phi_n)-phi_n; % (eq. 5.24 in AGMA 908-B89)
inv_phi_np=inv_phi_n+sn/n ; % (eq. 5.23 in AGMA 908-B89)
phi_nL=tan(phi_nW)-inv_phi_np; % Load angle (eq. 5.22 in AGMA 908-B89)

r_nL=r_nb/cos(phi_nL); %(eq. 5.28 in AGMA 908-B89)

%% Tool Geometry

n_c=10000; % Tool tooth number
n_o=n_c/cos(psi)^3; % Virtual tooth number of tool (eq. 5.29 in AGMA 908-B89)
r_no=n_o/2;
r_nbo=r_no*cos(phi);
R_oc=r_no+1.25;

s_no=pi/2;
x_o=(s_no-pi/2)/(2*tan(phi_n));  %Addendum modification factor of the tool
x_o=0 when s_no=pi/2 (eq. 5.35 in AGMA 908-B89)
rho_ao=0.38;
h_ao=R_oc-r_no;
r_no_s=r_no+h_ao-rho_ao; % Radius to center "S" of tool tip radius, (eq. 5.39
in AGMA 908-B89)

phi_ns=acos(r_nbo/r_no_s); % Pressure angle at point "S" on tool (eq. 5.40 in
AGMA 908-B89)
inv_phi_ns=tan(phi_ns)-phi_ns; % (eq. 5.41 in AGMA 908-B89)
inv_phi_npo=inv_phi_n+s_no/(2*r_no);
lamda_ns=(inv_phi_npo-inv_phi_ns-rho_ao/r_nbo)*2;  % Angle to center "S" of
tool tip radius (eq. 5.47 in AGMA 908-B89)

%% Generating Pressure angle

inv_phi_n2dot=inv_phi_n+2*(x_g+x_o)*tan(phi_n)/(n+n_o);
phi_n2dot(1)=(3*inv_phi_n2dot)^0.33; % First trial value ,(eq. 5.49 in AGMA
908-B89)
for i=1:100
```

```
phi_n2dot(i+1)=phi_n2dot(i)+(inv_phi_n2dot+phi_n2dot(i)-
tan(phi_n2dot(i)))/tan(phi_n2dot(i))^2;

if  abs(phi_n2dot(i+1)-phi_n2dot(i))>1e-6

  phi_n2dot(i)=phi_n2dot(i+1);
else
    phi_n2dot_final=phi_n2dot(i+1);
end
end
r_2dot_n= r_n*cos(phi_n)/cos(phi_n2dot_final); % The generating pitch radius
of virtualspur gear, (eq. 5.51 in AGMA 908-B89)
r_2dot_no= r_no*cos(phi_n)/cos(phi_n2dot_final); % The generating pitch radius
of virtualspur tool, (eq. 5.52 in AGMA 908-B89)

%% Algorithm for determing the critical point

alfa_n=pi/4; % Angle of surface normal
for i=1:10
mu_no= acos(r_2dot_no*cos(alfa_n)/r_no_s)-alfa_n; % (eq. 5.53inv  in AGMA 908-
B89)

K_S=r_2dot_no*sin(alfa_n)-r_no_s*sin(alfa_n+mu_no);

K_F=K_S-rho_ao; %(eq. 5.53  in AGMA 908-B89)

theta_no=mu_no-lamda_ns/2+pi/n_o;
theta_n=n_o/n*theta_no;
beta_n=alfa_n-theta_n;

eps_nF=r_2dot_n*sin(theta_n)+K_F*cos(beta_n);
eta_nF=r_2dot_n*cos(theta_n)+K_F*sin(beta_n);
h_F=r_nL-eta_nF; % Height of Lewis parabola (eq. 5.62 in AGMA 908-B89)

y=2*h_F*tan(beta_n)-eps_nF;

k1=(2*h_F/cos(beta_n));
k2=K_F*sin(beta_n);
k3=n_o/n1*((r_2dot_no*sin(alfa_n)/(r_no_s*sin(alfa_n+mu_no))-1));
k4=(2*eps_nF*tan(beta_n)-eta_nF-2*h_F/cos(beta_n)^2);
k5=r_2dot_no*(cos(alfa_n)-sin(alfa_n)/tan(alfa_n+mu_no));
k6=((1+sin(beta_n)^2)/cos(beta_n));
y_dott=k1-k2+k3*k4-k5*k6;

alfa_n=alfa_n-y/y_dott;
alfa_n1(i)=alfa_n;
end

sF=2*eps_nF;

%% Radius of curvature of root fillet
rho_F=rho_ao+(r_2dot_no-r_no_s)^2/((r_2dot_n*r_2dot_no/(r_2dot_n+r_2dot_no))-
(r_2dot_no-r_no_s));
```

```
%% Helical factor
Ch=1 ; % Helical factor, for spur =1 (eq. 5.69 in AGMA 908-B89)

%%  Helix angle factor
K_psi=cos(psi_r)*cos(psi);

%% Tooth form factor
Y=K_psi/((cos(phi_nL)/cos(phi_nr))*((6*h_F/(sF^2*Ch))-(tan(phi_nL)/sF)));

%% sress correction factor

H=0.331-0.436*phi_n; %(eq. 5.74 in AGMA 908-B89)
L=0.324-0.492*phi_n; %(eq. 5.75 in AGMA 908-B89)
M=0.261+0.545*phi_n; %(eq. 5.76 in AGMA 908-B89)
Kf=H+(sF/rho_F)^L*(sF/h_F)^M; % (eq. 5.72 in AGMA 908-B89)

%%  Helical overlap factor
C_psi=1;  % (eq. 4.11 in AGMA 908-B89)

%% Bending strength geometry factor
Y_J=Y*C_psi/(Kf*m_N);
end
```

F.7 Geometry factor pitting

```
function Z_I=Geometry_factor_pitting(n1,n2,gear_type,x)
% gear type =1 for external gear and =0 for internal.

%% Basic gear gometry (dimensionless)
mG=n2/n1; % Gear ratio
phi_n=20*pi/180; % standard normal pressure angle in rad
psi=0; % Standard helix angle

R1=n1/(2*cos(psi)); % Standard (reference) pitch radius  R1 and R2 (eq. 3.2
and 3.3 in AGMA 908-B89)
R2=R1*mG;

phi=atan(tan(phi_n)/cos(psi)); % Standard transverse pressure angle (eq. 3.4
in AGMA 908-B89)

Rb1=R1*cos(phi); % Pinion base radius (eq. 3.5 in AGMA 908-B89)
Rb2=Rb1*mG; % Gear base radius (eq. 3.6 in AGMA 908-B89)

Ro1=(n1+2)/2+x;
if gear_type==1
Ro2=(n2+2)/2-x;
elseif gear_type==0
Ro2=(n2+2)/2-x;
end

Pb=2*pi*Rb1/n1; % Transverse base pitch (eq. 3.8 in AGMA 908-B89)
PN=pi*cos(phi_n); % Normal base pitch (eq. 3.9 in AGMA 908-B89)

if gear_type==1
Cr=(n1+n2)/2; % Operating center distance for external gears
elseif gear_type==0
Cr=(n2-n1)/2; % Operating center distance for internal gears
end

if gear_type==1
phi_r= acos((Rb2+Rb1)/Cr); % operating transverse pressure angle
elseif gear_type==0
phi_r= acos((Rb2-Rb1)/Cr);
end

C6=Cr*sin(phi_r); % (eq. 3.11 in AGMA 908-B89)
C5=(Ro1^2-Rb1^2)^0.5; % (eq. 3.15 in AGMA 908-B89)
C2=C5-Pb; % (eq. 3.16 in AGMA 908-B89)

if gear_type==1
C1=(C6-(Ro2^2-Rb2^2)^0.50); % (eq. 3.13 in AGMA 908-B89)
elseif gear_type==0
C1=-(C6-(Ro2^2-Rb2^2)^0.50);
end

Z=C5-C1; % (eq. 3.17 in AGMA 908-B89)
mp=Z/Pb; % (eq. 3.18 in AGMA 908-B89)
if mp<2
 mN=1; % (eq. 3.25 in AGMA 908-B89)
end
```

```
%% Operating pitch diameter of pinion
if gear_type==1
d=2*Cr/(mG+1);     %   (eq. 4.2 in AGMA 908-B89)
elseif gear_type==0
d=2*Cr/(mG-1);
end

%% Spur and Low Axial contact ratio helical gears
rho1=C2; % (eq. 4.6 in AGMA 908-B89)
if gear_type == 1
rho2=C6-rho1; % (eq. 4.7 in AGMA 908-B89)
elseif gear_type== 0
rho2=C6+rho1;
end

%% Helical overlap factor
C_psi=1; % for spur gear (eq. 4.11 in AGMA 908-B89)

%% Geometry factor for pitting resistance
if gear_type==1
Z_I= cos(phi_r)*C_psi^2/((1/rho1+1/rho2)*d*mN); %(eq. 4.1 in AGMA 908-B89)
elseif gear_type==0
Z_I= cos(phi_r)*C_psi^2/((1/rho1-1/rho2)*d*mN);
end

end
```

F.8 Allowable bending stress

```
function S_F = allowable_bending_stress_AGMA(s_F,L,omega,n_planets,s_FP,i)

% i=1 for planet and i=0 for sun

q=n_planets;
n_L=60*L*omega*q;
Y_N= 1.3558*n_L^(-0.0178);

%Temperature factor
Y_omeg= 1;
%Reliability factor
Y_Z= 1;

% Reverse loading factor
if i==1
Y_a=0.7;
elseif i==0
Y_a=1;
end

%Safty factor for pitting(flank safety)
S_F=s_FP*Y_N*Y_a/(s_F*Y_omeg*Y_Z);
end
```

F.9 Allowable contact stress

```
function S_H = allowable_contact_stress_AGMA(s_H,L,omega,n_planets,s_HP)

% Allowable contact stress number (N/mm^2)
%s_HP= 1551 ; % Carburized and hardened, grade 2

% Stress cycle factor for pitting resistance
% L= life(hours)
% omega= speed(rpm)
% q= number of contacts per revolution.

q=n_planets;
n_L=60*L*omega*q;
Z_N= 1.4488*n_L^(-0.023);

%Hardness ratio factor for pitting resistance
Z_W=1 ;

%Temperature factor
Y_omeg= 1;

%Reliability factor
Y_Z= 1;

%Safty factor for pitting(flank safety)
S_H=s_HP*Z_N*Z_W/(s_H*Y_omeg*Y_Z);

end
```

F.10 Forces planet shaft

```
z1=45;        % Number of teeths
z2=14;
z3=21;
z4=z1+z2+z3;
mg=1+((z4*z2)/(z3*z1)); % Reduction ratio
Tin=6950/mg; %Input torque
N_planets=5; % Number of planets
m=2.5/1000;  % module
P_angle=20;  % Pressure angle
d1=m*z1;  % pitch diameter
d2=m*z2;
d3=m*z3;
d4=m*z4;
r1=d1/2; % Pitch radius
r2=d2/2;
r3=d3/2;
r4=d4/2;

F1_2t = (Tin/r1)/N_planets; % Tangential Force acting between sun and planet
1. positive
F1_2r = F1_2t*tand(P_angle);  % Radial force  negative
F3_4t = ((F1_2t*r2)/r3);     % Tangential Force acting between ring and planet
2. positive
F3_4r = F3_4t*tand(P_angle);   % Radial force positive
Fout_t=F1_2t+F3_4t;     % Max tangential load
Fout_t20=0.2*(F1_2t+F3_4t); % Dynamic_load 20% of max
b_p1=0.034;  % width of planet 1  [m]
b_p2=0.017;  % width of planet 2 [m]
washer_space=0.001; % space between planet and carrier
space=0.003; % space between planet1 and 2  [m]
shaft_supp=0.010; % How much of the shaft that goes into the carrier[m]

ratio_t=F3_4t/F1_2t; % ratio between tangential forces
ratio_r=F3_4r/F1_2r; % ratio between radial forces
Fout_r=F1_2r-F3_4r; % Max radial load
Fout_r20=0.2*(F1_2r-F3_4r);% Dynamic_load 20% of max
Dr=shaft_supp+washer_space+b_p1/2+(1-ratio_r)*(b_p1/2+space+b_p2/2); % Where
the total radial force will act on the shaft [mm]
Dt=shaft_supp+washer_space+b_p1/2+(1-ratio_t)*(b_p1/2+space+b_p2/2); % Where
the total tangential force will act on the shaft [mm]

% distance between where the forces act
L1=washer_space+b_p2/2;
L2=space+b_p1/2+b_p2/2;
L3=washer_space+b_p1/2;
L=L1+L2+L3+shaft_supp*2; % Planet Shaft lenght

Fo_t = (F3_4t*(L1)+(F1_2t*L1+L2))/(L1+L2+L3); % Tangential Force acting at
outer mounting point of the planet shaft.
Fo_r = Fo_t*tand(P_angle); % Radial outer force
Fi_t = F1_2t+F3_4t-Fo_t;  % Tangential Force acting at inner mounting point of
the planet shaft.
Fi_r = Fi_t*tand(P_angle); %Radial inner force
```

G

Load distribution factors

The factors for calculating the Load distribution factor are presented and explained in following text and is based on ANSI/AGMA 2101-C95[4].The is calculated according Eq. G.1 as mentioned in section 2.7.3.1

$$K_H = K_{H\beta} = 1 + K_{Hmc}(K_{Hpf}K_{Hpm} + K_{Hma}K_{He}) \tag{G.1}$$

where
K_{Hmc} = lead correction factor
K_{Hpf} = pinion proportion factor
K_{Hpm} = pinion proportion modifier
K_{Hma} = mesh alignment factor
K_{He} = mesh alignment correction factor

The **lead correction factor** is used to modified the peak load when the gear has crowning or lead modification is applied:
K_{Hmc} = 1.0 for gear with unmodified leads
K_{Hmc} = 0.8 for gear with leads properly modified by crowning or lead correction.

The pinion proportion factor is used to account for deflections due to load. The deflections are usually higher for gears with high aspect ratio(wide gears). The value for K_{Hpf} is determined by following equations
when $b \leq 25$

$$K_{Hpf} = \frac{b}{(10)d_{w1}} - 0.025 \tag{G.2}$$

when $25 \leq b \leq 432$

$$K_{Hpf} = \frac{b}{(10)d_{w1}} - 0.0375 + 0.00049b \tag{G.3}$$

when $432 \leq b \leq 1020$

$$K_{Hpf} = \frac{b}{(10)d_{w1}} - 0.1109 + 0.000492b - 0.000000353b^2 \tag{G.4}$$

The pinion proportion modifier K_{Hpm} alters the pinion proportion factor,K_{Hpf}, with based on the pinion location relative to the bearing.

$K_{Hpm} = 1.0$ for straddle mounted pinion with $(S_1/S) < 0.175$
$K_{Hpm} = 1.1$ for straddle mounted pinion with $(S_1/S) \geq 0.175$ where
S_1 is the offset of the pinion(see Fig.G.1). S is the bearing span (see Fig.G.1).

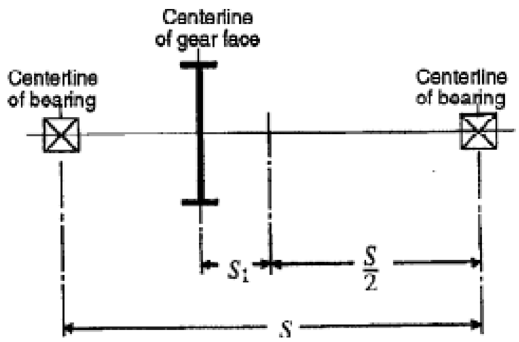


Figure G.1: Evaluation of S and S_1[4]

The mesh alignment factor, K_{Hma}, is intended to account for misalignment of the axes of rotation of the pitch cylinders of the mating gear elements from all causes other than elastic deformation. K_{Hma} is defined as

$$K_{Hma} = A + B(b) + C(b)^2 \tag{G.5}$$

Values for A,B and C can be seen in table **??**

Curve	**A**	**B**	**C**
Curve 1 Open gearing	$2.47\text{x}10^{-1}$	$0.657\text{x}10^{-3}$	$-1.186\text{x}10^{-7}$
Curve 2 Commercial enclosed gear unit	$1.27\text{x}10^{-1}$	$0.622\text{x}10^{-3}$	$-1.69\text{x}10^{-7}$
Curve 3 Precision enclosed gear unit	$0.675\text{x}10^{-1}$	$0.504\text{x}10^{-3}$	$-1.44\text{x}10^{-7}$
Curve 4 Extra precision enclosed gear unit	$0.380\text{x}10^{-1}$	$0.402\text{x}10^{-3}$	$-1.27\text{x}10^{-7}$

Table G.1: Empirical constants, A, B and C[4]

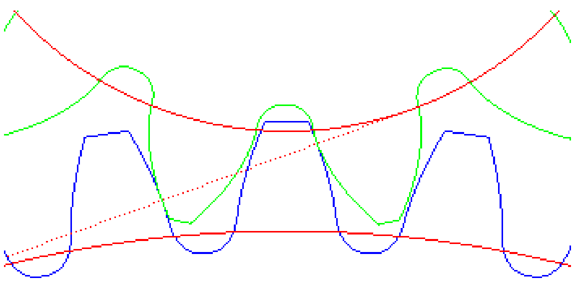
The mesh alignment correction factor,K_{He}, is used if the assembly or manufacturing techniques improve the effective mesh alignment. Suggested values for K_{He} are as follows

$K_{He} = 0.8$ when the gearing is adjusted at assembly.
$K_{He} = 0.8$ when lapping is used to improve the gearing compatibility .
$K_{He} = 1$ for everything else .

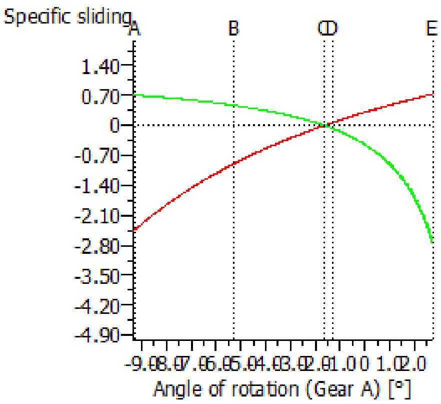
H

KISSsoft results

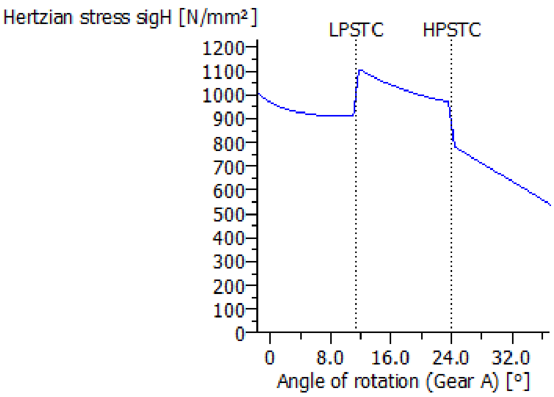
H.1 Stage 1



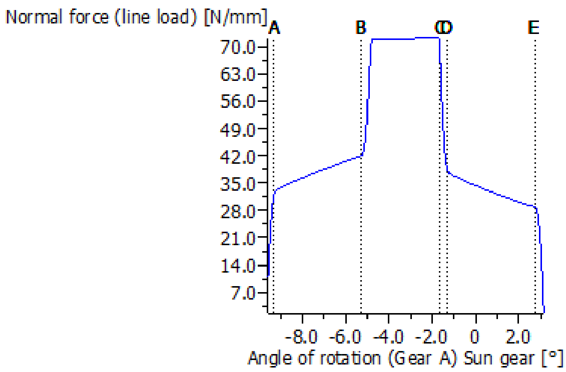
(a) Meshing between sun and planet 1



(b) Specific sliding for sun and planet 1



(c) Hertzian pressure for sun and planet 1



(d) Normal force for sun and planet 1

Figure H.1: Graphical results from KISSsoft for stage 1

KISSsoft
Calculation programs for machine design

KISSsoft Release 03/2016 E

KISSsoft evaluation

File

Name : stage1
Changed by: siguðursvavar on: 14.05.2017 at: 20:23:17

Important hint: At least one warning has occurred during the calculation:

1-> According AGMA 908-B89:
The tooth form factors Kf and J for internal toothing or racks are not defined in AGMA 908.
Thus the root strength of internal toothing (or racks) cannot be calculated!

Remark:
The form factors may be determined by using the graphical method (see corresponding option in dialog 'Define details of strength').

CALCULATION OF A SPUR PLANETARY GEAR STAGE

Drawing or article number:
Gear 1: 0.000.0
Gear 2: 0.000.0
Gear 3: 0.000.0

Calculation method AGMA 2001-C95

		Sun	Planets	Internal gear
Number of planets	[p]	(1)	5	(1)
Power (kW)	[P]		42.025	
Transmitted power (hp)	[P]		56.356	
Transmitted power (ft*lb/s)	[P]		30996.0	
Speed (1/min)	[n]	630.0		0.0
Speed difference for planet bearing calculation (1/min)	[n2]		1265.6	
Speed planet carrier (1/min)	[nSteg]		236.2	
Torque (Nm)	[T]	637.0	0.0	1061.7
Torque Pl.-Carrier (Nm)	[TSteg]		1698.667	

Overload factor	[Ko]	1.35
Power distribution factor	[Kgam]	1.30
	[KAeff = KA*Kgam]	1.76
Required service life (h)	[H]	5000.00

Gear driving (+) / driven (-) + -/+ -
Working flank gear 1: Right flank
Sense of rotation gear 1 clockwise
Gearbox type: Precision gearing in closed housing

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

		GEAR 1	GEAR 2	GEAR 3
Center distance (in, mm)	[a]	2.9035,	73.750	

Centre distance tolerance ISO 286:2010 Measure js7

Normal Diametral Pitch (1/in)	[Pnd]	10.16000		
Normal module (in, mm)	[mn]	0.09843, 2.5000		
Pressure angle at normal section (°)	[alfn]	20.0000		
Helix angle at reference circle (°)	[beta]	0.0000		
Number of teeth	[z]	45	14	-75
Facewidth (mm)	[b]	34.00	34.00	34.00
Hand of gear	Spur gear			

Planetary axles can be placed in regular pitch.: 72°

Accuracy grade	[Q-AGMA2000-A88]	11	11	11
Inner diameter (mm)	[di]	0.00	0.00	
External diameter (mm)	[di]			0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00	
Outer diameter of gear rim (mm)	[dbi]			0.00

Material

Gear 1: Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened
AGMA 2001-C95

Gear 2: Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened
AGMA 2001-C95

Gear 3: Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened
AGMA 2001-C95

		------- GEAR 1 ------------	GEAR 2 ------------	GEAR 3 ---
Surface hardness		HRC 60	HRC 60	HRC 60
		(lb/in²), (N/mm²)	(lb/in²), (N/mm²)	(lb/in²), (N/mm²)
Allowable bending stress number	[sat]	65000, 448.2	65000, 448.2	65000, 448.2
Allowable contact stress number	[sac]	225000, 1551.3	225000, 1551.3	225000, 1551.3
Tensile strength (N/mm²)	[σB]	966.00	966.00	966.00
Yield point (N/mm²)	[σS]	822.00	822.00	822.00
Young's modulus (N/mm²)	[E]	206842	206842	206842
Poisson's ratio	[v]	0.300	0.300	0.300
Roughness average value DS, flank (µm)	[RAH]	0.63	0.63	0.63
Roughness average value DS, root (µm)	[RAF]	2.40	2.40	2.40
Mean roughness height, Rz, flank (µm)	[RZH]	5.00	5.00	5.00
Mean roughness height, Rz, root (µm)	[RZF]	16.00	16.00	16.00

Gear reference profile 1 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Gear reference profile 2 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000

Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Gear reference profile 3 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250	1.250
Tooth root radius Refer. profile	[rofP*]	0.380	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (μm)	[Ca]	1.80	1.80	1.80

Lubrication type		Oil bath lubrication
Type of oil		Oil: ISO-VG 220
Lubricant base		Mineral-oil base
Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50
Specific density at 15 °C (kg/dm³)	[roOil]	0.895
Oil temperature (°C)	[TS]	70.000

		------- GEAR 1 ------------	GEAR 2 ------------	GEAR 3 ---
Overall transmission ratio	[itot]	2.667		
Gear ratio	[u]	0.311	-5.357	
Transverse module (mm)	[mt]	2.500		
Pressure angle at pitch circle (°)	[alft]	20.000		
Working transverse pressure angle (°)	[alfwt]	20.000	13.701	
	[alfwt.e/i]	20.032 / 19.968	13.653 / 13.748	
Working pressure angle at normal section (°)	[alfwn]	20.000	13.701	
Helix angle at operating pitch circle (°)	[betaw]	0.000	0.000	
Base helix angle (°)	[betab]	0.000		
Reference centre distance (mm)	[ad]	73.750	76.250	
Sum of profile shift coefficients	[Summexi]	0.0000	0.8581	
Profile shift coefficient	[x]	-0.4000	0.4000	0.4581
Tooth thickness (Arc) (module) (module)	[sn*]	1.2796	1.8620	1.9043
Tip alteration (mm)	[k*mn]	0.000	0.000	-0.150
Reference diameter (mm)	[d]	112.500	35.000	187.500
Base diameter (mm)	[db]	105.715	32.889	176.192
Tip diameter (mm)	[da]	115.500	42.000	180.509
(mm)	[da.e/i]	115.500 / 115.490	42.000 / 41.990	180.509 / 180.519

Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010	-0.000 / 0.010
Tip form diameter (mm)	[dFa]	115.500	42.000	180.509
(mm)	[dFa.e/i]	115.500 / 115.490	42.000 / 41.990	180.509 / 180.519
Active tip diameter (mm)	[dNa.e/i]	115.500 / 115.490	42.000 / 41.990	180.509 / 180.519
Operating pitch diameter (mm)	[dw]	112.500	35.000 / 33.852	181.352
(mm)	[dw.e]	112.523	35.007 / 33.846	181.316
(mm)	[dw.i]	112.477	34.993 / 33.859	181.389
Root diameter (mm)	[df]	104.250	30.750	191.459
Generating Profile shift coefficient	[xE.e/i]	-0.4330 / -0.4659	0.3758 / 0.3483	0.4141 / 0.3702
Manufactured root diameter with xE (mm)	[df.e]	104.085	30.629	191.679
(mm)	[df.i]	103.920	30.492	191.899
Theoretical tip clearance (mm)	[c]	0.625	0.625/ 0.980	1.130
Tip clearance upper allowance (mm)	[c.e]	0.774	0.810/ 1.220	1.279
Tip clearance lower allowance (mm)	[c.i]	0.670	0.692/ 1.075	1.175
Active root diameter (mm)	[dNf]	108.478	33.122/ 33.170	186.471
(mm)	[dNf.e]	108.502	33.136/ 33.193	186.425
(mm)	[dNf.i]	108.459	33.112/ 33.154	186.513
Root form diameter (mm)	[dFf]	107.239	33.045	189.312
(mm)	[dFf.e/i]	107.159 / 107.081	33.012 / 32.980	189.644 / 189.965
Internal toothing: Calculation dFf with pinion type cutter (z0= 24, x0= 0.000)				
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.710 / 0.650	0.078 / 0.050	1.770 / 1.566
Addendum (mm)	[ha = mn * (haP*+x)]	1.500	3.500	3.495
(mm)	[ha.e/i]	1.500 / 1.495	3.500 / 3.495	3.495 / 3.490
Dedendum (mm)	[hf = mn * (hfP*-x)]	4.125	2.125	1.980
(mm)	[hf.e/i]	4.207 / 4.290	2.185 / 2.254	2.090 / 2.200
Roll angle at dFa (°)	[xsi_dFa.e/i]	25.215 / 25.202	45.505 / 45.477	12.761 / 12.776
Roll angle to dNf (°)	[xsi_dNf.e/i]	13.241 / 13.137	7.032 / 6.683	
	[xsi_dNf.e/i]		7.804 / 7.282	19.808 / 19.896
Roll angle at dFf (°)	[xsi_dFf.e/i]	9.500 / 9.239	4.959 / 4.259	22.812 / 23.093
Tooth height (mm)	[h]	5.625	5.625	5.475
Virtual gear no. of teeth	[zn]	45.000	14.000	-75.000
Normal tooth thickness at tip circle (mm)	[san]	2.060	1.046	2.538
(mm)	[san.e/i]	2.002 / 1.936	1.000 / 0.933	2.464 / 2.384
Normal space width at root circle (mm)	[efn]	0.000	0.000	1.578
(mm)	[efn.e/i]	0.000 / 0.000	0.000 / 0.000	1.567 / 1.556
Max. sliding velocity at tip (m/s)	[vga]	0.699	1.229/ 0.976	0.879
Specific sliding at the tip	[zetaa]	0.729	0.710/ 0.564	0.412
Specific sliding at the root	[zetaf]	-2.451	-2.689/ -0.701	-1.292
Sliding factor on tip	[Kga]	0.301	0.530/ 0.435	0.089
Sliding factor on root	[Kgf]	-0.530	-0.301/ -0.089	-0.435
Pitch on reference circle (mm)	[pt]	7.854		
Base pitch (mm)	[pbt]	7.380		
Transverse pitch on contact-path (mm)	[pet]	7.380		
Length of path of contact (mm)	[ga]	11.099	10.907	
(mm)	[ga.e/i]	11.142 / 11.034	10.970 / 10.813	
Length T1-A (mm)	[T1A]	12.164	13.060/ 2.153	19.621
Length T1-B (mm)	[T1B]	15.882	9.342/ 5.680	23.148
Length T1-C (mm)	[T1C]	19.239	5.985/ 4.009	21.477
Length T1-D (mm)	[T1D]	19.544	5.680/ 9.534	27.001
Length T1-E (mm)	[T1E]	23.262	1.962/ 13.060	30.528
Diameter of single contact point B (mm)	[d-B]	110.384	37.826/ 34.796	182.173
(mm)	[d-B.e]	110.384	37.783/ 34.796	182.205
(mm)	[d-B.i]	110.377	37.882/ 34.791	182.137

Diameter of single contact point D (mm)	[d-D]	112.710	34.796/ 38.017	184.282
(mm)	[d-D.e]	112.680	34.796/ 37.953	184.282
(mm)	[d-D.i]	112.746	34.791/ 38.104	184.296

Transverse contact ratio	[eps_a]	1.504	1.478
Transverse contact ratio with allowances	[eps_a.e/i]	1.510 / 1.495	1.486 / 1.465
Overlap ratio	[eps_b]	0.000	0.000
Total contact ratio	[eps_g]	1.504	1.478
Total contact ratio with allowances	[eps_g.e/i]	1.510 / 1.495	1.486 / 1.465

2. FACTORS OF GENERAL INFLUENCE

------- Gear 1 ------------ Gear 2 ------------ Gear 3 ---

Calculated with the operating pitch circle:					
Nominal circumferential force (N)	[Ftw]	2264.889	2341.665		
Axial force (N)	[Fa]	0.0	0.0	0.0	
Axial force (total) (N)	[Fatot=Fa*	5]		0.0	0.0
Radial force (N)	[Fr]	824.352	570.863		
Net face width of narrowest member (in)	[F]	1.34 (34.00 mm)	1.34 (34.00 mm)		
Nominal force at operating pitch dia. (lb)	[Wt]	508.92 (2264.89 N)	526.17 (2341.66 N)		
Pitch line velocity (ft/min)	[vt]	456.57 (2.32 m/s)	441.60 (2.24 m/s)		
Gear unit type: Precision enclosed gear unit					
Mesh alignment factor	[Cma]	0.084	0.084		
Mounting procedure: Without contact improvement at assembly					
Mesh alignment correction factor	[Ce]	1.000			
Gearing: without longitudinal flank modification					
Lead correction factor	[Cmc]	1.000	1.000		
Pinion proportion factor	[Cpf]	0.076	0.080		
Pinion proportion modifier	[Cpm]	1.000	1.000		
Small offset [s1/s < 0.175]					
Face load distribution factor	[Cmf]	1.161	1.164		
Load distribution factor	[Km]	1.161	1.164		
Transmission accuracy number introduced:					
Transmission accuracy grade number	[Qv]	11			
(VpA replaced by fpe as defined in ISO or DIN)					
Dynamic factor	[Kv]	1.054			
Number of load cycles (in mio.)	[NL]	590.6	379.7	354.4	

3. TOOTH ROOT STRENGTH

------- GEAR 1 ------------ GEAR 2 ------------ GEAR 3 ---

Rim thickness factor	[KB]	1.000	1.000	1.000
Size factor	[KS]	1.000	1.000	0.000
Limiting Variation in action (in/10000)	[LimVarAc]	4.0	4.0	
Load sharing:				
0 = No (Loaded at tip) 1 = Yes (Loaded at HPSTC)		1	0	
Calc. as helical gear (0) / as LACR (1)		0	0/ 0	0
Load angle (°)	[phinL]	18.75	24.19/ 24.86	0.00
Calculation of factor Y following	AGMA 908-B89			
Heigth of Lewis parabola (in)	[hF]	0.119	0.099/ 0.103	0.000
Heigth of Lewis parabola (mm)	[hF]	3.027	2.524/ 2.605	0.000
Tooth thickness at critical section (in)	[sF]	0.187	0.200/ 0.199	0.000
Tooth thickness at critical section (mm)	[sF]	4.745	5.073/ 5.058	0.000
Radius at curvature of fillet curve (in)	[roF]	0.045	0.041/ 0.041	0.000
Radius at curvature of fillet curve (mm)	[roF]	1.133	1.036/ 1.036	0.000

Helical factor	[Ch]	1.00	1.00		
Helix angle factor	[Kpsi]	1.00	1.00		
Tooth form factor Y	[Y]	0.540	0.824/ 0.825	0.000	
Stress correction factor	[Kf]	1.698	1.917/ 1.890	0.000	
Bending strength geometry factor J	[J]	0.318	0.430/ 0.436	0.000	
Bending stress number (lb/in²)	[st]	26073.1	19290.9/ 19687.8	0.0	
Bending stress number (N/mm²)	[st]	179.8	133.0/ 135.7	0.0	
Stress cycle factor (for general applications)	[YN]	0.946	0.954	0.955	
Allowable bending stress number (lb/in²)	[sat]	65000.0	65000.0	65000.0	
Allowable bending stress number (N/mm²)	[sat]	448.2	448.2	448.2	
Temperature factor	[KT]	1.000	1.000	1.000	
Reliability factor (99.00 %)	[KR]	1.000			
Reverse loading factor	[-]	1.000	0.700	1.000	
Effective allow. b.s.n. (lb/in²)	[sateff]	61514.8	43400.4/ 43400.4	62076.7	
Effective allow. b.s.n. (N/mm²)	[sateff]	424.1	299.2/ 299.2	428.0	
Bending strength power rating (hp)	[Patu]	233.4(174.01 kW)	222.5(165.93 kW)/ 218.0(162.59 kW)		0.0(0.00 kW)

Note: Bending strength power rating calculated with Kγ=1, Ko=1, KR=1, SFmin=1

Unit load (lb/in²)	[UL]	3863.32	3994.28		
Allowable unit load (lb/in²)	[Uat]	9114.8	8691.7/ 8805.1	0.0	
Required safety factor	[SFmin]	1.000	1.000	1.000	
Safety factor (Bending)	[sateff/st]	2.359	2.250/ 2.204	0.000	
Transmittable power including SFmin	[Patu/SFmin]	233.4(174.01 kW)	222.5(165.93 kW)/ 218.0(162.59 kW)		0.0(0.00 kW)

Note: Transmittable power including SFmin calculated with Kγ=1, Ko=1, KR=1

(Note: Materials with HB > 400: Yield strength not checked.)

4. SAFETY AGAINST PITTING (TOOTH FLANK)

		------- Gear 1	------------ Gear 2	------------ Gear 3 ---	
		(√lb/in), (√N/mm)	(√lb/in), (√N/mm)		
Elastic coefficient	[Cp]	2290.00, 190.20	2290.00, 190.20		
Size factor	[Ks]	1.000	1.000	1.000	
Load sharing ratio	[mN]	1.000	1.000		
Helical overlap factor	[Cpsi]	1.000	1.000		
Geometry factor I	[I]	0.118	0.216		
		(lb/in²), (N/mm²)	(lb/in²), (N/mm²)		
Contact stress number	[sc]	162178.9, 1118.2	124137.1, 855.9		
Stress cycle factor (for general applications)	[ZN]	0.910	0.920	0.921	
Surface condition factor	[Cf]	1.000	1.000/ 1.000	1.000	
Hardness ratio factor	[CH]	1.000	1.000/ 1.000	1.000	
Temperature factor	[KT]	1.000	1.000	1.000	
Reliability factor	[KR]	99.000	ZS.CRAGMA		
Allowable contact stress number (lb/in²)	[sac]	225000.0	225000.0	225000.0	
Allowable contact stress number (N/mm²)	[sac]	1551.3	1551.3	1551.3	
Effective allow. c.s.n. (lb/in²)	[saceff]	204857.0	206949.4/206949.4	207278.0	
Effective allow. c.s.n. (N/mm²)	[saceff]	1412.4	1426.9/ 1426.9	1429.1	
Pitting resistance power rating (hp)	[Pacu]	157.8(117.68 kW)	161.1(120.10 kW)/ 274.9(204.98 kW)		275.8(205.63 kW)

Note: Pitting resistance power rating calculated with Kγ=1, Ko=1, KR=1, SHmin=1

Contact load factor (lb/in²)	[K]	361.8	239.9		
Allowable contact load factor (lb/in²)	[Kac]	577.3	589.1/ 666.8	668.9	

Required safety factor	[SHmin]	1.000	1.000	1.000
Safety factor (Pitting)	[saceff/sc]	1.263	1.276/ 1.667	1.670
Transmittable power including SHmin (hp)	[Pacu/SHmin^2]	157.8(117.68 kW)	161.1(120.10 kW)/ 274.9(204.98 kW)	275.8(205.63 kW)

Note: Transmittable power including SHmin calculated with Kγ=1, Ko=1, KR=1

SERVICE FACTORS (with Kγ= 1.000):

Service factor for tooth root	[KSF]	4.141	3.948/ 3.869	0.000
Service factor for pitting	[CSF]	2.800	2.858/ 4.878	4.893
Service factor for gear set	[SF]	2.800	3.869	

SERVICE FACTORS (with Kγ= 1.300):

Service factor for tooth root	[KSF]	3.185	3.037/ 2.976	0.000
Service factor for pitting	[CSF]	2.154	2.198/ 3.752	3.764
Service factor for gear set	[SF]	2.154	2.976	

Note: Service factors calculated with Ko=1, KR=1, SFmin=1, SHmin=1

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Pairing Gear 1- 2:
Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

Pairing Gear 2- 3:
Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Results from AGMA 925-A03 (Details see in the specific calculation sheet)

Probability of wear (%)	[Pwear]	78.859
Probability of scuffing (%)	[Pscuff]	5% or lower

6. MEASUREMENTS FOR TOOTH THICKNESS

		------- GEAR 1 ------------	GEAR 2 ------------	GEAR 3 ---
Tooth thickness deviation	DIN 3967 d26	DIN 3967 d26	DIN 3967 d26	
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.060/ -0.120	-0.044/ -0.094	-0.080/ -0.160
Number of teeth spanned	[k]	5.000	3.000	-8.000
(Internal toothing: k = (Measurement gap number)				
Base tangent length (no backlash) (mm)	[Wk]	34.103	19.625	-57.195
Actual base tangent length ('span') (mm)	[Wk.e/i]	34.047/ 33.990	19.584/ 19.537	-57.270/ -57.345
Diameter of contact point (mm)	[dMWk.m]	111.054	38.266	185.278
Theoretical diameter of ball/pin (mm)	[DM]	4.132	5.160	4.009
Effective diameter of ball/pin (mm)	[DMeff]	4.250	5.250	4.000
Radial single-ball measurement backlash free (mm)	[MrK]	58.191	22.520	90.023
Radial single-ball measurement (mm)	[MrK.e/i]	58.106/ 58.020	22.484/ 22.442	90.152/ 90.279
Diameter of contact point (mm)	[dMMr.m]	110.558	36.996	185.653
Diametral measurement over two balls without clearance (mm)	[MdK]	116.314	45.041	180.005
Diametral two ball measure (mm)	[MdK.e/i]	116.144/ 115.972	44.967/ 44.883	180.264/ 180.518
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	116.144/ 115.972	44.967/ 44.883	180.264/

180.518
Measurement over 3 pins (axial) according to AGMA 2002 (mm)

	[dk3A.e/i]	116.144/ 115.972	44.967/ 44.883	180.264/ 180.518
Effective dimensions over 3 pins (mm)	[Md3R.e/i]	116.076/ 115.904	0.000/ 0.000	180.223/ 180.477
Tooth thickness (chordal) in pitch diameter (mm)	[sc]	3.199	4.641	4.760
(mm)	[sc.e/i]	3.139/ 3.079	4.597/ 4.547	4.680/ 4.600
Reference chordal height from da.m (mm)	[ha]	1.520	3.652	3.463
Tooth thickness (Arc) (mm)	[sn]	3.199	4.655	4.761
(mm)	[sn.e/i]	3.139/ 3.079	4.611/ 4.561	4.681/ 4.601
Backlash free center distance (mm)	[aControl.e/i]	73.606/ 73.451	73.990/ 74.229	
Backlash free center distance, allowances (mm)	[jta]	-0.144/ -0.299	0.240/ 0.479	
dNf.i with aControl (mm)	[dNf0.i]	108.094	32.892	187.765
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.468	-0.060	0.939
Tip clearance	[c0.i(aControl)]	0.387	0.409	0.711
Centre distance allowances (mm)	[Aa.e/i]	0.015/ -0.015	-0.015/ 0.015	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.011/ -0.011	0.007/ -0.007	
Radial clearance (mm)	[jrw]	0.314/ 0.129	0.494/ 0.225	
Circumferential backlash (transverse section) (mm)	[jtw]	0.225/ 0.093	0.253/ 0.113	
Normal backlash (mm)	[jnw]	0.211/ 0.087	0.238/ 0.106	
Entire torsional angle (°)	[j.tSys]	0.1788/ 0.0871		

(j.tSys: Torsional angle of planet carrier for blocked shaft)

7. GEAR ACCURACY

		------- GEAR 1	------------ GEAR 2	------------ GEAR 3 ---
Following AGMA 2000-A88				
Accuracy grade	[Q-AGMA2000]	11	11	11
Pitch Variation Allowable (µm)	[VpA]	7.20	5.90	7.90
Runout Radial Tolerance (µm)	[VrT]	25.00	19.00	28.00
Profile Tolerance (µm)	[VphiT]	9.10	7.60	9.90
Tooth Alignment Tolerance (µm)	[VpsiT]	8.20	8.20	8.20
Composite Tolerance, Tooth-to-Tooth (µm)	[VqT]	13.00	16.00	13.00
Composite Tolerance, Total (µm)	[VcqT]	39.00	35.00	42.00

(AGMA <-> ISO: VpA <-> fpbT, VrT <-> FrT, VpsiT <-> FbT, VqT <-> fidT, VcqT <-> FidT)

Following AGMA 2015-1-A01 & 2015-2-A06				
Accuracy grade	[Q-AGMA2015]	A 6	A 6	A 6
Single pitch deviation (µm)	[fptT]	9.00	8.50	9.00
Total cumulative pitch deviation (µm)	[FpT]	34.00	31.00	37.00
Profile form deviation (µm)	[ffaT]	9.00	8.00	9.50
Profile slope deviation (µm)	[fHaT]	7.50	6.50	8.00
Total profile deviation (µm)	[FaT]	11.00	10.00	12.00
Helix form deviation (µm)	[ffbT]	9.00	8.50	9.50
Helix slope deviation (µm)	[fHbT]	9.00	8.50	9.50
Total helix deviation (µm)	[FbT]	13.00	12.00	13.00
Single flank composite, total (µm)	[FisT]	37.00	34.00	41.00
Single flank composite, tooth-to-tooth (µm)	[fisT]	3.40	3.10	3.70
Radial composite, total (µm)	[FidT]	34.00	32.00	37.00
Radial composite, tooth-to-tooth (µm)	[fidT]	6.50	6.00	7.00

According DIN 58405:1972 (Precision Mechanics):

Tooth-to-tooth composite error (µm)	[fi"]	10.00	8.00	10.00
Composite error (µm)	[Fi"]	28.00	22.00	28.00
Axis alignment error (µm)	[fp]	12.54	12.54	12.54
Flank direction error (µm)	[fbeta]	7.14	7.14	7.14
Runout (µm)	[Trk, Fr]	28.00	21.00	28.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality) 6)

Maximum value for deviation error of axis (µm)	[fSigbet]	7.80	7.80
Maximum value for inclination error of axes (µm)	[fSigdel]	15.60	15.60

8. ADDITIONAL DATA

Mass - calculated with da (kg)	[Mass]	2.789	0.369	1.904
Total mass (kg)	[Mass]	6.537		

Moment of inertia (System referenced to wheel 1):
calculation without consideration of the exact tooth shape

single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.00379	0.00005	0.01610
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.00523		
Mean coeff. of friction (acc. Niemann)	[mum]	0.107	0.113	
Wear sliding coef. by Niemann	[zetw]	1.086	0.795	

Meshpower (kW)	26.266	26.266
Gear power loss (kW)	0.117	0.118
Total power loss (kW)	1.177	
Total efficiency	0.972	

9. DETERMINATION OF TOOTH FORM

Data for the tooth form calculation :
Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.00
Required safety for tooth flank	[SHmin]	1.00

Service life (calculated with required safeties):
System service life (h) [Hatt] > 1000000

Tooth root service life (h)	[HFatt]	1e+006	1e+006	1e+006
Tooth flank service life (h)	[HHatt]	1e+006	1e+006	1e+006

Note: The entry 1e+006 h means that the Service life > 1,000,000 h.

Damage calculated on the basis of the required service life (5000.0 h)

F1%	F2%	F3%	H1%	H2%	H3%
0.00	0.00	0.00	0.00	0.00	0.00

REMARKS:

- Symbols used in []: [xx,yy] xx as used in AGMA 2001-D04, yy as used in AGMA 2101-D04
- Specifications with [.e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances

  Specifications with [.m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness

deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances

The calculation is done for the operating pitch circle.

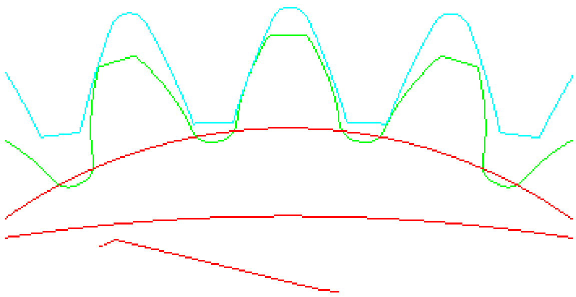
sateff = sat*KL/KT/KR*Kwb/SF (SF = 1.0)

LACR = Spur gear or helical gear with eps.b < 1.0

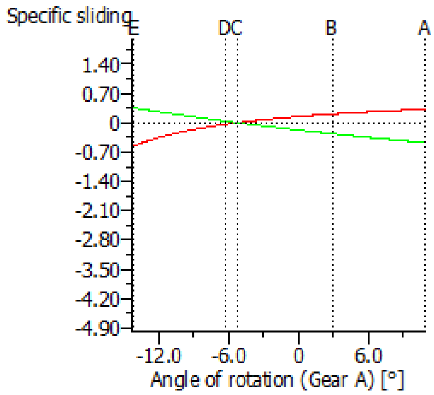
PSTC = Point of Single Tooth Contact

End of Report lines: 501

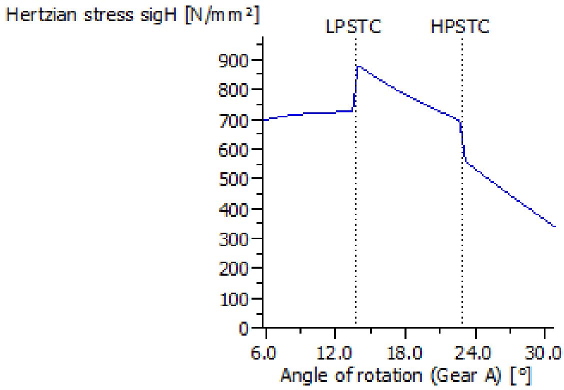
H.2 Stage 2



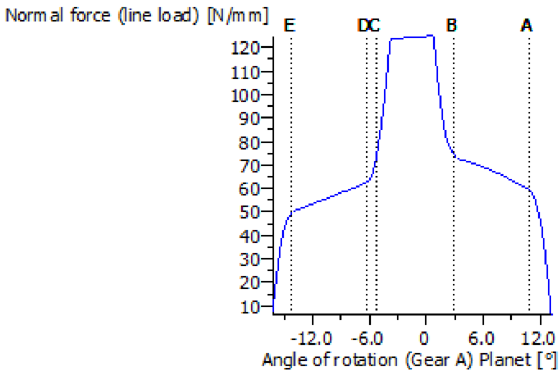
(a) Meshing between ring and planet 2



(b) Specific sliding for ring and planet 2



(c) Hertzian pressure for ring and planet 2



(d) Normal force for ring and planet 2

Figure H.2: Graphical results from KISSsoft for stage 2

KISSsoft Release 03/2016 E

KISSsoft evaluation

File

Name : stage2

Changed by: siguðursvavar on: 14.05.2017 at: 21:02:27

Important hint: At least one warning has occurred during the calculation:

1-> Tooth form factor Y:
For low grade gears (with high pitch deviation) an application of force at tip is required.

Do you want to change your input in the dialog 'Define details of strength' ?

2-> Tooth form factor Y:
For low grade gears (with high pitch deviation) an application of force at tip is required.

Do you want to change your input in the dialog 'Define details of strength' ?

CALCULATION OF A SPUR PLANETARY GEAR STAGE

Drawing or article number:
Gear 1: 0.000.0
Gear 2: 0.000.0
Gear 3: 0.000.0

Calculation method AGMA 2001-C95

		Sun	Planets	Internal gear
Number of planets	[p]	(1)	5	(1)
Power (kW)	[P]		42.067	
Transmitted power (hp)	[P]		56.413	
Transmitted power (ft*lb/s)	[P]		31026.9	
Speed (1/min)	[n]	949.6		0.0
Speed difference for planet bearing calculation (1/min)	[n2]		1101.0	
Speed planet carrier (1/min)	[nSteg]		289.0	
Torque (Nm)	[T]	423.0	0.0	967.0
Torque Pl.-Carrier (Nm)	[TSteg]		1390.000	

Overload factor	[Ko]		1.35	
Power distribution factor	[Kgam]		1.30	
	[KAeff = KA*Kgam]		1.76	
Required service life (h)	[H]		5000.00	
Gear driving (+) / driven (-)	-	+/-		+

Working flank gear 1: Right flank
Sense of rotation gear 1 counterclockwise
Gearbox type: Precision gearing in closed housing

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

------- GEAR 1 ------------ GEAR 2 ------------ GEAR 3 ---

Center distance (in, mm)	[a]	2.9035, 73.750		
Centre distance tolerance	ISO 286:2010 Measure js7			
Normal Diametral Pitch (1/in)	[Pnd]	10.16000		
Normal module (in, mm)	[mn]	0.09843, 2.5000		
Pressure angle at normal section (°)	[alfn]	20.0000		
Helix angle at reference circle (°)	[beta]	0.0000		
Number of teeth	[z]	35	21	-80
Facewidth (mm)	[b]	17.00	17.00	17.00
Hand of gear	Spur gear			
Planetary axles can be placed in regular pitch.:		72°		
Accuracy grade	[Q-AGMA2000-A88]	11	11	11
Inner diameter (mm)	[di]	0.00	0.00	
External diameter (mm)	[di]			0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00	
Outer diameter of gear rim (mm)	[dbi]			0.00

Material

Gear 1: Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened AGMA 2001-C95

Gear 2: Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened AGMA 2001-C95

Gear 3: Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened AGMA 2001-C95

		------- GEAR 1	------------ GEAR 2	------------ GEAR 3 ---
Surface hardness		HRC 60	HRC 60	HRC 60
		(lb/in²), (N/mm²)	(lb/in²), (N/mm²)	(lb/in²), (N/mm²)
Allowable bending stress number	[sat]	65000, 448.2	65000, 448.2	65000, 448.2
Allowable contact stress number	[sac]	225000, 1551.3	225000, 1551.3	225000, 1551.3
Tensile strength (N/mm²)	[σB]	966.00	966.00	966.00
Yield point (N/mm²)	[σS]	822.00	822.00	822.00
Young's modulus (N/mm²)	[E]	206842	206842	206842
Poisson's ratio	[v]	0.300	0.300	0.300
Roughness average value DS, flank (µm)	[RAH]	0.63	0.63	0.63
Roughness average value DS, root (µm)	[RAF]	2.40	2.40	2.40
Mean roughness height, Rz, flank (µm)	[RZH]	5.00	5.00	5.00
Mean roughness height, Rz, root (µm)	[RZF]	16.00	16.00	16.00

Gear reference profile 1 :		
Reference profile 1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A		
Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	
Gear reference profile 2 :		
Reference profile 1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A		
Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)

Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Gear reference profile 3 :
Reference profile 1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250	1.250
Tooth root radius Refer. profile	[rofP*]	0.380	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (µm)	[Ca]	1.80	1.80	1.80

Lubrication type		Oil bath lubrication
Type of oil		Oil: ISO-VG 220
Lubricant base		Mineral-oil base
Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50
Specific density at 15 °C (kg/dm³)	[roOil]	0.895
Oil temperature (°C)	[TS]	70.000

		------- GEAR 1		------------ GEAR 2		------------ GEAR 3 ---
Overall transmission ratio	[itot]		0.304			
Gear ratio	[u]		0.600		-3.810	
Transverse module (mm)	[mt]		2.500			
Pressure angle at pitch circle (°)	[alft]		20.000			
Working transverse pressure angle (°)	[alfwt]		26.886		20.000	
	[alfwt.e/i]		26.909 /	26.863	19.968 /	20.032
Working pressure angle at normal section (°)	[alfwn]		26.886		20.000	
Helix angle at operating pitch circle (°)	[betaw]		0.000		0.000	
Base helix angle (°)	[betab]		0.000			
Reference centre distance (mm)	[ad]		70.000		73.750	
Sum of profile shift coefficients	[Summexi]		1.7591		-0.0000	
Profile shift coefficient	[x]	1.2591		0.5000		-0.5000
Tooth thickness (Arc) (module) (module)	[sn*]	2.4873		1.9347		1.2068
Tip alteration (mm)	[k*mn]	-0.648		-0.648		0.000
Reference diameter (mm)	[d]	87.500		52.500		200.000

Base diameter (mm)	[db]	82.223	49.334	187.939
Tip diameter (mm)	[da]	97.499	58.704	197.500
(mm)	[da.e/i]	97.499 / 97.489	58.704 / 58.694	197.500 / 197.510
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010	-0.000 / 0.010
Tip form diameter (mm)	[dFa]	97.499	58.704	197.500
(mm)	[dFa.e/i]	97.499 / 97.489	58.704 / 58.694	197.500 / 197.510
Active tip diameter (mm)	[dNa.e/i]	97.499 / 97.489	58.704 / 58.694	197.500 / 197.510
Operating pitch diameter (mm)	[dw]	92.188	55.313 / 52.500	200.000
(mm)	[dw.e]	92.206	55.324 / 52.489	199.959
(mm)	[dw.i]	92.169	55.301 / 52.511	200.041
Root diameter (mm)	[df]	87.546	48.750	208.750
Generating Profile shift coefficient	[xE.e/i]	1.2261 / 1.1932	0.4670 / 0.4340	-0.5439 / -0.5879
Manufactured root diameter with xE (mm)	[df.e]	87.381	48.585	208.970
(mm)	[df.i]	87.216	48.420	209.189
Theoretical tip clearance (mm)	[c]	0.625	0.625/ 1.273	0.625
Tip clearance upper allowance (mm)	[c.e]	0.810	0.810/ 1.513	0.810
Tip clearance lower allowance (mm)	[c.i]	0.693	0.693/ 1.368	0.692
Active root diameter (mm)	[dNf]	89.317	51.365/ 50.389	205.155
(mm)	[dNf.e]	89.350	51.389/ 50.414	205.112
(mm)	[dNf.i]	89.291	51.347/ 50.371	205.190
Root form diameter (mm)	[dFf]	88.867	50.470	207.459
(mm)	[dFf.e/i]	88.685 / 88.506	50.370 / 50.275	207.675 / 207.887
Internal toothing: Calculation dFf with pinion type cutter (z0= 26, x0= 0.000)				
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.422 / 0.303	0.069 / 0.001	1.388 / 1.242
Addendum (mm)	[ha = mn * (haP*+x)]	5.000	3.102	1.250
(mm)	[ha.e/i]	5.000 / 4.995	3.102 / 3.097	1.250 / 1.245
Dedendum (mm)	[hf = mn * (hfP*-x)]	-0.023	1.875	4.375
(mm)	[hf.e/i]	0.060 / 0.142	1.957 / 2.040	4.485 / 4.595
Roll angle at dFa (°)	[xsi_dFa.e/i]	36.512 / 36.499	36.952 / 36.930	18.507 / 18.517
Roll angle to dNf (°)	[xsi_dNf.e/i]	24.367 / 24.262	16.710 / 16.535	
	[xsi_dNf.e/i]		12.054 / 11.813	25.047 / 25.106
Roll angle at dFf (°)	[xsi_dFf.e/i]	23.158 / 22.822	11.805 / 11.245	26.938 / 27.091
Tooth height (mm)	[h]	4.977	4.977	5.625
Virtual gear no. of teeth	[zn]	35.000	21.000	-80.000
Normal tooth thickness at tip circle (mm)	[san]	1.568	2.049	2.125
(mm)	[san.e/i]	1.507 / 1.434	1.988 / 1.915	2.050 / 1.967
Normal space width at root circle (mm)	[efn]	1.653	0.000	1.244
(mm)	[efn.e/i]	1.650 / 1.648	0.000 / 0.000	1.222 / 1.200
Max. sliding velocity at tip (m/s)	[vga]	0.628	0.988/ 0.327	1.464
Specific sliding at the tip	[zetaa]	0.545	0.342/ 0.321	0.356
Specific sliding at the root	[zetaf]	-0.520	-1.198/ -0.553	-0.473
Sliding factor on tip	[Kga]	0.310	0.197/ 0.195	0.108
Sliding factor on root	[Kgf]	-0.197	-0.310/ -0.108	-0.195
Pitch on reference circle (mm)	[pt]		7.854	
Base pitch (mm)	[pbt]		7.380	
Transverse pitch on contact-path (mm)	[pet]		7.380	
Length of path of contact (mm)	[ga]	8.757	10.779	
(mm)	[ga.e/i]	8.790 / 8.705	10.823 / 10.710	
Length T1-A (mm)	[T1A]	26.199	7.152/ 15.908	41.132
Length T1-B (mm)	[T1B]	24.822	8.528/ 12.510	37.734
Length T1-C (mm)	[T1C]	20.844	12.506/ 8.978	34.202
Length T1-D (mm)	[T1D]	18.818	14.532/ 8.528	33.752
Length T1-E (mm)	[T1E]	17.442	15.908/ 5.129	30.353

Diameter of single contact point B (mm)	[d-B]	96.048	52.199/ 55.315	202.525
(mm)	[d-B.e]	90.428	57.225/ 52.199	199.724
(mm)	[d-B.i]	90.420	57.302/ 52.193	199.658
Diameter of single contact point D (mm)	[d-D]	90.428	57.259/ 52.199	199.694
(mm)	[d-D.e]	96.014	52.199/ 55.276	202.525
(mm)	[d-D.i]	96.092	52.193/ 55.370	202.537

Transverse contact ratio	[eps_a]	1.187	1.461
Transverse contact ratio with allowances	[eps_a.e/i]	1.191 / 1.180	1.466 / 1.451
Overlap ratio	[eps_b]	0.000	0.000
Total contact ratio	[eps_g]	1.187	1.461
Total contact ratio with allowances	[eps_g.e/i]	1.191 / 1.180	1.466 / 1.451

2. FACTORS OF GENERAL INFLUENCE

------- Gear 1 ------------ Gear 2 ------------ Gear 3 ---

Calculated with the operating pitch circle:

Nominal circumferential force (N)	[Ftw]		1835.578	1933.913	
Axial force (N)	[Fa]	0.0		0.0	0.0
Axial force (total) (N)	[Fatot=Fa*	5]		0.0	0.0
Radial force (N)	[Fr]		930.660	703.887	
Net face width of narrowest member (in)	[F]		0.67 (17.00 mm)	0.67 (17.00 mm)	
Nominal force at operating pitch dia. (lb)	[Wt]		412.45 (1835.58 N)	434.55 (1933.91 N)	
Pitch line velocity (ft/min)	[vt]		627.66 (3.19 m/s)	595.75 (3.03 m/s)	

Gear unit type: Precision enclosed gear unit

Mesh alignment factor	[Cma]	0.076	0.076

Mounting procedure: Without contact improvement at assembly

Mesh alignment correction factor	[Ce]	1.000	

Gearing: without longitudinal flank modification

Lead correction factor	[Cmc]	1.000	1.000
Pinion proportion factor	[Cpf]	0.025	0.025
Pinion proportion modifier	[Cpm]	1.000	1.000
Small offset [s1/s < 0.175]			
Face load distribution factor	[Cmf]	1.101	1.101
Load distribution factor	[Km]	1.101	1.101

Transmission accuracy number introduced:

Transmission accuracy grade number	[Qv]		11	
(VpA replaced by fpe as defined in ISO or DIN)				
Dynamic factor	[Kv]		1.062	
Number of load cycles (in mio.)	[NL]	990.9	330.3	433.5

3. TOOTH ROOT STRENGTH

------- GEAR 1 ------------ GEAR 2 ------------ GEAR 3 ---

Rim thickness factor	[KB]	1.000	1.000	1.000
Size factor	[KS]	1.000	1.000	1.000
Limiting Variation in action (in/10000)	[LimVarAc]	3.0	3.0	

Load sharing:

0 = No (Loaded at tip) 1 = Yes (Loaded at HPSTC)		1	0	
Calc. as helical gear (0) / as LACR (1)		0	0/ 0	0
Load angle (°)	[phinL]	29.73	27.72/ 23.02	20.65

Determination of factor Y with AGMA 908-B89, for internal toothing by graphical method

Heigth of Lewis parabola (in)	[hF]	0.129	0.127/ 0.091	0.143
Heigth of Lewis parabola (mm)	[hF]	3.267	3.230/ 2.311	3.635
Tooth thickness at critical section (in)	[sF]	0.231	0.210/ 0.216	0.259

Tooth thickness at critical section (mm)	[sF]	5.878	5.346/ 5.489	6.588
Radius at curvature of fillet curve (in)	[roF]	0.038	0.039/ 0.039	0.046
Radius at curvature of fillet curve (mm)	[roF]	0.966	0.991/ 0.991	1.164
Helical factor	[Ch]	1.00	1.00	
Helix angle factor	[Kpsi]	1.00	1.00	
Tooth form factor Y	[Y]	0.873	0.695/ 1.067	0.902
Stress correction factor	[Kf]	1.887	1.795/ 2.088	1.875
Bending strength geometry factor J	[J]	0.463	0.387/ 0.511	0.481
Bending stress number (lb/in²)	[st]	27769.3	33200.4/ 26457.9	28108.5
Bending stress number (N/mm²)	[st]	191.5	228.9/ 182.4	193.8
Stress cycle factor (for general applications)	[YN]	0.938	0.956	0.952
Allowable bending stress number (lb/in²)	[sat]	65000.0	65000.0	65000.0
Allowable bending stress number (N/mm²)	[sat]	448.2	448.2	448.2
Temperature factor	[KT]	1.000	1.000	1.000
Reliability factor (99.00 %)	[KR]	1.000		
Reverse loading factor	[-]	1.000	0.700	1.000
Effective allow. b.s.n. (lb/in²)	[sateff]	60950.9	43508.2/ 43508.2	61854.4
Effective allow. b.s.n. (N/mm²)	[sateff]	420.2	300.0/ 300.0	426.5
Bending strength power rating (hp)	[Patu]	217.3(162.04 kW)	129.7(96.75 kW)/ 162.8(121.40 kW)	217.9(162.46 kW)

Note: Bending strength power rating calculated with Kγ=1, Ko=1, KR=1, SFmin=1

Unit load (lb/in²)	[UL]	6262.06	6597.53	
Allowable unit load (lb/in²)	[Uat]	13744.6	8206.2/ 10849.2	14518.2
Required safety factor	[SFmin]	1.000	1.000	1.000
Safety factor (Bending)	[sateff/st]	2.195	1.310/ 1.644	2.201
Transmittable power including SFmin	[Patu/SFmin]	217.3(162.04 kW)	129.7(96.75 kW)/ 162.8(121.40 kW)	217.9(162.46 kW)

Note: Transmittable power including SFmin calculated with Kγ=1, Ko=1, KR=1

(Note: Materials with HB > 400: Yield strength not checked.)

4. SAFETY AGAINST PITTING (TOOTH FLANK)

		------- Gear 1 ------------	Gear 2 ------------	Gear 3 ---
		(√lb/in), (√N/mm)	(√lb/in), (√N/mm)	
Elastic coefficient	[Cp]	2290.00, 190.20	2290.00, 190.20	
Size factor	[Ks]	1.000	1.000	1.000
Load sharing ratio	[mN]	1.000	1.000	
Helical overlap factor	[Cpsi]	1.000	1.000	
Geometry factor I	[I]	0.102	0.204	
		(lb/in²), (N/mm²)	(lb/in²), (N/mm²)	
Contact stress number	[sc]	172556.8, 1189.7	128605.3, 886.7	
Stress cycle factor (for general applications)	[ZN]	0.900	0.923	0.917
Surface condition factor	[Cf]	1.000	1.000/ 1.000	1.000
Hardness ratio factor	[CH]	1.000	1.000/ 1.000	1.000
Temperature factor	[KT]	1.000	1.000	1.000
Reliability factor	[KR]	99.000	ZS.CRAGMA	
Allowable contact stress number (lb/in²)	[sac]	225000.0	225000.0	225000.0
Allowable contact stress number (N/mm²)	[sac]	1551.3	1551.3	1551.3
Effective allow. c.s.n. (lb/in²)	[saceff]	202433.6	207613.9/207613.9	206319.5
Effective allow. c.s.n. (N/mm²)	[saceff]	1395.7	1431.4/ 1431.4	1422.5
Pitting resistance power rating (hp)	[Pacu]	136.3(101.61 kW)	143.3(106.87 kW)/ 258.0(192.40 kW)	254.8(190.01 kW)

Note: Pitting resistance power rating calculated with Kγ=1, Ko=1, KR=1, SHmin=1

Contact load factor (lb/in²)	[K]	452.8	231.7	
Allowable contact load factor (lb/in²)	[Kac]	623.2	655.5/ 603.8	596.3
Required safety factor	[SHmin]	1.000	1.000	1.000
Safety factor (Pitting)	[saceff/sc]	1.173	1.203/ 1.614	1.604
Transmittable power including SHmin (hp)	[Pacu/SHmin^2]	136.3(101.61 kW)	143.3(106.87 kW)/ 258.0(192.40 kW)	254.8(190.01 kW)

Note: Transmittable power including SHmin calculated with Kγ=1, Ko=1, KR=1

SERVICE FACTORS (with Kγ= 1.000):

Service factor for tooth root	[KSF]	3.852	2.300/ 2.886	3.862
Service factor for pitting	[CSF]	2.415	2.541/ 4.574	4.517
Service factor for gear set	[SF]	2.300	2.886	

SERVICE FACTORS (with Kγ= 1.300):

Service factor for tooth root	[KSF]	2.963	1.769/ 2.220	2.971
Service factor for pitting	[CSF]	1.858	1.954/ 3.518	3.475
Service factor for gear set	[SF]	1.769	2.220	

Note: Service factors calculated with Ko=1, KR=1, SFmin=1, SHmin=1

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Pairing Gear 1- 2:
Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

Pairing Gear 2- 3:
Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Results from AGMA 925-A03 (Details see in the specific calculation sheet)

Probability of wear (%)	[Pwear]	82.668
Probability of scuffing (%)	[Pscuff]	5% or lower

6. MEASUREMENTS FOR TOOTH THICKNESS

		------- GEAR 1	------------ GEAR 2	------------ GEAR 3 ---
Tooth thickness deviation		DIN 3967 d26	DIN 3967 d26	DIN 3967 d26
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.060/ -0.120	-0.060/ -0.120	-0.080/ -0.160
Number of teeth spanned	[k]	6.000	4.000	-10.000
(Internal toothing: k = (Measurement gap number)				
Base tangent length (no backlash) (mm)	[Wk]	43.970	27.421	-73.769
Actual base tangent length ('span') (mm)	[Wk.e/i]	43.914/ 43.858	27.365/ 27.309	-73.844/ -73.920
Diameter of contact point (mm)	[dMWk.m]	93.202	56.402	201.939
Theoretical diameter of ball/pin (mm)	[DM]	5.584	4.951	4.207
Effective diameter of ball/pin (mm)	[DMeff]	6.000	5.000	4.250
Radial single-ball measurement backlash free (mm)	[MrK]	51.695	31.228	98.300
Radial single-ball measurement (mm)	[MrK.e/i]	51.643/ 51.590	31.173/ 31.118	98.406/ 98.511
Diameter of contact point (mm)	[dMMr.m]	94.167	54.918	202.722
Diametral measurement over two balls without clearance (mm)	[MdK]	103.293	62.296	196.600

Diametral two ball measure (mm)	[MdK.e/i]	103.187/ 103.082	62.186/ 62.075	196.812/ 197.023
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	103.187/ 103.082	62.186/ 62.075	196.812/ 197.023
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	103.187/ 103.082	62.186/ 62.075	196.812/ 197.023
Effective dimensions over 3 pins (mm)	[Md3R.e/i]	103.090/ 102.984	62.026/ 61.916	-0.000/ -0.000
Tooth thickness (chordal) in pitch diameter (mm)	[sc]	6.213	4.830	3.017
(mm)	[sc.e/i]	6.153/ 6.093	4.770/ 4.710	2.937/ 2.857
Reference chordal height from da.m (mm)	[ha]	5.108	3.211	1.236
Tooth thickness (Arc) (mm)	[sn]	6.218	4.837	3.017
(mm)	[sn.e/i]	6.158/ 6.098	4.777/ 4.717	2.937/ 2.857
Backlash free center distance (mm)	[aControl.e/i]	73.625/ 73.499	73.940/ 74.127	
Backlash free center distance, allowances (mm)	[jta]	-0.125/ -0.251	0.190/ 0.377	
dNf.i with aControl (mm)	[dNf0.i]	88.886	49.993	206.033
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.101	-0.188	0.821
Tip clearance	[c0.i(aControl)]	0.457	0.457	0.330
Centre distance allowances (mm)	[Aa.e/i]	0.015/ -0.015	-0.015/ 0.015	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.015/ -0.015	0.011/ -0.011	
Radial clearance (mm)	[jrw]	0.266/ 0.110	0.392/ 0.175	
Circumferential backlash (transverse section) (mm)	[jtw]	0.268/ 0.111	0.291/ 0.129	
Normal backlash (mm)	[jnw]	0.252/ 0.105	0.273/ 0.121	
Entire torsional angle (°)	[j.tSys]	0.2073/ 0.1036		

(j.tSys: Torsional angle of planet carrier for blocked shaft)

7. GEAR ACCURACY

		------- GEAR 1	------------ GEAR 2	------------ GEAR 3 ---
Following AGMA 2000-A88				
Accuracy grade	[Q-AGMA2000]	11	11	11
Pitch Variation Allowable (µm)	[VpA]	6.90	6.30	8.00
Runout Radial Tolerance (µm)	[VrT]	24.00	21.00	29.00
Profile Tolerance (µm)	[VphiT]	8.80	8.10	10.00
Tooth Alignment Tolerance (µm)	[VpsiT]	6.60	6.60	6.60
Composite Tolerance, Tooth-to-Tooth (µm)	[VqT]	13.00	14.00	13.00
Composite Tolerance, Total (µm)	[VcqT]	37.00	34.00	43.00

(AGMA <-> ISO: VpA <-> fpbT, VrT <-> FrT, VpsiT <-> FbT, VqT <-> fidT, VcqT <-> FidT)

Following AGMA 2015-1-A01 & 2015-2-A06				
Accuracy grade	[Q-AGMA2015]	A 6	A 6	A 6
Single pitch deviation (µm)	[fptT]	9.00	8.50	9.50
Total cumulative pitch deviation (µm)	[FpT]	33.00	32.00	38.00
Profile form deviation (µm)	[ffaT]	8.50	8.00	9.50
Profile slope deviation (µm)	[fHaT]	7.00	6.50	8.00
Total profile deviation (µm)	[FaT]	11.00	10.00	13.00
Helix form deviation (µm)	[ffbT]	8.00	7.50	8.50
Helix slope deviation (µm)	[fHbT]	8.00	7.50	8.50
Total helix deviation (µm)	[FbT]	11.00	11.00	12.00
Single flank composite, total (µm)	[FisT]	37.00	35.00	42.00
Single flank composite, tooth-to-tooth (µm)	[fisT]	3.30	3.20	3.80
Radial composite, total (µm)	[FidT]	34.00	32.00	38.00

Radial composite, tooth-to-tooth (µm)	[fidT]	6.00	6.00	7.00

According DIN 58405:1972 (Precision Mechanics):

Tooth-to-tooth composite error (µm)	[fi"]	9.00	9.00	10.00
Composite error (µm)	[Fi"]	25.00	25.00	28.00
Axis alignment error (µm)	[fp]	12.54	12.54	12.54
Flank direction error (µm)	[fbeta]	5.00	5.00	5.00
Runout (µm)	[Trk, Fr]	24.00	24.00	28.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality) 6)

Maximum value for deviation error of axis (µm)	[fSigbet]	7.15	7.15
Maximum value for inclination error of axes (µm)	[fSigdel]	14.30	14.30

8. ADDITIONAL DATA

Mass - calculated with da (kg)	[Mass]	0.994	0.360	1.050
Total mass (kg)	[Mass]	3.845		

Moment of inertia (System referenced to wheel 1):
calculation without consideration of the exact tooth shape

single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.0009524	0.0001079	0.01043
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.00191		
Mean coeff. of friction (acc. Niemann)	[mum]	0.078	0.075	
Wear sliding coef. by Niemann	[zetw]	0.553	0.502	

Meshpower (kW)	29.264	29.264
Gear power loss (kW)	0.060	0.034
Total power loss (kW)	0.468	
Total efficiency	0.989	

9. DETERMINATION OF TOOTH FORM

Data for the tooth form calculation :

Calculation of Gear 1
Tooth form, Sun gear, Step 1: Automatic (final machining)
haP*= 0.789, hfP*= 1.250, rofP*= 0.380

Calculation of Gear 2
Tooth form, Planets, Step 1: Automatic (final machining)
haP*= 0.789, hfP*= 1.250, rofP*= 0.380

Calculation of Gear 3
Tooth form, Internal gear, Step 1: Automatic (final machining)
z0= 26, x0=0.0000, da0= 71.434 mm, a0= -68.823 mm
haP0*= 1.287, roaP0*= 0.000, hfP0*= 1.028, rofP0*= 0.380

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.00		
Required safety for tooth flank	[SHmin]	1.00		

Service life (calculated with required safeties):

System service life (h)	[Hatt]	> 1000000		
Tooth root service life (h)	[HFatt]	1e+006	1e+006	1e+006
Tooth flank service life (h)	[HHatt]	1e+006	1e+006	1e+006

Note: The entry 1e+006 h means that the Service life > 1,000,000 h.

Damage calculated on the basis of the required service life (5000.0 h)

F1%	F2%	F3%	H1%	H2%	H3%
0.00	0.00	0.00	0.10	0.03	0.00

REMARKS:

- Symbols used in []: [xx,yy] xx as used in AGMA 2001-D04, yy as used in AGMA 2101-D04
- Specifications with [.e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances

  Specifications with [.m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness

deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances

The calculation is done for the operating pitch circle.

sateff = sat*KL/KT/KR*Kwb/SF (SF = 1.0)

LACR = Spur gear or helical gear with eps.b < 1.0

PSTC = Point of Single Tooth Contact

End of Report lines: 516

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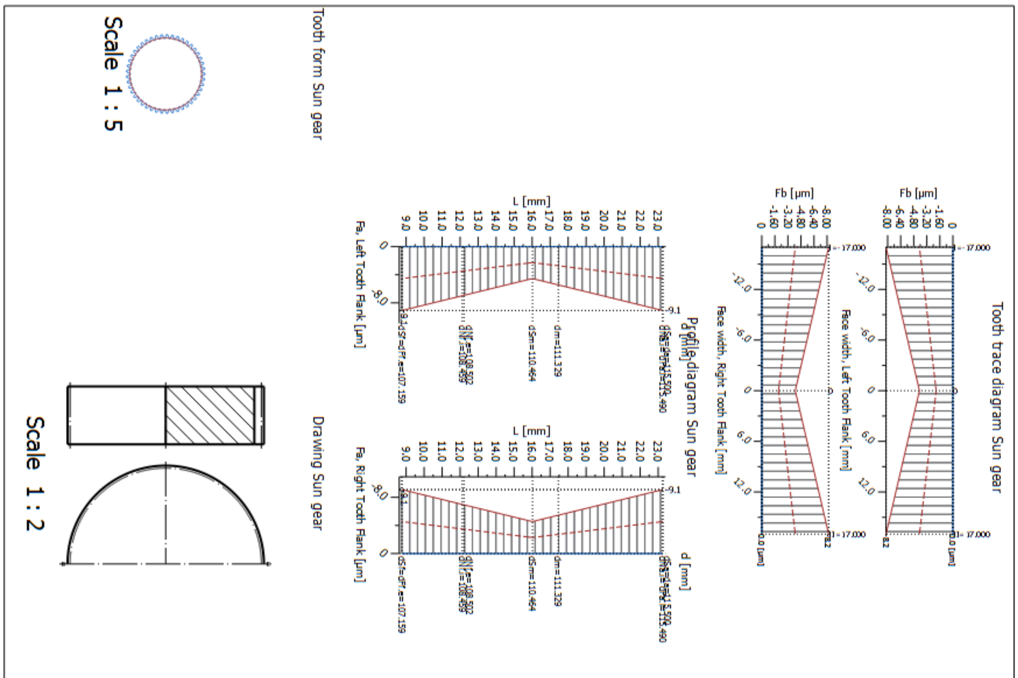
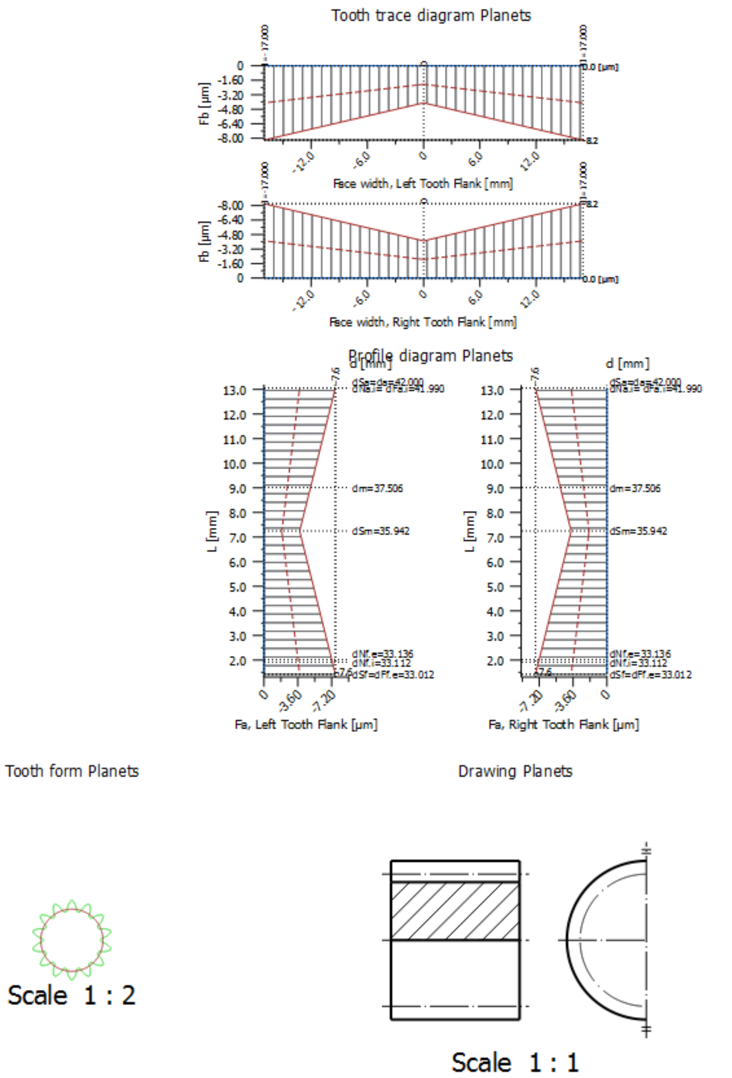
KISSsoft Maufacturing Drawings



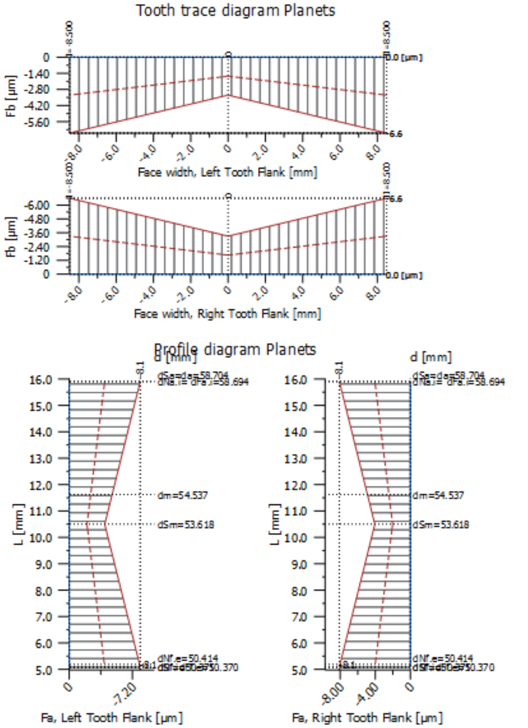
MANUFACTURING DATA FOR CYLINDRICAL GEARS		
Drawing or article number		0.000.0
Number of teeth	[z]	45
Facewidth (mm)	[b]	34.000
Normal Diametral Pitch (1/in)	[Pnd]	10.16000
Transverse Diametral Pitch (1/in)	[Pd]	10.16000
Normal module (in, mm)	[mn]	0.09843, 2.500
Helix angle (°)	[beta]	0.000 (0°0'0")
Hand of gear		Spur gear
Pressure angle at normal section (°)	[alfn]	20.000 (20°0'0")
Material		Steel, Grade 2, HRC58
Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Addendum coefficient	[haP*]	1.000
Dedendum coefficient	[hfP*]	1.250
Tip radius factor	[rhoaP*]	0.000
Root radius factor	[rhofP*]	0.380
Tip form height coefficient	[hFaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle (°)	[alfprP]	0.000
Ramp angle (°)	[alfKP]	0.000
not topping		
Accuracy grade according to AGMA2000		11
Profile shift coefficient	[x]	-0.400
Reference diameter (mm)	[d]	112.500
Tip diameter (mm)	[da]	115.500 (0.000 / -0.010)
Root diameter (mm)	[df]	104.250 (-0.165 / -0.330)
Tooth thickness deviation		DIN 3967 d26
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.060/-0.120
Number of teeth spanned	[k]	5
Base tangent length (no backlash) (mm)	[Wk]	34.103
Actual base tangent length ('span') (mm)	[Wk.e/i]	34.047/33.990
Effective diameter of ball/pin (mm)	[DMeff]	4.250
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	116.144/115.972
Measurement over 3 pins (axial) according to AGMA 2002 (mm)		
[MdR.e/i] 116.144/115.972		
Chordal tooth thickness (without back lash) (mm)	[sc]	3.199
Actual chordal tooth thickness (mm)	[sc.e/i]	3.139/3.079
Reference chordal height (mm)	[ha]	1.520



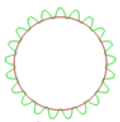
MANUFACTURING DATA FOR CYLINDRICAL GEARS		
Drawing or article number		0.000.0
Number of teeth	[z]	14
Facewidth (mm)	[b]	34.000
Normal Diametral Pitch (1/in)	[Pnd]	10.16000
Transverse Diametral Pitch (1/in)	[Pd]	10.16000
Normal module (in, mm)	[mn]	0.09843, 2.500
Helix angle (°)	[beta]	0.000 (0°0'0")
Hand of gear		Spur gear
Pressure angle at normal section (°)	[alfn]	20.000 (20°0'0")
Material		Steel, Grade 2, HRC58
Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Addendum coefficient	[haP*]	1.000
Dedendum coefficient	[hfP*]	1.250
Tip radius factor	[rhoaP*]	0.000
Root radius factor	[rhofP*]	0.380
Tip form height coefficient	[hFaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle (°)	[alfprP]	0.000
Ramp angle (°)	[alfKP]	0.000
not topping		
Accuracy grade according to AGMA2000		11
Profile shift coefficient	[x]	0.400
Reference diameter (mm)	[d]	35.000
Tip diameter (mm)	[da]	42.000 (0.000 / -0.010)
Root diameter (mm)	[df]	30.750 (-0.121 / -0.258)
Tooth thickness deviation		DIN 3967 d26
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.044/-0.094
Number of teeth spanned	[k]	3
Base tangent length (no backlash) (mm)	[Wk]	19.625
Actual base tangent length ('span') (mm)	[Wk.e/i]	19.584/19.537
Effective diameter of ball/pin (mm)	[DMeff]	5.250
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	44.967/44.883
Measurement over 3 pins (axial) according to AGMA 2002 (mm)		
[MdR.e/i] 44.967/44.883		
Chordal tooth thickness (without back lash) (mm)	[sc]	4.641
Actual chordal tooth thickness (mm)	[sc.e/i]	4.597/4.547
Reference chordal height (mm)	[ha]	3.652

Tooth trace diagram Planets

Profile diagram Planets

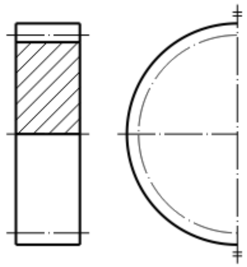


Tooth form Planets



Scale 1 : 2

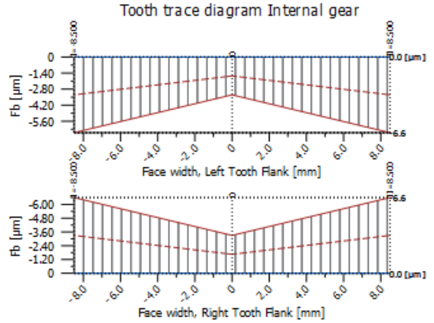
Drawing Planets



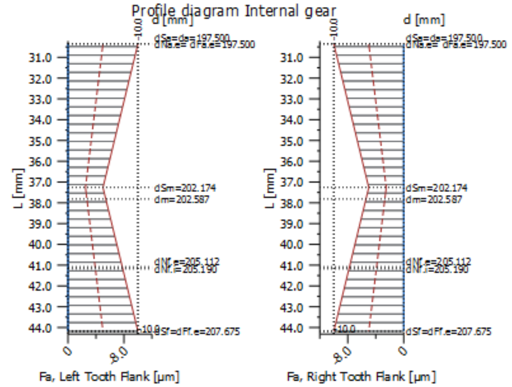
Scale 1 : 1

MANUFACTURING DATA FOR CYLINDRICAL GEARS		
Drawing or article number		0.000.0
Number of teeth	[z]	21
Facewidth (mm)	[b]	17.000
Normal Diametral Pitch (1/in)	[Pnd]	10.16000
Transverse Diametral Pitch (1/in)	[Pd]	10.16000
Normal module (in, mm)	[mn]	0.09843, 2.500
Helix angle (°)	[beta]	0.000 (0°0'0")
Hand of gear		Spur gear
Pressure angle at normal section (°)	[alfn]	20.000 (20°0'0")
Material		Steel, Grade 2, HRC58
Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Addendum coefficient	[haP*]	1.000
Dedendum coefficient	[hfP*]	1.250
Tip radius factor	[rhoaP*]	0.000
Root radius factor	[rhofP*]	0.380
Tip form height coefficient	[hFaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle (°)	[alfprP]	0.000
Ramp angle (°)	[alfKP]	0.000
not topping		
Accuracy grade according to AGMA2000		11
Profile shift coefficient	[x]	0.500
Reference diameter (mm)	[d]	52.500
Tip diameter (mm)	[da]	58.704 (0.000 / -0.010)
Root diameter (mm)	[df]	48.750 (-0.165 / -0.330)
Tooth thickness deviation		DIN 3967 d26
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.060/-0.120
Number of teeth spanned	[k]	4
Base tangent length (no backlash) (mm)	[Wk]	27.421
Actual base tangent length ('span') (mm)	[Wk.e/i]	27.365/27.309
Effective diameter of ball/pin (mm)	[DMeff]	5.000
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	62.186/62.075
Measurement over 3 pins (axial) according to AGMA 2002 (mm)		
[MdR.e/i] 62.186/62.075		
Chordal tooth thickness (without back lash) (mm)	[sc]	4.830
Actual chordal tooth thickness (mm)	[sc.e/i]	4.770/4.710
Reference chordal height (mm)	[ha]	3.211

Tooth trace diagram Internal gear

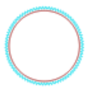


Profile diagram Internal gear



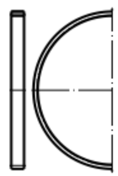
MANUFACTURING DATA FOR CYLINDRICAL GEARS		
Drawing or article number		0.000.0
Number of teeth	[z]	-80
Facewidth (mm)	[b]	17.000
Normal Diametral Pitch (1/in)	[Pnd]	10.16000
Transverse Diametral Pitch (1/in)	[Pd]	10.16000
Normal module (in, mm)	[mn]	0.09843, 2.500
Helix angle (°)	[beta]	0.000 (0°0'0")
Hand of gear		Spur gear
Pressure angle at normal section (°)	[alfn]	20.000 (20°0'0")
Material		Steel, Grade 2, HRC58
Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Addendum coefficient	[haP*]	1.000
Dedendum coefficient	[hfP*]	1.250
Tip radius factor	[rhoaP*]	0.000
Root radius factor	[rhofP*]	0.380
Tip form height coefficient	[hFaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle (°)	[alfprP]	0.000
Ramp angle (°)	[alfKP]	0.000
not topping		
Accuracy grade according to AGMA2000		11
Profile shift coefficient	[x]	-0.500
Reference diameter (mm)	[d]	200.000
Tip diameter (mm)	[da]	197.500 (-0.000 / 0.010)
Root diameter (mm)	[df]	208.750 (0.220 / 0.440)
Tooth thickness deviation		DIN 3967 d26
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.080/-0.160
Dimension gap number	[k]	10
Base tangent length (no backlash) (mm)	[Wk]	73.769
Actual base tangent length ('span') (mm)	[Wk.e/i]	73.844/73.920
Effective diameter of ball/pin (mm)	[DMeff]	4.250
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	196.812/197.023
Measurement over 3 pins (axial) according to AGMA 2002 (mm) [MdR.e/i] 196.812/197.023		
Chordal tooth thickness (without back lash) (mm)	[sc]	3.017
Actual chordal tooth thickness (mm)	[sc.e/i]	2.937/2.857
Reference chordal height (mm)	[ha]	1.236

Tooth form Internal gear



Scale 1 : 10

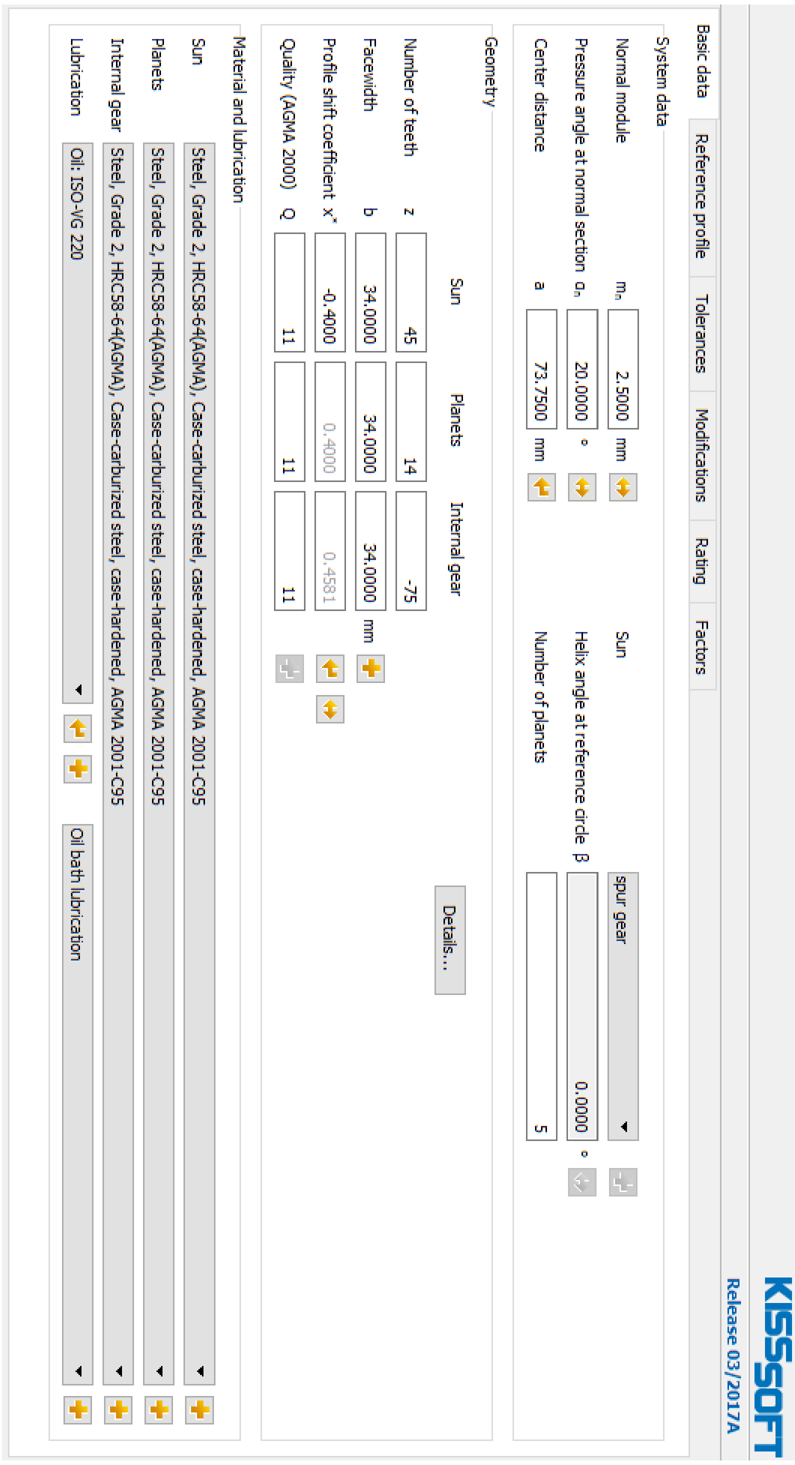
Drawing Internal gear

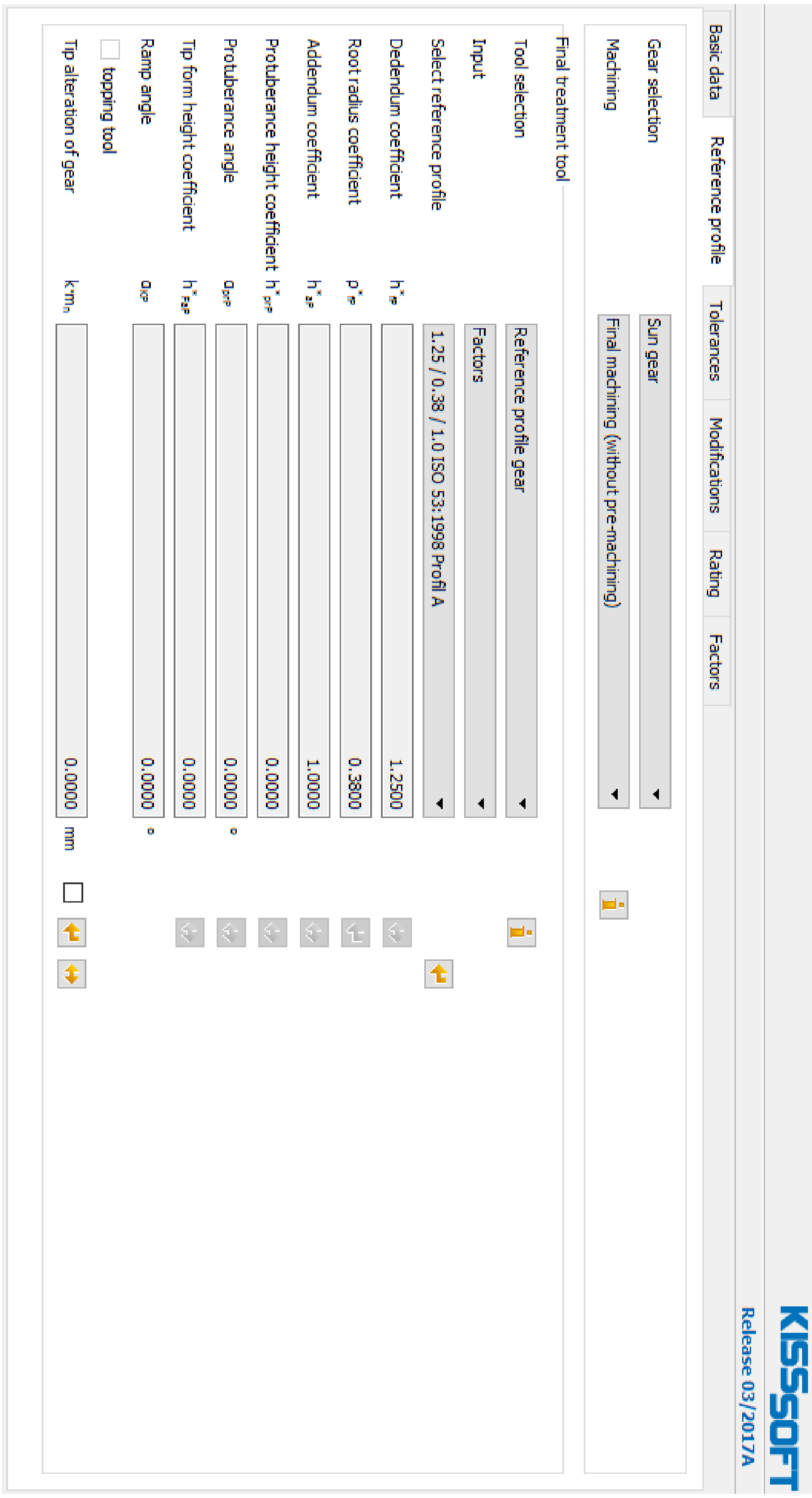


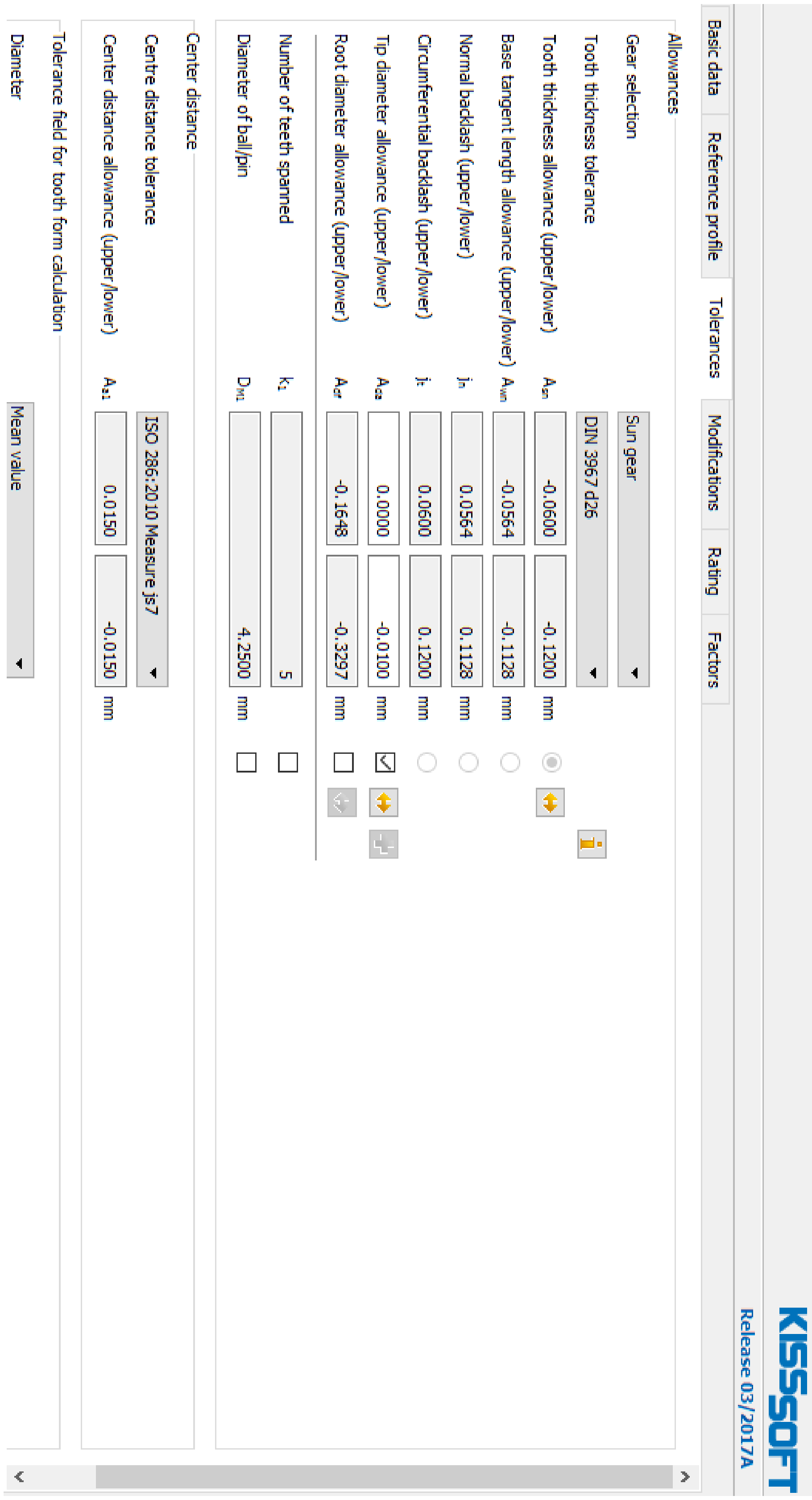
Scale 1 : 5

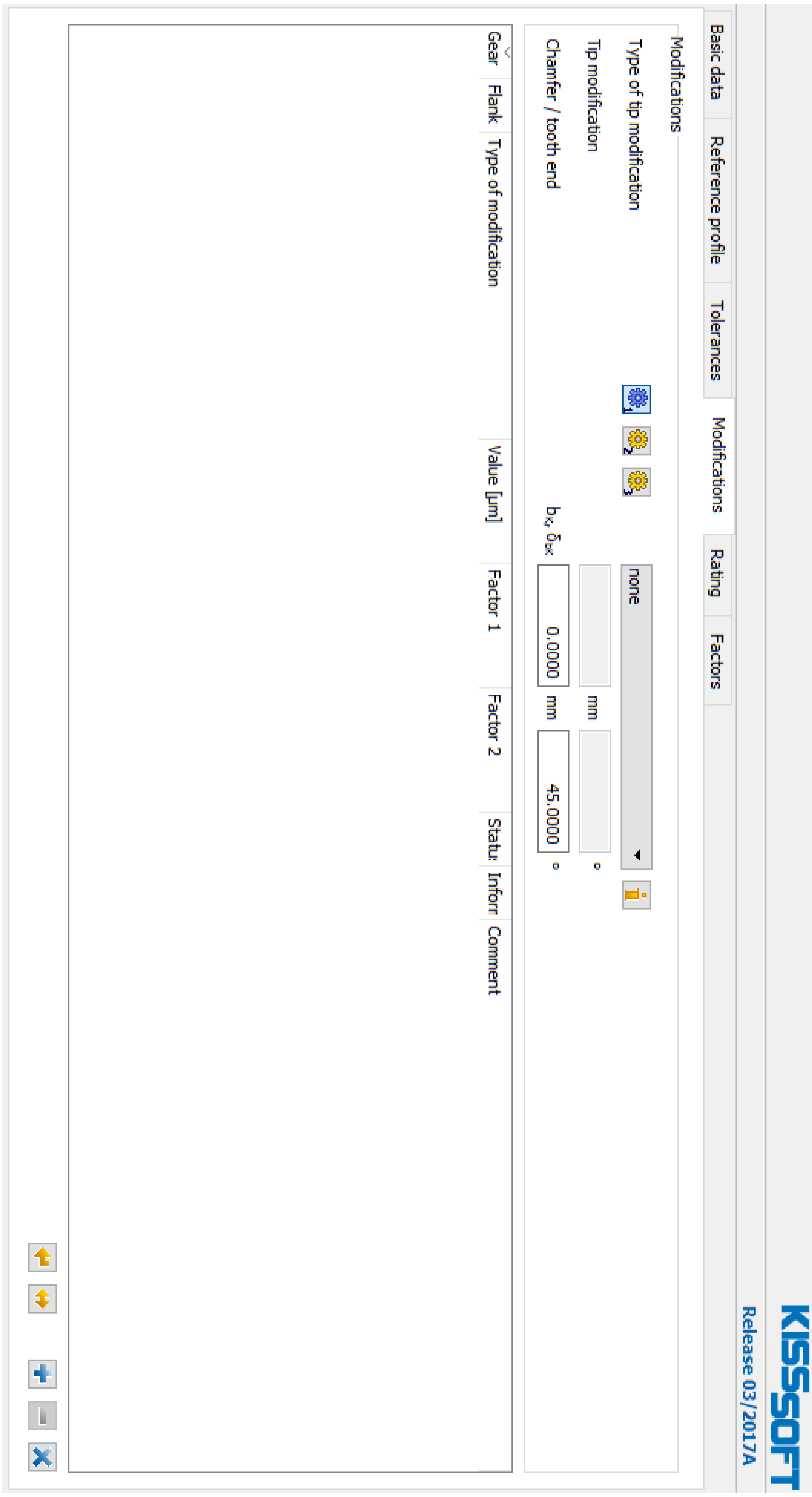
J

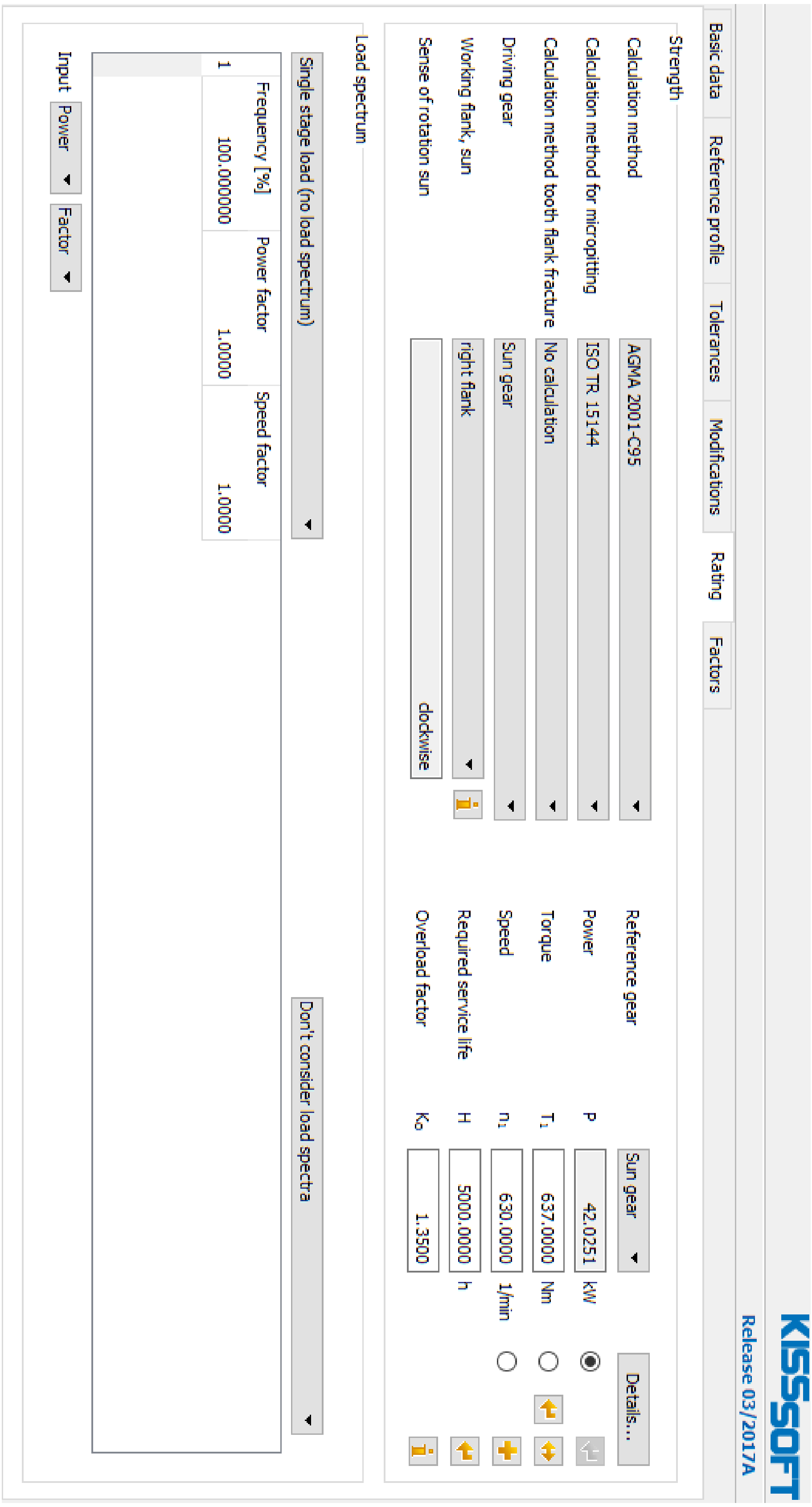
KISSsoft Input Parameters

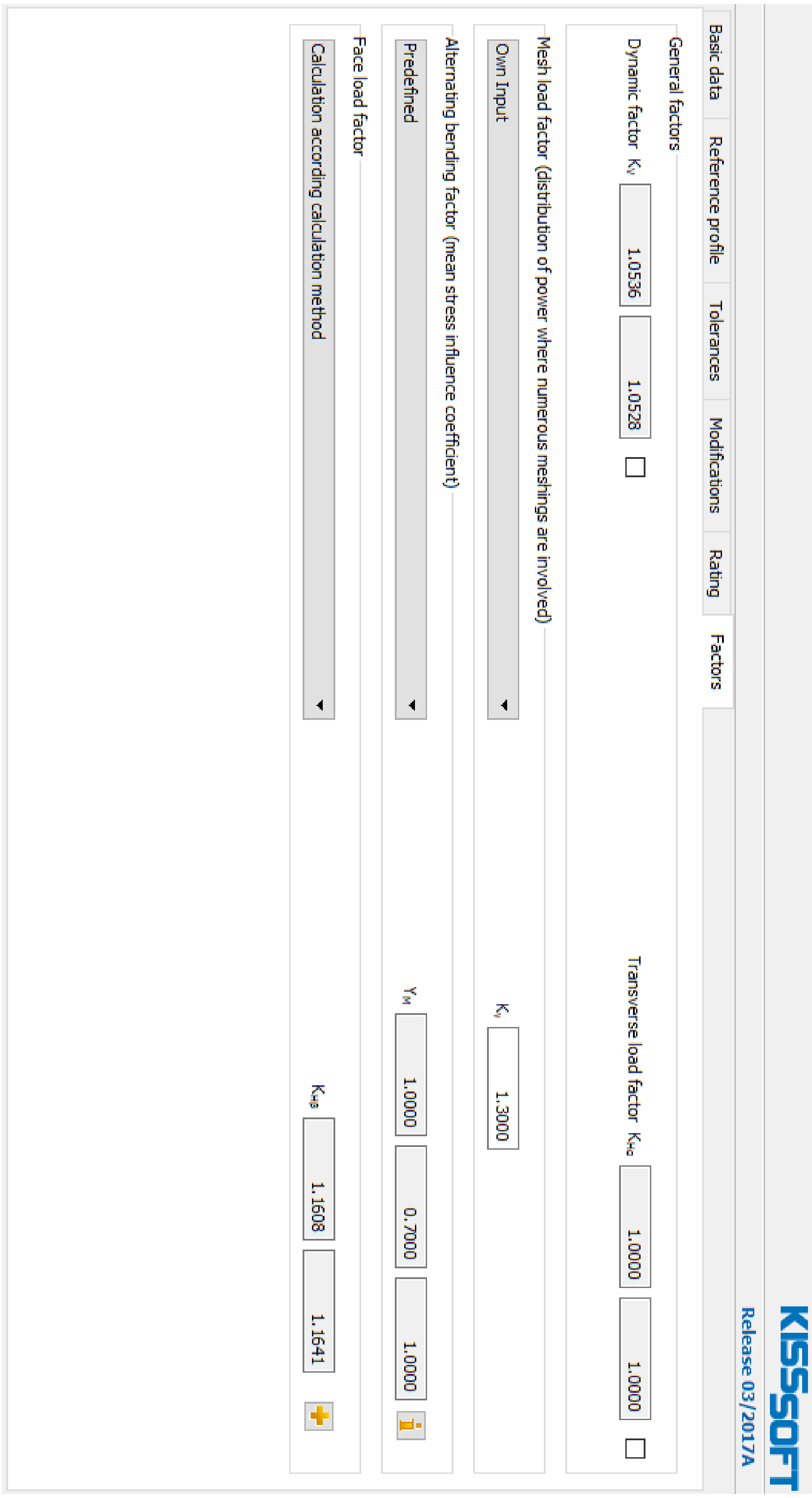


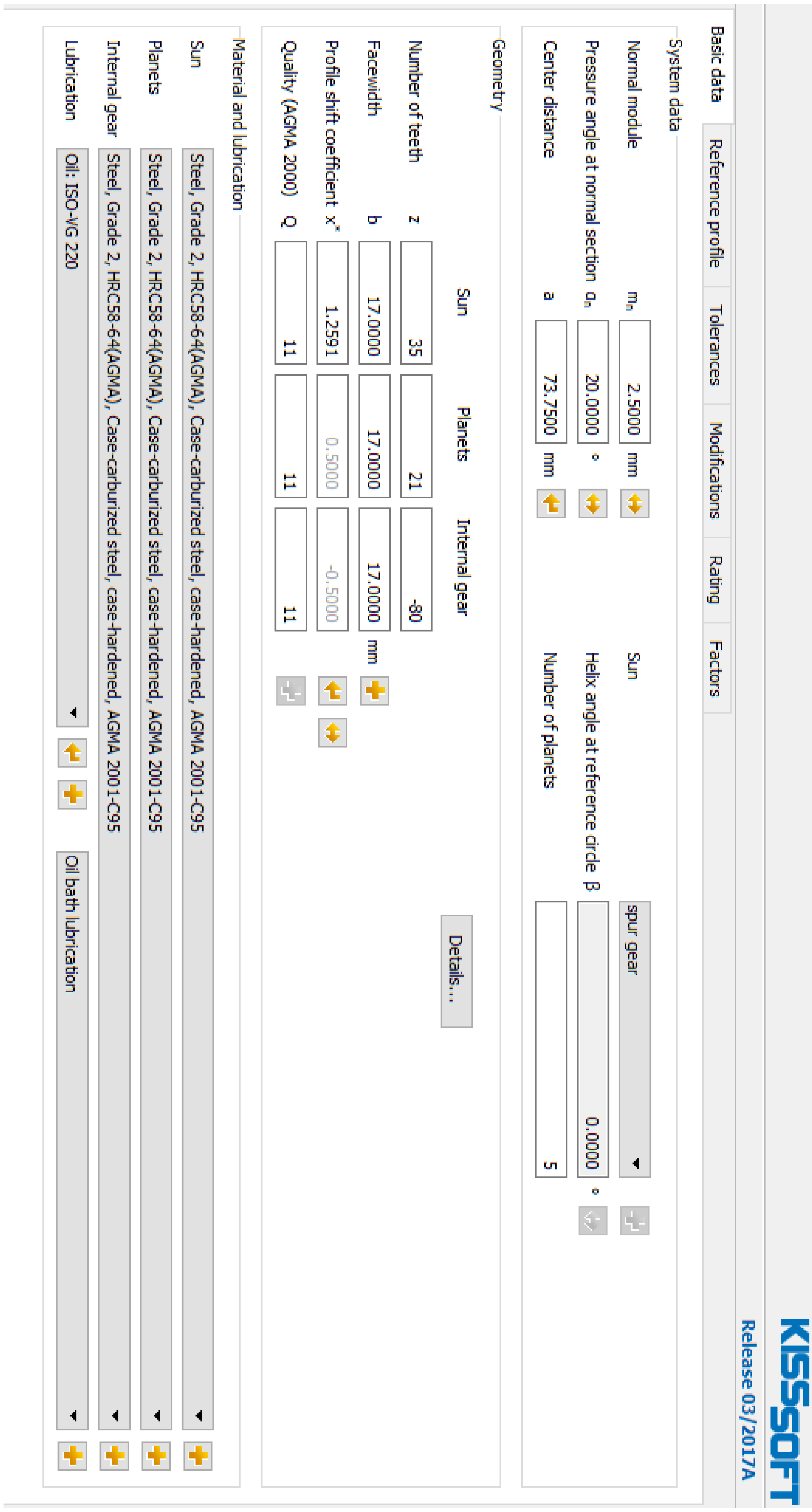

Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
Gear selection
Sun gear
Machining
Final machining (without pre-machining)
Final treatment tool
Tool selection
Reference profile gear
Input
Factors
Select reference profile
1.25 / 0.38 / 1.0 ISO 53:1998 Profil A
Dedendum coefficient
1.2500
Root radius coefficient
0.3800
Addendum coefficient
1.0000
Protuberance height coefficient
0.0000
Protuberance angle
0.0000
Tip form height coefficient
0.0000
Ramp angle
0.0000
topping tool
Tip alteration of gear
0.0000
mm
KISSSOFT
Release 03/2017A

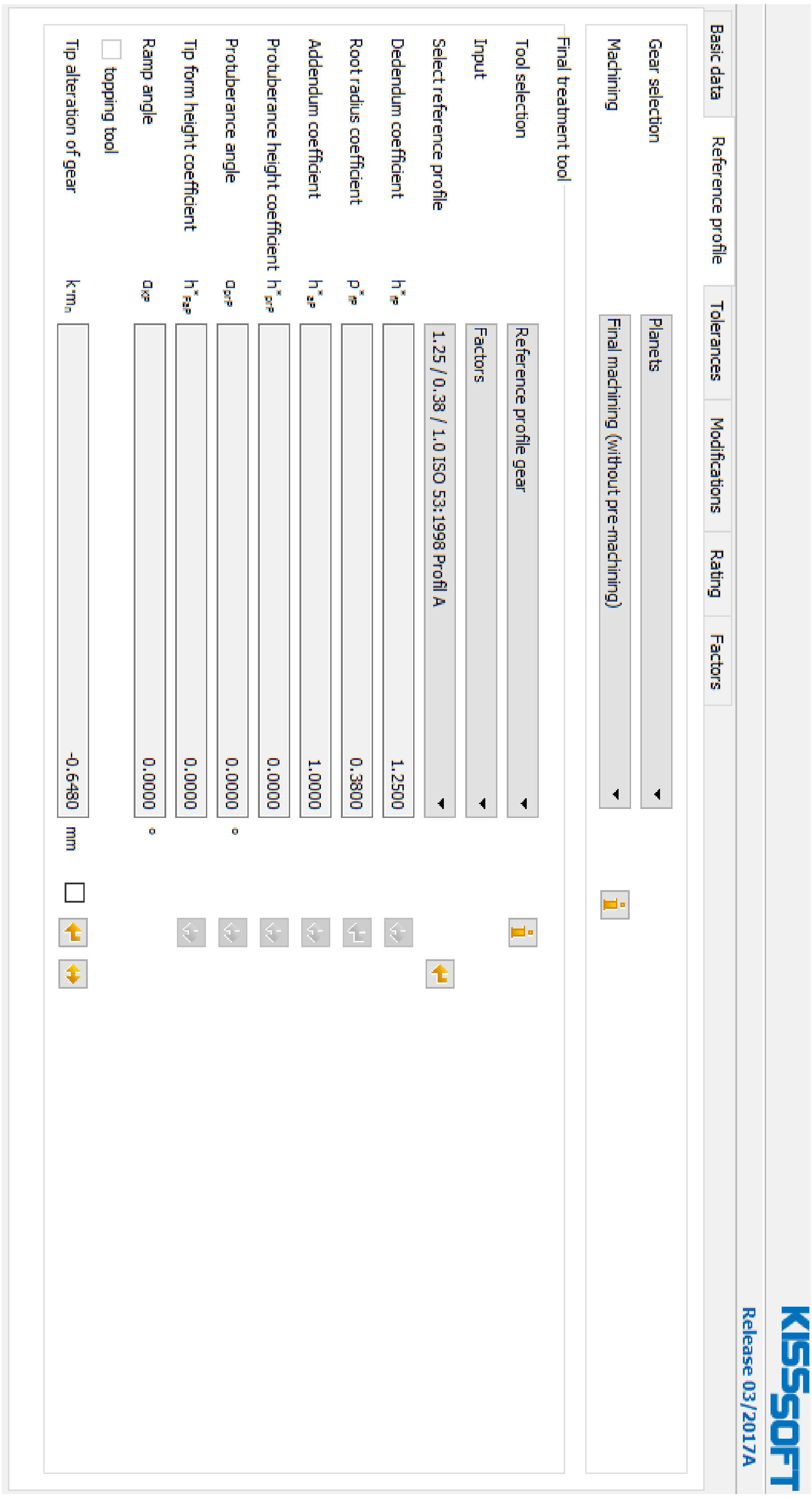

KISSsoft
Release 03/2017A
Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
Allowances
Gear selection
Sun gear
Tooth thickness tolerance
DIN 3967 d26
Tooth thickness allowance (upper/lower)
-0.0600
-0.1200
mm
Base tangent length allowance (upper/lower)
-0.0564
-0.1128
mm
Normal backlash (upper/lower)
0.0564
0.1128
mm
Circumferential backlash (upper/lower)
0.0600
0.1200
mm
Tip diameter allowance (upper/lower)
0.0000
-0.0100
mm
Root diameter allowance (upper/lower)
-0.1648
-0.3297
mm
Number of teeth spanned
5
Diameter of ball/pin
4.2500
mm
Center distance
Centre distance tolerance
ISO 286:2010 Measure js7
Center distance allowance (upper/lower)
0.0150
-0.0150
mm
Tolerance field for tooth form calculation
Diameter
Mean value

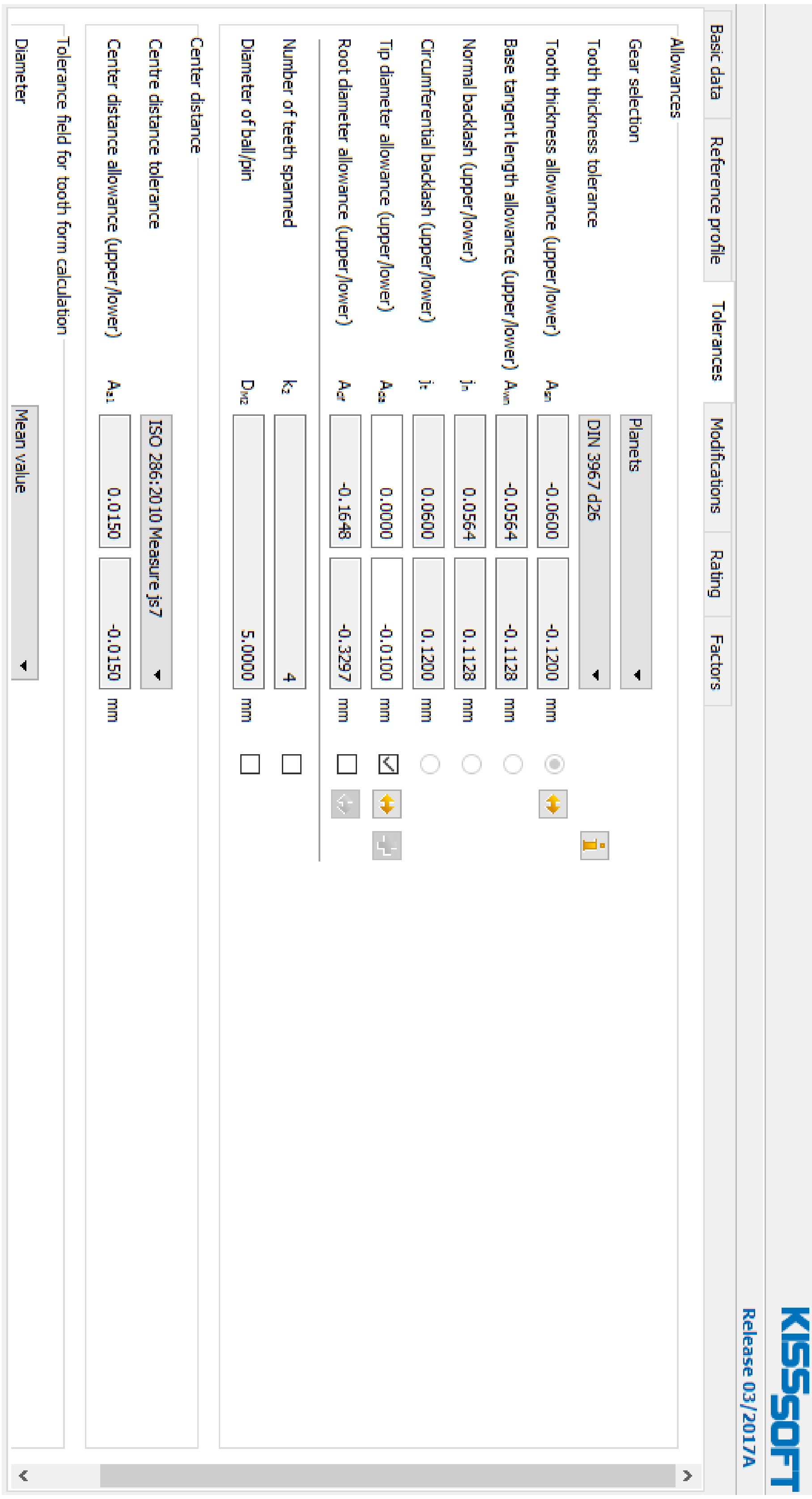

Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
Modifications
Type of tip modification
none
Tip modification
mm
Chamfer / tooth end
0.0000
mm
45.0000
Gear
Flank
Type of modification
Value [µm]
Factor 1
Factor 2
Statu
Inforr
Comment
Release 03/2017A

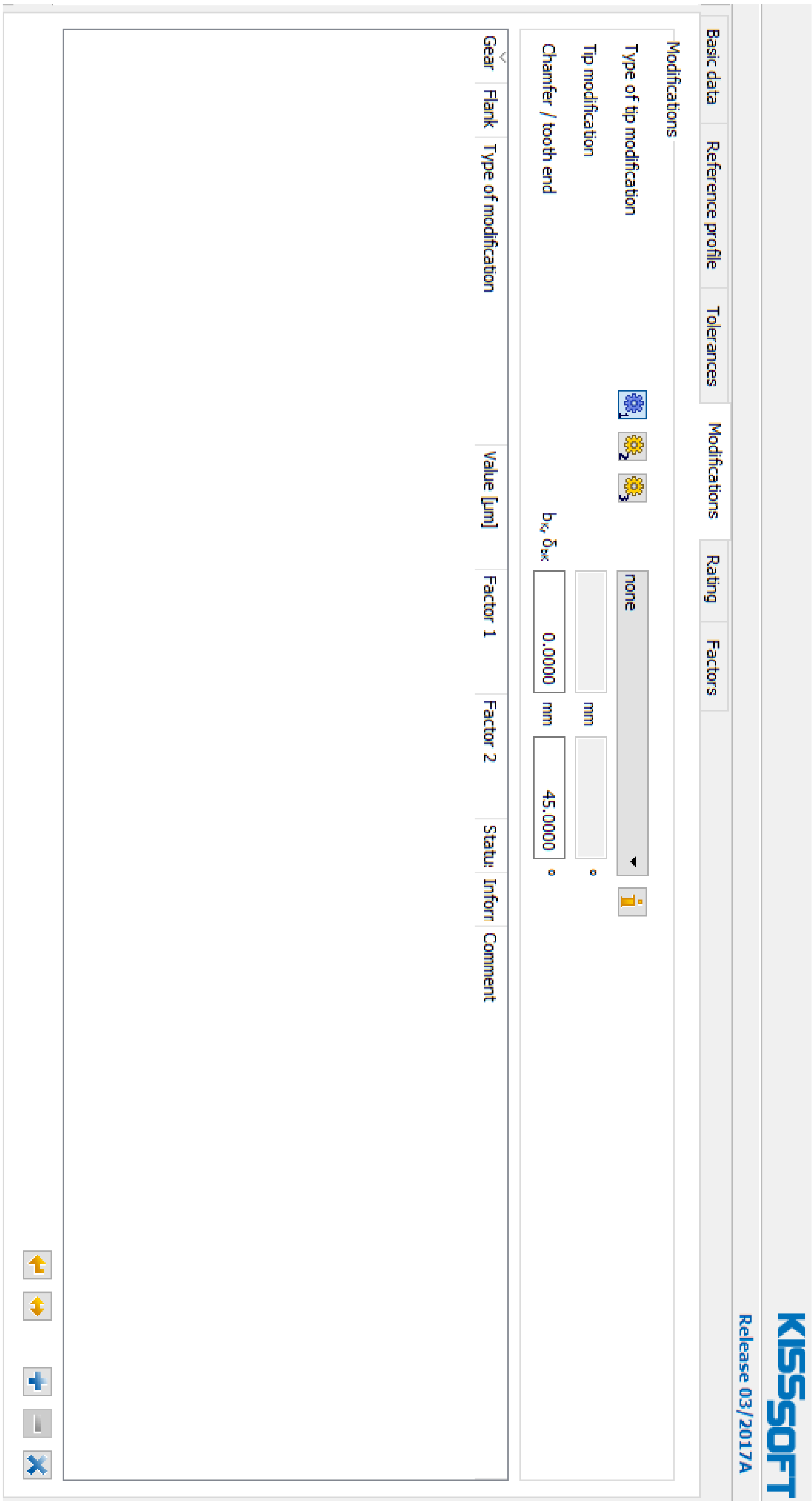

KISSSOFT
Release 03/2017A
Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
Strength
Calculation method
AGMA 2001-C95
Calculation method for micropitting
ISO TR 15144
Calculation method tooth flank fracture
No calculation
Driving gear
Sun gear
Working flank, sun
right flank
Sense of rotation sun
clockwise
Reference gear
Sun gear
Details...
Power
P
42.0251
kW
Torque
T1
637.0000
Nm
Speed
n1
630.0000
1/min
Required service life
H
5000.0000
h
Overload factor
Ko
1.3500
Load spectrum
Single stage load (no load spectrum)
Don't consider load spectra
Frequency [%]
Power factor
Speed factor
1
100.000000
1.0000
1.0000
Input
Power
Factor

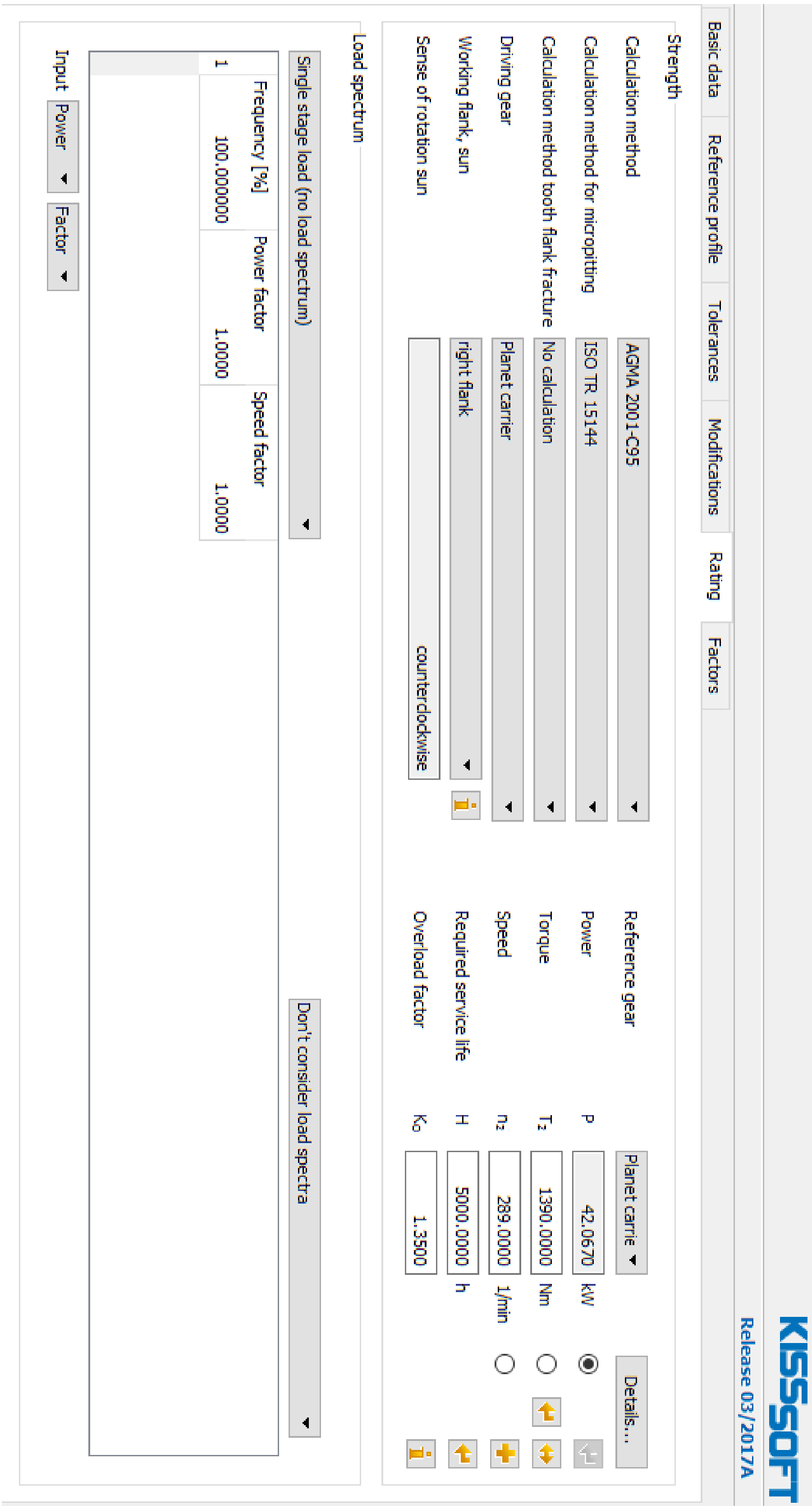

Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
KISSSOFT
Release 03/2017A
General factors
Dynamic factor K_V
1.0536
1.0528
Transverse load factor $K_{H\alpha}$
1.0000
1.0000
Mesh load factor (distribution of power where numerous meshings are involved)
Own Input
K_γ
1.3000
Alternating bending factor (mean stress influence coefficient)
Predefined
Y_M
1.0000
0.7000
1.0000
Face load factor
Calculation according calculation method
$K_{H\beta}$
1.1608
1.1641

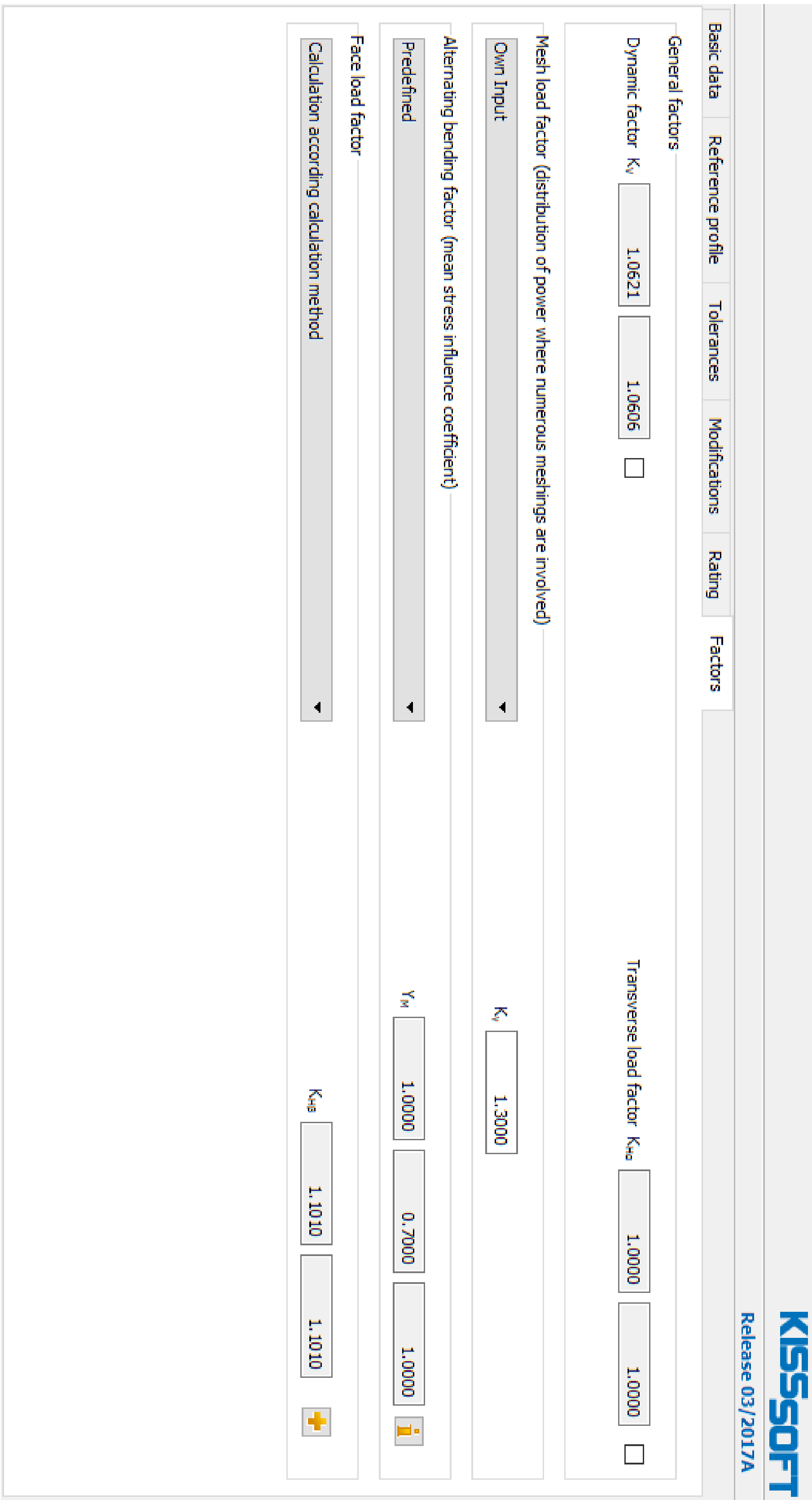

KISSSOFT
Release 03/2017A
Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
System data
Normal module
mn
2.5000
mm
Sun
spur gear
Pressure angle at normal section αn
20.0000
°
Helix angle at reference circle β
0.0000
°
Center distance
a
73.7500
mm
Number of planets
5
Geometry
Sun
Planets
Internal gear
Details...
Number of teeth
z
35
21
-80
Facewidth
b
17.0000
17.0000
17.0000
mm
Profile shift coefficient x*
1.2591
0.5000
-0.5000
Quality (AGMA 2000)
Q
11
11
11
Material and lubrication
Sun
Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened, AGMA 2001-C95
Planets
Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened, AGMA 2001-C95
Internal gear
Steel, Grade 2, HRC58-64(AGMA), Case-carburized steel, case-hardened, AGMA 2001-C95
Lubrication
Oil: ISO-VG 220
Oil bath lubrication


KISSSOFT
Release 03/2017A
Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
Gear selection
Planets
Machining
Final machining (without pre-machining)
Final treatment tool
Tool selection
Reference profile gear
Input
Factors
Select reference profile
1.25 / 0.38 / 1.0 ISO 53:1998 Profil A
Dedendum coefficient
1.2500
Root radius coefficient
0.3800
Addendum coefficient
1.0000
Protuberance height coefficient
0.000
Protuberance angle
0.0000
Tip form height coefficient
0.0000
Ramp angle
0.0000
topping tool
Tip alteration of gear
-0.6480
mm


Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
KISSSOFT
Release 03/2017A
Allowances
Gear selection
Planets
Tooth thickness tolerance
DIN 3967 d26
Tooth thickness allowance (upper/lower)
-0.0600
-0.1200
mm
Base tangent length allowance (upper/lower)
-0.0564
-0.1128
mm
Normal backlash (upper/lower)
0.0564
0.1128
mm
Circumferential backlash (upper/lower)
0.0600
0.1200
mm
Tip diameter allowance (upper/lower)
0.0000
-0.0100
mm
Root diameter allowance (upper/lower)
-0.1648
-0.3297
mm
Number of teeth spanned
4
Diameter of ball/pin
5.0000
mm
Center distance
Centre distance tolerance
ISO 286:2010 Measure js7
Center distance allowance (upper/lower)
0.0150
-0.0150
mm
Tolerance field for tooth form calculation
Diameter
Mean value


Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
Modifications
Type of tip modification
Tip modification
Chamfer / tooth end
none
mm
mm
0.0000
45.0000
Gear
Flank
Type of modification
Value [µm]
Factor 1
Factor 2
Comment
KISSSOFT
Release 03/2017A


KISSSOFT
Release 03/2017A
Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
Strength
Calculation method
AGMA 2001-C95
Calculation method for micropitting
ISO TR 15144
Calculation method tooth flank fracture
No calculation
Driving gear
Planet carrier
Working flank, sun
right flank
Sense of rotation sun
counterclockwise
Reference gear
Planet carrie
Details...
Power
P
42.0670
kW
Torque
T2
1390.0000
Nm
Speed
n2
289.0000
1/min
Required service life
H
5000.0000
h
Overload factor
Ko
1.3500
Load spectrum
Single stage load (no load spectrum)
Don't consider load spectra
Frequency [%]
Power factor
Speed factor
1
100.000000
1.0000
1.0000
Input
Power
Factor



KISSSOFT
Release 03/2017A
Basic data
Reference profile
Tolerances
Modifications
Rating
Factors
General factors
Dynamic factor K_v
1.0621
1.0606
Transverse load factor $K_{H\alpha}$
1.0000
1.0000
Mesh load factor (distribution of power where numerous meshings are involved)
Own Input
K_γ
1.3000
Alternating bending factor (mean stress influence coefficient)
Predefined
Y_M
1.0000
0.7000
1.0000
Face load factor
Calculation according calculation method
$K_{H\beta}$
1.1010
1.1010