
High-speed Ravigneaux Planetary Gearbox for an Electric Vehicle:

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Dissertation submitted to
Faculdade de Engenharia da Universidade do Porto
for the degree of:

Mestre em Engenharia Mecânica

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Porto, 2022

The work presented in this dissertation was performed at the
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Abstract

Nowadays, environmental concerns and strong governmental regulations have pushed electric vehicles to be presented as a viable solution to consumers while being allegedly in line with the new global paradigm of sustainability.

In order to simplify the powertrain system, battery electric vehicles currently often use a single driving motor with a fixed ratio transmission; however, the performance requirements of the power battery pack and driving motor are relatively high. Battery electric vehicles can be fitted with multispeed gearboxes to increase the effectiveness of an electric driving system while also meeting vehicle driveability criteria and lowering manufacturing costs.

A two-speed transmission system appears to be suitable for BEVs in terms of dynamic performance, energy efficiency, and cost-saving. It can improve the gradeability at low-speed and efficiency at high speed, thereby reducing the performance requirements of the power battery pack and driving motor.

This thesis focuses on designing a two speed planetary transmission for an electric vehicle. The planetary arrangement offers significant weight and space reduction over parallel shaft gears, resulting in a compact design suitable with electric car powertrains.

Keywords: Electric Vehicles, Multispeed, Planetary transmissions, Ravigneaux

Resumo

Atualmente, preocupações ambientais e fortes incentivos governamentais têm permitido que os veículos elétricos sejam apresentados como uma alternativa viável para os consumidores, ao mesmo tempo que se enquadram nos novos padrões de sustentabilidade globais.

De forma a simplificar o sistema de tração, os veículos elétricos normalmente usam um único motor elétrico com uma razão de transmissão fixa; contudo, isto requer um maior desempenho por parte das baterias e do motor. Os veículos elétricos podem ser equipados com transmissões de múltiplas velocidades de forma a aumentar o rendimento do sistema de propulsão elétrico, enquanto garante os critérios de desempenho e diminui os custos de produção.

Uma transmissão com duas velocidades aparenta uma solução apropriada para veículos elétricos em termos de desempenho dinâmico, eficiência energética e redução de custos. Permite melhorar a capacidade de subir rampas a baixas velocidades e a eficiência energética a altas velocidades, o que leva a uma redução dos requisitos de desempenho do motor e das baterias.

Esta tese foca-se em desenvolver uma transmissão planetária de duas velocidades, para um veículo elétrico. A arquitetura planetária garante a redução de peso e espaço em comparação com as caixas de velocidade tradicionais, resultando numa solução compacta apropriada a sistemas de propulsão de veículos elétricos.

Palavras-chave: Veículos elétricos, Múltiplas velocidades, Transmissões planetárias, Ravigneaux

Резюме

В настоящее время забота об окружающей среде и сильные правительственные стимулы позволили представить электромобили как жизнеспособную альтернативу для потребителей, отвечающую новым мировым стандартам устойчивости.

Чтобы упростить систему тяги, в электромобилях обычно используется один электродвигатель с фиксированным передаточным числом, однако, это требует большей производительности от аккумуляторов и двигателя. Электромобили могут быть оснащены многоскоростными трансмиссиями, чтобы повысить производительность электрической силовой установки, при этом соблюдая критерии производительности и снижая производственные затраты.

Двухступенчатая коробка передач представляется подходящим решением для электромобилей с точки зрения динамических характеристик, энергоэффективности и экономии средств. Улучшает способность преодолевать подъемы на низких скоростях и энергоэффективность на высоких скоростях, что приводит к снижению требований к производительности двигателя и аккумуляторов.

В данной диссертации основное внимание уделяется разработке двухступенчатой планетарной коробки передач для электромобиля. Планетарная архитектура гарантирует экономию веса и места по сравнению с традиционными коробками передач, что приводит к компактному решению, подходящему для силовых установок электромобилей.

To my mother Karina

‘The only true wisdom is in knowing you know nothing.’

Socrates

Acknowledgements

I want to express my gratitude to my supervisor, Professor José Almacinha, and my co-advisor, Professor Jorge Seabra, for giving me the chance to work on this project and for their support, direction, comments, and advice.

I must acknowledge the Faculdade de Engenharia da Universidade do Porto, CETRIB (Unit of Tribology, Vibrations) for the chance to complete this dissertation and the conditions offered.

I would also like to thank my friends and girlfriend for all the support given, which helped me achieve my goals while having a healthy study/work environment.

My mother and family deserve the deepest gratitude for helping me get my master's degree; without them, I would not be the person I am today.

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Introduction

1.1 Introduction

Given the state of power battery and driving electric motor technology development today, it is essential to pursue a variety of strategies for enhancing powertrain effectiveness in order to increase one-charged mileage and decrease the design requirements of the power battery and driving electric motor in order to reduce manufacturing costs. Numerous studies show that a two-speed transmission system that has been optimized can greatly reduce the size of the powertrain while also making up the deficiency of the driving motor and power batteries.

1.2 Objectives

This thesis is focused on the development of a high speed planetary transmission for an electric vehicle, based on the general design of a two speed Ravigneaux transmission. The main objectives are:

1. Understand the role of planetary type transmissions in the growing technology of multi speed transmissions intended for electric vehicles;
2. Make a dynamic performance analysis on a powertrain that incorporates a two speed transmission, with the goal of estimating the needed gear ratios for a set of requirements;
3. Build the dynamic model of the main components of the Ravigneaux transmission and define its kinematics;
4. Design the transmission in **KISSsys** and size its gears, including the final drive, according to strenght calculations performed in **KISSsoft**;
5. Choose the transmission's bearings, power couplings, shafts, as well as other components, based on appropriate **KISSsoft** calculations;
6. Design the 3D CAD model of the transmission in **Solidworks**, giving special attention to the complexity of assembly of a planetary transmission;
7. Present the 2D technical drawings of the transmission, and relevant individual components.

1.3 Dissertation Layout

The dissertation was organized into chapters in the manner deemed to be most appropriate. The chapters are followed by the references, and the appendices are located at the very end. The following is a brief overview of each chapter:

Chapter 2

This chapter provides the framework for the research of this thesis. Initially, an overview on the need for the pursue of strategies for enhancing powertrain effectiveness is done, followed by the required background information on relevant aspects of the advantages of multi-speed transmissions in electric vehicles. This is followed by a detailed literature review that identifies the state-of-the-art in multi-speed electric vehicles.

Chapter 3

The general design characteristics are evaluated. The required vehicle specifications and vehicle performance are reviewed. A dynamic performance analysis is performed in order to estimate the needed gear ratios to satisfy the set requirements.

Chapter 4

The working principles of the two speed planetary transmissions are detailed. Following that, the kinematics of the transmission are mathematically defined.

Chapter 5

The mating conditions of planetary transmissions are detailed, specifically the ones necessary for six pinion Ravigneaux architectures. The sizing of gears is done according to the kinematics, mating conditions and strenght calculations performed in **KISSsoft**.

Chapter 6

In this chapter, the shaft design and bearing selection is detailed. Special attention is given to the planet's bearings which are the critical bearings in a planetary transmission. The torque transmitting splines for each shaft are also reviewed.

Chapter 7

Different types of one-way clutches are analysed, and appropriate selection of a one-way clutch is done, which is a component that plays a role in the gear shifting mechanism of the transmission proposed here.

Chapter 8

This chapter is dedicated to the design of the of the band brakes, which were intended to be electrically actuated. The choice of a suitable electric motor is addressed. Furthermore, the assembly of the mechanism is also detailed.

Chapter 9

In this chapter, the exploded view of the full transmission is shown along with the list of its parts, and the complete assembly procedure is detailed.

Background Information and Literature Review

2.1 Background information

2.1.1 History of EVs

Gustave Trouvé, a french man, created the first electric car in 1881. It was a tricycle with a 0.1 horsepower DC motor that was powered by lead-acid batteries. The entire vehicle, including the driver, weighted around 160 kg. Two British professors created a vehicle comparable to this in 1883. Because the technology was not yet sophisticated enough to compete with horse carriages, these early realizations received little public notice. Potential buyers were uninterested in 15 km/h speeds and a 16 km range. The 1864 Paris to Rouen race changed everything: the 1135 kilometers were covered in 48 hours and 53 minutes by an electric vehicle, with an average speed of 23.3 km/h. This speed was significantly superior to anything else that could be done [1].

The first commercial electric vehicle was Morris and Salom's Electroboat. This vehicle was used as a taxi in New York City by a company that was founded by the vehicle's creators. Despite a greater purchasing price, the Electroboat proved to be more lucrative than horse cabs. It might be utilized for three 4-hour shifts with 90-minute breaks in between. It had two 1.5 horsepower motors that gave it a top speed of 32 km/h and a range of 40 km [1].

The introduction of regenerative braking by a M.A. Darracq on his 1897 coupé was the most significant technical achievement of the era. This technology allows the vehicle's kinetic energy to be recovered while braking and recharging the batteries, considerably increasing the operating range. It is one of the most important contributions to electric and hybrid car technology since it adds more to energy efficiency in urban driving than anything else [1].

In addition, among the most significant electric vehicles of that era was the first vehicle ever to reach 100 km/h. It was "La Jamais Contente" (figure 2.1) built by a french man called Camille Jenatzy.

Electric vehicles began to fade as gasoline vehicles became more powerful, flexible, and, above all, easier to handle. Their exorbitant price didn't help, but it was their limited driving range and performance that truly hurt them when compared to gasoline-powered vehicles. Around 1905, the last commercially significant electric vehicles were released.

In the 1960s and 1970s, there was a resurgence of interest in electric vehicles, but it wasn't until the early 1990s that a more significant push for driving electric vehicles was realized. Regulations, acts, and laws were mostly responsible for this increase. The first mass-produced all-electric car, the EV1, was built by General Motors (GM) in 1996. The EV1 had a top speed of 130 km/h and a range of around 100 km. However, by 2002, GM



Figure 2.1: Camille Jenatton with his wife on the victory parade on 1 May 1899 after the 100 km/h record-breaking run on 29 April 1899 [2].

had abandoned the EV1 due to a lack of profit in the electric vehicle market. Several other electric vehicles of the time met the same fate.

Approximately half a decade after the EV1's cancellation, GM along with other significant vehicle manufacturers, like as Nissan, were looking to re-enter the market. Elon Musk and Tesla Motors (formed in 2003) were in the midst of mass production of the Roadster, an all-electric sports automobile capable of traveling 320 kilometers on a single charge. With a resurgence of interest, the focus shifted to mass-market, low-cost electric vehicles. GM began work on the Chevrolet Volt, a plug-in hybrid electric vehicle with a battery and a small gasoline engine, while Nissan prepared the LEAF, their all-electric vehicle. Electric vehicles appeared to be on the rise, thanks to increased competition and better technology.

Today's electric vehicle (EV) technology can be broken down into four different types. Battery Electric Vehicles (BEVs), Plug-In Hybrid Electric Vehicles (PHEVs), Extended Range Electric Vehicles (EREVs), and Hybrid Electric Vehicles (HEVs). Taken together, EVs represent a significant portion of the world's mobility. In fact, they are currently experiencing a fast market growth that will be analysed in the following section.

2.1.2 Global EV Growth

Due to their increasing practicality, EVs are rapidly gaining market dominance. This is aided by recent technological developments, particularly in the fields of energy storage and charging systems. In contrast to conventional vehicles with internal combustion engines (ICEs), the market share of EV is predicted to grow further in light of recent advances in global emission standards. The International Energy Agency’s (IEA) latest report (Global EV Outlook 2021, [3]) showed that electric car registrations increased 41 % in 2020, despite the pandemic-related worldwide downturn in car sales in which global car sales dropped 16 %. Sales of electric cars were 4.6 % of total car sales. It also stated that there were 10 million electric cars on the world’s roads at the end of 2020 (a 43 % increase over 2019), following a decade of rapid growth (see figure 2.2). For the first time, Europe surpassed the People’s Republic of China (“China”) as the world’s largest EV market, which shows Europe’s clear commitment to the transition to EVs.

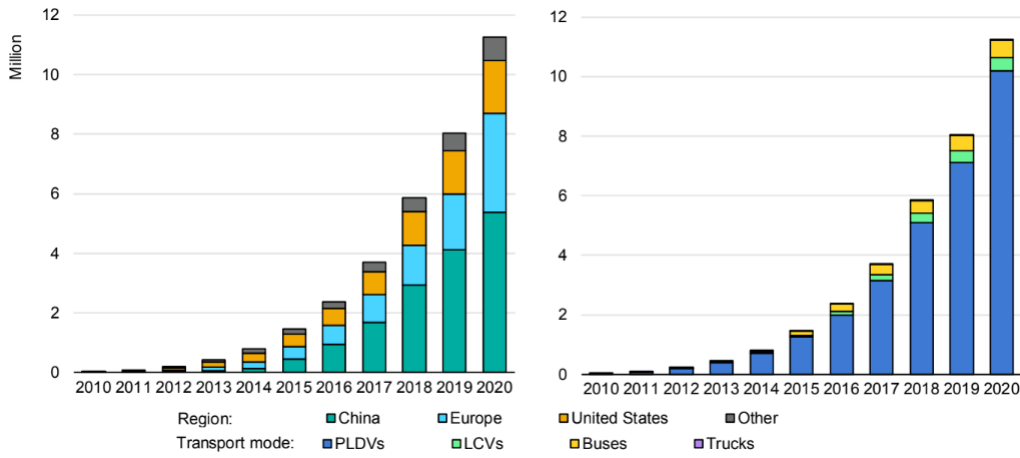


Figure 2.2: Global electric vehicle stock by region (left) and transport mode (right), 2010-2020 [3].

The cost of batteries and electric vehicles is decreasing [3]. Infrastructure for charging is growing (see figure 2.3). This approach encourages the electrification of two- and three-wheelers, light-duty vehicles (LDVs) (cars and vans), taxis and shared vehicles, buses, and heavy-duty vehicles with short-range requirements, such as city deliveries.

The number of electric vehicle models offered to customers is continuing to grow. The IEA’s report states that vehicle manufacturers have declared more aggressive electrification plans. In fact, 18 of the top 20 vehicle manufacturers in the world, which account for over 90 % of new car registrations in 2020, have indicated plans to expand their model portfolio and swiftly scale up production of light-duty electric vehicles. Some examples of these announcements include: Volvo has announced they will only sell electric cars from 2030 [4]; in Europe, Ford will also only be selling electric cars from 2030 [5]; Volkswagen aims for 70 % electric car sales in Europe, and 50 % in China and the United States [6]; General Motors plans to offer only electric LDVs by 2035 [7]. Electric heavy-duty vehicle model availability is also expanding, with four major truck manufacturers projecting an all-electric future.

Governments around the world paid USD 14 billion to encourage the purchase of electric vehicles, which increased by 25 % in 2019 due to stronger incentives in Europe. Nonetheless, over the last five years, the share of government subsidies in total spending on electric cars has fallen, indicating that EVs are becoming more appealing to customers.

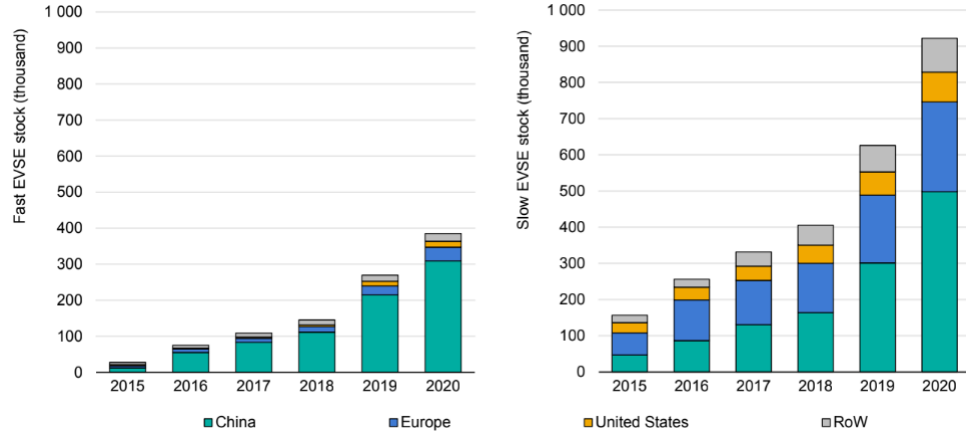


Figure 2.3: Stock of fast and slow publicly accessible chargers for electric light-duty vehicles, 2015-2020 [3].

Many governments have also committed to electrification targets or ICE bans for cars, or even announced net-zero pledges. The countries which have these targets in law or have proposed legislation are shown in figure 2.4, along with their electrification goal.

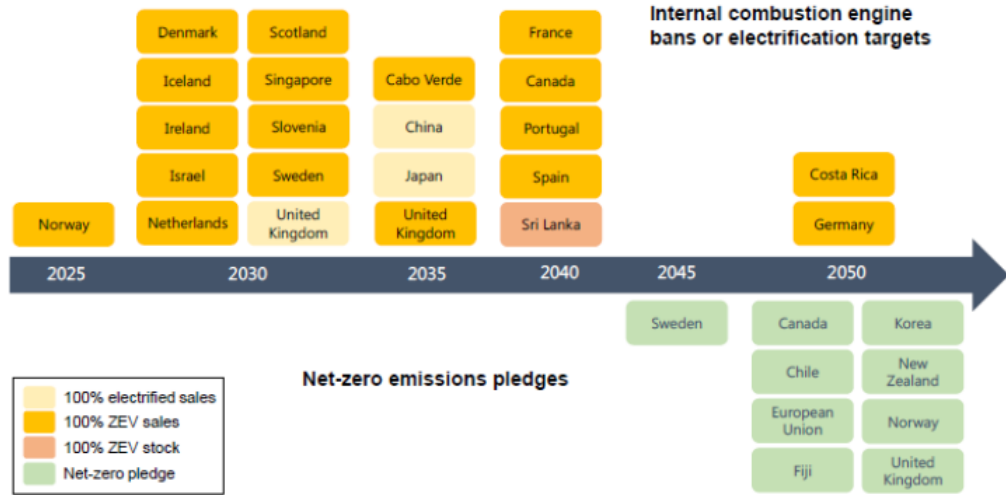


Figure 2.4: Electrification targets of various countries [3].

2.1.3 Challenges from EV's Popularity

The growing popularity of electric vehicles has brought with it a slew of new issues that must not be overlooked. These issues can be divided into three categories: challenges in ensuring the long-term viability of EV production, challenges in meeting the rising demand for electricity as a result of EV penetration, and challenges in managing the ICE-to-EV transition in terms of vehicle numbers and the industry ecosystem. In the context of EV production sustainability, [8] stated that, while EVs create zero emissions, the same cannot be said for their manufacturing. This is due to the fact that the manufacturing of motors and batteries uses a substantial amount of depletable elements, such as heavy rare earth minerals. Furthermore, when compared to the production of ICE vehicles, the method produces higher levels of heavy metal pollutants, such as lead, nickel, and molybdenum, which has been argued to be harmful to human health. The carbon impact of these

activities is now relatively high due to their localization, according to a study by Steve Voller *et al.* [9]. These operations are currently concentrated in China, South Korea, and Japan, where a large amount of the power is generated from fossil fuels, leaving a large carbon footprint. It also suggested that the trend of growing battery size be stopped because it is linked to the previously noted carbon footprint issue. This recommendation can be realized by increasing the efficiency of EV powertrains.

The growing popularity of electric vehicles also produces an increase in electricity demand, which leads to the second problem posed by the growing popularity of electric vehicles. According to [10], the amount of electricity consumed by EVs on a daily basis is around the same as the average daily electricity consumption of a typical american household. This study also stated that in the UK, when electric vehicles reach 20 % of its vehicle fleet, peak electricity demand is predicted to rise by 36 %. To meet this challenge, two fundamental strategies must be seriously considered: efficient power grid management, which can be accomplished through the use of vehicle-to-grid technology or extensive battery swapping, which is already being done by a chinese EV company named NIO [11], and efficient, sustainable, and cost-effective EV powertrains, which include the use of optimal motors, transmissions, and batteries, with possibly significant carry-over technologies from ICE vehicles.

Finally, efficient management of the ICE-to-EV migration is essential in order to have a smooth transition phase. If the overall number of existing ICE vehicles, especially those with low emission standards, is not dramatically decreased, simply increasing the market proportion of EVs would not suffice. Furthermore, any migration must be managed in the context of the current industrial supply chain. For example, a suitable strategy must be prepared for current ICE-related production units that are likely to be shut down once EVs overtake ICE vehicles in terms of market dominance. One strategy in this regard is to repurpose existing manufacturing operations to focus on electric vehicle-related items. This is less popular, however, due to the high expenses of educating existing staff and updating plants [12]. EV powertrain retrofitting of existing ICE vehicles is one approach that can be used to speed the ICE-to-EV transition. The objective is to not only speed EV market penetration, but also to make use of existing resources, in this case, current ICE vehicles on the road, so that the net number of vehicles on the road does not rise. This approach, however, presents challenges in terms of public acceptance and vehicle homologations. In general, public acceptability of EV retrofitting can be gradually increased by effective communication between the government, technology providers, and the general public, emphasizing the benefits in terms of sustainability, long-term financial savings, and lower emissions. Simultaneously, EV powertrain compatibility and flexibility should be increased in order to lower initial retrofitting costs. Such powertrains can also help with homologations, which is a key roadblock in the implementation of EV retrofitting.

It can be noted above that, while increasing EV market share reduces global carbon emissions from one perspective, it nevertheless poses numerous key economic and overall sustainability issues. If these issues are not addressed, the benefits of electric vehicles will be negated.

2.1.4 EV Powertrains

EV powertrains are made up primarily of batteries, an electric motor, and a transmission, and their efficiency and practicality may be measured. A high-efficiency EV powertrain means that the vehicle's energy consumption (kWh) per distance (km) may be kept as low as possible, allowing it to travel further. The goal is to make the powertrain components as cost-effective as possible, with low production and operating (i.e.,

maintenance) costs, and to make them sustainable, i.e., with a low carbon footprint from manufacture to application.

The battery, which is the initial component of an EV powertrain, stores electricity for use by the electric motor. To guarantee that the powertrain is both efficient and practical, the battery must have a high energy density, which allows it to store large amounts of electric power without adding to its weight.

The electric motor is the next main component of an EV powertrain, and it is responsible for transforming the electricity from the battery into mechanical power to move the car utilizing the electromagnetic induction method. An inverter regulates the required current flow from the battery to meet the driving circumstances and controls the motor. Permanent magnet synchronous motors (PMSMs) and induction motors (IMs) are the two most common types of motors used in electric vehicles. Permanent magnetic materials in either the stator or the rotor generate the magnetic field required to rotate the rotor in PMSMs. In IMs, on the other hand, the electromagnetic field is generated by current flowing through the rotor conductor. The differences between these types of motors will be discussed in detail later on when choosing the electric motor that will be part of the design. Compared to ICEs, both types have a smaller volume and a higher power to weight ratio, which contribute positively to the power consumption of an EV. Not only that, but these motors have a strong torque capability at low rpm, which eliminates the need for high gear ratios for vehicle start-stop. This explains why most existing electric vehicles lack multispeed gearboxes. Figure 2.5 shows the typical performance characteristics of electric motors and gasoline engines, in which the electric motor's high torque availability at low rpm can be seen. The gasoline engine, on the other hand, can only operate from an idle speed, from which the torque-speed curve evolves and presents a relatively flat profile (compared with an ideal one). As a result, a multigear transmission is frequently used to modify it [1]. Electric motors have extremely high peak efficiency, reaching values of over 85 % [13]. However, such efficiency is only achievable within a narrow range of motor speeds; thus, under a variety of driving circumstances, the powertrain's efficiency typically falls far below that. Not only that, but the manufacturing of electric motors necessitates the use of heavy rare earth minerals, resulting in high costs and less long-term production. As a result, sustainable and cost-effective alternatives to realizing the true potential of electric vehicles in terms of range and performance are desired.

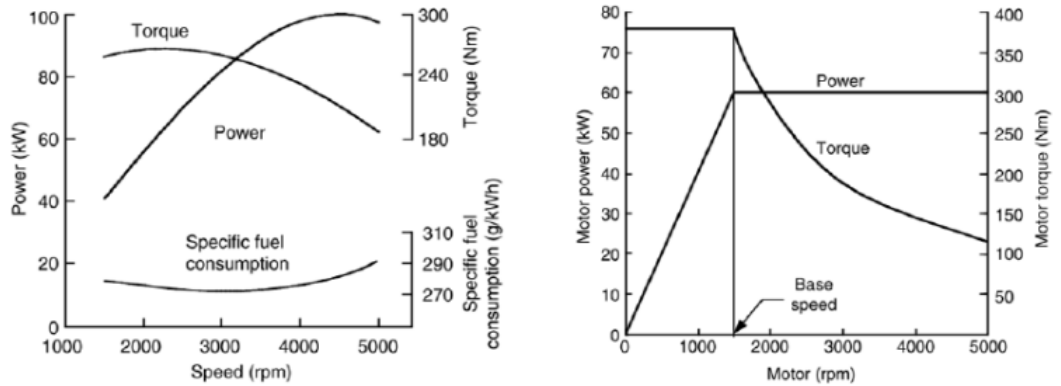


Figure 2.5: Typical performance characteristics of gasoline engines (left) and electric motors for traction (right) [1].

The transmission is the final significant component of an electric vehicle's powertrain, and it is responsible for efficiently transmitting power from the motor to the wheels. The transmission utilized in present EVs usually only provides a single speed ratio due to the characteristics of the basic electric motors employed. The fundamental advantage of single-speed transmissions is their simple structure, which results in reduced production and maintenance costs. However, this restricts the electric motor's ability to work optimally under a variety of driving circumstances. As a result, EVs' true potential in terms of driving range and power consumption is difficult to realize. There is a clear need of optimizing the power flow of EVs, in order for the electric motor to have the ability to work more efficiently and effectively, resulting in reduced battery power usage.

2.2 Literature Review

2.2.1 Power Flow Optimization in EVs

It's difficult to ensure that a single motor EV powertrain can perform ideally for diverse driving conditions, especially when the vehicle is traveling at high speed and low load, and as a result, their efficiency typically drops to around 60 % [14]. This will consume a lot of energy and restrict the driving range. Single-speed transmission BEVs frequently use higher-power electric motors to match the performance of two-speed transmission BEVs. As a result, BEV batteries must increase capacity and output power to fulfill the demands of electric motors. Allowing various power flow arrangements in the powertrain is one option for avoiding this. Multispeed transmissions can prevent the electric motor from working in the low-efficiency speed range, allowing it to work as much as possible in the high-efficiency speed range, decreasing losses and increasing transmission range. In terms of design complexity, an EV's multispeed transmission should be simpler than those seen in current ICE-powered vehicles. This is due to a number of causes, the most important of which is the demand for moving-off parts in traditional ICE cars. A moving-off device, such as a dry friction clutch or torque converter, is usually necessary to facilitate the vehicle's start-stop condition due to the ICE idle speed condition. Moving-off parts are no longer necessary in an EV because the motor's torque is available from as low as 0 rpm. Furthermore, the removal of moving-off elements allows for the implementation of a considerably simpler transmission control algorithm, as the moving-off element is no longer required to produce optimal driving comfort during start-stop conditions (see figure 2.6).

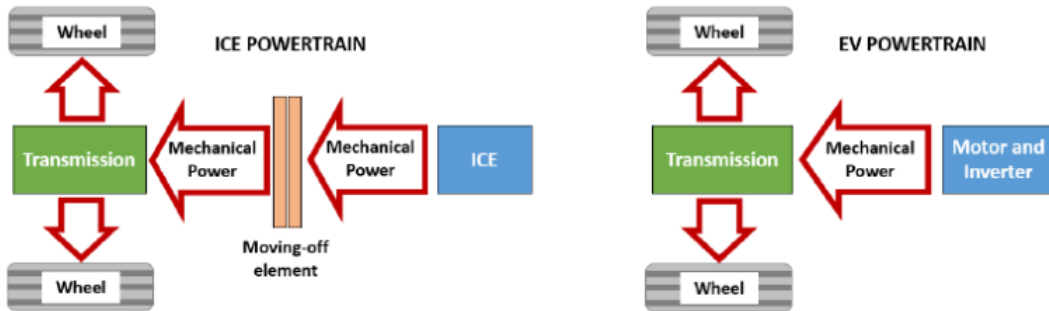


Figure 2.6: Powertrain layout schematics for an ICE vehicle (left) and an electric vehicle (right) [15].

Another way of allowing flexible power flow is utilizing a multi-motor configuration. In [16], a proposed solution where two smaller motors coupled via a planetary gear, was compared with a standard configuration that used one larger motor directly connected to the drive wheels; the results showed that the dual-motor solution resulted in an average efficiency improvement of about 9 %.

In summary, it can be concluded that optimizing EV powertrains requires flexible motor power flow, optimized powertrain components, and control, which can be accomplished by optimizing multi-motor designs or integrating a multispeed transmission in the EV powertrain.

2.2.2 The Benefits of Multispeed Transmissions for BEVs

Similar to ICE vehicles, the major motive for using multi speed discrete transmission in an EV is to provide the best gear ratio in the powertrain so that the motor can work efficiently in a variety of driving circumstances. Because EV motors can produce significant torque output at low rpm, the number of gears in an EV is predicted to be relatively low, possibly as few as two, compared to those in an ICE car. Recent studies comparing the efficiency of an EV powertrain with a single-speed gearbox vs a two-speed transmission are detailed in [17-21]. In the work done by Nikolay Hinov *et al.* [17], a regular electric vehicle model was simulated and the comparison was carried out using simulation model-based estimation, which took into account vehicle parameters, reference motor data, and three driving cycles: the New European Driving Cycle (NEDC), the Worldwide Light Duty Test Cycle (WLTC), and the US Environmental Protection Agency (EPA) Federal Test Procedure for city driving (FTP-75). The results showed that a dual-speed gearbox led to better efficiency, within the range of 1.7 % to 2.4 %. It should be noted, however, that [17] focused solely on obtaining gear ratios for powertrain efficiency, with little regard for driving performance.

In the work by Antti Ritari *et al.* [18], the comparison was carried out on an electric bus model with a two-speed dual clutch transmission (DCT) that works in a specialized city driving cycle. Since it can pre-select the next gear before engaging the second clutch, a DCT provides quick gear shifting. A single clutch automated manual transmission (AMT) and a traditional automatic transmission do not have this capacity. The results showed that the efficiency improvement was in the range of 2 % to 3.2 %, which is similar with the findings in [17], and that using a two-speed transmission allowed for the use of a less expensive motor with a narrow high efficiency range.

Bingzhao Gao *et al.* [20] proposed a two-speed automatic manual transmission for an electric passenger vehicle. Vehicle performance was compared between a fixed-ratio gearbox and the proposed 2-speed automatic manual transmission in this study. To improve driveability and efficiency, the gear ratio and gearshift schedules were initially optimized using a dynamic programming process that included three different types of driving cycles. The comparison results reveal that the acceleration time from 0 to 80 km/h is lowered by 1.1 seconds, and the top speed is increased by 31.1 % from 113 km/h to 148 km/h. In addition, the average energy consumption is reduced by 7–11 % over the three drive cycles studied, implying that the driving range can be increased.

Senqi Tan *et al.* [19] simulated an electric dump truck operating on a specific route, with a defined starting point and end point without complex traffic conditions. This study compared the energy consumption using a two speed transmission consisting of two planetary sets against a single speed transmission. The findings show that the two-speed gearbox allowed for an energy consumption reduction of 6.1 %.

The research discussed above shows that multi speed discrete gearboxes have the potential to improve the efficiency of electric vehicle powertrains. However, these studies did not go into enough detail about gear ratio optimization, which is critical for achieving both powertrain efficiency and optimal driving performance. Due to the multi-objective nature of the problem, finding the ideal two-speed gear ratios is difficult. The optimal ratios, for example, should be able to produce desirable driving performance (in terms of acceleration rate and top speed) while consuming the least amount of energy. These goals frequently contradict one another, and they are influenced by a variety of factors such as road gradient and instantaneous vehicle speed. Thus, advanced optimization techniques are required, for instance, the final paper reviewed here, Kihan Kwon *et al.* [21] focused on optimizing the gear ratios of two-speed DCTs (dual-clutch transmissions) based not only on motor efficiency but also transmission efficiency. The transmission efficiency model was created by accounting for losses at the gear meshing, bearings, clutch, and concentric shaft. As a result, different gear ratios provided varying torque transmission efficiency between the input and output shafts. The best gear ratios for the EV powertrain were chosen based on the model, and its performance was compared to a single-speed EV powertrain for the WLTP cycle, which demonstrated a maximum improvement in energy efficiency and dynamic performance of 12.1 % and 10.7 %, respectively.

Besides the strong scientific support behind the benefits of using two-speed transmissions in EV powertrains for improving energy efficiency and dynamic performance, there are automakers that have already implemented a multi speed transmission into their designs. The Porsche Taycan is the world's first production electric vehicle with a multi-speed transmission [22]. The Porsche Taycan, like Tesla's dual-motor vehicles, has one motor on each axle and only one gear on the front motor. The rear motor, on the other hand, has an automatic transmission with two gears. The Taycan's automatic transmission (see figure 2.7) has two gears that are used to achieve distinct performance goals. The first gear has a 15:1 gear ratio and is used to accelerate the vehicle. The car can cruise more efficiently and attain higher top speeds in second gear, which has an estimated 8:1 gear ratio. It features incredible acceleration off the line, sustained acceleration into greater speeds, and an impressive top speed for an electric car.

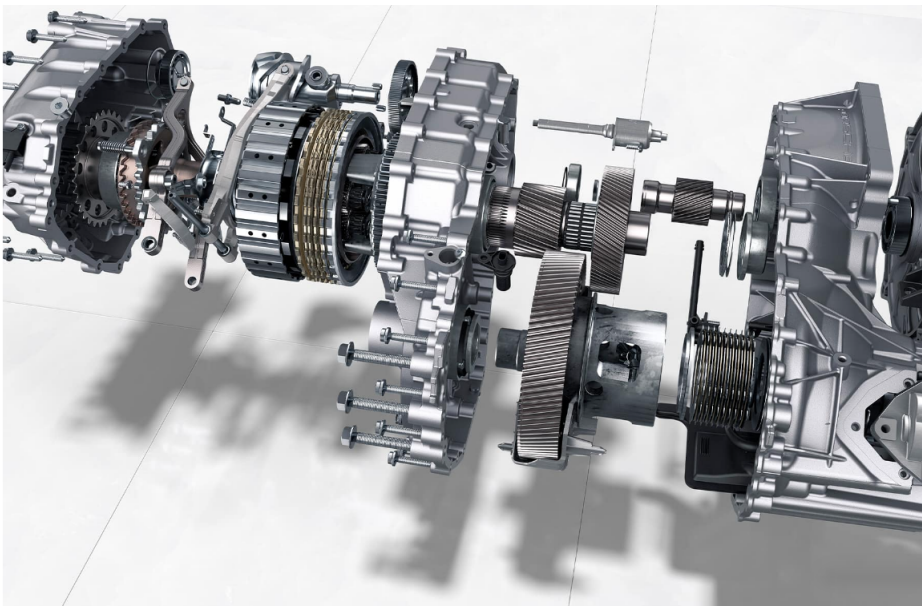


Figure 2.7: Porsche Taycan's two-speed transmission [22].

Multispeed gearboxes are also used in Rimac’s electric supercars and some Formula E cars to achieve the same goals. A multispeed transmission’s advantages may not be limited to luxury brands or race automobiles. In the future, other automakers may be tempted to use multi speed gearboxes in their electric vehicles. ZF, a German auto parts manufacturer, claimed in July 2019 that it had created a novel electric powertrain idea that combines an electric motor and a two-speed automatic gearbox into a single unit [23]. Like its 8HP eight-speed automatic transmission, which is used in everything from Volkswagens and BMWs to Jeeps, RAM trucks, and Aston Martins, the business could conceivably make this equipment available for any automaker to buy and connect into an electric vehicle design. The transmission also provides a large range boost, which could be one of the most appealing features for automakers. When compared to a one-speed transmission, ZF claims that its two-speed unit increases range by up to 5 %. No vehicles are using this yet and it’s somewhat unclear if this is production-ready or just a prototype, but ZF clearly believes there will be a market for this kind of drive unit, otherwise the company would not be developing it.

According to the literature review done above, BEVs with multispeed gearboxes have a number of advantages over single-speed equivalents, including improved dynamic performance, improved electric motor operation efficiency, and reduced weight and volume of powertrains. The next section will go through the many types of multispeed transmissions used in BEVs, as well as their layouts and structures.

2.2.3 Multi speed transmission architectures

Different multiple speed transmission layouts planned for or implemented in modern BEVs are discussed in this section. In general, passenger car transmissions are classified into the following main designs and formats [38]: manual transmissions (MT); automated manual transmissions (AMT); automatic transmissions with various gear ratios which include dual clutch transmissions (DCT) and the conventional automatic transmission (AT); hybrid drives; and mechanical continuously variable transmissions (CVT). Several topologies of multi speed gearboxes used in BEVs are currently being researched, and the ones found to be the most relevant in the literature, will be described. These architectures include automated manual transmissions (AMT) [24, 37], inverse automated manual transmissions (I-AMT) [20], dual-clutch transmissions (DCT) [25, 26], continuously variable transmissions (CVT) [27-30] and planetary type transmissions [31-36], which is a design implemented into ATs and often used in AMTs.

1) Automated manual transmission (AMT)

The AMT, as illustrated in figure 2.8 is a variation of the traditional manual transmission in which electrohydraulic or electromechanical actuators engage the synchronizer(s) and shift gears [37]. Since ICEs cannot work below specific speeds and controlling their speed during gear changes is difficult, clutches or torque converters are required for start-ups, idle running, and gear changing. Electric motors, on the other hand, can be controlled at a wide range of speeds and this distinction allows for the development of innovative electric vehicle transmissions. In comparison to manual transmission (MT), the AMT improves driving comfort and shift quality (minimum shifting time drops from 0.3 seconds to 0.1 seconds with AMT). Furthermore, when compared to automatic transmissions (AT), vehicle dynamic performance and driving quality can be significantly improved for high-end sport cars. It’s worth noting that one of the AMT’s flaws could be a reduction in driving comfort caused by a lack of traction during gear shift actuation. However,

this flaw is not due to the AMT's inherent limits; it may be overcome with careful gear change control [37]. Despite the fact that the AMT has never been widely used in ICE vehicles [38], it is a good contender for EV applications because of its low manufacturing and maintenance costs, simplicity, ease of deployment, and efficiency.

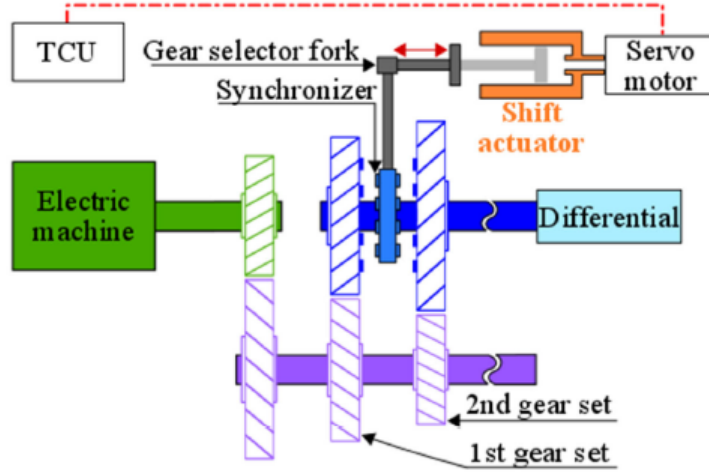


Figure 2.8: N-speed clutch less automated manual transmission (AMT) [39].

2) Inverse-Automated Manual Transmission (I-AMT)

An I-AMT is quite similar to an AMT, except instead of a synchronizer, it has a clutch to link the second gear, as seen in figure 2.9. The synchronizer engages or disengages the first gear, and power is blended in using the clutch. This unique arrangement, as proposed for EVs in [19], decreases torque interruption during shifting by around 50 % when compared to a typical AMT. It may be able to fully compensate torque and achieve seamless gear shifting with enhanced controls [19].

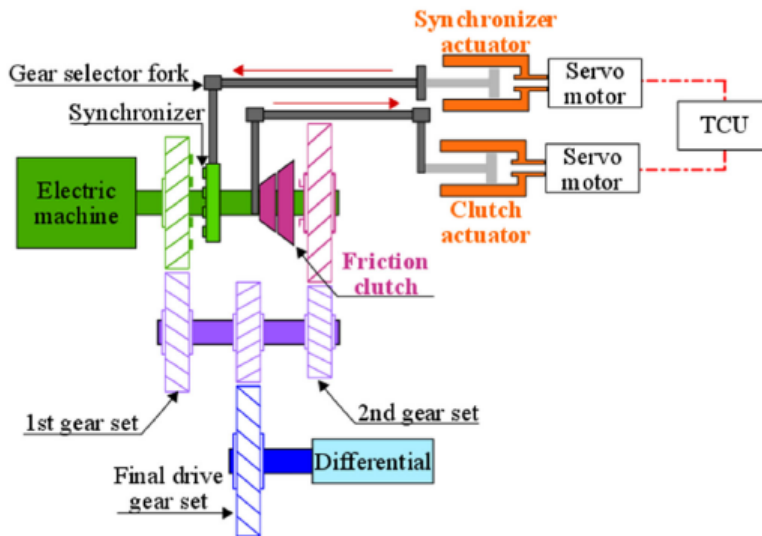


Figure 2.9: Two-speed inverse-automated manual transmission (I-AMT) [39].

3) Dual-Clutch Transmission (DCT)

In the 1940s, dual clutch transmissions were already being developed. The original goal was to equip huge commercial vehicles with technologies that would allow them to drive without losing power. However, serial production was not achieved. Porsche and Audi resurrected this transmission concept in the 1980s, developing a twin clutch transmission for racing cars. Because the systems' control quality was not yet enough, these transmissions were not suitable for serial manufacturing [38].

The purpose of this model was to blend the benefits of manual and automatic transmissions in one vehicle. Manual transmissions provide a high level of efficiency, a gear ratio spread that may be freely selected over a wide variety of gear ratios, as well as sportiness, driving dynamics, and driving pleasure. Traditional automatic gearboxes are distinguished by their ease of use when moving off due to the torque converter, as well as automatic shifting without power interruption [38].

The DCT, as shown in figure 2.10 has two shafts connected to the motor via independent clutches, one for odd gears and one for even gears, as shown in figure 2.10. Smooth shifting can be accomplished by disengaging one clutch while engaging the other and concurrently regulating both clutches [38].

The type of clutch utilized, whether wet or dry, has a significant impact on DCT design and efficiency. A wet clutch has oil on the clutch plates, which provides a strong cooling mechanism but also causes high drag and fluid churning loss, whereas a dry clutch does not have oil on the plates, which reduces loss but also makes it difficult to remove heat [25, 38].

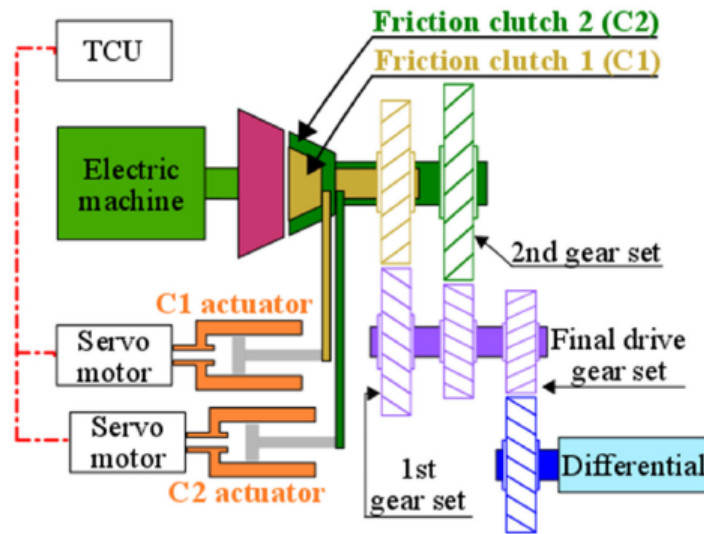


Figure 2.10: Two-speed dual clutch transmission (DCT) [39].

4) Continuously Variable Transmissions (CVT)

As shown in figure 2.11, a CVT uses tapered discs and a belt or chain to allow the gear ratio to adjust continuously from the minimum to the maximum ratio. A CVT can maintain the motor operating at its peak efficiency points for a significantly longer period of time than a traditional gearbox with fixed ratios. The friction between the belt or chain and the taper discs, which are pushed together or apart to regulate the gear ratio, causes significant power loss in CVTs [38]. Power is also required to provide pressure to

the taper discs in order to keep them in place, and the chain or belt may slip in some circumstances, such as when an ICE vehicle starts from a stop, resulting in further loss. Because of all of these losses, the CVT is the least efficient gearbox type. However, many studies have investigated CVTs for EV applications due to their smooth operation, lack of torque interruption, and ability to operate the traction machine over a wide range of speeds [27-30].

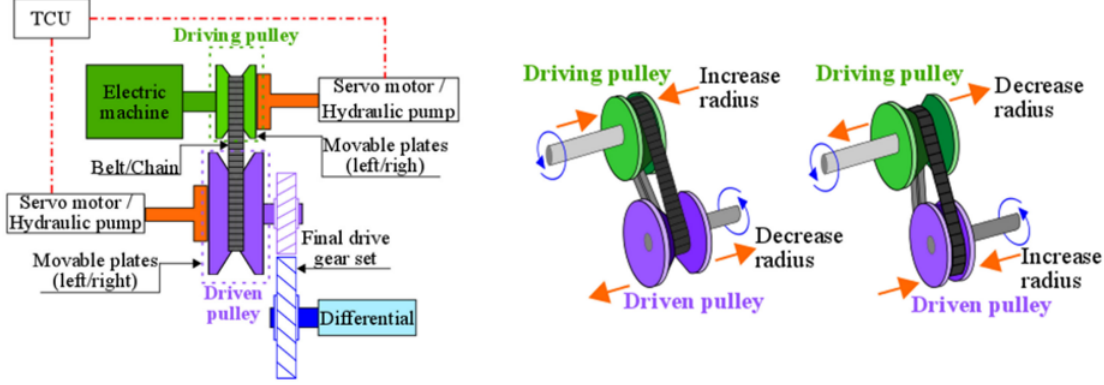


Figure 2.11: Continuously variable transmission (CVT) [39].

5) Planetary Type Transmissions

Epicyclic or planetary gearing arrangements are comprised of four different elements that produce a wide range of speed ratios in a compact layout. These elements are: (1) Annulus, an internally toothed ring gear co-axial with the gear train; (2) Planets, externally toothed gears which mesh with the sun and annulus; (3) Sun gear, an externally toothed ring gear co-axial with the gear train; and (4) Planet Carrier, a support structure for the planets, co-axial with the train. A simple planetary gear arrangement is illustrated in figure 2.12. The name "epicyclic" is derived from the curve traced by a point on the circumference of a circle as it rolls on the circumference of a second fixed circle.

By fixing one of the co-axial members and using the remaining two for input and output, three types of simple single-stage epicyclic gearing are possible. Generally, these are called planetary, star, and solar arrangements [40], which are shown in figures 2.13 to 2.15.

The main benefits of epicyclic gears over parallel shaft gears are significant weight and space reductions. These benefits result from the use of several planets, which enables the load to be transmitted through multiple tooth contacts, and from the coaxial arrangement of the input and output shafts, which results in a more compact layout. Each solar gear teeth engagement would have to support one third of the total load for a planetary system with three planets. As a result, the sun gear would have dimensions that are one-third of the pinion of a parallel shaft gear-train intended to transmit an equivalent torque [41].

It should be pointed out that six planets do not necessarily give a gearing twice the capacity of a similar one having only three planets. It is impossible to guarantee equal loading among more than three planets due to quality and accuracy of the manufacturing process. Due to this, three planet configurations are favored, while there are commercially available systems with up to eight planets.

A second advantage of using multiple planets is that when two or more planets are spaced symmetrically on the carrier, the radial loads of the planets offset each other. As

a result, the bearings and gear housing for the co-axial parts must only be constructed to keep the gearing properly aligned and endure loads imposed on by external factors [41].

Epicyclic gearing also has the advantage that its smaller gears can be manufactured more precisely and easily than parallel shaft gearing's larger ones. They are simpler to work with and harden, and distortion is not a major issue during hardening [41].

Use of smaller components gives lower pitch line velocities. This accounts for epicyclic gear trains being more quiet than parallel shaft gear trains. Having more teeth in mesh, not shifting the load so abruptly, also reduces the noise level.

Single-stage epicyclic gear trains are typically more efficient than identical parallel shaft gear trains because bearing losses and power losses from tooth friction are minimized. Tooth friction losses are approximately proportional to the tooth load and the pitch line velocities. With smaller tooth loads and slower pitch line velocities, the friction loss in epicyclic gears is less than parallel shaft gears running at the same rotational speed with the same load. Bearing losses are dependent on bearing size, which are smaller on epicyclic gears since no tooth reaction loads are carried.

Industrial applications for epicyclic gear systems date back a long way. A sun and planet gear configuration utilized in one of James Watt's early engines was patented in 1781. Epicyclic gear trains were unable to develop because internal gear manufacturing advances lagged behind those of external gears. The performance of epicyclic gear systems decreased at greater loads when industrial applications demanded transmissions with higher power ratings because improper manufacture and assembly methods prevented the planet gears from achieving load equalization [41].

Presently, as manufacturing technologies have been greatly enhanced, planetary gears are frequently used as main reduction gears in propulsion gas turbines for merchant ships. They are widely used in rotor drive gearboxes for helicopter aircraft, and to produce wheel drives for agricultural equipment. They are also becoming a preferred choice in multi speed gearboxes for electric vehicles, because their inherent advantage of providing constant meshing, allows for the elimination of clutches or torque converters. This allows for its perpetual connection to the electric motor and final drive, overcoming the limitation of torque interruption.

A review on proposed designs for multi speed planetary transmissions for EVs was done, and the most relevant works are listed next. In the work by Dele Meng *et al.* [31], an in-wheel two-speed automated manual transmission (IW-AMT) with a selectable one-way-clutch and a dual-stage planetary gear set was proposed. This work intended to evaluate the quality of shifting. In the work by Tao Wang *et al.* [32], a compact two-speed clutchless seamless transmission was designed, which was comprised of a dual-stage planetary gear set with common ring and common sun gears. In the study done by Yang Tian *et al.* [33], a novel two-speed planetary AMT (PAMT) was analysed, which used a one-way-clutch and band brakes to achieve gear shifting. This proposed gearbox inspired this thesis' transmission design and will be analysed in more detail further ahead. Another two-speed PAMT was designed by Nong Zhang *et al.* [34], and it featured a single stage planetary set. In the work of Shengnan Fang *et al.* [35], a two-speed uninterrupted mechanical transmission (UMT) composed of a planetary gear set, brake belt, and centrifugal clutch that realized seamless switching between two gears was proposed. Finally, in the work of Jae-Oh Han *et al.* [36], a two-speed transmission with a double planetary stage architecture was designed, in which the ring gear served as the system input.

Today's modern transmissions use a combination of multiple planetary gear sets referred to as a compound planetary gear set. The most renown and interesting concepts of compound planetary gear sets were studied and will be presented next.

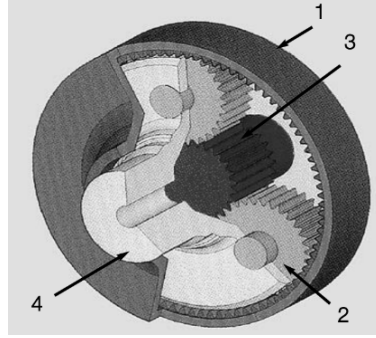


Figure 2.12: Planetary Gear Set [42].

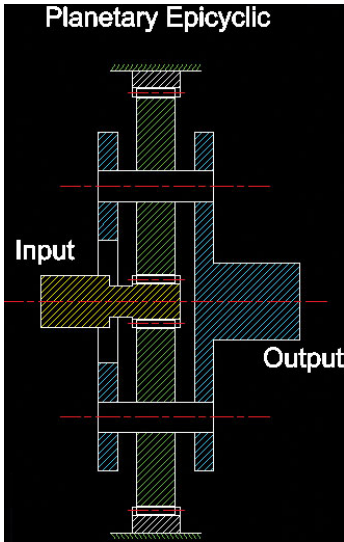


Figure 2.13: Planetary arrangement [40].

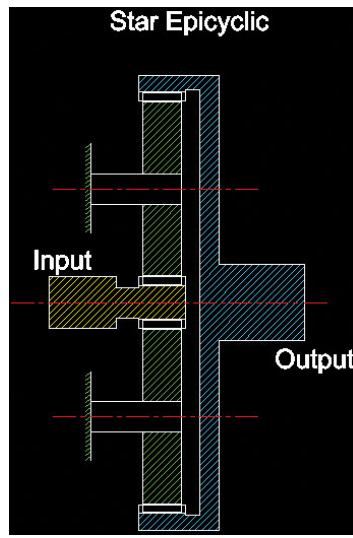


Figure 2.14: Star arrangement [40].

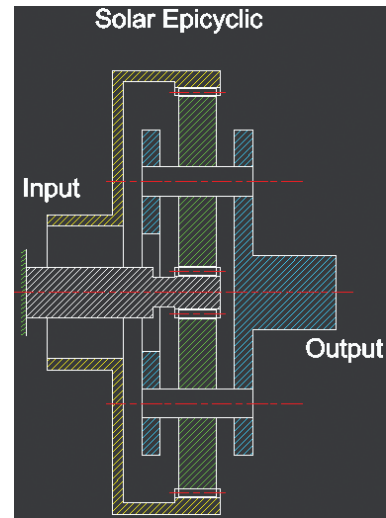


Figure 2.15: Solar arrangement [40].

5.1) Wilson Planetary Gear Set

The usage of planetary gears in vehicle transmissions begins in 1929 with Walter Gordon Wilson's "Wilson" gearbox model. Walter Gordon Wilson was born in Blackrock, Ireland, in 1874 and graduated in mechanical engineering. He began working in the British military sector in 1904. During World War I, he added planetary gears to the Mark V tank, allowing it to be driven by a single driver rather than the four required previously. Wilson refined the transmission used in the Mark V tank after the war, giving birth to the pre-selector gearbox in 1928. He formed Improved Gears Ltd with JD Siddeley to advance the Wilson Gearbox's commercial concept; the company later changed its name to Self-Changing Gears Ltd. The Wilson gearbox was then used in buses in England. However, due to its complicated construction, it fell out of favor for a time [43]. The Wilson Gear Set (see figure 2.16) consists of three planetary gear sets. The ring gear of the first gear set, the planetary carrier of the second gear set and the ring gear of the third planetary gear set are directly connected to the "Pot". The "Pot" is a cylindrical device that slides over all of the components to unitize the individual gear sets into an assembly. Out of the three sun gears, sun gear 2 and 3 are attached, commonly referred to as the "Double Sun Gear".

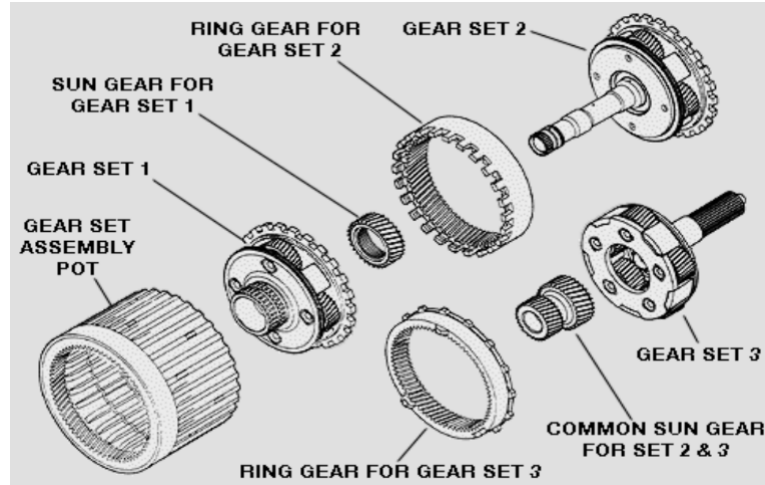


Figure 2.16: Wilson Planetary Gear Set [42].

5.2) Simpson Planetary Gear Set

Howard W. Simpson was born in Kalamazoo, Michigan, in 1892. After studying engineering at the University of Michigan, he went to work for Henry Ford's farm tractor company, the Henry Ford & Son (Fordson) Tractor Plant, after World War I. Simpson left Ford Motor Company in 1938 and the following year went to work for Detroit Harvester, where he served as chief engineer until 1943. He subsequently became a freelance engineering consultant, again working primarily in transmission and drivetrain design. In 1944, he developed and applied for a patent on a new hydraulically operated six-speed planetary transmission for tractors [44], followed in 1946 by a closely related three-speed passenger car unit [45]. Afterwards, Simpson filled a patent for an automatic planetary transmission for a self propelled vehicle in 1951 [46].

The Simpson planetary gear set (see figure 2.17) is the planetary mechanism that laid the groundwork for modern automated transmission systems. It features two ring Gears, one rear input ring and one attached to the rear planetary carrier. Two planetary carriers contain a set of three planetary pinions each. And a common sun gear meshes with both sets of planetary pinions. This simple arrangement provides three forward speeds, one reverse speed, and one neutral speed.

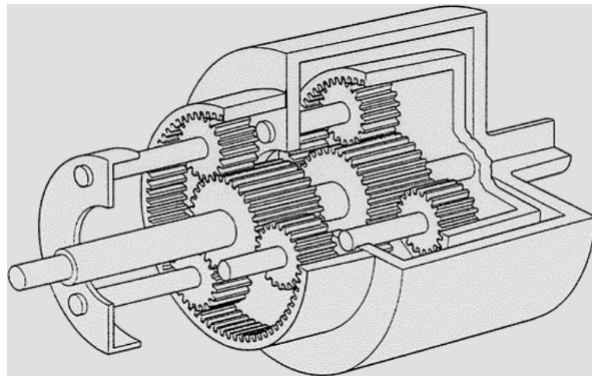


Figure 2.17: Simpson Planetary Gear Set [42].

5.3) Ravigneaux Planetary Gear Set

The Ravigneaux mechanism represents modern and compact planetary gear set design, being widely used in automatic transmissions and subject to numerous research. It arose as a result of improvement over earlier planetary automobile transmissions, such as the Simpson transmission, which despite having only being patented in 1944, had already been introduced to the automotive industry by Howard W. Simpson [43]. This design was patented by Paul Ravigneaux in 1940 in France [47].

This thesis' aim is to design a two speed transmission with a Ravigneaux type construction, inspired in the gearbox proposed in [33]. That being said, this type of gear set will be analysed in further detail than the rest.

When compared to normal planetary mechanisms, the building of a Ravigneaux planetary mechanism is more complicated. Ravigneaux planetary mechanisms come in a variety of forms, but the seven-link configuration has seen the most use in practical automatic transmission systems. The seven link configuration (see figure 2.18) is capable of four forward gears and one reverse; it includes a single carrier containing two sets of planet pinions, long pinions, and short pinions. Each pinion set includes three pinions, each pinion being mutually spaced angularly about a central axis, each short pinion meshing with a corresponding long pinion. There are two sun gears, a short sun gear meshing with the three short pinions, and a long sun gear meshing with the three long pinions. There is one ring gear, which meshes with the three long pinions. Both planet pinion sets are supported for rotation on the carrier. Each long pinion is in mesh with a short pinion, the ring gear, and the long sun gear. Each short pinion is in mesh with a long pinion and the short sun gear.

ZF HP 14 was one of the first transmission systems to use the seven-link Ravigneaux gear set (figure 2.19). It is based on a single Ravigneaux planetary gear set and a transmission that was introduced in 1984. There were four forward speeds, one backward speed, and a neutral speed available [38].

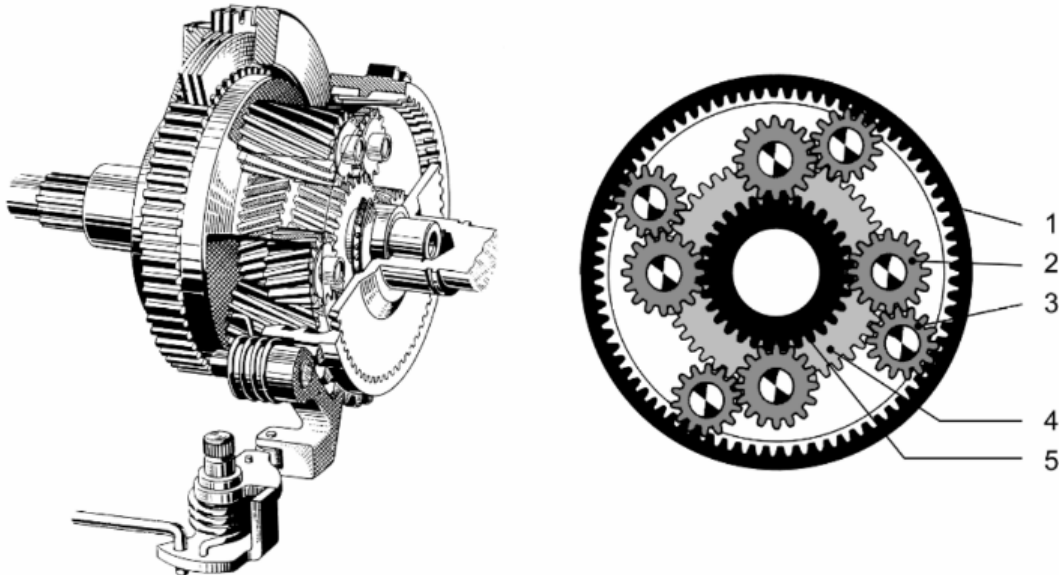


Figure 2.18: Seven-link Ravigneaux Planetary Gear Set. 1-Common ring gear; 2-narrow planetary gear; 3-broad planetary gear; 4-large sun gear; 5-small sun gear [38].

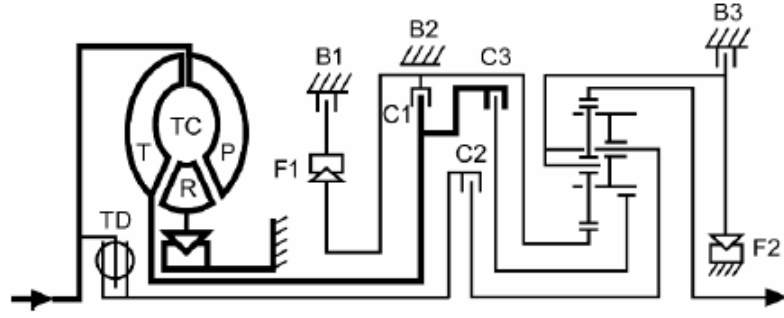


Figure 2.19: ZF HP 14 gearbox diagram [38].

An eight-link Ravigneaux planetary gear set (figure 2.20) is more complicated than the well-known seven-link arrangement. One more ring gear is the main difference between these two layouts. This arrangement was presented in one of the first patents filled by P. Ravigneaux [47].

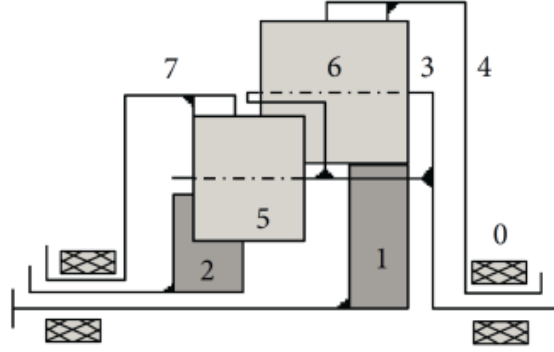


Figure 2.20: Eight-link Ravigneaux planetary mechanism [48].

When compared to the Ravigneaux gear sets previously stated, the nine-link planetary mechanism (figure 2.21) is notably different. Three sun gears (1, 2 and 8) as well as three planets (5, 6 and 6¹) and two ring gears make up this mechanism (4 and 7). This layout uses a highly complex planet carrier (3), which is comprised of three distinct planetary gears mounted on two spider shafts. This planetary gear set's use in automatic transmissions is severely limited due to its increased complexity.

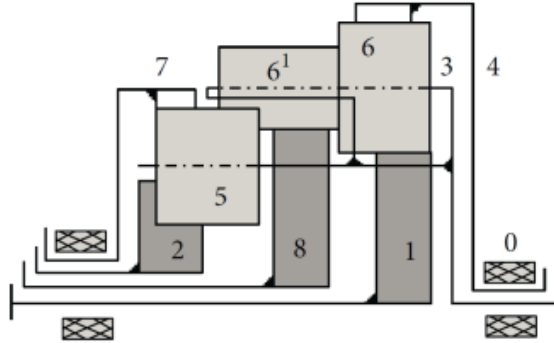


Figure 2.21: Nine-link Ravigneaux planetary mechanism [48].

The Lepelletier concept was used to create the world's first six-speed automatic gearbox system. ZF introduced this transmission (figure 2.24) to the market in 2001 under the name ZF 6 HP 26 [38]. This was also one of the first automatic gearbox systems with two overdrive speeds to be commercially available.

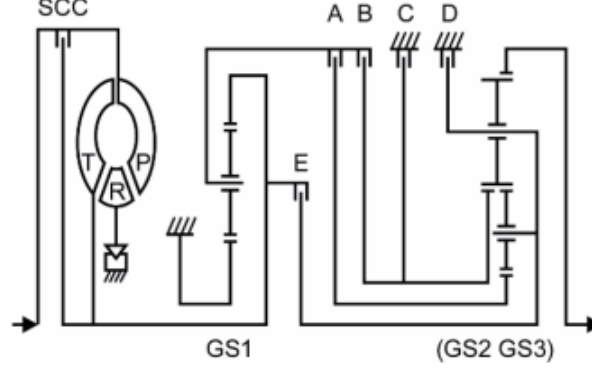


Figure 2.24: Gearbox diagram of ZF 6 HP 26 automatic transmission [38].

The development of automatic transmissions has accelerated in recent years. Recent innovations by manufacturers, were mostly inspired by Ravigneaux/Lepelletier planetary mechanisms. Cutting edge automatic transmissions indicate that simple planetary gear sets will probably rule the industry in the coming years [48]. However, although automatic transmissions are widely implemented in the automotive industry, their application in the electric vehicle segment is highly limited because of the preference for single gear ratio powertrains. For the growth of electric vehicles to continue, the optimization of power flow must be taken into consideration when designing an EV powertrain, which is why multiple speed automatic transmissions will start to be incorporated more often. The planetary architectures present themselves as a suitable choice for EV powertrains, because of their high load capacity and compact design. With this in mind, this work proposes a design for a two speed Ravigneaux transmission intended for an electric vehicle.

Performance Requirements of the Powertrain

3.1 Vehicle Specifications

The choice and study of the vehicle specifications began by firstly choosing a vehicle segment in which this type of transmission was considered interesting to be analysed. This process involved researching different electric vehicles available on the market ranging from low performance cars with urban driving profiles (highlighted in green) to high performance electric supercars (highlighted in red), along with their performance and motor characteristics [50]. The vehicles analysed are listed below in table 3.1. The goal was to select a vehicle which used a single speed transmission, along with its weight and electric motor, in order to observe the impact of implementing a two speed transmission on the vehicle's performance. After some discussion, it was decided that an appropriate and interesting choice would be the Audi Q4 40 e-tron.

Based on this vehicle's performance abilities and data on existing BEVs retrieved from [51], estimates on the goal performance requirements for this project were set.

Table 3.1: EVs' performance characteristics. Highlighted in green are urban/city EVs; in blue are medium performance EVs; and in red are high performance EVs and super cars [50].

Model	Power [kW]	Power [hp]	Peak Torque [Nm]	Top Speed [km/h]	0-100km/h [s]	Weight [kg]
Rimac Nevera	1427	1912	2360	412	1,97	2150
Porsche Taycan Turbo S Cross Turismo	560	750	1049	250	2,9	2320
Audi e-tron GT RS	475	637	830	250	3,3	2420
Tesla Model Y Performance	340	456	639	241	3,7	1950
Tesla model 3 Long Range	324	434	493	233	4,4	1920
BMW i4 eDrive 40	250	335	430	190	5,7	2125
Mercedes EQS 450+	245	328	568	210	6,2	2480
Hyundai Ioniq 5 Longe Range AWD	225	302	605	185	5,2	2100
VW ID4 Pro S AWD	220	295	460	-	5,4	2220
Ford Mustang Mach-E SR AWD	198	265	580	180	6,3	2100
Audi Q4 40 e-tron	150	201	310	160	8,5	2050
Chevrolet Bolt EV	150	201	360	145	6,9	1616
Mini Cooper SE level III	135	181	270	150	7,3	1365
Kia EV6 Light RWD SR	125	168	-	185	8	1820
Hyundai Ioniq Electric	88	118	295	165	9,9	1600

Vehicle performance can be summarized in the following single quantitative measures:

- Top speed;
- Minimum time to accelerate from 0-100 km/h;
- Gradeability.

Although the selected goal performance parameters will be explained in detail in the following sections, a brief rundown of them is done here. The top speed was set to slightly higher than the one achieved on the select car, so as to take advantage of the two speed configuration. A study on the acceleration time from 0 to 100 km/h was carried out with different transmission ratios and gear change speeds, with the goal of achieving better acceleration performance than the Audi Q4 40 e-tron. The gradeability, which describes describes the maximum grade that a vehicle climb at a certain speed while using the maximum power from the power train [1], will be set accordingly to general gradeability design goals retrieved from [51]. Based on the data consulted, and the fact that the chosen EV is considered to be on the higher end of performance when it comes to city cars, the powertrain designed in this thesis should allow for starting at a 25 % uphill grade and also for driving at a speed of 130 km/h at a grade of 6 %. The estimated performance goals that will restrict the design of the two speed transmission are summarized in table 3.2.

Table 3.2: Goal performance requirements set for the design of the transmission.

Goal performance requirements	
Top Speed	180 km/h
Acceleration 0-100 km/h	≤ 8.5 s
Gradeability at start	25 %
Gradeability at 130 km/h	6 %

3.2 Electric motor

The electric motor converts electrical energy into mechanical work and vice versa. An electric motor may supply high power and torque to a transaxle or differential for propulsion. In comparison to an internal combustion engine, an electric motor may provide instantaneous power and torque. Electric motors also have a far greater energy conversion efficiency than internal combustion engines (between 80 % and 95 %) [52].

Induction motors (IM), permanent magnet synchronous motors (PMSM), permanent magnet-brushless DC motors (PM-BLDC), and switching reluctance motors (SRM) are all employed as electric drives in electric vehicles [53]. Due to their great efficiency and power density, IM and PMSM are regarded the most desirable motors for use in EVs [54].

IM has a high efficiency, starting torque, and power, as well as a simple construction, low cost, robustness, and low maintenance. IMs can operate in any hostile environment and are unaffected by speed limitations. The IM's control system, on the other hand, is quite complicated and still has a difficulty with power density. The quantity of overall losses, which can be divided into losses in the magnetic circuit (iron losses), losses in the windings (copper losses), losses in the converter (commutation and stray losses), and mechanical losses, determines the energy efficiency of this motor [55].

PMSM has various unique capabilities, including the ability to provide consistent torque while maintaining high efficiency, high power density, and low energy consumption. Furthermore, because the PMSM rotor lacks a coil and brushes, it generates very

little heat. PM-SM is suited for EVs and HEVs since it features highly conductive materials and high permeability on the permanent magnets. On the other hand, due of the permanent magnet within, the initial cost of this motor is typically high, because PM material resources are restricted and expensive [55].

SRM is also a motor type that is preferred in electric vehicles. In comparison to the others, it has the simplest configuration. It just has a rotor (moving portion) and a stator (non-moving part), with the stator being the sole part with winding. The SRM is less expensive than PM motors since it lacks a permanent magnet. Furthermore, SRM is fault-tolerant, which means that a fault in one phase will not effect the other phases. Despite various difficulties like as acoustic noise, torque ripple, converter topology challenges, and electromagnetic interference that need to be addressed, SRM is still regarded a physically good contender for EVs and HEVs due to its sturdy construction and low cost [55].

Figure 3.1 shows the efficiency map of the electric motors detailed here, and it can be observed that each electric motor has its optimal area for both the driving cycle and the braking cycle.

As previously stated, the electric motor chosen for the powertrain here considered was the one from the selected EV (Audi Q4 40 e-tron). This vehicle features a permanent magnet synchronous motor that provides 150 kW of peak power and 90 kW of continuous power (figure 3.2). The motor data was retrieved from [56], and is detailed in table 3.3.

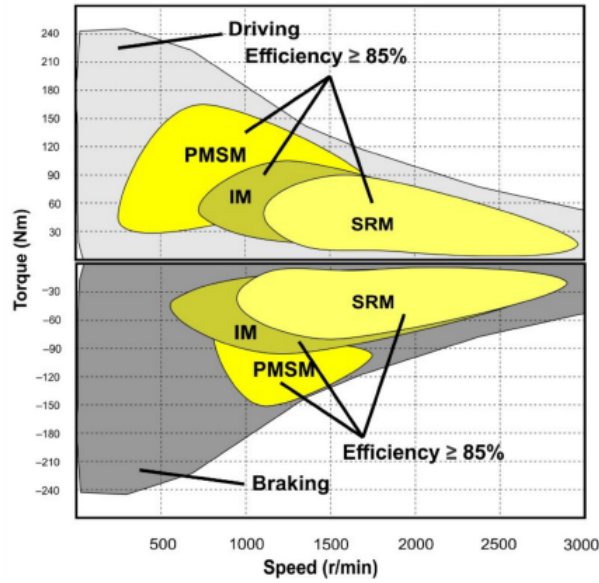


Figure 3.1: Efficiency map of electric motors [55].

Table 3.3: Technical data of the Audi Q4 40 e-tron electric motor [56].

Motor characteristics	
Type	PMSM
Power (Peak/Continuous) [kW]	150 / 90
Torque (Peak/Continuous) [Nm]	310 / 183
Base Speed [rpm]	4700
Maximum Speed [rpm]	16000

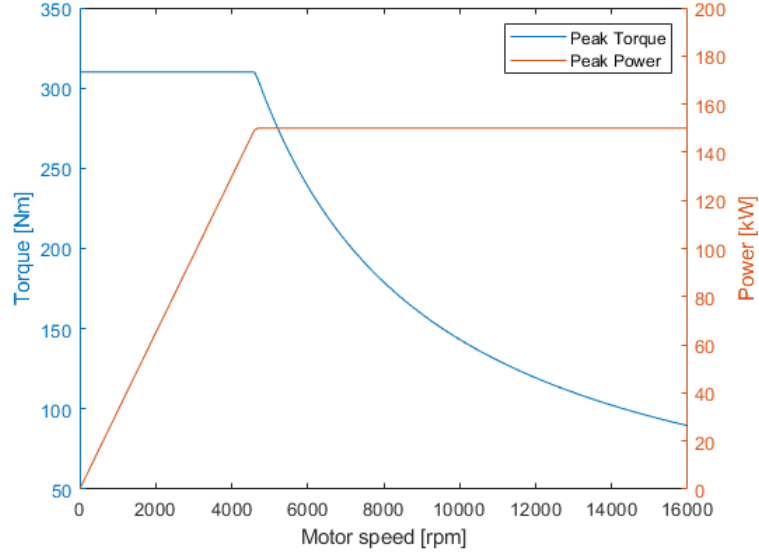


Figure 3.2: Torque-power peak curve of the Audi Q4 40 e-tron electric motor [56].

3.3 Vehicle Performance Analysis

As stated in the beginning of this chapter, a vehicle's driving performance is usually evaluated by its acceleration time, maximum speed, and gradeability. In EVs, proper motor power rating and transmission parameters are the primary considerations to meet the performance specification [1]. It should be noted that designing a power transmission structure does not follow a precise sequence of steps, since the design and selection of individual components is interdependent. Instead, it follows an iterative process in which compromises have to be done in order to achieve the goal requirements [57].

3.3.1 General Description of Vehicle Movement

In order to make an analysis on the performance capabilities of the powertrain here designed, the basic fundamentals of vehicle operation must be understood. This section will therefore focus on defining the forces acting on the vehicle and the equations that define its movement, based on the general principles of mechanics.

The forces exerted on a vehicle traveling up a gradient are presented in figure (3.3). The car is propelled forward by the tractive effort, F_t , in the contact area between the driving wheels' tires and the road surface. The motor torque generates this force, which is transmitted to the driven wheels via the transmission and final drive. Vehicle resistance opposing its movement includes rolling resistance force (F_{rr}) of the tires, appearing in figure 3.3 as rolling resistance torque T_{rf} and T_{rr} , aerodynamic drag (F_a), and gradient force (F_g). All these resistances will be discussed in detail in the following sections.

According to Newton's second law:

$$\frac{dV}{dt} = \frac{F_t - (F_{rr} + F_a + F_g)}{\delta M_v}, \quad (3.1)$$

where V is vehicle speed, F_t is the total tractive effort of the vehicle, M_v is the mass of the vehicle and δ is the mass factor, which is an effect of rotating components in the power train [1]. Equation (3.1) shows that the speed and acceleration of the vehicle depend on the tractive effort, resistance to movement and vehicle mass.

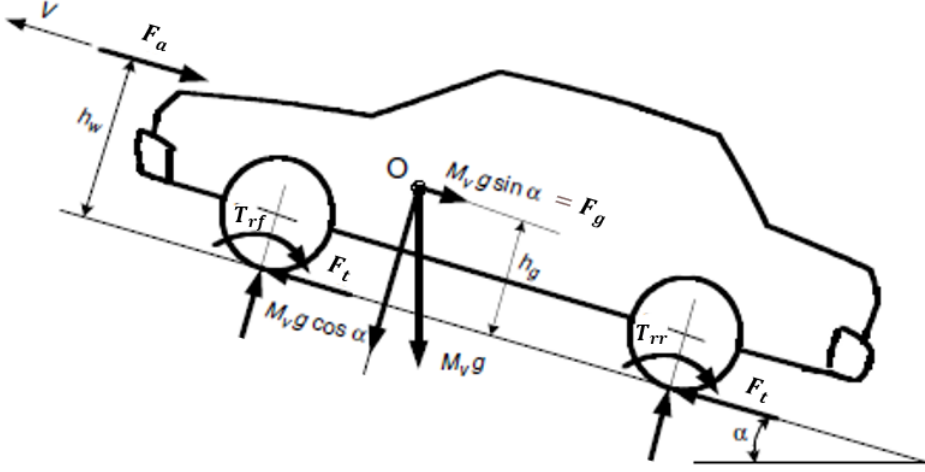


Figure 3.3: Forces acting on a vehicle [1].

Vehicle Resistance

As stated, the vehicle resistance opposing its movement includes the rolling resistance of the tires, aerodynamic drag and grading resistance.

1) *Rolling Resistance*

The rolling resistance of the tires is attributed to the hysteresis experienced in the tire materials. When the tire is rolling, it deflects, causing an asymmetric distribution of the ground reaction forces.

The pressure in the leading half of the contact area is larger than in the trailing half, resulting in a forwardly shifted ground reaction force which creates a moment (T_r) that opposes the rolling of the wheel [1]. This moment is represented in figure 3.4 and can be expressed as

$$T_r = Pa. \quad (3.2)$$

To keep the wheel rolling, a force F , acting on the center of the wheels, is required to balance this rolling resistant moment. This force is expressed as

$$F = \frac{T_r}{r_d} = \frac{Pa}{r_d} = P\mu, \quad (3.3)$$

where r_d is the tire radius, μ , which is given by a/r_d , represents the tire resistance coefficient. This coefficient is a function of the tire characteristics such as tire material, structure, temperature, inflation pressure and the road characteristics that include road roughness, material and the presence or absence of liquids on the road. The chosen value for this coefficient was $\mu = 0.009$, based on [51] and [1].

When considering the slope of the road, α , and the vertical gravitational force ($P = M_v g$) the rolling resistance force F_{rr} can be written as

$$F_{rr} = M_v g \mu \cos \alpha. \quad (3.4)$$

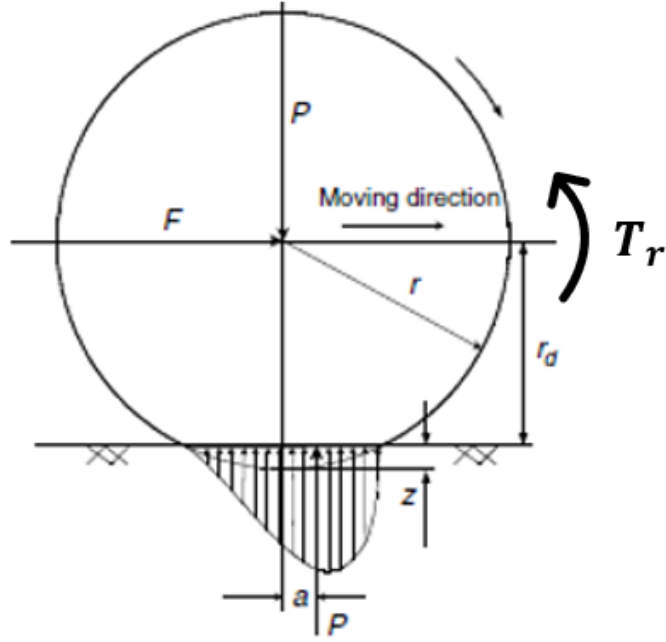


Figure 3.4: The deflection and rolling resistance on a road surface [1].

2) Aerodynamic Drag

Aerodynamic drag corresponds to the resistive force that a vehicle encounters when it travels at a certain speed through air. It mainly results from two components: shape drag and skin friction. Shape drag is the resulting force from both the high pressure area created in front of the vehicle and the low pressure area behind the vehicle that oppose its movement. Skin friction relates to the friction between the molecules of air that travel at gradually lower speeds the further they are from the vehicle skin. Aerodynamic drag F_a is a function of vehicle speed V , vehicle frontal area A , shape of the vehicle, air density ρ and it is expressed as

$$F_a = \frac{1}{2} \rho A C_d (V + V_w)^2, \quad (3.5)$$

where C_d is the aerodynamic drag coefficient that characterizes the shape of the vehicle and V_w is the component of wind speed. According to [51], $C_d = 0.28$ is an appropriate value for battery electric vehicles. The air density will be considered as $\rho = 1.225 \text{ kg/m}^3$ as also suggested by [51].

3) Grading Resistance

The grading resistance F_g is the vehicle's weight component in the direction of motion, when it goes up or down a slope. It can be expressed as

$$F_g = M_v g \sin \alpha. \quad (3.6)$$

The sum of all resistive components is also referred to as Road Load (F_{RL}):

$$F_{RL} = F_{rr} + F_a + F_g. \quad (3.7)$$

Therefore, equation (3.1) can also be written as

$$\delta M_v \frac{dV}{dt} = F_t - F_{RL}. \quad (3.8)$$

Tractive Effort

The power plant transmits torque to the driven wheels, and the torque felt on the driven wheels can be expressed as

$$T_w = i_{fd} i \eta_d T_p, \quad (3.9)$$

where i_{fd} is the gear ratio of the final drive, i is the gear ratio of the transmission, T_p is the torque output from the power plant and η_d is the driveline efficiency, which for simplification purposes will be assumed to be 1. The tractive effort of the driven wheels can now be expressed as:

$$F_t = \frac{T_w}{r_d} = \frac{T_p i_{fd} i}{r_d}. \quad (3.10)$$

For simplification purposes, the calculations were done considering a global transmission ratio (i_G):

$$i_G = i_{fd} i. \quad (3.11)$$

The tractive effort can therefore be rewritten as

$$F_t = \frac{T_w}{r_d} = \frac{T_p i_G}{r_d}. \quad (3.12)$$

Having defined the forces acting on the vehicle during its operation, the needed gear ratios for the two speed transmission could now be analysed based on the chosen traction motor and the goal performance characteristics. The parameters chosen for the calculations performed next are summarized in table 3.4. Both the mass of the vehicle and its tire radius are the same as the chosen EV [56] (Audi Q4 40 e-tron).

Table 3.4: Considered parameters for the calculations done in the dynamic performance analysis, based on [51, 1, 56].

Parameter	Designation	Unit	Value
Mass	M_v	kg	2050
Area	A	m ²	2.56
Tire radius	r_d	m	0.3688
Gravitational acceleration	g	m/s ²	9.81
Air density	ρ	kg/m ³	1.225
Aerodynamic drag coefficient	C_d	-	0.29
Tire resistance coefficient	μ	-	0.009

3.3.2 Maximum Speed

The maximum speed of a vehicle is found when the tractive effort (F_t) curve intersects the road load curve (F_{RL}), in the tractive effort vs. vehicle speed diagram [1]. In this diagram, shown in figure 3.5, this point of intersection is point f.

However, this intersection point does not exist in most vehicle powertrain designs, which use large traction motors or large gear ratios. In these cases, the speed of the vehicle is limited by the maximum speed of the electric motor [1]. Equation (3.13) will determine the maximum speed (V_{max}) of the vehicle, where N_m is the maximum speed of the motor ($N_m=16\,000$ rpm).

$$V_{max} = \frac{\pi N_m r_d}{30 i_{Gmin}}. \quad (3.13)$$

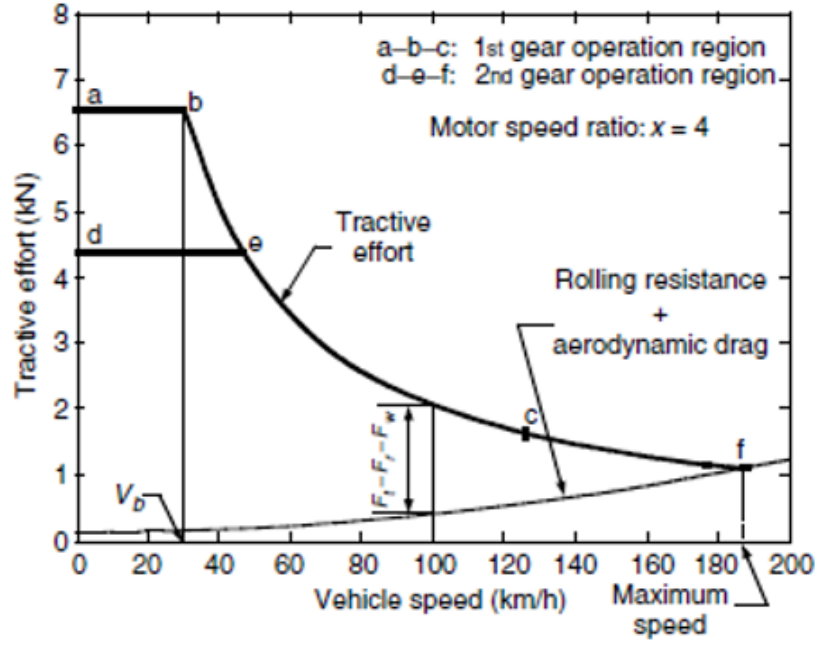


Figure 3.5: Tractive effort vs. vehicle speed of a general two speed transmission [1].

With a defined value for the target top speed, equation (3.13) will give the minimum required global gear ratio (i_{Gmin}), which in a two speed transmission corresponds to the second global gear ratio ($i_{Gmin} = i_{G2}$). Thus, rearranging,

$$i_{G2} = \frac{\pi N_m r_d}{30 V_{max}}. \quad (3.14)$$

The previously set goal for the top speed was to be higher than the selected motor's vehicle, which is 160 km/h. However, like in most modern cars, this value is electronically limited so as to protect key components of the vehicle and follow legal regulations. Although no information on the actual maximum speed of the Audi Q4 40 e-tron was found, it was estimated to be around 170 km/h, based on information found on its transmission and final drive gear ratios [56]. Having initially set the goal of achieving a higher maximum speed than the Audi, a top speed target of 180 km/h was locked for this design. The global transmission ratio needed for the vehicle to achieve this speed, is at maximum - $i_{G2} \approx 12.5$. Table 3.5 summarizes the conclusions drawn from analysing the design goal top speed.

Table 3.5: Top speed requirement results.

Requirement	Transmission design goal
Top Speed - 180 km/h	$i_{G2} \leq 12.5$

3.3.3 Gradeability

A vehicle's gradeability is defined as the maximum grade that a vehicle can climb at a certain speed while using the powertrain's maximum power [51]. With α being the road angle, gradeability can be expressed as

$$Grade [\%] = \tan \alpha \times 100 \quad (3.15)$$

Gradeability requirements for general vehicles are based on two situations: the ability to start in an uphill of a certain grade, and the ability to travel at a certain speed at a specific grade. According to [51], general gradeability design goals are that cars should be able to start in a 25 % uphill grade, drive at a speed of 88 km/h in a 3 % grade, and at 72 km/h in a 6 % grade. Considering that typical grade levels on highways rarely surpass grades of 6 % [51], and that the performance levels of the powertrain here considered is higher than most cars intended for city driving, the following gradeability requirements were set as design goals:

- a) The vehicle must be able to start in a 25 % uphill grade;
- b) The vehicle must be able to travel at a speed of 130km/h at a grade of 6%.

In order to evaluate these requirements and design the gear ratios according to them, the equilibrium equation must be considered, where the tractive force must be enough to overcome the road load:

$$F_t = F_{RL}. \quad (3.16)$$

a) The vehicle must be able to start in a 25 % uphill grade

This requirement corresponds to a situation where the vehicle is initially stopped, which means the electric motor is in the constant torque phase (peak torque phase). There is no aerodynamic drag ($F_a=0$) since the velocity of the vehicle is 0. Based on this, equation (3.16) can be written as

$$\frac{T_P i_G}{r_d} = \mu M_v g \cos \alpha + M_v g \sin \alpha. \quad (3.17)$$

Since the motor is in its constant torque phase, it outputs its peak torque ($T_p=T_{peak}$). And the global gear ratio here calculated corresponds to the minimum global first gear ratio (i_{G1}) needed to overcome the road load. Thus,

$$i_{G1} = \frac{r_d}{T_{peak}} M_v g (\mu \cos \alpha + \sin \alpha). \quad (3.18)$$

With a peak torque of 310 Nm and using the parameters in table 3.4, $i_{G1} \approx 6$. This value is significantly inferior to the minimum second gear ratio needed (calculated in 3.3.2), $i_{G2} \approx 12.5$, which leads to conclude that this requirement is easily assured with the traction motor selected, and therefore would not restrict the gear specification design process.

b) The vehicle must be able to travel at a speed of 130km/h at a grade of 6 %

This requirement involves analysing a high speed situation, in which the electric motor is its constant power phase. In this case, the tractive force is expressed as

$$F_t = \frac{P_{peak}}{V}, \quad (3.19)$$

where P_{peak} is the motor's peak power, and V is the vehicle's speed. Substituting F_t in the equilibrium equation (3.16):

$$\frac{P_{peak}}{V} = \mu M_v g \cos \alpha + M_v g \sin \alpha + \frac{1}{2} \rho A C_d V^2 \quad (3.20)$$

rearranging,

$$P_{peak} = (\mu M_v g \cos \alpha + M_v g \sin \alpha + \frac{1}{2}\rho AC_d V^2)V. \quad (3.21)$$

From equation (3.19), we see that when the motor is in its constant power phase, the tractive force is independent of the gear ratio. Therefore, the ability to drive at a certain speed (higher than the motor's base speed) and grade, is solely dependent on the motor's peak power output (equation (3.21)). For the vehicle to be able to drive at a speed of 130 km/h at a grade of 6 %, a motor capable of outputting a minimum of 68 kW is needed. Since the electric motor here used has a peak output power of 150 kW, this requirement is safely assured. The conclusions drawn from analysing the gradeability requirements are summarized in table 3.6.

Table 3.6: Gradeability requirements results.

Requirement	Transmission design goal
Uphill start at 25 % grade	$i_{G1} \geq 6$
Travel at a 130 km/h at 6 % grade	$P_{peak} \geq 68 \text{ kW}$

3.3.4 Acceleration Performance

Acceleration performance of a vehicle is evaluated by the time used to accelerate the vehicle from a low-speed V_1 (usually zero) to a higher speed V_2 [1]. For passenger cars, acceleration performance is more important than maximum cruising speed and gradeability, since it is the acceleration requirement that dictates the power rating of the electric motor.

In this work, the power rating has been set from the beginning and therefore, an analysis will be done on the acceleration performance that can be extracted from the motor considering the two speed transmission design. The second gear ratio will be set according to the requirement of top speed set analysed in 3.3.2 and the first gear ratio will be chosen based on the maximum decrease in acceleration time achieved, since the gradeability requirements did not restrict the first gear ratio in this project.

In order to evaluate the acceleration performance, Newton's second law must be again considered:

$$\delta M_v \frac{dV}{dt} = F_t - F_{RL}. \quad (3.22)$$

Substituting the road load (F_{RL}) for its components and rearranging,

$$F_t = F_{rr} + F_g + F_a + \delta M_v \frac{dV}{dt} \quad (3.23)$$

and considering a flat road ($\alpha=0$), F_g will be equal to 0, thus,

$$F_t = \mu M_v g + \frac{1}{2}\rho AC_d V^2 + \delta M_v \frac{dV}{dt}. \quad (3.24)$$

The velocity can then be written as,

$$V = \int_{V_1}^{V_2} \frac{F_t - \mu M_v g - \frac{1}{2}\rho AC_d V^2}{\delta M_v} dt. \quad (3.25)$$

As was shown in 3.3.3, the tractive force is only dependent on the gear ratio when the motor is in its constant torque phase. When it reaches the base speed (V_{base}), it enters

its constant power phase and the tractive force becomes dependent solely on the vehicle's speed and the electric motor's power output (P). Therefore, and considering that the intended acceleration performance analysis was from 0 to 100 km/h ($V_1 = 0$ and $V_2 = 100$), the integral can be split into two:

$$V = \int_0^{V_{base}} \frac{\frac{T_p i_G}{r_d} - \mu M_v g - \frac{1}{2} \rho A C_d V^2}{\delta M_v} dt + \int_{V_{base}}^{100} \frac{\frac{P}{V} - \mu M_v g - \frac{1}{2} \rho A C_d V^2}{\delta M_v} dt. \quad (3.26)$$

This equation is difficult to solve analytically, since the torque of the powerplant is a function of speed, which in turn is a function of vehicle speed. Therefore, it was solved numerically through MATLAB. Plots were drawn to show the evolution of vehicle speed with time.

In order to evaluate the increase in acceleration performance when adopting a two speed transmission, an initial analysis was made on the acceleration achieved using a single speed transmission with a global gear ratio of $i_G=12.5$, which is the gear ratio needed for the vehicle to achieve the desired top speed. The vehicle's velocity evolution with time is shown in 3.6. The MATLAB script employed, estimated an acceleration time of 9.8 seconds.

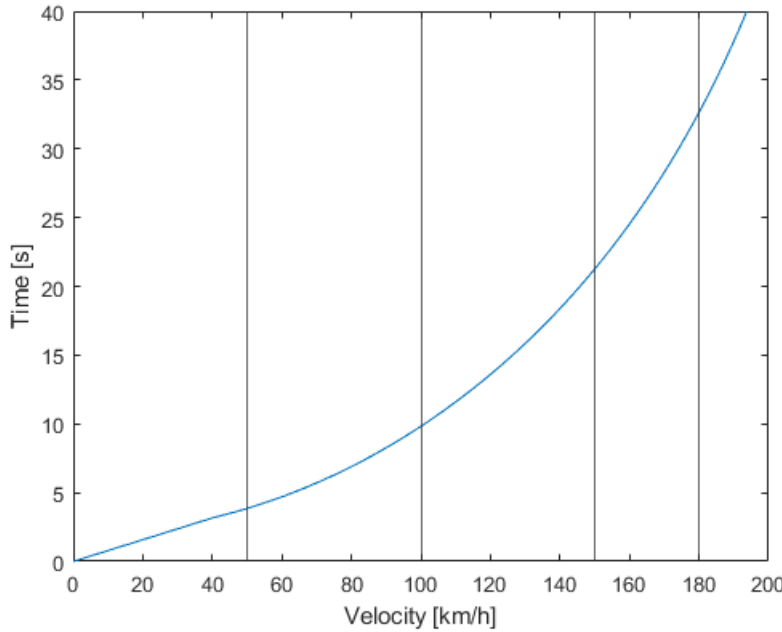


Figure 3.6: Acceleration performance considering a single speed transmission with a global gear ratio of $i_G=12.5$.

A study was then performed on the acceleration performance of the two speed transmission, in order to find the optimal first gear ratio when accelerating the vehicle from 0 to 100 km/h, so as to compare with the performance achieved with a single speed configuration. The second gear ratio was kept at $i_{G2}=12.5$. The analysis also took into account different speeds at which the gear shift occurred, and is illustrated in figure 3.7.

From this analysis, it was concluded that first gear ratios ranging from $i_{G1}=16$ to $i_{G1}=19$ resulted in the largest decreases (approximately 3 %) in acceleration, when compared to the single speed gearbox.

Based on the results, the lowest global first gear ratio capable of providing the maximum increase in acceleration performance was chosen - $i_{G1}=16$. It should be noted that

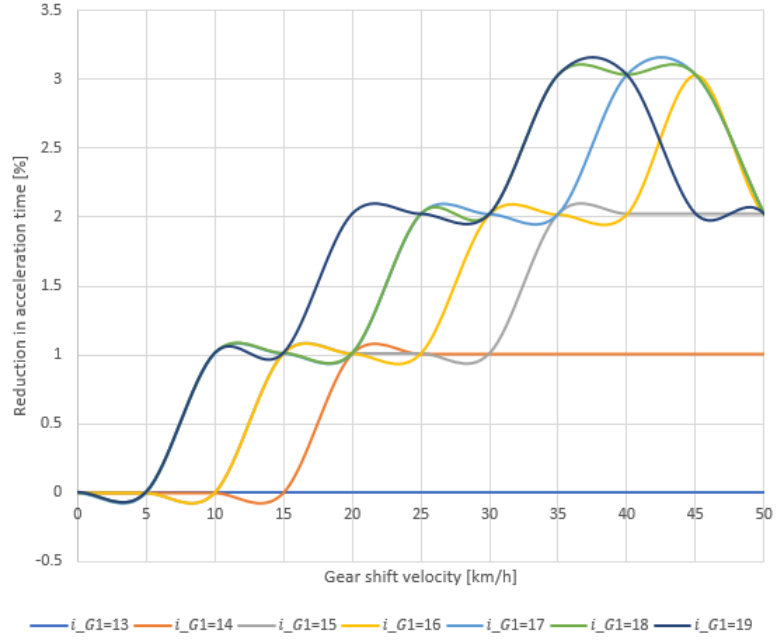


Figure 3.7: Reduction in acceleration time of an EV using a two speed transmission, compared to a single speed one. The gear ratio of the single speed is equal to the second gear ratio of the two speed speed configuration. Results are shown for different first gear ratios, and for different speeds at which the vehicle shifts gears.

for $i_{G1}=16$, the maximum acceleration performance is achieved considering that the gear shift occurs when the vehicle reaches a speed of 45 km/h.

Figure 3.8 shows the evolution of vehicle speed with time, under maximum motor output conditions, considering 3 different transmissions: a single speed transmission with a gear ratio equal to the first gear ratio of the two speed transmission designed here - $i_G=16$; another single speed with a global gear ratio now equal to the second gear ratio of the two-speed transmission - $i_G=12.5$; and finally, the two speed with $i_{G1}=16$ and $i_{G2}=12.5$. Although the acceleration time from 0 to 50 km/h is similar, it is clear the the two speed transmission provides superior acceleration performance from 50 to 100 km/h. Table 3.7 shows the difference in acceleration times for the three transmissions considered.

Table 3.7: Acceleration times from 0 to 50 km/h, and 0 to 100 km/h for the three transmission configurations considered.

	0-50 km/h	0-100 km/h
$i_G=16$	3.7 s	10.7 s
$i_G=12.5$	3.9 s	9.8 s
$i_{G1}=16$ & $i_{G2}=12.5$	3.7 s	9.6 s

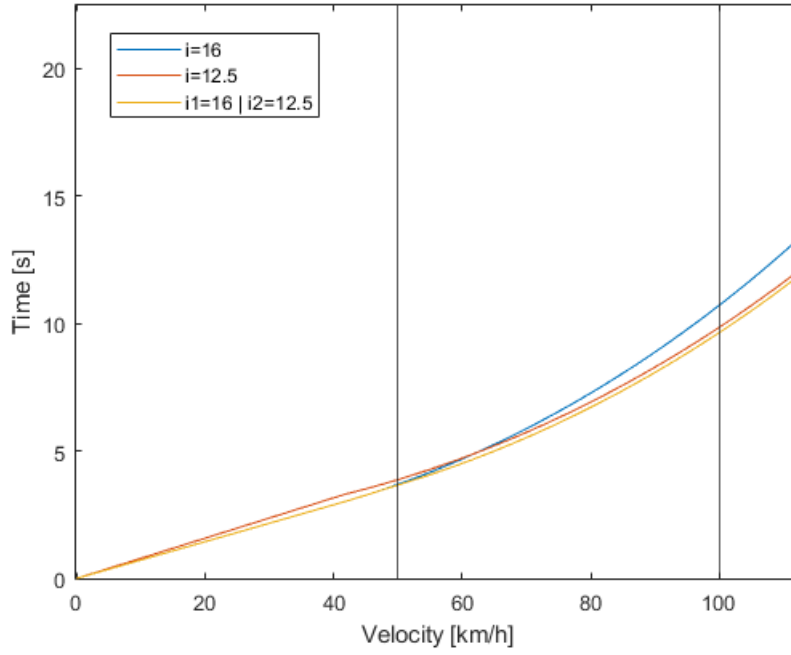


Figure 3.8: Vehicle speed with time, for the three different transmission configurations considered.

It can be seen from these results that the optimal gear ratios obtained allow for an acceleration time of 9.6 seconds, which does not correspond to the initial goal of achieving a lower acceleration time than the selected EV's (8.5 seconds). This can be attributed to the fact that all calculations were made based on estimated parameters and not real life conditions. Nevertheless, the design process proceeded with these gear ratios in mind. The conclusions drawn from this acceleration performance analysis are summarized in table 3.8.

Table 3.8: Acceleration performance requirement results.

Requirement	Transmission design goal
Fastest acceleration from 0-100 km/h	$i_{G1}=16$

3.4 Results from the Performance Requirements Analysis

The transmission design conditions that were imposed by the initially set performance requirements are summarized below in table 3.9.

Table 3.9: Transmission design goals based on all of the initially set performance requirements.

Requirement	Transmission design goal
Top Speed - 180 km/h	$i_{G2} \leq 12.5$
Start at uphill grade of 25 %	$i_{G1} \geq 6$
Travel at 130 km/h at 6 % grade	$P \geq 68$ kW
Fastest acceleration from 0-100 km/h	$i_{G1}=16$

3. Performance Requirements of the Powertrain

Since the gradeability requirements were easily assured, they were not restrictive on the gear ratio specification process. Therefore, only the top speed imposed on the project and the acceleration performance, dictated the gear ratios for the two speed transmission. Based on this, the gear ratios chosen at this phase of the design process were the following:

Table 3.10: Chosen gear ratios based on the dynamic performance requirements.

First global gear ratio	$i_{G1}=16$
Second gear global ratio	$i_{G2}=12.5$

As stated in 3.13, it is possible to estimate the potential top speed of the vehicle (if not restricted by the maximum speed of the electric motor) by finding the maximum road load that the vehicle is able to overcome with the provided powertrain. With the first and second global gear ratios of the transmission now defined, it was possible to make this analysis. A plot (figure 3.9) was drawn to show the evolution of both the vehicle's tractive effort and the road load, in relation to the vehicle's speed. The road load was calculated for all the levels of grade studied here.

Several conclusions can be drawn from the plotted data. The theoretical top speed of the vehicle corresponds to the point at which the vehicle's tractive effort is equal to resistive forces on a flat road (0 % grade), which in this case would be close to 250 km/h. It is also possible to observe that the vehicle is able to achieve considerably high speeds (bellow the 180 km/h limited by the motor's rotation), at significant levels of grade (6 % and 25 %). Finally, it can be noted that for low speeds, the first gear allows the vehicle to overcome greater levels of road load.

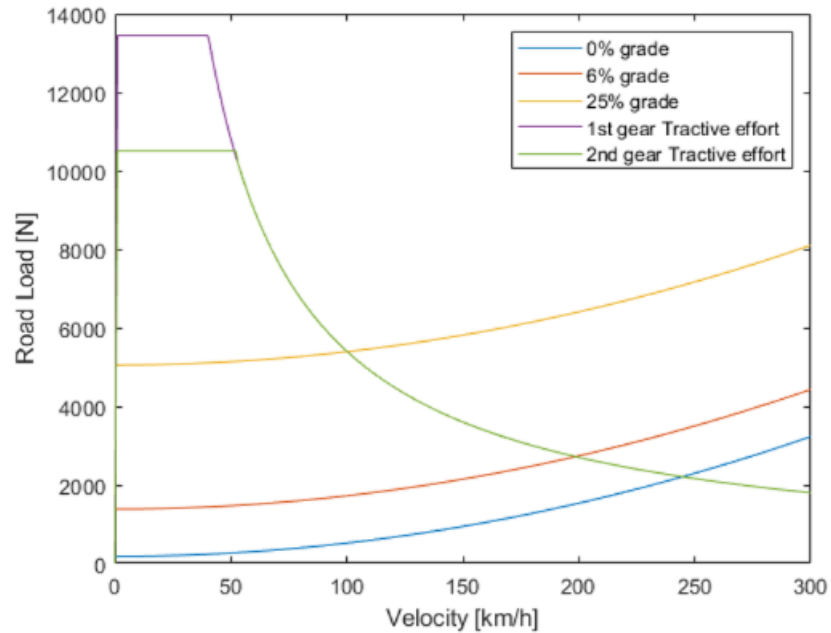


Figure 3.9: Tractive effort vs. vehicle speed of the proposed two speed transmission.

3.5 Expected lifetime and Operating Regime

The major purpose, as previously stated, is to design a two speed transmission for a city car such as the Audi Q4 e-tron. The car is expected to be driven in a city context for the most of its life, but it should also be able to drive on highways, given its relatively high performance characteristics. A value between 300 000 and 400 000 km is an adequate assumption for the vehicle's driving range during the transmission's expected lifetime [60]. According to data retrieved on the estimated life spent by cars at various speeds [58, 59, 60], it was possible to estimate the proposed transmission's expected mileage. This data is shown in table 3.11.

Table 3.11: Considered operating regime and expected lifetime for the proposed two speed transmission.

Vehicle speed [km/h]	Usage factor [%]	Distance [km]	Time spent [h]	Gear
30	15	52500	1750	1 st
50	45	157500	3150	2 nd
90	20	70000	778	2 nd
120	20	70000	583	2 nd
Total	100	350000	6261	

The proposed two speed transmission is expected to last 6261 hours, while being subject to its expected normal driving conditions. Seen as the gear shift speed was set to 45 km/h, it can be seen that the transmission will be in second gear for most of the time. Therefore, careful consideration was given to the gears that transmit power in second gear, during the gear design analysis that will be detailed further ahead. Table 3.12 summarizes the expected lifetime of the two speed transmission, for each gear. Reverse gear was not considered, since its usage is very limited.

Table 3.12: Expected lifetime for each gear of the proposed two speed transmission.

Gear ratio	1 st	2 nd
Expected lifetime [h]	1750	4511

Transmission layout and Kinematic Analysis

It is necessary to determine the input and output speeds in relation to the gears' teeth numbers in order to design the transmission gears in accordance with the previously established gear ratios. As a result, the kinematics of the transmission will be examined in this chapter, along with a mathematical definition of the angular speeds of all rotating elements. There are two sections in this chapter. The layout of the proposed two-speed Ravigneaux transmission is established in the first section, and the kinematic analysis of the transmission is covered in the second.

4.1 Layout and Working Principles

The novel two speed transmission designed in this thesis is based on the model proposed in [33], and its layout is shown in figure 4.1. It is comprised of a compound planetary gear system, two brakes and a one-way clutch (OWC). The compound planetary gear system is a variation of the 7-link Ravigneaux gear set (explained in detail in 2.2.3), since the input power is only being given to one of the sun gears. It consists of a ring gear (R), a planet carrier (C), a small sun gear (SS), a large sun gear (LS), three inner planet pinions (IP), and three outer planet pinions (OP). As shown in this figure, the input of the transmission is the small sun gear, which is connected to the electric motor through the input shaft. The output of the transmission is the ring gear, which is attached to final drive.

The author of this design idea names this transmission as a Uninterrupted Planetary Automatic Transmission (UPAT) [33]. Its ability to provide uninterrupted torque is ensured by the OWC and two band brakes adopted in the gearshift system. The band brakes are electrically actuated (using a screw-nut pair), instead of the hydraulically actuated brakes adopted in traditional ATs.

A traditional 7-link Ravigneaux gear set, in which the input power can be provided to both sun gears and the carrier, is capable of providing four different gear ratios and one reverse. However, as stated above, this design features a single input shaft connected to the small sun gear in order to ensure only two possible gear ratios.

The working principles of the gearshift system are presented in table 4.1, where '0' indicates the element does not work, and '1' means the element is operating. When the OWC is working, it means that it is preventing the carrier from rotating, where as when it is not working, it allows its free rotation.

- Parking can be achieved by engaging both the carrier brake (C brake) and the large sun brake (LS brake);
- First gear is achieved by engaging C brake, and allowing the free rotation of the LS;

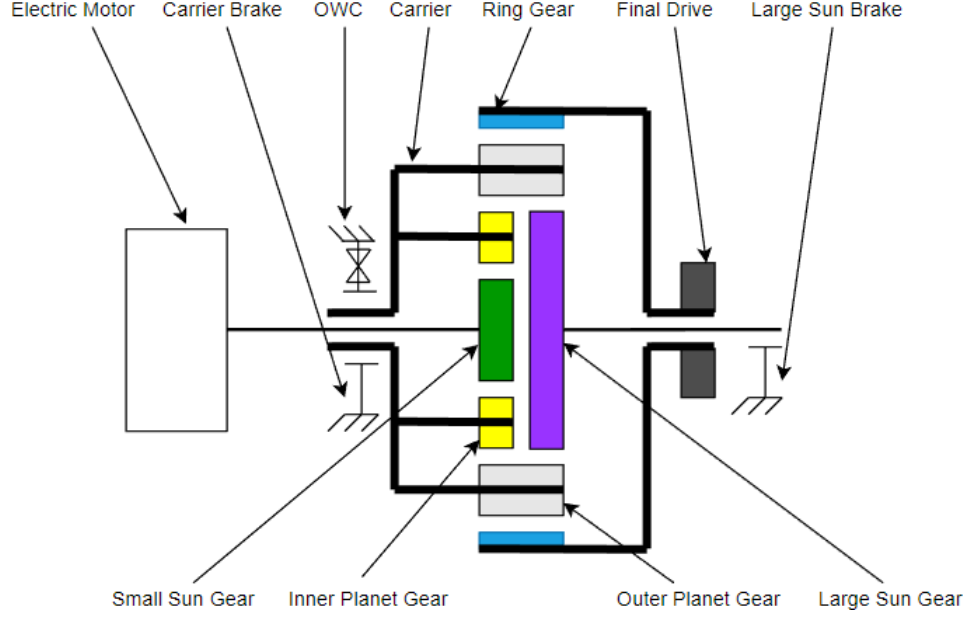


Figure 4.1: Layout of the proposed two speed transmission.

- Second gear is achieved when the LS is stopped by the large sun brake (LS brake), and the carrier is allowed to rotate freely;
- Reverse gear is achieved by grounding the carrier with the OWC.

When the upshift from first to second gear happens, C brake must be fully disengaged quickly, and then LS brake is controlled to engage gradually until LS speed reaches zero, which means the upshift process is finished. When shifting from second gear back to first gear, the LS brake is released gradually until the torque applied on itself is zero, while simultaneously C brake should be controlled to engage quickly. In the condition of reverse gear driving, C is grounded by the OWC, since the counter-clockwise rotation of the traction motor corresponds to the direction of rotation to which the OWC locks the carrier to the housing.

The final drive gear was intended to be mounted to the ring gear case in order to transmit the output power. The development of the final drive system was not one of the main goals of this work, and therefore, did not receive careful consideration. Nevertheless, its design is summarized in the following chapter.

Table 4.1: Gear shifting principles of the proposed two-speed transmission.

Gear	OWC	C Brake	LS Brake
Parking	0	1	1
1st	0	1	0
2nd	0	0	1
Reverse	1	0	0

4.2 Kinematic Analysis of the Transmission

Figure 4.2 shows the configuration of the proposed planetary gear set, along with the reference points in which the kinematic analysis will be based on. Table 4.2 describes each rotational member that will be present in this same analysis. In this section, the kinematic equations of this transmission are defined through the analysis of linear speed relations on contact points and through geometric relations intrinsic to this configuration.

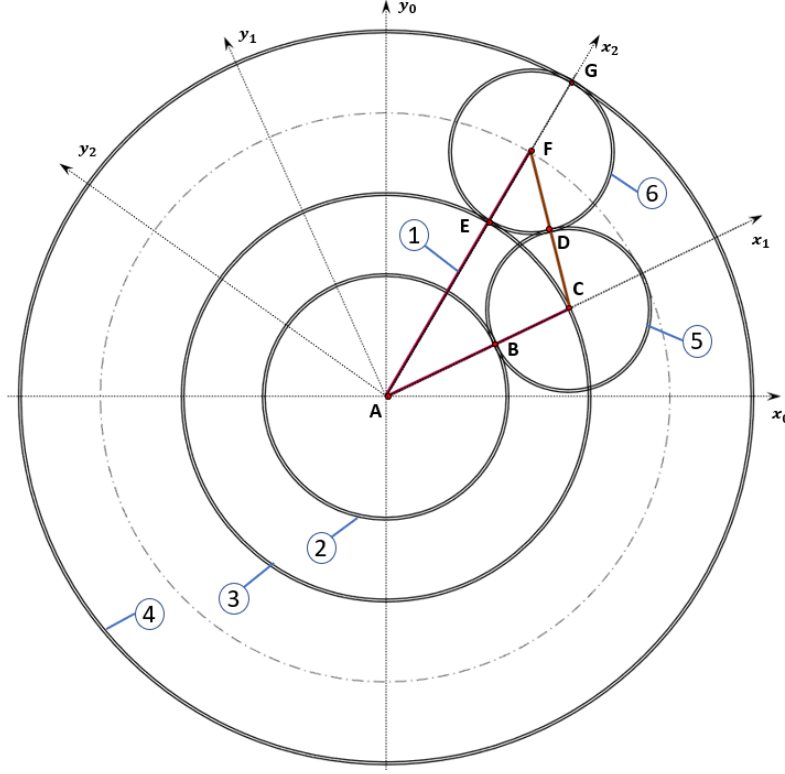


Figure 4.2: 2D spatial configuration of the planetary gear set.

Table 4.2: Description of the bodies comprising the compound planetary gear set.

Body	Carrier C 1	Small Sun SS 2	Large Sun LS 3	Ring R 4	Inner Planet IP 5	Outer Planet OP 6
Reference point	A	A	A	A	C	F
Radius [m]	$R_C \mid R_F$	R_2	R_3	R_4	R_5	R_6
Angular speed [rad/s]	$\vec{\omega}_{10}$	$\vec{\omega}_{20}$	$\vec{\omega}_{30}$	$\vec{\omega}_{40}$	$\vec{\omega}_{50}$	$\vec{\omega}_{60}$
Linear speed [m/s]	$\vec{v}_{10}^A = 0$	$\vec{v}_{20}^A = 0$	$\vec{v}_{30}^A = 0$	$\vec{v}_{40}^A = 0$	$\vec{v}_{50}^C = \vec{v}_{10}^C$	$\vec{v}_{60}^F = \vec{v}_{10}^F$

Some geometric restrictions could be immediately set:

$$R_C = R_2 + R_5 \quad (4.1)$$

$$R_F = R_3 + R_6 \quad (4.2)$$

$$R_4 = R_3 + 2R_6 \quad (4.3)$$

$$\overline{CF} = R_P = R_5 + R_6 \quad (4.4)$$

Assuming the rolling contact of gears at points B, E, G and D, is done without sliding, and considering the reference axis, S_1 and S_2 , the following kinematic restrictions could be set:

$$\vec{v}_{20|S_1}^B = \vec{v}_{50|S_1}^B \quad (4.5)$$

$$\vec{v}_{30|S_2}^E = \vec{v}_{60|S_2}^E \quad (4.6)$$

$$\vec{v}_{40|S_2}^G = \vec{v}_{60|S_2}^G \quad (4.7)$$

$$\vec{v}_{50|S_1}^D = \vec{v}_{60|S_1}^D \quad (4.8)$$

In this kinematic analysis, Mozzi's first equation will be used, which can be expressed as [61]

$$\vec{v}_P = \vec{v}_{O_1} + \vec{\omega} \times \overrightarrow{O_1P} \quad (4.9)$$

This equation indicates that the velocity of a generic point P (shown in figure 4.3) of a moving body, can be obtained by the sum of two vectors:

1. a constant vector $\vec{v}_{O_1}$, which is the point of an arbitrary point O_1 taken as a pole. This vector, in a given instant, is the same for all the points belonging to a body and can be considered as the translatory component;
2. the velocity $\vec{\omega} \times \overrightarrow{O_1P}$, that the point P would have if it was rotating at a constant angular velocity $\vec{\omega}$, around O_1 .

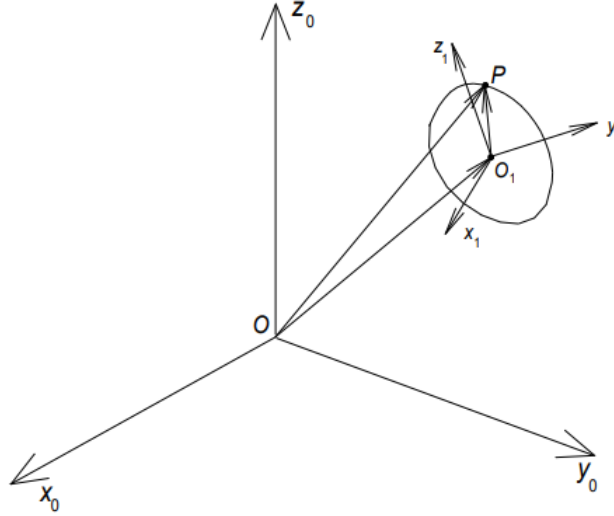


Figure 4.3: Body in movement relatively to S_0 [61].

Assuming the rolling contact with no sliding on point B, and the reference axis S_1 (A, x_1, y_1, z_1), from equation (4.5):

$$\vec{v}_{20|S_1}^B = \vec{v}_{50|S_1}^B. \quad (4.10)$$

Applying Mozzi's first equation between points A and B, and knowing that point A is fixed ($\vec{v}_{20|S_1}^A = 0$), $\vec{v}_{20|S_1}^B$ can be defined as

$$\vec{v}_{20|S_1}^B = \vec{v}_{20|S_1}^A + \vec{\omega}_{20|S_1} \times \overrightarrow{AB}_{|S_1} = \begin{Bmatrix} 0 \\ 0 \\ \omega_{20} \end{Bmatrix} \times \begin{Bmatrix} R_2 \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ R_2 \omega_{20} \\ 0 \end{Bmatrix}. \quad (4.11)$$

Mozzi's first equation is used again, and $\vec{v}_{50}^B$ can be expressed as

$$\vec{v}_{50|S1}^B = \vec{v}_{50|S1}^C + \vec{\omega}_{50|S1} \times \vec{CB}_{|S1}. \quad (4.12)$$

The inner planet (5) and the carrier (1) both share point C, thus,

$$\vec{v}_{50|S1}^C = \vec{v}_{10|S1}^C \quad (4.13)$$

and applying Mozzi's first equation again, $\vec{v}_{10|S1}^C$ can be expressed as

$$\vec{v}_{10|S1}^C = \vec{v}_{10|S1}^A + \vec{\omega}_{10|S1} \times \vec{AC}_{|S1} = \begin{Bmatrix} 0 \\ 0 \\ \omega_{10} \end{Bmatrix} \times \begin{Bmatrix} R_C \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ R_C \omega_{10} \\ 0 \end{Bmatrix}. \quad (4.14)$$

Replacing (4.14) in (4.12),

$$\vec{v}_{50|S1}^B = \begin{Bmatrix} 0 \\ R_C \omega_{10} \\ 0 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ \omega_{50} \end{Bmatrix} \times \begin{Bmatrix} -R_5 \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ R_C \omega_{10} - R_5 \omega_{50} \\ 0 \end{Bmatrix}. \quad (4.15)$$

Finally, equation (4.5) can be expressed as

$$\begin{aligned} \vec{v}_{20|S1}^B &= \vec{v}_{50|S1}^B \\ \Leftrightarrow R_2 \omega_{20} &= R_C \omega_{10} - R_5 \omega_{50} \\ \Leftrightarrow R_2 \omega_{20} &= (R_2 + R_5) \omega_{10} - R_5 \omega_{50}. \end{aligned} \quad (4.16)$$

Following a similar procedure to the one above, considering rolling without sliding on point E and the reference axis S_2 (A, x_2 , y_2 , z_2), from equation 4.6:

$$\vec{v}_{30|S2}^E = \vec{v}_{60|S2}^E. \quad (4.17)$$

Where $\vec{v}_{30|S2}^E$ can be calculated through Mozzi's first equation,

$$\vec{v}_{30|S2}^E = \vec{v}_{30|S2}^A + \vec{\omega}_{30|S2} \times \vec{AE}_{|S2} = \begin{Bmatrix} 0 \\ 0 \\ \omega_{30} \end{Bmatrix} \times \begin{Bmatrix} R_3 \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ R_3 \omega_{30} \\ 0 \end{Bmatrix}. \quad (4.18)$$

Similarly, Mozzi's first equation can be used to obtain $\vec{v}_{60|S2}^E$,

$$\vec{v}_{60|S2}^E = \vec{v}_{60|S2}^F + \vec{\omega}_{60|S2} \times \vec{FE}_{|S2}. \quad (4.19)$$

Point F is common to both the outer planet (6) and the carrier (1), therefore,

$$\vec{v}_{60|S2}^F = \vec{v}_{10|S2}^F \quad (4.20)$$

and through Mozzi's first equation, $\vec{v}_{10|S2}^F$ can be expressed as

$$\vec{v}_{10|S2}^F = \vec{v}_{10|S2}^A + \vec{\omega}_{10|S2} \times \vec{AF}_{|S2} = \begin{Bmatrix} 0 \\ 0 \\ \omega_{10} \end{Bmatrix} \times \begin{Bmatrix} R_F \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ R_F \omega_{10} \\ 0 \end{Bmatrix}. \quad (4.21)$$

Replacing (4.21) in (4.19),

$$\vec{v}_{60|S_2}^E = \begin{Bmatrix} 0 \\ R_F\omega_{10} \\ 0 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ \omega_{60} \end{Bmatrix} \times \begin{Bmatrix} -R_6 \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ R_F\omega_{10} - R_6\omega_{60} \\ 0 \end{Bmatrix}. \quad (4.22)$$

Finally, equation (4.6) can be expressed as

$$\begin{aligned} \vec{v}_{30|S_2}^E &= \vec{v}_{60|S_2}^E \\ \Leftrightarrow R_3\omega_{30} &= R_F\omega_{10} - R_6\omega_{60} \\ \Leftrightarrow R_3\omega_{30} &= (R_3 + R_6)\omega_{10} - R_6\omega_{60}. \end{aligned} \quad (4.23)$$

Assuming rolling without sliding on point G as well, with reference axis S_2 - equation (4.7):

$$\vec{v}_{40|S_2}^G = \vec{v}_{60|S_2}^G. \quad (4.24)$$

Since the ring gear (4) only rotates around the fixed point A, $v_{40|S_2}^G$ can be easily obtained through Mozzi's first equation as follows,

$$\vec{v}_{40|S_2}^G = \vec{v}_{40|S_2}^A + \vec{\omega}_{40|S_2} \times \vec{AG}_{|S_2} = \begin{Bmatrix} 0 \\ 0 \\ \omega_{40} \end{Bmatrix} \times \begin{Bmatrix} R_4 \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ (R_3 + 2R_6)\omega_{40} \\ 0 \end{Bmatrix}. \quad (4.25)$$

$\vec{v}_{60|S_2}^G$ can also be obtained using Mozzi's first equation,

$$\vec{v}_{60|S_2}^G = \vec{v}_{60|S_2}^F + \vec{\omega}_{60|S_2} \times \vec{FG}_{|S_2} \quad (4.26)$$

and seen as point F also belongs to the carrier (1), $\vec{v}_{60|S_2}^F$ is equal to $\vec{v}_{10|S_2}^F$, which has been calculated in (4.21), therefore,

$$\vec{v}_{60|S_2}^F = \vec{v}_{10|S_2}^F = \begin{Bmatrix} 0 \\ R_F\omega_{10} \\ 0 \end{Bmatrix}. \quad (4.27)$$

Replacing (4.27) in (4.26),

$$\vec{v}_{60|S_2}^G = \begin{Bmatrix} 0 \\ R_F\omega_{10} \\ 0 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ \omega_{60} \end{Bmatrix} \times \begin{Bmatrix} R_6 \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} 0 \\ R_F\omega_{10} + R_6\omega_{60} \\ 0 \end{Bmatrix}. \quad (4.28)$$

Finally, replacing (4.25) and (4.28) in equation (4.7), it is possible to obtain the following equation

$$\begin{aligned} \vec{v}_{40|S_2}^G &= \vec{v}_{60|S_2}^G \\ \Leftrightarrow R_4\omega_{40} &= R_F\omega_{10} + R_6\omega_{60} \\ \Leftrightarrow R_4\omega_{40} &= (R_3 + R_6)\omega_{10} + R_6\omega_{60}. \end{aligned} \quad (4.29)$$

The last contact point of rolling without sliding, point D, will now be analysed through equation (4.8), considering the reference axis S_1 :

$$\vec{v}_{50|S_1}^D = \vec{v}_{60|S_1}^D. \quad (4.30)$$

Using Mozzi's first equation, $\vec{v}_{50|S1}^D$ can be expressed as

$$\vec{v}_{50|S1}^D = \vec{v}_{50|S1}^C + \vec{\omega}_{50|S1} \times \overrightarrow{CD}_{|S1} \quad (4.31)$$

where $\vec{v}_{50|S1}^C$ has been calculated in equation (4.13), and $\overrightarrow{CD}_{|S1}$ has to be projected to S_1 . For that, refer to figure 4.4, in which the angles necessary for the rest of the kinematic analysis have been highlighted.

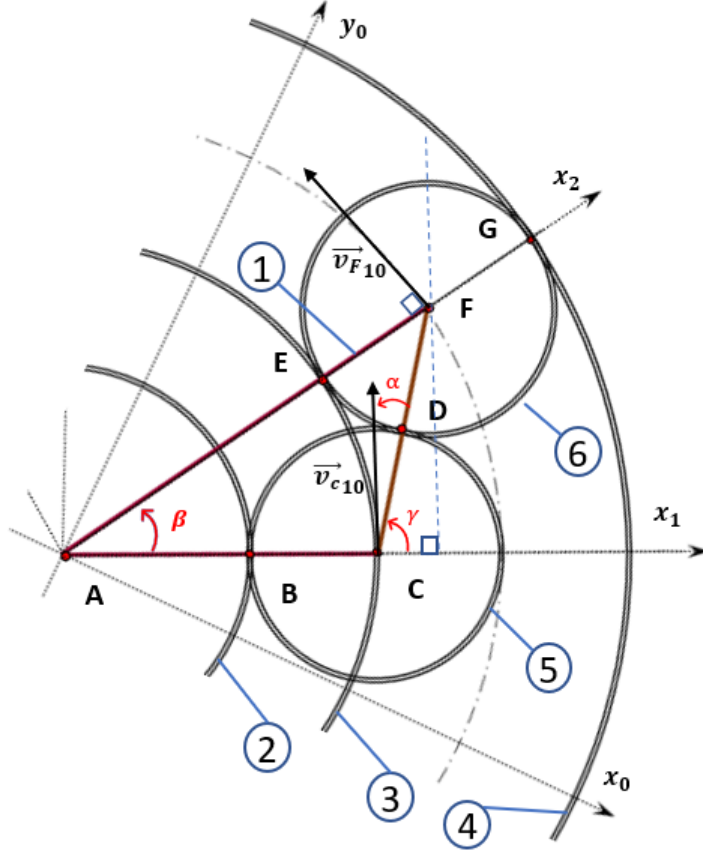


Figure 4.4: Representation of the angles necessary for the kinematic analysis.

Considering angle α , equation (4.31) can now be expressed as,

$$\vec{v}_{50|S1}^D = \begin{Bmatrix} 0 \\ R_C \omega_{10} \\ 0 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ \omega_{50} \end{Bmatrix} \times \begin{Bmatrix} R_5 \sin \alpha \\ R_5 \cos \alpha \\ 0 \end{Bmatrix} = \begin{Bmatrix} -R_5 \omega_{50} \cos \alpha \\ R_C \omega_{10} + R_5 \omega_{50} \sin \alpha \\ 0 \end{Bmatrix}. \quad (4.32)$$

Mozzi's first equation can also be used to obtain $\vec{v}_{60|S1}^D$, as follows,

$$\vec{v}_{60|S1}^D = \vec{v}_{60|S1}^F + \vec{\omega}_{60|S1} \times \overrightarrow{FD}_{|S1} \quad (4.33)$$

where $\vec{v}_{60|S1}^F$ has been calculated in equation (4.27). Therefore, considering now angles α and β , equation (4.33) can be rewritten as

$$\vec{v}_{60|S1}^D = \begin{Bmatrix} -R_F \omega_{10} \sin \beta \\ R_F \omega_{10} \cos \beta \\ 0 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ \omega_{60} \end{Bmatrix} \times \begin{Bmatrix} -R_6 \sin \alpha \\ -R_6 \cos \alpha \\ 0 \end{Bmatrix} = \begin{Bmatrix} -R_F \omega_{10} \sin \beta + R_6 \omega_{60} \cos \alpha \\ R_F \omega_{10} \cos \beta - R_6 \omega_{60} \sin \alpha \\ 0 \end{Bmatrix}. \quad (4.34)$$

Finally, replacing (4.32) and (4.34) in the initial equation (4.8), the two following equations are obtained,

$$-R_5\omega_{50}\cos\alpha = -R_F\omega_{10}\sin\beta + R_6\omega_{60}\cos\alpha \quad (4.35)$$

and,

$$R_C\omega_{10} + R_5\omega_{50}\sin\alpha = R_F\omega_{10}\cos\beta - R_6\omega_{60}\sin\alpha. \quad (4.36)$$

Analysing figure 4.4, it can be concluded that

$$\begin{aligned} (R_5 + R_6)\sin\gamma &= (R_5 + R_6)\cos\alpha \\ \Leftrightarrow \sin\gamma &= \cos\alpha. \end{aligned} \quad (4.37)$$

Replacing (4.37) in equation (4.35) and rearranging,

$$\frac{R_5\omega_{50} + R_6\omega_{60}}{\sin\beta} = \frac{R_F\omega_{10}}{\sin\gamma} \quad (4.38)$$

In summary, the kinematics of this planetary transmission are defined by the four following equations,

$$R_2\omega_{20} = (R_2 + R_5)\omega_{10} - R_5\omega_{50} \quad (4.39)$$

$$R_3\omega_{30} = (R_3 + R_6)\omega_{10} - R_6\omega_{60} \quad (4.40)$$

$$R_4\omega_{40} = (R_3 + R_6)\omega_{10} + R_6\omega_{60} \quad (4.41)$$

$$\frac{R_5\omega_{50} + R_6\omega_{60}}{\sin\beta} = \frac{R_F\omega_{10}}{\sin\gamma} \quad (4.42)$$

According to equations (4.39)-(4.42), ω_{20} and ω_{40} could be derived. From the sum of equations (4.39) and (4.40), ω_{20} is obtained as follows,

$$\begin{aligned} R_2\omega_{20} + R_3\omega_{30} &= (R_2 + R_5)\omega_{10} + (R_3 + R_6)\omega_{10} - R_5\omega_{50} - R_6\omega_{60} \\ \Leftrightarrow R_2\omega_{20} + R_3\omega_{30} &= (R_C + R_F)\omega_{10} - (R_5\omega_{50} + R_6\omega_{60}) \\ \Leftrightarrow R_2\omega_{20} + R_3\omega_{30} &= (R_C + R_F)\omega_{10} - R_F\omega_{10}\frac{\sin\beta}{\sin\gamma} \\ \Leftrightarrow R_2\omega_{20} + R_3\omega_{30} &= \left[R_F\left(1 - \frac{\sin\beta}{\sin\gamma}\right) + R_C \right] \omega_{10} \\ \Leftrightarrow \omega_{20} &= \frac{[(R_3 + R_6)\frac{\sin\gamma - \sin\beta}{\sin\gamma} + R_2 + R_5]\omega_{10} - R_3\omega_{30}}{R_2}. \end{aligned} \quad (4.43)$$

Next, rearranging equation (4.41), ω_{40} can be defined as follows,

$$\begin{aligned} R_4\omega_{40} &= (R_3 + R_6)\omega_{10} + R_6\omega_{60} \\ \Leftrightarrow R_4\omega_{40} &= (R_3 + R_6)\omega_{10} + R_6\left[-\frac{R_3}{R_6}\omega_{30} + \frac{R_3 + R_6}{R_6}\omega_{10}\right] \\ \Leftrightarrow \omega_{40} &= \frac{2(R_3 + R_6)\omega_{10} - R_3\omega_{30}}{R_4}. \end{aligned} \quad (4.44)$$

Finally, it was necessary to define $\sin\gamma$ and $\sin\beta$ according to the gears' radiuses. From figure 4.4, it can be concluded that,

$$R_F\cos\beta = R_C + R_P\cos\gamma \quad (4.45)$$

and,

$$R_F \sin \beta = R_P \sin \gamma. \quad (4.46)$$

Recall that $R_P = R_5 + R_6$. Taking both equations to the power of two, equation (4.45) can be expressed as

$$R_F^2 \cos^2 \beta = R_C^2 + R_P^2 \cos^2 \gamma + 2R_C R_P \cos \gamma \quad (4.47)$$

and equation (4.46) as

$$R_F^2 \sin^2 \beta = R_P^2 \sin^2 \gamma. \quad (4.48)$$

Taking the sum of equations (4.47) and (4.48), comes,

$$R_F^2 (\cos^2 \beta + \sin^2 \beta) = R_C^2 + R_P^2 (\cos^2 \gamma + \sin^2 \gamma) + 2R_C R_P \cos \gamma \quad (4.49)$$

and knowing that for any angle x ,

$$\cos^2 x + \sin^2 x = 1 \quad (4.50)$$

equation (4.49) can be simplified as follows,

$$R_F^2 = R_C^2 + R_P^2 + 2R_C R_P \cos \gamma \quad (4.51)$$

rearranging,

$$\cos \gamma = \frac{R_F^2 - R_C^2 - R_P^2}{2R_C R_P}. \quad (4.52)$$

And according to equation (4.50), $\sin \gamma$ can be expressed as

$$\sin \gamma = (1 - \cos^2 \gamma)^{1/2}. \quad (4.53)$$

Referring now to equation (4.45), and rearranging,

$$\begin{aligned} R_F \cos \beta &= R_C + R_P \cos \gamma \\ \cos \beta &= \frac{1}{R_F} (R_C + R_P \cos \gamma). \end{aligned} \quad (4.54)$$

Replacing equation (4.52) in equation (4.54), comes,

$$\cos \beta = \frac{1}{R_F} \left(R_C + \frac{R_F^2 - R_C^2 - R_P^2}{2R_C} \right). \quad (4.55)$$

And $\sin \beta$ can be expressed as

$$\sin \beta = (1 - \cos^2 \beta)^{1/2}. \quad (4.56)$$

It was now possible to define the transmission ratios. According to equations (4.43) and (4.44), the transmission ratio i can be expressed as

$$i = \frac{\omega_{20}}{\omega_{40}} = \frac{R_4}{R_2} \frac{(R_F \frac{\sin \gamma - \sin \beta}{\sin \gamma} + R_C) \omega_{10} - R_3 \omega_{30}}{2R_F \omega_{10} - R_3 \omega_{30}}. \quad (4.57)$$

According to equation (4.57), the two speeds gears and reverse gear intended for this two-speed transmission are defined through the following gear ratios:

- **First gear ratio:** When the carrier is grounded by C brake, which means that $\omega_{10}=0$, the first gear ratio (i_1) is achieved. The first gear ratio can be expressed as

$$i_1 = \frac{R_4}{R_2}. \quad (4.58)$$

- **Second gear ratio:** When the large sun is grounded by LS brake, which means that $\omega_{30}=0$, the second gear is achieved. The second gear ratio (i_2) can be expressed as

$$i_2 = \frac{R_4}{R_2} \frac{R_F \frac{\sin \gamma - \sin \beta}{\sin \gamma} + R_C}{2R_F}. \quad (4.59)$$

- **Reverse gear ratio:** When the motor is rotating in the counterclockwise direction, the carrier is stopped by the OWC, and reverse gear is achieved. The reverse gear ratio (i_R) is therefore equal to the first gear ratio and can be expressed as

$$i_R = \frac{R_4}{R_2}. \quad (4.60)$$

Summary of the Kinematic Analysis

- **Geometric constraints** - equations (4.1), (4.2), (4.3) and (4.4):

$$R_C = R_2 + R_5$$

$$R_F = R_3 + R_5$$

$$R_4 = R_3 + 2R_6$$

$$R_P = R_5 + R_6$$

- **Points of rolling contact without sliding** - equations (4.5), (4.6), (4.7) and (4.8):

$$\vec{v}_{20|S1}^B = \vec{v}_{50|S1}^B$$

$$\vec{v}_{30|S2}^E = \vec{v}_{60|S2}^E$$

$$\vec{v}_{40|S2}^G = \vec{v}_{60|S2}^G$$

$$\vec{v}_{50|S1}^D = \vec{v}_{60|S1}^D$$

- **Speed of the input gear** (ω_{20}) - equation (4.43):

$$\omega_{20} = \frac{[(R_3 + R_6) \frac{\sin \gamma - \sin \beta}{\sin \gamma} + R_2 + R_5] \omega_{10} - R_3 \omega_{30}}{R_2}$$

- **Speed of the output gear** (ω_{40}) - equation (4.44):

$$\omega_{40} = \frac{2(R_3 + R_6) \omega_{10} - R_3 \omega_{30}}{R_4}$$

- **Angles between the planet gears (γ and β)** - refer to figure 4.4 and equations (4.52) and (4.55):

$$\cos \gamma = \frac{R_F^2 - R_C^2 - R_P^2}{2R_C R_P}$$

$$\cos \beta = \frac{1}{R_F} \left(R_C + \frac{R_F^2 - R_C^2 - R_P^2}{2R_C} \right)$$

- **Transmission ratio (i)** - equation (4.57):

$$i = \frac{\omega_{20}}{\omega_{40}} = \frac{R_4}{R_2} \frac{(R_F \frac{\sin \gamma - \sin \beta}{\sin \gamma} + R_C) \omega_{10} - R_3 \omega_{30}}{2R_F \omega_{10} - R_3 \omega_{30}}.$$

Gear Design

5.1 Mating Conditions for Planetary Gears

In this section, the mating conditions for the Ravigneaux gear set proposed in this thesis will be described, in order to later define the number of gear teeth of all gears. Since planetary systems involve multiple power paths and simultaneous meshing of several gears, it is important to guarantee these conditions, otherwise, proper operation of the gear set will not be obtained.

Unfortunately, there few references covering these criteria for Ravigneaux gear sets. Ford's patent on the carrier assembly for Ravigneaux transmissions [62] covers some important topics, particularly regarding the assembly of planetary systems with more than six pinions. The work of Naiwei Zou *et al.* [63] summarizes the teeth matching conditions for general planetary systems, and in the work of Robert G. Parker [64], an extensive analysis on mesh phasing relationships in planetary gears was done.

Nevertheless, the information found in these sources was combined with the knowledge gained from the process of designing this transmission, and it was possible to summarize the criteria needed to ensure the complete assembly and proper functioning of a six pinion Ravigneaux gear set on the three following conditions.

5.1.1 Assembly Condition

Planetary and epicyclic gears rely on symmetry to achieve the large torque-weight ratios and compactness that make them widely used in multiple applications. In order to assemble any planetary gear set, the number of planetary gears and the number of teeth of both suns and the ring gear, must meet a certain relation, or, the assembly will be unable to be done due to the interference between planetary gears and sun gear and (or) ring gear. This relation refers to the equal spacing between pinions, which can be achieved by the following conditions:

$$\frac{z_4 - z_2}{n_p} = N \quad (5.1)$$

and,

$$\frac{z_4 - z_3}{n_p} = N, \quad (5.2)$$

where z is the number of teeth on the respective gear (shown in figure 5.1), n_p is the number of planetary gear pairs, and N is an integer. In the case of the six-pinion Ravigneaux gear set proposed here, $n_p=3$. In contrast, Ford's patent [62] covers Ravigneaux gear sets with eight and ten pinions, which require more than this condition to ensure the assembly of

all gears. When designing an eight-pinion ravigneaux gear set for example, the increased number of meshes makes it necessary to analyse the mesh phasing of all different meshes, and guaranteeing that the phase error is equal for the different load paths, otherwise it might not be possible to fully assemble the gear set.

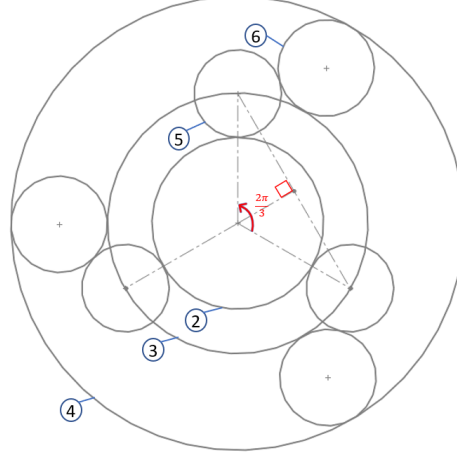


Figure 5.1: Six-pinion Ravigneaux gear set configuration. 1- Carrier (not shown); 2- Small Sun Gear; 3- Large Sun Gear; 4- Ring Gear; 5- Inner Planet Gear; 6- Outer Planet Gear.

5.1.2 Neighbour Condition

When designing a planetary gear set, there must not be any interference between two non-meshing neighbouring gears. Therefore, this condition refers to ensuring that the center distance of two non meshing gears is longer than their tip diameter. In a dual pin planetary gear system, such as the 6 pinion Ravigneaux gear set, the following non-interferences must be guaranteed. Note that these expressions serve only as approximations to the design process, since it is not being taken into account the tip diameters, but the reference diameters instead. For the complete formulas, refer to [65].

1) Non-interference Condition of Inner Planet Gears and Ring Gear

A certain interval must be preserved between the inner planet gear tip and the ring gear tip. Therefore, the relation of teeth number for each gear can be expressed as follows,

$$z_4 - z_2 > 2z_5. \quad (5.3)$$

2) Non-interference Condition of Outer Planet Gears and Small Sun Gear

For Ravigneaux gear sets, it necessary to guarantee that the long pinions/outer planet gears do not come into contact with the small sun gear. For that, it is only necessary to design a large sun gear (which meshes with the outer planets), that is sufficiently bigger than the small sun gear:

$$z_3 > z_2. \quad (5.4)$$

3) Non-interference Condition of Inner Planet Gears

A certain interval must be preserved between neighbouring inner planet gears, as shown in figure 5.1, so that their gear tips do not interfere. From the 3D design phase of this

work, it was possible to conclude that this condition and the following, were easy to ensure in a six-pinion Ravigneaux gear set intended to be used as a two speed transmission for an electric vehicle. The needed gear ratios for desirable performance dictate the gear teeth numbers of both sun gears in a way that the three planet gear pairs are sufficiently spaced apart. Nevertheless, the condition can be expressed as follows,

$$(z_2 + z_5) \sin \left(\frac{\pi}{n_p} \right) \geq z_5. \quad (5.5)$$

4) Non-interference Condition of Outer Planet Gears

Similar to the non-interference condition between neighbouring inner planet gears described above, the tip diameters of neighbouring outer planet gears must not come into contact. And, as was stated, this condition was found to be easily assured in a six pinion Ravigneaux gear set intended for an electric vehicle's two speed transmission. The mathematical formulation of this condition differs greatly between different planetary architectures, and it can be found in [63].

5.1.3 Coaxiality Condition

The coaxiality condition means that the center of rotation of all main members to lie strongly in the same straight line. For a Ravigneaux gear, the coaxility of the outer planet must be guaranteed, since it is the planet gear that meshes with the ring gear. This condition can be expressed as

$$z_4 - z_3 = 2z_6. \quad (5.6)$$

The coaxiality of the inner planet gear is automatically guaranteed by the neighbour condition of non-interference between inner planet gears and the ring gear, and is less restrictive since the inner planet does not mesh with the ring gear.

5.2 Planetary Gears Sizing

5.2.1 Teeth Number Specification

Having kinematically defined the transmission ratios in 4.2, and the conditions that need to be respected when designing a Ravigneaux architecture, in 5.1, it was now possible to choose the number of teeth of all planetary gears, according to the goal transmission ratios estimated when analysing the dynamic performance requirements, in chapter 3.

An expected gear ratio for the final drive had to be considered in order to define the transmission ratios based on the required global gear ratios previously set. It was agreed with the project advisors that the final drive would have a gear ratio of four ($i_{fd}=4$). Thus, the goal transmission ratios were calculated and are displayed bellow in table 5.1.

With these goal values in mind, all of the kinematic equations and mating conditions were inserted into an excel sheet. Through an iterative process, the gear teeth numbers that satisfied all of these requirements were discovered and are displayed in table 5.2.

Table 5.1: Goal transmission ratios, based on the performance requirements, considering a final drive gear ratio of 4.

Goal 1 st gear ratio	$i_1=4$
Goal 2 nd gear ratio	$i_2=3.125$

Table 5.2: Number gear teeth for all planetary gears of the proposed two speed transmission.

Planetary member	Small Sun SS	Large Sun LS	Ring R	Inner Planet IP	Outer Planet OP
Number of gear teeth	$z_2=45$	$z_3=93$	$z_4=183$	$z_5=52$	$z_6=45$

The transmission ratios in table 5.3 correspond to these teeth numbers. It was not possible to simultaneously satisfy both gear ratios specified based on the performance requirements, since achieving the second gear ratio meant the first gear ratio had to be substantially higher than the goal one. The teeth numbers were chosen to satisfy the first gear ratio because a second gear ratio lower than desired, still resulted in the required top speed.

Table 5.3: Definitive gear ratios set for the design of the two speed transmission.

Definitive 1 st gear ratio	$i_1=4.07$
Definitive 2 nd gear ratio	$i_2=2.03$

5.2.2 Strength Calculations and Module Sizing

With the gears' teeth number set, it was now possible establish the remaining main parameters of the gears, which are the gear's face widths, profile shifts and module, in order to be able to start the 3D modelling process. To effectively choose these parameters, it was necessary to accurately simulate the kinematics of the Ravigneaux transmission, so as to consider the load distribution and the different load paths inherent to a planetary system. This was achieved through the software **KISSsys**, which is a module of the software **KISSsoft**. **KISSsys** is used to model complex systems such as complete transmissions. In practice, the full system is modelled into the **KISSsys** suite, while the calculations are performed by various **KISSsoft** sub-models.

It is worth noting that the parameter specification did not take a meticulous approach because that wasn't the thesis's main purpose. Instead, the parameter analysis was carried out just for the purpose of sizing the gears and confirming that they could safely withstand the power supplied by the chosen electric motor in order to begin the 3D modeling of the transmission, which is the topic of this thesis. Once the transmission was modeled in **KISSsys** (shown in figure 5.2), the face widths were chosen based on rough estimates of the transmission's overall width. The gears' module was determined via an iterative procedure to determine the smallest value that would assure the gears could withstand the power flow. Since the planet gears mesh with multiple gears, the profile shifts could not be determined by analyzing individual gear pairs. Instead, the **KISSsoft** sub-models calculated them automatically, finding solutions for all gears based on the optimal specific sliding and on ensuring that there is no interference between gears.

The transmission was run in second gear for the strength calculations of the gears because it is the gear that is anticipated to be used the most (as shown in 3.5). The safety factors for the tooth root stress (SF) and contact stress (SH) were calculated taking as reference the DIN 3990:1987 Method B standard. The material used for all gears was the steel alloy 18CrNiMo7-6, with a yield limit $\sigma_y=850$ MPa and ultimate strength $\sigma_{max}=1200$ MPa.

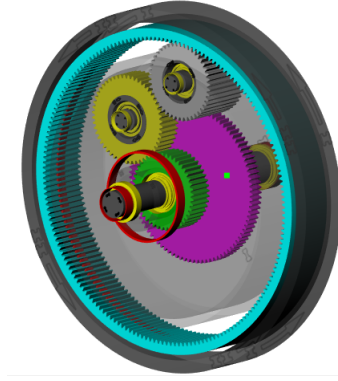


Figure 5.2: KISSsys model of the two speed Ravigneaux transmission. Only one planetary gear pair out of the three is shown.

In table 5.4, the results from the gears' module analysis is shown, where relevant characteristics of the gears' meshes are highlighted. It was agreed with the project advisors that the gears used would be spur gears with a pressure angle α_n equal to 20° , for simplification purposes so as to not introduce major thrust forces. The face widths were fixed for all the computations, and the input data was to the motor's peak torque characteristics ($P=150$ kW, $T=310$ Nm and $N=4700$ rpm), since it corresponds to the most demanding situation, in which the gears experience the highest loads.

Table 5.4: Gear strength calculations for three different values of gear module, considering the transmission was operating in second gear.

Meshing	SS + IP		OP + IP		OP + LS		OP + R	
Gear	SS	IP	OP	IP	LS	OP	OP	R
Number of teeth	45	52	45	52	93	45	45	-183
Facewidth [mm]	20	20	50	20	10	50	50	50
Sum of profile shifts [mm]	1.1611		1.1572		1.3926		1.3926	
Module - m [mm]	1							
Center distance [mm]	49.577		49.574		70.305		70.332	
Diameter [mm]	45	52	45	52	93	45	45	183
Power Loss [W]	217.6		217.6		86.7		39.3	
Meshing efficiency [%]	99.37		99.37		99.49		99.77	
SF	0.85	0.84	0.91	0.81	0.80	0.93	1.9	1.9
SH	0.95	0.96	0.93	0.94	1.05	1.03	2.25	2.27
Module - m [mm]	1.25							
Center distance [mm]	61.971		61.967		87.881		87.915	
Diameter [mm]	56.25	65	56.25	65	116.25	56.25	56.25	228.75
Power Loss [W]	180.2		180.2		71.8		34	
Meshing efficiency [%]	99.48		99.47		99.58		99.8	
SF	1.2523	1.2424	1.3529	1.1934	1.1865	1.3821	2.8027	2.9668
SH	1.1639	1.1729	1.1367	1.1455	1.2703	1.2501	2.7647	2.7969
Module - m [mm]	1.5							
Center distance [mm]	74.366		74.36		105.46		105.5	
Diameter [mm]	67.5	78	67.5	78	139.5	67.5	67.5	274.5
Power Loss [W]	154.1		154.3		61.5		30.2	
Meshing efficiency [%]	99.55		99.55		99.64		99.82	
SF	1.7225	1.7089	1.8793	1.6216	1.5931	1.9225	3.891	4.13
SH	1.3692	1.3798	1.3265	1.3368	1.4787	1.4551	3.2543	3.2922

Part 11 of DIN 3990 states that the minimum safety factor for the tooth root strength is 1.4 and the minimum safety factor for the flank contact pressure is 1.0. On the basis of this data, it was possible to draw the conclusion that only gear modules with a 1.5 or higher would guarantee the gears' strength under the electric motor's peak power and peak torque output. It was also possible to draw the conclusion that the meshing efficiency of all the gear pairs rose with the increase in module. More detailed information regarding the gears' strength calculations can be found in Appendix A.

The final results for gear sizing are provided in the table 5.5. The completed 3D model of the gears is displayed in figure 5.3. It should be mentioned that the final design of the gears was the outcome of a parallel and integrated analysis on the 3D CAD model, where adjustments to the face widths were necessary to meet space requirement needs.

Table 5.5: Summary of the gears' sizing calculations, considering the peak power and peak torque output from the electric motor.

Gear	Inner Planet IP	Outer Planet OP	Small Sun SS	Large Sun LS	Ring R
Number of teeth	52	45	45	93	-183
Face width [mm]	23	50	23	13	50
Module [mm]	1.5				
Profile shift [mm]	0.5966	0.5606	0.5645	0.8320	-1.9839
Reference diameter [mm]	78	67.5	67.5	139.5	274.5
Base diameter [mm]	73.296	63.429	63.429	131.087	257.946
Tip diameter [mm]	82.540	71.932	71.941	144.734	277.452

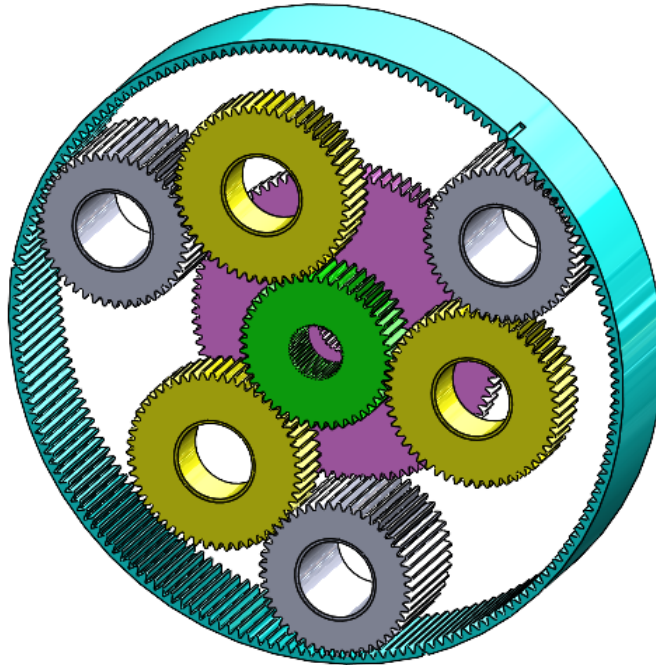


Figure 5.3: 3D model of the gears' final configuration on the carrier.

5.3 Final Drive Sizing

This thesis's main focus is to design and model the proposed two speed transmission, including the housing. To be able to construct the housing according to the size and positioning of the final gear pair, it was therefore required to design the final driving gear pair. The sizing was performed in KISSsoft, according to the standard ISO 6336:2019, and considering the peak power and peak torque output from the electric motor. Firstly, a *Rough sizing* was performed, allowing the module and dimensions of the gear pair to be estimated based on the transmission ratio and the number of pinion teeth range provided. The transmission ratio for the final drive had already been previously set as $i_{fd}=4$. Secondly, a *Fine Sizing* was carried out in order to calculate all the gear pair characteristics that satisfied a provided range of module values and interaxis. Finally, tuning the face width it was possible to find a compromise in order to reach the minimum safety factor for bending ($SF_{min}=1.4$) and for flank strength ($SH_{min}=1$) The results of the final drive gear pair sizing can be found in Appendix A, and are summarized bellow in table 5.6. The driven gear was solely used as a tool to properly design the housing; just the driving gear was modelled in 3D CAD, as seen in figure 5.4.

Table 5.6: Summary of the final drive gear pair sizing calculations, considering the peak power and peak torque output from the electric motor.

	Gear 1	Gear 2
Number of teeth	27	111
Face width [mm]	43	40
Module [mm]	3.5	
Center distance [mm]	247	
Pressure angle [°]	25	
Helix angle [°]	12.5	
Sum of profile shifts [mm]	-0.1035	
Transmission ratio - i_{fd}	4.111	
Reference diameter [mm]	96.794	397.933
Base diameter [mm]	87.343	359.077
Tip diameter [mm]	102.882	405.116
SF	2.570	2.511
SH	1.373	1.464

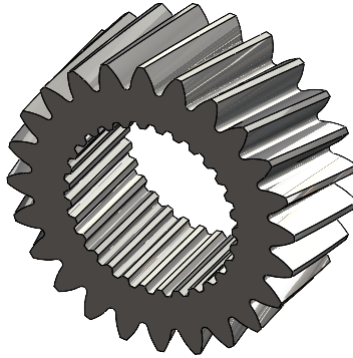


Figure 5.4: 3D model of the driving gear of the final drive gear pair.

Shaft and Bearing Sizing

6.1 Rotational Speeds of the Planetary Members

Mechanical transmissions have always used rolling bearings. They maintain the shaft's alignment by supporting it. Additionally, they allow the shaft to freely rotate, reducing friction, and they withstand forces acting on the shaft and convey them to the housing. Rolling bearings in electric car transmissions must withstand high rotating speeds, high working temperatures, and, when employing a light alloy housing (such as aluminum), a heat expansion compensation is essential. This chapter will cover the design of the shafts, the rolling bearings that support them, and the splines implemented to transfer torque between shafts and gears.

In order to determine which rotating components of the transmission needed greater consideration when choosing the bearings, the speed of each planetary member was analysed using the equations derived from the kinematic analysis described in 4.2. Table 6.1 displays the corresponding rotational speeds of all planetary members at the motor's base speed of 4700 rpm. It was also crucial to analyse the angular speeds of the planets relative to the carrier, since the planet gears (IP and OP) rotate around pins that are fixed to the spinning carrier. All angular speeds could be calculated using the following expression,

$$\omega_{j1} = \omega_{j0} - \omega_{10}, \quad (6.1)$$

where ω_{j1} is the speed of the member j relative to the carrier (1) and ω_{j0} is the absolute angular speed of the member j .

It can be observed that both sun gears and planet gears experience the highest rotational speeds. However, the sun gears benefit from the planetary system's advantage of allowing power split. Because of the circular symmetry in which the planets are placed around the sun, the radial loads of the planets offset each other. Therefore, the bearings and the gear housing for the co-axial elements must be designed only to maintain proper alignment of the gearing and withstand loads imposed by external conditions. In fact, the shaft and bearing analysis performed in **KISSsoft** showed that the bearings supporting the sun shafts sustained very little loading. On the other hand, the unsymmetrical meshing and the high rotational speed experienced by the planets make the planet bearings a critical component of the transmission.

Based on the above, attention was taken in selecting the bearings that would hold the planets on the pins, and this process is described in more depth in the following section. The bearings supporting the sun shafts were solely selected in accordance with the 3D modeling phase's requirements for space and ensuring the permissible speeds of the bearings were higher than the shafts' speeds.

Table 6.1: Absolute angular speeds and relative angular speeds (relative to the carrier) of the planetary members, considering an input speed equal to the electric motor's base speed (4700 rpm) and maximum speed (16 000 rpm). 1- Carrier; 2- Small Sun; 3- Large Sun; 4- Ring; 5- Inner Planet; 6- Outer Planet.

	N = 4700 rpm			N = 16 000 rpm		
	Absolute Angular Speed [rpm]	Relative Angular Speed [rpm]		Absolute Angular Speed [rpm]	Relative Angular Speed [rpm]	
1st gear	ω_{10}	0	-	ω_{10}	0	-
	ω_{20}	4700	ω_{21}	ω_{20}	16 000	ω_{21}
	ω_{30}	-2274	ω_{31}	ω_{30}	-7744	ω_{31}
	ω_{40}	1156	ω_{41}	ω_{40}	3936	ω_{41}
	ω_{50}	-4067	ω_{51}	ω_{50}	-13 850	ω_{51}
	ω_{60}	4700	ω_{61}	ω_{60}	16 000	ω_{61}
2nd gear	ω_{10}	1533	-	ω_{10}	5220	-
	ω_{20}	4700	ω_{21}	ω_{20}	16 000	ω_{21}
	ω_{30}	0	ω_{31}	ω_{30}	0	ω_{31}
	ω_{40}	2311	ω_{41}	ω_{40}	7871	ω_{41}
	ω_{50}	-1208	ω_{51}	ω_{50}	-4115	ω_{51}
	ω_{60}	4700	ω_{61}	ω_{60}	16 000	ω_{61}

6.2 Planet Pin Bearings

The planet rolling bearings are the ones that allow the rotation of the planets on their pin shafts. As stated above, these bearings are a critical component in a planetary transmission, and therefore deserve careful attention.

At first, needle bearings were taken into consideration because they are a popular option for planet supports. In the work of Marco Cova[66], a planetary geartrain for a four-wheel drive Formula Student EV was designed, in which needle bearings were used to hold the planets on the pins. And in the study of Giovanni Cattabriga [67], needle bearings were incorporated into the design of a planetary speed multiplier for a regenerative shock absorber to be used in a Formula Student EV. Given their greater width, they offer significant support while maintaining a small radial dimension. It should be noted that since the rolling elements rotate directly on the surface of the pin, the surface must be manufactured appropriately (machining, heat treatment and tolerances).

Following that, the integration of cylindrical roller bearings was examined, and a comparison between their rating life and that of needle bearings was made. Both analyses were conducted with the transmission's most demanding conditions in mind: peak torque and peak power output; peak power and maximum speed. The objective was to confirm that the transmission's predicted lifetime under these circumstances was satisfied by the bearings' rated life. According to approximate estimates, an EV operates at maximum electric motor power output 15 % of the time, which indicates that 15 % of the transmission's projected lifetime will be spent operating in these conditions. This would be equivalent to 263 hours of driving in first gear and 677 hours in second gear, as determined by the analysis of the transmission's predicted mileage outlined in 3.5.

The calculations performed to verify if the bearings satisfy these requirements (table 6.2) are detailed next, for each planet. The rolling bearings considered are provided in Appendix C, and detailed in table (6.3).

Table 6.2: Transmission's expected lifetime operating under peak power conditions, which has to be verified by the bearings' rating life.

	1 st gear	2 nd gear
Transmission's expected lifetime under peak power output [h]	263	677

Table 6.3: Characteristics of the two planet pin bearings analysed.

Designation	SKF NUJ 202 ECP	SKF K 15x19x10
Type	Cylindrical roller bearing (single row)	Needle cage bearing
Inner diameter [mm]	15	15
External diameter [mm]	35	19
Width [mm]	11	10
Dynamic load rating [kN]	12.5	7.21
Static load rating [kN]	10.2	9.3
Reference speed [rpm]	22 000	24 000
Limiting speed [rpm]	26 000	28 000

i) Inner Planet Bearings

Figure 6.1 illustrates the forces that the inner planet is subject to when driving in first gear and second gear. The two blue arrows represent the meshing loads and the yellow arrows represent the torque transmitted by the carrier, which is fixed in first gear.

**Figure 6.1:** Forces acting on the inner planet during operation, represented in KISSsoft.

As mentioned earlier, the rating life of both needle cage bearings and cylindrical roller bearings was studied under two output conditions: peak power (P_{max}) and peak (T_{max}) torque, which corresponds to the motor's state at its base speed; and peak power and maximum speed (N_{max}) of the motor. This analysis was done for both gears of the two speed transmission, and is summarized in table 6.4.

From the data shown in the table, it was possible to make the following conclusions:

- Operating under P_{max} and T_{max} is more demanding to the inner planet in terms of strength, when compared to P_{max} and N_{max} conditions;
- The operating regime of the first gear is more severe on the inner planet than the second gear;
- The needle cage bearing gives an extremely limited bearing life when driving in first gear;

Table 6.4: Summary of the calculations done on inner planet's bearing rating life, considering two motor output conditions.

		P_{max} and T_{max}		P_{max} and N_{max}	
		Minimum bearing life [h]	Static Safety factor	Minimum bearing life [h]	Static Safety factor
1^{st} gear	Needle cage bearing	73	3.06	1328	10.57
	Cylindrical roller bearing	462	3.37	8389	11.6
2^{nd} gear	Needle cage bearing	108	3.06	1975	10.57
	Cylindrical roller bearing	685	3.37	12445	11.6

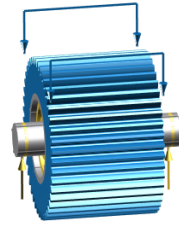
- The cylindrical roller bearing ensures the required lifetime driving in both gears.

Based on the above, the bearing chosen to support the inner planets on their pins was the single row cylindrical roller bearing SKF NJ 202 ECP, whose detailed information is provided in Appendix C.

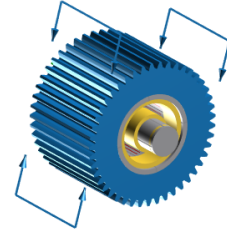
ii) Outer Planet Bearings

Figure 6.2 now illustrates the forces that the outer planet is subject to when driving in first gear and second gear. Since the carrier is locked in first gear, the pin ends, which are fixed to the carrier, experience reacting torque. There are only two meshing loads in first gear, which correspond to the meshing with the inner planet and with the ring gear, because the large sun gear rotates freely and avoids the introduction of meshing loads.

The large sun is locked when in second gear, adding a third meshing load. In addition, the carrier rotates freely, preventing reacting torques.



(a) First gear



(b) Second gear

Figure 6.2: Forces acting on the inner planet during operation, represented in KISSsoft

The outer planets underwent the same analysis as the inner planets in order to compare the use of needle bearings or cylindrical roller bearings as the outer planets' supporting elements on their pins. This analysis is summarized below in table 6.5.

From the table shown, the following conclusions were drawn:

- Similar to the inner planet, the outer planet is more severely loaded under P_{max} and T_{max} conditions and whilst operating in first gear;
- The outer planet's bearings have a shorter rated life than those of the inner planet when used in first gear and under P_{max} and T_{max} conditions. On the other hand,

Table 6.5: Summary of the calculations done on outer planet's bearing rating life, considering two motor output conditions.

		P_{max} and T_{max}		P_{max} and N_{max}	
		Minimum bearing life [h]	Static Safety factor	Minimum bearing life [h]	Static Safety factor
1st gear	Needle cage bearing	45	2.78	901	9.82
	Cylindrical roller bearing	313	3.13	5673	10.78
2nd gear	Needle cage bearing	743	5.7	13347	19.59
	Cylindrical roller bearing	4682	6.26	84758	21.55

when in second gear and under the identical conditions, the outer planet's bearings guarantee a longer lifetime. This led to the conclusion that the first gear is more demanding on the outer planet while the second gear is more demanding on the inner planet;

- The cylindrical roller bearings also guaranteed a longer bearing life for the bearings on the outer planet, as was evident from the examination of the bearings on the inner planet.

This analysis led to the decision to also employ the cylindrical roller bearings to support the outer planets on the planet pins.

6.3 Sun Shafts and Bearings

As previously stated, the sun shafts' bearings do not experience significant radial loads because of the circular symmetry in which the planets are placed around the sun gears. The radial loads can be offset thanks to this arrangement. Additionally, since the power is transferred by spur gears, the choice of bearings would not be influenced by the axial loads. In fact, through the calculations performed in the **KISSsoft** sub-models that comprised the **KISSsys** transmission model (available in Appendix A), it was possible to verify that the sun shaft bearings supported very small loads. Therefore, the choice of bearings meant meeting the specifications for the shafts' rotational speeds and the space requirements, which were only clear during the 3D CAD modeling process. The selected rolling bearing type for all transmission bearings is the most widely used one, which is the deep groove ball bearing (single row). Its simple design makes them an affordable choice, while still being robust in operation. They require little maintenance and are able to withstand very high speeds, which is relevant for electric vehicle transmissions. This section will also cover the splines selected to transfer the torque in each shaft.

6.3.1 Small Sun Shaft

The small sun shaft (SS shaft) rotates continuously while the transmission is in operation, since it corresponds to the input shaft. The material chosen for the SS shaft was the 18CrNiMo7-6 steel, same as the small sun gear. As previously mentioned, the bearings were selected in accordance with the shafts' rotational speeds and the space constraints

observed during the modeling phase. Due to its function as the input shaft, it would rotate at the same rate as the electric motor, reaching a maximum speed of 16 000 rpm. To sustain the SS shaft, it was decided to utilise two bearings, one of which would be fixed to the housing and the other of which would slide on the carrier's surface. Their placement can be seen in figure 6.3, which shows the SS shaft coupled to the small sun gear.

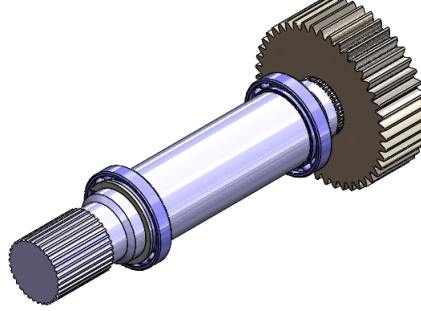


Figure 6.3: 3D representation of the small sun shaft, rolling bearings and small sun gear

Small Sun Shaft Bearings

The two rolling bearings selected were both the same, which was the SKF 61806 single row deep groove ball bearing (provided in Appendix C). They support a limiting speed of 20 000 rpm, sufficiently covering the speed of the SS shaft. The bearing is shown in figure 6.4, and its characteristics are summarized in table 6.6.



Figure 6.4: SKF 61806 bearing.

Table 6.6: Characteristics of the small sun shaft's rolling bearings.

Designation	SKF 61806
Type	Single row deep groove ball bearing
Inner diameter [mm]	30
External diameter [mm]	42
Width [mm]	7
Dynamic load rating [kN]	4.1
Static load rating [kN]	2.9
Reference speed [rpm]	32 000
Limiting speed [rpm]	20 000

Small Sun Shaft Splines

Splines, due to their complexity, are more expensive to manufacture than keyways, which are also used for torque transmission. However, since the load distribution between

teeth is equivalent in a spline connection, a longer fatigue life is expected compared to a key connection. Additionally, they ensure high torque transmission, which is the major reason why they were the preferred choice in this design. Straight-sided splines and involute splines are the two most common types. The latter offers more advantages, including higher strength and higher manufacturing accuracy [59], which is why they were the selected type.

The standard DIN 5480 was used to size the splines, and according to this standard, fitted splined connection are designated by: the main standard number, N for a hub or W for a shaft, the reference diameter, the module, number of teeth and finally the tolerance class. The calculations regarding the SS shaft splines are provided in Appendix B. The selected splines for the input and the small sun gear coupling were, respectively:

- DIN 5480 - W 28 x 0.8 x 34 x 8h. with a length of 30 mm;
- DIN 5480 - N 25 x 0.5 x 48 x 8h, with a length of 23 mm.

6.3.2 Large Sun Shaft

The large sun shaft (LS shaft) is locked in second gear, and rotates to a maximum of 7740 rpm in first gear. It couples with the large sun gear and with the brake drum that is part of the LS shaft band brake. Similar to the SS shaft, two rolling bearings were also used to support the LS shaft, one of which was fixed to the housing and the other was allowed to slide freely in one direction on the surface of the ring gear case. The LS shaft's rolling bearing placement is shown in figure 6.5, in which the large sun gear is also illustrated.

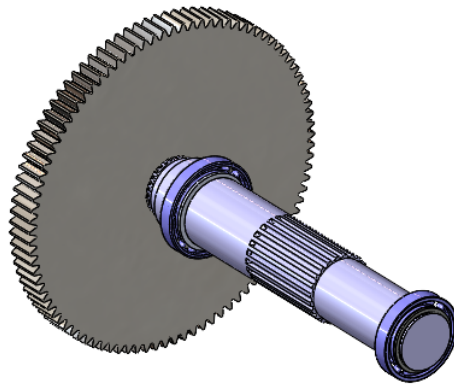


Figure 6.5: 3D representation of the large sun shaft, rolling bearings and large sun gear.

Large Sun Shaft Bearings

Table 6.7 lists the characteristics of the two rolling bearings chosen to support the LS shaft. Due to the planned assembly sequence, it was discovered during the 3D modeling process that the rolling bearing on the shaft's end, which will be fixed to the housing, had to be smaller than the bearing next to the large sun gear. The rolling bearing next to the large sun gear is the same as the ones used in the SS shaft, the SKF 61806. The bearing placed on the shaft's end is the SKF 61805. Both have limiting speeds significantly higher than the LS shaft's maximum rotation speed.

Table 6.7: Characteristics of the large sun shaft's rolling bearings.

Designation	SKF 61806	SKF 61805
Type	Single row deep groove ball bearing	Single row deep groove ball bearing
Inner diameter [mm]	30	25
External diameter [mm]	42	37
Width [mm]	7	7
Dynamic load rating [kN]	4.1	4.36
Static load rating [kN]	2.9	2.6
Reference speed [rpm]	32 000	38 000
Limiting speed [rpm]	20 000	24 000

Large Sun Shaft Splines

As already mentioned, the LS brake drum and the LS gear are connected by the LS shaft. Only when the LS shaft is engaged by the band brake, in second gear, does torque transfer take place, which corresponds to 320 Nm of torque under conditions of maximum torque output. The selected splines for the large sun gear and large sun brake drum coupling were, respectively:

- DIN 5480 - W 26 x 0.8 x 31 x 8h, with 13 mm of length;
- DIN 5480 - W 31 x 1.25 x 23 x 8h, with 40 mm of length.

One-Way Clutch

7.1 One-Way Clutch Working Principles

One-way clutches (OWC) are employed in a variety of applications to transfer power from one shaft to another in a selective way. Both shafts can rotate, but for backstopping applications, one is usually fixed to a stationary (reaction) member or housing. One-way clutches transmit torque when the rotation of one shaft relative to the other results in a type of wedging action to occur inside of the one way clutch. The wedging action is different for every type of one way clutch. When the relative rotation direction of the shafts reverses the wedging action is released and the clutch can freewheel or “over run”.

One-way clutches exist in a wide variety of design types. All types have an inner and an outer race. In some designs these races are concentrically located with the locking members placed in between. This can be seen in the roller and sprag types shown in figure 7.1. The Mechanical Diode (MD) style shown in figure 7.1 uses two plates placed facing each other. The locking member is placed axially between the two plates.

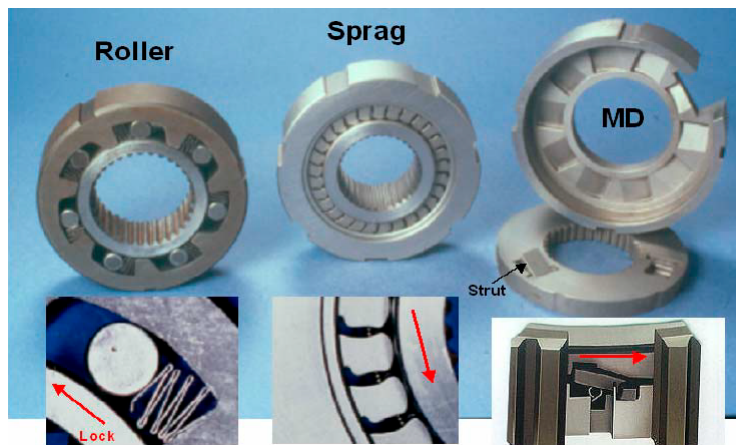


Figure 7.1: Three basic types of one-way clutches [68].

In order to explain the principle of torque transmission, the sprag type one-way clutch will be now taken as an example. Figure 7.2 shows that the cage evenly distributes the sprags across the circle of the inner and outer rings, and that spring force keeps them in constant contact with the rings. Because the sprags are larger than the distance between the inner and outer rings, they are positioned at an angle to the raceway surfaces of the inner and outer rings. When the sprags spin around their centers, the sprag surfaces in

touch with the inner and outer surfaces are shaped in a way that makes the sprag height appear to increase.

The sprags incline clockwise and come into contact with the inner and outer rings to transmit torque as the outer ring rotates clockwise as illustrated in the figure 7.3.

The sprags incline counterclockwise to disengage from the inner and outer rings as the outer ring rotates counterclockwise as shown in figure 7.4, freewheeling the outer ring as a result.

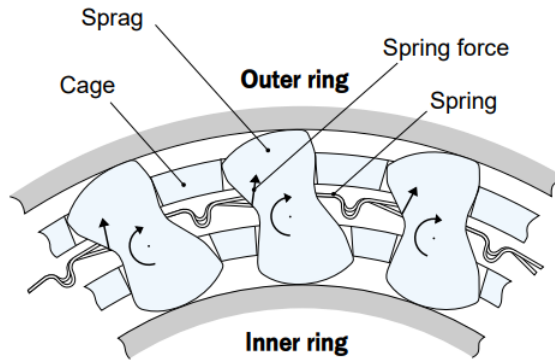


Figure 7.2: Sprag shape and arrangement [69]

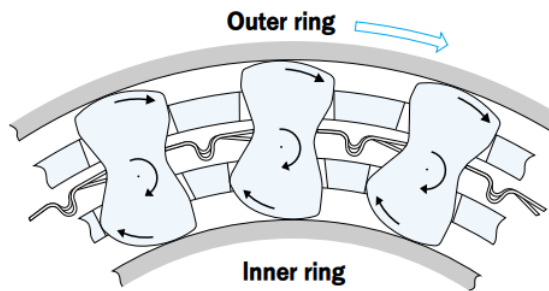


Figure 7.3: Sprag Position in torque transmission [69]

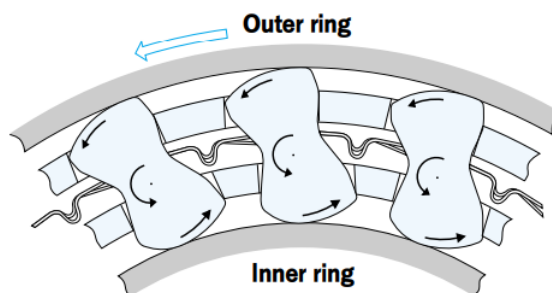


Figure 7.4: Sprag position in freewheeling [69]

7.2 Selection of One-Way Clutch

The two speed transmission designed in this thesis features a OWC that is part of the gear shifting mechanism, and ensures the reverse gear operation as was explained in 4.1. To engage reverse gear, the OWC has to provide enough stopping torque to stop the carrier from rotating, and lock it to the housing. The necessary braking torque, under maximum torque output from the electric motor, was evaluated and it was found that it corresponded to approximately 950 Nm. Besides this, the OWC has to be able to freewheel at the maximum rotational speed of the carrier when second gear is engaged. This maximum speed corresponded to roughly 5200 rpm. These requirements for the selection of the OWC are summarized in table 7.1.

Table 7.1: Requirements for the selection of the OWC.

Braking Torque [Nm]	950
Max. Freewheeling Speed [rpm]	5200

Finding an OWC that allowed for such high freewheeling speeds while also ensuring a high braking torque was challenging. A number of OWC producers, including Koyo [69], Stieber [70], Renold [71], and Tsubaki [72], were consulted. Even though only the latter one offered OWCs that guaranteed high braking torque and high freewheeling speeds, the criteria of permitting 5200 rpm of freewheeling speed could not be met. However, since no alternate solutions could be found, the transmission design was carried out using the OWC chosen from Tsubaki, which is shown in figure 7.5, and detailed in table 7.2. Full information about this OWC is available in Appendix D.



Figure 7.5: Tsubaki one-way clutches [72].

Table 7.2: Main characteristics of the selected OWC - Tsubaki BR48HT-R76B [72].

Designation	Tsubaki BR48HT-R76B
Braking Torque [Nm]	1100
Max. Freewheeling Speed [rpm]	3600

This OWC features a cam design, instead of the roller or sprag type described above. According to Tsubaki, this design assures a longer service life when compared to these conventional types. The cam used in the BR-HT Cam Clutch offers a unique cross section (shown in figure 7.6) that provides positive mechanical engagement only when needed. Otherwise, the Cam Clutch rotates freely with absolutely no mechanical contact in the clutch mechanism, which Tsubaki refers to as the "Lift-Off" design.

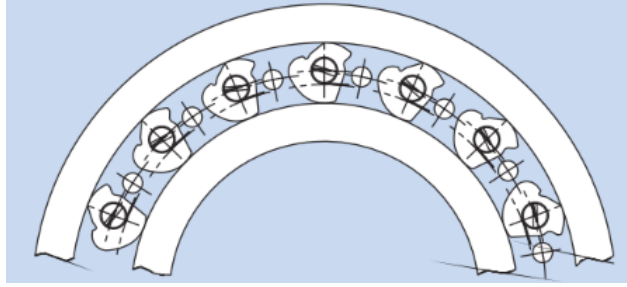
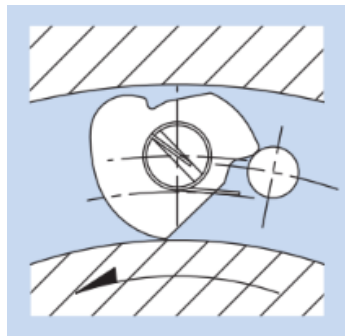


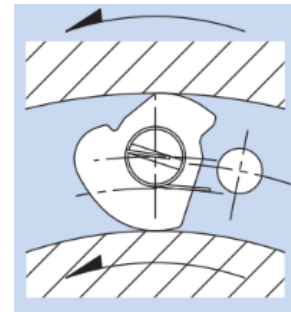
Figure 7.6: Representation of the Tsubaki Cam Clutch design [72].

The working principles of this type of OWC will now be explained, and it should be noted that this OWC is intended to be used with a rotating inner race and a fixed outer race. However, in the transmission designed in this thesis, it was intended to fix the inner race to the housing and to make use of the rotating outer race instead. Since this OWC was the only one that met the necessary dynamic and spatial requirements, the design moved forward with it while keeping in mind that the OWC's mechanism had to be inverted for it to function properly. It is suggested as future work, to design an appropriate OWC to be implemented into this transmission.

The inner and outer races are locked together when the Cam Clutch is in the stationary position. As represented in figure 7.7a, the cam releases from the inner race when the inner side (load side) overruns at high speed. When the inner race stops, the cam rotates back into an engaged position. If the inner races tries to rotate in the reverse direction, the cams then provide backstopping the (figure 7.7b). For this design to be properly implemented into the two speed transmission studied here, the cams would have to be turned over, so that the load side would be the outer race, and the anchored side the inner race.



(a) Freewheeling.



(b) Backstopping position.

Figure 7.7: Operation of the Tsubaki OWC [72].

7.3 Mounting Solution of the One-Way Clutch

As was already mentioned, the OWC locks the carrier to engage reverse gear in the two speed transmission that was designed here. To do this, the carrier is attached to the carrier brake drum, which is connected to the outer race of the OWC. The outer race cannot rotate in the engaging direction, which was defined as the direction in which the reverse gear rotates, because the inner race is secured to the housing by a key. The OWC freewheels in the other direction, which corresponds to the first and second gear driving. However, when the vehicle is in first gear, the band brake engages, locking the carrier brake drum and stopping the OWC from rotating as well.

The OWC is one of the first components to be mounted to the transmission's housing. The carrier brake drum is first connected to its outer race using eight M8 screws, as illustrated in the figure 7.8a. Then, using washers and retaining rings, the two components are locked to a sleeve that is fixed to the housing's front cover. The key is mounted on the sleeve together with the OWC. The assembled OWC and the carrier brake drum are shown in figure 7.8b.

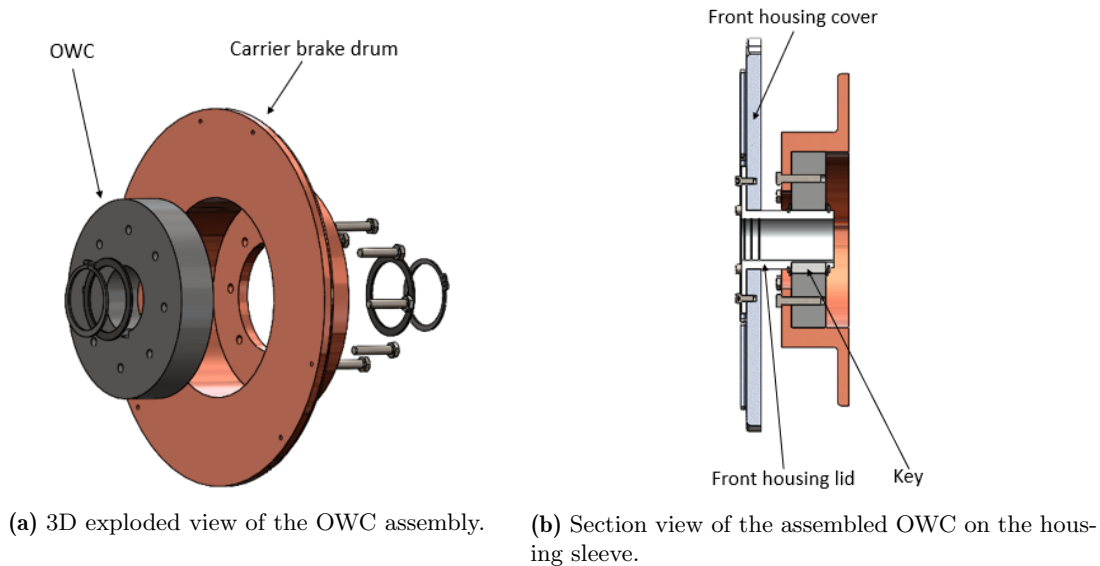


Figure 7.8: OWC assembly and mounting.

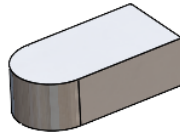


Figure 7.9: 3D model of the OWC's key.

This key was sized in KISSsoft according to the DIN 6885 standard and can be defined as follows:

- DIN 6885 C 18 x 11 x 35

It was intended for the key to have one of its ends machined in order to provide a larger and more stable contact surface with the locking washer, therefore being defined as a type C key, as shown in figure 7.9.

Band Brake Design

8.1 Band Brake Working Principles

The two-speed planetary transmission has two band brakes built into its design as a stopping system and as a key component of the gear shifting system. By progressively activating or deactivating each other in order to transfer torque between each one of them, it is possible to achieve a non-abrupt gear shift. In fact, planetary transmissions often use band brakes because it is frequently necessary to stop components with large diameters, which is advantageous for band brakes.

This chapter will concentrate on designing the two band brakes, including the activation system and selecting a suitable friction material. In this first section, a detailed explanation of how a basic band brake works will be provided.

A band brake is made of a flexible leather band, one or more ropes, or a steel disc that is lined with friction material and encircles a portion of the drum's circumference. A band brake is referred to as a basic band brake, as shown in figure 8.1, if one end of the band is attached to the fulcrum, which is a fixed pin on the lever (point B), and the other end is attached to the lever at a certain distance from the fulcrum. The brakes are applied when a force P is supplied to the lever at point C, causing it to rotate around the fulcrum pin B and tighten the band around the drum [73]. The friction between the band and the drum provides the braking force. The force P on the lever at C may be determined as discussed below. Considering:

- T_1 - Tension in the tight side of the band;
- T_2 - Tension in the slack side of the band;
- θ - angle of wrap of the band on the drum;
- μ - Coefficient of friction between the band and the drum;
- R - Radius of the drum;
- t - Thickness of the band;
- b - Width of the band.

It is known that the relation between tensions T_1 and T_2 is expressed as follows [73],

$$\frac{T_1}{T_2} = e^{\mu\theta}. \quad (8.1)$$

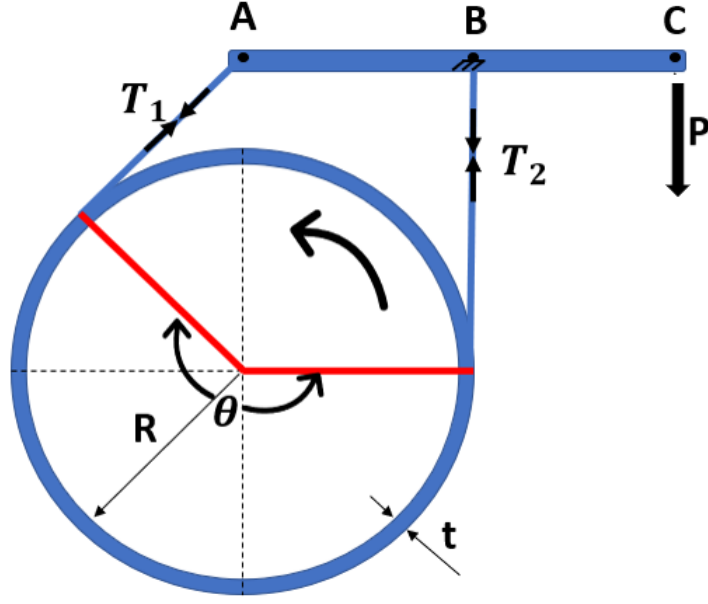


Figure 8.1: Simple band brake, considering the anticlockwise rotation of the drum.

And the braking force (F_b), a scalar, is given by,

$$F_b = T_1 - T_2 \quad (8.2)$$

which means that the braking torque (T_b) can be expressed as

$$T_b = (T_1 - T_2)R. \quad (8.3)$$

Since T_1 is the tension on the tight side, it will be larger than T_2 , which means that for the friction material to not yield, the stress due to T_1 cannot overcome the maximum permissible stress σ_{max} ,

$$\begin{aligned} \sigma_{max} &\geq \frac{T_1}{bt} \\ \frac{\sigma_y}{N} &\geq \frac{T_1}{bt}, \end{aligned} \quad (8.4)$$

where σ_y is the friction material's tensile stress, and N is the safety coefficient. Expressing T_1 as the maximum permissible value,

$$T_1 = \frac{\sigma_y}{N} bt. \quad (8.5)$$

According to equations (8.1) and (8.5), T_2 can be expressed as

$$\begin{aligned} T_2 &= \frac{T_1}{e^{\mu\theta}} \\ T_2 &= \frac{\frac{\sigma_y}{N} bt}{e^{\mu\theta}}. \end{aligned} \quad (8.6)$$

Replacing equations (8.5) and (8.6) in equation (8.3), the braking torque can now be expressed as

$$T_b = \frac{\sigma_y}{N} bt \left(1 - \frac{1}{e^{\mu\theta}}\right) R \quad (8.7)$$

8.2 Friction Material

In order to select the suitable friction material for both band brakes included in this transmission's design, the necessary braking torques had to be first assessed. The carrier brake (C brake) must have enough stopping torque to block the carrier during first gear, while operating under maximum torque output conditions. And the large sun brake (LS brake), must be able to stop the large sun shaft during second gear, while operating under maximum torque output from the engine as well. These maximum torque values were retrieved from the KISSsoft calculations performed on the transmission KISSsys model, which are available in Appendix A, and are summarized bellow in table.

Table 8.1: Necessary braking torque values for both band brakes.

	C Brake	LS Brake
Maximum braking torque [Nm]	950	320

The chosen friction material was the Thermofiber 1700 (figure 8.2), which is provided by ProTec *Friction Group*. The TF1700 is a high-content Kevlar material designed for high performance motorsports clutch, severe service industrial brake and clutch applications. It has a semi-flexible structure which allows for the engaging and disengaging of the band brake. According to ProTec, it is suitable for both dry and wet applications, which is an important characteristic considering the band brakes will be in contact with the transmission's lubricant. Its characteristics are provided in appendix D, and are detailed in table 8.2.



Figure 8.2: Thermofiber 1700, provided by ProTec *Friction Group*.

Table 8.2: Characteristics of the selected friction material.

Material	Thermofiber 1700
Structure	Semi-flexible
Main fiber	Kevlar
Type of service	Dry & wet
Dynamic coefficient of friction - μ	0.70
Tensile Strenght [MPa]	72

The band brakes' geometries, which can be described by their internal radius (R), thickness (t), width (b), and angle of wrap (θ), were developed in an iterative process that was done simultaneously with the bands' 3D modeling phase, where space constraints had to be addressed. Table 8.3 contains the bands' geometries that satisfied the space and stopping torque requirements, as well as the corresponding stopping torques and the bands' tensions, which were obtained considering a coefficient of safety $N=1.3$.

Table 8.3: Band brakes' geometry characteristics and corresponding maximum braking torques.

Band brake	C Brake	LS Brake
R [mm]	110	100
b [mm]	38.5	25
t [mm]	5	4
θ [°]	300	300
T_b [Nm]	1111	525
T_1 [N]	10 365	5385
T_2 [N]	1265	567

The 3D model of the C brake band, which is similar to the LS brake band, is shown in figure 8.3. The friction material in a band brake only wraps around the projected angle, hence a strap fastened to the friction material is required to secure the band to the pins. The band brakes designed here, however, were modelled as a single component for the purpose of simplification.



Figure 8.3: 3D representation of the modelled band.

8.3 Activation system

While traditional automatic planetary transmissions adopt hydraulic actuators to engage and disengage the band brakes, it was decided to adopt electric actuators, since they are efficient, easy to control, and provided a quick response. This electric actuation is based on a screw-nut motor that has also been used in other planetary transmission systems aimed at EVs. The design that inspired this thesis [33] used the same band brake arrangement as the one being considered here. In the work of Ali Pakniyat *et al.* [32], an electrically activated band brake was used to stop the ring gear. A similar method was also employed in the work of Shengnan Fang *et al.* [35] to prevent the carrier from rotating in both directions, as illustrated in figure 8.4.

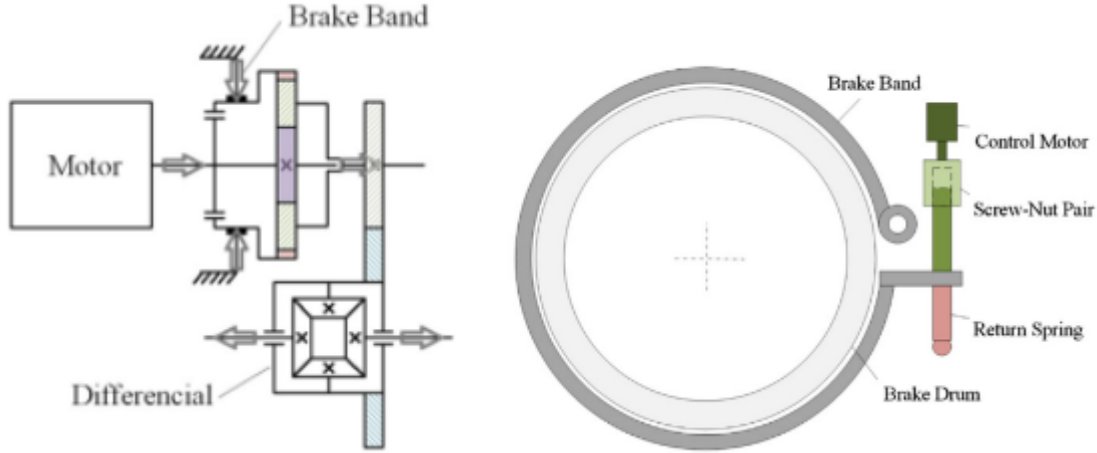


Figure 8.4: Electrically actuated band brake used to stop the carrier from the planetary transmission designed in [35].

8.3.1 Selection of the Electric Motor

The mechanism developed to activate the band brakes will be described in detail in this section, starting by sizing the required screw-nut pair and the suitable electric motor.

The electric actuation system was sized based on the band brake needing the highest braking torque, which corresponds to the C band brake. Then, in an effort to simplify this design process, the same system was employed to activate the LS band.

The relation between the electrically applied normal force (F_N) and the braking torque can be expressed as [32]

$$T_b = F_N(e^{\mu\theta} - 1)R \quad (8.8)$$

rearranging,

$$F_N = \frac{T_b}{(e^{\mu\theta} - 1)R}. \quad (8.9)$$

Knowing that C brake's maximum stopping torque allowed by the friction material is 1111 Nm, the electrically applied normal force is $F_N=265$ N.

In order to choose the appropriate electric motor, it was necessary to calculate the lead screw's torque and its angular speed. Since this system was not one of the main goals of the thesis, the lead screw was not selected based on a strength study but rather on the space needs found during the 3D modeling phase, which defined a lead screw with 13 mm of diameter (d) and a 3 mm lead (p) as a suitable choice.

The screw's torque (C) could now be calculated according to the following expression [74],

$$C = \frac{F_N p}{2\pi\eta}, \quad (8.10)$$

where η is the screw-nut pair's efficiency and can be defined as follows,

$$\eta = \frac{1 - f \tan \alpha}{1 + \frac{f}{\tan \alpha}} \quad (8.11)$$

and f is the screw nut's pair coefficient of friction, which was considered to be $f=0.2$ [74], and α can be expressed as

$$\alpha = \arctan\left(\frac{p}{\pi d}\right). \quad (8.12)$$

Calculating η and α and replacing in equation (8.10), the screw's torque is equal to $C=0.452$ Nm.

The lead screw's rotating speed (V_s) in rpm can be expressed as

$$V_s = \frac{L}{T_p} 60, \quad (8.13)$$

where L is the travel distance of the nut on the screw and T is the desired time to complete the band's engagement. It was decided that the band would be completely engaged once the screw travelled 20 mm ($L=20$ mm), in 2 seconds ($T=2$ seconds). Replacing these values in equation 8.13, the screw's angular velocity would have to be equal to $V_s=200$ rpm.

The electric motor's required power could now be calculated as follows,

$$Q = CV_s \frac{2\pi}{60}. \quad (8.14)$$

Having calculated $C=0.452$ Nm, and $V_s=200$ rpm, the power needed for the electric band actuation system is equal to $Q=9.46$ W.

It was difficult to find an electric motor with these specific characteristics, and therefore, it was chosen solely based on the motor's power output. The selected electric motor (figure 8.5) is provided by SPG Motor and outputs 10 W of power. Its detailed information is provided in Appendix D, and is summarized in table 8.4.



Figure 8.5: Representation of the selected electric motor for the band brakes' activation.

Table 8.4: Characteristics of the selected electric motor for the band brakes' activation.

Electric Motor	S6D10-12 (SPG Motor)
Torque [Nm]	0.032
Speed [rpm]	3000
Power [W]	10

8.3.2 3D Model of the Activation System

In this section, the 3D model of the bands' activation system will be shown. Since extra components were needed for its appropriate installation and operation, its integration into the transmission will also be examined in detail. Lead screws were intended to be axially supported by ball bearings and positioned vertically within the housing of the gearbox. The assembled band brake system is shown in figure 8.6, with the fixed end (bottom end) of the band installed on a cylindrical piece that remains stationary on the lead screw shaft and the moving end of the band attached to the nut (top end).

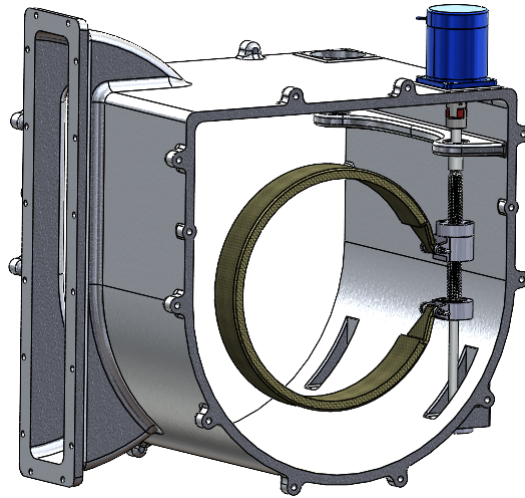


Figure 8.6: 3D representation of the assembled band brake system.

A closer inspection of how the band's ends are attached to the lead screw shaft is seen in figure 8.7. Locking pins were used to lock the band and transmit the normal force. As seen in the figure 8.8, these pins have folding latches that secure them once they are attached. These locking pins are provided by NORELEM, and appendix D contains detailed information about them.

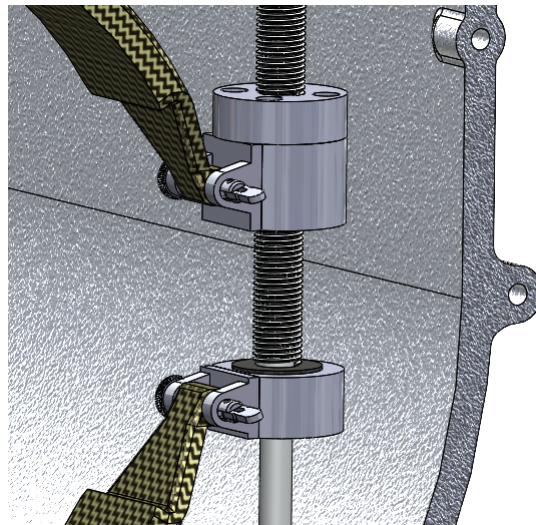


Figure 8.7: Closer look at the band's ends attachment to the lead screw shaft.

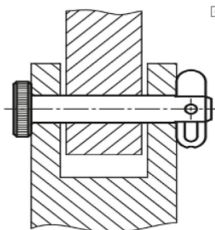


Figure 8.8: 2D representation of the pins in their locked position.

As shown in figure 8.9, the cylindrical component that holds the fixed end of the band is installed on the shaft along with a bushing that has a larger diameter face on its bottom end. This allows the cylindrical piece to be held vertically by a retaining ring. The top of this cylindrical piece is supported against a shoulder of the lead screw shaft, as was seen above in figure 8.7.

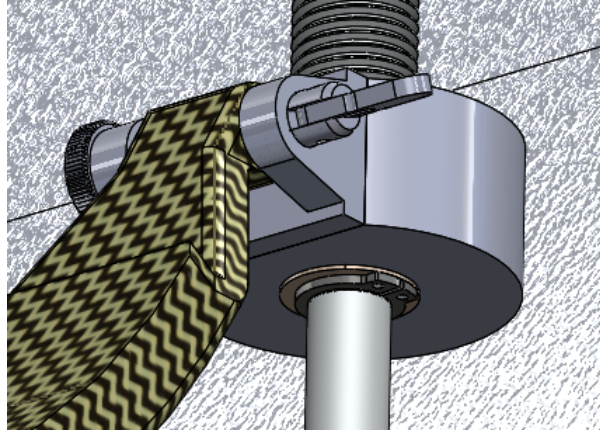
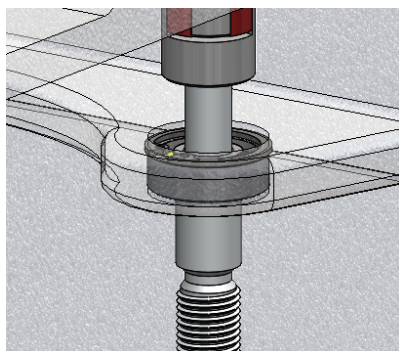


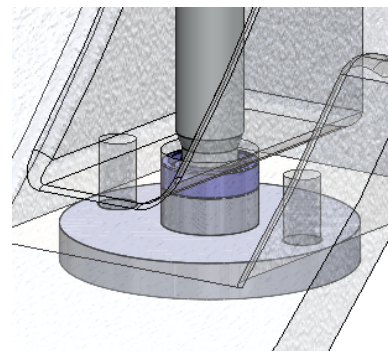
Figure 8.9: Closer look at the band's fixed end attachment to the lead screw shaft

As previously mentioned, ball bearings are positioned on the shaft's ends to support the lead screw shaft. Figure provides 8.10 a closer look at each bearing's mounting approach. The bearing that supports the top end of the shaft is a SKF 6000 deep groove ball bearing and it sits on a shaft shoulder. A retaining mounted on the housing locks this bearing, thus stopping the shaft from moving in the upwards direction.

Regarding the bearing mounted on the bottom, it is a SKF 618/6 deep groove ball bearing that sits on a small circular lid, which is fixed to the housing through 2 screws. This bearing's other side is held against a shaft shoulder, and therefore impedes the shaft from moving in the downwards direction. The bearings' detailed information is provided in Appendix B.



(a) Top rolling bearing - SK 6000 deep groove ball bearing.



(b) Bottom rolling bearing - SKF 618/6 deep groove ball bearing.

Figure 8.10: Lead screw shaft's bearings mounting approaches.

The cylindrical surface of the motor that sticks out from the fixing surface, as can be seen in the motor's representation in figure 8.5, is used to center the motor, which is positioned on a machined surface of the housing. The motor is secured to the housing with four screws, as shown bellow in figure 8.11.

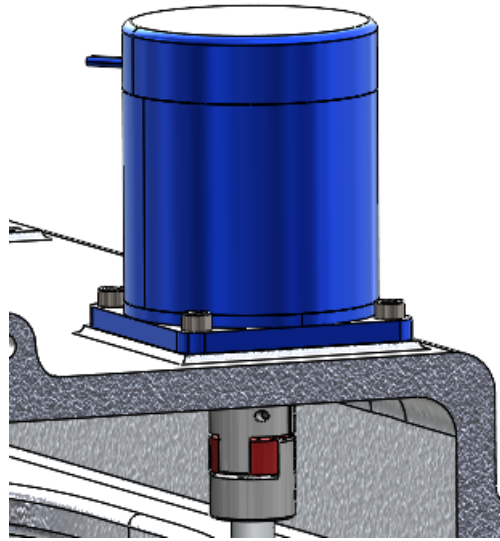


Figure 8.11: Electric motor's fixing to the housing.

The coupling between the electric motor's shaft and the lead screw shaft was done through a jaw coupling provided by Elesa + Granter. The selected jaw coupling was the GN 2240-20-B6-8-AL-RS (available in Appendix D), whose representation is shown below in figure 8.12. The jaws are fixed to the shafts using screws that tighten the jaws, permitting the transfer of power, rather of the torque being transferred through keys. Keys were not required because the jaw coupling's rated torque (3 Nm) was substantially more than the lead screw's torque (0.452 Nm).

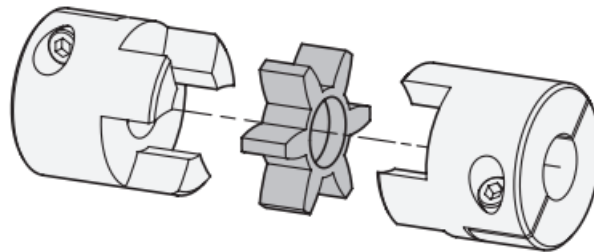


Figure 8.12: Representation of the jaw coupling used to couple to electric motor's shaft to the lead screw shaft.

Full Transmission Model & Assembly

9.1 Exploded View and List of Parts

This chapter will outline the assembly steps for the suggested two-speed transmission and display the fully assembled model. Because of the transmission's complexity, not all assembly stages will be fully detailed. Instead, the most important sub-assemblies will be examined, and a general overview of the assembly order will be provided.

A detailed step-by-step assembly is provided in the video available in [76]. For the transmission's 2D technical drawing, refer to Appendix E.

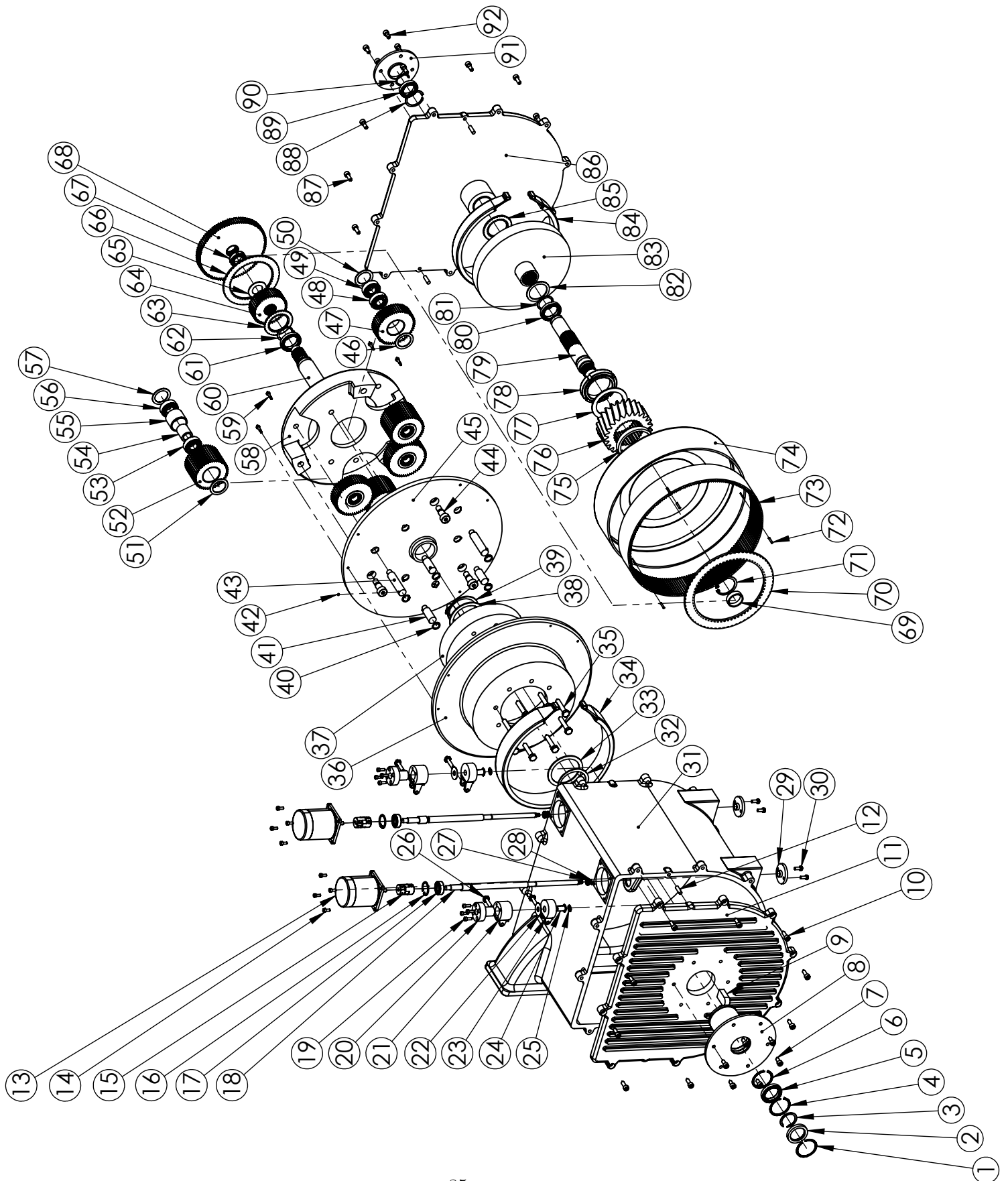
The sub-assembly of the planet carrier and its planets, as well as that of the power output provider—the ring gear case, which houses both the ring gear and the final drive gear—will be covered in the section that follows. Assembling the OWC onto the transmission has already been described in (7.3)

Table 9.1 provides the list of parts, which are numbered in the exploded view of the transmission shown in page 83.

Table 9.1: List of parts.

Parts	
1- Internal Retaining Ring VHM-42	47- 3 x Inner Planet Gear
2- Seal SKF 30x42x6	48- 3x Roller Bearing SKF NJ 202 ECP
3- External Retaining Ring VHM-42	49- 3x Roller Bearing SKF NJ 202 ECP
4- Internal Retaining Ring VHM-42	50- 3x Inner Planet Washer
5- 3x Roller Bearing SKF 61806	51- 3x Outer Planet Washer
6- Internal Retaining Ring VHM-42	52- 3x Outer Planet Gear
7- 6x Hex Socket Head Cap Screw ISO 4762-M6x12-12.9	53- 3x Roller Bearing SKF NJ 202 ECP
8- Front Housing Lid	54- 3x Outer Planet Small Spacer
9- Key DIN 6885 C 18x11x35	55- 3x Outer Planet Small Spacer
10- 10x Hex Socket Head Cap Screw ISO 4762-M6x16-12.9	56- 3x Roller Bearing SKF NJ 202 ECP
11- Front Housing Cover	57- 3x Outer Planet Washer
12- 4x Cylindrical Pin ISO 2338-6m6x20-St	58- Carrier Body
13- 8x Hex Socket Head Cap Screws ISO 4762-M4x12.12.9	59- 6x Precision Construction Screws 7534 4x10
14- 2x Electric Motor	60- Small Sun Shaft

15- 2x Jaw Coupling GN 2240-20-B6-8-AL-RS	61- Roller Bearing SKF 61806
16- 2x Internal Circlip DIN 471-26x1.2-St	62- External Circlip DIN 471-30x1.5-St
17- 2x Roller Bearing SKF 6000	63- Thrust Bearing SKF AXK 4060
18- Lead Screw Shaft	64- Small Sun Gear
19- 8x Hex Socket Head Cap Screw ISO 4762-M4x12-12.9	65- Washer ZTM 2238-015
20- 2x Trapezoidal Nut Type FCS	66- Thrust Bearing SKF AXK 90120
21- 2x Holder for Moving Locking Pin	67- Lock Nut SKF KMFE 4
22- 2x Lead Screw Shaft Washer	68- Large Sun Gear
23- 2x Holder for Fixed Locking Pin	69- Large Sun Spacer
24- 2x Bushing GFM-1416-03	70- Thrust Bearing SKF AXK 150190
25- 2x External Circlip DIN 471-9x1-St	71- Internal Circlip DIN-42x1-8-St
26- 2x Locking Pin with Folding Latch	72- 3x Cylindrical Pin ISO 2338-3m6x16-St
27- 2x Roller Bearing SKF W 618/6	73- Ring Gear
28- 2x External Retaining Ring DIN 471-6x0.7-St	74- Ring Gear Case
29- 2x Lead Screw Shaft Lid	75- Final Drive Spacer
30- 4x Hex Screw ISO 4017-M5x12-12.9	76- Final Drive Gear
31- Housing	77- Final Drive Washer
32- External Retaining Ring DIN 471-60x0.7	78- Lock Nut SKF KMFE 11
33- One-Way Clutch Washer	79- Small Sun Shaft
34- Carrier Brake Band	80- Roller Bearing SKF 61806
35- 8x Hex Bolt ISO 4014-M8x40-12.9	81- External Ring DIN 471-30x1.5-St
36- Carrier Brake Drum	82- Large Sun Drum Washer
37- One-Way Clutch Tsubaki BR48HT-R76B	83- Large Sun Brake Drum
38- One-Way Clutch Washer	84- Large Sun Brake Band
39- External Retaining Ring DIN 471-60x2-St	85- Thrust Bearing SKF AXK 4060
40- 6x Internal Retaining Ring DIN 471-15x1-St	86- Rear Housing Cover
41- 3x Inner Planet Pin	87- 10x Socket Head Cap Screw ISO 4762-M6x16-12.9
42- 6x Locking Sphere	88- Internal Retaining Ring FH-037
43- 3x Outer Planet Pin	89- RB SKF 61805
44- 3x Precision Construction Screw 7534-12x16	90- External Retaining Ring FS-025
45- Carrier Plate	91- Rear Housing Lid
46- 3x Inner Planet Washer	92- 4x Hex Socket Head Cap Screw ISO 4762 M6x12-12.9



9.2 Sub-Assemblies

9.2.1 Planet Carrier Sub-Assembly

i) Planet Carrier

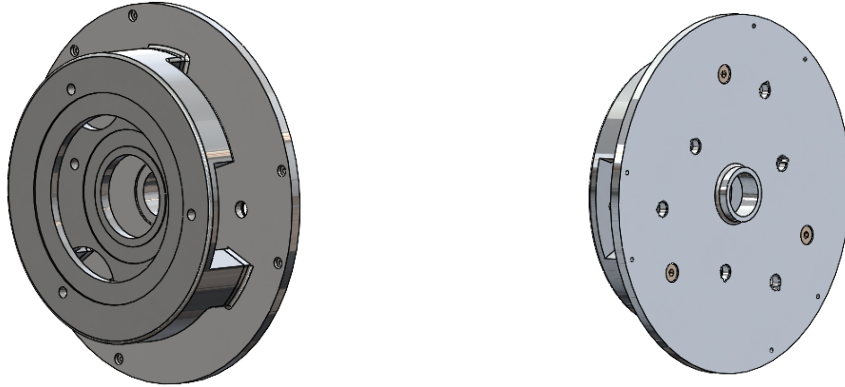
The planet carrier is by nature a complex part of a Ravigneaux planetary transmission. Its large dimensions are inherent, as a result of the double pinion arrangement and space for a long pinion. Some designs have made use of these large dimensions by integrating the carrier into an assembly with additional transmission parts, such as a friction clutch and a one-way clutch. An example of such design is illustrated in figure 75, which shows a Ravigneaux planet carrier, a clutch hub and a one-way clutch as an integrated assembly.



Figure 9.1: Example of a Ravigneaux planet carrier [75].

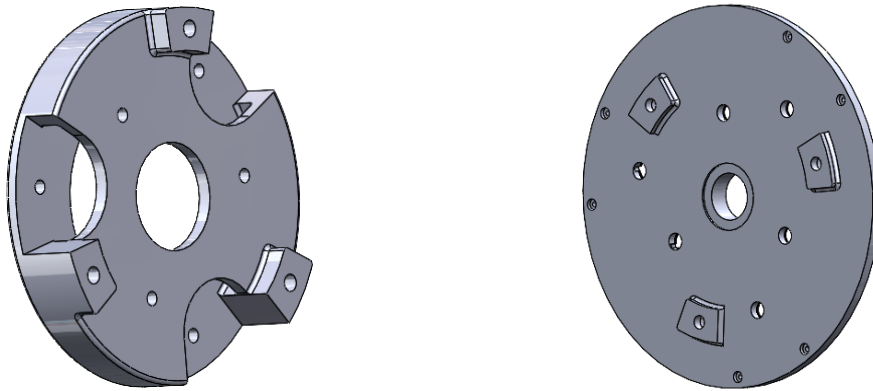
However, the carrier considered for this transmission took a different approach because it was made to be fixed to a drum that would perform the braking. Its assembled 3D model is displayed bellow in figure 9.2.

Because of its complexity, the planet carrier is often manufactured in two separate parts in order to allow the machining of contact surfaces. The design of the carrier started by considering it as a single component, but further development into the transmission's design led to the decision of diving it into two parts. The two parts that make up the planet carrier are shown in figure 9.3.



(a) Back view of the assembled planet carrier.

(b) Front view of the assembled planet carrier.

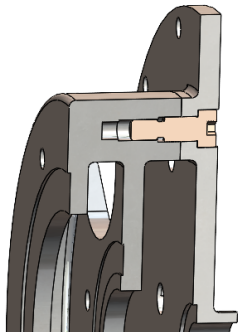
Figure 9.2: 3D model of the assembled planet carrier.

(a) Carrier body.

(b) Carrier plate.

Figure 9.3: 3D model of the carrier's separate parts.

The carrier body (9.3a) and carrier plate (9.3b) are connected by three precision construction screws, which ensure both the alignment and fixation of the two components. The screws are inserted through the carrier plate and fastened to the carrier body as illustrated in figure 9.4a. These screws are provided by NORELEM, whose catalog is in the Appendix D.



(a) Section view of the carrier parts' connection.



(b) NORELEM precision construction screw (similar to ISO 7379) used to join the carrier's parts.

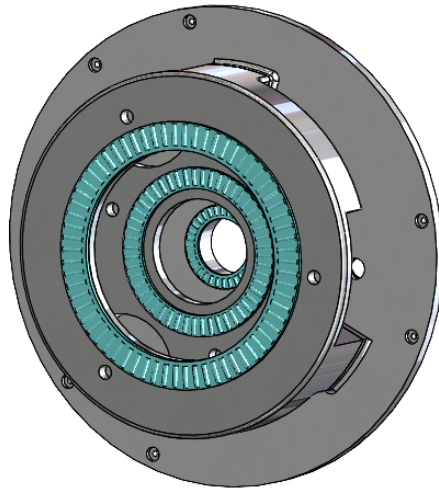
Figure 9.4: Fixation and alignment approach between the two carrier parts.

The carrier body and carrier plate can both be cast in terms of production, but ad-

ditional machining operations must be performed. The carrier will support needle roller thrust bearings, as shown in figure 9.5a, and these bearings require an appropriate surface finishing on the faces that will serve as raceways.

In order to ensure the parallelism and clearance between the carrier and the planet gears' surfaces, it will also be necessary to machine the carrier's surfaces that will be adjacent to the planets. The section that follows will provide more information on the planet gears' assemblies.

The thrust bearings used, which are represented in figure 9.5b, are provided by SKF and are listed in table 9.2 (refer to appendix B for detailed information).



(a) 3D model.



(b) SKF needle roller thrust bearing.

Figure 9.5: Thrust bearings placement in the carrier.

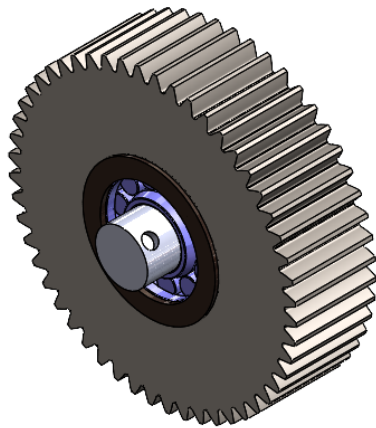
Table 9.2: Thrust bearings incorporated into the carrier assembly.

Component in contact	Roller needle thrust bearing
Small sun gear	SKF AXK 4060
Large sun gear	SKF AXK 90120
Ring gear case	SKF AXK 150190

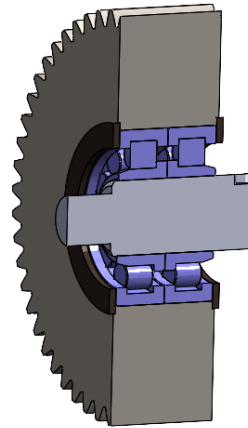
ii) Inner Planet

Figure 9.6 illustrates the inner planet assembly in both a 3D and sectional view. The washers (shown in black) and the shoulders on the inner races of the rolling bearings—which were the subject of the discussion in 6.2—are used to axially fix the rolling bearings inside the planet gears. The faces of the gears are machined to fit the washers, which are in contact with the carrier's surface and ensure the gear's axial positioning.

The pins are machined to have a shoulder that makes contact with the surface of the carrier and serves to axially locate the pin. The sphere (part 42) that is inserted into the small hole visible in figure 9.6b, stops the rotation of the pin relative to the carrier. Then, as will be illustrated further below, an internal retaining ring is used to axially fix the pin and the sphere to the carrier.



(a) 3D view of the inner planet assembly.

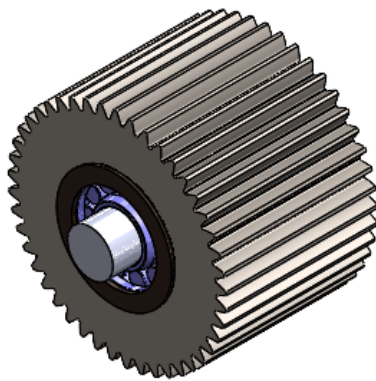


(b) Section view of the inner planet assembly.

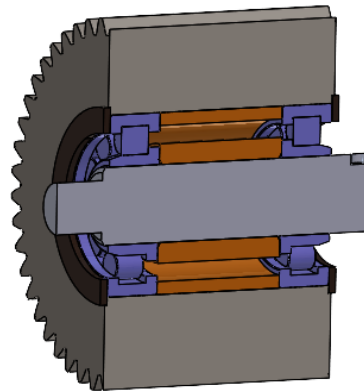
Figure 9.6: 3D model of the inner planet assembly.

iii) Outer Planet

Figure now 9.7 illustrates the outer planet assembly in both a 3D and sectional view. Since the outer planet is longer than the inner planet, in addition to the washers, two spacers (represented in brown) had to be introduced to axially position the the bearings. As in the case of the inner planet's pin, the outer planet's pin has a shoulder to locate the pin on the carrier, and a hole to insert a sphere that prevents the pin to rotate in relation to the carrier.



(a) 3D view of the outer planet assembly.



(b) Section view of the outer planet assembly.

Figure 9.7: 3D model of the outer planet assembly.

iv) Carrier & Planet Gears Assembly

Having described the carrier and the planet gears, the assembly of all these components can be understood. The carrier and planets are displayed in an axonometric view in figure 9.8a, and the position of the planets is shown in the carrier's frontal view (figure 9.8b). Although the next chapter will detail the full assembly instructions, a basic explanation of the assembly of the planet gears to the carrier is provided here. First, the spacers, washers, and bearings for the planets are put together and correctly aligned in the carrier. Following the insertion of the small spheres and planet pins, the internal retaining rings are finally positioned to lock the pins in the carrier.

The location of the sphere and internal retaining ring are shown in detail in figure 9.9, where the carrier plate has been rendered transparent.

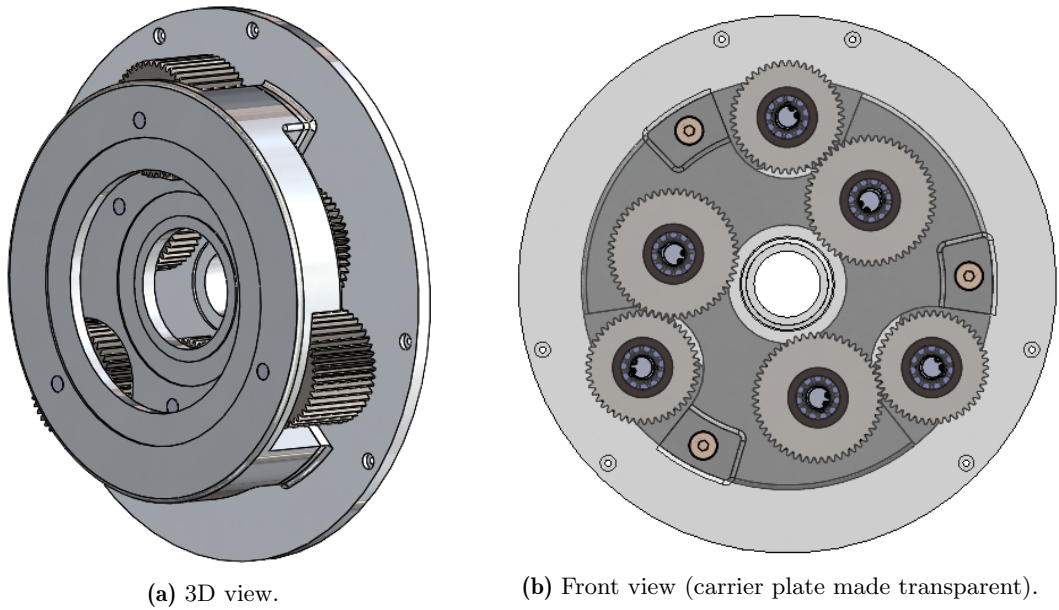


Figure 9.8: 3D model of the carrier and planet gears assembly.

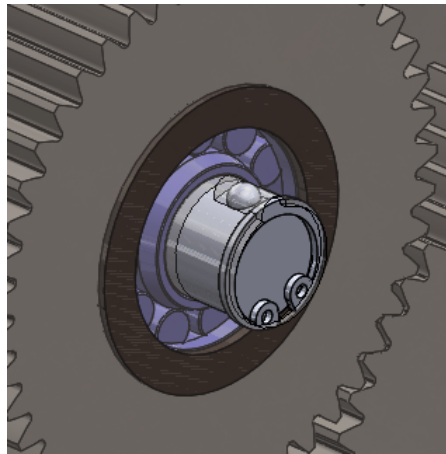


Figure 9.9: Positioning of the locking sphere and internal retaining elastic ring (carrier made transparent).

9.2.2 Ring Gear Case Sub-Assembly

Another assembly that merits consideration is the ring gear case and its parts because it is the component that transmits the output power. Therefore, a quick description of how the parts were designed and put together is provided here. Firstly, three press-fitted cylindrical pins are used to permanently lock the internally toothed ring gear to the ring case. This assembly step is shown in an exploded format in figure 9.10.

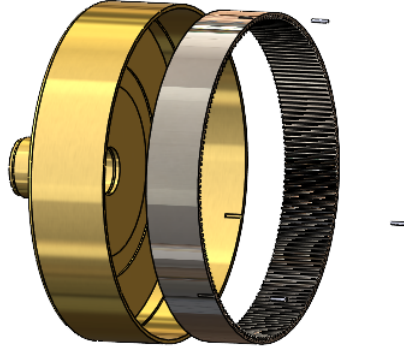


Figure 9.10: Exploded view of the ring gear assembly on to the ring case.

The final drive gear is then assembled, being axially positioned by a spacer and a lock nut-washer pair. Despite not being shown, the case's end is threaded, to which the lock nut is fastened, ensuring that the final drive gear is fixed to the case. Power is transmitted through splines. Figure 9.11 illustrates this, which is the last step of the ring gear case assembly. The assembled ring gear case is shown in figure 9.12.

The lock nut, which is shown in figure 9.13, is provided by SKF and uses a screw that makes contact with the threaded shaft, to ensure the lock nut does not unscrew. For detailed information about this lock nut, refer to Appendix D.

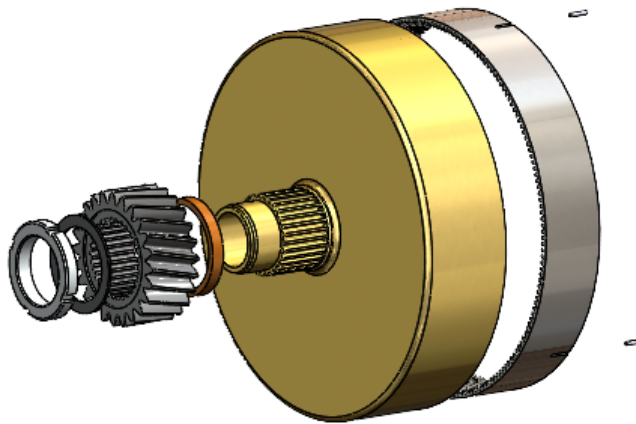


Figure 9.11: Exploded view of the ring gear case's parts.



(a) Back view of the assembled ring gear case. (b) Front view of the assembled ring gear case.

Figure 9.12: 3D model of the assembled ring gear case.



(a) 3D view of the SKF KMFE 11 lock nut. (b) Section view of the SKF KMFE 11 lock nut.

Figure 9.13: SKF KMFE 11 lock nut.

9.3 Assembly Procedures and Final 3D Model

After discussing the transmission's primary sub-assemblies, this section will focus on the transmission's entire assembly process.

Firstly, the carrier's band brake system is assembled. The bottom roller bearing (27) is placed on the housing, being held by the bottom lid (29). The lead screw shaft (18) is then inserted together with the upper roller bearing (17). As it enters the housing, the pin holders (21 and 23) are assembled with their respective components to the shaft, in which is placed an external retaining ring (25) to retain the holder for the fixed pin (23). After mounting the pin holders, the retaining ring (28) for the bottom bearing is placed on the shaft and the shaft can then be inserted to its final position. The top bearing is then fixed by an internal ring (26) and the shaft becomes completely fixed. Afterwards, the shaft is coupled with the electric motor (14), which is fixed to the housing by 4 screws (13). Following that, the carrier's brake band is assembled (34), with the locking pins (26), into its most open position in order to allow the carrier brake drum (36) to be later inserted inside the band. Figure 9.14 shows the carrier's band brake system fully assembled.

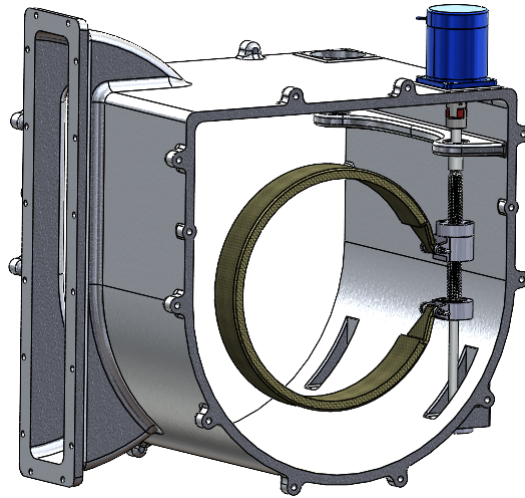
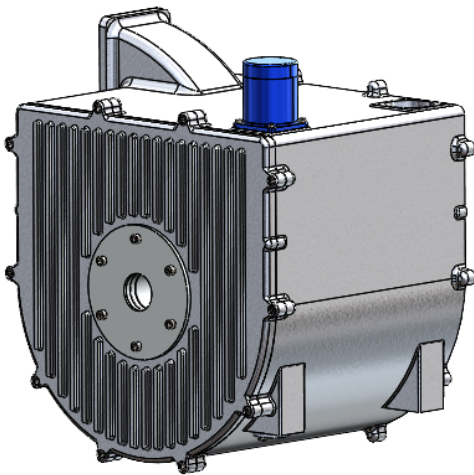


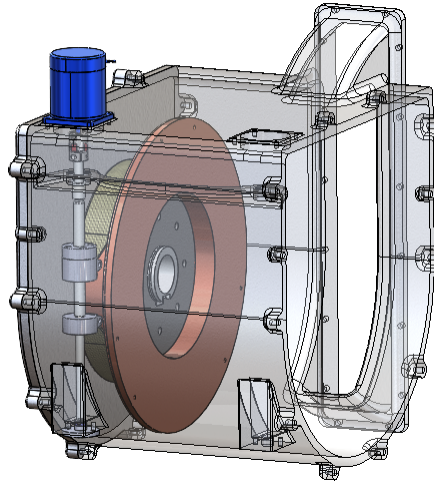
Figure 9.14: Carrier's band brake system assembled.

Next, the OWC is assembled on the housing, as was described in 7.3. The housing's front cover (11) and lid (8) are fixed to the main housing (31). Following that, the OWC is fixed to the lid's sleeve by washers and retaining rings. The carrier brake drum, which is screwed to the OWC's outer race, is inserted through the brake band. Once the OWC sub assembly is placed, the brake band is tightened. Figure 9.15 shows the the transmission after this step of the assembly.

The carrier sub-assembly, which has already been previously detailed, is then inserted and fixed to the brake drum through 6 precision construction screws (59), as shown bellow in figure 9.16. The carrier is placed together with the thrust bearings (63, 66 and 70).



(a) Front housing cover assembled.



(b) OWC and drum assembled.

Figure 9.15: Assembly of the OWC and brake drum on the transmission.

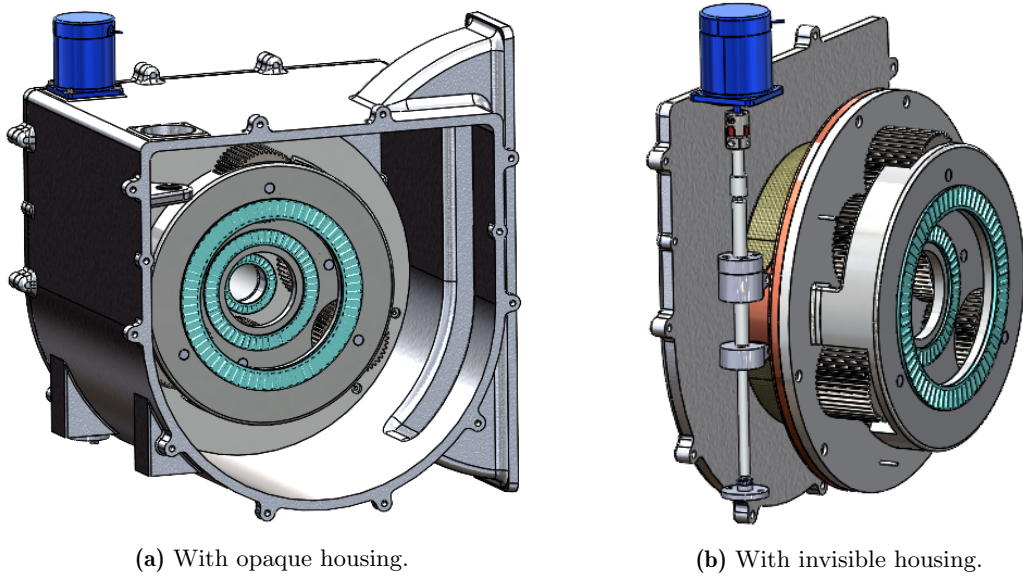


Figure 9.16: Assembly of the carrier and planets on the transmission.

Following that, the small sun gear (64) and the small sun shaft (60) are assembled. The shaft is inserted first with bearing (61) and ring (62), and the small sun gear is then placed on the shaft's spline. The sun gear is fixed by a washer (65) and a lock nut (67), as can be seen on the section view of the transmission at this stage of the assembly, in figure 9.17

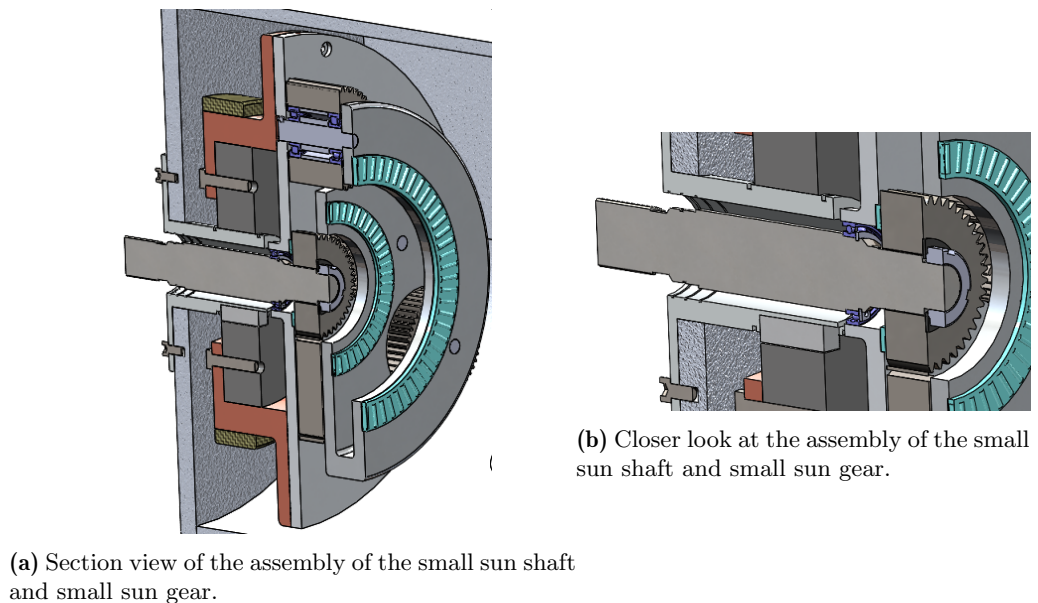


Figure 9.17: Assembly of the small sun shaft and small sun gear.

Next, the large sun gear (68) is assembled together with its spacer (69). The ring gear case sub-assembly, which has also been previously described, is mounted right after the large sun gear, completing the planetary gears' assembly. Before placing the ring gear case, an internal retaining ring (71) is placed inside its sleeve, as can be seen in the section view of the transmission shown below in figure 9.18.

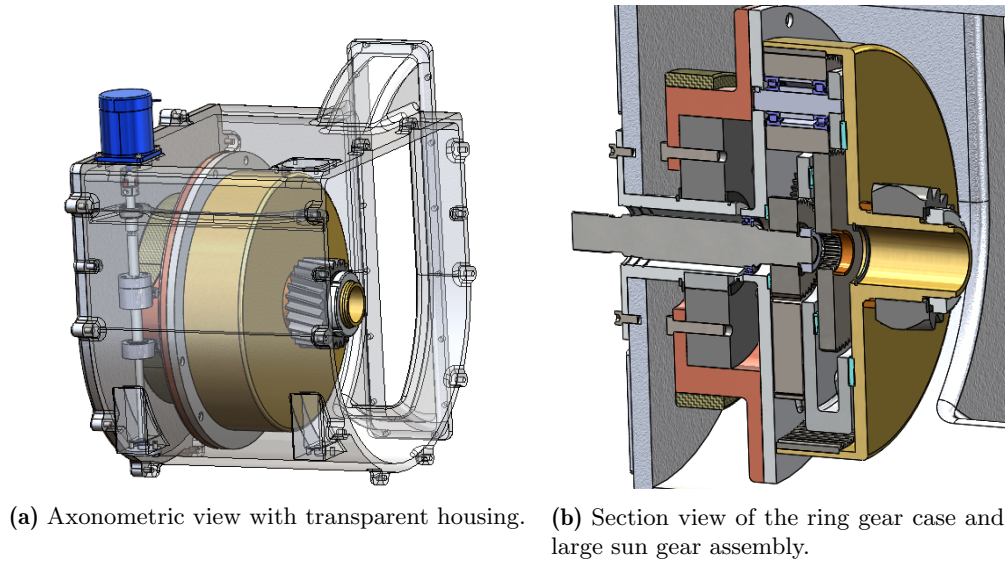


Figure 9.18: Assembly large sun gear and ring case sub-assembly.

After placing the ring gear case, the large sun shaft (79) is then inserted together with a roller bearing (80) and an external retaining ring (81). After placing the shaft, the large sun brake drum (83) can also be placed on its spline, making sure that a washer (82) is placed between the drum and the ring gear case. Figure 9.19 now shows the transmission after having assembled these components.

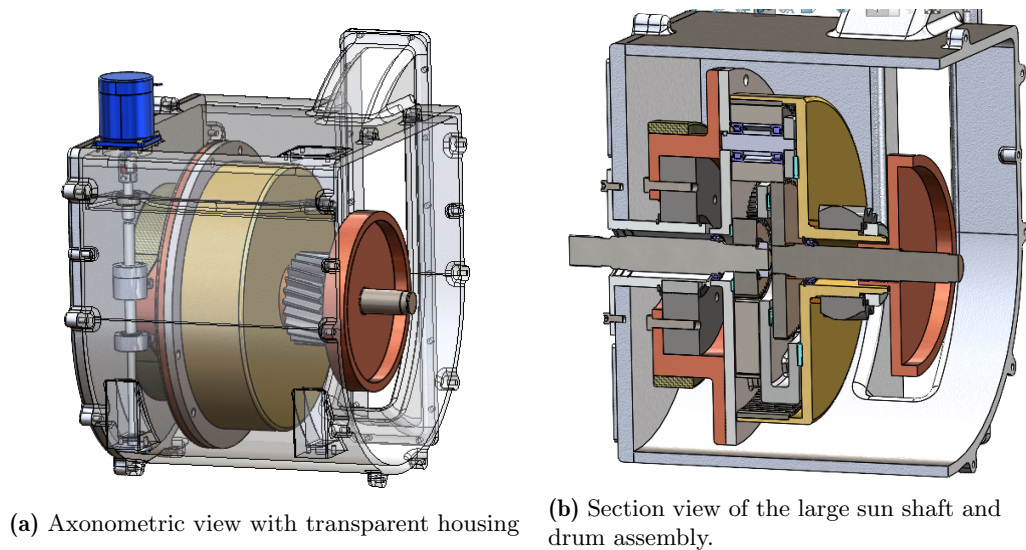


Figure 9.19: Assembly of the large sun shaft and large sun drum

After the large sun drum is mounted, the brake band (84) and its activation system can be assembled. The assembly procedures are the same as the ones detailed for the carrier's band brake. Figure 9.20 shows the transmission after having assembled the large sun band brake.

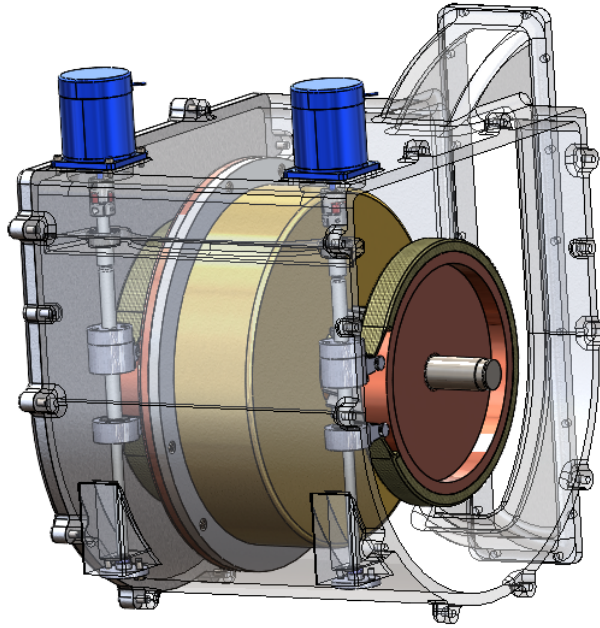
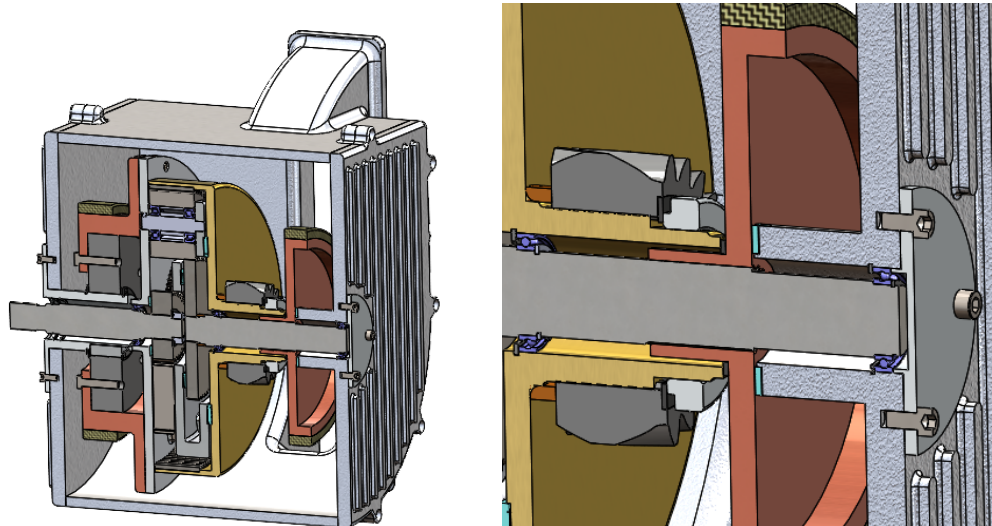


Figure 9.20: Assembled band brake system for the large sun shaft.

The rear housing cover (86) can now be fixed to the housing together with an internal ring (88). The cover is aligned through 2 cylindrical pins (12) and fixed by 10 screws (87). This cover is inserted together with a thrust bearing (85) that will axially fix the large sun brake drum. The large sun shaft's last roller bearing (89) is then inserted on the shaft, followed by an external retaining ring (90). Finally, the bearing is fixed by the rear housing cover's lid (91), which is screwed to the rear housing cover through 4 screws (92). These assembled components can be viewed in figure 9.21.



(a) Section view of all the transmission's components. (b) Closer look at the assembled large sun shaft's second rolling bearing.

Figure 9.21: Assembly of the large sun shaft's second bearing and the rear housing cover.

Lastly, the small sun shaft's second roller bearing is assembled and the input entrance is sealed. An internal retaining ring (6) is first placed on the front housing lid's sleeve, followed by the roller bearing (5) and by two other elastic rings, an internal (4) and an external one (3). Then, the seal (2) is placed on the shaft and fixed by another internal ring (1). Figure 9.22 shows a closer look at the positioning of these mentioned components. This completes the transmission's assembly which can be fully visualized in figure 9.23.

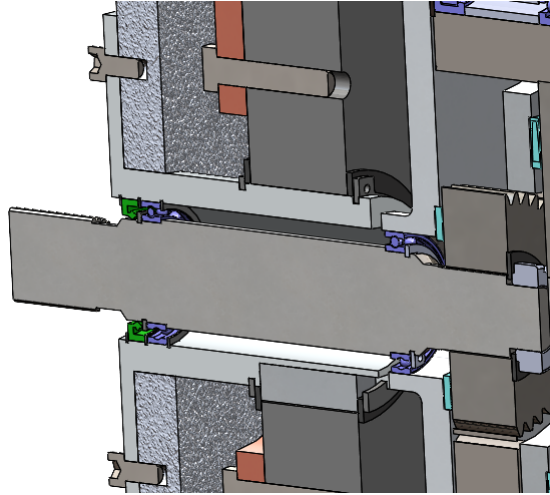


Figure 9.22: Assembly of the input shaft's seal and last roller bearing.

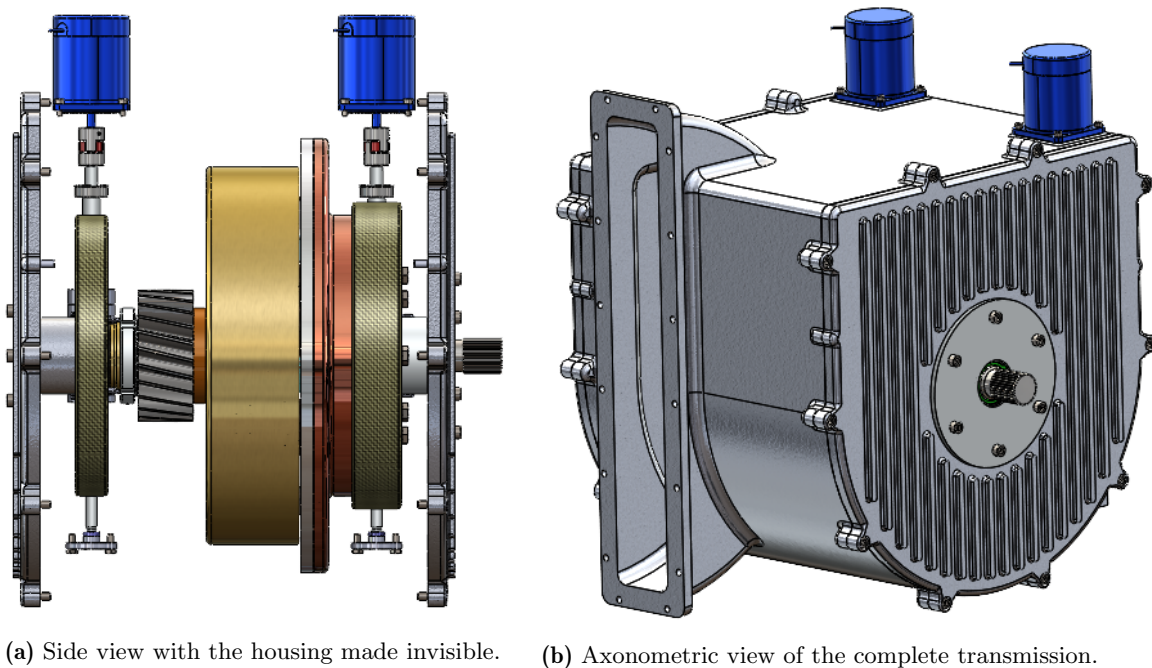


Figure 9.23: 3D model of the complete transmission.

Conclusion

10.1 Conclusions

The development of a two-speed transmission for an electric car was the main topic of this thesis. This transmission consisted in a planetary architecture combined with two band brake systems that provided the stopping of the planetary members. The transmission was meant to be a part of a single motor powertrain that was taken from an existing EV on the market. A list of the findings that could be reached during this work is listed next.

The background investigation carried out in chapter 2 was crucial to comprehending the current paradigm of the electric vehicle business. Despite the significant increase in EV sales, there are still several problems caused by their widespread use that need to be resolved. Increasing the market share of EVs may, from one angle, result in a reduction in global carbon emissions, but it also raises a number of significant economic and sustainability concerns. The advantages of electric vehicles will be defeated if these problems are not resolved. One of the ways to tackle these issues is to optimize the power flow in EVs, which can be achieved through the implementation of multispeed transmissions.

From the performance analysis described in chapter 3, it was possible to conclude that the use of two gears, resulted in better acceleration performance and allowed for a higher vehicle top speed, when compared to a single gear. The total lifetime of the transmission was also estimated, allowing for the conclusion that the second gear will likely be used far more frequently than the first.

The topic of gear sizing was covered in Chapter 5, which started with an understanding of the mating conditions required to provide a functional planetary transmission. The gears may not be able to be assembled due to mismatched teeth if the assembly requirement is not met. The two desired gear ratios specified in chapter 3 were taken into consideration when determining the number of teeth on the gears. The Ravigneaux transmission's geometric complexity prevented simultaneous satisfaction of both gear ratios, leading to a compromise for the second gear ratio. The module was determined using KISSsoft's strength calculations, which mandated a 1.5 mm module. This strength analysis also made it possible to see that the power loss for each gear mesh was significantly reduced as a result of the increase in module.

The bearing sizing detailed in chapter 6 showed interesting conclusions. Due to the transmission's circular symmetry in which the planets are placed around the sun, the radial loads of the planets offset each other. Therefore, the bearings for the co-axial elements only had to be designed to maintain proper alignment of the gearing and withstand loads

imposed by external conditions. The planet bearings, however, required careful selection because of the unsymmetrical meshing and high rotational speeds of the planet gears. This analysis concluded that the first gear was more demanding on the inner planet bearings and the second gear was more demanding on the outer planet. Furthermore, a comparison was drawn between needle bearings and cylindrical roller bearings, which allowed to conclude that the cylindrical roller bearings were the appropriate choice, since the needle bearings did not ensure a sufficient rating life.

Chapter 7's discussion of how to select the best one-way clutch revealed that there aren't many one-way clutches on the market today that can guarantee both high braking torques and high freewheeling speeds. In fact, the one-way clutch used did not ensure a maximum freewheeling speed that satisfied the maximum rotating speed of the electric motor. However, due to the lack of alternatives, this one-way clutch was chosen and was incorporated into the design.

The band brakes used in the transmission's braking mechanism were described in detail in Chapter 8. Due to its quicker response, electric actuation was chosen over hydraulic actuation. After conducting an analysis to determine the best friction material, it was determined that a kevlar-based material suitable for both dry and wet applications offered the required braking torques. The selection of the electric motor, which led to a 10 W electric motor, was also covered in this section. This electric motor drives a lead screw that moves a trapezoidal nut to engage and disengage the brake band.

The whole transmission is shown in an exploded form in chapter 9 along with a list of all of its components. Here, a broad description of the assembly process is also provided, paying special attention to relevant sub-assemblies such as the planet carrier and the ring gear case. The complexity of the planet carrier prevented adequate surface finishing if it were to be cast as a single piece, so it had to be designed in two parts. When designing the transmission in 3D, the assembly processes were also taken into account, which led to the conclusion that designing the transmission to be sequentially assembled from the same side was a good design approach.

10.2 Future Work

The design of the planetary transmission proposed in this thesis covered a wide range of topics, which require a deeper analysis. Therefore, the following tasks are suggested as future work:

- Perform a thermal analysis on the planet's bearings during maximum speed working conditions;
- Perform a finite element analysis of the planet carrier considering both gears working under critical conditions;
- Design a differential to be implemented into this gearbox design;
- Develop a gear shifting control strategy;
- Study the power loss of the transmission;
- Study and develop a lubrication system.

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Appendix - Gear sizing KISSsoft report

KISSsoft calculations regarding all gear meshes, shafts and bearings of the transmission, considering second gear driving and maximum torque and base speed output conditions. These calculations were done for three different modules, but only the calculations done for the selected module are provided. The final drive sizing calculation, along with its spline is also provided.

- Planetary gears for $m = 1.5$ mm;
- Final drive gear;
- Final drive gear splines.

A.1 Planetary gears for $m = 1.5$ mm

2ndGear_m=1.5

Power inputs:

_O.Input			
Speed	[1/min]	4700	
Torque	[Nm]	310	
Power	[kW]	152.577	

Power outputs:

_O.Output			
Speed	[1/min]	2311.475	
Torque	[Nm]	-630.333	
Power	[kW]	152.577	

KISSsoft	
Inhalt	
_O.GroupBox.CarrierShaft.CarrierShaft_calc_____	3
_O.GroupBox.CarrierShaft.Planet1.Planet1_calc_____	11
_O.GroupBox.CarrierShaft.Planet2.Planet2_calc_____	21
_O.GroupBox.MainAxis_____	31
_O.GroupBox.PlanetPlanetConstraint1.Planet1Planet2_calc_____	51
_O.GroupBox.RingPlanetConstraint1.PlanetRing1_calc_____	62
_O.GroupBox.RingShaft.RingShaft_calc_____	73
_O.GroupBox.Sun1Shaft.Sun1Shaft_calc_____	80
_O.GroupBox.Sun2Shaft.Sun2Shaft_calc_____	88
_O.GroupBox.SunPlanetConstraint1.SunPlanet1_calc_____	96
_O.GroupBox.SunPlanetConstraint2.Sun2Planet2_calc_____	107

_O.GroupBox.CarrierShaft.CarrierShaft_calc**1 Input data**

Coordinate system shaft: see picture W-002

1.1 Shafts**1.1.1 Shaft 'CarrierShaft'**

Drawing

Initial position (mm) 0.000

Length (mm) 106.000

Speed (1/min) 1532.609

Direction of rotation: clockwise

Material C45 (1)

Young's modulus (N/mm²) 206000.000Poisson's ratio ν 0.300Density (kg/m³) 7830.000Coefficient of thermal expansion (10⁻⁶/K) 11.500

Temperature (°C) 20.000

Weight of shaft (kg) 3.445

Weight of shaft, including additional masses (kg) 3.445

Mass moment of inertia (kg*m²) 0.013Momentum of mass GD2 (Nm²) 0.505**1.2 Weight force**

The direction of the weight is not considered

1.3 Shaft modeling**1.4 Shear deformations**

Consider deformations due to shearing

Shear correction factor 1.100

1.5 Rolling bearings

Contact angle of rolling bearings is considered

Tolerance field: Mean value

1.6 Housing

Reference temperature (°C) 20.000

1.7 Load applications

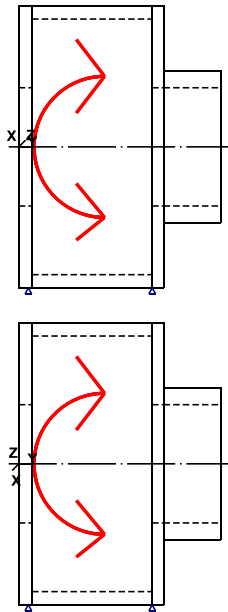


Figure: Load applications

2 Shaft definition (CarrierShaft)

2.1 Outer contour

2.1.1 Cylinder (Cylinder), 0.000 mm ... 76.000 mm

Diameter (mm)	[d]	148.0000
Length (mm)	[l]	76.0000
Surface roughness (µm)	[Rz]	8.0000

2.1.2 Cylinder (Cylinder), 76.000 mm ... 106.000 mm

Diameter (mm)	[d]	80.0000
Length (mm)	[l]	30.0000
Surface roughness (µm)	[Rz]	8.0000

2.2 Inner contour

2.2.1 Cylindrical bore (Cylindrical bore), 0.000 mm ... 7.000 mm

Diameter (mm)	[d]	62.0000
Length (mm)	[l]	7.0000
Surface roughness (µm)	[Rz]	8.0000

2.2.2 Cylindrical bore (Cylindrical bore), 7.000 mm ... 70.000 mm

Diameter (mm)	[d]	134.0000
Length (mm)	[l]	63.0000
Surface roughness (µm)	[Rz]	8.0000

2.2.3 Cylindrical bore (Cylindrical bore), 70.000 mm ... 106.000 mm

Diameter (mm)	[d]	62.0000
Length (mm)	[l]	36.0000
Surface roughness (μm)	[Rz]	8.0000

2.3 Forces**2.3.1 Coupling (PlanetCarrierCoupling(PlanetCarrierConstraint1))**

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	45.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		20.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

2.3.2 Coupling (PlanetCarrierCoupling(PlanetCarrierConstraint2))

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	45.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		20.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

2.3.3 Coupling (PlanetCarrierCoupling(PlanetPlanetConstraint1))

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	45.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000

Direction of the radial force (°)	0.0000	
Axial force factor (-)	0.0000	
Length of load application (mm)	20.0000	
Power (kW)	107.2459	driving (output)
Torque (Nm)	-668.2222	
Axial force (N)	0.0000	
Shearing force X (N)	0.0000	
Shearing force Z (N)	0.0000	
Bending moment X (Nm)	0.0000	
Bending moment Z (Nm)	0.0000	
Mass (kg)	0.0000	
Mass moment of inertia Jp (kg*m²)	0.0000	
Mass moment of inertia Jxx (kg*m²)	0.0000	
Mass moment of inertia Jzz (kg*m²)	0.0000	
Eccentricity (mm)	0.0000	

2.3.4 Coupling (PlanetCarrierCoupling(RingPlanetConstraint1))

Position on shaft (mm)	[y _{local}]	45.0000	
Position in global system (mm)	[y _{global}]	45.0000	
Effective diameter (mm)		170.0000	
Radial force factor (-)		0.0000	
Direction of the radial force (°)		0.0000	
Axial force factor (-)		0.0000	
Length of load application (mm)		20.0000	
Power (kW)		76.2883	driving (output)
Torque (Nm)		-475.3333	
Axial force (N)		0.0000	
Shearing force X (N)		0.0000	
Shearing force Z (N)		0.0000	
Bending moment X (Nm)		0.0000	
Bending moment Z (Nm)		0.0000	
Mass (kg)		0.0000	
Mass moment of inertia Jp (kg*m²)		0.0000	
Mass moment of inertia Jxx (kg*m²)		0.0000	
Mass moment of inertia Jzz (kg*m²)		0.0000	
Eccentricity (mm)		0.0000	

2.3.5 Coupling (PlanetCarrierCoupling(SunPlanetConstraint1))

Position on shaft (mm)	[y _{local}]	45.0000	
Position in global system (mm)	[y _{global}]	45.0000	
Effective diameter (mm)		170.0000	
Radial force factor (-)		0.0000	
Direction of the radial force (°)		0.0000	
Axial force factor (-)		0.0000	
Length of load application (mm)		20.0000	
Power (kW)		76.2883	driven (input)
Torque (Nm)		475.3333	
Axial force (N)		0.0000	
Shearing force X (N)		0.0000	
Shearing force Z (N)		0.0000	
Bending moment X (Nm)		0.0000	
Bending moment Z (Nm)		0.0000	
Mass (kg)		0.0000	
Mass moment of inertia Jp (kg*m²)		0.0000	
Mass moment of inertia Jxx (kg*m²)		0.0000	
Mass moment of inertia Jzz (kg*m²)		0.0000	
Eccentricity (mm)		0.0000	

2.3.6 Coupling (PlanetCarrierCoupling(SunPlanetConstraint2))

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	45.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		20.0000
Power (kW)		107.2459 driven (input)
Torque (Nm)		668.2222
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

2.4 Bearing**2.4.1 Support8 (Own input)**

Bearing position (mm)	[y _{local}]	5.000
Bearing position (mm)	[y _{global}]	5.000
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: freeRy: freeRz: free		

2.4.2 Support9 (Own input)

Bearing position (mm)	[y _{local}]	70.000
Bearing position (mm)	[y _{global}]	70.000
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: freeRy: freeRz: free		

3 Gears

4 Results

4.1 Shafts

Maximum deflection (μm)	0.000
Position of the maximum (mm)	0.000
Mass center of gravity (mm)	44.461
Total axial load (N)	0.000
Torsion of the shaft under torque ($^{\circ}$)	-0.000
(Difference between left and right shaft end)	

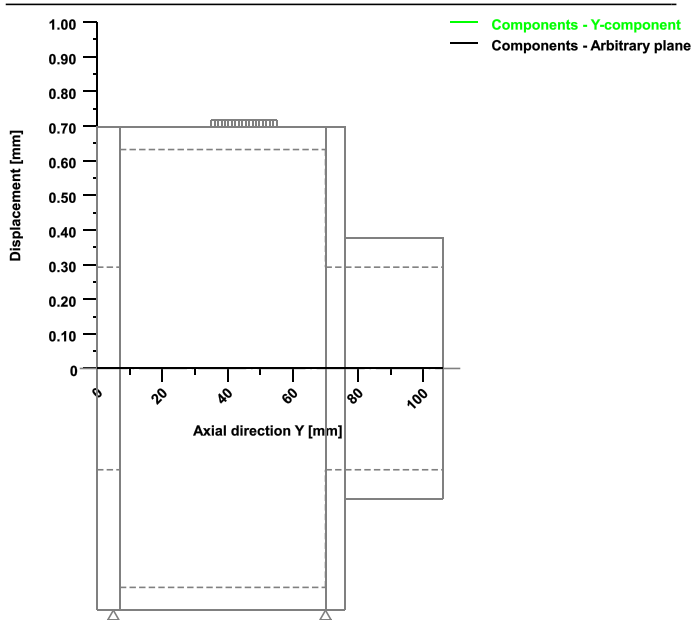
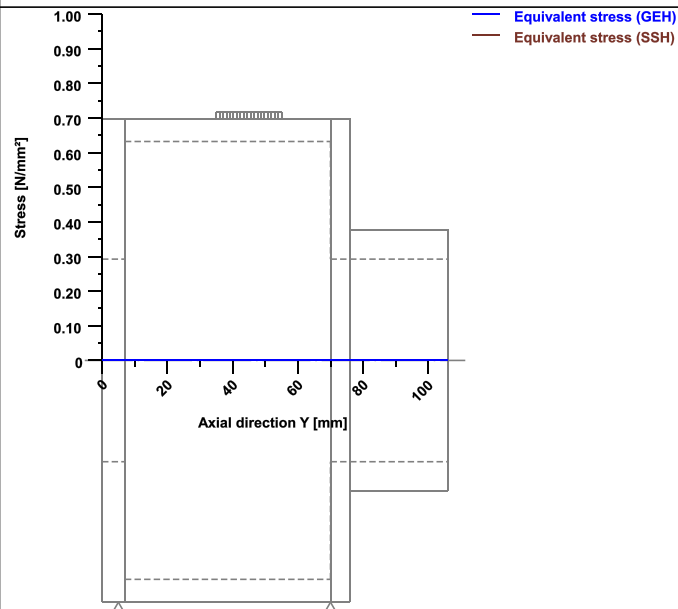


Figure: Deformation (bending etc.) (Arbitrary plane 0 127)

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Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\text{sigV} = ((\text{sigB} + \text{sigZ}, D)^2 + 3 * (\text{tauT} + \text{tauS})^2)^{1/2}$

SSH(Tresca): $\text{sigV} = ((\text{sigB} - \text{sigZ}, D)^2 + 4 * (\text{tauT} + \text{tauS})^2)^{1/2}$

Figure: Equivalent stress

4.2 Bearing

4.3 'CarrierShaft' Bearing 'Support8'

Position (Y-coordinate)	[y]	5.00	mm		
Bearing reaction force	[F _x]			-0.000	kN
Bearing reaction force	[F _y]			0.000	kN
Bearing reaction force	[F _z]			-0.000	kN
Bearing reaction force	[F _t]			0.000	kN
Displacement of bearing	[u _x]			0.000	µm
Displacement of bearing	[u _y]			0.000	µm
Displacement of bearing	[u _z]			0.000	µm
Displacement of bearing	[u _t]			0.000	µm
Misalignment of bearing	[r _x]			0.000	mrاد
Misalignment of bearing	[r _y]			-0.000	mrاد
Misalignment of bearing	[r _z]			0.000	mrاد
Misalignment of bearing	[r _t]			0.000	mrاد

4.4 'CarrierShaft' Bearing 'Support9'

Position (Y-coordinate)	[y]	70.00	mm		
Bearing reaction force	[F _x]			-0.000	kN
Bearing reaction force	[F _y]			0.000	kN

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Bearing reaction force	[F _z]	0.000	kN
Bearing reaction force	[F _t]	0.000	kN
Displacement of bearing	[u _x]	0.000	µm
Displacement of bearing	[u _y]	-0.000	µm
Displacement of bearing	[u _z]	0.000	µm
Displacement of bearing	[u _i]	0.000	µm
Misalignment of bearing	[r _x]	-0.000	mrاد
Misalignment of bearing	[r _y]	0.000	mrاد
Misalignment of bearing	[r _z]	-0.000	mrاد
Misalignment of bearing	[r _i]	0.000	mrاد
Note: Utilization = (Lreq/Lh)^(1/k)			
Ball bearing: k = 3, roller bearing: k = 10/3			

_O.GroupBox.CarrierShaft.Planet1.Planet1_calc**1 Input data**

Coordinate system shaft: see picture W-002

1.1 Shafts**1.1.1 Shaft 'PlanetGearShaft'**

Drawing	
Initial position (mm)	20.000
Length (mm)	50.000
Speed (1/min)	4700.000
Direction of rotation:	clockwise
Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	0.115
Weight of shaft, including additional masses (kg)	0.115
Mass moment of inertia (kg*mm ²)	40.718
Momentum of mass GD2 (Nm ²)	0.002

1.1.2 Shaft 'PlanetPinShaft'

Drawing	
Initial position (mm)	10.000
Length (mm)	71.000
Speed (1/min)	1532.609
Direction of rotation:	clockwise
Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (g)	98.241
Weight of shaft, including additional masses (g)	98.241
Mass moment of inertia (kg*mm ²)	2.763
Momentum of mass GD2 (Nm ²)	0.000

1.2 Weight force

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Weight towards	0.000		
Weight towards		0.000	
Weight towards			-1.000

1.3 Shaft modeling

1.4 Shear deformations

Consider deformations due to shearing	
Shear correction factor	1.100

1.5 Rolling bearings

Rolling bearing stiffness is calculated from inner bearing geometry	
Tolerance field:	Mean value

1.6 Housing

Reference temperature (°C)	20.000
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1.7 Load applications

Figure: Load applications

2 Shaft definition (PlanetGearShaft)

2.1 Outer contour

2.1.1 Cylinder (Cylinder),	0.000 mm ... 50.000 mm	
Diameter (mm)	[d]	40.0000

Length (mm)	[l]	50.0000
Surface roughness (μm)	[Rz]	8.0000

2.2 Inner contour

2.2.1 Cylindrical bore (Cylindrical bore), 0.000 mm ... 50.000 mm

Diameter (mm)	[d]	35.0000
Length (mm)	[l]	50.0000
Surface roughness (μm)	[Rz]	8.0000

2.3 Forces

2.3.1 Cylindrical gear (PlanetGear1(44.8))

Position on shaft (mm)	[y _{local}]	25.0000
Position in global system (mm)	[y _{global}]	45.0000
Operating pitch diameter (mm)		68.9943
Spur gear		
Working pressure angle at normal section (°)		23.1701
Position of contact (°)		44.8202
Length of load application (mm)		50.0000
Power (kW)		50.8589 driven (input)
Torque (Nm)		103.3333
Axial force (N)		-0.0000
Shearing force X (N)		1202.0823
Shearing force Z (N)		-3028.3687
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		-0.0000

2.3.2 Cylindrical gear (PlanetGear1(-180.0))

Position on shaft (mm)	[y _{local}]	25.0000
Position in global system (mm)	[y _{global}]	45.0000
Operating pitch diameter (mm)		68.8032
Spur gear		
Working pressure angle at normal section (°)		22.7954
Position of contact (°)		-180.0000
Length of load application (mm)		50.0000
Power (kW)		25.4294 driving (output)
Torque (Nm)		-51.6667
Axial force (N)		0.0000
Shearing force X (N)		631.1862
Shearing force Z (N)		-1501.8684
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		-0.0000

2.3.3 Cylindrical gear (PlanetGear1(0.0))

Position on shaft (mm)	[y _{local}]	25.0000
Position in global system (mm)	[y _{global}]	45.0000
Operating pitch diameter (mm)		68.7767
Spur gear		
Working pressure angle at normal section (°)		22.7428
Position of contact (°)		0.0000
Length of load application (mm)		50.0000
Power (kW)		25.4294 driving (output)

Torque (Nm)	-51.6667
Axial force (N)	0.0000
Shearing force X (N)	-629.8062
Shearing force Z (N)	1502.4476
Bending moment X (Nm)	-0.0000
Bending moment Z (Nm)	0.0000

3 Shaft definition (PlanetPinShaft)

3.1 Outer contour

3.1.1 Cylinder (Cylinder), 0.000 mm ... 71.000 mm

Diameter (mm)	[d]	15.0000
Length (mm)	[l]	71.0000
Surface roughness (µm)	[Rz]	8.0000

3.2 Forces

3.2.1 Coupling (PlanetCarrierCoupling1(PlanetCarrierConstraint1))

Position on shaft (mm)	[y _{local}]	34.0000
Position in global system (mm)	[y _{global}]	44.0000
Effective diameter (mm)		35.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		15.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

3.3 Bearing

3.3.1 Support1 (Own input)

Bearing position (mm)	[y _{local}]	3.000
Bearing position (mm)	[y _{global}]	13.000
Degrees of freedom		
X: fixedY: fixedZ: fixed		
Rx: fixedRy: freeRz: fixed		

3.3.2 Support2 (Own input)

Bearing position (mm)	[y _{lokal}]	68.000
Bearing position (mm)	[y _{global}]	78.000
Degrees of freedom		
X: fixedY: fixedZ: fixed		
Rx: fixedRy: freeRz: fixed		

4 Connections**4.1 ConnectionRollerBearing1 (SKF NJ 202 ECP)**

Position (mm)		27.000 mm
Shaft 'PlanetPinShaft' <-> Shaft 'PlanetGearShaft'		
Free bearing		
d = 15.000(mm), D = 35.000 (mm), b = 11.000 (mm), r = 0.600 (mm)		
C = 12.500(kN), C0 = 10.200 (kN), Cu = 1.200 (kN)		
Ctheo = 12.627 (kN), C0theo = 10.200 (kN)		
fC = 1.000 (kN), fC0 = 1.000 (kN)		
Calculation with approximate bearings internal geometry (*)		
Z = 11, Dpw = 24.800 (mm), Dw = 5.500 (mm)		
Lwe = 4.924 (mm)		
Diameter, external race (mm)	[d _o]	30.300
Diameter, internal race (mm)	[d _i]	19.300
Bearing clearance		0.00 µm

4.2 ConnectionRollerBearing2 (SKF NJ 202 ECP)

Position (mm)		63.000 mm
Shaft 'PlanetPinShaft' <-> Shaft 'PlanetGearShaft'		
Free bearing		
d = 15.000(mm), D = 35.000 (mm), b = 11.000 (mm), r = 0.600 (mm)		
C = 12.500(kN), C0 = 10.200 (kN), Cu = 1.200 (kN)		
Ctheo = 12.627 (kN), C0theo = 10.200 (kN)		
fC = 1.000 (kN), fC0 = 1.000 (kN)		
Calculation with approximate bearings internal geometry (*)		
Z = 11, Dpw = 24.800 (mm), Dw = 5.500 (mm)		
Lwe = 4.924 (mm)		
Diameter, external race (mm)	[d _o]	30.300
Diameter, internal race (mm)	[d _i]	19.300
Bearing clearance		0.00 µm

5 Gears

6 Results

6.1 Shafts

Maximum deflection	14.156(μm) (PlanetGearShaft pos = 47.273 mm)
Mass center of gravity	
PlanetGearShaft (mm)	25.000
PlanetPinShaft (mm)	35.500
Total axial load	
PlanetGearShaft (N)	0.000
PlanetPinShaft (N)	0.000
Torsion of the shafts under torque	
(Difference between left and right shaft end)	
PlanetGearShaft (°)	-0.000
PlanetPinShaft (°)	-0.000

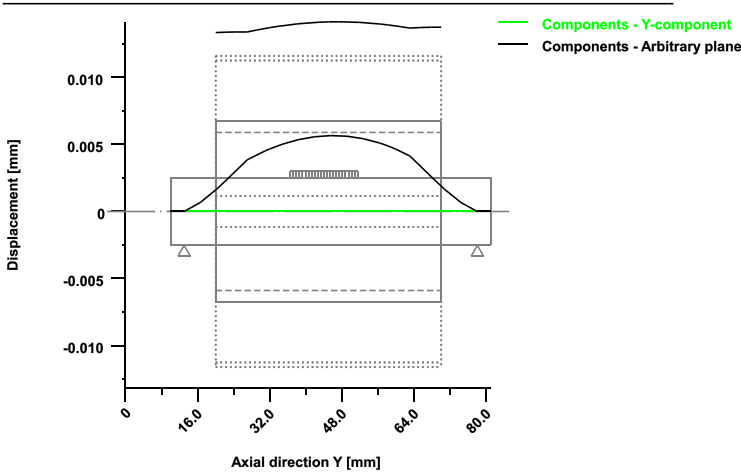
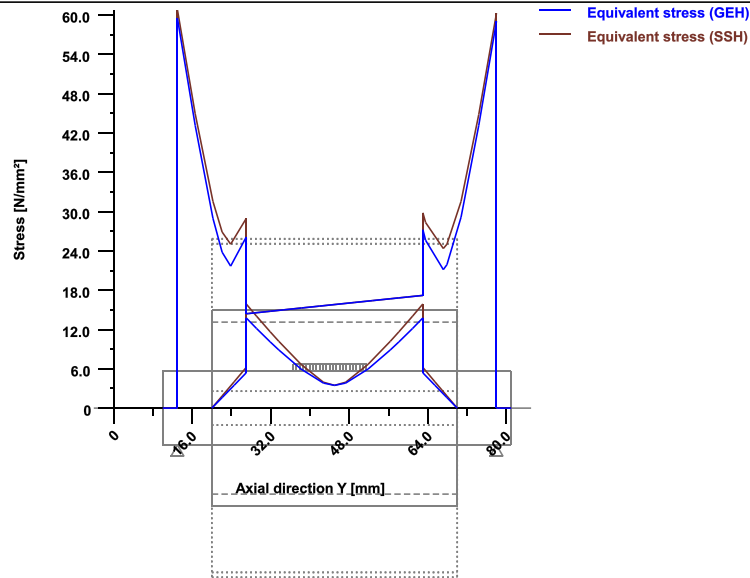


Figure: Deformation (bending etc.) (Arbitrary plane 291.8272251 127)

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Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\text{sigV} = ((\text{sigB} + \text{sigZ}, D)^2 + 3 * (\text{tauT} + \text{tauS})^2)^{1/2}$

SSH(Tresca): $\text{sigV} = ((\text{sigB} - \text{sigZ}, D)^2 + 4 * (\text{tauT} + \text{tauS})^2)^{1/2}$

Figure: Equivalent stress

6.2 Bearing

Probability of failure	[n]	10.00	%		
Axial clearance (ISO 281)	[u _A]			10.00	μm
Lubricant	ISO-VG 220				
Lubricant - service temperature	[T _S]			70.00	°C
Rolling bearing stiffness calculated from internal geometry					

6.3 'PlanetPinShaft' Bearing 'Support1'

Position (Y-coordinate)	[y]	3.00	mm		
Bearing reaction force	[F _x]			-0.611	kN
Bearing reaction force	[F _y]			0.000	kN
Bearing reaction force	[F _z]			1.539	kN
Bearing reaction force	[F _i]			1.656	kN
Bearing reaction moment	[M _x]			17.05	Nm
Bearing reaction moment	[M _y]			0.00	Nm
Bearing reaction moment	[M _z]			6.77	Nm
Bearing reaction moment	[M _i]			18.35	Nm
Displacement of bearing	[u _x]			0.000	μm
Displacement of bearing	[u _y]			0.000	μm
Displacement of bearing	[u _z]			0.000	μm
Displacement of bearing	[u _i]			0.000	μm
Misalignment of bearing	[r _x]			0.000	mrad
Misalignment of bearing	[r _y]			-0.000	mrad

Misalignment of bearing	[r _z]	0.000	mrاد
Misalignment of bearing	[r _i]	0.000	mrاد

6.4 'PlanetPinShaft' Bearing 'Support2'

Position (Y-coordinate)	[y]	68.00	mm		
Bearing reaction force	[F _x]			-0.592	kN
Bearing reaction force	[F _y]			-0.000	kN
Bearing reaction force	[F _z]			1.491	kN
Bearing reaction force	[F _i]			1.604	kN
Bearing reaction moment	[M _x]			-17.00	Nm
Bearing reaction moment	[M _y]			0.00	Nm
Bearing reaction moment	[M _z]			-6.75	Nm
Bearing reaction moment	[M _i]			18.29	Nm
Displacement of bearing	[u _x]			0.000	μm
Displacement of bearing	[u _y]			0.000	μm
Displacement of bearing	[u _z]			0.000	μm
Displacement of bearing	[u _i]			0.000	μm
Misalignment of bearing	[r _x]			0.000	mrاد
Misalignment of bearing	[r _y]			-0.000	mrاد
Misalignment of bearing	[r _z]			0.000	mrاد
Misalignment of bearing	[r _i]			0.000	mrاد

6.5 Rolling bearing 'ConnectionRollerBearing1'

Position (Y-coordinate)	[y]	27.00	mm		
Dynamic equivalent load	[P]	1.63	kN		
Static equivalent load	[P ₀]			1.63	kN
Minimum EHL lubricant film thickness	[h _{min}]			0.132	μm
Life modification factor for reliability	[a ₁]			1.000	

6.5.1 Results according to ISO 281

Lubricant	ISO-VG 220				
Load ratio	[C/P]			7.670	
Operating viscosity	[v]			48.884	mm ² /s
Reference viscosity	[v ₁]			0.000	mm ² /s
Viscosity ratio	[κ]			0.000	
Basic bearing rating life	[L _{nh}]			4682.04	h
Static safety factor	[S ₀]			6.26	

6.5.2 Calculation with approximate bearings internal geometry (ISO/TS 16281)

Operating bearing clearance	[Pd]			0.000	μm
Reference rating life	[L _{nh}]			18034.98	h
Bearing reaction force	[F _x]			0.602	kN
Bearing reaction force	[F _y]			0.000	kN
Bearing reaction force	[F _z]			-1.515	kN
Bearing reaction force	[F _i]			1.630	kN
Bearing reaction moment	[M _x]			0.06	Nm
Bearing reaction moment	[M _y]			-0.00	Nm
Bearing reaction moment	[M _z]			0.02	Nm
Bearing reaction moment	[M _i]			0.07	Nm
Displacement of bearing	[u _x]			-3.559	μm
Displacement of bearing	[u _y]			-0.000	μm

Displacement of bearing	$[u_z]$	8.851	μm
Displacement of bearing	$[u_i]$	9.540	μm
Misalignment of bearing	$[r_x]$	-0.156	mrad
Misalignment of bearing	$[r_y]$	0.000	mrad
Misalignment of bearing	$[r_z]$	-0.062	mrad
Misalignment of bearing	$[r_i]$	0.167	mrad
Oil level	$[H]$	0.000	mm
Rolling moment of friction	$[M_r]$	0.029	Nm
Sliding moment of friction	$[M_{sl}]$	0.001	Nm
Moment of friction, seals	$[M_{seal}]$	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	$[M_{drag}]$	0.000	Nm
Torque of friction	$[M_{loss}]$	0.030	Nm
Power loss	$[P_{loss}]$	10.028	W

The moment of friction is calculated according to the details in SKF Catalog 2018.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

6.6 Rolling bearing 'ConnectionRollerBearing2'

Position (Y-coordinate)	$[y]$	63.00	mm
Dynamic equivalent load	$[P]$	1.63	kN
Static equivalent load	$[P_0]$	1.63	kN
Minimum EHL lubricant film thickness	$[h_{min}]$	0.132	μm
Life modification factor for reliability	$[a_1]$	1.000	

6.6.1 Results according to ISO 281

Lubricant	ISO-VG 220		
Load ratio	$[C/P]$	7.671	
Operating viscosity	$[v]$	48.884	mm^2/s
Reference viscosity	$[v_1]$	0.000	mm^2/s
Viscosity ratio	$[k]$	0.000	
Basic bearing rating life	$[L_{nh}]$	4685.10	h
Static safety factor	$[S_0]$	6.26	

6.6.2 Calculation with approximate bearings internal geometry (ISO/TS 16281)

Operating bearing clearance	$[Pd]$	0.000	μm
Reference rating life	$[L_{nh}]$	18038.23	h
Bearing reaction force	$[F_x]$	0.602	kN
Bearing reaction force	$[F_y]$	0.000	kN
Bearing reaction force	$[F_z]$	-1.514	kN
Bearing reaction force	$[F_i]$	1.629	kN
Bearing reaction moment	$[M_x]$	-0.07	Nm
Bearing reaction moment	$[M_y]$	-0.00	Nm
Bearing reaction moment	$[M_z]$	-0.03	Nm
Bearing reaction moment	$[M_i]$	0.07	Nm
Displacement of bearing	$[u_x]$	-3.559	μm
Displacement of bearing	$[u_y]$	0.000	μm
Displacement of bearing	$[u_z]$	8.850	μm
Displacement of bearing	$[u_i]$	9.539	μm
Misalignment of bearing	$[r_x]$	0.169	mrad
Misalignment of bearing	$[r_y]$	0.000	mrad
Misalignment of bearing	$[r_z]$	0.067	mrad
Misalignment of bearing	$[r_i]$	0.182	mrad
Oil level	$[H]$	0.000	mm

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Rolling moment of friction	[M _r]	0.029	Nm
Sliding moment of friction	[M _{sl}]	0.001	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.030	Nm
Power loss	[P _{loss}]	10.027	W
The moment of friction is calculated according to the details in SKF Catalog 2018.			
The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.			
(*) Note about roller bearings with an approximated bearing geometry: The internal geometry of these bearings has not been input in the database. The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog). For this reason, the geometry may be different from the actual geometry. In some situations, this may result in significant variations in roller bearing stiffness.			
6.6.3 Damage (%)		[Lreq] (0.000)	
Bin no	B1	B2	
1	0.00	0.00	

Σ	0.00	0.00	
6.6.4 Utilization (%)		[Lreq] (0.000)	
B1	B2		
0.00	0.00		
Note: Utilization = (Lreq/Lh)^(1/k)			
Ball bearing: k = 3, roller bearing: k = 10/3			
B1 : ConnectionRollerBearing1 (Connecting rolling bearing)			
B2 : ConnectionRollerBearing2 (Connecting rolling bearing)			
6.6.5 Calculation of the factors required to define reliability R(t) using the Weibull distribution. t in (h)			
Reliability not calculated			

_O.GroupBox.CarrierShaft.Planet2.Planet2_calc**1 Input data**

Coordinate system shaft: see picture W-002

1.1 Shafts**1.1.1 Shaft 'PlanetGearShaft'**

Drawing	
Initial position (mm)	7.000
Length (mm)	24.000
Speed (1/min)	1208.403
Direction of rotation:	counterclockwise
Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	0.118
Weight of shaft, including additional masses (kg)	0.118
Mass moment of inertia (kg*mm ²)	47.967
Momentum of mass GD2 (Nm ²)	0.002

1.1.2 Shaft 'PlanetPinShaft'

Drawing	
Initial position (mm)	0.000
Length (mm)	41.000
Speed (1/min)	1532.609
Direction of rotation:	clockwise
Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (g)	56.731
Weight of shaft, including additional masses (g)	56.731
Mass moment of inertia (kg*mm ²)	1.596
Momentum of mass GD2 (Nm ²)	0.000

1.2 Weight force

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Weight towards	0.000		
Weight towards		0.000	
Weight towards			-1.000

1.3 Shaft modeling

1.4 Shear deformations

Consider deformations due to shearing	
Shear correction factor	1.100

1.5 Rolling bearings

Rolling bearing stiffness is calculated from inner bearing geometry
Tolerance field: Mean value

1.6 Housing

Reference temperature (°C)	20.000
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1.7 Load applications

Figure: Load applications

2 Shaft definition (PlanetGearShaft)

2.1 Outer contour

2.1.1 Cylinder (Cylinder), Diameter (mm)	0.000 mm ... 24.000 mm [d]	45.0000
---	-------------------------------	---------

Length (mm)	[l]	24.0000
Surface roughness (μm)	[Rz]	8.0000

2.2 Inner contour**2.2.1 Cylindrical bore (Cylindrical bore), 0.000 mm ... 24.000 mm**

Diameter (mm)	[d]	35.0000
Length (mm)	[l]	24.0000
Surface roughness (μm)	[Rz]	8.0000

2.3 Forces**2.3.1 Cylindrical gear (PlanetGear2(-135.2))**

Position on shaft (mm)	[y _{local}]	12.0000
Position in global system (mm)	[y _{global}]	19.0000
Operating pitch diameter (mm)		79.7267
Spur gear		
Working pressure angle at normal section (°)		23.1701
Position of contact (°)		-135.1798
Length of load application (mm)		23.0000
Power (kW)		15.1103 driving (output)
Torque (Nm)		119.4074
Axial force (N)		-0.0000
Shearing force X (N)		-1202.0823
Shearing force Z (N)		3028.3687
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		0.0000

2.3.2 Cylindrical gear (PlanetGear2(-44.8))

Position on shaft (mm)	[y _{local}]	12.0000
Position in global system (mm)	[y _{global}]	19.0000
Operating pitch diameter (mm)		79.7321
Spur gear		
Working pressure angle at normal section (°)		23.1792
Position of contact (°)		-44.8163
Length of load application (mm)		23.0000
Power (kW)		15.1103 driven (input)
Torque (Nm)		-119.4074
Axial force (N)		0.0000
Shearing force X (N)		1201.3946
Shearing force Z (N)		3028.6416
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

3 Shaft definition (PlanetPinShaft)**3.1 Outer contour****3.1.1 Cylinder (Cylinder), 0.000 mm ... 41.000 mm**

KISSsoft		
Diameter (mm)	[d]	15.0000
Length (mm)	[l]	41.0000
Surface roughness (µm)	[Rz]	8.0000
3.2 Forces		
3.2.1 Coupling (PlanetCarrierCoupling2(PlanetCarrierConstraint2))		
Position on shaft (mm)	[y _{local}]	20.0000
Position in global system (mm)	[y _{global}]	20.0000
Effective diameter (mm)		25.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		15.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000
3.3 Bearing		
3.3.1 Support1 (Own input)		
Bearing position (mm)	[y _{local}]	1.000
Bearing position (mm)	[y _{global}]	1.000
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: fixedRy: freeRz: fixed		
3.3.2 Support2 (Own input)		
Bearing position (mm)	[y _{local}]	39.000
Bearing position (mm)	[y _{global}]	39.000
Degrees of freedom		
X: fixedY: fixedZ: fixed		
Rx: fixedRy: freeRz: fixed		
4 Connections		
4.1 ConnectionRollerBearing1 (SKF NJ 202 ECP)		

Position (mm) 13.000 mm
 Shaft 'PlanetPinShaft' <-> Shaft 'PlanetGearShaft'
 Free bearing
 $d = 15.000(\text{mm})$, $D = 35.000(\text{mm})$, $b = 11.000(\text{mm})$, $r = 0.600(\text{mm})$
 $C = 12.500(\text{kN})$, $C_0 = 10.200(\text{kN})$, $C_u = 1.200(\text{kN})$
 $C_{\text{theo}} = 12.627(\text{kN})$, $C_{0\text{theo}} = 10.200(\text{kN})$
 $f_C = 1.000(\text{kN})$, $f_{C0} = 1.000(\text{kN})$
 Calculation with approximate bearings internal geometry (*)
 $Z = 11$, $D_{pw} = 24.800(\text{mm})$, $D_w = 5.500(\text{mm})$
 $L_{we} = 4.924(\text{mm})$
 Diameter, external race (mm) [d_o] 30.300
 Diameter, internal race (mm) [d_i] 19.300
 Bearing clearance 0.00 μm

4.2 ConnectionRollerBearing2 (SKF NJ 202 ECP)

Position (mm) 25.000 mm
 Shaft 'PlanetPinShaft' <-> Shaft 'PlanetGearShaft'
 Free bearing
 $d = 15.000(\text{mm})$, $D = 35.000(\text{mm})$, $b = 11.000(\text{mm})$, $r = 0.600(\text{mm})$
 $C = 12.500(\text{kN})$, $C_0 = 10.200(\text{kN})$, $C_u = 1.200(\text{kN})$
 $C_{\text{theo}} = 12.627(\text{kN})$, $C_{0\text{theo}} = 10.200(\text{kN})$
 $f_C = 1.000(\text{kN})$, $f_{C0} = 1.000(\text{kN})$
 Calculation with approximate bearings internal geometry (*)
 $Z = 11$, $D_{pw} = 24.800(\text{mm})$, $D_w = 5.500(\text{mm})$
 $L_{we} = 4.924(\text{mm})$
 Diameter, external race (mm) [d_o] 30.300
 Diameter, internal race (mm) [d_i] 19.300
 Bearing clearance 0.00 μm

5 Gears

6 Results

6.1 Shafts

Maximum deflection	20.715(μm) (PlanetGearShaft pos = 31.000 mm)
Mass center of gravity	
PlanetGearShaft (mm)	12.000
PlanetPinShaft (mm)	20.500
Total axial load	
PlanetGearShaft (N)	0.000
PlanetPinShaft (N)	0.000
Torsion of the shafts under torque	
(Difference between left and right shaft end)	
PlanetGearShaft (°)	-0.000
PlanetPinShaft (°)	-0.000

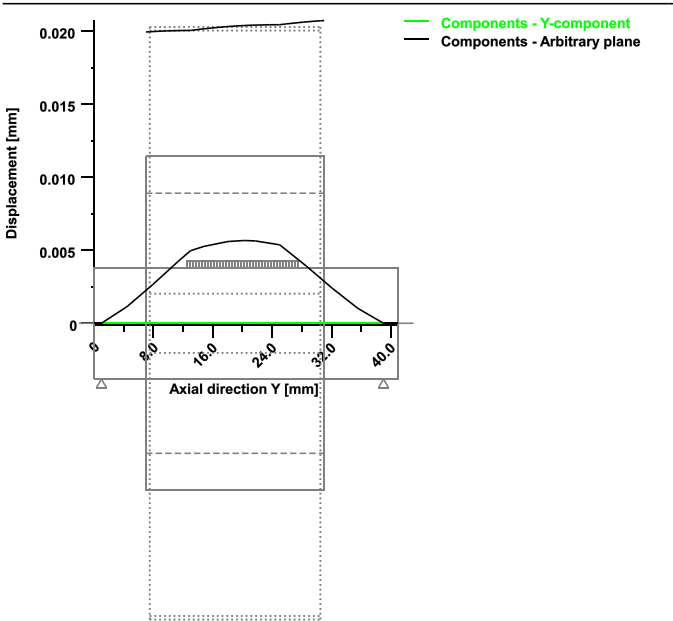
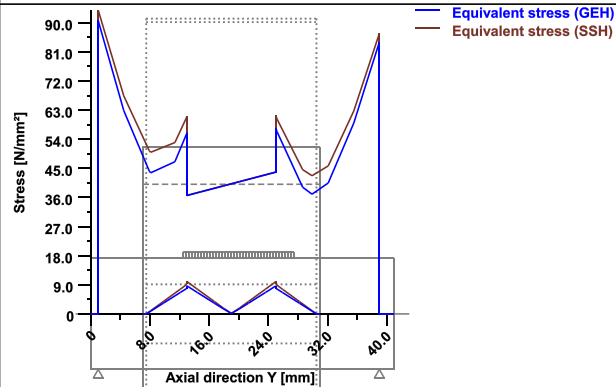


Figure: Deformation (bending etc.) (Arbitrary plane 90.37381429 127)

KISSsoft


Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\text{sigV} = ((\text{sigB} + \text{sigZ}, D)^2 + 3 * (\text{tauT} + \text{tauS})^2)^{1/2}$

SSH(Tresca): $\text{sigV} = ((\text{sigB} - \text{sigZ}, D)^2 + 4 * (\text{tauT} + \text{tauS})^2)^{1/2}$

Figure: Equivalent stress

6.2 Bearing

Probability of failure	[n]	10.00	%	
Axial clearance (ISO 281)	[u _A]		10.00	μm
Lubricant	ISO-VG 220			
Lubricant - service temperature	[T _B]		70.00	°C
Rolling bearing stiffness calculated from internal geometry				

6.3 'PlanetPinShaft' Bearing 'Support1'

Position (Y-coordinate)	[y]	1.00	mm	
Bearing reaction force	[F _x]		0.000	kN
Bearing reaction force	[F _y]		0.000	kN
Bearing reaction force	[F _z]		-3.229	kN
Bearing reaction force	[F _i]		3.229	kN
Bearing reaction moment	[M _x]		-26.58	Nm
Bearing reaction moment	[M _y]		0.00	Nm
Bearing reaction moment	[M _z]		-0.00	Nm
Bearing reaction moment	[M _i]		26.58	Nm
Displacement of bearing	[u _x]		0.000	μm
Displacement of bearing	[u _y]		0.000	μm
Displacement of bearing	[u _z]		0.000	μm
Displacement of bearing	[u _i]		0.000	μm
Misalignment of bearing	[r _x]		0.000	mrad
Misalignment of bearing	[r _y]		-0.000	mrad

Misalignment of bearing	[r _z]	0.000	mrاد
Misalignment of bearing	[r _i]	0.000	mrاد

6.4 'PlanetPinShaft' Bearing 'Support2'

Position (Y-coordinate)	[y]	39.00	mm		
Bearing reaction force	[F _x]			0.000	kN
Bearing reaction force	[F _y]			-0.000	kN
Bearing reaction force	[F _z]			-2.827	kN
Bearing reaction force	[F _i]			2.827	kN
Bearing reaction moment	[M _x]			25.00	Nm
Bearing reaction moment	[M _y]			0.00	Nm
Bearing reaction moment	[M _z]			0.00	Nm
Bearing reaction moment	[M _i]			25.00	Nm
Displacement of bearing	[u _x]			0.000	μm
Displacement of bearing	[u _y]			0.000	μm
Displacement of bearing	[u _z]			0.000	μm
Displacement of bearing	[u _i]			0.000	μm
Misalignment of bearing	[r _x]			0.000	mrاد
Misalignment of bearing	[r _y]			-0.000	mrاد
Misalignment of bearing	[r _z]			0.000	mrاد
Misalignment of bearing	[r _i]			0.000	mrاد

6.5 Rolling bearing 'ConnectionRollerBearing1'

Position (Y-coordinate)	[y]	13.00	mm		
Dynamic equivalent load	[P]	3.03	kN		
Static equivalent load	[P ₀]			3.03	kN
Minimum EHL lubricant film thickness	[h _{min}]			0.110	μm
Life modification factor for reliability	[a ₁]			1.000	

6.5.1 Results according to ISO 281

Lubricant	ISO-VG 220				
Load ratio	[C/P]			4.126	
Operating viscosity	[v]			48.884	mm ² /s
Reference viscosity	[v ₁]			0.000	mm ² /s
Viscosity ratio	[κ]			0.000	
Basic bearing rating life	[L _{nh}]			685.10	h
Static safety factor	[S ₀]			3.37	

6.5.2 Calculation with approximate bearings internal geometry (ISO/TS 16281)

Operating bearing clearance	[Pd]			0.000	μm
Reference rating life	[L _{nh}]			2327.08	h
Bearing reaction force	[F _x]			-0.000	kN
Bearing reaction force	[F _y]			0.000	kN
Bearing reaction force	[F _z]			3.029	kN
Bearing reaction force	[F _i]			3.029	kN
Bearing reaction moment	[M _x]			-0.07	Nm
Bearing reaction moment	[M _y]			0.00	Nm
Bearing reaction moment	[M _z]			-0.00	Nm
Bearing reaction moment	[M _i]			0.07	Nm
Displacement of bearing	[u _x]			0.135	μm
Displacement of bearing	[u _y]			0.000	μm

Displacement of bearing	[u_z]	-15.071	μm
Displacement of bearing	[u_i]	15.071	μm
Misalignment of bearing	[r_x]	0.137	mrad
Misalignment of bearing	[r_y]	0.000	mrad
Misalignment of bearing	[r_z]	0.000	mrad
Misalignment of bearing	[r_i]	0.137	mrad
Oil level	[H]	0.000 mm	
Rolling moment of friction	[M_r]	0.033	Nm
Sliding moment of friction	[M_{sl}]	0.002	Nm
Moment of friction, seals	[M_{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	[M_{drag}]	0.000	Nm
Torque of friction	[M_{loss}]	0.035	Nm
Power loss	[P_{loss}]	10.072	W

The moment of friction is calculated according to the details in SKF Catalog 2018.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

6.6 Rolling bearing 'ConnectionRollerBearing2'

Position (Y-coordinate)	[y]	25.00	mm
Dynamic equivalent load	[P]	3.03	kN
Static equivalent load	[P_0]	3.03	kN
Minimum EHL lubricant film thickness	[h_{min}]	0.110	μm
Life modification factor for reliability	[a_1]	1.000	

6.6.1 Results according to ISO 281

Lubricant	ISO-VG 220		
Load ratio	[C/P]	4.130	
Operating viscosity	[ν]	48.884	mm^2/s
Reference viscosity	[ν_1]	0.000	mm^2/s
Viscosity ratio	[κ]	0.000	
Basic bearing rating life	[L_{nh}]	687.43	h
Static safety factor	[S_0]	3.37	

6.6.2 Calculation with approximate bearings internal geometry (ISO/TS 16281)

Operating bearing clearance	[Pd]	0.000	μm
Reference rating life	[L_{nh}]	2334.09	h
Bearing reaction force	[F_x]	-0.000	kN
Bearing reaction force	[F_y]	0.000	kN
Bearing reaction force	[F_z]	3.026	kN
Bearing reaction force	[F_i]	3.026	kN
Bearing reaction moment	[M_x]	0.08	Nm
Bearing reaction moment	[M_y]	0.00	Nm
Bearing reaction moment	[M_z]	0.00	Nm
Bearing reaction moment	[M_i]	0.08	Nm
Displacement of bearing	[u_x]	0.135	μm
Displacement of bearing	[u_y]	0.000	μm
Displacement of bearing	[u_z]	-15.059	μm
Displacement of bearing	[u_i]	15.060	μm
Misalignment of bearing	[r_x]	-0.175	mrad
Misalignment of bearing	[r_y]	0.000	mrad
Misalignment of bearing	[r_z]	0.000	mrad
Misalignment of bearing	[r_i]	0.175	mrad
Oil level	[H]	0.000 mm	

KISSsoft			
Rolling moment of friction	[M _r]	0.033	Nm
Sliding moment of friction	[M _{sl}]	0.002	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.035	Nm
Power loss	[P _{loss}]	10.068	W
The moment of friction is calculated according to the details in SKF Catalog 2018.			
The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.			
(*) Note about roller bearings with an approximated bearing geometry: The internal geometry of these bearings has not been input in the database. The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog). For this reason, the geometry may be different from the actual geometry. In some situations, this may result in significant variations in roller bearing stiffness.			
6.6.3 Damage (%)		[Lreq] (0.000)	
Bin no	B1	B2	
1	0.00	0.00	

Σ	0.00	0.00	
6.6.4 Utilization (%)		[Lreq] (0.000)	
B1	B2		
0.00	0.00		
Note: Utilization = (Lreq/Lh)^(1/k)			
Ball bearing: k = 3, roller bearing: k = 10/3			
B1 : ConnectionRollerBearing1 (Connecting rolling bearing)			
B2 : ConnectionRollerBearing2 (Connecting rolling bearing)			
6.6.5 Calculation of the factors required to define reliability R(t) using the Weibull distribution. t in (h)			
Reliability not calculated			

_O.GroupBox.MainAxis**1 Input data**

Coordinate system shaft: see picture W-002

1.1 Shafts**1.1.1 Shaft 'CarrierShaft'**

Drawing

Initial position (mm) 26.000

Length (mm) 106.000

Speed (1/min) 1532.609

Direction of rotation: clockwise

Material

C45 (1)

Young's modulus (N/mm²) 206000.000Poisson's ratio ν 0.300Density (kg/m³) 7830.000Coefficient of thermal expansion (10⁻⁶/K) 11.500

Temperature (°C) 20.000

Weight of shaft (kg) 3.445

Weight of shaft, including additional masses (kg) 3.445

Mass moment of inertia (kg*m²) 0.013Momentum of mass GD2 (Nm²) 0.505**1.1.2 Shaft 'RingShaft'**

Drawing

Initial position (mm) 40.000

Length (mm) 50.000

Speed (1/min) 2311.475

Direction of rotation: clockwise

Material

C45 (1)

Young's modulus (N/mm²) 206000.000Poisson's ratio ν 0.300Density (kg/m³) 7830.000Coefficient of thermal expansion (10⁻⁶/K) 11.500

Temperature (°C) 20.000

Weight of shaft (kg) 3.129

Weight of shaft, including additional masses (kg) 3.129

Mass moment of inertia (kg*m²) 0.035Momentum of mass GD2 (Nm²) 1.384**1.1.3 Shaft 'Sun1Shaft'**

Drawing

KISSsoft			
Initial position (mm)		-61.000	
Length (mm)		113.000	
Speed (1/min)		0.000	
Direction of rotation:	clockwise		
Material	C45 (1)		
Young's modulus (N/mm ²)		206000.000	
Poisson's ratio nu		0.300	
Density (kg/m ³)		7830.000	
Coefficient of thermal expansion (10 ⁻⁶ /K)		11.500	
Temperature (°C)		20.000	
Weight of shaft (kg)		0.625	
Weight of shaft, including additional masses (kg)		0.625	
Mass moment of inertia (kg*mm ²)		70.360	
Momentum of mass GD2 (Nm ²)		0.003	
1.1.4 Shaft 'Sun2Shaft'			
Drawing			
Initial position (mm)		70.000	
Length (mm)		87.000	
Speed (1/min)		4700.000	
Direction of rotation:	clockwise		
Material	C45 (1)		
Young's modulus (N/mm ²)		206000.000	
Poisson's ratio nu		0.300	
Density (kg/m ³)		7830.000	
Coefficient of thermal expansion (10 ⁻⁶ /K)		11.500	
Temperature (°C)		20.000	
Weight of shaft (kg)		0.482	
Weight of shaft, including additional masses (kg)		0.482	
Mass moment of inertia (kg*mm ²)		54.171	
Momentum of mass GD2 (Nm ²)		0.002	
1.2 Weight force			
Weight towards		0.000	
Weight towards		0.000	
Weight towards			-1.000
1.3 Shaft modeling			
1.4 Shear deformations			
Consider deformations due to shearing			
Shear correction factor		1.100	
1.5 Rolling bearings			
Rolling bearing stiffness is calculated from inner bearing geometry			
Tolerance field:	Mean value		
1.6 Housing			
Reference temperature (°C)		20.000	
1.7 Load applications			

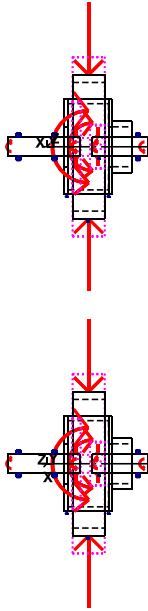


Figure: Load applications

2 Shaft definition (CarrierShaft)

2.1 Outer contour

2.1.1 Cylinder (Cylinder), 0.000 mm ... 76.000 mm

Diameter (mm)	[d]	148.0000
Length (mm)	[l]	76.0000
Surface roughness (μm)	[Rz]	8.0000

2.1.2 Cylinder (Cylinder), 76.000 mm ... 106.000 mm

Diameter (mm)	[d]	80.0000
Length (mm)	[l]	30.0000
Surface roughness (μm)	[Rz]	8.0000

2.2 Inner contour

2.2.1 Cylindrical bore (Cylindrical bore), 0.000 mm ... 7.000 mm

Diameter (mm)	[d]	62.0000
Length (mm)	[l]	7.0000
Surface roughness (μm)	[Rz]	8.0000

2.2.2 Cylindrical bore (Cylindrical bore), 7.000 mm ... 70.000 mm

Diameter (mm)	[d]	134.0000
Length (mm)	[l]	63.0000
Surface roughness (μm)	[Rz]	8.0000

2.2.3 Cylindrical bore (Cylindrical bore), 70.000 mm ... 106.000 mm

Diameter (mm)	[d]	62.0000
Length (mm)	[l]	36.0000
Surface roughness (µm)	[Rz]	8.0000

2.3 Forces
2.3.1 Coupling (PlanetCarrierCoupling(PlanetCarrierConstraint1))

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	71.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		20.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

2.3.2 Coupling (PlanetCarrierCoupling(PlanetCarrierConstraint2))

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	71.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		20.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

2.3.3 Coupling (PlanetCarrierCoupling(PlanetPlanetConstraint1))

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	71.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000

Direction of the radial force (°)	0.0000	
Axial force factor (-)	0.0000	
Length of load application (mm)	20.0000	
Power (kW)	107.2459	driving (output)
Torque (Nm)	-668.2222	
Axial force (N)	0.0000	
Shearing force X (N)	0.0000	
Shearing force Z (N)	0.0000	
Bending moment X (Nm)	0.0000	
Bending moment Z (Nm)	0.0000	
Mass (kg)	0.0000	
Mass moment of inertia J_p (kg*m²)	0.0000	
Mass moment of inertia J_{xx} (kg*m²)	0.0000	
Mass moment of inertia J_{zz} (kg*m²)	0.0000	
Eccentricity (mm)	0.0000	

2.3.4 Coupling (PlanetCarrierCoupling(RingPlanetConstraint1))

Position on shaft (mm)	[y _{local}]	45.0000	
Position in global system (mm)	[y _{global}]	71.0000	
Effective diameter (mm)		170.0000	
Radial force factor (-)		0.0000	
Direction of the radial force (°)		0.0000	
Axial force factor (-)		0.0000	
Length of load application (mm)		20.0000	
Power (kW)		76.2883	driving (output)
Torque (Nm)		-475.3333	
Axial force (N)		0.0000	
Shearing force X (N)		0.0000	
Shearing force Z (N)		0.0000	
Bending moment X (Nm)		0.0000	
Bending moment Z (Nm)		0.0000	
Mass (kg)		0.0000	
Mass moment of inertia J_p (kg*m²)		0.0000	
Mass moment of inertia J_{xx} (kg*m²)		0.0000	
Mass moment of inertia J_{zz} (kg*m²)		0.0000	
Eccentricity (mm)		0.0000	

2.3.5 Coupling (PlanetCarrierCoupling(SunPlanetConstraint1))

Position on shaft (mm)	[y _{local}]	45.0000	
Position in global system (mm)	[y _{global}]	71.0000	
Effective diameter (mm)		170.0000	
Radial force factor (-)		0.0000	
Direction of the radial force (°)		0.0000	
Axial force factor (-)		0.0000	
Length of load application (mm)		20.0000	
Power (kW)		76.2883	driven (input)
Torque (Nm)		475.3333	
Axial force (N)		0.0000	
Shearing force X (N)		0.0000	
Shearing force Z (N)		0.0000	
Bending moment X (Nm)		0.0000	
Bending moment Z (Nm)		0.0000	
Mass (kg)		0.0000	
Mass moment of inertia J_p (kg*m²)		0.0000	
Mass moment of inertia J_{xx} (kg*m²)		0.0000	
Mass moment of inertia J_{zz} (kg*m²)		0.0000	
Eccentricity (mm)		0.0000	

2.3.6 Coupling (PlanetCarrierCoupling(SunPlanetConstraint2))

Position on shaft (mm)	[y _{local}]	45.0000
Position in global system (mm)	[y _{global}]	71.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		20.0000
Power (kW)		107.2459 driven (input)
Torque (Nm)		668.2222
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

2.4 Bearing**2.4.1 Support8 (Own input)**

Bearing position (mm)	[y _{local}]	5.000
Bearing position (mm)	[y _{global}]	31.000
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: freeRy: freeRz: free		

2.4.2 Support9 (Own input)

Bearing position (mm)	[y _{local}]	70.000
Bearing position (mm)	[y _{global}]	96.000
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: freeRy: freeRz: free		

3 Shaft definition (RingShaft)**3.1 Outer contour****3.1.1 Cylinder (Cylinder), 0.000 mm ... 50.000 mm**

Diameter (mm)	[d]	224.0000
Length (mm)	[l]	50.0000
Surface roughness (µm)	[Rz]	8.0000

3.2 Inner contour

3.2.1 Cylindrical bore (Cylindrical bore), 0.000 mm ... 50.000 mm

Diameter (mm)	[d]	200.0000
Length (mm)	[l]	50.0000
Surface roughness (μm)	[Rz]	8.0000

3.3 Forces**3.3.1 Coupling (CouplingOutput(Output))**

Position on shaft (mm)	[y _{local}]	24.0000
Position in global system (mm)	[y _{global}]	64.0000
Effective diameter (mm)		230.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		10.0000
Power (kW)		152.5766 driving (output)
Torque (Nm)		-630.3330
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

3.3.2 Cylindrical gear (RingGear(180.0))

Position on shaft (mm)	[y _{local}]	25.0000
Position in global system (mm)	[y _{global}]	65.0000
Operating pitch diameter (mm)		-279.7996
Spur gear		
Working pressure angle at normal section (°)		22.7954
Position of contact (°)		180.0000
Length of load application (mm)		50.0000
Power (kW)		50.8589 driven (input)
Torque (Nm)		210.1111
Axial force (N)		-0.0000
Shearing force X (N)		-631.1862
Shearing force Z (N)		1501.8684
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

3.3.3 Cylindrical gear (RingGear(300.0))

Position on shaft (mm)	[y _{local}]	25.0000
Position in global system (mm)	[y _{global}]	65.0000
Operating pitch diameter (mm)		-279.7996
Spur gear		
Working pressure angle at normal section (°)		22.7954
Position of contact (°)		300.0000
Length of load application (mm)		50.0000
Power (kW)		50.8589 driven (input)

KISSsoft		
Torque (Nm)		210.1111
Axial force (N)		-0.0000
Shearing force X (N)		-985.0631
Shearing force Z (N)		-1297.5575
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		-0.0000
3.3.4 Cylindrical gear (RingGear(420.0))		
Position on shaft (mm)	[y _{local}]	25.0000
Position in global system (mm)	[y _{global}]	65.0000
Operating pitch diameter (mm)		-279.7996
Spur gear		
Working pressure angle at normal section (°)		22.7954
Position of contact (°)		420.0000
Length of load application (mm)		50.0000
Power (kW)		50.8589 driven (input)
Torque (Nm)		210.1111
Axial force (N)		-0.0000
Shearing force X (N)		1616.2493
Shearing force Z (N)		-204.3109
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		-0.0000
3.4 Bearing		
3.4.1 Support7 (Own input)		
Bearing position (mm)	[y _{local}]	24.000
Bearing position (mm)	[y _{global}]	64.000
Degrees of freedom		
X: fixedY: fixedZ: fixed		
Rx: fixedRy: freeRz: fixed		
4 Shaft definition (Sun1Shaft)		
4.1 Outer contour		
4.1.1 Cylinder (Cylinder), 0.000 mm ... 113.000 mm		
Diameter (mm)	[d]	30.0000
Length (mm)	[l]	113.0000
Surface roughness (µm)	[Rz]	8.0000
4.2 Forces		
4.2.1 Coupling (CouplingStopLS(StopLS))		
Position on shaft (mm)	[y _{local}]	6.0000
Position in global system (mm)	[y _{global}]	-55.0000
Effective diameter (mm)		40.0000

Radial force factor (-)	0.0000
Direction of the radial force (°)	0.0000
Axial force factor (-)	0.0000
Length of load application (mm)	10.0000
Power (kW)	0.0000
Torque (Nm)	-320.3333
Axial force (N)	0.0000
Shearing force X (N)	0.0000
Shearing force Z (N)	0.0000
Bending moment X (Nm)	0.0000
Bending moment Z (Nm)	0.0000
Mass (kg)	0.0000
Mass moment of inertia J_p (kg*m ²)	0.0000
Mass moment of inertia J_{xx} (kg*m ²)	0.0000
Mass moment of inertia J_{zz} (kg*m ²)	0.0000
Eccentricity (mm)	0.0000

4.2.2 Cylindrical gear (Sun1Gear(180.0))

Position on shaft (mm)	[y _{local}]	107.0000
Position in global system (mm)	[y _{global}]	46.0000
Operating pitch diameter (mm)		142.1384
Spur gear		
Working pressure angle at normal section (°)		22.7428
Position of contact (°)		180.0000
Length of load application (mm)		13.0000
Power (kW)		0.0000
Torque (Nm)		106.7778
Axial force (N)		-0.0000
Shearing force X (N)		629.8062
Shearing force Z (N)		1502.4476
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

4.2.3 Cylindrical gear (Sun1Gear(300.0))

Position on shaft (mm)	[y _{local}]	107.0000
Position in global system (mm)	[y _{global}]	46.0000
Operating pitch diameter (mm)		142.1384
Spur gear		
Working pressure angle at normal section (°)		22.7428
Position of contact (°)		300.0000
Length of load application (mm)		13.0000
Power (kW)		0.0000
Torque (Nm)		106.7778
Axial force (N)		-0.0000
Shearing force X (N)		-1616.0609
Shearing force Z (N)		-205.7956
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		-0.0000

4.2.4 Cylindrical gear (Sun1Gear(420.0))

Position on shaft (mm)	[y _{local}]	107.0000
Position in global system (mm)	[y _{global}]	46.0000
Operating pitch diameter (mm)		142.1384
Spur gear		
Working pressure angle at normal section (°)		22.7428
Position of contact (°)		420.0000

Length of load application (mm)	13.0000
Power (kW)	0.0000
Torque (Nm)	106.7778
Axial force (N)	-0.0000
Shearing force X (N)	986.2547
Shearing force Z (N)	-1296.6520
Bending moment X (Nm)	0.0000
Bending moment Z (Nm)	-0.0000

4.3 Bearing

4.3.1 RollerBearing3 (SKF 61806)

Bearing type		Deep groove ball bearing (single row)
Bearing position (mm)	[y _{lokal}]	17.000
Bearing position (mm)	[y _{global}]	-44.000
Attachment of external ring		Set fixed bearing left
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	42.000
Width (mm)	[b]	7.000
Corner radius (mm)	[r]	0.600
Number of rolling bodies	[Z]	17
Rolling body reference circle (mm)	[D _{pw}]	36.000
Diameter rolling body (mm)	[D _w]	3.500
Diameter, external race (mm)	[d _o]	39.506
Diameter, internal race (mm)	[d _i]	32.494
Radius of curvature, external race (mm)	[r _o]	1.855
Radius of curvature, internal race (mm)	[r _i]	1.820
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		ISO 5753-1:2009 C0 (12.50 µm)
Axial clearance		Pe = 92.70 µm
Basic static load rating (kN)	[C ₀]	2.900
Basic dynamic load rating (kN)	[C]	4.100
Fatigue load limit (kN)	[C _u]	0.146
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	4.521
Basic static load rating (kN)	[C _{0theo}]	3.421
Correction factor Basic dynamic load rating		
	[f _c]	1.000
Correction factor Basic static load rating		
	[f _{c0}]	1.000

4.3.2 RollerBearing4 (SKF 61806)

Bearing type		Deep groove ball bearing (single row)
Bearing position (mm)	[y _{lokal}]	73.000
Bearing position (mm)	[y _{global}]	12.000
Attachment of external ring		Set fixed bearing left
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	42.000
Width (mm)	[b]	7.000
Corner radius (mm)	[r]	0.600
Number of rolling bodies	[Z]	17
Rolling body reference circle (mm)	[D _{pw}]	36.000

Diameter rolling body (mm)	[D _w]	3.500
Diameter, external race (mm)	[d _o]	39.506
Diameter, internal race (mm)	[d _i]	32.494
Radius of curvature, external race (mm)	[r _o]	1.855
Radius of curvature, internal race (mm)	[r _i]	1.820
Calculation with approximate bearings internal geometry (*)		
Bearing clearance	ISO 5753-1:2009 C0 (12.50 µm)	
Axial clearance	Pe = 92.70 µm	
Basic static load rating (kN)	[C ₀]	2.900
Basic dynamic load rating (kN)	[C]	4.100
Fatigue load limit (kN)	[C _u]	0.146
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	4.521
Basic static load rating (kN)	[C _{0theo}]	3.421
Correction factor Basic dynamic load rating	[f _c]	1.000
Correction factor Basic static load rating	[f _{c0}]	1.000

5 Shaft definition (Sun2Shaft)

5.1 Outer contour

5.1.1 Cylinder (Cylinder), 0.000 mm ... 87.000 mm

Diameter (mm)	[d]	30.0000
Length (mm)	[l]	87.0000
Surface roughness (µm)	[Rz]	8.0000

5.2 Forces

5.2.1 Coupling (CouplingInput(Input))

Position on shaft (mm)	[y _{local}]	82.0000
Position in global system (mm)	[y _{global}]	152.0000
Effective diameter (mm)		40.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		10.0000
Power (kW)		152.5767 driven (input)
Torque (Nm)		310.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia J _p (kg*m ²)		0.0000
Mass moment of inertia J _{xx} (kg*m ²)		0.0000
Mass moment of inertia J _{zz} (kg*m ²)		0.0000
Eccentricity (mm)		0.0000

5.2.2 Cylindrical gear (Sun2Gear(135.2))

Position on shaft (mm)	[y _{local}]	9.0000
Position in global system (mm)	[y _{global}]	79.0000
Operating pitch diameter (mm)		68.9990
Spur gear		
Working pressure angle at normal section (°)		23.1792
Position of contact (°)		135.1837
Length of load application (mm)		23.0000
Power (kW)		50.8589 driving (output)
Torque (Nm)		-103.3333
Axial force (N)		0.0000
Shearing force X (N)		-1201.3946
Shearing force Z (N)		-3028.6416
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		-0.0000

5.2.3 Cylindrical gear (Sun2Gear(255.2))

Position on shaft (mm)	[y _{local}]	9.0000
Position in global system (mm)	[y _{global}]	79.0000
Operating pitch diameter (mm)		68.9990
Spur gear		
Working pressure angle at normal section (°)		23.1792
Position of contact (°)		255.1837
Length of load application (mm)		23.0000
Power (kW)		50.8589 driving (output)
Torque (Nm)		-103.3333
Axial force (N)		0.0000
Shearing force X (N)		3223.5779
Shearing force Z (N)		473.8826
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		-0.0000

5.2.4 Cylindrical gear (Sun2Gear(375.2))

Position on shaft (mm)	[y _{local}]	9.0000
Position in global system (mm)	[y _{global}]	79.0000
Operating pitch diameter (mm)		68.9990
Spur gear		
Working pressure angle at normal section (°)		23.1792
Position of contact (°)		375.1837
Length of load application (mm)		23.0000
Power (kW)		50.8589 driving (output)
Torque (Nm)		-103.3333
Axial force (N)		0.0000
Shearing force X (N)		-2022.1833
Shearing force Z (N)		2554.7591
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		0.0000

5.3 Bearing

5.3.1 RollerBearing5 (SKF 61806)

Bearing type	Deep groove ball bearing (single row)
--------------	---------------------------------------

Bearing position (mm)	[y _{lokal}]	72.000
Bearing position (mm)	[y _{global}]	142.000
Attachment of external ring	Set fixed bearing right	
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	42.000
Width (mm)	[b]	7.000
Corner radius (mm)	[r]	0.600
Number of rolling bodies	[Z]	17
Rolling body reference circle (mm)	[D _{pw}]	36.000
Diameter rolling body (mm)	[D _w]	3.500
Diameter, external race (mm)	[d _o]	39.506
Diameter, internal race (mm)	[d _i]	32.494
Radius of curvature, external race (mm)	[r _o]	1.855
Radius of curvature, internal race (mm)	[r _i]	1.820
Calculation with approximate bearings internal geometry (*)		
Bearing clearance	ISO 5753-1:2009 C0 (12.50 µm)	
Axial clearance	Pe = 92.70 µm	
Basic static load rating (kN)	[C ₀]	2.900
Basic dynamic load rating (kN)	[C]	4.100
Fatigue load limit (kN)	[C _u]	0.146
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	4.521
Basic static load rating (kN)	[C _{0theo}]	3.421
Correction factor Basic dynamic load rating	[f _c]	1.000
Correction factor Basic static load rating	[f _{c0}]	1.000

5.3.2 Support8 (Free bearing)

Bearing position (mm)	[y _{lokal}]	11.000
Bearing position (mm)	[y _{global}]	81.000
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: freeRy: freeRz: free		

6 Gears

7 Results

7.1 Shafts

Maximum deflection	8.396(μm) (Sun2Shaft pos =	157.000	mm)
Mass center of gravity			
CarrierShaft (mm)			44.461
RingShaft (mm)			25.000
Sun1Shaft (mm)			56.500
Sun2Shaft (mm)			43.500
Total axial load			
CarrierShaft (N)			0.000
RingShaft (N)			0.000
Sun1Shaft (N)			0.000
Sun2Shaft (N)			0.000
Torsion of the shafts under torque			
(Difference between left and right shaft end)			
CarrierShaft (°)			-0.000
RingShaft (°)			0.000
Sun1Shaft (°)			0.294
Sun2Shaft (°)			0.205

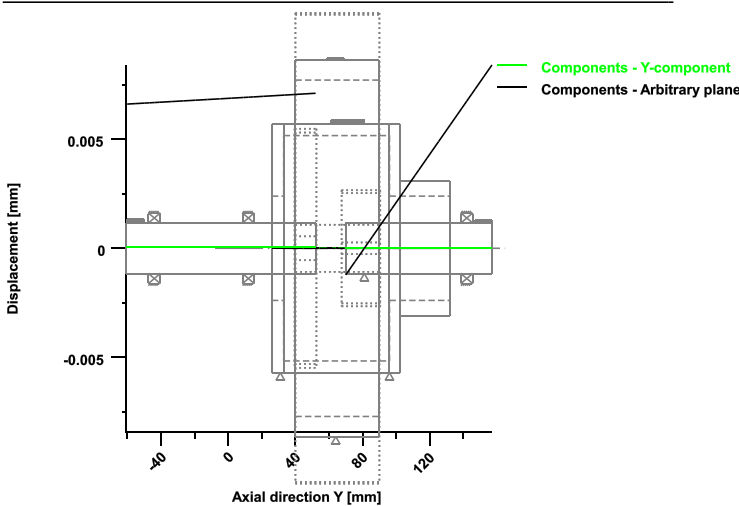
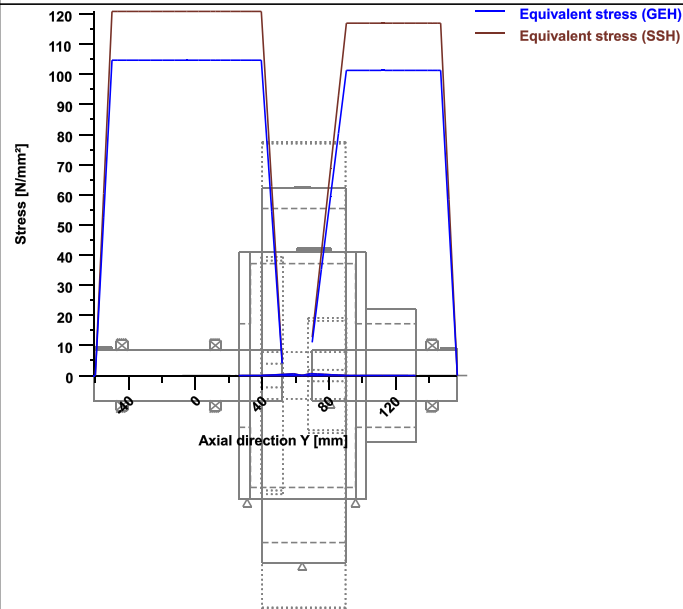


Figure: Deformation (bending etc.) (Arbitrary plane 270.0125262 127)

KISSsoft


Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\text{sigV} = ((\text{sigB} + \text{sigZ}, D)^2 + 3 * (\text{tauT} + \text{tauS})^2)^{1/2}$

SSH(Tresca): $\text{sigV} = ((\text{sigB} - \text{sigZ}, D)^2 + 4 * (\text{tauT} + \text{tauS})^2)^{1/2}$

Figure: Equivalent stress

7.2 Bearing

Probability of failure	[n]	10.00	%	
Axial clearance (ISO 281)	[u _A]			10.00 μm
Lubricant	ISO-VG 220			
Lubricant - service temperature	[T _B]			70.00 °C
Rolling bearing stiffness calculated from internal geometry				

7.3 'CarrierShaft' Bearing 'Support8'

Position (Y-coordinate)	[y]	5.00	mm	
Bearing reaction force	[F _x]			0.000 kN
Bearing reaction force	[F _y]			0.000 kN
Bearing reaction force	[F _z]			0.013 kN
Bearing reaction force	[F _i]			0.013 kN
Displacement of bearing	[u _x]			0.000 μm
Displacement of bearing	[u _y]			0.000 μm
Displacement of bearing	[u _z]			0.000 μm
Displacement of bearing	[u _i]			0.000 μm
Misalignment of bearing	[r _x]			-0.000 mrad
Misalignment of bearing	[r _y]			-0.000 mrad
Misalignment of bearing	[r _z]			0.000 mrad
Misalignment of bearing	[r _i]			0.000 mrad

7.4 'CarrierShaft' Bearing 'Support9'

Position (Y-coordinate)	[y]	70.00	mm		
Bearing reaction force	[F _x]			-0.000	kN
Bearing reaction force	[F _y]			0.000	kN
Bearing reaction force	[F _z]			0.021	kN
Bearing reaction force	[F _t]			0.021	kN
Displacement of bearing	[u _x]			0.000	µm
Displacement of bearing	[u _y]			-0.000	µm
Displacement of bearing	[u _z]			0.000	µm
Displacement of bearing	[u _t]			0.000	µm
Misalignment of bearing	[r _x]			-0.000	mrad
Misalignment of bearing	[r _y]			-0.000	mrad
Misalignment of bearing	[r _z]			-0.000	mrad
Misalignment of bearing	[r _t]			0.000	mrad

7.5 'RingShaft' Bearing 'Support7'

Position (Y-coordinate)	[y]	24.00	mm		
Bearing reaction force	[F _x]			-0.000	kN
Bearing reaction force	[F _y]			-0.000	kN
Bearing reaction force	[F _z]			0.031	kN
Bearing reaction force	[F _t]			0.031	kN
Bearing reaction moment	[M _x]			0.03	Nm
Bearing reaction moment	[M _y]			0.00	Nm
Bearing reaction moment	[M _z]			0.00	Nm
Bearing reaction moment	[M _t]			0.03	Nm
Displacement of bearing	[u _x]			0.000	µm
Displacement of bearing	[u _y]			0.000	µm
Displacement of bearing	[u _z]			0.000	µm
Displacement of bearing	[u _t]			0.000	µm
Misalignment of bearing	[r _x]			0.000	mrad
Misalignment of bearing	[r _y]			-0.000	mrad
Misalignment of bearing	[r _z]			0.000	mrad
Misalignment of bearing	[r _t]			0.000	mrad

7.6 'Sun1Shaft' Rolling bearing 'RollerBearing3'

Position (Y-coordinate)	[y]	17.00	mm		
Dynamic equivalent load	[P]	0.00	kN		
Static equivalent load	[P ₀]			0.00	kN
Minimum EHL lubricant film thickness	[h _{min}]			0.000	µm
Spin to roll ratio	[ω _s /ω _{roll}]	0.000			
Life modification factor for reliability	[a ₁]			1.000	

7.6.1 Results according to ISO 281

Lubricant	ISO-VG 220				
Load ratio	[C/P]		2260.116		
Operating viscosity	[ν]		48.884	mm ² /s	
Reference viscosity	[ν ₁]		0.000	mm ² /s	
Viscosity ratio	[κ]		0.000		
Basic bearing rating life	[L _{nh}]		> 1000000	h	
Static safety factor	[S ₀]		1598.62		

7.6.2 Calculation with approximate bearings internal geometry (ISO/TS 16281)

Operating bearing clearance	[Pd]	12.500	μm
Reference rating life	[L _{nh}]	> 1000000	h
Modified reference rating life	[L _{nmh}]	> 1000000	h
Bearing reaction force	[F _x]	-0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	0.002	kN
Bearing reaction force	[F _i]	0.002	kN
Bearing reaction moment	[M _x]	0.00	Nm
Bearing reaction moment	[M _y]	0.00	Nm
Bearing reaction moment	[M _z]	0.00	Nm
Bearing reaction moment	[M _i]	0.00	Nm
Displacement of bearing	[u _x]	0.054	μm
Displacement of bearing	[u _y]	0.070	μm
Displacement of bearing	[u _z]	-6.674	μm
Displacement of bearing	[u _i]	6.674	μm
Misalignment of bearing	[r _x]	-0.004	mrad
Misalignment of bearing	[r _y]	0.559	mrad
Misalignment of bearing	[r _z]	-0.002	mrad
Misalignment of bearing	[r _i]	0.005	mrad
Oil level	[H]	0.000 mm	
Rolling moment of friction	[M _r]	0.000	Nm
Sliding moment of friction	[M _{sl}]	0.000	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.000	Nm
Power loss	[P _{loss}]	0.000	W

The moment of friction is calculated according to the details in SKF Catalog 2018.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

7.7 'Sun1Shaft' Rolling bearing 'RollerBearing4'

Position (Y-coordinate)	[y]	73.00 mm	
Dynamic equivalent load	[P]	0.00 kN	
Static equivalent load	[P ₀]		0.00 kN
Minimum EHL lubricant film thickness	[h _{min}]		0.000 μm
Spin to roll ratio	[ω_s/ω_{roll}]	0.000	
Life modification factor for reliability	[a ₁]		1.000

7.7.1 Results according to ISO 281

Lubricant	ISO-VG 220		
Load ratio	[C/P]	955.533	
Operating viscosity	[ν]	48.884	mm^2/s
Reference viscosity	[ν_1]	0.000	mm^2/s
Viscosity ratio	[κ]	0.000	
Basic bearing rating life	[L _{nh}]	> 1000000	h
Static safety factor	[S ₀]	675.86	

7.7.2 Calculation with approximate bearings internal geometry (ISO/TS 16281)

Operating bearing clearance	[Pd]	12.500	μm
Reference rating life	[L _{nh}]	> 1000000	h

Modified reference rating life	[L _{nmh}]	> 1000000	h
Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	0.004	kN
Bearing reaction force	[F _i]	0.004	kN
Bearing reaction moment	[M _x]	0.00	Nm
Bearing reaction moment	[M _y]	0.00	Nm
Bearing reaction moment	[M _z]	0.00	Nm
Bearing reaction moment	[M _i]	0.00	Nm
Displacement of bearing	[u _x]	0.184	µm
Displacement of bearing	[u _y]	0.070	µm
Displacement of bearing	[u _z]	-6.919	µm
Displacement of bearing	[u _i]	6.921	µm
Misalignment of bearing	[r _x]	-0.004	mrاد
Misalignment of bearing	[r _y]	3.406	mrاد
Misalignment of bearing	[r _z]	-0.002	mrاد
Misalignment of bearing	[r _i]	0.005	mrاد
Oil level	[H]	0.000 mm	
Rolling moment of friction	[M _r]	0.000	Nm
Sliding moment of friction	[M _{sl}]	0.000	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.000	Nm
Power loss	[P _{loss}]	0.000	W

The moment of friction is calculated according to the details in SKF Catalog 2018.

The calculation is always performed with a coefficient for additives in the lubricant µbl=0.15.

7.8 'Sun2Shaft' Rolling bearing 'RollerBearing5'

Position (Y-coordinate)	[y]	72.00 mm	
Dynamic equivalent load	[P]	0.00 kN	
Static equivalent load	[P ₀]		0.00 kN
Minimum EHL lubricant film thickness	[h _{min}]		0.256 µm
Spin to roll ratio	[ω _s /ω _{roll}]	0.000	
Life modification factor for reliability	[a ₁]		1.000

7.8.1 Results according to ISO 281

Lubricant	ISO-VG 220		
Load ratio	[C/P]	1681.264	
Operating viscosity	[v]	48.884	mm ² /s
Reference viscosity	[v ₁]	0.000	mm ² /s
Viscosity ratio	[κ]	0.000	
Basic bearing rating life	[L _{nh}]	> 1000000	h
Static safety factor	[S ₀]	1189.19	

7.8.2 Calculation with approximate bearings internal geometry (ISO/TS 16281)

Operating bearing clearance	[Pd]	12.500	µm
Reference rating life	[L _{nh}]	> 1000000	h
Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	0.002	kN
Bearing reaction force	[F _i]	0.002	kN
Bearing reaction moment	[M _x]	0.00	Nm

Bearing reaction moment	$[M_y]$	0.00	Nm
Bearing reaction moment	$[M_z]$	0.00	Nm
Bearing reaction moment	$[M_i]$	0.00	Nm
Displacement of bearing	$[u_x]$	0.001	μm
Displacement of bearing	$[u_y]$	-0.000	μm
Displacement of bearing	$[u_z]$	-6.739	μm
Displacement of bearing	$[u_i]$	6.739	μm
Misalignment of bearing	$[r_x]$	-0.110	mrad
Misalignment of bearing	$[r_y]$	3.093	mrad
Misalignment of bearing	$[r_z]$	-0.000	mrad
Misalignment of bearing	$[r_i]$	0.110	mrad
Oil level	$[H]$	0.000 mm	
Rolling moment of friction	$[M_r]$	0.005	Nm
Sliding moment of friction	$[M_{sl}]$	0.000	Nm
Moment of friction, seals	$[M_{seal}]$	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	$[M_{drag}]$	0.000	Nm
Torque of friction	$[M_{loss}]$	0.005	Nm
Power loss	$[P_{loss}]$	2.428	W

The moment of friction is calculated according to the details in SKF Catalog 2018.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

7.9 'Sun2Shaft' Bearing 'Support8'

Position (Y-coordinate)	$[y]$	11.00 mm	
Bearing reaction force	$[F_x]$	-0.000	kN
Bearing reaction force	$[F_y]$	0.000	kN
Bearing reaction force	$[F_z]$	0.002	kN
Bearing reaction force	$[F_i]$	0.002	kN
Displacement of bearing	$[u_x]$	0.000	μm
Displacement of bearing	$[u_y]$	0.000	μm
Displacement of bearing	$[u_z]$	0.000	μm
Displacement of bearing	$[u_i]$	0.000	μm
Misalignment of bearing	$[r_x]$	-0.111	mrad
Misalignment of bearing	$[r_y]$	0.188	mrad
Misalignment of bearing	$[r_z]$	-0.000	mrad
Misalignment of bearing	$[r_i]$	0.111	mrad

(*) Note about roller bearings with an approximated bearing geometry:

The internal geometry of these bearings has not been input in the database.

The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog).

For this reason, the geometry may be different from the actual geometry.

In some situations, this may result in significant variations in roller bearing stiffness.

7.9.1 Damage (%)

[Lreq] (0.000)

Bin no	B1	B2	B3
1	0.00	0.00	0.00

Σ	0.00	0.00	0.00

7.9.2 Utilization (%)

[Lreq] (0.000)

B1	B2	B3
0.00	0.00	0.00

Note: Utilization = $(L_{req}/L_h)^{1/k}$

KISSsoft

Ball bearing: $k = 3$, roller bearing: $k = 10/3$

B1 : RollerBearing3

B2 : RollerBearing4

B3 : RollerBearing5

7.9.3 Calculation of the factors required to define reliability $R(t)$ using the Weibull distribution. t in (h)

Reliability not calculated

_O.GroupBox.PlanetPlanetConstraint1.Planet1Planet2_calc

1 Overview

Calculation method	DIN 3990:1987 Method B	
Drawing or article number:		
Gear 1:	PlanetGear1(PlanetPlanetConstraint1)	
Gear 2:	PlanetGear2(PlanetPlanetConstraint1)	
	----- Gear 1 -----	Gear 2 --
Power (kW)	[P]	34.274
Speed (1/min)	[n]	3167.4 2741.0
Torque (Nm)	[T]	103.3 119.4
Application factor	[KA]	1.25
Required service life (h)	[H]	20000.00
Gear driving (+) / driven (-)	-	+
Working flank gear 1:	Left flank	
Gear 1 direction of rotation:	Clockwise	

2 Tooth geometry and material

Geometry calculation according to	ISO 21771:2007	
	----- Gear 1 -----	Gear 2 --
Center distance (mm)	[a]	74.360
Center distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	1.5000
Normal pressure angle (°)	[α_n]	20.0000
Helix angle at reference circle (°)	[β]	0.0000
Number of teeth	[z]	45 52
Facewidth (mm)	[b]	50.00 23.00
Hand of gear	Spur gear	
Accuracy grade	[Q-DIN 3961:1978]	6 6
Inner diameter (mm)	[di]	40.00 40.00
Inner diameter of gear rim (mm)	[dbi]	0.00 0.00

2.1 Material

Gear 1
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy $J=12\text{mm}<\text{HRC}28$
 Gear 2
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy $J=12\text{mm}<\text{HRC}28$

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

Surface hardness		HRC 61	HRC 61
Fatigue strength, tooth root stress (N/mm ²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1200.00	1200.00
Yield point (N/mm ²)	[σS]	850.00	850.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

2.2 Reference profiles

Gear reference profile 1:		
Reference profile, Own Input	1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A	
Dedendum coefficient	[hfP*]	1.250
Root radius factor	[pfP*]	0.380 (pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	

Gear reference profile 2:		
Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A	
Dedendum coefficient	[hfP*]	1.250
Root radius factor	[pfP*]	0.380 (pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	

2.2.1 Information on final machining

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

Type of profile modification:	none (only running-in)	
Tip relief by running in (μm)	[Ca L/R]	2.0 / 2.0 2.0 / 2.0

2.3 Lubrication

Lubrication type	Oil bath lubrication	
Type of oil	ISO-VG 220	
Lubricant base	Mineral-oil base	
Oil nominal kinematic viscosity at 40°C (mm ² /s)	[ν40]	220.00
Oil nominal kinematic viscosity at 100°C (mm ² /s)	[ν100]	17.50
Specific density at 15°C (kg/dm ³)	[ρ]	0.895
Oil temperature (°C)	[TS]	70.000

2.4 Gear pair

Overall transmission ratio	[itot]	-0.865
Gear ratio	[u]	1.156
Transverse module (mm)	[mt]	1.500
Transverse pressure angle (°)	[ot]	20.000
Working pressure angle (°)	[owt]	23.170
	[owt.e/i]	23.197 / 23.143
Working pressure angle at normal section (°)	[awn]	23.170
Helix angle at operating pitch circle (°)	[βw]	0.000
Base helix angle (°)	[βb]	0.000
Reference center distance (mm)	[ad]	72.750
Pitch on reference circle (mm)	[pt]	4.712
Base pitch (mm)	[pbt]	4.428
Transverse pitch on contact-path (mm)	[pet]	4.428
Sum of profile shift coefficients	[Σxi]	1.1572
Transverse contact ratio	[εα]	1.509
Transverse contact ratio with allowances	[εα.e/m/i]	1.517 / 1.506 / 1.495
Overlap ratio	[εβ]	0.000
Total contact ratio	[εγ]	1.509
Total contact ratio with allowances	[εγ.e/m/i]	1.517 / 1.506 / 1.495
Length of path of contact (mm)	[ga, e/i]	6.681 (6.720 / 6.622)
Length T1-A (mm)	[T1A]	16.963 (16.963 / 16.952)
Length T1-B (mm)	[T1B]	14.709 (14.671 / 14.758)
Length T1-C (mm)	[T1C]	13.573 (13.556 / 13.591)
Length T1-D (mm)	[T1D]	12.534 (12.534 / 12.524)
Length T1-E (mm)	[T1E]	10.281 (10.243 / 10.330)
Length T2-A (mm)	[T2A]	12.295 (12.257 / 12.344)
Length T2-B (mm)	[T2B]	14.549 (14.549 / 14.538)
Length T2-C (mm)	[T2C]	15.685 (15.664 / 15.705)
Length T2-D (mm)	[T2D]	16.724 (16.686 / 16.772)
Length T2-E (mm)	[T2E]	18.977 (18.977 / 18.966)
Length T1-T2 (mm)	[T1T2]	29.258 (29.220 / 29.296)
Minimal length of contact line (mm)	[Lmin]	23.000

2.5 Gear 1

Profile shift coefficient	[x]	0.5606
Tooth thickness, arc, in module	[sn*]	1.9789
Tip alteration (mm)	[k*mn]	-0.125
Reference diameter (mm)	[d]	67.500
Base diameter (mm)	[db]	63.429
Tip diameter (mm)	[da]	71.932
(mm)	[da.e/i]	71.932 / 71.922
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010
Tip form diameter (mm)	[dFa]	71.932
(mm)	[dFa.e/i]	71.932 / 71.922
Root diameter (mm)	[df]	65.432

Generating Profile shift coefficient	[xE.e/i]	0.4965 / 0.4599
Generated root diameter with xE (mm)	[df.e/i]	65.239 / 65.130
Root form diameter (mm)	[dFf]	66.281
(mm)	[dFf.e/i]	66.120 / 66.030
Involute length (mm)	[l_dFa-l_dFf]	3.078
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	2.216
(mm)	[ha.e/i]	2.216 / 2.211
Dedendum (mm)	[hf=mn*(hfP*-x)]	1.034
(mm)	[hf.e/i]	1.130 / 1.185
Tooth height (mm)	[h]	3.250
Virtual gear no. of teeth	[zn]	45.000
Normal tooth thickness at tip circle (mm)	[san]	1.091
(mm)	[san.e/i]	1.022 / 0.974
Normal tooth thickness at tip form circle (mm)	[sFan]	1.091
(mm)	[sFan.e/i]	1.022 / 0.974
Normal space width at root circle (mm)	[efn]	1.057
(mm)	[efn.e/i]	1.074 / 1.085

2.6 Gear 2

Profile shift coefficient	[x]	0.5966
Tooth thickness, arc, in module	[sn*]	2.0051
Tip alteration (mm)	[k*mn]	-0.125
Reference diameter (mm)	[d]	78.000
Base diameter (mm)	[db]	73.296
Tip diameter (mm)	[da]	82.540
(mm)	[da.e/i]	82.540 / 82.530
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010
Tip form diameter (mm)	[dFa]	82.540
(mm)	[dFa.e/i]	82.540 / 82.530
Root diameter (mm)	[df]	76.040
Generating Profile shift coefficient	[xE.e/i]	0.5325 / 0.4958
Generated root diameter with xE (mm)	[df.e/i]	75.847 / 75.737
Root form diameter (mm)	[dFf]	76.862
(mm)	[dFf.e/i]	76.694 / 76.600
Involute length (mm)	[l_dFa-l_dFf]	3.087
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	2.270
(mm)	[ha.e/i]	2.270 / 2.265
Dedendum (mm)	[hf=mn*(hfP*-x)]	0.980
(mm)	[hf.e/i]	1.076 / 1.131
Tooth height (mm)	[h]	3.250
Virtual gear no. of teeth	[zn]	52.000
Normal tooth thickness at tip circle (mm)	[san]	1.110
(mm)	[san.e/i]	1.041 / 0.993
Normal tooth thickness at tip form circle (mm)	[sFan]	1.110
(mm)	[sFan.e/i]	1.041 / 0.993
Normal space width at root circle (mm)	[efn]	1.039
(mm)	[efn.e/i]	1.053 / 1.061

2.7 Gear specific pair data Gear pair 1, Gear 1

Operating pitch diameter (mm)	[dw]	68.994
(mm)	[dw.e/i]	69.008 / 68.980
Active tip diameter (mm)	[dNa]	71.932
(mm)	[dNa.e/i]	71.932 / 71.922
Theoretical tip clearance (mm)	[c]	0.375
Effective tip clearance (mm)	[c.e/i]	0.546 / 0.456

Active root diameter (mm)	[dNf]	66.679
(mm)	[dNf.e/l]	66.709 / 66.655
Reserve (dNf-dFf)/2 (mm)	[cF.e/l]	0.340 / 0.268
Max. sliding velocity at tip (m/s)	[vga]	2.037
Specific sliding at the tip	[ζa]	0.373
Specific sliding at the root	[ζf]	-0.597
Mean specific sliding	[ζm]	0.373
Sliding factor on tip	[Kga]	0.183
Sliding factor on root	[Kgf]	-0.178
Roll angle at dFa (°)	[ξdFa.e/l]	30.645 / 30.625
Roll angle to dNa (°)	[ξdNa.e/l]	30.645 / 30.625
Roll angle to dNf (°)	[ξdNf.e/l]	18.662 / 18.505
Roll angle at dFf (°)	[ξdFf.e/l]	16.865 / 16.575
Diameter of single contact point B (mm)	[d-B]	69.919 (69.887 / 69.961)
Diameter of single contact point D (mm)	[d-D]	68.203 (68.203 / 68.196)
Addendum contact ratio	[ε]	0.765 (0.769 / 0.759)

2.8 Gear specific pair data Gear pair 1, Gear 2

Operating pitch diameter (mm)	[dw]	79.727
(mm)	[dw.e/l]	79.743 / 79.711
Active tip diameter (mm)	[dNa]	82.540
(mm)	[dNa.e/l]	82.540 / 82.530
Theoretical tip clearance (mm)	[c]	0.375
Effective tip clearance (mm)	[c.e/l]	0.546 / 0.456
Active root diameter (mm)	[dNf]	77.311
(mm)	[dNf.e/l]	77.342 / 77.287
Reserve (dNf-dFf)/2 (mm)	[cF.e/l]	0.371 / 0.296
Max. sliding velocity at tip (m/s)	[vga]	2.097
Specific sliding at the tip	[ζa]	0.374
Specific sliding at the root	[ζf]	-0.594
Mean specific sliding	[ζm]	0.373
Sliding factor on tip	[Kga]	0.178
Sliding factor on root	[Kgf]	-0.183
Roll angle at dFa (°)	[ξdFa.e/l]	29.669 / 29.652
Roll angle to dNa (°)	[ξdNa.e/l]	29.669 / 29.652
Roll angle to dNf (°)	[ξdNf.e/l]	19.299 / 19.163
Roll angle at dFf (°)	[ξdFf.e/l]	17.648 / 17.397
Diameter of single contact point B (mm)	[d-B]	78.860 (78.860 / 78.852)
Diameter of single contact point D (mm)	[d-D]	80.567 (80.535 / 80.607)
Addendum contact ratio	[ε]	0.743 (0.748 / 0.736)

3 General influence factors

		----- Gear 1 ----- Gear 2 --
Nominal circum. force at pitch circle (N)	[Ft]	3061.7
Axial force (N)	[Fa]	0.0
Radial force (N)	[Fr]	1114.4
Normal force (N)	[Fnorm]	3258.2
Nominal circumferential force per mm (N/mm)	[w]	133.12
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	2995.4
Axial force (N)	[Faw]	0.0
Radial force (N)	[Frw]	1282.0
Circumferential speed reference circle (m/s)	[v]	11.19

KISSsoft			
Circumferential speed operating pitch circle (m/s)	[v(dw)]	11.44	
Running-in value (μm)	[yp]	0.5	
Running-in value (μm)	[yf]	0.4	
Tolerances fpe, ff and fH β according to the tolerances in section 7			
Correction factor	[CM]	0.800	
Gear blank factor	[CR]	1.000	
Basic rack factor	[CBS]	0.975	
Material coefficient	[E/Est]	1.000	
Singular tooth stiffness (N/mm/ μm)	[c]	15.913	
Meshing stiffness (N/mm/ μm)	[cv]	21.985	
Reduced mass (kg/mm)	[mRed]	0.00879	
Resonance speed (min-1)	[nE1]	10610	
Resonance ratio (-)	[N]	0.299	
Subcritical range			
Running-in value (μm)	[yd]	0.5	
Bearing distance l of pinion shaft (mm)	[l]	100.000	
Distance s of pinion shaft (mm)	[s]	10.000	
Outside diameter of pinion shaft (mm)	[dsh]	50.000	
Load according to Figure 6.8, DIN 3990-1:1987	[-]	4	
0:6.8a, 1:6.8b, 2:6.8c, 3:6.8d, 4:6.8e			
Coefficient K' according to Figure 6.8, DIN 3990-1:1987	[K]	-1.00	
Without stiffening			
Tooth trace deviation (active) (μm)	[Ffy]	5.33	
from deformation of shaft (μm)	[fsh*B1]	0.17	
fsh = (μm) 0.17			
B1 = 1.00			
fH β 5 = (μm) 6.50			
Tooth without flank line modification			
Position of contact pattern: favorable			
from production tolerances (μm)	[fm α *B2]	9.00	
B2 = 1.00			
Tooth trace deviation, theoretical (μm)	[Ffx]	6.27	
Running-in value (μm)	[y β]	0.94	
Dynamic factor	[Kv]	1.169	
Face load factor - flank	[KH β]	1.301	
- Tooth root	[KF β]	1.253	
- Scuffing	[KB β]	1.301	
Transverse load factor - flank	[KH α]	1.000	
- Tooth root	[KF α]	1.000	
- Scuffing	[KB α]	1.000	
Number of load cycles (in mio.)	[NL]	3800.870	3289.214
4 Tooth root load capacity			
Calculation of Tooth form coefficients according method: B			
----- Gear 1 ----- Gear 2 --			
Calculated with generating profile shift coefficient	[xE.e]	0.4965	0.5325
Tooth form factor	[YF]	1.26	1.26
Stress correction factor	[YS]	2.27	2.29
Load application angle (°)	[α Fen]	23.26	23.13
Load application diameter (mm)	[den]	69.919	80.567

Bending moment arm (mm)	[hF]	1.68	1.70
Tooth thickness at root (mm)	[sFn]	3.43	3.45
Tooth root radius (mm)	[pF]	0.62	0.61
Bending moment arm (-)	[hF/mn]	1.120	1.134
Tooth thickness at root (-)	[sFn/mn]	2.286	2.302
Tooth root radius (-)	[pF/mn]	0.411	0.403
Calculation cross section diameter (mm)	[dsFn]	65.771	76.382
Tangents on calculation cross section (°)	[αsFn]	30.000	30.000
Notch parameter	[qs]	2.782	2.853
Contact ratio factor	[Yε]	1.000	
Helix angle factor	[Yβ]	1.000	
Effective facewidth (mm)	[beff]	26.00	23.00
Nominal stress at tooth root (N/mm²)	[σF0]	224.05	255.40
Tooth root stress (N/mm²)	[σF]	410.07	467.44
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdrelT]	1.003	1.003
Surface factor	[YRrelT]	0.957	0.957
Size factor, tooth root	[YX]	1.000	1.000
Life factor	[YNT]	1.000	1.000
$Y_{drelT} \cdot Y_{RrelT} \cdot Y_X \cdot Y_{NT}$		0.959	0.960
Alternating bending factor, mean stress influence coefficient	[YM]	1.000	1.000
Stress correction factor	[YST]	2.00	
$YST \cdot \sigma_{Flim}$ (N/mm²)	[σFE]	860.00	860.00
Permissible tooth root stress σ_{FG}/SF_{min} (N/mm²)	[σFP]	589.24	589.62
Limit strength tooth root (N/mm²)	[σFG]	824.93	825.46
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=σFG/σF]	2.01	1.77
Transmittable power (kW)	[kWRating]	49.25	43.23

5 Flank safety

		----- Gear 1 -----	Gear 2 --
Zone factor	[ZH]		2.300
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zε]		0.911
Helix angle factor	[Zβ]		1.000
Effective facewidth (mm)	[beff]		23.00
Nominal contact stress (N/mm²)	[σH0]		763.19
Contact stress at operating pitch circle (N/mm²)	[σHw]		1052.25
Single tooth contact factor	[ZB,ZD]	1.01	1.00
Contact stress (N/mm²)	[σHB, σHD]	1060.43	1052.25
Lubrication factor for NL	[ZL]	1.020	1.020
Speed factor for NL	[ZV]	1.003	1.003
Roughness factor for NL	[ZR]	0.956	0.956
Material hardening factor for NL	[ZW]	1.000	1.000
Life factor	[ZNT]	1.000	1.000
	[ZL*ZV*ZR*ZNT]	0.978	0.978
Limited pitting is permitted:		No	
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress, σ_{HG}/SH_{min} (N/mm²)	[σHP]	1466.62	1466.62
Pitting stress limit (N/mm²)	[σHG]	1466.62	1466.62
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress on operating pitch circle	[SHw]	1.39	1.39

KISSsoft

Safety against pressure, σHG/σHBD Single contact	[SHBD]	1.38	1.39
Safety regarding transmittable torque	[(SHBD)^2]	1.91	1.94
Transmittable power (kW)	[kWRating]	65.56	66.58

6 Micropitting

Calculation method according to

ISO/TS 6336-22:2018

Calculation has not been carried out, lubricant: Load stage micropitting test not known

7 Scuffing load capacity

Calculation method according to

DIN 3990:1987

Helical load factor for scuffing	[KBy]	1.000	
Lubrication coefficient for lubrication type	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Relative structural factor, scuffing	[XWrelT]	1.000	
Thermal contact factor (N/mm/s^0.5/K)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	10.46	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	23.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	253.056	
Angle factor	[Xαβ]	1.024	
ε1: 0.765 , ε2: 0.743			

7.1 Flash temperature-criteria

Tooth mass temperature (°C)	[θMB]	76.42	
θMB = θoil + XS*0.47*θflamax			
Maximum flash temperature (°C)	[θflamax]	13.67	
Scuffing temperature (°C)	[θS]	408.58	
Γ coordinates (point of highest temperature)	[Γ]	-0.205	
[Γ.A]= 0.250 [Γ.E]= -0.243			
Highest contact temp. (°C)	[θB]	90.09	
Flash factor (°K*N^-.75*s^0.5*m^-.5*mm)	[XM]	50.058	
Geometry factor	[XB]	0.142	
Load sharing factor	[XΓ]	0.409	
Dynamic viscosity (mPa*s)	[ηM]	32.59	(76.4 °C)
Coefficient of friction	[μm]	0.064	
Required safety	[SBmin]	2.000	
Margin of safety for scuffing, flash temperature	[SB]	16.845	

7.2 Integral temperature-criteria

Tooth mass temperature (°C)	[θMC]	77.06	
θMC = θoil + XS*0.70*θflaint			
Integral scuffing temperature (°C)	[θSint]	408.58	
Flash factor (°K*N^-.75*s^0.5*m^-.5*mm)	[XM]	50.058	
Contact ratio factor	[Xε]	0.279	
Dynamic viscosity (mPa*s)	[ηOil]	41.90	(70.0 °C)
Mean coefficient of friction	[μm]	0.059	
Geometry factor	[XBE]	0.169	

Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.011
Mean flash temperature (°C)	[θflaint]	10.09
Integral tooth flank temperature (°C)	[θint]	92.19
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	4.432
Safety referring to transmittable torque	[SSL]	15.260

8 Measurements for tooth thickness

		----- Gear 1 -----	Gear 2 --
Tooth thickness tolerance		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070 / -0.110	-0.070 / -0.110
Number of teeth spanned	[k]	6.000	7.000
Base tangent length (no backlash) (mm)	[Wk]	25.876	30.488
Base tangent length with allowance (mm)	[Wk.e/i]	25.810 / 25.772	30.422 / 30.384
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	68.472	79.352
Theoretical diameter of ball/pin (mm)	[DM]	2.737	2.719
Effective diameter of ball/pin (mm)	[DMeff]	2.750	2.750
Radial single-ball measurement backlash free (mm)	[MrK]	36.550	41.863
Radial single-ball measurement (mm)	[MrK.e/i]	36.473 / 36.429	41.785 / 41.740
Diameter of measuring circle (mm)	[dMMr.m]	69.023	79.659
Diametral measurement over two balls without clearance (mm)	[MdK]	73.056	83.726
Diametral two ball measure (mm)	[MdK.e/i]	72.904 / 72.816	83.570 / 83.481
Diametral measurement over pins without clearance (mm)	[MdR]	73.056	83.726
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	72.904 / 72.816	83.570 / 83.481
Dimensions over 3 pins without clearance (mm)	[Md3R]	73.014	0.000
Measurement over 3 pins with allowance (mm)	[Md3R.e/i]	72.861 / 72.773	0.000 / 0.000
Chordal tooth thickness (no backlash) (mm)	[sc]	2.967	3.007
Normal chordal tooth thickness with allowance (mm)	[sc.e/i]	2.898 / 2.859	2.938 / 2.898
Reference chordal height from da.m (mm)	[ha]	2.246	2.296
Tooth thickness, arc (mm)	[sn]	2.968	3.008
(mm)	[sn.e/i]	2.898 / 2.858	2.938 / 2.898
Backlash free center distance (mm)	[aControl.e/i]	74.192 / 74.095	
Backlash free center distance, allowances (mm)	[jta]	-0.168 / -0.265	
dNf.i with aControl (mm)	[dNf0.i]	66.272	76.889
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.076	0.097
Tip clearance (mm)	[c0.i(aControl)]	0.206	0.206
Center distance allowances (mm)	[Aa.e/i]	0.015 / -0.015	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.013 / -0.013	
Radial backlash (mm)	[jrw.e/i]	0.280 / 0.153	
Circumferential backlash (transverse section) (mm)	[jtw.e/i]	0.238 / 0.130	
Normal backlash (mm)	[jn.e/i]	0.217 / 0.121	
Torsional angle on input with output fixed:			
Total torsional angle (°)	[j.tSys]	0.3417/0.1872	

9 Toothing tolerances

	----- Gear 1 -----	Gear 2 --
According to DIN 3961:1978		

KISSsoft			
Accuracy grade	[Q-DIN3961]	6	6
Profile form deviation (µm)	[ff]	6.00	6.00
Profile slope deviation (µm)	[fHa]	5.00	5.00
Total profile deviation (µm)	[Ff]	8.00	8.00
Helix form deviation (µm)	[fbf]	8.00	5.50
Helix slope deviation (µm)	[fHb]	10.00	9.00
Total helix deviation (µm)	[Fb]	13.00	10.00
Normal base pitch deviation (µm)	[fpe]	7.00	7.00
Single pitch deviation (µm)	[fp]	7.00	7.00
Adjacent pitch difference (µm)	[fu]	9.00	9.00
Total cumulative pitch deviation (µm)	[Fp]	25.00	25.00
Sector pitch deviation over z/8 pitches (µm)	[Fpz/8]	15.00	15.00
Runout (µm)	[Fr]	16.00	16.00
Tooth Thickness Variation (µm)	[Rs]	10.00	10.00
Single flank composite, total (µm)	[Fi']	26.00	26.00
Single flank composite, tooth-to-tooth (µm)	[fi']	11.00	11.00
Radial composite, total (µm)	[Fi'']	20.00	20.00
Radial composite, tooth-to-tooth (µm)	[fi'']	8.00	8.00
According DIN 58405:1972 (Precision Mechanics)			
Tooth-to-tooth composite error (µm)	[fi'']	8.00	8.00
Composite error (µm)	[Fi'']	22.00	22.00
Axis alignment error (µm)	[fp]	12.64	12.64
Flank direction error (µm)	[fβ]	10.50	5.00
Runout (µm)	[Trk, Fr]	24.00	24.00
Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)			
6			
Maximum value for deviation error of axis (µm)	[fΣβ]	26.09	(Fβ=12.00)
Maximum value for inclination error of axes (µm)	[fΣδ]	52.17	
10 Modifying and defining the tooth form			
10.1 Data for the tooth form calculation			
Data not available.			
Please run the calculation in the "Tooth form" tab and open the main report again.			
11 Supplementary data			
Maximal possible center distance (eps_a=1.0)	[aMAX]	75.276	
Mass (kg)	[m]	0.958	0.663
Total mass (kg)	[mGes]	1.621	
Moment of inertia for system, relative to the input:			
calculation without consideration of the exact tooth shape			
Single gears, (da+df)/2...di (kg*m²)	[J]	0.0007569	0.0006535
System (da+df)/2...di (kg*m²)	[J]	0.001664	
Torsional stiffness at driving gear with fixed driven gear:			
Torsional stiffness (MNm/rad)	[cr]	0.664	
Torsion when subjected to nominal torque (°)	[δcr]	0.010	
Mean coefficient of friction (as defined in Niemann)	[μm]	0.054	
Wear sliding coef. by Niemann	[ζw]	0.563	
Loss factor	[HV]	0.082	
Gear power loss (kW)	[PVZ]	0.152	
Meshing efficiency (%)	[ηz]	99.557	

Sound pressure level according to Masuda, without contact analysis

[dB(A)]

72.2

12 Service life, damage

Required safety for tooth root

[SFmin]

1.40

Required safety for tooth flank

[SHmin]

1.00

Required service life

[H]

20000.00

Service life (calculated with required safeties):

System service life (h)

[Hatt]

> 1000000

Tooth root service life (h)

[HFatt]

1e+06

1e+06

Tooth flank service life (h)

[HHatt]

1e+06

1e+06

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

Damage relative to the required service life (H, 20000.0 h)

F1%

F2%

H1%

H2%

0.00

0.0000

0.0000

0.0000

13 Remarks

- Specifications with [..e/i] imply: Maximum [e] and minimum value [i] for
Taking all tolerances into account
Specifications with [..m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness allowance are taken into account.
The maximum and minimum clearance according to
the largest or smallest allowances are defined..
The calculation is performed for the operating pitch circle.
- Details of calculation method:
cy according to Method B
Kv according to Method B
KH β and KF β according to Method C
KH α , KF α according to Method B

_O.GroupBox.RingPlanetConstraint1.PlanetRing1_c alc

1 Overview

Calculation method	DIN 3990:1987 Method B		
Drawing or article number:			
Gear 1:	PlanetGear1(RingPlanetConstraint1)		
Gear 2:	RingGear(RingPlanetConstraint1)		
During calculation, the system takes into account the fact that this gear is a planet gear:			
	1		
Speed planet carrier (1/min)	[nSteg]	1532.6	
Absolute speed (1/min)	[n]	2311.5	
	----- Gear 1 ----- Gear 2 -----		
Power (kW)	[P]	17.137	
Relative speed (1/min)	[n]	3167.4	778.9
Torque (Nm)	[T]	51.7	210.1
Application factor	[KA]	1.25	
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		+	-
Working flank gear 1:	Right flank		
Gear 1 direction of rotation:	Clockwise		

2 Tooth geometry and material

Geometry calculation according to	ISO 21771:2007	
	----- Gear 1 ----- Gear 2 --	
Center distance (mm)	[a]	105.498
Center distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	1.5000
Normal pressure angle (°)	[on]	20.0000
Helix angle at reference circle (°)	[β]	0.0000
Number of teeth	[z]	45 -183
Facewidth (mm)	[b]	50.00 50.00
Hand of gear	Spur gear	
Axial offset (mm)	[bv]	20.00
Accuracy grade	[Q-DIN 3961:1978]	6 6
Inner diameter (mm)	[di]	40.00
External diameter (mm)	[d _e]	294.70
Inner diameter of gear rim (mm)	[db _i]	0.00
Outer diameter of gear rim (mm)	[db _e]	0.00

2.1 Material

Gear 1

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ J=12mm<HRC28

Gear 2

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ J=12mm<HRC28

		----- Gear 1 -----	Gear 2 --
Surface hardness		HRC 61	HRC 61
Fatigue strength. tooth root stress (N/mm ²)	[σF _{lim}]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σH _{lim}]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1200.00	1200.00
Yield point (N/mm ²)	[σS]	850.00	850.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

2.2 Reference profiles

Gear reference profile 1:

Reference profile, Own Input

1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[αprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[αKP]	0.000	
		not topping	

Gear reference profile 2:

Reference profile, Own Input

1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[αprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[αKP]	0.000	
		not topping	

2.2.1 Information on final machining

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

Type of profile modification:

none (only running-in)

Tip relief by running in (μm)

[Ca L/R] 2.0 / 2.0 2.0 / 2.0

2.3 Lubrication

Lubrication type	Oil bath lubrication	
Type of oil	ISO-VG 220	
Lubricant base	Mineral-oil base	
Oil nominal kinematic viscosity at 40°C (mm²/s)	[v40]	220.00
Oil nominal kinematic viscosity at 100°C (mm²/s)	[v100]	17.50
Specific density at 15°C (kg/dm³)	[ρ]	0.895
Oil temperature (°C)	[TS]	70.000

2.4 Gear pair

Overall transmission ratio	[itot]	4.067
Gear ratio	[u]	-4.067
Transverse module (mm)	[mt]	1.500
Transverse pressure angle (°)	[αt]	20.000
Working pressure angle (°)	[αwt]	22.795
	[αwt.e/i]	22.773 / 22.818
Working pressure angle at normal section (°)	[αwn]	22.795
Helix angle at operating pitch circle (°)	[βw]	0.000
Base helix angle (°)	[βb]	0.000
Reference center distance (mm)	[ad]	103.500
Pitch on reference circle (mm)	[pt]	4.712
Base pitch (mm)	[pbt]	4.428
Transverse pitch on contact-path (mm)	[pet]	4.428
Sum of profile shift coefficients	[Σxi]	-1.4233
Transverse contact ratio	[εα]	1.646
Transverse contact ratio with allowances	[εα.e/m/i]	1.656 / 1.641 / 1.626
Overlap ratio	[εβ]	0.000
Total contact ratio	[εγ]	1.646
Total contact ratio with allowances	[εγ.e/m/i]	1.656 / 1.641 / 1.626
Length of path of contact (mm)	[ga, e/i]	7.289 (7.334 / 7.199)
Length T1-A (mm)	[T1A]	10.222 (10.177 / 10.281)
Length T1-B (mm)	[T1B]	13.083 (13.083 / 13.052)
Length T1-C (mm)	[T1C]	13.329 (13.343 / 13.314)
Length T1-D (mm)	[T1D]	14.650 (14.605 / 14.709)
Length T1-E (mm)	[T1E]	17.511 (17.511 / 17.480)
Length T2-A (mm)	[T2A]	51.097 (51.097 / 51.110)
Length T2-B (mm)	[T2B]	53.957 (54.003 / 53.881)
Length T2-C (mm)	[T2C]	54.203 (54.263 / 54.143)
Length T2-D (mm)	[T2D]	55.525 (55.525 / 55.538)
Length T2-E (mm)	[T2E]	58.386 (58.431 / 58.309)
Length T1-T2 (mm)	[T1T2]	40.874 (40.920 / 40.829)
Minimal length of contact line (mm)	[Lmin]	30.000

2.5 Gear 1

Profile shift coefficient	[x]	0.5606
Tooth thickness, arc, in module	[sn*]	1.9789
Tip alteration (mm)	[k*mn]	0.137

Reference diameter (mm)	[d]	67.500
Base diameter (mm)	[db]	63.429
Tip diameter (mm)	[da]	72.456
(mm)	[da.e/i]	72.456 / 72.426
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.030
Tip form diameter (mm)	[dFa]	72.456
(mm)	[dFa.e/i]	72.456 / 72.426
Root diameter (mm)	[df]	65.432
Generating Profile shift coefficient	[xE.e/i]	0.4965 / 0.4599
Generated root diameter with xE (mm)	[df.e/i]	65.239 / 65.130
Root form diameter (mm)	[dFf]	66.281
(mm)	[dFf.e/i]	66.120 / 66.030
Internal toothing: Calculation dFf with pinion type cutter ($z_0=45$, $x_0=0.000$)		
Involute length (mm)	[l_dFa-l_dFf]	3.377
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	2.478
(mm)	[ha.e/i]	2.478 / 2.463
Dedendum (mm)	[hf=mn*(hfP*-x)]	1.034
(mm)	[hf.e/i]	1.130 / 1.185
Tooth height (mm)	[h]	3.512
Virtual gear no. of teeth	[zn]	45.000
Normal tooth thickness at tip circle (mm)	[san]	0.813
(mm)	[san.e/i]	0.754 / 0.695
Normal tooth thickness at tip form circle (mm)	[sFan]	0.813
(mm)	[sFan.e/i]	0.754 / 0.695
Normal space width at root circle (mm)	[efn]	1.057
(mm)	[efn.e/i]	1.074 / 1.085

2.6 Gear 2

Profile shift coefficient	[x]	-1.9839
Tooth thickness, arc, in module	[sn*]	0.1266
Tip alteration (mm)	[k*mn]	0.000
Reference diameter (mm)	[d]	274.500
Base diameter (mm)	[db]	257.946
Tip diameter (mm)	[da]	277.452
(mm)	[da.e/i]	277.462 / 277.452
Tip diameter allowances (mm)	[Ada.e/i]	0.010 / -0.000
Tip form diameter (mm)	[dFa]	277.452
(mm)	[dFa.e/i]	277.462 / 277.452
Root diameter (mm)	[df]	284.202
Generating Profile shift coefficient	[xE.e/i]	-2.0709 / -2.1167
Generated root diameter with xE (mm)	[df.e/i]	284.600 / 284.463
Root form diameter (mm)	[dFf]	283.641
(mm)	[dFf.e/i]	283.986 / 283.868
Internal toothing: Calculation dFf with pinion type cutter ($z_0=50$, $x_0=0.000$)		
Involute length (mm)	[l_dFa-l_dFf]	3.366
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	-1.476
(mm)	[ha.e/i]	-1.476 / -1.481
Dedendum (mm)	[hf=mn*(hfP*-x)]	4.851
(mm)	[hf.e/i]	4.981 / 5.050
Tooth height (mm)	[h]	3.375
Virtual gear no. of teeth	[zn]	183.000
Normal tooth thickness at tip circle (mm)	[san]	1.320
(mm)	[san.e/i]	1.228 / 1.174
Normal tooth thickness at tip form circle (mm)	[sFan]	1.320
(mm)	[sFan.e/i]	1.228 / 1.174

Normal space width at root circle (mm)	[efn]	0.589
(mm)	[efn.e/i]	0.567 / 0.555

2.7 Gear specific pair data Gear pair 1, Gear 1

Operating pitch diameter (mm)	[dw]	68.803
(mm)	[dw.e/i]	68.792 / 68.815
Active tip diameter (mm)	[dNa]	72.456
(mm)	[dNa.e/i]	72.456 / 72.426
Theoretical tip clearance (mm)	[c]	0.375
Effective tip clearance (mm)	[c.e/i]	0.606 / 0.488
Active root diameter (mm)	[dNf]	66.643
(mm)	[dNf.e/i]	66.679 / 66.615
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.324 / 0.248
Max. sliding velocity at tip (m/s)	[vga]	1.046
Specific sliding at the tip	[ζa]	0.180
Specific sliding at the root	[ζf]	-0.229
Mean specific sliding	[ζm]	0.183
Sliding factor on tip	[Kga]	0.092
Sliding factor on root	[Kgf]	-0.068
Roll angle at dFa (°)	[ξdFa.e/i]	31.636 / 31.580
Roll angle to dNa (°)	[ξdNa.e/i]	31.636 / 31.580
Roll angle to dNf (°)	[ξdNf.e/i]	18.574 / 18.386
Roll angle at dFf (°)	[ξdFf.e/i]	16.865 / 16.575
Diameter of single contact point B (mm)	[d-B]	68.614 (68.614 / 68.591)
Diameter of single contact point D (mm)	[d-D]	69.870 (69.832 / 69.919)
Addendum contact ratio	[ε]	0.945 (0.941 / 0.941)

2.8 Gear specific pair data Gear pair 1, Gear 2

Operating pitch diameter (mm)	[dw]	279.800
(mm)	[dw.e/i]	279.846 / 279.753
Active tip diameter (mm)	[dNa]	277.452
(mm)	[dNa.e/i]	277.462 / 277.452
Theoretical tip clearance (mm)	[c]	0.512
Effective tip clearance (mm)	[c.e/i]	0.685 / 0.590
Active root diameter (mm)	[dNf]	283.146
(mm)	[dNf.e/i]	283.183 / 283.083
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.452 / 0.342
Max. sliding velocity at tip (m/s)	[vga]	0.777
Specific sliding at the tip	[ζa]	0.186
Specific sliding at the root	[ζf]	-0.220
Mean specific sliding	[ζm]	0.183
Sliding factor on tip	[Kga]	0.068
Sliding factor on root	[Kgf]	-0.092
Roll angle at dFa (°)	[ξdFa.e/i]	22.700 / 22.706
Roll angle to dNa (°)	[ξdNa.e/i]	22.700 / 22.706
Roll angle to dNf (°)	[ξdNf.e/i]	25.904 / 25.958
Roll angle at dFf (°)	[ξdFf.e/i]	26.324 / 26.387
Diameter of single contact point B (mm)	[d-B]	279.610 (279.645 / 279.551)
Diameter of single contact point D (mm)	[d-D]	280.834 (280.834 / 280.845)
Addendum contact ratio	[ε]	0.702 (0.715 / 0.685)

3 General influence factors

		----- Gear 1 ----- Gear 2 --
Nominal circum. force at pitch circle (N)	[Ft]	1530.9
Axial force (N)	[Fa]	0.0
Radial force (N)	[Fr]	557.2
Normal force (N)	[Fnorm]	1629.1
Nominal circumferential force per mm (N/mm)	[w]	51.03
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	1501.9
Axial force (N)	[Faw]	0.0
Radial force (N)	[Frw]	631.2
Circumferential speed reference circle (m/s)	[v]	11.19
Circumferential speed operating pitch circle (m/s)	[v(dw)]	11.41
Running-in value (μm)	[yp]	0.6
Running-in value (μm)	[yf]	0.4
Tolerances fpe, ff and fH β according to the tolerances in section 7		
Correction factor	[CM]	0.800
Gear blank factor	[CR]	1.000
Basic rack factor	[CBS]	0.975
Material coefficient	[E/Est]	1.000
Singular tooth stiffness (N/mm/ μm)	[c']	8.530
Meshing stiffness (N/mm/ μm)	[cv]	12.663
Reduced mass (kg/mm)	[mRed]	0.01225
Resonance speed (min-1)	[nE1]	6824
Resonance ratio (-)	[N]	0.464
Subcritical range		
Running-in value (μm)	[ya]	0.6
Bearing distance l of pinion shaft (mm)	[l]	100.000
Distance s of pinion shaft (mm)	[s]	10.000
Outside diameter of pinion shaft (mm)	[dsh]	50.000
Load according to Figure 6.8, DIN 3990-1:1987 0:6.8a, 1:6.8b, 2:6.8c, 3:6.8d, 4:6.8e	[-]	4
Coefficient K' according to Figure 6.8, DIN 3990-1:1987	[K']	-1.00
Without stiffening		
Tooth trace deviation (active) (μm)	[F β y]	5.38
from deformation of shaft (μm)	[fsh*B1]	0.13
fsh = (μm) 0.13		
B1 = 1.00		
fH β 5 = (μm) 6.50		
Tooth without flank line modification		
Position of contact pattern: favorable		
from production tolerances (μm)	[fm α *B2]	9.00
B2 = 1.00		
Tooth trace deviation, theoretical (μm)	[F β x]	6.33
Running-in value (μm)	[y β]	0.95
Dynamic factor	[Kv]	1.342
Face load factor - flank		
- Tooth root	[KH β]	1.398
- Scuffing	[KF β]	1.346
	[KB β]	1.398
Transverse load factor - flank		
- Tooth root	[KH α]	1.000
- Scuffing	[KF α]	1.000
	[KB α]	1.000

KISSsoft			
Number of load cycles (in mio.)	[NL]	3800.870	2803.920
4 Tooth root load capacity			
Calculation of tooth form coefficients with graphical method			
Determination of greatest value for YF * YS on the effective tooth form			
		----- Gear 1 -----	Gear 2 --
Tooth form factor	[YF]	1.23	0.89
Stress correction factor	[YS]	2.32	3.06
Load application angle (°)	[αFen]	23.04	23.17
Load application diameter (mm)	[den]	69.87	-279.61
Bending moment arm (mm)	[hF]	1.72	1.84
Tooth thickness at root (mm)	[sFn]	3.52	4.27
Tooth root radius (mm)	[ρF]	0.60	0.45
Bending moment arm (-)	[hF/mn]	1.146	1.228
Tooth thickness at root (-)	[sFn/mn]	2.344	2.848
Tooth root radius (-)	[ρF/mn]	0.401	0.297
Calculation cross section diameter (mm)	[dsFn]	65.584	284.284
Tangents on calculation cross section (°)	[αsFn]	37.896	46.744
Notch parameter	[qs]	2.924	4.793
Contact ratio factor	[Yε]		1.000
Helix angle factor	[Yβ]		1.000
Effective facewidth (mm)	[beff]	31.50	31.50
Nominal stress at tooth root (N/mm²)	[σF0]	92.14	88.09
Tooth root stress (N/mm²)	[σF]	208.04	198.87
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdrelT]	1.004	1.019
Surface factor	[YRrelT]	0.957	0.957
Size factor, tooth root	[YX]	1.000	1.000
Life factor	[YNT]	1.000	1.000
YdrelT*YRrelT*YX*YNT		0.960	0.975
Alternating bending factor, mean stress influence coefficient			
	[YM]	1.000	1.000
Stress correction factor	[YST]		2.00
YST*σFlim (N/mm²)	[σFE]	860.00	860.00
Permissible tooth root stress σFG/SFmin (N/mm²)	[σFP]	589.98	598.63
Limit strength tooth root (N/mm²)	[σFG]	825.97	838.08
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=σFG/σF]	3.97	4.21
Transmittable power (kW)	[kWRating]	48.60	51.59
5 Flank safety			
		----- Gear 1 -----	Gear 2 --
Zone factor	[ZH]		2.321
Elasticity factor (√N/mm²)	[ZE]		189.812
Contact ratio factor	[Zε]		0.886
Helix angle factor	[Zβ]		1.000
Effective facewidth (mm)	[beff]		30.00
Nominal contact stress (N/mm²)	[σH0]		294.71
Contact stress at operating pitch circle (N/mm²)	[σHw]		451.35
Single tooth contact factor	[ZB,ZD]	1.01	1.00
Contact stress (N/mm²)	[σHB, σHD]	456.60	451.35
Lubrication factor for NL	[ZL]	1.020	1.020

Speed factor for NL	[ZV]	1.003	1.003
Roughness factor for NL	[ZR]	0.977	0.977
Material hardening factor for NL	[ZW]	1.000	1.000
Life factor	[ZNT]	1.000	1.000
	[ZL*ZV*ZR*ZNT]	1.000	1.000
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress, $\sigma_{HG}/\sigma_{Hmin}$ (N/mm ²)	[σ_{HP}]	1500.31	1500.31
Pitting stress limit (N/mm ²)	[σ_{HG}]	1500.31	1500.31
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress on operating pitch circle			
	[SHw]	3.32	3.32
Safety against pressure, σ_{HG}/σ_{HBD} Single contact	[SHBD]	3.29	3.32
Safety regarding transmittable torque	[$(SHBD)^2$]	10.80	11.05
Transmittable power (kW)	[kWRating]	185.03	189.36

6 Micropitting

Calculation method according to ISO/TS 6336-22:2018
 Calculation has not been carried out, lubricant: Load stage micropitting test not known

7 Scuffing load capacity

Calculation method according to	DIN 3990:1987		
Helical load factor for scuffing	[KBy]	1.000	
Lubrication coefficient for lubrication type	[XS]	1.000	
Scuffing test and load stage	[FZGtest] FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12	
Relative structural factor, scuffing	[XWrelT]	1.000	
Thermal contact factor (N/mm/s ^{0.5} /K)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	4.49	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	30.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	119.685	
Angle factor	[X $\alpha\beta$]	1.018	
ε_1 : 0.945, ε_2 : 0.702			

7.1 Flash temperature-criteria

Tooth mass temperature (°C)	[θ_{MB}]	70.87	
$\theta_{MB} = \theta_{oil} + XS \cdot 0.47 \cdot \theta_{flamax}$			
Maximum flash temperature (°C)	[θ_{flamax}]	1.85	
Scuffing temperature (°C)	[θ_S]	408.58	
Γ coordinates (point of highest temperature)	[Γ]	-0.233	
[$\Gamma.A$]= -0.233 [$\Gamma.E$]= 0.314			
Highest contact temp. (°C)	[θ_B]	72.72	
Flash factor (°K*N ⁻¹ ·75*s ^{0.5} *m ⁻¹ ·5*mm)	[XM]	50.058	
Geometry factor	[XB]	0.065	
Load sharing factor	[X Γ]	0.333	
Dynamic viscosity (mPa*s)	[η_M]	40.46	(70.9 °C)
Coefficient of friction	[μ_m]	0.044	
Required safety	[SBmin]	2.000	

Margin of safety for scuffing, flash temperature	[SB]	124.085
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7.2 Integral temperature-criteria

Tooth mass temperature (°C)	[θMC]	70.75
$\theta_{MC} = \theta_{oil} + X_S \cdot 0.70 \cdot \theta_{flint}$		
Integral scuffing temperature (°C)	[θSint]	408.58
Flash factor (°K·N ⁻¹ ·s ⁻¹ ·m ⁻¹ ·mm)	[XM]	50.058
Contact ratio factor	[Xε]	0.235
Dynamic viscosity (mPa·s)	[ηOil]	41.90 (70.0 °C)
Mean coefficient of friction	[μ _m]	0.039
Geometry factor	[XBE]	0.062
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.025
Mean flash temperature (°C)	[θflint]	1.07
Integral tooth flank temperature (°C)	[θint]	72.36
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	5.646
Safety referring to transmittable torque	[SSL]	143.260

8 Measurements for tooth thickness

		----- Gear 1 -----	Gear 2 --
Tooth thickness tolerance		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070 / -0.110	-0.095 / -0.145
Number of teeth spanned	[k]	6.000	24.000
For internal toothing: k = measurement gap number			
Base tangent length (no backlash) (mm)	[Wk]	25.876	109.943
Base tangent length with allowance (mm)	[Wk.e/i]	25.810 / 25.772	110.032 / 110.079
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.089 / -0.136
Diameter of measuring circle (mm)	[dMWk.m]	68.472	280.443
Theoretical diameter of ball/pin (mm)	[DM]	2.737	2.426
Effective diameter of ball/pin (mm)	[DMeff]	2.750	2.500
Radial single-ball measurement backlash free (mm)	[MrK]	36.550	138.407
Radial single-ball measurement (mm)	[MrK.e/i]	36.473 / 36.429	138.584 / 138.523
Diameter of measuring circle (mm)	[dMMr.m]	69.023	280.582
Diametral measurement over two balls without clearance (mm)	[MdK]	73.056	276.805
Diametral two ball measure (mm)	[MdK.e/i]	72.904 / 72.816	277.158 / 277.037
Diametral measurement over pins without clearance (mm)	[MdR]	73.056	276.805
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	72.904 / 72.816	277.158 / 277.037
Dimensions over 3 pins without clearance (mm)	[Md3R]	73.014	276.794
Measurement over 3 pins with allowance (mm)	[Md3R.e/i]	72.861 / 72.773	277.148 / 277.026
Chordal tooth thickness (no backlash) (mm)	[sc]	2.967	0.190
Normal chordal tooth thickness with allowance (mm)	[sc.e/i]	2.898 / 2.859	0.095 / 0.045
Reference chordal height from da.m (mm)	[ha]	2.503	-1.478
Tooth thickness, arc (mm)	[sn]	2.968	0.190
(mm)	[sn.e/i]	2.898 / 2.858	0.095 / 0.045
Backlash free center distance (mm)	[aControl.e/i]	105.697 / 105.805	
Backlash free center distance, allowances (mm)	[ita]	0.199 / 0.307	
dNf.i with aControl (mm)	[dNf0.i]	66.178	283.797
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.029	0.035
Tip clearance (mm)	[c0.i(aControl)]	0.198	0.301

Center distance allowances (mm)	[Aa.e/l]	-0.018 / 0.018
Circumferential backlash from Aa (mm)	[ltw_Aa.e/l]	0.015 / -0.015
Radial backlash (mm)	[lrw.e/l]	0.324 / 0.182
Circumferential backlash (transverse section) (mm)	[ltw.e/l]	0.275 / 0.153
Normal backlash (mm)	[ln.e/l]	0.252 / 0.143
Torsional angle on input with output fixed:		
Total torsional angle (°)	[j.tSys]	0.4574/0.2556

9 Toothing tolerances

----- Gear 1 ----- Gear 2 --			
According to DIN 3961:1978			
Accuracy grade	[Q-DIN3961]	6	6
Profile form deviation (μm)	[ff]	6.00	6.00
Profile slope deviation (μm)	[fHa]	5.00	5.00
Total profile deviation (μm)	[Ff]	8.00	8.00
Helix form deviation (μm)	[fbf]	8.00	8.00
Helix slope deviation (μm)	[fHb]	10.00	10.00
Total helix deviation (μm)	[Fb]	13.00	13.00
Normal base pitch deviation (μm)	[fpe]	7.00	8.00
Single pitch deviation (μm)	[fp]	7.00	8.00
Adjacent pitch difference (μm)	[fu]	9.00	10.00
Total cumulative pitch deviation (μm)	[FP]	25.00	29.00
Sector pitch deviation over $z/8$ pitches (μm)	[Fpz/8]	15.00	18.00
Runout (μm)	[Fr]	16.00	19.00
Tooth Thickness Variation (μm)	[Rs]	10.00	11.00
Single flank composite, total (μm)	[Fi]	26.00	30.00
Single flank composite, tooth-to-tooth (μm)	[fi]	11.00	11.00
Radial composite, total (μm)	[Fi"]	20.00	24.00
Radial composite, tooth-to-tooth (μm)	[fi"]	8.00	10.00
According DIN 58405:1972 (Precision Mechanics)			
Tooth-to-tooth composite error (μm)	[fi"]	8.00	10.00
Composite error (μm)	[Fi"]	22.00	28.00
Axis alignment error (μm)	[fp]	17.93	17.93
Flank direction error (μm)	[fiβ]	10.50	10.50
Runout (μm)	[Trk, Fr]	24.00	30.00
Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)			
6			
Maximum value for deviation error of axis (μm)	[fΣβ]	21.67	(Fβ=13.00)
Maximum value for inclination error of axes (μm)	[fΣδ]	43.33	

10 Modifying and defining the tooth form

10.1 Data for the tooth form calculation

Calculation of Gear 1

Tooth form, Gear 1, Step 1: Automatic (final machining)
 $haP^* = 1.169$, $hfP^* = 1.250$, $pfP^* = 0.380$

Calculation of Gear 2

Tooth form, Gear 2, Step 1: Automatic (final machining)
 $z_0 = 50$, $x_0 = 0.0000$, $da_0 = 79.315 \text{ mm}$, $a_0 = -102.608 \text{ mm}$
 $haP_0^* = 1.438$, $paP_0^* = 0.380$, $hfP_0^* = 0.920$, $pfP_0^* = 0.000$

11 Supplementary data

Maximal possible center distance (eps_a=1.0)	[aMAX]	104.423	
Mass (kg)	[m]	0.970	2.455
Total mass (kg)	[mGes]	3.425	
calculation without consideration of the exact tooth shape			
Single gears, (da+df)/2...di (kg*m²)	[J]	0.00077	0.05086
Torsional stiffness at driving gear with fixed driven gear:			
Torsional stiffness (MNm/rad)	[cr]	0.375	
Torsion when subjected to nominal torque (°)	[δcr]	0.008	
Mean coefficient of friction (as defined in Niemann)	[μm]	0.045	
Wear sliding coef. by Niemann	[ζw]	0.301	
Loss factor	[HV]	0.039	
Gear power loss (kW)	[PVZ]	0.030	
Meshing efficiency (%)	[ηz]	99.824	
Sound pressure level according to Masuda, without contact analysis			
	[dB(A)]	72.9	

12 Service life, damage

Required safety for tooth root	[SFmin]	1.40	
Required safety for tooth flank	[SHmin]	1.00	
Required service life	[H]	20000.00	
Service life (calculated with required safeties):			
System service life (h)	[Hatt]	>	1000000
Tooth root service life (h)	[HFatt]	1e+06	1e+06
Tooth flank service life (h)	[HHatt]	1e+06	1e+06
Note: The entry 1e+006 h means that the Service life > 1,000,000 h.			

Damage relative to the required service life (H, 20000.0 h)

F1%	F2%	H1%	H2%
0.00	0.0000	0.0000	0.0000

13 Remarks

- Specifications with [e/l] imply: Maximum [e] and minimum value [l] for Taking all tolerances into account
Specifications with [m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness allowance are taken into account.
The maximum and minimum clearance according to the largest or smallest allowances are defined..
The calculation is performed for the operating pitch circle.
- Details of calculation method:
cy according to Method B
Kv according to Method B
KHβ and KFβ according to Method C
KHα, KFα according to Method B

_O.GroupBox.RingShaft.RingShaft_calc**1 Input data**

Coordinate system shaft: see picture W-002

1.1 Shafts**1.1.1 Shaft 'RingShaft'**

Drawing	
Initial position (mm)	0.000
Length (mm)	50.000
Speed (1/min)	2311.475
Direction of rotation:	clockwise
Material	C45 (1)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	3.129
Weight of shaft, including additional masses (kg)	3.129
Mass moment of inertia (kg*m ²)	0.035
Momentum of mass GD2 (Nm ²)	1.384

1.2 Weight force

The direction of the weight is not considered

1.3 Shaft modeling**1.4 Shear deformations**

Consider deformations due to shearing	
Shear correction factor	1.100

1.5 Rolling bearings

Contact angle of rolling bearings is considered	
Tolerance field:	Mean value

1.6 Housing

Reference temperature (°C)	20.000
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1.7 Load applications



Figure: Load applications

2 Shaft definition (RingShaft)

2.1 Outer contour

2.1.1 Cylinder (Cylinder), 0.000 mm ... 50.000 mm

Diameter (mm)	[d]	224.0000
Length (mm)	[l]	50.0000
Surface roughness (µm)	[Rz]	8.0000

2.2 Inner contour

2.2.1 Cylindrical bore (Cylindrical bore), 0.000 mm ... 50.000 mm

Diameter (mm)	[d]	200.0000
Length (mm)	[l]	50.0000
Surface roughness (µm)	[Rz]	8.0000

2.3 Forces

2.3.1 Coupling (CouplingOutput(Output))

Position on shaft (mm)	[y _{local}]	24.0000
Position in global system (mm)	[y _{global}]	24.0000
Effective diameter (mm)		230.0000
Radial force factor (-)		0.0000

Direction of the radial force (°)	0.0000	
Axial force factor (-)	0.0000	
Length of load application (mm)	10.0000	
Power (kW)	152.5766	driving (output)
Torque (Nm)	-630.3330	
Axial force (N)	0.0000	
Shearing force X (N)	0.0000	
Shearing force Z (N)	0.0000	
Bending moment X (Nm)	0.0000	
Bending moment Z (Nm)	0.0000	
Mass (kg)	0.0000	
Mass moment of inertia J_p (kg*m²)	0.0000	
Mass moment of inertia J_{xx} (kg*m²)	0.0000	
Mass moment of inertia J_{zz} (kg*m²)	0.0000	
Eccentricity (mm)	0.0000	

2.3.2 Cylindrical gear (RingGear(180.0))

Position on shaft (mm)	[y _{local}]	25.0000	
Position in global system (mm)	[y _{global}]	25.0000	
Operating pitch diameter (mm)		-279.7996	
Spur gear			
Working pressure angle at normal section (°)		22.7954	
Position of contact (°)		180.0000	
Length of load application (mm)		50.0000	
Power (kW)		50.8589	driven (input)
Torque (Nm)		210.1111	
Axial force (N)		-0.0000	
Shearing force X (N)		-631.1862	
Shearing force Z (N)		1501.8684	
Bending moment X (Nm)		0.0000	
Bending moment Z (Nm)		0.0000	

2.3.3 Cylindrical gear (RingGear(300.0))

Position on shaft (mm)	[y _{local}]	25.0000	
Position in global system (mm)	[y _{global}]	25.0000	
Operating pitch diameter (mm)		-279.7996	
Spur gear			
Working pressure angle at normal section (°)		22.7954	
Position of contact (°)		300.0000	
Length of load application (mm)		50.0000	
Power (kW)		50.8589	driven (input)
Torque (Nm)		210.1111	
Axial force (N)		-0.0000	
Shearing force X (N)		-985.0631	
Shearing force Z (N)		-1297.5575	
Bending moment X (Nm)		-0.0000	
Bending moment Z (Nm)		-0.0000	

2.3.4 Cylindrical gear (RingGear(420.0))

Position on shaft (mm)	[y _{local}]	25.0000	
Position in global system (mm)	[y _{global}]	25.0000	
Operating pitch diameter (mm)		-279.7996	
Spur gear			
Working pressure angle at normal section (°)		22.7954	
Position of contact (°)		420.0000	
Length of load application (mm)		50.0000	

KISSsoft		
Power (kW)	50.8589	driven (input)
Torque (Nm)	210.1111	
Axial force (N)	-0.0000	
Shearing force X (N)	1616.2493	
Shearing force Z (N)	-204.3109	
Bending moment X (Nm)	0.0000	
Bending moment Z (Nm)	-0.0000	
2.4 Bearing		
2.4.1 Support7 (Own input)		
Bearing position (mm)	[y _{lokal}]	24.000
Bearing position (mm)	[y _{global}]	24.000
Degrees of freedom		
X: fixedY: fixedZ: fixed		
Rx: fixedRy: freeRz: fixed		
3 Gears		

4 Results

4.1 Shafts

Maximum deflection (μm)	0.000
Position of the maximum (mm)	50.000
Mass center of gravity (mm)	25.000
Total axial load (N)	0.000
Torsion of the shaft under torque ($^\circ$)	0.000
(Difference between left and right shaft end)	

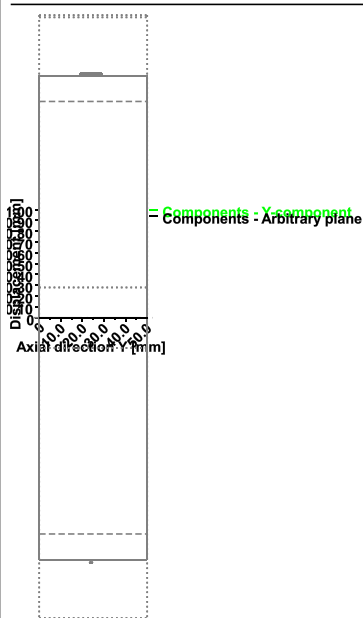
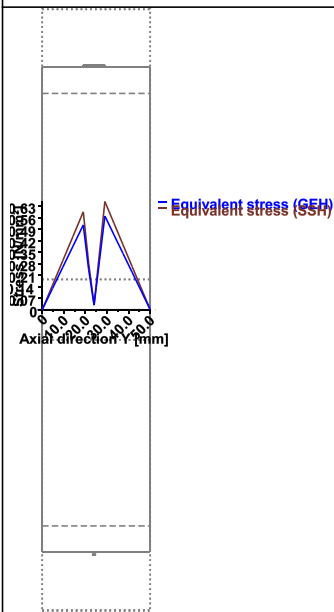


Figure: Deformation (bending etc.) (Arbitrary plane 15.69856797 127)



Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\text{sigV} = ((\text{sigB} + \text{sigZ}, D)^2 + 3 * (\text{tauT} + \text{tauS})^2)^{1/2}$

SSH(Tresca): $\text{sigV} = ((\text{sigB} - \text{sigZ}, D)^2 + 4 * (\text{tauT} + \text{tauS})^2)^{1/2}$

Figure: Equivalent stress

4.2 Bearing

4.3 'RingShaft' Bearing 'Support7'

Position (Y-coordinate)	[y]	24.00	mm		
Bearing reaction force	[F _x]	-0.000	kN		
Bearing reaction force	[F _y]	-0.000	kN		
Bearing reaction force	[F _z]	-0.000	kN		
Bearing reaction force	[F _t]	0.000	kN		
Bearing reaction moment	[M _x]	-0.00	Nm		
Bearing reaction moment	[M _y]	0.00	Nm		
Bearing reaction moment	[M _z]	0.00	Nm		
Bearing reaction moment	[M _t]	0.00	Nm		
Displacement of bearing	[u _x]	0.000	μm		
Displacement of bearing	[u _y]	0.000	μm		
Displacement of bearing	[u _z]	0.000	μm		
Displacement of bearing	[u _t]	0.000	μm		
Misalignment of bearing	[r _x]	0.000	mrاد		
Misalignment of bearing	[r _y]	-0.000	mrاد		
Misalignment of bearing	[r _z]	0.000	mrاد		
Misalignment of bearing	[r _t]	0.000	mrاد		

Note: Utilization = $(L_{req}/L_h)^{(1/k)}$

Ball bearing: k = 3, roller bearing: k = 10/3

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_O.GroupBox.Sun1Shaft.Sun1Shaft_calc**1 Input data**

Coordinate system shaft: see picture W-002

1.1 Shafts**1.1.1 Shaft 'Sun1Shaft'**

Drawing

Initial position (mm) 0.000

Length (mm) 113.000

Speed (1/min) 0.000

Direction of rotation: clockwise

Material C45 (1)

Young's modulus (N/mm²) 206000.000

Poisson's ratio nu 0.300

Density (kg/m³) 7830.000Coefficient of thermal expansion (10⁻⁶/K) 11.500

Temperature (°C) 20.000

Weight of shaft (kg) 0.625

Weight of shaft, including additional masses (kg) 0.625

Mass moment of inertia (kg*mm²) 70.360Momentum of mass GD2 (Nm²) 0.003**1.2 Weight force**

Weight towards 0.000

Weight towards 0.000

Weight towards -1.000

1.3 Shaft modeling**1.4 Shear deformations**

Consider deformations due to shearing

Shear correction factor 1.100

1.5 Rolling bearings

Contact angle of rolling bearings is considered

Tolerance field: Mean value

1.6 Housing

Reference temperature (°C) 20.000

1.7 Load applications

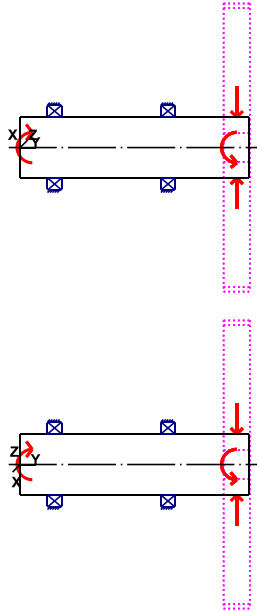


Figure: Load applications

2 Shaft definition (Sun1Shaft)

2.1 Outer contour

2.1.1 Cylinder (Cylinder),	0.000 mm ... 113.000 mm	
Diameter (mm)	[d]	30.0000
Length (mm)	[l]	113.0000
Surface roughness (μm)	[Rz]	8.0000

2.2 Forces

2.2.1 Coupling (CouplingStopLS(StopLS))

Position on shaft (mm)	[y _{local}]	6.0000
Position in global system (mm)	[y _{global}]	6.0000
Effective diameter (mm)		40.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		10.0000
Power (kW)		0.0000
Torque (Nm)		-320.3333
Axial force (N)		0.0000

Shearing force X (N)	0.0000
Shearing force Z (N)	0.0000
Bending moment X (Nm)	0.0000
Bending moment Z (Nm)	0.0000
Mass (kg)	0.0000
Mass moment of inertia Jp (kg*m²)	0.0000
Mass moment of inertia Jxx (kg*m²)	0.0000
Mass moment of inertia Jzz (kg*m²)	0.0000
Eccentricity (mm)	0.0000

2.2.2 Cylindrical gear (Sun1Gear(180.0))

Position on shaft (mm)	[y _{local}]	107.0000
Position in global system (mm)	[y _{global}]	107.0000
Operating pitch diameter (mm)		142.1384
Spur gear		
Working pressure angle at normal section (°)		22.7428
Position of contact (°)		180.0000
Length of load application (mm)		13.0000
Power (kW)		0.0000
Torque (Nm)		106.7778
Axial force (N)		-0.0000
Shearing force X (N)		629.8062
Shearing force Z (N)		1502.4476
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

2.2.3 Cylindrical gear (Sun1Gear(300.0))

Position on shaft (mm)	[y _{local}]	107.0000
Position in global system (mm)	[y _{global}]	107.0000
Operating pitch diameter (mm)		142.1384
Spur gear		
Working pressure angle at normal section (°)		22.7428
Position of contact (°)		300.0000
Length of load application (mm)		13.0000
Power (kW)		0.0000
Torque (Nm)		106.7778
Axial force (N)		-0.0000
Shearing force X (N)		-1616.0609
Shearing force Z (N)		-205.7956
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		-0.0000

2.2.4 Cylindrical gear (Sun1Gear(420.0))

Position on shaft (mm)	[y _{local}]	107.0000
Position in global system (mm)	[y _{global}]	107.0000
Operating pitch diameter (mm)		142.1384
Spur gear		
Working pressure angle at normal section (°)		22.7428
Position of contact (°)		420.0000
Length of load application (mm)		13.0000
Power (kW)		0.0000
Torque (Nm)		106.7778
Axial force (N)		-0.0000
Shearing force X (N)		986.2547
Shearing force Z (N)		-1296.6520
Bending moment X (Nm)		0.0000

Bending moment Z (Nm) -0.0000

2.3 Bearing

2.3.1 RollerBearing3 (SKF 61806)

Bearing type	Deep groove ball bearing (single row)	
Bearing position (mm)	[y _{lokal}]	17.000
Bearing position (mm)	[y _{global}]	17.000
Attachment of external ring	Set fixed bearing left	
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	42.000
Width (mm)	[b]	7.000
Corner radius (mm)	[r]	0.600
Basic static load rating (kN)	[C ₀]	2.900
Basic dynamic load rating (kN)	[C]	4.100
Fatigue load limit (kN)	[C _u]	0.146
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000
Correction factor Basic dynamic load rating	[f _c]	1.000
Correction factor Basic static load rating	[f _{c0}]	1.000

2.3.2 RollerBearing4 (SKF 61806)

Bearing type	Deep groove ball bearing (single row)	
Bearing position (mm)	[y _{lokal}]	73.000
Bearing position (mm)	[y _{global}]	73.000
Attachment of external ring	Set fixed bearing left	
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	42.000
Width (mm)	[b]	7.000
Corner radius (mm)	[r]	0.600
Basic static load rating (kN)	[C ₀]	2.900
Basic dynamic load rating (kN)	[C]	4.100
Fatigue load limit (kN)	[C _u]	0.146
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000
Correction factor Basic dynamic load rating	[f _c]	1.000
Correction factor Basic static load rating	[f _{c0}]	1.000

3 Gears

4 Results

4.1 Shafts

Maximum deflection (μm)	6.256
Position of the maximum (mm)	113.000
Mass center of gravity (mm)	56.500
Total axial load (N)	0.000
Torsion of the shaft under torque (°)	0.294
(Difference between left and right shaft end)	

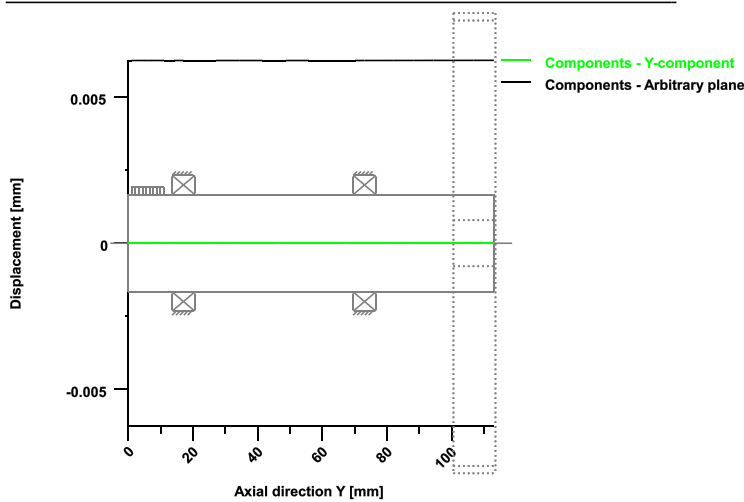
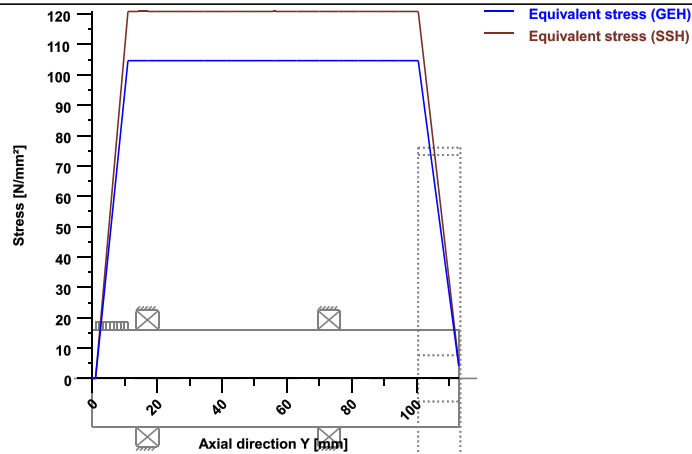


Figure: Deformation (bending etc.) (Arbitrary plane 270 127)

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Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\text{sigV} = ((\text{sigB} + \text{sigZ}, D)^2 + 3 * (\text{tauT} + \text{tauS})^2)^{1/2}$

SSH(Tresca): $\text{sigV} = ((\text{sigB} - \text{sigZ}, D)^2 + 4 * (\text{tauT} + \text{tauS})^2)^{1/2}$

Figure: Equivalent stress

4.2 Bearing

Probability of failure	[n]	10.00	%		
Axial clearance (ISO 281)	[u _A]			10.00	μm
Lubricant		ISO-VG 220			
Lubricant - service temperature	[T _B]			70.00	°C
Rolling bearings, classical calculation (contact angle considered)					

4.3 'Sun1Shaft' Rolling bearing 'RollerBearing3'

Position (Y-coordinate)	[y]	17.00	mm		
Dynamic equivalent load	[P]	0.00	kN		
Static equivalent load	[P ₀]			0.00	kN
Life modification factor for reliability	[a ₁]			1.000	

4.3.1 Results according to ISO 281

Lubricant	ISO-VG 220				
Load ratio	[C/P]	2279.016			
Operating viscosity	[v]	48.884	mm²/s		
Reference viscosity	[v ₁]	0.000	mm²/s		
Viscosity ratio	[κ]	0.000			
Basic bearing rating life	[L _{nh}]	> 1000000	h		
Static safety factor	[S ₀]	1611.99			

Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	0.002	kN
Bearing reaction force	[F _i]	0.002	kN
Displacement of bearing	[u _x]	-0.000	µm
Displacement of bearing	[u _y]	0.000	µm
Displacement of bearing	[u _z]	-6.250	µm
Displacement of bearing	[u _i]	6.250	µm
Misalignment of bearing	[r _x]	0.000	mrاد
Misalignment of bearing	[r _y]	0.559	mrاد
Misalignment of bearing	[r _z]	-0.000	mrاد
Misalignment of bearing	[r _i]	0.000	mrاد
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _r]	0.000	Nm
Sliding moment of friction	[M _{sl}]	0.000	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.000	Nm
Power loss	[P _{loss}]	0.000	W

The moment of friction is calculated according to the details in SKF Catalog 2018.

The calculation is always performed with a coefficient for additives in the lubricant µbl=0.15.

4.4 'Sun1Shaft' Rolling bearing 'RollerBearing4'

Position (Y-coordinate)	[y]	73.00	mm
Dynamic equivalent load	[P]	0.00	kN
Static equivalent load	[P ₀]	0.00	kN
Life modification factor for reliability	[a ₁]	1.000	

4.4.1 Results according to ISO 281

Lubricant	ISO-VG 220		
Load ratio	[C/P]	952.032	
Operating viscosity	[v]	48.884	mm²/s
Reference viscosity	[v ₁]	0.000	mm²/s
Viscosity ratio	[κ]	0.000	
Basic bearing rating life	[L _{nh}]	> 1000000	h
Static safety factor	[S ₀]	673.39	
Bearing reaction force	[F _x]	-0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	0.004	kN
Bearing reaction force	[F _i]	0.004	kN
Displacement of bearing	[u _x]	0.000	µm
Displacement of bearing	[u _y]	-0.000	µm
Displacement of bearing	[u _z]	-6.250	µm
Displacement of bearing	[u _i]	6.250	µm
Misalignment of bearing	[r _x]	-0.000	mrاد
Misalignment of bearing	[r _y]	3.406	mrاد
Misalignment of bearing	[r _z]	-0.000	mrاد
Misalignment of bearing	[r _i]	0.000	mrاد
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _r]	0.000	Nm
Sliding moment of friction	[M _{sl}]	0.000	Nm

Moment of friction, seals	$[M_{\text{seal}}]$	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	$[M_{\text{drag}}]$	0.000	Nm
Torque of friction	$[M_{\text{loss}}]$	0.000	Nm
Power loss	$[P_{\text{loss}}]$	0.000	W

The moment of friction is calculated according to the details in SKF Catalog 2018.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

4.4.2 Damage (%) [Lreq] (20000.000)

Bin no	B1	B2
1	2.00	2.00

Σ 2.00 2.00

4.4.3 Utilization (%) [Lreq] (20000.000)

B1	B2
27.14	27.14

Note: Utilization = $(L_{\text{req}}/L_h)^{(1/k)}$

Ball bearing: $k = 3$, roller bearing: $k = 10/3$

B1 : RollerBearing3

B2 : RollerBearing4

4.4.4 Calculation of the factors required to define reliability $R(t)$ using the Weibull distribution. t in (h)

Reliability not calculated

_O.GroupBox.Sun2Shaft.Sun2Shaft_calc

1 Input data

Coordinate system shaft: see picture W-002

1.1 Shafts

1.1.1 Shaft 'Sun2Shaft'

Drawing		
Initial position (mm)		0.000
Length (mm)		87.000
Speed (1/min)		4700.000
Direction of rotation:	clockwise	
Material	C45 (1)	
Young's modulus (N/mm ²)		206000.000
Poisson's ratio nu		0.300
Density (kg/m ³)		7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)		11.500
Temperature (°C)		20.000
Weight of shaft (kg)		0.482
Weight of shaft, including additional masses (kg)		0.482
Mass moment of inertia (kg*mm ²)		54.171
Momentum of mass GD2 (Nm ²)		0.002

1.2 Weight force

Weight towards	0.000		
Weight towards		0.000	
Weight towards			-1.000

1.3 Shaft modeling

1.4 Shear deformations

Consider deformations due to shearing

Shear correction factor	1.100
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1.5 Rolling bearings

Contact angle of rolling bearings is considered

Tolerance field:	Mean value
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1.6 Housing

Reference temperature (°C)	20.000
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1.7 Load applications

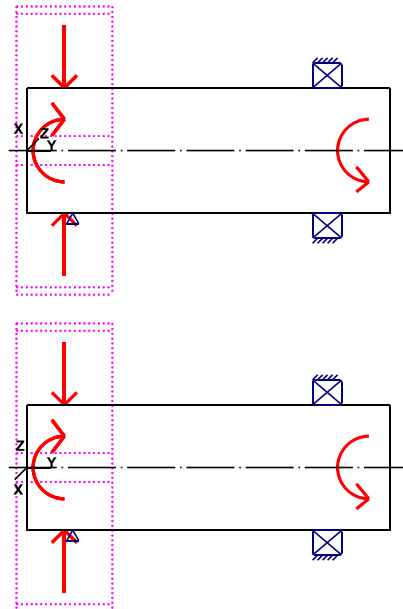


Figure: Load applications

2 Shaft definition (Sun2Shaft)

2.1 Outer contour

2.1.1 Cylinder (Cylinder), 0.000 mm ... 87.000 mm

Diameter (mm)	[d]	30.0000
Length (mm)	[l]	87.0000
Surface roughness (μm)	[Rz]	8.0000

2.2 Forces

2.2.1 Coupling (CouplingInput(Input))

Position on shaft (mm)	[y _{local}]	82.0000
Position in global system (mm)	[y _{global}]	82.0000
Effective diameter (mm)		40.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		10.0000
Power (kW)		152.5767 driven (input)
Torque (Nm)		310.0000
Axial force (N)		0.0000

Shearing force X (N)	0.0000
Shearing force Z (N)	0.0000
Bending moment X (Nm)	0.0000
Bending moment Z (Nm)	0.0000
Mass (kg)	0.0000
Mass moment of inertia Jp (kg*m²)	0.0000
Mass moment of inertia Jxx (kg*m²)	0.0000
Mass moment of inertia Jzz (kg*m²)	0.0000
Eccentricity (mm)	0.0000

2.2.2 Cylindrical gear (Sun2Gear(135.2))

Position on shaft (mm)	[y _{local}]	9.0000
Position in global system (mm)	[y _{global}]	9.0000
Operating pitch diameter (mm)		68.9990
Spur gear		
Working pressure angle at normal section (°)		23.1792
Position of contact (°)		135.1837
Length of load application (mm)		23.0000
Power (kW)		50.8589 driving (output)
Torque (Nm)		-103.3333
Axial force (N)		0.0000
Shearing force X (N)		-1201.3946
Shearing force Z (N)		-3028.6416
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		-0.0000

2.2.3 Cylindrical gear (Sun2Gear(255.2))

Position on shaft (mm)	[y _{local}]	9.0000
Position in global system (mm)	[y _{global}]	9.0000
Operating pitch diameter (mm)		68.9990
Spur gear		
Working pressure angle at normal section (°)		23.1792
Position of contact (°)		255.1837
Length of load application (mm)		23.0000
Power (kW)		50.8589 driving (output)
Torque (Nm)		-103.3333
Axial force (N)		0.0000
Shearing force X (N)		3223.5779
Shearing force Z (N)		473.8826
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		-0.0000

2.2.4 Cylindrical gear (Sun2Gear(375.2))

Position on shaft (mm)	[y _{local}]	9.0000
Position in global system (mm)	[y _{global}]	9.0000
Operating pitch diameter (mm)		68.9990
Spur gear		
Working pressure angle at normal section (°)		23.1792
Position of contact (°)		375.1837
Length of load application (mm)		23.0000
Power (kW)		50.8589 driving (output)
Torque (Nm)		-103.3333
Axial force (N)		0.0000
Shearing force X (N)		-2022.1833
Shearing force Z (N)		2554.7591
Bending moment X (Nm)		-0.0000

Bending moment Z (Nm) 0.0000

2.3 Bearing

2.3.1 RollerBearing5 (SKF 61806)

Bearing type Deep groove ball bearing (single row)

Bearing position (mm)	[y _{lokal}]	72.000
Bearing position (mm)	[y _{global}]	72.000
Attachment of external ring		Set fixed bearing right
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	42.000
Width (mm)	[b]	7.000
Corner radius (mm)	[r]	0.600
Basic static load rating (kN)	[C ₀]	2.900
Basic dynamic load rating (kN)	[C]	4.100
Fatigue load limit (kN)	[C _u]	0.146
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000
Correction factor Basic dynamic load rating	[f _c]	1.000
Correction factor Basic static load rating	[f _{c0}]	1.000

2.3.2 Support8 (Free bearing)

Bearing position (mm)	[y _{lokal}]	11.000
Bearing position (mm)	[y _{global}]	11.000
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: freeRy: freeRz: free		

3 Gears

4 Results

4.1 Shafts

Maximum deflection (µm)	7.787
Position of the maximum (mm)	87.000
Mass center of gravity (mm)	43.500
Total axial load (N)	0.000
Torsion of the shaft under torque (°)	0.205
(Difference between left and right shaft end)	

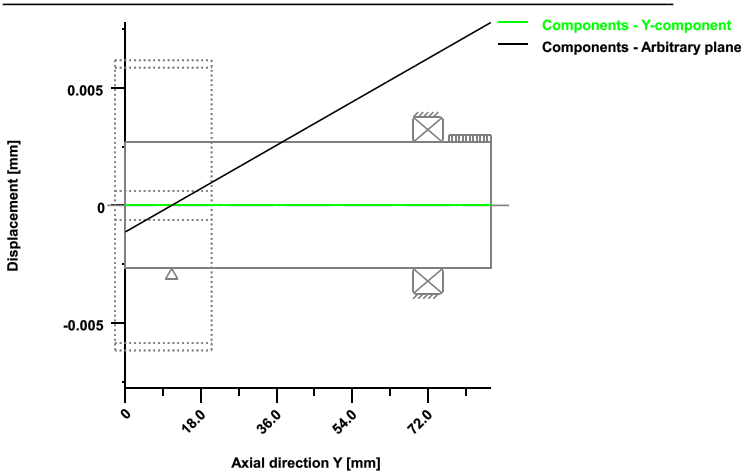
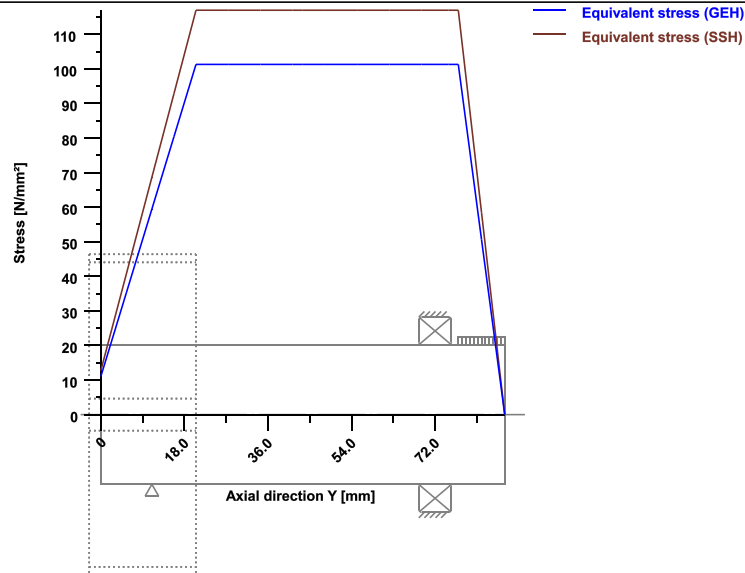


Figure: Deformation (bending etc.) (Arbitrary plane 270 127)



Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\text{sigV} = ((\text{sigB} + \text{sigZ}, D)^2 + 3 * (\text{tauT} + \text{tauS})^2)^{1/2}$

SSH(Tresca): $\text{sigV} = ((\text{sigB} - \text{sigZ}, D)^2 + 4 * (\text{tauT} + \text{tauS})^2)^{1/2}$

Figure: Equivalent stress

4.2 Bearing

Probability of failure	[n]	10.00	%		
Axial clearance (ISO 281)	[u _A]			10.00	µm
Lubricant		ISO-VG 220			
Lubricant - service temperature	[T _B]			70.00	°C
Rolling bearings, classical calculation (contact angle considered)					

4.3 'Sun2Shaft' Rolling bearing 'RollerBearing5'

Position (Y-coordinate)	[y]	72.00	mm		
Dynamic equivalent load	[P]	0.00	kN		
Static equivalent load	[P ₀]			0.00	kN
Life modification factor for reliability	[a ₁]			1.000	

4.3.1 Results according to ISO 281

Lubricant	ISO-VG 220				
Load ratio	[C/P]			1778.901	
Operating viscosity	[v]			48.884	mm²/s
Reference viscosity	[v ₁]			0.000	mm²/s
Viscosity ratio	[κ]			0.000	
Basic bearing rating life	[L _{nh}]			> 1000000	h
Static safety factor	[S ₀]			1258.25	

Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	0.002	kN
Bearing reaction force	[F _i]	0.002	kN
Displacement of bearing	[u _x]	-0.000	µm
Displacement of bearing	[u _y]	-0.000	µm
Displacement of bearing	[u _z]	-6.250	µm
Displacement of bearing	[u _i]	6.250	µm
Misalignment of bearing	[r _x]	-0.102	mrاد
Misalignment of bearing	[r _y]	3.093	mrاد
Misalignment of bearing	[r _z]	0.000	mrاد
Misalignment of bearing	[r _i]	0.102	mrاد
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _n]	0.005	Nm
Sliding moment of friction	[M _{sl}]	0.000	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 17000/1 EN:2018			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.005	Nm
Power loss	[P _{loss}]	2.428	W
The moment of friction is calculated according to the details in SKF Catalog 2018.			
The calculation is always performed with a coefficient for additives in the lubricant µbl=0.15.			

4.4 'Sun2Shaft' Bearing 'Support8'

Position (Y-coordinate)	[y]	11.00	mm
Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	0.002	kN
Bearing reaction force	[F _i]	0.002	kN
Displacement of bearing	[u _x]	0.000	µm
Displacement of bearing	[u _y]	0.000	µm
Displacement of bearing	[u _z]	0.000	µm
Displacement of bearing	[u _i]	0.000	µm
Misalignment of bearing	[r _x]	-0.102	mrاد
Misalignment of bearing	[r _y]	0.188	mrاد
Misalignment of bearing	[r _z]	0.000	mrاد
Misalignment of bearing	[r _i]	0.102	mrاد

4.4.1 Damage (%) [Lreq] (20000.000)

Bin no	B1
1	2.00

Σ	2.00

4.4.2 Utilization (%) [Lreq] (20000.000)

B1
 27.14
 Note: Utilization = (Lreq/Lh)^(1/k)
 Ball bearing: k = 3, roller bearing: k = 10/3
 B1 : RollerBearing5

KISSsoft

4.4.3 Calculation of the factors required to define reliability $R(t)$ using the Weibull distribution. t in (h)

Reliability not calculated

_O.GroupBox.SunPlanetConstraint1.SunPlanet1_c lc

1 Overview

Calculation method	DIN 3990:1987 Method B		
Drawing or article number:			
Gear 1:	Sun1Gear(SunPlanetConstraint1)		
Gear 2:	PlanetGear1(SunPlanetConstraint1)		
During calculation, the system takes into account the fact that this gear is a planet gear:			
	2		
Speed planet carrier (1/min)	[nSteg]	1532.6	
Absolute speed (1/min)	[n]	0.0	
	----- Gear 1 ----- Gear 2 -----		
Power (kW)	[P]	17.137	
Relative speed (1/min)	[n]	1532.6	3167.4
Torque (Nm)	[T]	106.8	51.7
Application factor	[KA]	1.25	
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		-	+
Working flank gear 1:	Right flank		
Gear 1 direction of rotation:	Counterclockwise		

2 Tooth geometry and material

Geometry calculation according to	ISO 21771:2007	
	----- Gear 1 ----- Gear 2 --	
Center distance (mm)	[a]	105.458
Center distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	1.5000
Normal pressure angle (°)	[on]	20.0000
Helix angle at reference circle (°)	[β]	0.0000
Number of teeth	[z]	93 45
Facewidth (mm)	[b]	13.00 50.00
Hand of gear	Spur gear	
Axial offset (mm)	[bv]	17.50
Accuracy grade	[Q-DIN 3961:1978]	6 6
Inner diameter (mm)	[di]	20.00 40.00
Inner diameter of gear rim (mm)	[dbi]	0.00 0.00

2.1 Material

Gear 1

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy $J=12\text{mm}$ <HRC28

Gear 2

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy $J=12\text{mm}$ <HRC28

		----- Gear 1 -----	Gear 2 --
Surface hardness		HRC 61	HRC 61
Fatigue strength, tooth root stress (N/mm ²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1200.00	1200.00
Yield point (N/mm ²)	[σS]	850.00	850.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

2.2 Reference profiles

Gear reference profile 1:

Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[αprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[αKP]	0.000	
	not topping		

Gear reference profile 2:

Reference profile, Own Input	1.25 / 0.38 / 1.0 ISO 53.2:1997 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[αprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[αKP]	0.000	
	not topping		

2.2.1 Information on final machining

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

Type of profile modification:

none (only running-in)

Tip relief by running in (μm)

[Ca L/R] 2.0 / 2.0 2.0 / 2.0

2.3 Lubrication

KISSsoft		
Lubrication type	Oil bath lubrication	
Type of oil	ISO-VG 220	
Lubricant base	Mineral-oil base	
Oil nominal kinematic viscosity at 40°C (mm²/s)	[v40]	220.00
Oil nominal kinematic viscosity at 100°C (mm²/s)	[v100]	17.50
Specific density at 15°C (kg/dm³)	[ρ]	0.895
Oil temperature (°C)	[TS]	70.000
2.4 Gear pair		
Overall transmission ratio	[itot]	-2.067
Gear ratio	[u]	2.067
Transverse module (mm)	[mt]	1.500
Transverse pressure angle (°)	[αt]	20.000
Working pressure angle (°)	[αwt]	22.743
	[αwt.e/l]	22.765 / 22.720
Working pressure angle at normal section (°)	[αwn]	22.743
Helix angle at operating pitch circle (°)	[βw]	0.000
Base helix angle (°)	[βb]	0.000
Reference center distance (mm)	[ad]	103.500
Pitch on reference circle (mm)	[pt]	4.712
Base pitch (mm)	[pbt]	4.428
Transverse pitch on contact-path (mm)	[pet]	4.428
Sum of profile shift coefficients	[Σx _i]	1.3926
Transverse contact ratio	[εα]	1.548
Transverse contact ratio with allowances	[εα.e/m/l]	1.559 / 1.546 / 1.533
Overlap ratio	[εβ]	0.000
Total contact ratio	[εγ]	1.548
Total contact ratio with allowances	[εγ.e/m/l]	1.559 / 1.546 / 1.533
Length of path of contact (mm)	[ga, e/l]	6.857 (6.902 / 6.789)
Length T1-A (mm)	[T1A]	30.676 (30.676 / 30.664)
Length T1-B (mm)	[T1B]	28.248 (28.202 / 28.304)
Length T1-C (mm)	[T1C]	27.475 (27.444 / 27.505)
Length T1-D (mm)	[T1D]	26.248 (26.248 / 26.236)
Length T1-E (mm)	[T1E]	23.820 (23.774 / 23.875)
Length T2-A (mm)	[T2A]	10.093 (10.048 / 10.150)
Length T2-B (mm)	[T2B]	12.522 (12.522 / 12.511)
Length T2-C (mm)	[T2C]	13.294 (13.280 / 13.309)
Length T2-D (mm)	[T2D]	14.521 (14.476 / 14.578)
Length T2-E (mm)	[T2E]	16.950 (16.950 / 16.939)
Length T1-T2 (mm)	[T1T2]	40.769 (40.724 / 40.815)
Minimal length of contact line (mm)	[Lmin]	13.000
2.5 Gear 1		
Profile shift coefficient	[x]	0.8320
Tooth thickness, arc, in module	[sn*]	2.1764
Tip alteration (mm)	[k*mn]	-0.131
Reference diameter (mm)	[d]	139.500

Base diameter (mm)	[db]	131.087
Tip diameter (mm)	[da]	144.734
(mm)	[da.e/i]	144.734 / 144.724
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010
Tip form diameter (mm)	[dFa]	144.734
(mm)	[dFa.e/i]	144.734 / 144.724
Root diameter (mm)	[df]	138.246
Generating Profile shift coefficient	[xE.e/i]	0.7450 / 0.6992
Generated root diameter with xE (mm)	[df.e/i]	137.985 / 137.848
Root form diameter (mm)	[dFf]	139.003
(mm)	[dFf.e/i]	138.751 / 138.620
Involute length (mm)	[l_dFa-l_dFf]	3.101
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	2.617
(mm)	[ha.e/i]	2.617 / 2.612
Dedendum (mm)	[hf=mn*(hfP*-x)]	0.627
(mm)	[hf.e/i]	0.757 / 0.826
Tooth height (mm)	[h]	3.244
Virtual gear no. of teeth	[zn]	93.000
Normal tooth thickness at tip circle (mm)	[san]	1.161
(mm)	[san.e/i]	1.067 / 1.011
Normal tooth thickness at tip form circle (mm)	[sFan]	1.161
(mm)	[sFan.e/i]	1.067 / 1.011
Normal space width at root circle (mm)	[efn]	0.998
(mm)	[efn.e/i]	1.004 / 1.007

2.6 Gear 2

Profile shift coefficient	[x]	0.5606
Tooth thickness, arc, in module	[sn*]	1.9789
Tip alteration (mm)	[k*mn]	-0.131
Reference diameter (mm)	[d]	67.500
Base diameter (mm)	[db]	63.429
Tip diameter (mm)	[da]	71.920
(mm)	[da.e/i]	71.920 / 71.910
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010
Tip form diameter (mm)	[dFa]	71.920
(mm)	[dFa.e/i]	71.920 / 71.910
Root diameter (mm)	[df]	65.432
Generating Profile shift coefficient	[xE.e/i]	0.4965 / 0.4599
Generated root diameter with xE (mm)	[df.e/i]	65.239 / 65.130
Root form diameter (mm)	[dFf]	66.281
(mm)	[dFf.e/i]	66.120 / 66.030
Involute length (mm)	[l_dFa-l_dFf]	3.072
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	2.210
(mm)	[ha.e/i]	2.210 / 2.205
Dedendum (mm)	[hf=mn*(hfP*-x)]	1.034
(mm)	[hf.e/i]	1.130 / 1.185
Tooth height (mm)	[h]	3.244
Virtual gear no. of teeth	[zn]	45.000
Normal tooth thickness at tip circle (mm)	[san]	1.097
(mm)	[san.e/i]	1.028 / 0.980
Normal tooth thickness at tip form circle (mm)	[sFan]	1.097
(mm)	[sFan.e/i]	1.028 / 0.980
Normal space width at root circle (mm)	[efn]	1.057
(mm)	[efn.e/i]	1.074 / 1.085

2.7 Gear specific pair data Gear pair 1, Gear 1

Operating pitch diameter (mm)	[dw]	142.138
(mm)	[dw.e/i]	142.162 /142.115
Active tip diameter (mm)	[dNa]	144.734
(mm)	[dNa.e/i]	144.734 /144.724
Theoretical tip clearance (mm)	[c]	0.375
Effective tip clearance (mm)	[c.e/i]	0.548 /0.453
Active root diameter (mm)	[dNf]	139.475
(mm)	[dNf.e/i]	139.513 /139.444
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.447 /0.347
Max. sliding velocity at tip (m/s)	[vga]	1.799
Specific sliding at the tip	[ζa]	0.320
Specific sliding at the root	[ζf]	-0.471
Mean specific sliding	[ζm]	0.320
Sliding factor on tip	[Kga]	0.138
Sliding factor on root	[Kgf]	-0.158
Roll angle at dFa (°)	[ξdFa.e/i]	26.816 /26.806
Roll angle to dNa (°)	[ξdNa.e/i]	26.816 /26.806
Roll angle to dNf (°)	[ξdNf.e/i]	20.871 /20.783
Roll angle at dFf (°)	[ξdFf.e/i]	19.876 /19.701
Diameter of single contact point B (mm)	[d-B]	142.743 (142.707 / 142.787)
Diameter of single contact point D (mm)	[d-D]	141.208 (141.208 / 141.199)
Addendum contact ratio	[ε]	0.723 (0.730 / 0.713)

2.8 Gear specific pair data Gear pair 1, Gear 2

Operating pitch diameter (mm)	[dw]	68.777
(mm)	[dw.e/i]	68.788 /68.765
Active tip diameter (mm)	[dNa]	71.920
(mm)	[dNa.e/i]	71.920 /71.910
Theoretical tip clearance (mm)	[c]	0.375
Effective tip clearance (mm)	[c.e/i]	0.596 /0.488
Active root diameter (mm)	[dNf]	66.564
(mm)	[dNf.e/i]	66.599 /66.537
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.284 /0.208
Max. sliding velocity at tip (m/s)	[vga]	1.576
Specific sliding at the tip	[ζa]	0.320
Specific sliding at the root	[ζf]	-0.471
Mean specific sliding	[ζm]	0.320
Sliding factor on tip	[Kga]	0.158
Sliding factor on root	[Kgf]	-0.138
Roll angle at dFa (°)	[ξdFa.e/i]	30.622 /30.602
Roll angle to dNa (°)	[ξdNa.e/i]	30.622 /30.602
Roll angle to dNf (°)	[ξdNf.e/i]	18.338 /18.153
Roll angle at dFf (°)	[ξdFf.e/i]	16.865 /16.575
Diameter of single contact point B (mm)	[d-B]	68.194 (68.194 / 68.186)
Diameter of single contact point D (mm)	[d-D]	69.762 (69.724 / 69.810)
Addendum contact ratio	[ε]	0.826 (0.829 / 0.820)

3 General influence factors

		----- Gear 1 ----- Gear 2 --
Nominal circum. force at pitch circle (N)	[Ft]	1530.9
Axial force (N)	[Fa]	0.0
Radial force (N)	[Fr]	557.2

Normal force (N)	[Fnorm]	1629.1
Nominal circumferential force per mm (N/mm)	[w]	117.76
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	1502.4
Axial force (N)	[Faw]	0.0
Radial force (N)	[Frw]	629.8
Circumferential speed reference circle (m/s)	[v]	11.19
Circumferential speed operating pitch circle (m/s)	[v(dw)]	11.41
Running-in value (μm)	[yp]	0.6
Running-in value (μm)	[yf]	0.4
Tolerances fpe, ff and fH β according to the tolerances in section 7		
Correction factor	[CM]	0.800
Gear blank factor	[CR]	1.000
Basic rack factor	[CBS]	0.975
Material coefficient	[E/Est]	1.000
Singular tooth stiffness (N/mm/ μm)	[c']	16.387
Meshing stiffness (N/mm/ μm)	[cy]	23.126
Reduced mass (kg/mm)	[mRed]	0.01243
Resonance speed (min-1)	[nE1]	4428
Resonance ratio (-)	[N]	0.346
Subcritical range		
Running-in value (μm)	[ya]	0.6
Bearing distance l of pinion shaft (mm)	[l]	100.000
Distance s of pinion shaft (mm)	[s]	10.000
Outside diameter of pinion shaft (mm)	[dsh]	50.000
Load according to Figure 6.8, DIN 3990-1:1987 0:6.8a, 1:6.8b, 2:6.8c, 3:6.8d, 4:6.8e	[-]	4
Coefficient K' according to Figure 6.8, DIN 3990-1:1987	[K]	-1.00
Without stiffening		
Tooth trace deviation (active) (μm)	[F β y]	5.04
from deformation of shaft (μm)	[fsh*B1]	0.05
fsh = (μm) 0.05		
B1 = 1.00		
fH β 5 = (μm) 6.00		
Tooth without flank line modification		
Position of contact pattern:	favorable	
from production tolerances (μm)	[fm α *B2]	8.00
B2 = 1.00		
Tooth trace deviation, theoretical (μm)	[F β x]	5.93
Running-in value (μm)	[y β]	0.89
Dynamic factor	[Kv]	1.226
Face load factor - flank	[KH β]	1.323
- Tooth root	[KF β]	1.234
- Scuffing	[KB β]	1.323
Transverse load factor - flank	[KH α]	1.000
- Tooth root	[KF α]	1.000
- Scuffing	[KB α]	1.000
Number of load cycles (in mio.)	[NL]	5517.391 3800.869

4 Tooth root load capacity

Calculation of Tooth form coefficients according method: B

		----- Gear 1 -----	Gear 2 --
Calculated with generating profile shift coefficient	[xE.e]	0.7450	0.4965
Tooth form factor	[YF]	1.23	1.19
Stress correction factor	[YS]	2.37	2.32
Load application angle (°)	[αFen]	22.54	22.92
Load application diameter (mm)	[d _{en}]	142.743	69.762
Bending moment arm (mm)	[hF]	1.72	1.59
Tooth thickness at root (mm)	[sFn]	3.52	3.43
Tooth root radius (mm)	[ρF]	0.57	0.62
Bending moment arm (-)	[hF/mn]	1.148	1.062
Tooth thickness at root (-)	[sFn/mn]	2.345	2.286
Tooth root radius (-)	[ρF/mn]	0.382	0.411
Calculation cross section diameter (mm)	[d _{sFn}]	138.528	65.771
Tangents on calculation cross section (°)	[α _{sFn}]	30.000	30.000
Notch parameter	[q _s]	3.068	2.782
Contact ratio factor	[Yε]		1.000
Helix angle factor	[Yβ]		1.000
Effective facewidth (mm)	[beff]	13.00	15.50
Nominal stress at tooth root (N/mm²)	[σF0]	228.83	182.43
Tooth root stress (N/mm²)	[σF]	432.75	344.99
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[Y _{drelT}]	1.005	1.003
Surface factor	[Y _{RrelT}]	0.957	0.957
Size factor, tooth root	[Y _X]	1.000	1.000
Life factor	[Y _{NT}]	1.000	1.000
Y _{drelT} *Y _{RrelT} *Y _X *Y _{NT}		0.962	0.959
Alternating bending factor, mean stress influence coefficient			
	[Y _M]	1.000	1.000
Stress correction factor	[Y _{ST}]		2.00
YST*σ _{Flim} (N/mm²)	[σ _{FE}]	860.00	860.00
Permissible tooth root stress σ _{FG} /SF _{min} (N/mm²)	[σ _{FP}]	590.72	589.24
Limit strength tooth root (N/mm²)	[σ _{FG}]	827.01	824.93
Required safety	[SF _{min}]	1.40	1.40
Safety for tooth root stress	[SF=σ _{FG} /σ _F]	1.91	2.39
Transmittable power (W)	[WR _{ating}]	23393.16	29269.79

5 Flank safety

		----- Gear 1 -----	Gear 2 --
Zone factor	[ZH]		2.324
Elasticity factor (√N/mm²)	[ZE]		189.812
Contact ratio factor	[Zε]		0.904
Helix angle factor	[Zβ]		1.000
Effective facewidth (mm)	[beff]		13.00
Nominal contact stress (N/mm²)	[σH0]		641.73
Contact stress at operating pitch circle (N/mm²)	[σHw]		913.75
Single tooth contact factor	[ZB,ZD]	1.00	1.02
Contact stress (N/mm²)	[σHB, σHD]	913.75	928.56
Lubrication factor for NL	[ZL]	1.020	1.020
Speed factor for NL	[ZV]	1.003	1.003
Roughness factor for NL	[ZR]	0.964	0.964
Material hardening factor for NL	[ZW]	1.000	1.000
Life factor	[ZNT]	1.000	1.000
	[ZL*ZV*ZR*ZNT]	0.987	0.987

Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress, $\sigma_{HG}/\sigma_{Hmin}$ (N/mm ²)	[σ_{HP}]	1480.34	1480.34
Pitting stress limit (N/mm ²)	[σ_{HG}]	1480.34	1480.34
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress on operating pitch circle			
	[SHw]	1.62	1.62
Safety against pressure, σ_{HG}/σ_{HBD} Single contact	[SHBD]	1.62	1.59
Safety regarding transmittable torque	[$(SHBD)^2$]	2.62	2.54
Transmittable power (W)	[WRating]	44978.68	43555.82

6 Micropitting

Calculation method according to ISO/TS 6336-22:2018
 Calculation has not been carried out, lubricant: Load stage micropitting test not known

7 Scuffing load capacity

Calculation method according to	DIN 3990:1987		
Helical load factor for scuffing	[KBy]	1.000	
Lubrication coefficient for lubrication type	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Relative structural factor, scuffing	[XWrelT]	1.000	
Thermal contact factor (N/mm/s ^{0.5} /K)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	8.98	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	13.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	238.749	
Angle factor	[X $\alpha\beta$]	1.017	
ε_1 : 0.723 , ε_2 : 0.826			

7.1 Flash temperature-criteria

Tooth mass temperature (°C)	[θ_{MB}]	74.42	
$\theta_{MB} = \theta_{oil} + XS \cdot 0.47 \cdot \theta_{flamax}$			
Maximum flash temperature (°C)	[θ_{flamax}]	9.41	
Scuffing temperature (°C)	[θ_S]	408.58	
Γ coordinates (point of highest temperature)	[Γ]	-0.241	
[$\Gamma.A$]= 0.117 [$\Gamma.E$]= -0.133			
Highest contact temp. (°C)	[θ_B]	83.83	
Flash factor (°K*N ^{0.75} *s ^{0.5} *m ^{0.5} mm)	[XM]	50.058	
Geometry factor	[XB]	0.144	
Load sharing factor	[X Γ]	0.333	
Dynamic viscosity (mPa*s)	[η_M]	35.40	(74.4 °C)
Coefficient of friction	[μ_m]	0.061	
Required safety	[SBmin]	2.000	
Margin of safety for scuffing, flash temperature	[SB]	24.466	

7.2 Integral temperature-criteria

Tooth mass temperature (°C)	[θ_{MC}]	74.58
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$\theta_{MC} = \theta_{oil} + X_S \cdot 0.70 \cdot \theta_{flint}$		
Integral scuffing temperature (°C)	[θ_{Sint}]	408.58
Flash factor ($^{\circ}K \cdot N^{\wedge-.75} \cdot s^{\wedge-.5} \cdot m^{\wedge-.5} \cdot mm$)	[XM]	50.058
Contact ratio factor	[X_{ε}]	0.260
Dynamic viscosity (mPa*s)	[η_{Oil}]	41.90 (70.0 °C)
Mean coefficient of friction	[μ_m]	0.055
Geometry factor	[XBE]	0.144
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.014
Mean flash temperature (°C)	[θ_{flint}]	6.54
Integral tooth flank temperature (°C)	[θ_{int}]	84.40
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	4.841
Safety referring to transmittable torque	[SSL]	23.521

8 Measurements for tooth thickness

		----- Gear 1 -----	Gear 2 --
Tooth thickness tolerance		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.095 / -0.145	-0.070 / -0.110
Number of teeth spanned	[k]	12.000	6.000
Base tangent length (no backlash) (mm)	[Wk]	53.732	25.876
Base tangent length with allowance (mm)	[Wk.e/i]	53.642 / 53.595	25.810 / 25.772
(mm)	[$\Delta Wk.e/i$]	-0.089 / -0.136	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	141.629	68.472
Theoretical diameter of ball/pin (mm)	[DM]	2.681	2.737
Effective diameter of ball/pin (mm)	[DMeff]	2.750	2.750
Radial single-ball measurement backlash free (mm)	[MrK]	72.985	36.550
Radial single-ball measurement (mm)	[MrK.e/i]	72.873 / 72.814	36.473 / 36.429
Diameter of measuring circle (mm)	[dMMr.m]	141.864	69.023
Diametral measurement over two balls without clearance (mm)	[MdK]	145.949	73.056
Diametral two ball measure (mm)	[MdK.e/i]	145.726 / 145.608	72.904 / 72.816
Diametral measurement over pins without clearance (mm)	[MdR]	145.949	73.056
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	145.726 / 145.608	72.904 / 72.816
Dimensions over 3 pins without clearance (mm)	[Md3R]	145.928	73.014
Measurement over 3 pins with allowance (mm)	[Md3R.e/i]	145.706 / 145.588	72.861 / 72.773
Chordal tooth thickness (no backlash) (mm)	[sc]	3.264	2.967
Normal chordal tooth thickness with allowance (mm)	[sc.e/i]	3.170 / 3.121	2.898 / 2.859
Reference chordal height from da.m (mm)	[ha]	2.634	2.240
Tooth thickness, arc (mm)	[sn]	3.265	2.968
(mm)	[sn.e/i]	3.170 / 3.120	2.898 / 2.858
Backlash free center distance (mm)	[aControl.e/i]	105.256 / 105.145	
Backlash free center distance, allowances (mm)	[ita]	-0.202 / -0.313	
dNf.i with aControl (mm)	[dNf0.i]	138.927	66.088
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.088	-0.016
Tip clearance (mm)	[c0.i(aControl)]	0.158	0.193
Center distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.015 / -0.015	
Radial backlash (mm)	[jrw.e/i]	0.330 / 0.184	
Circumferential backlash (transverse section) (mm)	[jtw.e/i]	0.274 / 0.153	
Normal backlash (mm)	[jn.e/i]	0.252 / 0.143	
Torsional angle on input with output fixed:			

Total torsional angle (°)	[j.tSys]	0.4573/0.2557
---------------------------	----------	---------------

9 Toothing tolerances

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

According to DIN 3961:1978

Accuracy grade	[Q-DIN3961]	6	6
Profile form deviation (µm)	[ff]	6.00	6.00
Profile slope deviation (µm)	[fHa]	5.00	5.00
Total profile deviation (µm)	[Ff]	8.00	8.00
Helix form deviation (µm)	[fbf]	4.00	8.00
Helix slope deviation (µm)	[fHb]	8.00	10.00
Total helix deviation (µm)	[Fb]	9.00	13.00
Normal base pitch deviation (µm)	[fpe]	8.00	7.00
Single pitch deviation (µm)	[fp]	8.00	7.00
Adjacent pitch difference (µm)	[fu]	10.00	9.00
Total cumulative pitch deviation (µm)	[Fp]	29.00	25.00
Sector pitch deviation over z/8 pitches (µm)	[Fpz/8]	18.00	15.00
Runout (µm)	[Fr]	19.00	16.00
Tooth Thickness Variation (µm)	[Rs]	11.00	10.00
Single flank composite, total (µm)	[Fi]	30.00	26.00
Single flank composite, tooth-to-tooth (µm)	[fi]	11.00	11.00
Radial composite, total (µm)	[Fi"]	24.00	20.00
Radial composite, tooth-to-tooth (µm)	[fi"]	10.00	8.00

According DIN 58405:1972 (Precision Mechanics)

Tooth-to-tooth composite error (µm)	[fi"]	9.00	8.00
Composite error (µm)	[Fi"]	25.00	22.00
Axis alignment error (µm)	[fp]	17.93	17.93
Flank direction error (µm)	[iβ]	5.00	10.50
Runout (µm)	[Trk, Fr]	28.00	24.00

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

6

Maximum value for deviation error of axis (µm)	[fΣβ]	42.31	(Fβ=11.00)
Maximum value for inclination error of axes (µm)	[fΣδ]	84.62	

10 Modifying and defining the tooth form

10.1 Data for the tooth form calculation

Data not available.

Please run the calculation in the "Tooth form" tab and open the main report again.

11 Supplementary data

Maximal possible center distance (eps_a=1.0)	[aMAX]	106.420	
Mass (kg)	[m]	1.568	0.958
Total mass (kg)	[mGes]	2.527	
calculation without consideration of the exact tooth shape			
Single gears, (da+df)/2...di (kg*m²)	[J]	0.00400	0.00076
Torsional stiffness at driving gear with fixed driven gear:			
Torsional stiffness (MNm/rad)	[cr]	0.297	
Torsion when subjected to nominal torque (°)	[δcr]	0.010	

KISSsoft

Mean coefficient of friction (as defined in Niemann)	[μ _m]	0.052	
Wear sliding coef. by Niemann	[ζ _w]	0.496	
Loss factor	[HV]	0.068	
Gear power loss (W)	[PVZ]	60.495	
Meshing efficiency (%)	[η _z]	99.647	
Sound pressure level according to Masuda, without contact analysis	[dB(A)]	68.7	

12 Service life, damage

Required safety for tooth root	[SF _{min}]	1.40	
Required safety for tooth flank	[SH _{min}]	1.00	
Required service life	[H]	20000.00	

Service life (calculated with required safeties):

System service life (h)	[H _{att}]	> 1000000	
-------------------------	---------------------	-----------	--

Tooth root service life (h)	[HF _{att}]	1e+06	1e+06
Tooth flank service life (h)	[HH _{att}]	1e+06	1e+06

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

Damage relative to the required service life (H, 20000.0 h)

F1%	F2%	H1%	H2%
0.00	0.0000	0.0000	0.0000

13 Remarks

Specifications with [e/i] imply: Maximum [e] and minimum value [i] for

Taking all tolerances into account

Specifications with [m] imply: Mean value within tolerance

For the backlash tolerance, the center distance tolerances and the tooth thickness allowance are taken into account.

The maximum and minimum clearance according to the largest or smallest allowances are defined..

The calculation is performed for the operating pitch circle.

Details of calculation method:

cy according to Method B

Kv according to Method B

KHβ and KFβ according to Method C

KHa, KFα according to Method B

_O.GroupBox.SunPlanetConstraint2.Sun2Planet2_c alc

1 Overview

Calculation method	DIN 3990:1987 Method B		
Drawing or article number:			
Gear 1:	Sun2Gear(SunPlanetConstraint2)		
Gear 2:	PlanetGear2(SunPlanetConstraint2)		
During calculation, the system takes into account the fact that this gear is a planet gear:			
	2		
Speed planet carrier (1/min)	[nSteg]		1532.6
Absolute speed (1/min)	[n]		4700.0
	----- Gear 1 ----- Gear 2 -----		
Power (kW)	[P]		34.274
Relative speed (1/min)	[n]	3167.4	2741.0
Torque (Nm)	[T]	103.3	119.4
Application factor	[KA]		1.25
Required service life (h)	[H]		20000.00
Gear driving (+) / driven (-)		+	-
Working flank gear 1:	Right flank		
Gear 1 direction of rotation:	Clockwise		

2 Tooth geometry and material

Geometry calculation according to	ISO 21771:2007	
	----- Gear 1 ----- Gear 2 --	
Center distance (mm)	[a]	74.366
Center distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	1.5000
Normal pressure angle (°)	[on]	20.0000
Helix angle at reference circle (°)	[β]	0.0000
Number of teeth	[z]	45 52
Facewidth (mm)	[b]	23.00 23.00
Hand of gear	Spur gear	
Accuracy grade	[Q-DIN 3961:1978]	6 6
Inner diameter (mm)	[di]	20.00 40.00
Inner diameter of gear rim (mm)	[dbi]	0.00 0.00

2.1 Material

Gear 1
18CrNiMo7-6, Case-carburized steel, case-hardened

KISSsoft

ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

Gear 2

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

		----- Gear 1 -----	Gear 2 --
Surface hardness		HRC 61	HRC 61
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm²)	[σB]	1200.00	1200.00
Yield point (N/mm²)	[σS]	850.00	850.00
Young's modulus (N/mm²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

2.2 Reference profiles

Gear reference profile 1:

Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[αprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[αKP]	0.000	
	not topping		

Gear reference profile 2:

Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.472)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[αprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[αKP]	0.000	
	not topping		

2.2.1 Information on final machining

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

Type of profile modification:

none (only running-in)

Tip relief by running in (μm)

[Ca L/R]

2.0 / 2.0

2.0 / 2.0

2.3 Lubrication

Lubrication type

Oil bath lubrication

Type of oil	ISO-VG 220	
Lubricant base	Mineral-oil base	
Oil nominal kinematic viscosity at 40°C (mm²/s)	[v40]	220.00
Oil nominal kinematic viscosity at 100°C (mm²/s)	[v100]	17.50
Specific density at 15°C (kg/dm³)	[ρ]	0.895
Oil temperature (°C)	[TS]	70.000

2.4 Gear pair		
Overall transmission ratio	[itot]	-1.156
Gear ratio	[u]	1.156
Transverse module (mm)	[mt]	1.500
Transverse pressure angle (°)	[αt]	20.000
Working pressure angle (°)	[αwt]	23.179
	[αwt.e/i]	23.206 / 23.152
Working pressure angle at normal section (°)	[αwn]	23.179
Helix angle at operating pitch circle (°)	[βw]	0.000
Base helix angle (°)	[βb]	0.000
Reference center distance (mm)	[ad]	72.750
Pitch on reference circle (mm)	[pt]	4.712
Base pitch (mm)	[pbt]	4.428
Transverse pitch on contact-path (mm)	[pet]	4.428
Sum of profile shift coefficients	[Σxi]	1.1611
Transverse contact ratio	[εα]	1.508
Transverse contact ratio with allowances	[εα.e/m/i]	1.516 / 1.505 / 1.494
Overlap ratio	[εβ]	0.000
Total contact ratio	[εγ]	1.508
Total contact ratio with allowances	[εγ.e/m/i]	1.516 / 1.505 / 1.494
Length of path of contact (mm)	[ga, e/i]	6.677 (6.715 / 6.617)
Length T1-A (mm)	[T1A]	10.296 (10.258 / 10.345)
Length T1-B (mm)	[T1B]	12.545 (12.545 / 12.534)
Length T1-C (mm)	[T1C]	13.579 (13.562 / 13.597)
Length T1-D (mm)	[T1D]	14.724 (14.686 / 14.773)
Length T1-E (mm)	[T1E]	16.973 (16.973 / 16.962)
Length T2-A (mm)	[T2A]	18.975 (18.975 / 18.964)
Length T2-B (mm)	[T2B]	16.726 (16.688 / 16.775)
Length T2-C (mm)	[T2C]	15.692 (15.671 / 15.712)
Length T2-D (mm)	[T2D]	14.547 (14.547 / 14.536)
Length T2-E (mm)	[T2E]	12.298 (12.260 / 12.347)
Length T1-T2 (mm)	[T1T2]	29.271 (29.233 / 29.309)
Minimal length of contact line (mm)	[Lmin]	23.000

2.5 Gear 1		
Profile shift coefficient	[x]	0.5645
Tooth thickness, arc, in module	[sn*]	1.9817
Tip alteration (mm)	[k*mn]	-0.126
Reference diameter (mm)	[d]	67.500
Base diameter (mm)	[db]	63.429

Tip diameter (mm)	[da]	71.941
(mm)	[da.e/i]	71.941 / 71.931
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010
Tip form diameter (mm)	[dFa]	71.941
(mm)	[dFa.e/i]	71.941 / 71.931
Root diameter (mm)	[df]	65.443
Generating Profile shift coefficient	[xE.e/i]	0.5004 / 0.4637
Generated root diameter with xE (mm)	[df.e/i]	65.251 / 65.141
Root form diameter (mm)	[dFf]	66.291
(mm)	[dFf.e/i]	66.130 / 66.039
Involute length (mm)	[l_dFa-l_dFf]	3.079
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	2.221
(mm)	[ha.e/i]	2.221 / 2.216
Dedendum (mm)	[hf=mn*(hfP*-x)]	1.028
(mm)	[hf.e/i]	1.124 / 1.179
Tooth height (mm)	[h]	3.249
Virtual gear no. of teeth	[zn]	45.000
Normal tooth thickness at tip circle (mm)	[san]	1.090
(mm)	[san.e/i]	1.021 / 0.973
Normal tooth thickness at tip form circle (mm)	[sFan]	1.090
(mm)	[sFan.e/i]	1.021 / 0.973
Normal space width at root circle (mm)	[efn]	1.056
(mm)	[efn.e/i]	1.073 / 1.083

2.6 Gear 2

Profile shift coefficient	[x]	0.5966
Tooth thickness, arc, in module	[sn*]	2.0051
Tip alteration (mm)	[k*mn]	-0.126
Reference diameter (mm)	[d]	78.000
Base diameter (mm)	[db]	73.296
Tip diameter (mm)	[da]	82.538
(mm)	[da.e/i]	82.538 / 82.528
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010
Tip form diameter (mm)	[dFa]	82.538
(mm)	[dFa.e/i]	82.538 / 82.528
Root diameter (mm)	[df]	76.040
Generating Profile shift coefficient	[xE.e/i]	0.5325 / 0.4958
Generated root diameter with xE (mm)	[df.e/i]	75.847 / 75.737
Root form diameter (mm)	[dFf]	76.862
(mm)	[dFf.e/i]	76.694 / 76.600
Involute length (mm)	[l_dFa-l_dFf]	3.086
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	2.269
(mm)	[ha.e/i]	2.269 / 2.264
Dedendum (mm)	[hf=mn*(hfP*-x)]	0.980
(mm)	[hf.e/i]	1.076 / 1.131
Tooth height (mm)	[h]	3.249
Virtual gear no. of teeth	[zn]	52.000
Normal tooth thickness at tip circle (mm)	[san]	1.111
(mm)	[san.e/i]	1.042 / 0.994
Normal tooth thickness at tip form circle (mm)	[sFan]	1.111
(mm)	[sFan.e/i]	1.042 / 0.994
Normal space width at root circle (mm)	[efn]	1.039
(mm)	[efn.e/i]	1.053 / 1.061

2.7 Gear specific pair data Gear pair 1, Gear 1

Operating pitch diameter (mm)	[dw]	68.999
(mm)	[dw.e/i]	69.013 / 68.985
Active tip diameter (mm)	[dNa]	71.941
(mm)	[dNa.e/i]	71.941 / 71.931
Theoretical tip clearance (mm)	[c]	0.375
Effective tip clearance (mm)	[c.e/i]	0.546 / 0.456
Active root diameter (mm)	[dNf]	66.688
(mm)	[dNf.e/i]	66.718 / 66.665
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.340 / 0.268
Max. sliding velocity at tip (m/s)	[vga]	2.100
Specific sliding at the tip	[ζa]	0.373
Specific sliding at the root	[ζf]	-0.595
Mean specific sliding	[ζm]	0.373
Sliding factor on tip	[Kga]	0.183
Sliding factor on root	[Kgf]	-0.178
Roll angle at dFa (°)	[ξdFa.e/i]	30.663 / 30.644
Roll angle to dNa (°)	[ξdNa.e/i]	30.663 / 30.644
Roll angle to dNf (°)	[ξdNf.e/i]	18.690 / 18.532
Roll angle at dFf (°)	[ξdFf.e/i]	16.896 / 16.605
Diameter of single contact point B (mm)	[d-B]	68.211 (68.211 / 68.203)
Diameter of single contact point D (mm)	[d-D]	69.932 (69.900 / 69.973)
Addendum contact ratio	[ε]	0.766 (0.770 / 0.760)

2.8 Gear specific pair data Gear pair 1, Gear 2

Operating pitch diameter (mm)	[dw]	79.732
(mm)	[dw.e/i]	79.748 / 79.716
Active tip diameter (mm)	[dNa]	82.538
(mm)	[dNa.e/i]	82.538 / 82.528
Theoretical tip clearance (mm)	[c]	0.375
Effective tip clearance (mm)	[c.e/i]	0.546 / 0.456
Active root diameter (mm)	[dNf]	77.313
(mm)	[dNf.e/i]	77.344 / 77.289
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.372 / 0.297
Max. sliding velocity at tip (m/s)	[vga]	2.031
Specific sliding at the tip	[ζa]	0.373
Specific sliding at the root	[ζf]	-0.595
Mean specific sliding	[ζm]	0.373
Sliding factor on tip	[Kga]	0.178
Sliding factor on root	[Kgf]	-0.183
Roll angle at dFa (°)	[ξdFa.e/i]	29.665 / 29.648
Roll angle to dNa (°)	[ξdNa.e/i]	29.665 / 29.648
Roll angle to dNf (°)	[ξdNf.e/i]	19.303 / 19.167
Roll angle at dFf (°)	[ξdFf.e/i]	17.648 / 17.397
Diameter of single contact point B (mm)	[d-B]	80.569 (80.537 / 80.610)
Diameter of single contact point D (mm)	[d-D]	78.859 (78.859 / 78.851)
Addendum contact ratio	[ε]	0.741 (0.746 / 0.734)

3 General influence factors

		----- Gear 1 ----- Gear 2 --
Nominal circum. force at pitch circle (N)	[Ft]	3061.7
Axial force (N)	[Fa]	0.0
Radial force (N)	[Fr]	1114.4
Normal force (N)	[Fnorm]	3258.2

KISSsoft			
Nominal circumferential force per mm (N/mm)	[w]	133.12	
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]	2995.2	
Axial force (N)	[Faw]	0.0	
Radial force (N)	[Frw]	1282.5	
Circumferential speed reference circle (m/s)	[v]	11.19	
Circumferential speed operating pitch circle (m/s)	[v(dw)]	11.44	
Running-in value (μm)	[yp]	0.5	
Running-in value (μm)	[yf]	0.4	
Tolerances fpe, ff and fH β according to the tolerances in section 7			
Correction factor	[CM]	0.800	
Gear blank factor	[CR]	1.000	
Basic rack factor	[CBS]	0.975	
Material coefficient	[E/Est]	1.000	
Singular tooth stiffness (N/mm/ μm)	[c]	15.916	
Meshing stiffness (N/mm/ μm)	[cv]	21.978	
Reduced mass (kg/mm)	[mRed]	0.00939	
Resonance speed (min-1)	[nE1]	10265	
Resonance ratio (-)	[N]	0.309	
Subcritical range			
Running-in value (μm)	[ya]	0.5	
Bearing distance l of pinion shaft (mm)	[l]	46.000	
Distance s of pinion shaft (mm)	[s]	4.600	
Outside diameter of pinion shaft (mm)	[dsh]	23.000	
Load according to Figure 6.8, DIN 3990-1:1987	[-]	4	
0:6.8a, 1:6.8b, 2:6.8c, 3:6.8d, 4:6.8e			
Coefficient K' according to Figure 6.8, DIN 3990-1:1987	[K]	-1.00	
Without stiffening			
Tooth trace deviation (active) (μm)	[F β y]	3.83	
from deformation of shaft (μm)	[fsh*B1]	1.59	
fsh = (μm) 1.59			
B1 = 1.00			
fH β 5 = (μm) 6.50			
Tooth without flank line modification			
Position of contact pattern:	favorable		
from production tolerances (μm)	[fm α *B2]	9.00	
B2 = 1.00			
Tooth trace deviation, theoretical (μm)	[F β x]	4.50	
Running-in value (μm)	[y β]	0.67	
Dynamic factor	[Kv]	1.174	
Face load factor - flank	[KH β]	1.215	
- Tooth root	[KF β]	1.182	
- Scuffing	[KB β]	1.215	
Transverse load factor - flank	[KH α]	1.000	
- Tooth root	[KF α]	1.000	
- Scuffing	[KB α]	1.000	
Number of load cycles (in mio.)	[NL]	11402.609	3289.214
4 Tooth root load capacity			
Calculation of Tooth form coefficients according method: B			

		----- Gear 1 -----	Gear 2 --
Calculated with generating profile shift coefficient	[xE.e]	0.5004	0.5325
Tooth form factor	[YF]	1.26	1.26
Stress correction factor	[YS]	2.27	2.29
Load application angle (°)	[αFen]	23.28	23.14
Load application diameter (mm)	[d _{en}]	69.932	80.569
Bending moment arm (mm)	[hF]	1.68	1.70
Tooth thickness at root (mm)	[sFn]	3.43	3.45
Tooth root radius (mm)	[ρF]	0.62	0.61
Bending moment arm (-)	[hF/mn]	1.120	1.135
Tooth thickness at root (-)	[sFn/mn]	2.287	2.302
Tooth root radius (-)	[ρF/mn]	0.410	0.403
Calculation cross section diameter (mm)	[d _{sFn}]	65.782	76.382
Tangents on calculation cross section (°)	[α _{sFn}]	30.000	30.000
Notch parameter	[q _s]	2.787	2.853
Contact ratio factor	[Yε]		1.000
Helix angle factor	[Yβ]		1.000
Effective facewidth (mm)	[beff]	23.00	23.00
Nominal stress at tooth root (N/mm²)	[σF0]	253.32	255.50
Tooth root stress (N/mm²)	[σF]	439.32	443.11
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdrelT]	1.003	1.003
Surface factor	[YRrelT]	0.957	0.957
Size factor, tooth root	[YX]	1.000	1.000
Life factor	[YNT]	1.000	1.000
$Y_{drelT} \cdot Y_{RrelT} \cdot Y_X \cdot Y_{NT}$		0.959	0.960
Alternating bending factor, mean stress influence coefficient			
	[YM]	1.000	1.000
Stress correction factor	[YST]		2.00
$YST \cdot \sigma_{Flim}$ (N/mm²)	[σFE]	860.00	860.00
Permissible tooth root stress σ_{FG}/SF_{min} (N/mm²)	[σFP]	589.27	589.62
Limit strength tooth root (N/mm²)	[σFG]	824.97	825.46
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=σFG/σF]	1.88	1.86
Transmittable power (kW)	[kWRating]	45.97	45.61

5 Flank safety

		----- Gear 1 -----	Gear 2 --
Zone factor	[ZH]		2.300
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zε]		0.911
Helix angle factor	[Zβ]		1.000
Effective facewidth (mm)	[beff]		23.00
Nominal contact stress (N/mm²)	[σH0]		763.18
Contact stress at operating pitch circle (N/mm²)	[σHw]		1019.23
Single tooth contact factor	[ZB,ZD]	1.01	1.00
Contact stress (N/mm²)	[σHB, σHD]	1027.11	1019.23
Lubrication factor for NL	[ZL]	1.020	1.020
Speed factor for NL	[ZV]	1.003	1.003
Roughness factor for NL	[ZR]	0.956	0.956
Material hardening factor for NL	[ZW]	1.000	1.000
Life factor	[ZNT]	1.000	1.000
	[ZL*ZV*ZR*ZNT]	0.978	0.978
Limited pitting is permitted:	No		

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Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress, $\sigma_{HG}/\sigma_{Hmin}$ (N/mm²)	[σ_{HP}]	1466.62	1466.62
Pitting stress limit (N/mm²)	[σ_{HG}]	1466.62	1466.62
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress on operating pitch circle			
	[SHw]	1.44	1.44
Safety against pressure, σ_{HG}/σ_{HBD} Single contact	[SHBD]	1.43	1.44
Safety regarding transmittable torque	[$(SHBD)^2$]	2.04	2.07
Transmittable power (kW)	[kWRating]	69.88	70.97

6 Micropitting

Calculation method according to

ISO/TS 6336-22:2018

Calculation has not been carried out, lubricant: Load stage micropitting test not known

7 Scuffing load capacity

Calculation method according to

DIN 3990:1987

Helical load factor for scuffing	[KBy]	1.000	
Lubrication coefficient for lubrication type	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Relative structural factor, scuffing	[XWreIT]	1.000	
Thermal contact factor (N/mm/s ^{0.5} /K)	[BM]	13.780	13.780
Relevant tip relief (µm)	[Ca]	2.00	2.00
Optimal tip relief (µm)	[Ceff]	10.45	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	23.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	237.425	
Angle factor	[Xαβ]	1.024	
ε1: 0.766 , ε2: 0.741			

7.1 Flash temperature-criteria

Tooth mass temperature (°C)	[θ_{MB}]	75.98	
$\theta_{MB} = \theta_{oil} + XS \cdot 0.47 \cdot \theta_{flamax}$			
Maximum flash temperature (°C)	[θ_{flamax}]	12.72	
Scuffing temperature (°C)	[θ_S]	408.58	
Γ coordinates (point of highest temperature)	[Γ]	-0.204	
[Γ.A]= -0.242 [Γ.E]= 0.250			
Highest contact temp. (°C)	[θ_B]	88.70	
Flash factor (°K*N ^{-0.75} *s ^{-0.5} *m ^{-0.5} *mm)	[XM]	50.058	
Geometry factor	[XB]	0.141	
Load sharing factor	[XΓ]	0.409	
Dynamic viscosity (mPa*s)	[ηM]	33.14	(76.0 °C)
Coefficient of friction	[μm]	0.062	
Required safety	[SBmin]	2.000	
Margin of safety for scuffing, flash temperature	[SB]	18.094	

7.2 Integral temperature-criteria

Tooth mass temperature (°C)	[θ_{MC}]	76.62	
$\theta_{MC} = \theta_{oil} + XS \cdot 0.70 \cdot \theta_{flaint}$			

Integral scuffing temperature (°C)	[θSint]	408.58
Flash factor (°K·N ⁻¹ ·75·s ⁻¹ ·m ⁻¹ ·5·mm)	[XM]	50.058
Contact ratio factor	[Xε]	0.278
Dynamic viscosity (mPa·s)	[ηOil]	41.90 (70.0 °C)
Mean coefficient of friction	[μ _m]	0.058
Geometry factor	[XBE]	0.169
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.011
Mean flash temperature (°C)	[θflaint]	9.46
Integral tooth flank temperature (°C)	[θint]	90.81
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	4.499
Safety referring to transmittable torque	[SSL]	16.270

8 Measurements for tooth thickness

		----- Gear 1 -----	Gear 2 --
Tooth thickness tolerance		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070 / -0.110	-0.070 / -0.110
Number of teeth spanned	[k]	6.000	7.000
Base tangent length (no backlash) (mm)	[Wk]	25.880	30.488
Base tangent length with allowance (mm)	[Wk.e/i]	25.814 / 25.776	30.422 / 30.384
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	68.474	79.352
Theoretical diameter of ball/pin (mm)	[DM]	2.739	2.719
Effective diameter of ball/pin (mm)	[DMeff]	2.750	2.750
Radial single-ball measurement backlash free (mm)	[MrK]	36.554	41.863
Radial single-ball measurement (mm)	[MrK.e/i]	36.478 / 36.434	41.785 / 41.740
Diameter of measuring circle (mm)	[dMMr.m]	69.031	79.659
Diametral measurement over two balls without clearance (mm)	[MdK]	73.066	83.726
Diametral two ball measure (mm)	[MdK.e/i]	72.913 / 72.825	83.570 / 83.481
Diametral measurement over pins without clearance (mm)	[MdR]	73.066	83.726
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	72.913 / 72.825	83.570 / 83.481
Dimensions over 3 pins without clearance (mm)	[Md3R]	73.023	0.000
Measurement over 3 pins with allowance (mm)	[Md3R.e/i]	72.870 / 72.782	0.000 / 0.000
Chordal tooth thickness (no backlash) (mm)	[sc]	2.972	3.007
Normal chordal tooth thickness with allowance (mm)	[sc.e/i]	2.903 / 2.863	2.938 / 2.898
Reference chordal height from da.m (mm)	[ha]	2.251	2.295
Tooth thickness, arc (mm)	[sn]	2.973	3.008
(mm)	[sn.e/i]	2.903 / 2.863	2.938 / 2.898
Backlash free center distance (mm)	[aControl.e/i]	74.197 / 74.100	
Backlash free center distance, allowances (mm)	[jta]	-0.168 / -0.265	
dNf.i with aControl (mm)	[dNf0.i]	66.281	76.891
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.075	0.098
Tip clearance (mm)	[c0.i(aControl)]	0.206	0.206
Center distance allowances (mm)	[Aa.e/i]	0.015 / -0.015	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.013 / -0.013	
Radial backlash (mm)	[jrw.e/i]	0.280 / 0.153	
Circumferential backlash (transverse section) (mm)	[jtw.e/i]	0.238 / 0.130	
Normal backlash (mm)	[jn.e/i]	0.217 / 0.121	
Torsional angle on input with output fixed:			
Total torsional angle (°)	[j.tSys]	0.3948/0.2163	

9 Toothing tolerances

		----- Gear 1 -----	Gear 2 --
According to DIN 3961:1978			
Accuracy grade	[Q-DIN3961]	6	6
Profile form deviation (µm)	[ff]	6.00	6.00
Profile slope deviation (µm)	[fHa]	5.00	5.00
Total profile deviation (µm)	[Ff]	8.00	8.00
Helix form deviation (µm)	[fβf]	5.50	5.50
Helix slope deviation (µm)	[fHβ]	9.00	9.00
Total helix deviation (µm)	[Fβ]	10.00	10.00
Normal base pitch deviation (µm)	[fpe]	7.00	7.00
Single pitch deviation (µm)	[fp]	7.00	7.00
Adjacent pitch difference (µm)	[fu]	9.00	9.00
Total cumulative pitch deviation (µm)	[Fp]	25.00	25.00
Sector pitch deviation over z/8 pitches (µm)	[Fpz/8]	15.00	15.00
Runout (µm)	[Fr]	16.00	16.00
Tooth Thickness Variation (µm)	[Rs]	10.00	10.00
Single flank composite, total (µm)	[Ft]	26.00	26.00
Single flank composite, tooth-to-tooth (µm)	[ft]	11.00	11.00
Radial composite, total (µm)	[F'']	20.00	20.00
Radial composite, tooth-to-tooth (µm)	[ft'']	8.00	8.00
According DIN 58405:1972 (Precision Mechanics)			
Tooth-to-tooth composite error (µm)	[ft'']	8.00	8.00
Composite error (µm)	[F'']	22.00	22.00
Axis alignment error (µm)	[fp]	12.64	12.64
Flank direction error (µm)	[fβ]	5.00	5.00
Runout (µm)	[Trk, Fr]	24.00	24.00
Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality 6)			
Maximum value for deviation error of axis (µm)	[fΣβ]	12.00	(Fβ=12.00)
Maximum value for inclination error of axes (µm)	[fΣδ]	24.00	

10 Modifying and defining the tooth form

10.1 Data for the tooth form calculation

Data not available.

Please run the calculation in the "Tooth form" tab and open the main report again.

11 Supplementary data

Maximal possible center distance (eps_a=1.0)	[aMAX]	75.279	
Mass (kg)	[m]	0.611	0.663
Total mass (kg)	[mGes]	1.274	
calculation without consideration of the exact tooth shape			
Single gears, (da+df)/2...di (kg*m²)	[J]	0.0003908	0.0006535
Torsional stiffness at driving gear with fixed driven gear:			
Torsional stiffness (MNm/rad)	[cr]	0.497	
Torsion when subjected to nominal torque (°)	[δcr]	0.012	
Mean coefficient of friction (as defined in Niemann)	[μm]	0.054	

Wear sliding coef. by Niemann	[ζw]	0.562
Loss factor	[HV]	0.082
Gear power loss (kW)	[PVZ]	0.152
Meshing efficiency (%)	[ηz]	99.558
Sound pressure level according to Masuda, without contact analysis	[dB(A)]	72.3

12 Service life, damage

Required safety for tooth root	[SFmin]	1.40
Required safety for tooth flank	[SHmin]	1.00
Required service life	[H]	20000.00
Service life (calculated with required safeties):		
System service life (h)	[Hatt]	> 1000000
Tooth root service life (h)	[HFatt]	1e+06
Tooth flank service life (h)	[HHatt]	1e+06

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

Damage relative to the required service life (H, 20000.0 h)

F1%	F2%	H1%	H2%
0.00	0.0000	0.0000	0.0000

13 Remarks

- Specifications with [./i] imply: Maximum [e] and minimum value [i] for Taking all tolerances into account
Specifications with [./m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness allowance are taken into account.
The maximum and minimum clearance according to the largest or smallest allowances are defined..
The calculation is performed for the operating pitch circle.
- Details of calculation method:
cy according to Method B
Kv according to Method B
KHβ and KFβ according to Method C
KHα, KFα according to Method B

A.2 Final drive gear

Calculation of a helical-toothed cylindrical gear pair

Final drive sizing

Changed on
Changed on

04.07.2022, 13:22:39
igorm

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1 Messages

Messages occurred during the calculation:

! Take note of the f_{ZCa} coefficient, which is new in ISO 6336-2:2019, and is used for defining the Hertzian pressure in the single tooth contact points (coefficients Z_B and Z_D). Enter the value in the Settings-Button dialog for root and flank calculation.

We suggest that you select the following option for the active calculation:
Single contact point factors for 'With suitable modifications (from experience) ($f_{ZCa} = 1.07$)'

! Note:
Gear 2:
Measuring the Base tangent length is critical,
it is better to use measurement over pins!

2 Overview

Calculation method	ISO 6336:2019	
Drawing or article number:		
Gear 1:	0.000.0	
Gear 2:	0.000.0	
	----- Gear 1 ----- Gear 2 --	
Power (kW)	[P]	150.000
Speed (1/min)	[n]	2263.0 550.5
Torque (Nm)	[T]	633.0 2602.2
Application factor	[KA]	1.25
Required service life (h)	[H]	4000.00
Gear driving (+) / driven (-)		+ -
Working flank gear 1:	Right flank	
Gear 1 direction of rotation:	Clockwise	

3 Tooth geometry and material

Geometry calculation according to	ISO 21771:2007	
	----- Gear 1 ----- Gear 2 --	
Center distance (mm)	[a]	247.000
Center distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	3.5000
Normal pressure angle (°)	[on]	25.0000
Helix angle at reference circle (°)	[β]	12.5000
Number of teeth	[z]	27 111
Facewidth (mm)	[b]	43.00 40.00
Hand of gear		right
left		
Accuracy grade	[Q-ISO 1328:2013]	A6 A6
Inner diameter (mm)	[di]	60.00 0.00
Inner diameter of gear rim (mm)	[dbi]	0.00 0.00

3.1 Material

Gear 1

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

Gear 2

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

		----- Gear 1 -----	Gear 2 --
Surface hardness		HRC 61	HRC 61
Material quality according to ISO 6336:2006 Normal. Life factors ZNT and YNT ≥ 0.85 .			
Fatigue strength. tooth root stress (N/mm ²)	[σ_{Flim}]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σ_{Hlim}]	1500.00	1500.00
Tensile strength (N/mm ²)	[σ_{B}]	1200.00	1200.00
Yield point (N/mm ²)	[σ_{S}]	850.00	850.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

3.2 Reference profiles
Gear reference profile 1:

Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.318)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[α_{prP}]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[α_{KP}]	0.000	
	not topping		

Gear reference profile 2:

Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[pfP*]	0.380	(pfPmax*=0.318)
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[paP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[α_{prP}]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[α_{KP}]	0.000	
	not topping		

3.2.1 Information on final machining

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[pfP*]	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[α_{prP}]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[α_{KP}]	0.000	0.000

Type of profile modification:

none (only running-in)

Tip relief by running in (μm)

[Ca L/R] 2.0 / 2.0 2.0 / 2.0

3.3 Lubrication

Lubrication type	Oil bath lubrication	
Type of oil	ISO-VG 220	
Lubricant base	Mineral-oil base	
Oil nominal kinematic viscosity at 40°C (mm²/s)	[v40]	220.00
Oil nominal kinematic viscosity at 100°C (mm²/s)	[v100]	17.50
Specific density at 15°C (kg/dm³)	[ρ]	0.895
Oil temperature (°C)	[TS]	70.000

3.4 Gear pair

Overall transmission ratio	[itot]	-4.111
Gear ratio	[u]	4.111
Transverse module (mm)	[mt]	3.585
Transverse pressure angle (°)	[αt]	25.531
Working pressure angle (°)	[αwt]	25.353
	[αwt.e/i]	25.365 / 25.342
Working pressure angle at normal section (°)	[αwn]	24.827
Helix angle at operating pitch circle (°)	[βw]	12.482
Base helix angle (°)	[βb]	11.313
Reference center distance (mm)	[ad]	247.364
Pitch on reference circle (mm)	[pt]	11.263
Base pitch (mm)	[pbt]	10.163
Transverse pitch on contact-path (mm)	[pet]	10.163
Sum of profile shift coefficients	[Σxi]	-0.1035
Transverse contact ratio	[εα]	1.496
Transverse contact ratio with allowances	[εα.e/m/i]	1.501 / 1.491 / 1.481
Overlap ratio	[εβ]	0.787
Total contact ratio	[εγ]	2.283
Total contact ratio with allowances	[εγ.e/m/i]	2.289 / 2.278 / 2.268
Length of path of contact (mm)	[ga, e/i]	15.204 (15.258 / 15.049)
Length T1-A (mm)	[T1A]	11.980 (11.926 / 12.102)
Length T1-B (mm)	[T1B]	17.022 (17.022 / 16.989)
Length T1-C (mm)	[T1C]	20.693 (20.683 / 20.704)
Length T1-D (mm)	[T1D]	22.143 (22.089 / 22.265)
Length T1-E (mm)	[T1E]	27.185 (27.185 / 27.151)
Length T2-A (mm)	[T2A]	93.785 (93.785 / 93.717)
Length T2-B (mm)	[T2B]	88.744 (88.690 / 88.831)
Length T2-C (mm)	[T2C]	85.072 (85.029 / 85.115)
Length T2-D (mm)	[T2D]	83.623 (83.623 / 83.554)
Length T2-E (mm)	[T2E]	78.581 (78.527 / 78.668)
Length T1-T2 (mm)	[T1T2]	105.765 (105.712 / 105.819)
Minimal length of contact line (mm)	[Lmin]	55.478

3.5 Gear 1

Lead height (mm)	[pz]	1371.655
Axial pitch (mm)	[px]	50.802
Profile shift coefficient	[x]	-0.1300
Tooth thickness, arc, in module	[sn*]	1.4496

Tip alteration (mm)	[k*mn]	-0.001
Reference diameter (mm)	[d]	96.794
Base diameter (mm)	[db]	87.343
Tip diameter (mm)	[da]	102.882
(mm)	[da.e/i]	102.882 /102.847
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.035
Tip form diameter (mm)	[dFa]	102.882
(mm)	[dFa.e/i]	102.882 /102.847
Root diameter (mm)	[df]	87.134
Generating Profile shift coefficient	[xE.e/i]	-0.1514 /-0.1637
Generated root diameter with xE (mm)	[df.e/i]	86.984 /86.899
Root form diameter (mm)	[dFf]	90.287
(mm)	[dFf.e/i]	90.199 /90.150
Involute length (mm)	[l_dFa-l_dFf]	6.964
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	3.044
(mm)	[ha.e/i]	3.044 /3.026
Dedendum (mm)	[hf=mn*(hfP*-x)]	4.830
(mm)	[hf.e/i]	4.905 /4.948
Tooth height (mm)	[h]	7.874
Virtual gear no. of teeth	[zn]	28.762
Normal tooth thickness at tip circle (mm)	[san]	2.006
(mm)	[san.e/i]	1.952 /1.889
Normal tooth thickness at tip form circle (mm)	[sFan]	2.006
(mm)	[sFan.e/i]	1.952 /1.889
Normal space width at root circle (mm)	[efn]	0.000
(mm)	[efn.e/i]	0.000 /0.000

3.6 Gear 2

Lead height (mm)	[pz]	5639.027
Axial pitch (mm)	[px]	50.802
Profile shift coefficient	[x]	0.0265
Tooth thickness, arc, in module	[sn*]	1.5955
Tip alteration (mm)	[k*mn]	-0.001
Reference diameter (mm)	[d]	397.933
Base diameter (mm)	[db]	359.077
Tip diameter (mm)	[da]	405.116
(mm)	[da.e/i]	405.116 /405.053
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.063
Tip form diameter (mm)	[dFa]	405.116
(mm)	[dFa.e/i]	405.116 /405.053
Root diameter (mm)	[df]	389.368
Generating Profile shift coefficient	[xE.e/i]	-0.0133 /-0.0317
Generated root diameter with xE (mm)	[df.e/i]	389.089 /388.960
Root form diameter (mm)	[dFf]	391.181
(mm)	[dFf.e/i]	390.925 /390.807
Involute length (mm)	[l_dFa-l_dFf]	7.726
Addendum, $m_n(h_{ap}^*+x+k)$ (mm)	[ha]	3.592
(mm)	[ha.e/i]	3.592 /3.560
Dedendum (mm)	[hf=mn*(hfP*-x)]	4.282
(mm)	[hf.e/i]	4.422 /4.486
Tooth height (mm)	[h]	7.874
Virtual gear no. of teeth	[zn]	118.245
Normal tooth thickness at tip circle (mm)	[san]	2.144
(mm)	[san.e/i]	2.043 /1.951
Normal tooth thickness at tip form circle (mm)	[sFan]	2.144
(mm)	[sFan.e/i]	2.043 /1.951

Normal space width at root circle (mm)	[efn]	1.584
(mm)	[efn.e/]	1.596 / 1.602

3.7 Gear specific pair data Gear pair 1, Gear 1

Operating pitch diameter (mm)	[dw]	96.652
(mm)	[dw.e/]	96.661 / 96.643
Active tip diameter (mm)	[dNa]	102.882
(mm)	[dNa.e/]	102.882 / 102.847
Theoretical tip clearance (mm)	[c]	0.875
Effective tip clearance (mm)	[c.e/]	1.119 / 0.991
Active root diameter (mm)	[dNf]	90.570
(mm)	[dNf.e/]	90.635 / 90.541
Reserve (dNf-dFf)/2 (mm)	[cF.e/]	0.242 / 0.171
Max. sliding velocity at tip (m/s)	[vga]	1.912
Specific sliding at the tip	[ζa]	0.297
Specific sliding at the root	[ζf]	-0.904
Mean specific sliding	[ζm]	0.399
Sliding factor on tip	[Kga]	0.167
Sliding factor on root	[Kgf]	-0.224
Roll angle at dFa (°)	[ξdFa.e/]	35.665 / 35.622
Roll angle to dNa (°)	[ξdNa.e/]	35.665 / 35.622
Roll angle to dNf (°)	[ξdNf.e/]	15.877 / 15.647
Roll angle at dFf (°)	[ξdFf.e/]	14.772 / 14.642
Diameter of single contact point B (mm)	[d-B]	93.743 (93.743 / 93.719)
Diameter of single contact point D (mm)	[d-D]	97.929 (97.880 / 98.039)
Addendum contact ratio	[ε]	0.639 (0.640 / 0.634)

3.8 Gear specific pair data Gear pair 1, Gear 2

Operating pitch diameter (mm)	[dw]	397.348
(mm)	[dw.e/]	397.385 / 397.311
Active tip diameter (mm)	[dNa]	405.116
(mm)	[dNa.e/]	405.116 / 405.053
Theoretical tip clearance (mm)	[c]	0.875
Effective tip clearance (mm)	[c.e/]	1.047 / 0.927
Active root diameter (mm)	[dNf]	391.964
(mm)	[dNf.e/]	392.034 / 391.921
Reserve (dNf-dFf)/2 (mm)	[cF.e/]	0.614 / 0.498
Max. sliding velocity at tip (m/s)	[vga]	2.567
Specific sliding at the tip	[ζa]	0.475
Specific sliding at the root	[ζf]	-0.422
Mean specific sliding	[ζm]	0.399
Sliding factor on tip	[Kga]	0.224
Sliding factor on root	[Kgf]	-0.167
Roll angle at dFa (°)	[ξdFa.e/]	29.930 / 29.908
Roll angle to dNa (°)	[ξdNa.e/]	29.930 / 29.908
Roll angle to dNf (°)	[ξdNf.e/]	25.105 / 25.060
Roll angle at dFf (°)	[ξdFf.e/]	24.661 / 24.613
Diameter of single contact point B (mm)	[d-B]	400.547 (400.500 / 400.624)
Diameter of single contact point D (mm)	[d-D]	396.115 (396.115 / 396.058)
Addendum contact ratio	[ε]	0.857 (0.862 / 0.846)

4 General influence factors

KISSsoft			
----- Gear 1 ----- Gear 2 --			
Nominal circum. force at pitch circle (N)	[Ft]	13078.5	
Axial force (N)	[Fa]	2899.4	
Radial force (N)	[Fr]	6246.7	
Normal force (N)	[Fnorm]	14780.9	
Nominal circumferential force per mm (N/mm)	[w]	326.96	
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]	13097.7	
Axial force (N)	[Faw]	2899.4	
Radial force (N)	[Frw]	6206.2	
Circumferential speed reference circle (m/s)	[v]	11.47	
Circumferential speed operating pitch circle (m/s)	[v(dw)]	11.45	
Running-in value (µm)	[yp]	0.6	
Running-in value (µm)	[yf]	0.8	
Correction factor	[CM]	0.800	
Gear blank factor	[CR]	1.000	
Basic rack factor	[CBS]	1.073	
Material coefficient	[E/Est]	1.000	
Singular tooth stiffness (N/mm/µm)	[c]	14.917	
Meshing stiffness (N/mm/µm)	[cya]	20.467	
Meshing stiffness (N/mm/µm)	[cyβ]	17.397	
Reduced mass (kg/mm)	[mRed]	0.02639	
Resonance speed (min-1)	[nE1]	9849	
Resonance ratio (-)	[N]	0.230	
Subcritical range			
Running-in value (µm)	[ya]	0.8	
Bearing distance l of pinion shaft (mm)	[l]	86.000	
Distance s of pinion shaft (mm)	[s]	8.600	
Outside diameter of pinion shaft (mm)	[dsh]	43.000	
Load in accordance with Figure 13, ISO 6336-1:2006 0:a), 1:b), 2:c), 3:d), 4:e)	[-]	4	
Coefficient K' according to Figure 13, ISO 6336-1:2006 Without stiffening	[K']	-1.00	
Tooth trace deviation (active) (µm)	[Fβy]	4.36	
from deformation of shaft (µm)	[fsh*B1]	1.40	
fsh = (µm) 2.80			
B1 = 0.50			
fHβ5 = (µm) 7.00			
Tooth trace:	width-crowned [Cβ = 0.5*(fmα+fsh)]		
Position of contact pattern:	favorable		
from production tolerances (µm)	[fmα*B2]	6.90	
B2 =			
0.50			
Tooth trace deviation, theoretical (µm)	[Fβx]	5.13	
Running-in value (µm)	[yβ]	0.77	
Dynamic factor	[Kv]	1.074	
Face load factor - flank	[KHβ]	1.086	
- Tooth root	[KFβ]	1.069	
- Scuffing	[KBβ]	1.086	
Transverse load factor - flank	[KHα]	1.068	
- Tooth root	[KFα]	1.068	
- Scuffing	[KBα]	1.068	
Number of load cycles (in mio.)	[NL]	543.120	132.110

5 Tooth root load capacity

Calculation of Tooth form coefficients according method: B

		----- Gear 1 -----	Gear 2 --
Calculated with generating profile shift coefficient	[xE.i]	-0.1637	-0.0317
Tooth form factor	[YF]	1.15	0.94
Stress correction factor	[YS]	1.89	2.30
Load application angle (°)	[αFen]	23.31	25.06
Load distribution influence factor	[fε]	0.848	
Load application diameter (mm)	[den]	97.572	400.199
Bending moment arm (mm)	[hF]	3.70	3.94
Tooth thickness at root (mm)	[sFn]	7.60	8.65
Tooth root radius (mm)	[ρF]	2.11	1.62
Bending moment arm (-)	[hF/mn]	1.057	1.126
Tooth thickness at root (-)	[sFn/mn]	2.173	2.472
Tooth root radius (-)	[ρF/mn]	0.604	0.462
Calculation cross section diameter (mm)	[dsFn]	88.398	390.356
Tangents on calculation cross section (°)	[αsFn]	30.000	30.000
Notch parameter	[qs]	1.799	2.678
Helix angle factor	[Yβ]	0.986	
Deep tooth factor	[YDT]	1.000	
Gear rim factor	[YB]	1.00	1.00
Effective facewidth (mm)	[beff]	43.00	40.00
Nominal stress at tooth root (N/mm²)	[σF0]	186.81	198.40
Tooth root stress (N/mm²)	[σF]	286.47	304.24
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdrelT]	0.993	1.002
Surface factor	[YRrelT]	0.957	0.957
Size factor, tooth root	[YX]	1.000	1.000
Life factor	[YNT]	0.901	0.927
$Y_{drelT} \cdot Y_{RrelT} \cdot Y_X \cdot Y_{NT}$		0.856	0.888
Alternating bending factor, mean stress influence coefficient	[YM]	1.000	1.000
Stress correction factor	[YST]	2.00	
$YST \cdot \sigma_{Flim}$ (N/mm²)	[σFE]	860.00	860.00
Permissible tooth root stress σ_{FG}/SF_{min} (N/mm²)	[σFp]	525.84	545.69
Limit strength tooth root (N/mm²)	[σFG]	736.18	763.97
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=σFG/σF]	2.57	2.51
Transmittable power (kW)	[kWRating]	275.33	269.04

6 Flank safety

		----- Gear 1 -----	Gear 2 --
Zone factor	[ZH]	2.255	
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]	189.812	
Contact ratio factor	[Zε]	0.839	
Helix angle factor	[Zβ]	1.012	
Effective facewidth (mm)	[beff]	40.00	
Nominal contact stress (N/mm²)	[σH0]	744.57	
Contact stress at operating pitch circle (N/mm²)	[σHw]	929.49	
Coefficient [fZCa] 1.20 (Helical gear sets without flank modifications)			
Single tooth contact factor	[ZB,ZD]	1.09	1.07
Contact stress (N/mm²)	[σHB, σHD]	1015.06	994.41

Lubrication factor for NL	[ZL]	1.020	1.020
Speed factor for NL	[ZV]	1.004	1.004
Roughness factor for NL	[ZR]	0.976	0.976
Material hardening factor for NL	[ZW]	1.000	1.000
Life factor	[ZNT]	0.929	0.971
	[ZL*ZV*ZR*ZNT]	0.929	0.970
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress, $\sigma_{HG}/\sigma_{Hmin}$ (N/mm ²)	[σ_{HP}]	1393.71	1455.47
Pitting stress limit (N/mm ²)	[σ_{HG}]	1393.71	1455.47
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress on operating pitch circle	[SHw]	1.50	1.57
Safety against pressure, σ_{HG}/σ_{HBD} Single contact	[SHBD]	1.37	1.46
Safety regarding transmittable torque	[$(\sigma_{HBD})^2$]	1.89	2.14
Transmittable power (kW)	[kWRating]	282.78	321.34

7 Micropitting

Calculation method according to ISO/TS 6336-22:2018
Calculation has not been carried out, lubricant: Load stage micropitting test not known

8 Scuffing load capacity

Calculation method according to		ISO/TS 6336-20/21:2017	
Helical load factor for scuffing	[KBy]	1.175	
Lubrication coefficient for lubrication type	[XS]	1.000	
Scuffing test and load stage	[FZGtest] FZG - Test A / 8.3 / 90 (ISO 14635 - 1)		12
Multiple meshing factor	[Xmp]	1.000	
Relative structural factor, scuffing	[XWrelT]	1.000	
Thermal contact factor (N/mm/s^0.5/K)	[BM]	13.780	13.780
Relevant tip relief (µm)	[Ca]	2.00	2.00
Optimal tip relief (µm)	[Ceff]	19.97	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	40.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	509.531	
KBy =1.175, wBt*KBy =598.953			
Angle factor	[Xαβ]	1.060	
ε1: 0.639 , ε2: 0.857			

8.1 Flash temperature-criteria

Lubricant factor	[XL]	0.830
Tooth mass temperature (°C)	[θ_{Mi}]	76.62
$\theta_{Mi} = \theta_{oil} + XS \cdot 0.47 \cdot Xmp \cdot \theta_{flm}$		
Average flash temperature (°C)	[θ_{flm}]	14.07
Scuffing temperature (°C)	[θ_S]	348.80
Γ coordinates (point of highest temperature)	[Γ]	-0.421
[$\Gamma.A$]= -0.421 [$\Gamma.E$]= 0.314		
Highest contact temp. (°C)	[θ_B]	105.37
Flash factor ($^{\circ}K \cdot N^{0.75} \cdot s^{0.5} \cdot m^{0.5} \cdot mm$)	[XM]	50.058

Approach factor	[XJ]	1.068
Load sharing factor	[XΓ]	0.412
Dynamic viscosity (mPa*s)	[ηM]	41.90 (70.0 °C)
Coefficient of friction	[μ _m]	0.048
Required safety	[SBmin]	2.000
Margin of safety for scuffing, flash temperature	[SB]	7.883

8.2 Integral temperature-criteria

Lubricant factor	[XL]	1.000
Tooth mass temperature (°C)	[θMC]	77.72
$\theta_{MC} = \theta_{oil} + X_S \cdot 0.70 \cdot \theta_{flint}$		
Integral scuffing temperature (°C)	[θSint]	360.78
Flash factor (°K*N ^{0.75} *s ^{0.5} *m ^{-0.5} *mm)	[XM]	50.058
Running-in factor, well run in	[XE]	1.000
Contact ratio factor	[Xε]	0.347
Dynamic viscosity (mPa*s)	[ηOil]	41.90 (70.0 °C)
Mean coefficient of friction	[μ _m]	0.046
Geometry factor	[XBE]	0.143
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.132
Mean flash temperature (°C)	[θflint]	11.03
Integral tooth flank temperature (°C)	[θint]	94.27
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	3.827
Safety referring to transmittable torque	[SSL]	11.979

9 Measurements for tooth thickness

		----- Gear 1 -----	Gear 2 --
		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness tolerance			
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070 / -0.110	-0.130 / -0.190
Number of teeth spanned	[k]	4.000	17.000
Base tangent length (no backlash) (mm)	[Wk]	37.238	175.788
Base tangent length with allowance (mm)	[Wk.e/i]	37.175 / 37.138	175.670 / 175.615
(mm)	[ΔWk.e/i]	-0.063 / -0.100	-0.118 / -0.172
Diameter of measuring circle (mm)	[dMWk.m]	94.638	398.245
Theoretical diameter of ball/pin (mm)	[DM]	6.120	6.109
Effective diameter of ball/pin (mm)	[DMeff]	6.500	6.500
Radial single-ball measurement backlash free (mm)	[MrK]	52.918	204.067
Radial single-ball measurement (mm)	[MrK.e/i]	52.850 / 52.811	203.933 / 203.871
Diameter of measuring circle (mm)	[dMMr.m]	96.308	398.499
Diametral measurement over two balls without clearance (mm)	[MdK]	105.669	408.094
Diametral two ball measure (mm)	[MdK.e/i]	105.533 / 105.455	407.826 / 407.701
Diametral measurement over pins without clearance (mm)	[MdR]	105.837	408.134
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	105.701 / 105.623	407.866 / 407.742
Measurement over 2 pins, free, according to AGMA 2002 (mm)	[dk2f.e/i]	105.524 / 105.446	407.824 / 407.699
Measurement over 2 pins, transverse, according to AGMA 2002 (mm)	[dk2t.e/i]	105.864 / 105.786	407.906 / 407.782
Measurement over 3 pins, axial, according to AGMA 2002 (mm)	[dk3A.e/i]	105.701 / 105.623	407.866 / 407.742
Chordal tooth thickness (no backlash) (mm)	[sc]	5.071	5.584
Normal chordal tooth thickness with allowance (mm)	[sc.e/i]	5.003 / 4.964	5.455 / 5.395

Reference chordal height from da.m (mm)	[ha]	3.099	3.595
Tooth thickness, arc (mm)	[sn]	5.073	5.584
(mm)	[sn.e/i]	5.003 / 4.963	5.454 / 5.394
Backlash free center distance (mm)	[aControl.e/i]	246.784 / 246.675	
Backlash free center distance, allowances (mm)	[jta]	-0.216 / -0.325	
dNf.i with aControl (mm)	[dNf0.i]	90.179	391.357
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	-0.010	0.216
Tip clearance (mm)	[c0.i(aControl)]	0.689	0.625
Center distance allowances (mm)	[Aa.e/i]	0.023 / -0.023	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.022 / -0.022	
Radial backlash (mm)	[jrw.e/i]	0.348 / 0.193	
Circumferential backlash (transverse section) (mm)	[jtw.e/i]	0.329 / 0.183	
Normal backlash (mm)	[jn.e/i]	0.291 / 0.162	
Torsional angle on input with output fixed:			
Total torsional angle (°)	[j.tSys]	0.3896/0.2167	

10 Toothing tolerances

		----- Gear 1 -----	Gear 2 --
According to ISO 1328-1:2013, ISO 1328-2:1997			
Accuracy grade	[Q]	A6	A6
Single pitch deviation (µm)	[fptT]	9.00	9.50
Base circle pitch deviation (µm)	[fpbT]	8.12	8.57
Sector pitch deviation over k/8 pitches (µm)	[Fpk/8T]	18.00	23.00
Profile form deviation (µm)	[fioT]	10.00	10.00
Profile slope deviation (µm)	[fHoT]	8.00	8.00
Total profile deviation (µm)	[FoT]	13.00	13.00
Helix form deviation (µm)	[fHβT]	11.00	12.00
Helix slope deviation (µm)	[fHβT]	9.50	10.00
Total helix deviation (µm)	[FβT]	14.00	15.00
Total cumulative pitch deviation (µm)	[FpT]	28.00	37.00
Adjacent pitch difference (µm)	[fuT]	13.00	14.00
Runout (µm)	[FrT]	26.00	33.00
Single flank composite, total (µm)	[FisT]	37.00	46.00
Single flank composite, tooth-to-tooth (µm)	[fisT]	9.00	9.00
Radial composite, total (µm)	[FidT]	36.00	52.00
Radial composite, tooth-to-tooth (µm)	[fidT]	14.00	15.00
FidT (F''), fidT (fi'') according to ISO 1328:1997 calculated with the geometric mean values for mn and d			
According to ISO 1328-2:2020			
Accuracy grade	[Q]	R40	R40
Radial composite, total (µm)	[FidT]	36.00	48.00
Radial composite, tooth-to-tooth (µm)	[fidT]	16.00	20.00

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

	6		
Maximum value for deviation error of axis (µm)	[fΣβ]	16.13	(Fβ=15.00)
Maximum value for inclination error of axes (µm)	[fΣδ]	32.25	

11 Modifying and defining the tooth form

11.1 Profile and flank line modifications

gear 1

Symmetric (both flanks)

- Flank line crowning
 $C_b = 3.000 \mu\text{m}$
 $r_{\text{crown}} = 77042\text{mm}$

gear 2**Symmetric (both flanks)**

- Flank line crowning
 $C_b = 3.000 \mu\text{m}$
 $r_{\text{crown}} = 66667\text{mm}$

11.2 Data for the tooth form calculation

Data not available.

Please run the calculation in the "Tooth form" tab and open the main report again.

12 Supplementary data

Mass (kg)	[m]	1.435	38.817
Total mass (kg)	[mGes]	40.252	
Moment of inertia for system, relative to the input: calculation without consideration of the exact tooth shape			
Single gears, $(d_a + d_f)/2 \dots d_i$ (kg*m ²)	[J]	0.00226	0.76567
System $(d_a + d_f)/2 \dots d_i$ (kg*m ²)	[J]	0.04757	
Torsional stiffness at driving gear with fixed driven gear:			
Torsional stiffness (MNm/rad)	[cr]	1.329	
Torsion when subjected to nominal torque (°)	[δcr]	0.027	
Mean coefficient of friction (as defined in Niemann)	[μ _m]	0.049	
Wear sliding coef. by Niemann	[ζw]	0.597	
Loss factor	[HV]	0.095	
Gear power loss (kW)	[PVZ]	0.696	
Meshing efficiency (%)	[ηz]	99.536	
Sound pressure level according to Masuda, without contact analysis			
	[dB(A)]	84.2	

13 Service life, damage

Required safety for tooth root	[SFmin]	1.40	
Required safety for tooth flank	[SHmin]	1.00	
Required service life	[H]	4000.00	
Service life (calculated with required safeties):			
System service life (h)	[Hatt]	>	1000000
Tooth root service life (h)	[HFatt]	1e+06	1e+06
Tooth flank service life (h)	[HHatt]	1e+06	1e+06
Note: The entry 1e+006 h means that the Service life > 1,000,000 h.			

Damage relative to the required service life (H, 4000.0 h)

F1%	F2%	H1%	H2%
0.00	0.0000	0.0000	0.0000

14 Remarks

- Specifications with [..e/î] imply: Maximum [e] and minimum value [î] for Taking all tolerances into account
 - Specifications with [..m] imply: Mean value within tolerance
 - For the backlash tolerance, the center distance tolerances and the tooth thickness allowance are taken into account.
- The maximum and minimum clearance according to the largest or smallest allowances are defined..
- The calculation is performed for the operating pitch circle.
- Calculation of Zbet according to Corrigendum 1 ISO 6336-2:2008 with $Z\beta = 1/(\cos(\beta)^{0.5})$
 - Details of calculation method:
 - cy according to Method B
 - Kv according to Method B
 - KH β and KF β according to Method C
 - fma according to equation 64, fsh according to 57/58, F β x according to 52/53/54
 - KH α , KF α according to Method B
 - The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRreIT and YX..

End of Report

lines:

622

A.3 Final drive gear splines

KISSsoft

Spline

Spline Final Drive

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8	Remarks	7

1 Messages

No messages occurred during the calculation.

2 Overview

Shaft DIN 5480 - W 68.00x2.00x32x8h
Hub DIN 5480 - N 68.00x2.00x32x9H

Drawing or article number:
Shaft: 0.000.0
Hub: 0.000.0

3 Tooth geometry and material

Normal module (mm)	[mn]	2.0000
Reference diameter (mm)	[dB]	68.00
Normal pressure angle (°)	[on]	30.000
		----- Shaft ----- Hub -----
Number of teeth	[z]	32 -32
Helix angle at reference circle (°)	[β]	0.0000
Facewidth (mm)	[b]	43.00 43.00
Hand of gear		Spur gear

3.1 Material

Gear 1
18CrNiMo7-6, Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

Gear 2
18CrNiMo7-6, Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

		----- Shaft ----- Hub ----
Surface hardness		HRC 61 HRC 61
Reference diameter material (mm)	[dB]	70 110
Tensile strength (N/mm ²)	[σB]	900.00 600.00
Yield point (N/mm ²)	[σS]	640.00 425.00
Young's modulus (N/mm ²)	[E]	206000 206000
Poisson's ratio	[ν]	0.300 0.300
Roughness average value DS, flank (μm)	[RAH]	0.00 0.00
Roughness average value DS, root (μm)	[RAF]	0.00 0.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80 4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00 20.00

3.2 Reference profiles

Gear reference profile 1:
Reference profile DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching
Dedendum coefficient [hfP*] 0.550
Root radius factor [pfP*] 0.160 (pfPmax*=0.810)

Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[qprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
not topping		
Gear reference profile 2:		
Reference profile	DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching	
Dedendum coefficient	[hfP*]	0.550
Root radius factor	[pfP*]	0.160 (pfPmax*=0.810)
Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[qprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
not topping		

3.2.1 Information on final machining

Dedendum reference profile	[hfP*]	0.550	0.550
Tooth root radius Refer. profile	[pfP*]	0.160	0.160
Addendum Reference profile	[haP*]	0.450	0.450
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[qprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[αKP]	0.000	0.000

Transverse module (mm)	[mt]	2.000
Transverse pressure angle (°)	[αt]	30.000
Base helix angle (°)	[βb]	0.000
Sum of profile shift coefficients	[Σxi]	0.0000

		----- Shaft -----	Hub -----
Profile shift coefficient	[x]	0.4500	-0.4500
Profile shift (x*m) (mm)	[x*m]	0.9000	-0.9000
Pitch diameter (mm)	[d]	64.000	64.000
Base diameter (mm)	[db]	55.426	55.426
Tip diameter (mm)	[da]	67.600	64.000
Effective tip diameter (mm)	[da.e/i]	67.600 /67.410	
Effective tip diameter (mm)	[da.e/i]		64.000 /64.190
Tip diameter allowances (mm)	[Ada.e/i]	0.000 /-0.190	
Tip diameter allowances (mm)	[Ada.e/i]		-0.000 / 0.190
Root diameter (mm)	[df]	63.600	68.000
Effective root diameter (mm)	[df.e/i]	63.565 /63.503	
Effective root diameter (mm)	[df.e/i]		68.052 /68.139
Root diameter allowances (mm)	[Adf.e/i]	-0.035 /-0.097	
Root diameter allowances (mm)	[Adf.e/i]		0.052 / 0.139
Generating Profile shift coefficient	[xE.e/i]	0.4413/0.4258	
Generating Profile shift coefficient	[xE.e/i]		-0.4630/-0.4846
Root form diameter (mm)	[dFf]	63.920	67.726
Effective root form diameter (mm)	[dFf.e/i]	63.886 /63.824	
Effective root form diameter (mm)	[dFf.e/i]		67.778 /67.866
(dFf2 calculated on the basis of the generated involute.)			
Root diameter allowances in accordance with DIN 5480 -1, Table 5:			
Root form diameter (mm)	[dFf]	63.920	67.680
Effective root form diameter (mm)	[dFf.e/i]	63.920 /63.858	
Effective root form diameter (mm)	[dFf.e/i]		67.680 /67.767

Active tip diameter (mm)	[dNa.e/l]	67.600 / 67.410
Active tip diameter (mm)	[dNa.e/l]	64.000 / 64.190
Active root diameter (mm)	[dNf.e/l]	64.000 / 64.190
Active root diameter (mm)	[dNf.e/l]	67.600 / 67.410
Tooth height (mm)	[h]	2.000 2.000
Theoretical tip clearance (mm)	[c]	0.200 0.200
Effective tip clearance (mm)	[c.e/l]	0.364 / 0.226
Effective tip clearance (mm)	[c.e/l]	0.343 / 0.217
Normal tooth thickness at tip circle (mm)	[san]	2.054 2.102
(mm)	[san.e/l]	2.159 / 1.994
(mm)	[san.e/l]	2.189 / 2.022
Normal space width at root circle (mm)	[efn]	1.862 1.783
(mm)	[efn.e/l]	1.861 / 1.860
(mm)	[efn.e/l]	1.779 / 1.773
Pitch on reference circle (mm)	[pt]	6.283
Base pitch (mm)	[pbt]	5.441
Transverse pitch on contact-path (mm)	[pet]	5.441

3.3 Measurements for tooth thickness

		----- Shaft -----	Hub -----
Tolerance class		8	9
Tooth thickness tolerance		DIN 5480 h	DIN 5480 H
Number of teeth spanned	[k]	6.0000	6.0000
Base tangent length (no backlash) (mm)	[Wk]	33.8069	33.8069
Diameter of measuring circle (mm)	[dMWk.m]	64.9178	64.9291
Theoretical diameter of ball/pin (mm)	[dm]	4.0164	3.6728
Effective diameter of ball/pin (mm)	[DMeff]	4.2500	3.7500
Theor. dimension over two balls (mm)	[MRe/Mri-ball]	72.6992	59.9898
Diametral measurement over pins without clearance (mm)	[MRe/Mri-pin]	72.6992	59.9898

3.4 Data for Actual Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Smax/Smin]	4.1608/ 4.1248
Tooth thickness / Spacewidth (mm)	[Emax/Emin]	4.2608/ 4.2108
Tooth thickness tolerance, normal section (mm)	[Tol.Smax/min]	-0.0200/-0.0560
Tooth space tolerance, normal section (mm)	[Tol.Emax/min]	0.0800/ 0.0300
Base tangent length (mm)	[Wk.Smax/Smin]	33.7896/33.7584
Base tangent length (mm)	[Wk.Smax/Smin]	33.8762/33.8329
Diametral two ball measure (mm)	[MRe/Mri-ball]	72.6697/72.6165
Diametral two ball measure (mm)	[MRe/Mri-ball]	60.1296/60.0423
Diametral measurement over pins (mm)	[MRe/Mri-pin]	72.6697/72.6165
Diametral measurement over pins (mm)	[MRe/Mri-pin]	60.1296/60.0423

3.5 Data for Effective Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Svmax/min]	4.1808/ 4.1608
Tooth thickness / Spacewidth (mm)	[Evmax/min]	4.2108/ 4.1808
Tooth thickness tolerance, normal section (mm)	[Tol.Svmax/min]	0.0000/-0.0200
Tooth space tolerance, normal section (mm)	[Tol.Evmax/min]	0.0300/ 0.0000
Base tangent length (mm)	[Wk.Svmax/min]	33.8069/33.7896
Base tangent length (mm)	[Wk.Svmax/min]	33.8329/33.8069
Diametral two ball measure (mm)	[MRe/Mri-ball]	72.6992/72.6697
Diametral two ball measure (mm)	[MRe/Mri-ball]	60.0423/59.9898
Diametral measurement over pins (mm)	[MRe/Mri-pin]	72.6992/72.6697
Diametral measurement over pins (mm)	[MRe/Mri-pin]	60.0423/59.9898

Tolerance data DIN 5480-1 (mm)	[TG]	0.0560	0.0800
(mm)	[Tact]	0.0360	0.0500
(mm)	[Teff]	0.0200	0.0300

Circumferential backlash, transverse section:

-Theoretical, without form errors (mm)	[jt.act]	0.1360/0.0500
-Effective, with form errors (mm)	[jt.eff]	0.0500/0.0000

Normal backlash:

-Theoretical, without form errors (mm)	[jn.act]	0.1178/0.0433
-Effective, with form errors (mm)	[jn.eff]	0.0433/0.0000

Radial backlash:

-Theoretical, without form errors (mm)	[jr.act]	0.0589/0.0217
-Effective, with form errors (mm)	[jr.eff]	0.0217/0.0000

Note: Take the values in "Actual dimensions" into account when checking splines with individual measurements (base tangent length/ measurement over pins)..

4 Toothing tolerances

		----- Shaft -----	Hub -----
According to	DIN 5480:2006:		
Tolerance class	[Q-DIN5480]	8	9
Total profile deviation (µm)	[Fa]	19.0	25.0
Total helix deviation (µm)	[Fb]	10.0	13.0
Single pitch deviation (µm)	[fp]	15.0	21.0
Total cumulative pitch deviation (µm)	[Fp]	36.0	50.0
Runout (µm)	[Fr]	50.0	50.0

5 Strength calculation

Calculation method:	G. Niemann, Machine Elements I, 4th edition.	
Centering:	flank centered	
Supporting length (mm)	[l _{tr}]	43.00
Maximal circumferential force (N)	[F _t]	9422.49
Maximal circumferential force per tooth (N)	[F _t /z]	294.45
Load application diameter (mm)	[d _m]	65.80
Tooth height (mm)	[h]	1.61
Distance a ₀ (mm)	[a ₀]	21.50
Length factor	[k _λ]	1.07
Share factor, equivalent load	[k _φ β _q]	2.00
Share factor, maximum load	[k _φ β _{max}]	1.70

The share factors k_φβ according to Niemann are determined according to the tolerance class specified in DIN 5480..

[Q] 9

Nominal torque (Nm)	[T _{enn}]	182.00
Application factor	[K _A]	1.25
Service torque (Nm)	[T _{eq}]	227.50
Maximum torque (Nm)	[T _{max}]	310.00
Torque curve:	No alternating torque	
Number of load peaks	[N _L]	1000
Number of change of load direction	[N _W]	1
Load direction changing coefficient	[f _w]	1.00
Tolerance field according to DIN 5480	"H9"	

5.1 Shaft

Width on shaft (mm)	[L_W]	43.00
Supporting surface (mm²)	[F _{lw} =l _{tr} *h*z]	2215.36
Tip form diameter (mm)	[dFa1.1]	67.41
Equivalent effective surface pressure (N/mm²)	[peq]	6.66
Maximum effective surface pressure (N/mm²)	[pmax]	7.72
Support factor	[fs]	1.30
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm²)	[pzuleq]	956.80
Permissible pressure (N/mm²)	[pzulmax]	1435.20
f _w * pzul / peq		143.57
f _L * pzul / pmax		185.93
Required safety	[Smin]	1.00
Minimal safety	[S]	143.57

5.2 Hub

Width on hub (mm)	[L_N]	43.00
Supporting surface (mm²)	[F _{ln} =l _{tr} *h*z]	2215.36
Tip form diameter (mm)	[dFa2.1]	-64.19
Small external diameter (mm)	[D1]	100.00
Big external diameter (mm)	[D2]	100.00
Width of hub-part with D2 (mm)	[c]	43.00
Equivalent diameter hub (mm)	[D]	100.00
Equivalent effective surface pressure (N/mm²)	[peq]	6.66
Maximum effective surface pressure (N/mm²)	[pmax]	7.72
Support factor	[fs]	1.50
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm²)	[pzuleq]	733.13
Permissible pressure (N/mm²)	[pzulmax]	1099.69
f _w * pzul / peq		110.00
f _L * pzul / pmax		142.46
Required safety	[Smin]	1.00
Minimal safety	[S]	110.00

6 Supplementary data

Moment of inertia (System referenced to wheel 1):
calculation without consideration of the exact tooth shape
Single gears ((da+df)/2...di) (kg*m²)
System, (da+df)/2...di (kg*m²)

[J]	0.0006091	0.002675
[J]	0.003284	

7 Modifying and defining the tooth form

7.1 Data for the tooth form calculation

Data not available.
Please run the calculation in the "Tooth form" tab and open the main report again.

8 Remarks



- Specifications with [e/l] imply: Maximum [e] and minimum value [l] for
Taking all tolerances into account
Specifications with [m] imply: Mean value within tolerance
- Concerning the calculation method:
 $h = (dFa1.i - ABS(dFa2.i)) / 2.0$
 $dm = (dFa1.i + ABS(dFa2.i)) / 2.0;$
 $Ft = Mmax * 2000 / dm$
Pressure load: $p(eq,max) = k\phi\beta(eq,max)*k1*M*2000/(dm*lh*z); pmax \geq peq$
If $pmax < peq$, $pmax = peq$ is set.
Coefficient for load direction changes according to DIN 6892:2012/ fig. 7
 $pzuleq = fs*fh*fw*(Rm,Rp)$
 $pzulmax = fs*fh*fl*(Rm,Rp)$
Rm for brittle material; Rp for ductile material

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Appendix - Splines & Key Calculations KISSsoft report

This appendix provides the shafts' splines calculations, as well as the calculation for the one-way clutch key.

- Small sun gear shaft - input spline;
- Small sun gear shaft - gear coupling spline;
- Large sun gear shaft - gear coupling spline spline;
- Large sun gear shaft - brake drum coupling spline;
- One-way clutch key.

B.1 Small sun gear shaft - input spline

KISSsoft

Spline

Spline Small sun input

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04.07.2022, 13:23:51
igorm

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1 Messages

No messages occurred during the calculation.

2 Overview

Shaft	DIN 5480 - W 28.00x0.80x34x8h
Hub	DIN 5480 - N 28.00x0.80x34x9H
Drawing or article number:	
Shaft:	0.000.0
Hub:	0.000.0

3 Tooth geometry and material

Normal module (mm)	[mn]	0.8000	
Reference diameter (mm)	[dB]	28.00	
Normal pressure angle (°)	[on]	30.000	
		----- Shaft ----- Hub -----	
Number of teeth	[z]	34	-34
Helix angle at reference circle (°)	[β]	0.0000	
Facewidth (mm)	[b]	30.00	30.00
Hand of gear			Spur gear

3.1 Material

Gear 1
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

Gear 2
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

		----- Shaft ----- Hub ----	
Surface hardness		HRC 61	HRC 61
Reference diameter material (mm)	[dB]	28	45
Tensile strength (N/mm ²)	[σB]	1100.00	900.00
Yield point (N/mm ²)	[σS]	745.00	640.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.00	0.00
Roughness average value DS, root (μm)	[RAF]	0.00	0.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

3.2 Reference profiles

Gear reference profile 1:		
Reference profile	DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching	
Dedendum coefficient	[hfP*]	0.550
Root radius factor	[pfP*]	0.160 (pfPmax*=0.810)

Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[qprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	
Gear reference profile 2:		
Reference profile	DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching	
Dedendum coefficient	[hfP*]	0.550
Root radius factor	[pfP*]	0.160 (pfPmax*=0.810)
Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[qprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	

3.2.1 Information on final machining

Dedendum reference profile	[hfP*]	0.550	0.550
Tooth root radius Refer. profile	[pfP*]	0.160	0.160
Addendum Reference profile	[haP*]	0.450	0.450
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[qprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[αKP]	0.000	0.000

Transverse module (mm)	[mt]	0.800
Transverse pressure angle (°)	[αt]	30.000
Base helix angle (°)	[βb]	0.000
Sum of profile shift coefficients	[Σxi]	0.0000

		----- Shaft -----	Hub -----
Profile shift coefficient	[x]	-0.0500	0.0500
Profile shift (x*m) (mm)	[x*m]	-0.0400	0.0400
Pitch diameter (mm)	[d]	27.200	27.200
Base diameter (mm)	[db]	23.556	23.556
Tip diameter (mm)	[da]	27.840	26.400
Effective tip diameter (mm)	[da.e/i]	27.840 /27.710	
Effective tip diameter (mm)	[da.e/i]		26.400 /26.530
Tip diameter allowances (mm)	[Ada.e/i]	0.000 /-0.130	
Tip diameter allowances (mm)	[Ada.e/i]		-0.000 / 0.130
Root diameter (mm)	[df]	26.240	28.000
Effective root diameter (mm)	[df.e/i]	26.211 /26.162	
Effective root diameter (mm)	[df.e/i]		28.040 /28.109
Root diameter allowances (mm)	[Adf.e/i]	-0.029 /-0.078	
Root diameter allowances (mm)	[Adf.e/i]		0.040 / 0.109
Generating Profile shift coefficient	[xE.e/i]	-0.0684/-0.0987	
Generating Profile shift coefficient	[xE.e/i]		0.0251/-0.0182
Root form diameter (mm)	[dFf]	26.407	27.880
Effective root form diameter (mm)	[dFf.e/i]	26.381 /26.337	
Effective root form diameter (mm)	[dFf.e/i]		27.920 /27.991
(dFf2 calculated on the basis of the generated involute.)			
Root diameter allowances in accordance with DIN 5480 -1, Table 5:			
Root form diameter (mm)	[dFf]	26.340	27.900
Effective root form diameter (mm)	[dFf.e/i]	26.340 /26.292	
Effective root form diameter (mm)	[dFf.e/i]		27.900 /27.969

Active tip diameter (mm)	[dNa.e/l]	27.840 / 27.710
Active tip diameter (mm)	[dNa.e/l]	26.400 / 26.530
Active root diameter (mm)	[dNf.e/l]	26.400 / 26.530
Active root diameter (mm)	[dNf.e/l]	27.840 / 27.710
Tooth height (mm)	[h]	0.800 / 0.800
Theoretical tip clearance (mm)	[c]	0.080 / 0.080
Effective tip clearance (mm)	[c.e/l]	0.200 / 0.100
Effective tip clearance (mm)	[c.e/l]	0.184 / 0.095
Normal tooth thickness at tip circle (mm)	[san]	0.848 / 0.837
(mm)	[san.e/l]	0.908 / 0.802
(mm)	[san.e/l]	0.886 / 0.776
Normal space width at root circle (mm)	[efn]	0.753 / 0.751
(mm)	[efn.e/l]	0.754 / 0.756
(mm)	[efn.e/l]	0.750 / 0.748
Pitch on reference circle (mm)	[pt]	2.513
Base pitch (mm)	[pbt]	2.177
Transverse pitch on contact-path (mm)	[pet]	2.177

3.3 Measurements for tooth thickness

		----- Shaft -----	Hub -----
Tolerance class		8	9
Tooth thickness tolerance		DIN 5480 h	DIN 5480 H
Number of teeth spanned	[k]	6.0000	6.0000
Base tangent length (no backlash) (mm)	[Wk]	13.1972	13.1972
Diameter of measuring circle (mm)	[dMWk.m]	26.9973	27.0057
Theoretical diameter of ball/pin (mm)	[dm]	1.4829	1.4062
Effective diameter of ball/pin (mm)	[DMeff]	1.5000	1.5000
Theor. dimension over two balls (mm)	[MRe/Mri-ball]	29.4165	24.7420
Diametral measurement over pins without clearance (mm)	[MRe/Mri-pin]	29.4165	24.7420

3.4 Data for Actual Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Smax/Smin]	1.1934/ 1.1654
Tooth thickness / Spacewidth (mm)	[Emax/Emin]	1.2734/ 1.2334
Tooth thickness tolerance, normal section (mm)	[Tol.Smax/min]	-0.0170/-0.0450
Tooth space tolerance, normal section (mm)	[Tol.Emax/min]	0.0630/ 0.0230
Base tangent length (mm)	[Wk.Smax/Smin]	13.1825/13.1583
Base tangent length (mm)	[Wk.Smax/Smin]	13.2518/13.2172
Diametral two ball measure (mm)	[MRe/Mri-ball]	29.3891/29.3437
Diametral two ball measure (mm)	[MRe/Mri-ball]	24.8646/24.7870
Diametral measurement over pins (mm)	[MRe/Mri-pin]	29.3891/29.3437
Diametral measurement over pins (mm)	[MRe/Mri-pin]	24.8646/24.7870

3.5 Data for Effective Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Svmax/min]	1.2104/ 1.1934
Tooth thickness / Spacewidth (mm)	[Evmax/min]	1.2334/ 1.2104
Tooth thickness tolerance, normal section (mm)	[Tol.Svmax/min]	0.0000/-0.0170
Tooth space tolerance, normal section (mm)	[Tol.Evmax/min]	0.0230/ 0.0000
Base tangent length (mm)	[Wk.Svmax/min]	13.1972/13.1825
Base tangent length (mm)	[Wk.Svmax/min]	13.2172/13.1972
Diametral two ball measure (mm)	[MRe/Mri-ball]	29.4165/29.3891
Diametral two ball measure (mm)	[MRe/Mri-ball]	24.7870/24.7420
Diametral measurement over pins (mm)	[MRe/Mri-pin]	29.4165/29.3891
Diametral measurement over pins (mm)	[MRe/Mri-pin]	24.7870/24.7420

Tolerance data DIN 5480-1 (mm)	[TG]	0.0450	0.0630
(mm)	[Tact]	0.0280	0.0400
(mm)	[Teff]	0.0170	0.0230

Circumferential backlash, transverse section:

-Theoretical, without form errors (mm)	[jt.act]	0.1080/0.0400
-Effective, with form errors (mm)	[jt.eff]	0.0400/0.0000

Normal backlash:

-Theoretical, without form errors (mm)	[jn.act]	0.0935/0.0346
-Effective, with form errors (mm)	[jn.eff]	0.0346/0.0000

Radial backlash:

-Theoretical, without form errors (mm)	[jr.act]	0.0468/0.0173
-Effective, with form errors (mm)	[jr.eff]	0.0173/0.0000

Note: Take the values in "Actual dimensions" into account when checking splines with individual measurements (base tangent length/ measurement over pins)..

4 Toothing tolerances

		----- Shaft -----	Hub -----
According to	DIN 5480:2006:		
Tolerance class	[Q-DIN5480]	8	9
Total profile deviation (µm)	[Fa]	15.0	21.0
Total helix deviation (µm)	[Fb]	9.0	11.0
Single pitch deviation (µm)	[fp]	12.0	17.0
Total cumulative pitch deviation (µm)	[Fp]	28.0	40.0
Runout (µm)	[Fr]	30.0	30.0

5 Strength calculation

Calculation method:	G. Niemann, Machine Elements I, 4th edition.	
Centering:	flank centered	
Supporting length (mm)	[l _{tr}]	30.00
Maximal circumferential force (N)	[F _t]	22861.36
Maximal circumferential force per tooth (N)	[F _t /z]	672.39
Load application diameter (mm)	[d _m]	27.12
Tooth height (mm)	[h]	0.59
Distance a ₀ (mm)	[a ₀]	15.00
Length factor	[k _λ]	1.18
Share factor, equivalent load	[k _φ β _q]	2.00
Share factor, maximum load	[k _φ β _{max}]	1.70

The share factors k_φβ according to Niemann are determined according to the tolerance class specified in DIN 5480..

[Q] 9

Nominal torque (Nm)	[T _{enn}]	183.00
Application factor	[K _A]	1.25
Service torque (Nm)	[T _{eq}]	228.75
Maximum torque (Nm)	[T _{max}]	310.00
Torque curve:	No alternating torque	
Number of load peaks	[N _L]	1000
Number of change of load direction	[N _W]	1
Load direction changing coefficient	[f _w]	1.00
Tolerance field according to DIN 5480	"H9"	

5.1 Shaft

Width on shaft (mm)	[L_W]	30.00
Supporting surface (mm²)	[Flw=ltr*h*z]	601.80
Tip form diameter (mm)	[dFa1.1]	27.71
Equivalent effective surface pressure (N/mm²)	[peq]	65.88
Maximum effective surface pressure (N/mm²)	[pmax]	75.88
Support factor	[fs]	1.30
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm²)	[pzuleq]	1113.77
Permissible pressure (N/mm²)	[pzulmax]	1670.66
fw * pzul / peq		16.91
fL * pzul / pmax		22.02
Required safety	[Smin]	1.00
Minimal safety	[S]	16.91

5.2 Hub

Width on hub (mm)	[L_N]	30.00
Supporting surface (mm²)	[Fln=ltr*h*z]	601.80
Tip form diameter (mm)	[dFa2.1]	-26.53
Small external diameter (mm)	[D1]	40.00
Big external diameter (mm)	[D2]	40.00
Width of hub-part with D2 (mm)	[c]	30.00
Equivalent diameter hub (mm)	[D]	40.00
Equivalent effective surface pressure (N/mm²)	[peq]	65.88
Maximum effective surface pressure (N/mm²)	[pmax]	75.88
Support factor	[fs]	1.50
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm²)	[pzuleq]	1104.00
Permissible pressure (N/mm²)	[pzulmax]	1656.00
fw * pzul / peq		16.76
fL * pzul / pmax		21.82
Required safety	[Smin]	1.00
Minimal safety	[S]	16.76

6 Supplementary data

Moment of inertia (System referenced to wheel 1):

calculation without consideration of the exact tooth shape

Single gears ((da+df)/2...di) (kg*m²)	[J]	1.222e-05	4.628e-05
System, (da+df)/2...di (kg*m²)	[J]	5.85e-05	

7 Modifying and defining the tooth form**7.1 Data for the tooth form calculation**

Data not available.

Please run the calculation in the "Tooth form" tab and open the main report again.

8 Remarks

- Specifications with [e/l] imply: Maximum [e] and minimum value [l] for
Taking all tolerances into account
Specifications with [m] imply: Mean value within tolerance
- Concerning the calculation method:
 $h = (dFa1.i - ABS(dFa2.i)) / 2.0$
 $dm = (dFa1.i + ABS(dFa2.i)) / 2.0;$
 $Ft = Mmax * 2000 / dm$
Pressure load: $p(eq,max) = k\phi\beta(eq,max)*k1*M*2000/(dm*t*h*z); pmax \geq peq$
If $pmax < peq$, $pmax = peq$ is set.
Coefficient for load direction changes according to DIN 6892:2012/ fig. 7
 $pzuleq = fs*fh*fw*(Rm,Rp)$
 $pzulmax = fs*fh*fl*(Rm,Rp)$
Rm for brittle material; Rp for ductile material

End of Report

lines: 304

B.2 Small sun gear shaft - gear coupling spline

Spline

Spline Small sun gear coupling

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1 Messages

No messages occurred during the calculation.

2 Overview

Shaft DIN 5480 - W 25.00x0.50x48x8-
 Hub DIN 5480 - N 25.00x0.50x48x9-
 Drawing or article number:
 Shaft: 0.000.0
 Hub: 0.000.0

3 Tooth geometry and material

Normal module (mm)	[mn]	0.5000
Reference diameter (mm)	[dB]	25.00
Normal pressure angle (°)	[on]	30.000
		----- Shaft ----- Hub -----
Number of teeth	[z]	48 -48
Helix angle at reference circle (°)	[β]	0.0000
Facewidth (mm)	[b]	23.00 23.00
Hand of gear		Spur gear

3.1 Material

Gear 1
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC
 Gear 2
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

		----- Shaft ----- Hub ----
Surface hardness		HRC 61 HRC 61
Reference diameter material (mm)	[dB]	25 35
Tensile strength (N/mm²)	[σB]	1100.00 1100.00
Yield point (N/mm²)	[σS]	745.00 745.00
Young's modulus (N/mm²)	[E]	206000 206000
Poisson's ratio	[ν]	0.300 0.300
Roughness average value DS, flank (μm)	[RAH]	0.00 0.00
Roughness average value DS, root (μm)	[RAF]	0.00 0.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80 4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00 20.00

3.2 Reference profiles

Gear reference profile 1:
 Reference profile DIN 5480:2006 0.60 / 0.16 / 0.45 Hobbing
 Dedendum coefficient [hfP*] 0.600
 Root radius factor [pfP*] 0.160 (pfPmax*=0.760)

Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	
Gear reference profile 2:		
Reference profile	DIN 5480:2006 0.65 / 0.16 / 0.45 Shaping	
Dedendum coefficient	[hfP*]	0.650
Root radius factor	[pfP*]	0.160 (pfPmax*=0.710)
Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	

3.2.1 Information on final machining

Dedendum reference profile	[hfP*]	0.600	0.650
Tooth root radius Refer. profile	[pfP*]	0.160	0.160
Addendum Reference profile	[haP*]	0.450	0.450
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[αprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[αKP]	0.000	0.000

Transverse module (mm)	[mt]	0.500
Transverse pressure angle (°)	[αt]	30.000
Base helix angle (°)	[βb]	0.000
Sum of profile shift coefficients	[Σxi]	0.0000

		----- Shaft -----	Hub -----
Profile shift coefficient	[x]	0.4500	-0.4500
Profile shift (x*m) (mm)	[x*m]	0.2250	-0.2250
Pitch diameter (mm)	[d]	24.000	24.000
Base diameter (mm)	[db]	20.785	20.785
Tip diameter (mm)	[da]	24.900	24.000
Effective tip diameter (mm)	[da.e/i]	24.900 /24.770	
Effective tip diameter (mm)	[da.e/i]		24.000 /24.130
Tip diameter allowances (mm)	[Ada.e/i]	0.000 /-0.130	
Tip diameter allowances (mm)	[Ada.e/i]		-0.000 / 0.130
Root diameter (mm)	[df]	23.850	25.100
Effective root diameter (mm)	[df.e/i]	23.824 /23.781	
Effective root diameter (mm)	[df.e/i]		25.135 /25.197
Root diameter allowances (mm)	[Adf.e/i]	-0.026 /-0.069	
Root diameter allowances (mm)	[Adf.e/i]		0.035 / 0.097
Generating Profile shift coefficient	[xE.e/i]	0.4240/0.3807	
Generating Profile shift coefficient	[xE.e/i]		-0.4846/-0.5470
Root form diameter (mm)	[dFf]	23.930	25.029
Effective root form diameter (mm)	[dFf.e/i]	23.905 /23.862	
Effective root form diameter (mm)	[dFf.e/i]		25.064 /25.127
(dFf2 calculated on the basis of the generated involute.)			
Root diameter allowances in accordance with DIN 5480 -1, Table 5:			
Root form diameter (mm)	[dFf]	23.944	24.956
Effective root form diameter (mm)	[dFf.e/i]	23.944 /23.901	
Effective root form diameter (mm)	[dFf.e/i]		24.956 /25.018

Active tip diameter (mm)	[dNa.e/l]	24.900 /24.770
Active tip diameter (mm)	[dNa.e/l]	24.000 /24.130
Active root diameter (mm)	[dNf.e/l]	24.000 /24.130
Active root diameter (mm)	[dNf.e/l]	24.900 /24.770
Tooth height (mm)	[h]	0.525 0.550
Theoretical tip clearance (mm)	[c]	0.100 0.075
Effective tip clearance (mm)	[c.e/l]	0.213 / 0.117
Effective tip clearance (mm)	[c.e/l]	0.175 / 0.088
Normal tooth thickness at tip circle (mm)	[san]	0.517 0.526
(mm)	[san.e/l]	0.584 / 0.476
(mm)	[san.e/l]	0.584 / 0.470
Normal space width at root circle (mm)	[efn]	0.437 0.387
(mm)	[efn.e/l]	0.437 / 0.437
(mm)	[efn.e/l]	0.385 / 0.381
Pitch on reference circle (mm)	[pt]	1.571
Base pitch (mm)	[pbt]	1.360
Transverse pitch on contact-path (mm)	[pet]	1.360

3.3 Measurements for tooth thickness

		----- Shaft -----	Hub -----
Tolerance class		8	9
Tooth thickness tolerance		No allowance	No allowance
Number of teeth spanned	[k]	9.0000	9.0000
Base tangent length (no backlash) (mm)	[Wk]	12.9052	12.9052
Diameter of measuring circle (mm)	[dMWk.m]	24.4617	24.4697
Theoretical diameter of ball/pin (mm)	[dm]	0.9699	0.9147
Effective diameter of ball/pin (mm)	[DMeff]	1.0000	1.0000
Theor. dimension over two balls (mm)	[MRe/Mri-ball]	26.0286	22.8080
Diametral measurement over pins without clearance (mm)	[MRe/Mri-pin]	26.0286	22.8080

3.4 Data for Actual Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Smax/Smin]	1.0302/ 1.0052
Tooth thickness / Spacewidth (mm)	[Emax/Emin]	1.1012/ 1.0652
Tooth thickness tolerance, normal section (mm)	[Tol.Smax/min]	-0.0150/-0.0400
Tooth space tolerance, normal section (mm)	[Tol.Emax/min]	0.0560/ 0.0200
Base tangent length (mm)	[Wk.Smax/Smin]	12.8922/12.8705
Base tangent length (mm)	[Wk.Smax/Smin]	12.9537/12.9225
Diametral two ball measure (mm)	[MRe/Mri-ball]	26.0053/25.9663
Diametral two ball measure (mm)	[MRe/Mri-ball]	22.9068/22.8434
Diametral measurement over pins (mm)	[MRe/Mri-pin]	26.0053/25.9663
Diametral measurement over pins (mm)	[MRe/Mri-pin]	22.9068/22.8434

3.5 Data for Effective Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Svmax/min]	1.0452/ 1.0302
Tooth thickness / Spacewidth (mm)	[Evmax/min]	1.0652/ 1.0452
Tooth thickness tolerance, normal section (mm)	[Tol.Svmax/min]	0.0000/-0.0150
Tooth space tolerance, normal section (mm)	[Tol.Evmax/min]	0.0200/ 0.0000
Base tangent length (mm)	[Wk.Svmax/min]	12.9052/12.8922
Base tangent length (mm)	[Wk.Svmax/min]	12.9225/12.9052
Diametral two ball measure (mm)	[MRe/Mri-ball]	26.0286/26.0053
Diametral two ball measure (mm)	[MRe/Mri-ball]	22.8434/22.8080
Diametral measurement over pins (mm)	[MRe/Mri-pin]	26.0286/26.0053
Diametral measurement over pins (mm)	[MRe/Mri-pin]	22.8434/22.8080

Tolerance data DIN 5480-1 (mm)	[TG]	0.0400	0.0560
(mm)	[Tact]	0.0250	0.0360
(mm)	[Teff]	0.0150	0.0200

Circumferential backlash, transverse section:

-Theoretical, without form errors (mm)	[jt.act]	0.0960/0.0350
-Effective, with form errors (mm)	[jt.eff]	0.0350/0.0000

Normal backlash:

-Theoretical, without form errors (mm)	[jn.act]	0.0831/0.0303
-Effective, with form errors (mm)	[jn.eff]	0.0303/0.0000

Radial backlash:

-Theoretical, without form errors (mm)	[jr.act]	0.0416/0.0152
-Effective, with form errors (mm)	[jr.eff]	0.0152/0.0000

Note: Take the values in "Actual dimensions" into account when checking splines with individual measurements (base tangent length/ measurement over pins)..

4 Toothing tolerances

		----- Shaft -----	Hub -----
According to	DIN 5480:2006:		
Tolerance class	[Q-DIN5480]	8	9
Total profile deviation (µm)	[Fa]	13.0	19.0
Total helix deviation (µm)	[Fb]	9.0	10.0
Single pitch deviation (µm)	[fp]	11.0	15.0
Total cumulative pitch deviation (µm)	[Fp]	25.0	36.0
Runout (µm)	[Fr]	30.0	30.0

5 Strength calculation

Calculation method:	G. Niemann, Machine Elements I, 4th edition.	
Centering:	flank centered	
Supporting length (mm)	[ltr]	23.00
Maximal circumferential force (N)	[Ft]	25357.87
Maximal circumferential force per tooth (N)	[Ft/z]	528.29
Load application diameter (mm)	[dm]	24.45
Tooth height (mm)	[h]	0.32
Distance a0 (mm)	[a0]	20.00
Length factor	[kλ]	1.08
Share factor, equivalent load	[kφβq]	2.00
Share factor, maximum load	[kφβmax]	1.70

The share factors $k_{\phi\beta}$ according to Niemann are determined according to the tolerance class specified in DIN 5480..

[Q] 9

Nominal torque (Nm)	[Tnenn]	110.00
Application factor	[KA]	1.25
Service torque (Nm)	[Teq]	137.50
Maximum torque (Nm)	[Tmax]	310.00
Torque curve:	No alternating torque	
Number of load peaks	[NL]	1000
Number of change of load direction	[NW]	1
Load direction changing coefficient	[fw]	1.00
Tolerance field according to DIN 5480	"H9"	

5.1 Shaft

Width on shaft (mm)	[L_W]	23.00
Supporting surface (mm ²)	[Flw=ltr*h*z]	353.28
Tip form diameter (mm)	[dFa1.1]	24.77
Equivalent effective surface pressure (N/mm ²)	[peq]	68.92
Maximum effective surface pressure (N/mm ²)	[pmax]	132.08
Support factor	[fs]	1.30
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm ²)	[pzuleq]	1113.77
Permissible pressure (N/mm ²)	[pzulmax]	1670.66
fw * pzul / peq		16.16
fL * pzul / pmax		12.65
Required safety	[Smin]	1.00
Minimal safety	[S]	12.65

5.2 Hub

Width on hub (mm)	[L_N]	23.00
Supporting surface (mm ²)	[Fln=ltr*h*z]	353.28
Tip form diameter (mm)	[dFa2.1]	-24.13
Small external diameter (mm)	[D1]	40.00
Big external diameter (mm)	[D2]	30.00
Width of hub-part with D2 (mm)	[c]	23.00
Equivalent diameter hub (mm)	[D]	30.00
Equivalent effective surface pressure (N/mm ²)	[peq]	68.92
Maximum effective surface pressure (N/mm ²)	[pmax]	132.08
Support factor	[fs]	1.50
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm ²)	[pzuleq]	1285.13
Permissible pressure (N/mm ²)	[pzulmax]	1927.69
fw * pzul / peq		18.65
fL * pzul / pmax		14.59
Required safety	[Smin]	1.00
Minimal safety	[S]	14.59

6 Supplementary data

Moment of inertia (System referenced to wheel 1):

calculation without consideration of the exact tooth shape

Single gears ((da+df)/2...di) (kg*m ²)	[J]	6.184e-06	7.83e-06
System, (da+df)/2...di (kg*m ²)	[J]	1.401e-05	

7 Modifying and defining the tooth form**7.1 Data for the tooth form calculation**

Data not available.

Please run the calculation in the "Tooth form" tab and open the main report again.

8 Remarks

- Specifications with [e/l] imply: Maximum [e] and minimum value [l] for
Taking all tolerances into account
Specifications with [m] imply: Mean value within tolerance

- Concerning the calculation method:

$$h = (dFa1.i - ABS(dFa2.i)) / 2.0$$

$$dm = (dFa1.i + ABS(dFa2.i)) / 2.0;$$

$$Ft = Mmax * 2000 / dm$$
 Pressure load: $p(eq,max) = k\phi\beta(eq,max)*k1*M*2000/(dm*t*h*z); pmax \geq peq$
 If $pmax < peq$, $pmax = peq$ is set.
 Coefficient for load direction changes according to DIN 6892:2012/ fig. 7
 $pzuleq = fs*fh*fw*(Rm,Rp)$
 $pzulmax = fs*fh*fl*(Rm,Rp)$
 Rm for brittle material; Rp for ductile material

End of Report

lines: 304

B.3 Large sun gear shaft - gear coupling spline spline

KISSsoft

Spline

Spline Large sun gear coupling

Changed on

04.07.2022, 13:24:53

Changed on

igorm

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1 Messages

Messages occurred during the calculation:

- i** Normally the force is absorbed in the middle of the notch.
 a_0 would then be the supporting length/2 = 6.50 mm.
 Your entry is probably incorrect.

2 Overview

Shaft DIN 5480 - W 26.00x0.80x31x8h
 Hub DIN 5480 - N 26.00x0.80x31x9H

Drawing or article number:

Shaft: 0.000.0
 Hub: 0.000.0

3 Tooth geometry and material

Normal module (mm)	[mn]	0.8000
Reference diameter (mm)	[dB]	26.00
Normal pressure angle (°)	[on]	30.000
		----- Shaft ----- Hub -----
Number of teeth	[z]	31 -31
Helix angle at reference circle (°)	[β]	0.0000
Facewidth (mm)	[b]	13.00 13.00
Hand of gear		Spur gear

3.1 Material

Gear 1

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

Gear 2

18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

		----- Shaft -----	Hub ----
		HRC 61	HRC 61
Surface hardness		26	35
Reference diameter material (mm)	[dB]	26	35
Tensile strength (N/mm ²)	[σB]	1100.00	1100.00
Yield point (N/mm ²)	[σS]	745.00	745.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.00	0.00
Roughness average value DS, root (μm)	[RAF]	0.00	0.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

3.2 Reference profiles

Gear reference profile 1:		
Reference profile	DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching	
Dedendum coefficient	[hfP*]	0.550
Root radius factor	[pfP*]	0.160 (pfPmax*=0.810)
Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
not topping		

Gear reference profile 2:		
Reference profile	DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching	
Dedendum coefficient	[hfP*]	0.550
Root radius factor	[pfP*]	0.160 (pfPmax*=0.810)
Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
not topping		

3.2.1 Information on final machining

Dedendum reference profile	[hfP*]	0.550	0.550
Tooth root radius Refer. profile	[pfP*]	0.160	0.160
Addendum Reference profile	[haP*]	0.450	0.450
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[αprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[αKP]	0.000	0.000

Transverse module (mm)	[mt]	0.800
Transverse pressure angle (°)	[αt]	30.000
Base helix angle (°)	[βb]	0.000
Sum of profile shift coefficients	[Σxi]	0.0000

		----- Shaft -----	Hub -----
Profile shift coefficient	[x]	0.2000	-0.2000
Profile shift (x*m) (mm)	[x*m]	0.1600	-0.1600
Pitch diameter (mm)	[d]	24.800	24.800
Base diameter (mm)	[db]	21.477	21.477
Tip diameter (mm)	[da]	25.840	24.400
Effective tip diameter (mm)	[da.e/i]	25.840 /25.710	
Effective tip diameter (mm)	[da.e/i]		24.400 /24.530
Tip diameter allowances (mm)	[Ada.e/i]	0.000 +/-0.130	
Tip diameter allowances (mm)	[Ada.e/i]		-0.000 / 0.130
Root diameter (mm)	[df]	24.240	26.000
Effective root diameter (mm)	[df.e/i]	24.211 /24.162	
Effective root diameter (mm)	[df.e/i]		26.040 /26.109
Root diameter allowances (mm)	[Adf.e/i]	-0.029 +/-0.078	
Root diameter allowances (mm)	[Adf.e/i]		0.040 / 0.109
Generating Profile shift coefficient	[xE.e/i]	0.1816/0.1513	
Generating Profile shift coefficient	[xE.e/i]		-0.2249/-0.2682
Root form diameter (mm)	[dFf]	24.379	25.886
Effective root form diameter (mm)	[dFf.e/i]	24.352 /24.306	
Effective root form diameter (mm)	[dFf.e/i]		25.926 /25.996
(dFf2 calculated on the basis of the generated involute.)			

Root diameter allowances in accordance with DIN 5480 -1, Table 5:

Root form diameter (mm)	[dFf]	24.340	25.900
Effective root form diameter (mm)	[dFf.e/i]	24.340 /24.292	
Effective root form diameter (mm)	[dFf.e/i]		25.900 /25.969
Active tip diameter (mm)	[dNa.e/i]	25.840 /25.710	
Active tip diameter (mm)	[dNa.e/i]		24.400 /24.530
Active root diameter (mm)	[dNf.e/i]	24.400 /24.530	
Active root diameter (mm)	[dNf.e/i]		25.840 /25.710
Tooth height (mm)	[h]	0.800	0.800
Theoretical tip clearance (mm)	[c]	0.080	0.080
Effective tip clearance (mm)	[c.e/i]	0.200 / 0.100	
Effective tip clearance (mm)	[c.e/i]		0.184 / 0.095
Normal tooth thickness at tip circle (mm)	[san]	0.840	0.833
(mm)	[san.e/i]	0.904 / 0.793	
(mm)	[san.e/i]		0.886 / 0.771
Normal space width at root circle (mm)	[efn]	0.743	0.737
(mm)	[efn.e/i]	0.743 / 0.744	
(mm)	[efn.e/i]		0.735 / 0.731
Pitch on reference circle (mm)	[pt]	2.513	
Base pitch (mm)	[pbt]	2.177	
Transverse pitch on contact-path (mm)	[pet]	2.177	

3.3 Measurements for tooth thickness

		----- Shaft -----	Hub -----
Tolerance class		8	9
Tooth thickness tolerance		DIN 5480 h	DIN 5480 H
Number of teeth spanned	[k]	6.0000	6.0000
Base tangent length (no backlash) (mm)	[Wk]	13.2855	13.2855
Diameter of measuring circle (mm)	[dMWk.m]	25.2505	25.2596
Theoretical diameter of ball/pin (mm)	[dm]	1.5417	1.4416
Effective diameter of ball/pin (mm)	[DMeff]	1.7500	1.5000
Theor. dimension over two balls (mm)	[MRe/Mri-ball]	28.0349	22.7483
Diametral measurement over pins without clearance (mm)	[MRe/Mri-pin]	28.0349	22.7483

3.4 Data for Actual Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Smax/Smin]	1.4244/ 1.3964
Tooth thickness / Spacewidth (mm)	[Emax/Emin]	1.5044/ 1.4644
Tooth thickness tolerance, normal section (mm)	[Tol.Smax/min]	-0.0170/-0.0450
Tooth space tolerance, normal section (mm)	[Tol.Emax/min]	0.0630/ 0.0230
Base tangent length (mm)	[Wk.Smax/Smin]	13.2708/13.2465
Base tangent length (mm)	[Wk.Smax/Smin]	13.3401/13.3054
Diametral two ball measure (mm)	[MRe/Mri-ball]	28.0094/27.9674
Diametral two ball measure (mm)	[MRe/Mri-ball]	22.8641/22.7908
Diametral measurement over pins (mm)	[MRe/Mri-pin]	28.0094/27.9674
Diametral measurement over pins (mm)	[MRe/Mri-pin]	22.8641/22.7908

3.5 Data for Effective Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Svmax/min]	1.4414/ 1.4244
Tooth thickness / Spacewidth (mm)	[Evmax/min]	1.4644/ 1.4414
Tooth thickness tolerance, normal section (mm)	[Tol.Svmax/min]	0.0000/-0.0170
Tooth space tolerance, normal section (mm)	[Tol.Evmax/min]	0.0230/ 0.0000
Base tangent length (mm)	[Wk.Svmax/min]	13.2855/13.2708
Base tangent length (mm)	[Wk.Svmax/min]	13.3054/13.2855

Diametral two ball measure (mm)	[MRe/Mri-ball]	28.0349/28.0094
Diametral two ball measure (mm)	[MRe/Mri-ball]	22.7908/22.7483
Diametral measurement over pins (mm)	[MRe/Mri-pin]	28.0349/28.0094
Diametral measurement over pins (mm)	[MRe/Mri-pin]	22.7908/22.7483

Tolerance data DIN 5480-1 (mm)	[TG]	0.0450	0.0630
(mm)	[Tact]	0.0280	0.0400
(mm)	[Teff]	0.0170	0.0230

Circumferential backlash, transverse section:

-Theoretical, without form errors (mm)	[jt.act]	0.1080/0.0400
-Effective, with form errors (mm)	[jt.eff]	0.0400/0.0000

Normal backlash:

-Theoretical, without form errors (mm)	[jn.act]	0.0935/0.0346
-Effective, with form errors (mm)	[jn.eff]	0.0346/0.0000

Radial backlash:

-Theoretical, without form errors (mm)	[jr.act]	0.0468/0.0173
-Effective, with form errors (mm)	[jr.eff]	0.0173/0.0000

Note: Take the values in "Actual dimensions" into account when checking splines with individual measurements (base tangent length/ measurement over pins)..

4 Toothing tolerances

		----- Shaft -----	Hub -----
According to	DIN 5480:2006:		
Tolerance class	[Q-DIN5480]	8	9
Total profile deviation (µm)	[Fa]	15.0	21.0
Total helix deviation (µm)	[Fb]	9.0	11.0
Single pitch deviation (µm)	[fp]	12.0	17.0
Total cumulative pitch deviation (µm)	[Fp]	28.0	40.0
Runout (µm)	[Fr]	30.0	30.0

5 Strength calculation

Calculation method: G. Niemann, Machine Elements I, 4th edition.

Centering:	flank centered	
Supporting length (mm)	[ltr]	13.00
Maximal circumferential force (N)	[Ft]	24681.53
Maximal circumferential force per tooth (N)	[Ft/z]	796.18
Load application diameter (mm)	[dm]	25.12
Tooth height (mm)	[h]	0.59
Distance a0 (mm)	[a0]	0.00
Length factor	[kλ]	1.10
Share factor, equivalent load	[kφβq]	2.00
Share factor, maximum load	[kφβmax]	1.70

The share factors $k\phi\beta$ according to Niemann are determined according to the tolerance class specified in DIN 5480..

[Q] 9

Nominal torque (Nm)	[Tnenn]	182.00
Application factor	[KA]	1.25
Service torque (Nm)	[Teq]	227.50
Maximum torque (Nm)	[Tmax]	310.00
Torque curve:	No alternating torque	
Number of load peaks	[NL]	1000
Number of change of load direction	[NW]	1

KISSsoft		
Load direction changing coefficient	[fw]	1.00
Tolerance field according to DIN 5480	"H9"	
5.1 Shaft		
Width on shaft (mm)	[l_W]	13.00
Supporting surface (mm²)	[Flw=ltr*h*z]	237.77
Tip form diameter (mm)	[dFa1.i]	25.71
Equivalent effective surface pressure (N/mm²)	[peq]	167.67
Maximum effective surface pressure (N/mm²)	[pmax]	194.20
Support factor	[fs]	1.30
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm²)	[pzuleq]	1113.77
Permissible pressure (N/mm²)	[pzulmax]	1670.66
fw * pzul / peq		6.64
fL * pzul / pmax		8.60
Required safety	[Smin]	1.00
Minimal safety	[S]	6.64
5.2 Hub		
Width on hub (mm)	[l_N]	13.00
Supporting surface (mm²)	[Fln=ltr*h*z]	237.77
Tip form diameter (mm)	[dFa2.i]	-24.53
Small external diameter (mm)	[D1]	30.00
Big external diameter (mm)	[D2]	30.00
Width of hub-part with D2 (mm)	[c]	13.00
Equivalent diameter hub (mm)	[D]	30.00
Equivalent effective surface pressure (N/mm²)	[peq]	167.67
Maximum effective surface pressure (N/mm²)	[pmax]	194.20
Support factor	[fs]	1.50
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm²)	[pzuleq]	1285.13
Permissible pressure (N/mm²)	[pzulmax]	1927.69
fw * pzul / peq		7.66
fL * pzul / pmax		9.93
Required safety	[Smin]	1.00
Minimal safety	[S]	7.66
6 Supplementary data		
Moment of inertia (System referenced to wheel 1):		
calculation without consideration of the exact tooth shape		
Single gears ((da+df)/2...di) (kg*m²)	[J]	3.892e-06 4.02e-06
System, (da+df)/2...di (kg*m²)	[J]	7.911e-06
7 Modifying and defining the tooth form		
7.1 Data for the tooth form calculation		
Data not available.		
Please run the calculation in the "Tooth form" tab and open the main report again.		

8 Remarks

- Specifications with [..e/l] imply: Maximum [e] and minimum value [l] for
 Taking all tolerances into account
 Specifications with [..m] imply: Mean value within tolerance

- Concerning the calculation method:

$$h = (dFa1.i - ABS(dFa2.i)) / 2.0$$

$$dm = (dFa1.i + ABS(dFa2.i)) / 2.0;$$

$$Ft = Mmax * 2000 / dm$$
 Pressure load: $p(eq,max) = k\phi\beta(eq,max)*k1*M*2000/(dm*l*h*z); pmax \geq peq$
 If $pmax < peq$, $pmax = peq$ is set.
 Coefficient for load direction changes according to DIN 6892:2012/ fig. 7
 $pzuleq = fs*fh*fw*(Rm,Rp)$
 $pzulmax = fs*fh*fl*(Rm,Rp)$
 Rm for brittle material; Rp for ductile material

End of Report

lines:

306

B.4 Large sun gear shaft - brake drum coupling spline

Spline

Spline Large sun brake drum coupling

Changed on
Changed on

04.07.2022, 13:25:22
igorm

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1 Messages

No messages occurred during the calculation.

2 Overview

Shaft DIN 5480 - W 31.00x1.25x23x8h
 Hub DIN 5480 - N 31.00x1.25x23x9H

Drawing or article number:
 Shaft: 0.000.0
 Hub: 0.000.0

3 Tooth geometry and material

Normal module (mm)	[mn]	1.2500
Reference diameter (mm)	[dB]	31.00
Normal pressure angle (°)	[on]	30.000
		----- Shaft ----- Hub -----
Number of teeth	[z]	23 -23
Helix angle at reference circle (°)	[β]	0.0000
Facewidth (mm)	[b]	40.00 40.00
Hand of gear		Spur gear

3.1 Material

Gear 1
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

Gear 2
 18CrNiMo7-6, Case-carburized steel, case-hardened
 ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

		----- Shaft ----- Hub ----
Surface hardness		HRC 61 HRC 61
Reference diameter material (mm)	[dB]	35 45
Tensile strength (N/mm²)	[σB]	1100.00 900.00
Yield point (N/mm²)	[σS]	745.00 640.00
Young's modulus (N/mm²)	[E]	206000 206000
Poisson's ratio	[ν]	0.300 0.300
Roughness average value DS, flank (μm)	[RAH]	0.00 0.00
Roughness average value DS, root (μm)	[RAF]	0.00 0.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80 4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00 20.00

3.2 Reference profiles

Gear reference profile 1:
 Reference profile DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching
 Dedendum coefficient [hfP*] 0.550
 Root radius factor [pfP*] 0.160 (pfPmax*=0.810)

Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	
Gear reference profile 2:		
Reference profile	DIN 5480:2006 0.55 / 0.16 / 0.45 Broaching	
Dedendum coefficient	[hfP*]	0.550
Root radius factor	[pfP*]	0.160 (pfPmax*=0.810)
Addendum coefficient	[haP*]	0.450
Tip radius factor	[paP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[αprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[αKP]	0.000
	not topping	

3.2.1 Information on final machining

Dedendum reference profile	[hfP*]	0.550	0.550
Tooth root radius Refer. profile	[pfP*]	0.160	0.160
Addendum Reference profile	[haP*]	0.450	0.450
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[αprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[αKP]	0.000	0.000

Transverse module (mm)	[mt]	1.250
Transverse pressure angle (°)	[αt]	30.000
Base helix angle (°)	[βb]	0.000
Sum of profile shift coefficients	[Σxi]	0.0000

		----- Shaft -----	Hub -----
Profile shift coefficient	[x]	0.3500	-0.3500
Profile shift (x*m) (mm)	[x*m]	0.4375	-0.4375
Pitch diameter (mm)	[d]	28.750	28.750
Base diameter (mm)	[db]	24.898	24.898
Tip diameter (mm)	[da]	30.750	28.500
Effective tip diameter (mm)	[da.e/i]	30.750 / 30.590	
Effective tip diameter (mm)	[da.e/i]		28.500 / 28.630
Tip diameter allowances (mm)	[Ada.e/i]	0.000	-0.160
Tip diameter allowances (mm)	[Ada.e/i]		-0.000 / 0.130
Root diameter (mm)	[df]	28.250	31.000
Effective root diameter (mm)	[df.e/i]	28.221 / 28.172	
Effective root diameter (mm)	[df.e/i]		31.040 / 31.109
Root diameter allowances (mm)	[Adf.e/i]	-0.029	-0.078
Root diameter allowances (mm)	[Adf.e/i]		0.040 / 0.109
Generating Profile shift coefficient	[xE.e/i]	0.3382 / 0.3188	
Generating Profile shift coefficient	[xE.e/i]		-0.3659 / -0.3936
Root form diameter (mm)	[dFf]	28.455	30.834
Effective root form diameter (mm)	[dFf.e/i]	28.426 / 28.380	
Effective root form diameter (mm)	[dFf.e/i]		30.875 / 30.945
(dFf2 calculated on the basis of the generated involute.)			
Root diameter allowances in accordance with DIN 5480 -1, Table 5:			
Root form diameter (mm)	[dFf]	28.440	30.810
Effective root form diameter (mm)	[dFf.e/i]	28.440 / 28.392	
Effective root form diameter (mm)	[dFf.e/i]		30.810 / 30.879

Active tip diameter (mm)	[dNa.e/l]	30.750 / 30.590
Active tip diameter (mm)	[dNa.e/l]	28.500 / 28.630
Active root diameter (mm)	[dNf.e/l]	28.500 / 28.630
Active root diameter (mm)	[dNf.e/l]	30.750 / 30.590
Tooth height (mm)	[h]	1.250 1.250
Theoretical tip clearance (mm)	[c]	0.125 0.125
Effective tip clearance (mm)	[c.e/l]	0.260 / 0.145
Effective tip clearance (mm)	[c.e/l]	0.229 / 0.140
Normal tooth thickness at tip circle (mm)	[san]	1.292 1.304
(mm)	[san.e/l]	1.382 / 1.244
(mm)	[san.e/l]	1.361 / 1.242
Normal space width at root circle (mm)	[efn]	1.157 1.118
(mm)	[efn.e/l]	1.157 / 1.156
(mm)	[efn.e/l]	1.115 / 1.109
Pitch on reference circle (mm)	[pt]	3.927
Base pitch (mm)	[pbt]	3.401
Transverse pitch on contact-path (mm)	[pet]	3.401

3.3 Measurements for tooth thickness

		----- Shaft -----	Hub -----
Tolerance class		8	9
Tooth thickness tolerance		DIN 5480 h	DIN 5480 H
Number of teeth spanned	[k]	5.0000	5.0000
Base tangent length (no backlash) (mm)	[Wk]	17.0797	17.0797
Diameter of measuring circle (mm)	[dMWk.m]	30.1892	30.1990
Theoretical diameter of ball/pin (mm)	[dm]	2.5517	2.2847
Effective diameter of ball/pin (mm)	[DMeff]	2.7500	2.5000
Theor. dimension over two balls (mm)	[MRe/Mri-ball]	34.0868	25.4291
Diametral measurement over pins without clearance (mm)	[MRe/Mri-pin]	34.0868	25.4291

3.4 Data for Actual Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Smax/Smin]	2.4517/ 2.4237
Tooth thickness / Spacewidth (mm)	[Emax/Emin]	2.5317/ 2.4917
Tooth thickness tolerance, normal section (mm)	[Tol.Smax/min]	-0.0170/-0.0450
Tooth space tolerance, normal section (mm)	[Tol.Emax/min]	0.0630/ 0.0230
Base tangent length (mm)	[Wk.Smax/Smin]	17.0650/17.0408
Base tangent length (mm)	[Wk.Smax/Smin]	17.1343/17.0997
Diametral two ball measure (mm)	[MRe/Mri-ball]	34.0627/34.0229
Diametral two ball measure (mm)	[MRe/Mri-ball]	25.5473/25.4725
Diametral measurement over pins (mm)	[MRe/Mri-pin]	34.0627/34.0229
Diametral measurement over pins (mm)	[MRe/Mri-pin]	25.5473/25.4725

3.5 Data for Effective Dimensions DIN 5480:2006

Tooth thickness / Spacewidth (mm)	[Svmax/min]	2.4687/ 2.4517
Tooth thickness / Spacewidth (mm)	[Evmax/min]	2.4917/ 2.4687
Tooth thickness tolerance, normal section (mm)	[Tol.Svmax/min]	0.0000/-0.0170
Tooth space tolerance, normal section (mm)	[Tol.Evmax/min]	0.0230/ 0.0000
Base tangent length (mm)	[Wk.Svmax/min]	17.0797/17.0650
Base tangent length (mm)	[Wk.Svmax/min]	17.0997/17.0797
Diametral two ball measure (mm)	[MRe/Mri-ball]	34.0868/34.0627
Diametral two ball measure (mm)	[MRe/Mri-ball]	25.4725/25.4291
Diametral measurement over pins (mm)	[MRe/Mri-pin]	34.0868/34.0627
Diametral measurement over pins (mm)	[MRe/Mri-pin]	25.4725/25.4291

Tolerance data DIN 5480-1 (mm)	[TG]	0.0450	0.0630
(mm)	[Tact]	0.0280	0.0400
(mm)	[Teff]	0.0170	0.0230

Circumferential backlash, transverse section:

-Theoretical, without form errors (mm)	[jt.act]	0.1080/0.0400
-Effective, with form errors (mm)	[jt.eff]	0.0400/0.0000

Normal backlash:

-Theoretical, without form errors (mm)	[jn.act]	0.0935/0.0346
-Effective, with form errors (mm)	[jn.eff]	0.0346/0.0000

Radial backlash:

-Theoretical, without form errors (mm)	[jr.act]	0.0468/0.0173
-Effective, with form errors (mm)	[jr.eff]	0.0173/0.0000

Note: Take the values in "Actual dimensions" into account when checking splines with individual measurements (base tangent length/ measurement over pins)..

4 Toothing tolerances

		----- Shaft -----	Hub -----
According to	DIN 5480:2006:		
Tolerance class	[Q-DIN5480]	8	9
Total profile deviation (µm)	[Fa]	15.0	21.0
Total helix deviation (µm)	[Fb]	9.0	11.0
Single pitch deviation (µm)	[fp]	12.0	17.0
Total cumulative pitch deviation (µm)	[Fp]	28.0	40.0
Runout (µm)	[Fr]	30.0	30.0

5 Strength calculation

Calculation method:	G. Niemann, Machine Elements I, 4th edition.	
Centering:	flank centered	
Supporting length (mm)	[ltr]	40.00
Maximal circumferential force (N)	[Ft]	21614.32
Maximal circumferential force per tooth (N)	[Ft/z]	939.75
Load application diameter (mm)	[dm]	29.61
Tooth height (mm)	[h]	0.98
Distance a0 (mm)	[a0]	20.00
Length factor	[kλ]	1.23
Share factor, equivalent load	[kφβq]	2.00
Share factor, maximum load	[kφβmax]	1.70

The share factors $k_{\phi\beta}$ according to Niemann are determined according to the tolerance class specified in DIN 5480..

[Q] 9

Nominal torque (Nm)	[Tnenn]	188.00
Application factor	[KA]	1.25
Service torque (Nm)	[Teq]	235.00
Maximum torque (Nm)	[Tmax]	320.00
Torque curve:	No alternating torque	
Number of load peaks	[NL]	1000
Number of change of load direction	[NW]	1
Load direction changing coefficient	[fw]	1.00
Tolerance field according to DIN 5480	"H9"	

5.1 Shaft

Width on shaft (mm)	[L_W]	40.00
Supporting surface (mm ²)	[Flw=ltr*h*z]	901.60
Tip form diameter (mm)	[dFa1.1]	30.59
Equivalent effective surface pressure (N/mm ²)	[peq]	43.37
Maximum effective surface pressure (N/mm ²)	[pmax]	50.20
Support factor	[fs]	1.30
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm ²)	[pzuleq]	1113.77
Permissible pressure (N/mm ²)	[pzulmax]	1670.66
fw * pzul / peq		25.68
fL * pzul / pmax		33.28
Required safety	[Smin]	1.00
Minimal safety	[S]	25.68

5.2 Hub

Width on hub (mm)	[L_N]	40.00
Supporting surface (mm ²)	[Fln=ltr*h*z]	901.60
Tip form diameter (mm)	[dFa2.1]	-28.63
Small external diameter (mm)	[D1]	40.00
Big external diameter (mm)	[D2]	40.00
Width of hub-part with D2 (mm)	[c]	40.00
Equivalent diameter hub (mm)	[D]	40.00
Equivalent effective surface pressure (N/mm ²)	[peq]	43.37
Maximum effective surface pressure (N/mm ²)	[pmax]	50.20
Support factor	[fs]	1.50
Load peak coefficient	[fL]	1.50
Hardness influence coefficient	[fH]	1.15
Permissible pressure (N/mm ²)	[pzuleq]	1104.00
Permissible pressure (N/mm ²)	[pzulmax]	1656.00
fw * pzul / peq		25.45
fL * pzul / pmax		32.99
Required safety	[Smin]	1.00
Minimal safety	[S]	25.45

6 Supplementary data

Moment of inertia (System referenced to wheel 1):

calculation without consideration of the exact tooth shape

Single gears ((da+df)/2...di) (kg*m ²)	[J]	2.308e-05	5.44e-05
System, (da+df)/2...di (kg*m ²)	[J]	7.748e-05	

7 Modifying and defining the tooth form**7.1 Data for the tooth form calculation**

Data not available.

Please run the calculation in the "Tooth form" tab and open the main report again.

8 Remarks

- Specifications with [e/l] imply: Maximum [e] and minimum value [l] for
 Taking all tolerances into account
 Specifications with [m] imply: Mean value within tolerance
 - Concerning the calculation method:

$$h = (dFa1.i - ABS(dFa2.i)) / 2.0$$

$$dm = (dFa1.i + ABS(dFa2.i)) / 2.0;$$

$$Ft = Mmax * 2000 / dm$$
 Pressure load: $p(eq,max) = k\phi\beta(eq,max)*k1*M*2000/(dm*t*h*z); pmax \geq peq$
 If $pmax < peq$, $pmax = peq$ is set.
 Coefficient for load direction changes according to DIN 6892:2012/ fig. 7
 $pzuleq = fs*fh*fw*(Rm,Rp)$
 $pzulmax = fs*fh*fl*(Rm,Rp)$
 Rm for brittle material; Rp for ductile material

End of Report

lines: 304

B.5 One-way clutch key

KISSsoft

Key

One way Clutch Key

Changed on
Changed on

04.07.2022, 13:25:51
igorm

KISSsoft – student license (not for commercial use) (581)
KISSsoft Release 2021 -SP1

KISSsoft		
Contents		
1	Messages	3
2	Overview	3
3	Shaft	3
4	Hub	4
5	Key	4
6	Remarks	5

1 Messages

No messages occurred during the calculation.

2 Overview

Calculation method: DIN 6892-B:2012

Label	DIN 6885.1:1968 Default	
Key width (mm)	[b]	18.00
Key height (mm)	[h]	11.00
Chamfer (mean value) (mm)	[r]	0.50
Shaft diameter (mm)	[d]	60.00
Nominal torque (Nm)	[T]	558.00
Application factor	[KA]	1.00
equivalent torque (Nm)	[Teq]	558.00
Maximum torque (Nm)	[Tmax]	950.67
Minimal frictional torque for interference fit (Nm)	[TRmin]	0.00
Torque curve: With alternating torque		
Maximum backwards torque (Nm)	[TmaxR]	0.00
Number of load peaks	[NL]	10000
Number of change of load direction	[NW]	1
Load direction changing coefficient	[fw]	1.00
Number of keys	[i]	1
Load factor	[phi]	1.00
equivalent circumferential stress (N)	[Feq]	18600.00
Maximal circumferential force (N)	[Fmax]	31689.00
Contact coefficient for equivalent surface pressure	[Kveq]	1.00
Contact coefficient for maximal surface pressure	[Kvmax]	1.00
Help coefficient	[Kle]	1.058
Load distribution coefficient	[KL]	1.058
Friction factor for equivalent torsional moment	[KReq]	1.000
Friction factor for maximum peak torsional moment	[KRmax]	1.000

3 Shaft

Material	C45 (1)			
Type	Through hardened steel			
Treatment	unalloyed, through hardened			
Tensile strength (N/mm²)	[Rm]	630.00	(d=40 -100 mm)	
Yield point (N/mm²)	[Re]	370.00	(d=40 -100 mm)	
Groove depth shaft (minimal value) (mm)	[t1]	7.00		
Chamfer on shaft (mm)	[s1]	0.01		
Supporting key length (mm)	[ltr]	35.00		
Supporting key height (mm)	[t1tr]	5.11		
Equivalent effective surface pressure (N/mm²)	[peq]	110.17		
Maximum effective surface pressure (N/mm²)	[pmax]	187.70		
Support factor	[fs]	1.30		

Hardness influence coefficient	[fH]	1.00
Permissible surface pressure (N/mm ²)	[pzul]	481.00
Load peak frequency coefficient	[fL]	1.40
Load direction changing coefficient	[fw]	1.00
Safety factor, equivalent load (fw*pzul/peq)	[SFeq]	4.37
Safety, maximum load (fL * pzul / pmax)	[SFmax]	3.59
Minimal safety	[SF]	3.59

4 Hub

Material	18CrNiMo7-6				
Type	Case-carburized steel				
Treatment	case-hardened				
Tensile strength (N/mm ²)	[Rm]	900.00	(d=40	-100	mm)
Yield point (N/mm ²)	[Re]	640.00	(d=40	-100	mm)
The material strength values are defined using the wall thickness as the raw diameter.					
Groove depth hub (minimal value) (mm)	[t2]	4.40			
Chamfer on hub (mm)	[s2]	0.01			
Supporting key length (mm)	[ltr]	35.00			
Supporting key height (mm)	[t2tr]	4.87			
Small outside diameter of hub (mm)	[D1]	80.00			
Big outside diameter of hub (mm)	[D2]	180.00			
Width of hub-part with D2 (mm)	[c]	35.00			
Equivalent diameter hub (mm)	[D]	180.00			
Distance a0 (Figure 2, DIN 6892) (mm)	[a0]	17.50			
Equivalent effective surface pressure (N/mm ²)	[peq]	115.37			
Maximum effective surface pressure (N/mm ²)	[pmax]	196.56			
Support factor	[fs]	1.50			
Hardness influence coefficient	[fH]	1.15			
Permissible surface pressure (N/mm ²)	[pzul]	1104.00			
Load peak frequency coefficient	[fL]	1.40			
Load direction changing coefficient	[fw]	1.00			
Safety factor, equivalent load (fw*pzul/peq)	[SFeq]	9.57			
Safety, maximum load (fL * pzul / pmax)	[SFmax]	7.86			
Minimal safety	[SF]	7.86			

5 Key

Material	C45 (1)				
Type	Through hardened steel				
Treatment	unalloyed, through hardened				
Tensile strength (N/mm ²)	[Rm]	700.00	(d= 0	-16	mm)
Yield point (N/mm ²)	[Re]	490.00	(d= 0	-16	mm)
Equivalent effective surface pressure (N/mm ²)	[peq]	110.17	/	115.37	
Maximum effective surface pressure (N/mm ²)	[pmax]	187.70	/	196.56	
Support factor	[fs]	1.10			
Hardness influence coefficient	[fH]	1.00			
Permissible surface pressure (N/mm ²)	[pzul]	539.00			
Load peak frequency coefficient	[fL]	1.40			
Load direction changing coefficient	[fw]	1.00			
Safety factor, equivalent load (fw*pzul/peq)	[SFeq]	4.67			
Safety, maximum load (fL * pzul / pmax)	[SFmax]	3.84			
Minimal safety	[SF]	3.84			
Cross section area (mm ²)	[b*ltr]	630.00			
Shear stress (N/mm ²)	[tau]	29.52			

KISSsoft		
6 Remarks		
Safety = Minimum (fw*pzul/peq, fL*pzul/pmax) Condition according to DIN 6892 Safety >= 1.0 Chamfer on key: Mean value as in examples in DIN 6892 Groove depth: Minimum value as in examples in DIN 6892		
End of Report	lines:	142

Appendix - Bearings & Seal

The following appendix consists of the rolling bearings product sheets from the SKF catalogue. The seal, which is also provided by SKF, is also listed here.

- Sun gear shafts - SKF 61806 and SKF 61805;
- Planet bearings - SKF NJ 202 ECP and SKF k 15x19x10;
- Lead screw shaft bearings - SKF 6000 and SKF 618/6;
- Thrust bearings - SKF AXK 4060, 90120 and 150190;
- Seal - SKF 30x42x6 HMS5 RG.

C.1 Sun gear shafts - SKF 61806 and SKF 61805



61806

Deep groove ball bearing

Single row deep groove ball bearings are particularly versatile, have low friction and are optimized for low noise and low vibration, which enables high rotational speeds. They accommodate radial and axial loads in both directions, are easy to mount, and require less maintenance than many other bearing types.

- Simple, versatile and robust design
- Low friction
- High-speed capability
- Accommodate radial and axial loads in both directions
- Require little maintenance

Overview

Dimensions

Bore diameter	30 mm
Outside diameter	42 mm
Width	7 mm

Performance

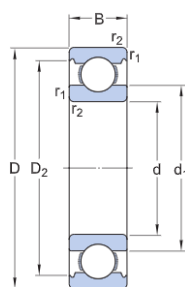
Basic dynamic load rating	4.1 kN
Basic static load rating	2.9 kN
Limiting speed	20000 r/min
Reference speed	32000 r/min

Properties

Bore type	Cylindrical
Cage	Sheet metal
Coating	Without
Filling slots	Without
Locating feature, bearing outer ring	None
Lubricant	None
Matched arrangement	No
Material, bearing	Bearing steel
Number of rows	1
Radial internal clearance	CN
Relubrication feature	Without
Sealing	Without

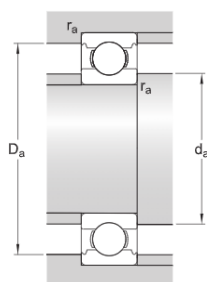


Technical Specification



Dimensions

d	30 mm	Bore diameter
D	42 mm	Outside diameter
B	7 mm	Width
d ₁	≈ 33.7 mm	Shoulder diameter
D ₁	≈ 38.35 mm	Shoulder diameter
D ₂	≈ 39.4 mm	Recess diameter
r _{1,2}	min. 0.6 mm	Chamfer dimension



Abutment dimensions

d _a	min. 32 mm	Diameter of shaft abutment
D _a	max. 40 mm	Diameter of housing abutment
r _a	max. 0.3 mm	Radius of shaft or housing fillet

Calculation data

Basic dynamic load rating	C	4.1 kN
Basic static load rating	C ₀	2.9 kN
Fatigue load limit	P _u	0.146 kN
Reference speed		32 000 r/min
Limiting speed		20 000 r/min
Minimum load factor	k _r	0.015
Calculation factor	f ₀	13.9

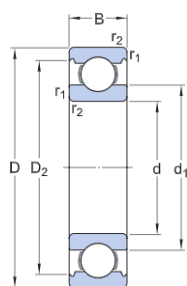


61805

- Popular item

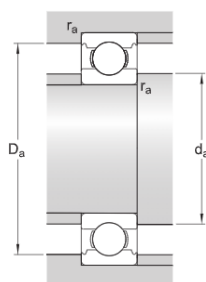
Deep groove ball bearings

Technical specification



DIMENSIONS

d	25 mm	Bore diameter
D	37 mm	Outside diameter
B	7 mm	Width
d ₁	≈ 28.5 mm	Shoulder diameter
D ₁	≈ 33.15 mm	Shoulder diameter
D ₂	≈ 34.2 mm	Recess diameter
r _{1,2}	min. 0.3 mm	Chamfer dimension



ABUTMENT DIMENSIONS

d _a	min. 27 mm	Diameter of shaft abutment
D _a	max. 35 mm	Diameter of housing abutment
r _a	max. 0.3 mm	Radius of shaft or housing fillet

CALCULATION DATA

Basic dynamic load rating	C	4.36 kN
Basic static load rating	C ₀	2.6 kN
Fatigue load limit	P _u	0.125 kN
Reference speed		38 000 r/min
Limiting speed		24 000 r/min
Minimum load factor	k _r	0.015
Calculation factor	f ₀	14.2

MASS

Mass bearing	0.021 kg
--------------	----------

TOLERANCE CLASS

Dimensional tolerances	Normal
Radial run-out	Normal

C.2 Planet bearings - SKF NJ 202 ECP and SKF k 15x19x10



NU 202 ECP

- Popular item

Cylindrical roller bearings, single row

Bearing data

Tolerances,

Normal (metric), P6, Normal (inch),

[Radial internal clearance,](#)

cylindrical bore, tapered bore,

[Axial internal clearance,](#)

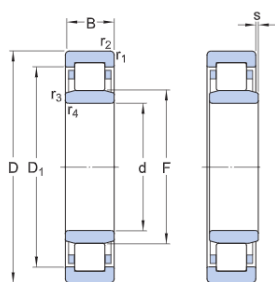
NUP, NJ + HJ

Bearing interfaces

[Seat tolerances for standard conditions,](#)

[Tolerances and resultant fit](#)

Technical specification

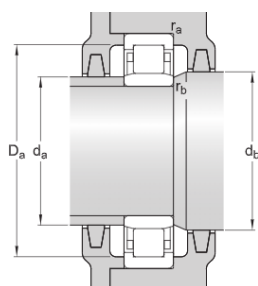


DIMENSIONS

d	15 mm	Bore diameter
D	35 mm	Outside diameter
B	11 mm	Width
D ₁	≈ 27.7 mm	Shoulder diameter of outer ring
F	19.3 mm	Chamfer dimension of loose flange ring
r _{1,2}	min. 0.6 mm	Chamfer dimension
r _{3,4}	min. 0.3 mm	Chamfer dimension
s	max. 1 mm	Permissible axial displacement

ABUTMENT DIMENSIONS

d _a	min. 17.4 mm	Diameter of spacer sleeve
d _a	max. 18.4 mm	Diameter of spacer sleeve
d _b	min. 21 mm	Diameter of shaft abutment
D _a	max. 31.3 mm	Diameter of housing abutment
r _a	max. 0.6 mm	Radius of fillet
r _b	max. 0.3 mm	Radius of fillet



CALCULATION DATA

Basic dynamic load rating	C	12.5 kN
Basic static load rating	C ₀	10.2 kN
Fatigue load limit	P _u	1.22 kN
Reference speed		22 000 r/min
Limiting speed		26 000 r/min
Minimum load factor	k _r	0.15
Limiting value	e	0.2
Axial load factor	Y	0.6

MASS

Mass	0.047 kg
------	----------

More information

Product details Designs and variants Bearing data Loads Temperature limits Permissible speed Design considerations Designation system	Engineering information Principles of rolling bearing selection General bearing knowledge Bearing selection process Bearing failure and how to prevent it	Tools SimPro Quick Bearing Select Engineering Calculator LubeSelect for SKF greases Heater selection tool Oil Injection Method Program Rolling bearings mounting and dismounting instructions
---	--	---



K 15X19X10

Single row needle roller and cage assembly

Needle roller and cage assemblies are self-contained, ready-to-mount bearings without inner and outer rings. In applications where the shaft and housing bore can serve as raceways, the assemblies can be used to create bearing arrangements that require minimal radial space. As the clearance of the bearing arrangement is only affected by the roller sorting and the finish of shaft and housing, very precise bearing arrangements can be achieved.

- High radial load carrying capacity
- High stiffness
- High precision
- Lowest cross-sectional height among rolling bearings
- Accommodate axial displacement in both directions

Overview

Dimensions

Bore diameter of needle roller complement	15 mm
Outside diameter	19 mm
Outside diameter of needle roller complement	19 mm
Width	10 mm

Performance

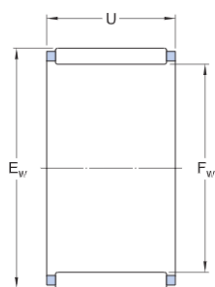
Basic dynamic load rating	7.21 kN
Basic static load rating	9.3 kN
Limiting speed	28000 r/min
Reference speed	24000 r/min

Properties

Aligning feature	Without
Bearing part	Needle roller and cage assembly
Cage	Sheet steel
Coating	Without
Lubricant	None
Material, bearing	Bearing steel
Number of rows	1
Outer ring type	Not applicable
Relubrication feature	Without
Sealing	Without
Sealing type	Not applicable
Tolerance class	Normal



Technical Specification



Dimensions

F_w	15 mm	Diameter under rollers
E_w	19 mm	Outside diameter roller complement
U	10 mm	Width needle roller and cage assembly

Calculation data

Basic dynamic load rating	C	7.21 kN
Basic static load rating	C_0	9.3 kN
Fatigue load limit	P_u	1.08 kN
Reference speed		24 000 r/min
Limiting speed		28 000 r/min

Mass

Mass needle roller and cage assembly	0.005 kg
--------------------------------------	----------

C.3 Lead screw shaft bearings - SKF 6000 and SKF 618/6



6000

- Item popular
- SKF Explorer

Rolamentos rígidos de esferas

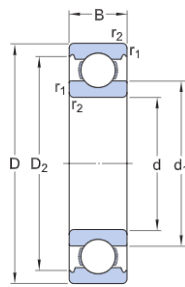
Dados do rolamento

Tolerâncias, Normal (métrico),
P6, P5, Normal (polegadas),
Folga interna radial, Classes C2
a C5

Interfaces do rolamento

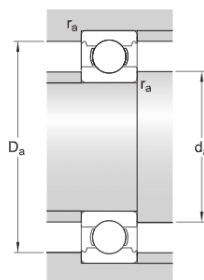
[Tolerâncias de assento para
condições padrão,](#)
[Tolerâncias e ajustes resultantes](#)

Especificações técnicas



DIMENSÕES

d	10 mm	Diâmetro do furo
D	26 mm	Diâmetro externo
B	8 mm	Largura
d ₁	≈ 14.8 mm	Diâmetro do ressalto
D ₂	≈ 22.6 mm	Diâmetro do rebaixo
r _{1,2}	min. 0.3 mm	Dimensão de chanfro



DIMENSÕES DO ENCOSTO

d _a min. 12 mm	Diâmetro do encosto do eixo
D _a max. 24 mm	Diâmetro do encosto do mancal
r _a max. 0.3 mm	Raio do eixo ou do filete do mancal

DADOS DE CÁLCULO

Classificação de carga dinâmica básica	C	4.75 kN
Classificação de carga estática básica	C ₀	1.96 kN
Limite de carga de fadiga	P _u	0.083 kN
Velocidade de referência		67 000 r/min
Velocidade-limite		40 000 r/min
Fator de carga mínima	k _r	0.025
Fator de cálculo	f ₀	12

MASSA

Massa do rolamento	0.019 kg
--------------------	----------

CLASSE DE TOLERÂNCIA

Tolerâncias dimensionais	P6
Desvio radial	P5



618/6

Deep groove ball bearings

Bearing data

Tolerances,

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

Radial internal clearance,

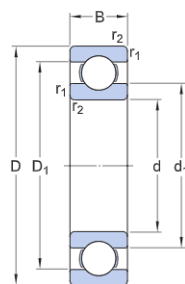
Classes C2 to C5

Bearing interfaces

Seat tolerances for standard conditions,

Tolerances and resultant fits

Technical specification

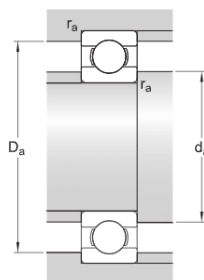


DIMENSIONS

d	6 mm	Bore diameter
D	13 mm	Outside diameter
B	3.5 mm	Width
d ₁	≈ 8 mm	Shoulder diameter
D ₁	≈ 11 mm	Shoulder diameter
r _{1,2}	min. 0.15 mm	Chamfer dimension

ABUTMENT DIMENSIONS

d _a	min. 6.8 mm	Diameter of shaft abutment
D _a	max. 12.2 mm	Diameter of housing abutment
r _a	max. 0.1 mm	Radius of shaft or housing fillet



CALCULATION DATA

Basic dynamic load rating	C	0.715 kN
Basic static load rating	C ₀	0.224 kN
Fatigue load limit	P _u	0.01 kN
Reference speed		110 000 r/min
Limiting speed		67 000 r/min
Minimum load factor	k _r	0.015
Calculation factor	f ₀	7

MASS

Mass bearing	0.004 kg
--------------	----------

TOLERANCE CLASS

Dimensional tolerances	Normal
Radial run-out	Normal

C.4 Thrust bearings - SKF AXK 4060, 90120 and 150190



AXK 4060

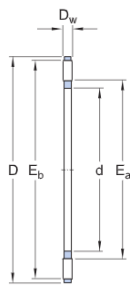
- Popular item

Needle roller thrust bearings

Bearing data, Design considerations
[Tolerances, table, values,](#)
[Design considerations, table](#)

Bearing interfaces
[Seat tolerances for standard conditions](#)

Technical specification



DIMENSIONS

d	40 mm	Bore diameter
D	60 mm	Outside diameter
D _w	3 mm	Diameter roller
E _a	min. 45 mm	Raceway diameter (min.) roller and cage thrust assembly
E _b	max. 58 mm	Raceway diameter (max.) roller and cage thrust assembly

CALCULATION DATA

Basic dynamic load rating	C	25 kN
Basic static load rating	C ₀	114 kN
Fatigue load limit	P _u	13.7 kN
Reference speed		2 800 r/min
Limiting speed		5 600 r/min

MASS

Mass roller and cage thrust assembly	0.016 kg
--------------------------------------	----------

ASSOCIATED PRODUCTS

Universal washer	LS 4060
Thin universal washer	AS 4060
Shaft washer	WS 81108
Housing washer	GS 81108

More information

Product details	Engineering information	Tools
Designs and variants	Principles of rolling bearing selection	SimPro Quick
Bearing data	General bearing knowledge	Bearing Select
Loads	Bearing selection process	Engineering Calculator
Temperature limits	Bearing failure and how to prevent it	LubeSelect for SKF greases
Permissible speed		Heater selection tool
Design considerations		
Designation system		



AXK 90120

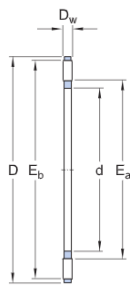
- Popular item

Needle roller thrust bearings

Bearing data, Design considerations
[Tolerances, table, values,](#)
[Design considerations, table](#)

Bearing interfaces
[Seat tolerances for standard conditions](#)

Technical specification



DIMENSIONS

d	90 mm	Bore diameter
D	120 mm	Outside diameter
D _w	4 mm	Diameter roller
E _a	min. 94 mm	Raceway diameter (min.) roller and cage thrust assembly
E _b	max. 118 mm	Raceway diameter (max.) roller and cage thrust assembly

CALCULATION DATA

Basic dynamic load rating	C	65.5 kN
Basic static load rating	C ₀	405 kN
Fatigue load limit	P _u	49 kN
Reference speed		1 500 r/min
Limiting speed		3 000 r/min

MASS

Mass roller and cage thrust assembly	0.086 kg
--------------------------------------	----------

ASSOCIATED PRODUCTS

Universal washer	LS 90120
Thin universal washer	AS 90120
Shaft washer	WS 81118
Housing washer	GS 81118

More information

Product details	Engineering information	Tools
Designs and variants	Principles of rolling bearing selection	SimPro Quick
Bearing data	General bearing knowledge	Bearing Select
Loads	Bearing selection process	Engineering Calculator
Temperature limits	Bearing failure and how to prevent it	LubeSelect for SKF greases
Permissible speed		Heater selection tool
Design considerations		
Designation system		



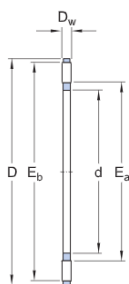
AXK 150190

Needle roller thrust bearings

Bearing data, Design considerations
[Tolerances, table, values,](#)
[Design considerations, table](#)

Bearing interfaces
[Seat tolerances for standard conditions](#)

Technical specification



DIMENSIONS

d	150 mm	Bore diameter
D	190 mm	Outside diameter
D _w	5 mm	Diameter roller
E _a	min. 156 mm	Raceway diameter (min.) roller and cage thrust assembly
E _b	max. 187 mm	Raceway diameter (max.) roller and cage thrust assembly

CALCULATION DATA

Basic dynamic load rating	C	120 kN
Basic static load rating	C ₀	950 kN
Fatigue load limit	P _u	102 kN
Reference speed		1 000 r/min
Limiting speed		2 000 r/min

MASS

Mass roller and cage thrust assembly	0.23 kg
--------------------------------------	---------

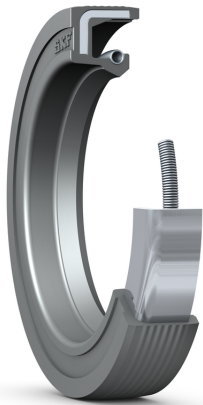
ASSOCIATED PRODUCTS

Universal washer	LS 150190
Thin universal washer	AS 150190
Shaft washer	WS 81130
Housing washer	GS 81130

More information

Product details Designs and variants Bearing data Loads Temperature limits Permissible speed Design considerations Designation system	Engineering information Principles of rolling bearing selection General bearing knowledge Bearing selection process Bearing failure and how to prevent it	Tools SimPro Quick Bearing Select Engineering Calculator LubeSelect for SKF greases Heater selection tool
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C.5 Seal - SKF 30x42x6 HMS5 RG



30X42X6 HMS5 RG

Radial shaft seal with rubber outside diameter and single sealing lip, for oil or grease, metric

Radial shaft seals are used between rotating and stationary machine components, or between components in relative motion. HMS5 seals are designed with a rubber outside diameter. They have a conventional sealing lip made of elastomer or thermoplastic with a garter spring to maintain the right radial load.

- High pumping rate
- With garter spring
- Accommodate misalignment

Overview

Dimensions

Housing bore diameter	42 mm
Nominal width	6 mm
Shaft diameter	30 mm

Properties

Auxiliary lip	No
Design	HMS5
Sealing lip material	Nitrile rubber (NBR)

Performance

Limiting speed	3660 r/min
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Appendix - Other Components

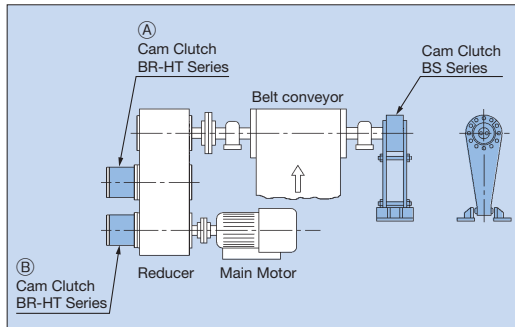
This chapter list the catalogues for all the remaining components that were used in the design of the transmission.

- One-way clutch - Tsubaki BR48HT-R76B;
- Brake bands' friction material;
- Electric motor;
- Brake bands' locking pins;
- Precision construction screws;
- Jaw coupling GN 2240-20-B6-8-AL-RS;
- Lock nuts SKF KMFE 4 and SKF KMFE 11;

D.1 One-way clutch - Tsubaki BR48HT-R76B

BR-HT SERIES

TYPICAL APPLICATION



Mounting position ①

Intermediate shaft of Reducer

Use: Backstop for medium speed overrunning

Mounting position ②

Input shaft of Reducer

Use: Backstop for high-speed overrunning

Other application

- Pump Backstopping
- Fan Backstopping
- Backstopping function in high speed inner race overrunning

SELECTION PROCEDURES

For Back Stop Application

Calculate the torque on the Cam Clutch according to the following formula:

$$T = \frac{9550 \times P}{N} \times SF$$

T: Loaded torque (N·m)

P: Nominal torque of Motor (kW)

N: Shaft speed (r/min)

SF: Service factor

- Notes:**
1. SF 1.6 (Service condition: For belt conveyor or bucket elevator)
 2. Select the clutch whose T.C. satisfies the calculated backstop torque (T).
 3. If there is a chance of motor stall, include the stall torque ratio in your calculations.

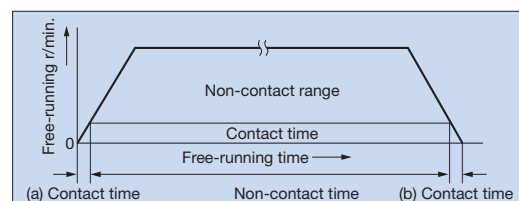
Select Clutch by:

- a) Design torque requirement and service factor
- b) The actual overrunning speed should be between MIN r/min and MAX r/min
- c) Bore

In case the overrunning speed is less than MIN r/min or higher than MAX r/min, please contact TSUBAKI.

SERVICE LIFE OF BR-HT SERIES

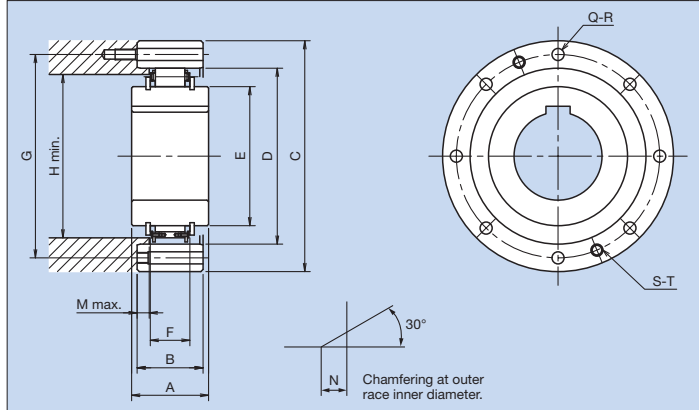
The service life of previous TSUBAKI Cam Clutch was determined as the frictional service life during free-running (clutch disengaged) and the fatigue service life of the engaged clutch. However, with BR-HT Series, free-running frictional service life is not a factor because there is no mechanical contact when the clutch is disengaged. As a result, service life is determined solely by the fatigue life of the engaged clutch.



Friction in the clutch mechanism only occurs during a very short period of time denoted by "a" and "b". "a" is the time during which the cam is engaged until the acceleration of inner race causes it to disengage. "b" is the time during which the cam engages when the inner race decelerates.

BR-HT SERIES CAM CLUTCH

MODELS BR15HT TO BR130HT



Format

BR40HT - R66B - 35

Bore size

Size & Length of cam
A, B, C, D type
WA/WB/WC/WD type
(W=Double cam cage)

Model name
HT=High Torque

Dimensions and Capacities

Dimensions in mm

Model	Bore Size (H7)	T.C. (N·m)	Inner Race Overrunning Speed (r/min)		Max. Engagement (r/min)	A	B	C	D (H7)	E	Mounting Holes		Removal Holes	F	Min. Mass (kg)	Max. Mass (kg)	H min.	M max.	Chamfer N
			Min.	Max.							PCD G	Q'ty-Size Q-R	Q'ty-Size S-T						
BR15HT-R31A	*20	105	880	3,600	550	24	25	85	55	30	70	6-M6	2-M6	17.0	—	0.8	45	3	1
BR18HT-R38A	*25	155	850	3,600	500	24	25	90	62	37	75	6-M6	2-M6	17.0	—	0.9	50	3	1
BR20HT-S20B	20	225	850	3,600	400	35	35	90	66	41	78	6-M6	2-M6	25.0	—	1.3	53	4	1
BR25HT-B46B	25 30	400	800	3,600	380	35	35	95	70	45	82	6-M6	2-M6	25.0	1.4	1.4	58	4	1
BR30HT-S30B	30	500	740	3,600	360	35	35	100	75	50	87	6-M6	2-M6	25.0	—	1.5	64	4	1
BR30HT-R51B	25 30 35 36	500	740	3,600	360	35	35	105	75	50	90	6-M6	2-M6	25.0	1.7	1.8	64	4	1
BR35HT-B56B	35 40	600	710	3,600	340	35	35	110	80	55	96	8-M6	2-M6	25.0	1.8	1.9	70	4	1
BR38HT-R61A	30 35 40 *45	425	740	3,600	400	25	25	120	85	60	105	6-M8	2-M8	19.0	1.6	1.8	74	3	1
BR40HT-S40B	40	850	670	3,600	320	35	35	125	90	65	108	8-M8	2-M8	25.0	—	2.4	82	4	1
BR40HT-R66B	35 40 45 *48	850	670	3,600	320	35	35	132	90	65	115	8-M8	2-M8	25.0	2.7	2.9	82	4	1
BR45HT-S45B	45	950	640	3,600	310	35	35	130	95	70	112	8-M8	2-M8	25.0	—	2.6	86	4	1
BR48HT-R76B	45 55 *60	1,100	620	3,600	300	35	35	140	100	75	125	8-M8	2-M8	25.0	2.9	3.3	92	4	1
BR50HT-B86B	40 45 50 60 65 *70	1,450	590	3,600	280	40	40	150	110	85	132	8-M8	2-M8	25.0	3.5	4.3	103	6.5	1
BR58HT-R101B	55 70 *80	1,800	550	3,600	260	50	50	175	125	100	155	8-M10	2-M10	25.0	5.8	6.7	117	11.5	1
BR60HT-B85A	45 50 60 65	2,400	420	3,600	230	60	50	175	125	92	155	8-M10	2-M10	36.0	6.8	7.6	110	6	1
BR70HT-B100A	45 50 55 60 70 75 *80	3,150	390	3,600	220	60	50	190	140	107	165	12-M10	2-M10	36.0	7.6	9.2	125	6	1.5
BR80HT-S80A	80	5,000	440	3,600	200	70	60	210	160	127	185	12-M10	2-M10	36.0	—	12	148	11	1.5
BR80HT-B120B	60 65 70 75 80 95	7,000	310	3,600	160	70	60	210	160	127	185	12-M10	2-M10	50.0	11	13	148	4	1.5
BR90HT-S90A	90	6,000	410	3,000	190	80	70	230	180	147	206	12-M12	2-M12	36.0	—	16	170	16	2
BR90HT-B140B	65 90 100 110	9,000	300	3,000	150	70	70	245	180	147	218	12-M12	2-M12	50.0	17	20	170	9	2
BR95HT-S100C	100	20,500	240	2,700	130	90	80	290	210	177	258	12-M16	2-M16	63.0	—	33	200	7.5	2
BR95HT-R170C	70 85 90 100 120 130	20,500	240	2,700	130	80	80	290	210	177	258	12-M16	2-M16	63.0	29	35	200	7.5	2
BR98HT-R200C	130 155	27,000	230	2,100	110	80	80	310	240	207	278	12-M16	2-M16	63.0	30	33	230	7.5	2
BR100HT-S100A	100	11,000	440	2,700	210	90	80	290	210	143	258	12-M16	2-M16	52.6	—	28	200	11.5	2
BR130HT-S130A	130	16,000	400	2,400	190	90	80	322	240	173	278	12-M16	2-M16	52.6	—	33	210	11.5	2

Notes: 1. T.C.---Torque Capacity

The maximum transmissible torque is twice the T.C.

2. Keyway No Mark on Bore Size...ISOR773/DIN6885.1

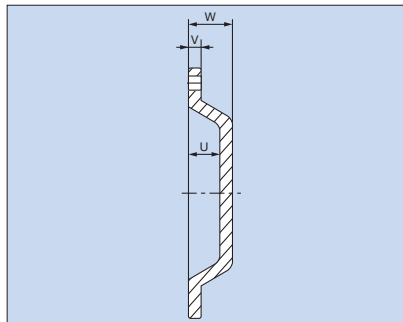
*Mark on Bore Size...DIN6885.3

3. Min. overrunning speed of inner race should not be below under continuous operation.

4. Max. engagement speed must not be exceeded when transmitting torque.

Recommended Cover Dimension Table

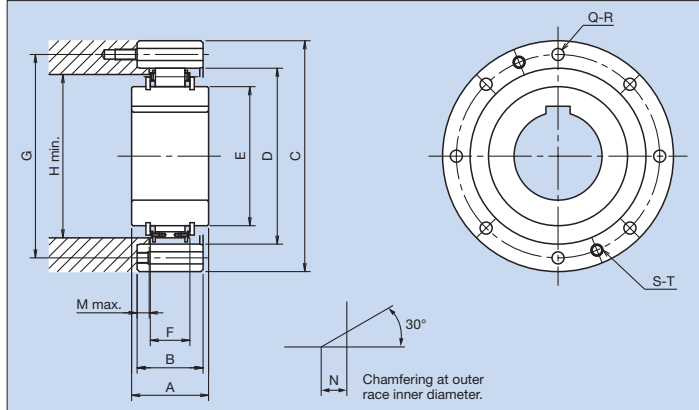
Dimensions in mm



BR-HT Size	Width		Depth
	W	V	
BR15HT to BR38HT	21	8	15
BR40HT to BR50HT	23	11	17
BR58HT to BR60HT	28	11	22
BR70HT to BR95HT-S100C	35	11	29
BR95HT-R170C to BR130HT	42	15	33
BR180HT	60	15	48
BR190HT to BR230HT	60	18	48
BR240HT to BR300HT	68	26	56

BR-HT SERIES CAM CLUTCH

MODELS BR180HT TO BR300HT



Format

BR40HT - R66B - 35

Bore size

Size & Length of cam
A, B, C, D type
WA/WB/WC/WD type
(W=Double cam cage)

Model name
HT=High Torque

Dimensions and Capacities

Dimensions in mm

Model	Bore Size (H7)	T.C. (N·m)	Inner Race Overrunning Speed (r/min)		Max. Engagement (r/min)	A	B	C	D (H7)	E	Mounting Holes		Removal Holes		F	Mass (kg)	H min.	M max.	Chamfer N
			Min.	Max.							PCD G	Q'ty-Size Q-R	Q'ty-Size S-T						
BR180HT-S180A	180	32,000	300	1,300	160	90	80	412	310	243	360	12-M20	2-M20	53	56	280	11.5	2	
BR180HT-S180C	180	53,000	250	1,300	120	120	120	422	310	243	370	18-M20	2-M20	83	85	280	16.5	2	
BR180HT-S180WA	180	64,000	300	1,300	160	160	160	412	310	243	360	24-M20	2-M20	106	107	280	30	2	
BR180HT-S180WC	180	106,000	250	1,300	120	240	240	425	310	243	370	24-M24	2-M20	166	174	280	35	2	
BR180HT-R240A	185	32,000	220	1,300	110	90	80	400	310	243	360	12-M20	2-M20	53	50	280	11.5	2	
BR180HT-R240D	185	64,000	210	1,300	100	120	125	420	310	243	370	16-M24	2-M24	96	84	280	12.5	2	
BR180HT-R240WB	185	70,000	220	1,300	110	160	160	412	310	243	360	24-M20	2-M20	140	100	280	8	2	
BR180HT-R240WD	185	128,000	210	1,300	100	240	240	425	310	243	370	24-M24	2-M24	192	163	280	22	2	
BR190HT-R260A	205	39,000	200	1,300	95	105	80	430	330	263	380	16-M20	2-M20	53	60	300	11.5	2	
BR220HT-S220A	220	45,000	280	1,100	140	105	80	470	360	293	410	16-M20	2-M20	53	74	330	11.5	2	
BR220HT-S220C	220	70,000	230	1,100	110	120	120	470	360	293	410	16-M24	2-M20	83	100	330	16.5	2	
BR220HT-S220WA	220	90,000	280	1,100	140	160	160	480	360	293	430	18-M24	2-M24	106	141	330	25	2	
BR220HT-S220WC	220	140,000	230	1,100	110	240	240	490	360	293	425	20-M30	2-M30	166	215	330	35	2	
BR220HT-R290B	230	60,000	195	1,100	115	105	80	460	360	293	410	16-M20	2-M20	70	87	330	3	2	
BR220HT-R290D	230	92,000	190	1,100	95	120	110	460	360	293	410	16-M20	2-M20	96	146	330	5	2	
BR220HT-R290WB	230	120,000	195	1,100	115	160	160	480	360	293	410	18-M24	2-M24	140	120	330	8	2	
BR220HT-R290WD	230	184,000	190	1,100	95	240	240	490	360	293	425	20-M30	2-M30	192	206	330	22	2	
BR230HT-R310B	240	70,000	190	1,100	90	110	125	497	380	313	450	24-M20	2-M20	70	110	350	25.5	3	
BR230HT-R310D	240	110,000	185	1,100	80	120	125	497	380	313	450	24-M20	2-M20	96	116	350	12.5	3	
BR240HT-S240A	240	54,000	220	1,100	120	105	90	500	390	323	440	16-M20	2-M20	53	91	360	16.5	3	
BR240HT-S240C	240	88,000	185	1,100	110	120	120	520	390	323	440	16-M24	2-M24	83	129	360	16.5	3	
BR240HT-S240WA	240	108,000	220	1,100	120	180	180	520	390	323	440	24-M24	2-M24	106	161	360	35	3	
BR240HT-S240WC	240	176,000	185	1,100	110	240	240	530	390	323	455	24-M30	2-M30	166	249	360	35	3	
BR240HT-R320B	250	77,000	190	1,100	115	105	80	490	390	323	440	16-M24	2-M24	70	78	360	3	3	
BR240HT-R320D	250	113,000	180	1,100	105	120	120	520	390	323	440	16-M24	2-M24	96	128	360	10	3	
BR240HT-R320WB	250	154,000	190	1,100	115	180	180	505	390	323	440	24-M24	2-M24	140	173	360	18	3	
BR240HT-R320WD	250	226,000	180	1,100	105	240	240	530	390	323	460	24-M30	2-M30	192	259	360	22	3	
BR260HT-S260A	260	66,000	250	1,000	130	105	105	550	430	363	500	16-M24	2-M24	57	122	400	22	3	
BR260HT-S260C	260	110,000	190	1,000	100	125	125	580	430	363	500	24-M24	2-M24	87	170	400	17	3	
BR260HT-S260WA	260	132,000	250	1,000	130	210	210	580	430	363	500	24-M24	2-M24	114	235	400	46	3	
BR260HT-S260WC	260	220,000	190	1,000	100	250	250	580	430	363	500	24-M30	2-M30	174	323	400	36	3	
BR260HT-R360D	280	150,000	170	1,000	90	125	120	540	430	363	500	24-M24	2-M24	100	127	400	8	3	
BR260HT-R360WB	280	196,000	175	1,000	95	210	210	550	430	363	500	24-M24	2-M24	148	227	400	29	3	
BR260HT-R360WD	280	300,000	170	1,000	90	250	250	580	430	363	500	24-M30	2-M30	200	311	400	23	3	
BR300HT-S300A	300	82,000	230	1,000	120	105	105	630	480	413	560	24-M24	2-M24	53	163	460	22	3	
BR300HT-S300C	300	140,000	200	1,000	95	125	125	630	480	413	560	24-M24	2-M24	83	198	460	17	3	
BR300HT-S300WA	300	164,000	230	1,000	120	210	210	630	480	413	560	24-M24	2-M24	106	324	460	46	3	
BR300HT-R410D	320	195,000	165	1,000	85	125	120	630	480	413	560	24-M24	2-M24	100	186	460	8	3	
BR300HT-R410WB	320	250,000	165	1,000	85	210	210	630	480	413	560	24-M24	2-M24	148	314	460	29	3	
BR300HT-R410WD	320	366,000	165	1,000	85	220	220	630	480	413	560	24-M30	2-M30	200	324	460	8	3	

Notes: 1. T.C.---Torque Capacity

The maximum transmissible torque is twice the T.C.

2. Keyway No Mark on Bore Size...ISOR773 DIN6885.1

3. Min. overrunning speed of inner race should not be below under continuous operation.

4. Max. engagement speed must not be exceeded when transmitting torque.

D.2 Brake bands' friction material



ID Material: 16
R. Antich
Revision: 1
Date: 8/25/20

45 Kensico Drive
Mt. Kisco, NY 10549
T. 914-244-3600
E. answers@protecfriiction.com
www.protecfriiction.com

TF1700

TF1700 is a high-content Kevlar material designed for high performance motorsports clutch, severe service industrial brake and clutch applications. TF1700 virtually eliminates clutch slippage with a high torque capacity that avoids uncontrollable glazing

Additionally, TF1700 exhibits extraordinary wear life properties. When given a mating surface, clutch flywheel, brake rotor, or brake drum, it will dramatically outlast organic friction wear performance .

TF1700 is completely non-abrasive against mating surfaces, promotes smooth engagement, resists very high surface pressures at the brake interface, and is an effective alternative for sintered materials since it handles high energy inputs in both dry and oil-immersed applications.

TF1700 is available in flat sheets and in finished custom shapes and sizes, with thicknesses as low as 0.040"

Material Data

Friction Properties (according to graphics)

Dynamic Friction Coefficient:	0.65±0.05	μ
Wear Rate:	60 (at 150 °C)	
T° Fading:	>500	°C

Physical Properties

Hardness (DIN53505):	85±5	Shore-D
Specific Gravity (ASTM D792):	1.27±0.1	gr/cm3

Mechanical Properties

Tensile Strength (ASTM D638):	70±5 N/mm ²
Compressive Strength (ISO 844:2014):	83±5 N/mm ²
Burst Resistant (200 x 137 x 3,5) 200°C:	18200±200 RPM
Poisson Coefficient (ASTM D638):	0.27±0.03
Young Modulus (ASTM D638):	7200±100 N/mm ²

Recommended Working Values

T° Max. Continuous Operation:	300 °C
T° Max. Intermittent Operation:	450 °C

Material Type : Paper Friction

Appearance / Formats



Applications

Agricultural and building machinery - Callipers for industrial applications - Car / motorcycle competition clutches - Clutch buttons - Friction Gasket - Heavy-duty industrial machinery - Micellaneous industrial brakes / clutches - Wet Friction -

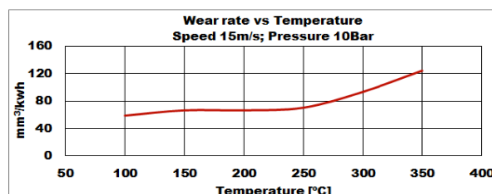
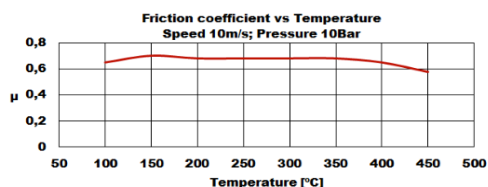
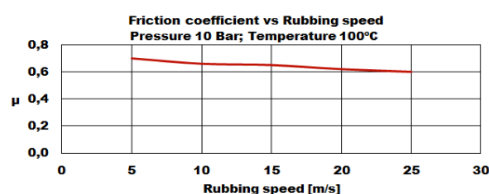
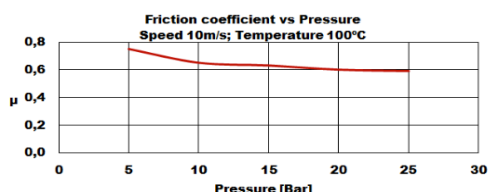
Price Level : \$ \$ \$ \$

Reach (EC) 1907/2006 - RoHS 2011/65/EU : Compliance

Others

Recommended Mating Surface:	Perlitic cast iron, hardness HB150-200
Recommended Adhesives:	Thermosetting adhesive
Oil Resistant:	Yes

The above data is taken from specific test parameters therefore results can vary in different application conditions



Rubbing speed, temperature and pressure are related. Changing any values will change others. The values shown represent typical conditions, but are not ultimate limits of the material

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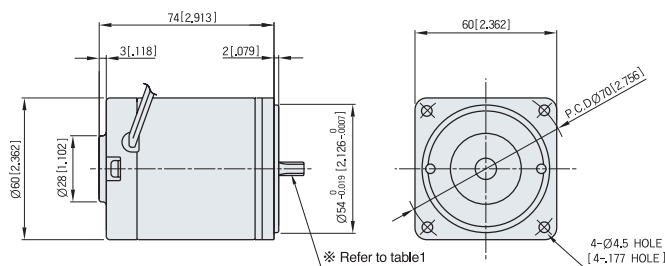
answers@protecfriiction.com

D.3 Electric motor



MOTOR DIMENSION

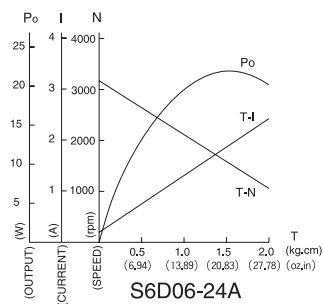
UNIT : mm[inch]



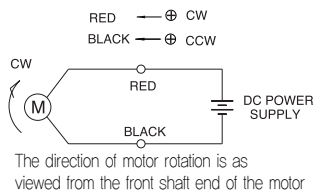
▼ SPEC for output shaft of motor-(Table1)

MODEL	GEARED TYPE S6D06-□A S6D10-□A	STRAIGHT TYPE S6D06-□S S6D10-□S	D-CUT TYPE S6D06-□D S6D10-□D
TYPE OF OUTPUT SHAFT			

CHARACTERISTIC CURVE



CIRCUIT DIAGRAM



MOTOR SPECIFICATION

(Voltage : DC 24V)

Type	Voltage	Output	No Load		At Rated Lod				Run Duty	Weight
	V		Speed	Current(MAX.)	Speed	Torque		Current		
		W	RPM	A	RPM			A	Hr	Kg
S6D06-12□	12	6	3100	0.70	2800	0.021	N-m	0.80	2000	0.6
						2.9	oz-in			
						0.21	kgf · cm			
S6D06-24□	24	6	3200	0.40	2950	0.020	N-m	0.50	2000	0.6
						2.8	oz-in			
						0.20	kgf · cm			
S6D06-90□	90	6	3000	0.10	2800	0.021	N-m	0.15	2000	0.6
						2.9	oz-in			
						0.21	kgf · cm			
S6D10-12□	12	10	3300	0.90	3000	0.032	N-m	1.60	2000	0.6
						4.6	oz-in			
						0.33	kgf · cm			
S6D10-24□	24	10	3300	0.50	3000	0.032	N-m	0.80	2000	0.6
						4.6	oz-in			
						0.33	kgf · cm			
S6D10-90□	90	10	3300	0.15	3100	0.030	N-m	0.20	2000	0.6
						4.3	oz-in			
						0.31	kgf · cm			

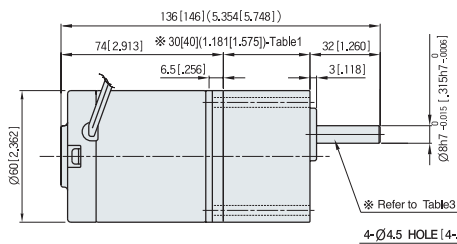
※ Run duty is valued at rated Voltage and No load

21C, for World geared motor!

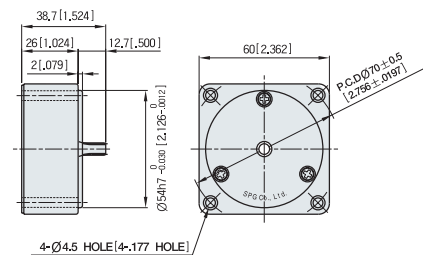


GEARED MOTOR SPECIFICATION

▼ GEARED MOTOR ※ MOTOR MODEL : S6D06-□A, S6D10-□A
※ HEAD MODEL : S6□A3□~S6□A250□



▼ INTER-DECIMAL GEAR HEAD
※ MODEL : S6GX10B



▼ ※30[40] – (Table1)

GEAR RATIO	SIZE:mm[inch]
S6 □ A3 □ ~ S6 □ A18 □	30[1.181]
S6 □ A20 □ ~ S6 □ A250 □	40[1.575]

▼ WEIGHT – (Table2)

PART		WEIGHT:kg(oz)
MOTOR		0.6/0.7(21.2/24.7)
DECIMAL GEAR HEAD		0.18(6.3)
GEAR HEAD	S6 □ A3 □ ~ S6 □ A18 □	0.24(8.5)
	S6 □ A20 □ ~ S6 □ A40 □	0.30(10.6)
	S6 □ A50 □ ~ S6 □ A250 □	0.33(11.6)

▼ SPEC for output shaft of gearhead-(Table3)

MODEL	TYPES OF OUTPUT SHAFT
STRAIGHT TYPE	32 (+0.256, -0) 28 (+0.315, -0) 121 (+0.2, -0)
D-CUT TYPE	32 (+0.256, -0) 28 (+0.315, -0) 121 (+0.2, -0) 7 (-0.5, -0.2) 1.275 (-0.001, -0.003)
KEY TYPE	32 (+0.256, -0) 28 (+0.315, -0) 121 (+0.2, -0) 10 (+0.005, -0.010) 1.8 (+0.1, -0) 1.071 (+0.0039, -0)

▼ KEY SPEC

GEAR HEAD

Technical drawing of a gear head showing two views: a front view and a side view.

Front View (Left):

- Shaft diameter: 12 ± 0.2 [12.0079]
- Gear thickness: 3 ± 0.05 [3.0010]

Side View (Right):

- Gear pitch diameter: 12 ± 0.2 [12.0079]
- Gear thickness: 3 ± 0.05 [3.0010]

GEAR HEAD RATED LOAD

1. S6D06-24A

MODEL	GEAR RATIO	3	3.6	5	6	7.5	9	10	12.5	15	18	20	25	30	36	40	50	60	75	90	100	120	150	180	200	250
	No Load rpm	1067	889	640	533	427	356	320	256	213	178	160	128	107	89	80	64	53	43	36	32	27	21	18	16	13
S&DA	N-m	0.05	0.06	0.08	0.10	0.12	0.14	0.16	0.20	0.24	0.29	0.29	0.36	0.43	0.51	0.57	0.64	0.77	0.96	1.16	1.29	1.54	1.93	2.31	2.57	2.94
	oz-in	6.7	8.1	11.2	13.5	16.9	20.2	22.5	28.1	33.7	40.5	40.5	50.6	60.7	72.8	80.9	91.0	109.3	136.6	163.9	182.1	218.5	273.1	327.8	364.2	416.3
	kgf·cm	0.5	0.6	0.8	1.0	1.2	1.5	1.6	2.0	2.4	2.9	2.9	3.6	4.4	5.2	5.8	6.6	7.9	9.8	11.8	13.1	15.7	19.7	23.6	26.2	30.6

2. S6D10-24A

MODEL	GEAR RATIO	3	3.6	5	6	7.5	9	10	12.5	15	18	20	25	30	36	40	50	60	75	90	100	120	150	180	200	250
	No Load rpm	1100	917	660	550	440	367	330	264	220	183	165	132	110	92	83	66	55	44	37	33	28	22	18	17	13
S6DA3B	N-m	0.08	0.09	0.13	0.16	0.20	0.24	0.26	0.33	0.39	0.47	0.47	0.59	0.71	0.85	0.94	1.06	1.27	1.59	1.91	2.12	2.55	2.94	2.94	2.94	2.94
	kgf·m	1.1	1.34	1.85	2.23	2.78	3.34	3.71	46.4	55.6	66.8	66.8	83.5	100.2	120.2	133.5	150.2	180.3	225.3	270.4	300.5	360.5	416.3	416.3	416.3	416.3
	ozf·cm	0.8	1.0	1.3	1.6	2.0	2.4	2.7	3.3	4.0	4.8	4.8	6.0	7.2	8.7	9.6	10.8	13.0	16.2	19.5	21.7	26.0	30.0	30.0	30.0	30.0

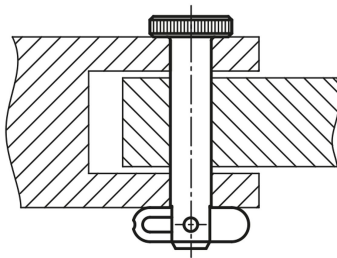
- The code in ■ of gearhead model is for gear ratio
- It is the permissible torque of the assembled motor and gearhead
- The permissible torque of the combination of the regular gearhead and the inter-decimal gearhead is $30\text{kgf} \cdot \text{cm}$
- ■ color indicates that the output shaft of the geared motor rotates in the same direction as the output shaft of the motor, Others indicate rotation in the opposite direction,
- No Load rpm equals Motor no load rpm divided by gear ratio. The actual rotation speed is less 2–20% than the displayed value according to the load
- There is no marked the 'L' code for it is an exclusive use,

D.4 Brake bands' locking pins

03400 Locking pins with folding latch



Item description/product images



Description

Material:

Steel electro zinc-plated.

Note:

Locking pins with folding latch are not only used for fast and easy fixing but also for permanent joining of movable parts and workpieces.

The wide folding latch lets you secure the parts to be joined over a large cross section. It can also work under pressure in the axial direction.

Indexing slots in the folding latch allow definite "Closed" and "Open" positions.

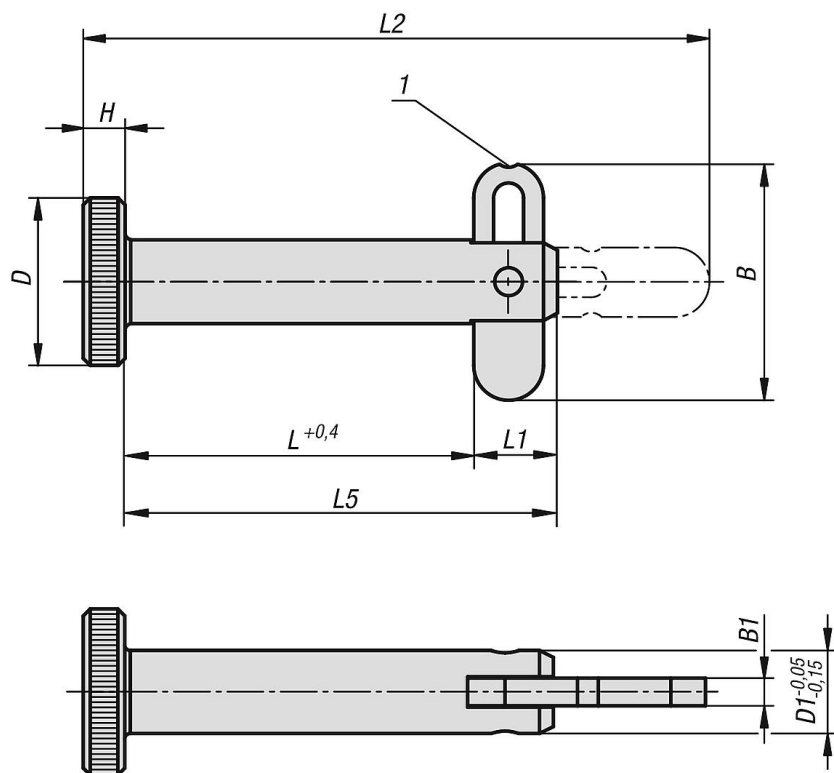
Drawing reference:

1) locking slot

03400 Locking pins with folding latch

norelem

Drawings



Overview of items

Order No.	B	B1	D	D1	H	L	L1	L2	L5	Receiving hole H11	Shearing force double shear max.kN	Extraction force F N
03400-06025	16,9	2	12	6	3	25	6	45	31	6	12	190
03400-06040	16,9	2	12	6	3	40	6	60	46	6	12	190
03400-06050	16,9	2	12	6	3	50	6	70	56	6	12	190
03400-08025	16,9	2	16	8	4	25	6	46	31	8	21	270
03400-08040	16,9	2	16	8	4	40	6	61	46	8	21	270
03400-08050	16,9	2	16	8	4	50	6	71	56	8	21	270

D.5 Precision construction screws

07534 Shoulder screws similar to DIN ISO 7379



Item description/product images



Description

Material:

Steel or stainless steel A2.

Version:

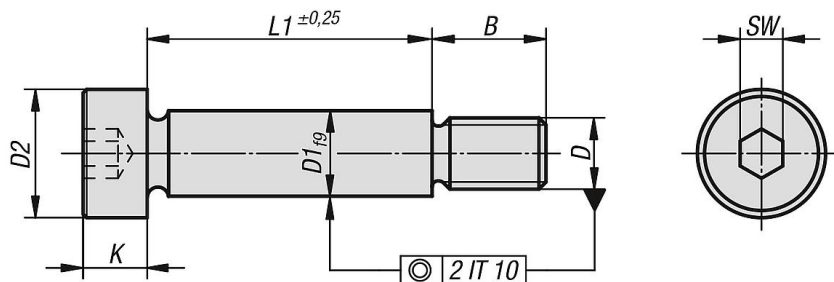
Grade 12.9. Shaft OD ground and bright.

Bright stainless steel or tempered steel.

Note:

Hexagon socket head shoulder screws are precision construction elements for many applications. As they can simplify complicated constructions, they are frequently chosen as the most cost-effective solution. Shoulder screws provide the decisive rationalising effect required today.

Drawings



Overview of items

Order No.	Main material	D1	D	D2	B	K	SW	L1
07534-04X06	steel	4	M3	7	7	3	2	6
07534-04X08	steel	4	M3	7	7	3	2	8
07534-04X10	steel	4	M3	7	7	3	2	10
07534-04X12	steel	4	M3	7	7	3	2	12
07534-04X16	steel	4	M3	7	7	3	2	16
07534-04X20	steel	4	M3	7	7	3	2	20
07534-05X08	steel	5	M4	9	8	4	2,5	8
07534-05X10	steel	5	M4	9	8	4	2,5	10
07534-05X16	steel	5	M4	9	8	4	2,5	16
07534-05X20	steel	5	M4	9	8	4	2,5	20
07534-05X30	steel	5	M4	9	8	4	2,5	30
07534-05X40	steel	5	M4	9	8	4	2,5	40
07534-06X16	steel	6	M5	10	9,5	4,5	3	16
07534-06X20	steel	6	M5	10	9,5	4,5	3	20
07534-06X25	steel	6	M5	10	9,5	4,5	3	25
07534-06X30	steel	6	M5	10	9,5	4,5	3	30
07534-06X40	steel	6	M5	10	9,5	4,5	3	40
07534-06X50	steel	6	M5	10	9,5	4,5	3	50
07534-06X60	steel	6	M5	10	9,5	4,5	3	60
07534-08X16	steel	8	M6	13	11	5,5	4	16
07534-08X20	steel	8	M6	13	11	5,5	4	20
07534-08X25	steel	8	M6	13	11	5,5	4	25
07534-08X30	steel	8	M6	13	11	5,5	4	30
07534-08X40	steel	8	M6	13	11	5,5	4	40

07534 Shoulder screws similar to DIN ISO 7379



Overview of items

Order No.	Main material	D1	D	D2	B	K	SW	L1
07534-08X50	steel	8	M6	13	11	5,5	4	50
07534-08X60	steel	8	M6	13	11	5,5	4	60
07534-10X16	steel	10	M8	16	13	7	5	16
07534-10X20	steel	10	M8	16	13	7	5	20
07534-10X25	steel	10	M8	16	13	7	5	25
07534-10X30	steel	10	M8	16	13	7	5	30
07534-10X40	steel	10	M8	16	13	7	5	40
07534-10X50	steel	10	M8	16	13	7	5	50
07534-10X60	steel	10	M8	16	13	7	5	60
07534-10X70	steel	10	M8	16	13	7	5	70
07534-10X80	steel	10	M8	16	13	7	5	80
07534-12X16	steel	12	M10	18	16	9	6	16
07534-12X20	steel	12	M10	18	16	9	6	20
07534-12X25	steel	12	M10	18	16	9	6	25
07534-12X30	steel	12	M10	18	16	9	6	30
07534-12X40	steel	12	M10	18	16	9	6	40
07534-12X50	steel	12	M10	18	16	9	6	50
07534-12X60	steel	12	M10	18	16	9	6	60
07534-12X70	steel	12	M10	18	16	9	6	70
07534-12X80	steel	12	M10	18	16	9	6	80
07534-12X90	steel	12	M10	18	16	9	6	90
07534-12X100	steel	12	M10	18	16	9	6	100
07534-16X30	steel	16	M12	24	18	11	8	30
07534-16X40	steel	16	M12	24	18	11	8	40
07534-16X50	steel	16	M12	24	18	11	8	50
07534-16X60	steel	16	M12	24	18	11	8	60
07534-16X70	steel	16	M12	24	18	11	8	70
07534-16X80	steel	16	M12	24	18	11	8	80
07534-16X90	steel	16	M12	24	18	11	8	90
07534-16X100	steel	16	M12	24	18	11	8	100
07534-16X120	steel	16	M12	24	18	11	8	120
07534-20X30	steel	20	M16	30	22	14	10	30
07534-20X40	steel	20	M16	30	22	14	10	40
07534-20X50	steel	20	M16	30	22	14	10	50
07534-20X60	steel	20	M16	30	22	14	10	60
07534-20X70	steel	20	M16	30	22	14	10	70
07534-20X80	steel	20	M16	30	22	14	10	80
07534-20X90	steel	20	M16	30	22	14	10	90
07534-20X100	steel	20	M16	30	22	14	10	100
07534-20X120	steel	20	M16	30	22	14	10	120
07534-104X06	stainless steel	4	M3	7	7	3	2	6
07534-104X08	stainless steel	4	M3	7	7	3	2	8
07534-104X10	stainless steel	4	M3	7	7	3	2	10
07534-104X16	stainless steel	4	M3	7	7	3	2	16
07534-104X20	stainless steel	4	M3	7	7	3	2	20
07534-105X08	stainless steel	5	M4	9	8	4	2,5	8
07534-105X10	stainless steel	5	M4	9	8	4	2,5	10
07534-105X16	stainless steel	5	M4	9	8	4	2,5	16
07534-105X20	stainless steel	5	M4	9	8	4	2,5	20
07534-105X30	stainless steel	5	M4	9	8	4	2,5	30
07534-105X40	stainless steel	5	M4	9	8	4	2,5	40
07534-106X16	stainless steel	6	M5	10	9,5	4,5	3	16
07534-106X20	stainless steel	6	M5	10	9,5	4,5	3	20
07534-106X25	stainless steel	6	M5	10	9,5	4,5	3	25
07534-106X30	stainless steel	6	M5	10	9,5	4,5	3	30
07534-106X40	stainless steel	6	M5	10	9,5	4,5	3	40
07534-106X50	stainless steel	6	M5	10	9,5	4,5	3	50
07534-106X60	stainless steel	6	M5	10	9,5	4,5	3	60
07534-108X16	stainless steel	8	M6	13	11	5,5	4	16
07534-108X20	stainless steel	8	M6	13	11	5,5	4	20
07534-108X25	stainless steel	8	M6	13	11	5,5	4	25
07534-108X30	stainless steel	8	M6	13	11	5,5	4	30
07534-108X40	stainless steel	8	M6	13	11	5,5	4	40
07534-108X50	stainless steel	8	M6	13	11	5,5	4	50
07534-108X60	stainless steel	8	M6	13	11	5,5	4	60

07534 Shoulder screws similar to DIN ISO 7379



Overview of items

Order No.	Main material	D1	D	D2	B	K	SW	L1
07534-110X16	stainless steel	10	M8	16	13	7	5	16
07534-110X20	stainless steel	10	M8	16	13	7	5	20
07534-110X25	stainless steel	10	M8	16	13	7	5	25
07534-110X30	stainless steel	10	M8	16	13	7	5	30
07534-110X40	stainless steel	10	M8	16	13	7	5	40
07534-110X50	stainless steel	10	M8	16	13	7	5	50
07534-110X60	stainless steel	10	M8	16	13	7	5	60
07534-110X70	stainless steel	10	M8	16	13	7	5	70
07534-110X80	stainless steel	10	M8	16	13	7	5	80
07534-112X16	stainless steel	12	M10	18	16	9	6	16
07534-112X20	stainless steel	12	M10	18	16	9	6	20
07534-112X25	stainless steel	12	M10	18	16	9	6	25
07534-112X30	stainless steel	12	M10	18	16	9	6	30
07534-112X40	stainless steel	12	M10	18	16	9	6	40
07534-112X50	stainless steel	12	M10	18	16	9	6	50
07534-112X60	stainless steel	12	M10	18	16	9	6	60
07534-112X70	stainless steel	12	M10	18	16	9	6	70
07534-112X80	stainless steel	12	M10	18	16	9	6	80
07534-112X90	stainless steel	12	M10	18	16	9	6	90
07534-112X100	stainless steel	12	M10	18	16	9	6	100
07534-116X30	stainless steel	16	M12	24	18	11	8	30
07534-116X40	stainless steel	16	M12	24	18	11	8	40
07534-116X50	stainless steel	16	M12	24	18	11	8	50
07534-116X60	stainless steel	16	M12	24	18	11	8	60
07534-116X70	stainless steel	16	M12	24	18	11	8	70
07534-116X80	stainless steel	16	M12	24	18	11	8	80
07534-116X90	stainless steel	16	M12	24	18	11	8	90
07534-116X100	stainless steel	16	M12	24	18	11	8	100
07534-116X120	stainless steel	16	M12	24	18	11	8	120
07534-120X30	stainless steel	20	M16	30	22	14	10	30
07534-120X40	stainless steel	20	M16	30	22	14	10	40
07534-120X50	stainless steel	20	M16	30	22	14	10	50
07534-120X60	stainless steel	20	M16	30	22	14	10	60
07534-120X70	stainless steel	20	M16	30	22	14	10	70
07534-120X80	stainless steel	20	M16	30	22	14	10	80
07534-120X90	stainless steel	20	M16	30	22	14	10	90
07534-120X100	stainless steel	20	M16	30	22	14	10	100
07534-120X120	stainless steel	20	M16	30	22	14	10	120

D.6 Jaw coupling GN 2240-20-B6-8-AL-RS

GN 2240



Elastomer jaw couplings

with clamping hub

SPECIFICATION

Bore codes

- Type **B**: without keyway
- Type **K**: with keyway (from $d_1 = 30$)

Hub

Aluminum **AL**
anodized, natural color

Coupling spider
Polyurethane (TPU)
temperature resistant up to 60 °C

Hardness

- 80 Shore A, blue **BS**
- 92 Shore A, white **WS**
- 98 Shore A, red **RS**

Socket cap screws DIN 912
Steel, blackened

Temperature range: -20 °C up to +60 °C

INFORMATION

Elastomer jaw couplings GN 2240 can transmit very high torques while compensating for shaft misalignments and runout tolerances. They are preferred in applications where the focus lies on pure torque and power transmission.

The choice of three coupling spiders with different hardness values allows the properties of the coupling to be optimally matched to the specific requirements. The clamping hubs and simple plug-in installation make jaw couplings very easy to assemble.

With the bore code K, the keyway is always integrated into both bores d_2 and d_3 .

ACCESSORY

- Coupling spiders GN 2240.1 (see page)

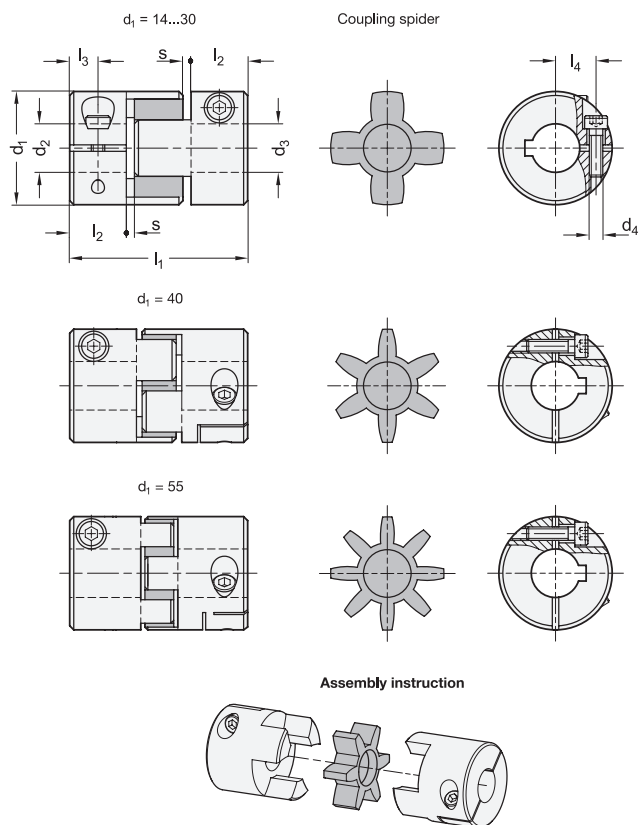
TECHNICAL INFORMATION

- Keyway P9 DIN 6885 (see page A16)
- ISO-Fundamental Tolerances (see page A21)
- Elastomer characteristics (see page A32)



GN 2240**Elastomer jaw couplings**

2/4

**TECHNICAL VALUES**

d1	Coupling spider	Shore hardness coupling spider	Rated torque in Nm	Max. torque in Nm	Max. speed (min ⁻¹)	Moment of inertia in kgm ²	Static torsional stiffness in Nm/rad	Max. shaft misalignment		
								lateral in mm	axial in mm	angular in °
14	BS	80A	0.7	1.4	45.000	2.0×10^{-7}	8	0.15	0.6	1
14	WS	92A	1.2	2.4	45.000	2.0×10^{-7}	14	0.1	0.6	1
14	RS	98A	2	4	45.000	2.0×10^{-7}	22	0.1	0.6	1
20	BS	80A	1.8	3.6	31.000	1.1×10^{-6}	16	0.2	0.8	1
20	WS	92A	3	6	31.000	1.1×10^{-6}	29	0.15	0.8	1
20	RS	98A	5	10	31.000	1.1×10^{-6}	55	0.1	0.8	1
30	BS	80A	4	8	21.000	6.2×10^{-6}	46	0.2	1	1
30	WS	92A	7.5	15	21.000	6.2×10^{-6}	73	0.15	1	1
30	RS	98A	12.5	25	21.000	6.2×10^{-6}	130	0.1	1	1
40	BS	80A	4.9	9.8	15.000	3.7×10^{-5}	380	0.15	1.2	1
40	WS	92A	10	20	15.000	3.7×10^{-5}	570	0.1	1.2	1
40	RS	98A	17	34	15.000	3.7×10^{-5}	1200	0.1	1.2	1
55	BS	80A	17	34	11.000	1.6×10^{-4}	1400	0.2	1.4	1
55	WS	92A	35	70	11.000	1.6×10^{-4}	1600	0.15	1.4	1
55	RS	98A	60	120	11.000	1.6×10^{-4}	2600	0.1	1.4	1

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12/2021

GN 2240

Elastomer jaw couplings

* Complete with shore hardness of the couplings (BS, RS, WS)

BS RS WS

GN 2240-B

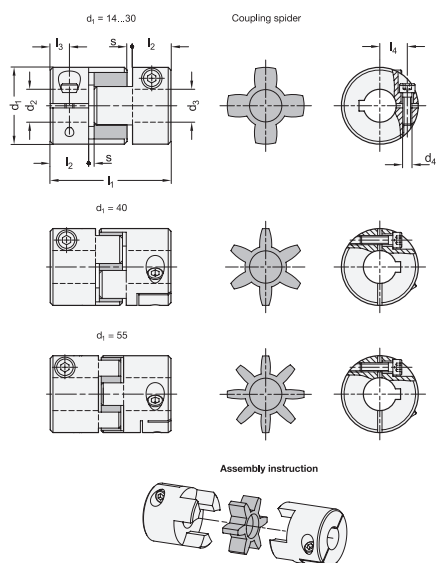
Description	d1	d2 - d3 H8 recommended shaft tolerance h7	d4	l1	l2 recommended shaft insertion depth	l3	l4	s recommended installation spacing	⚖
GN 2240-14-B3-3-AL-*	14	3-3	M 2	22	7	3.5	4	1	9
GN 2240-14-B3-4-AL-*	14	3-4	M 2	22	7	3.5	4	1	9
GN 2240-14-B3-5-AL-*	14	3-5	M 2	22	7	3.5	4	1	9
GN 2240-14-B3-6-AL-*	14	3-6	M 2 / M 1.6	22	7	3.5	4 / 5	1	9
GN 2240-14-B4-4-AL-*	14	4-4	M 2	22	7	3.5	4	1	9
GN 2240-14-B4-5-AL-*	14	4-5	M 2	22	7	3.5	4	1	9
GN 2240-14-B4-6-AL-*	14	4-6	M 2 / M 1.6	22	7	3.5	4 / 5	1	9
GN 2240-14-B5-5-AL-*	14	5-5	M 2	22	7	3.5	4	1	9
GN 2240-14-B5-6-AL-*	14	5-6	M 2 / M 1.6	22	7	3.5	4 / 5	1	9
GN 2240-14-B6-6-AL-*	14	6-6	M 1.6	22	7	3.5	5	1	9
GN 2240-20-B5-5-AL-*	20	5-5	M 2.5	30	10	5	6.5	1	22
GN 2240-20-B5-6-AL-*	20	5-6	M 2.5	30	10	5	6.5	1	22
GN 2240-20-B5-8-AL-*	20	5-8	M 2.5	30	10	5	6.5	1	22
GN 2240-20-B6-6-AL-*	20	6-6	M 2.5	30	10	5	6.5	1	22
GN 2240-20-B6-8-AL-*	20	6-8	M 2.5	30	10	5	6.5	1	22
GN 2240-20-B8-8-AL-*	20	8-8	M 2.5	30	10	5	6.5	1	22
GN 2240-30-B8-8-AL-*	30	8-8	M 4	35	11	5.5	10	1.5	51
GN 2240-30-B8-10-AL-*	30	8-10	M 4	35	11	5.5	10	1.5	51
GN 2240-30-B8-12-AL-*	30	8-12	M 4	35	11	5.5	10	1.5	51
GN 2240-30-B8-14-AL-*	30	8-14	M 4 / M 3	35	11	5.5	10 / 11	1.5	51
GN 2240-30-B10-10-AL-*	30	10-10	M 4	35	11	5.5	10	1.5	51
GN 2240-30-B10-12-AL-*	30	10-12	M 4	35	11	5.5	10	1.5	51
GN 2240-30-B10-14-AL-*	30	10-14	M 4 / M 3	35	11	5.5	10 / 11	1.5	51
GN 2240-30-B12-12-AL-*	30	12-12	M 4	35	11	5.5	10	1.5	51
GN 2240-30-B12-14-AL-*	30	12-14	M 4 / M 3	35	11	5.5	10 / 11	1.5	51
GN 2240-30-B14-14-AL-*	30	14-14	M 3	35	11	5.5	11	1.5	51
GN 2240-40-B12-12-AL-*	40	12-12	M 5	66	25	8.5	14	2	181
GN 2240-40-B12-14-AL-*	40	12-14	M 5	66	25	8.5	14	2	181
GN 2240-40-B12-15-AL-*	40	12-15	M 5	66	25	8.5	14	2	181
GN 2240-40-B12-16-AL-*	40	12-16	M 5	66	25	8.5	14	2	181
GN 2240-40-B14-14-AL-*	40	14-14	M 5	66	25	8.5	14	2	181
GN 2240-40-B14-15-AL-*	40	14-15	M 5	66	25	8.5	14	2	181
GN 2240-40-B14-16-AL-*	40	14-16	M 5	66	25	8.5	14	2	181
GN 2240-40-B15-15-AL-*	40	15-15	M 5	66	25	8.5	14	2	181
GN 2240-40-B15-16-AL-*	40	15-16	M 5	66	25	8.5	14	2	181
GN 2240-40-B16-16-AL-*	40	16-16	M 5	66	25	8.5	14	2	181
GN 2240-55-B18-18-AL-*	55	18-18	M 6	78	30	10.5	20	2	414
GN 2240-55-B18-19-AL-*	55	18-19	M 6	78	30	10.5	20	2	414
GN 2240-55-B18-20-AL-*	55	18-20	M 6	78	30	10.5	20	2	414
GN 2240-55-B18-25-AL-*	55	18-25	M 6	78	30	10.5	20	2	414
GN 2240-55-B19-19-AL-*	55	19-19	M 6	78	30	10.5	20	2	414
GN 2240-55-B19-20-AL-*	55	19-20	M 6	78	30	10.5	20	2	414
GN 2240-55-B19-25-AL-*	55	19-25	M 6	78	30	10.5	20	2	414
GN 2240-55-B20-20-AL-*	55	20-20	M 6	78	30	10.5	20	2	414
GN 2240-55-B20-25-AL-*	55	20-25	M 6	78	30	10.5	20	2	414
GN 2240-55-B25-25-AL-*	55	25-25	M 6	78	30	10.5	20	2	414

Weight type BS

GN 2240

Elastomer jaw couplings

4/4



* Complete with shore hardness of the couplings (BS, RS, WS)

BS RS WS

GN 2240-K

Description	d1	d2 - d3 H8 recommended shaft tolerance h7	d4	l1	l2 recommended shaft insertion depth	l3	l4	s recommended installation spacing	⚖
GN 2240-30-K8-8-AL-*	30	8-8	M 4	35	11	5.5	10	1.5	51
GN 2240-30-K8-10-AL-*	30	8-10	M 4	35	11	5.5	10	1.5	51
GN 2240-30-K8-12-AL-*	30	8-12	M 4	35	11	5.5	10	1.5	51
GN 2240-30-K8-14-AL-*	30	8-14	M 4 / M 3	35	11	5.5	10 / 11	1.5	51
GN 2240-30-K10-10-AL-*	30	10-10	M 4	35	11	5.5	10	1.5	51
GN 2240-30-K10-12-AL-*	30	10-12	M 4	35	11	5.5	10	1.5	51
GN 2240-30-K10-14-AL-*	30	10-14	M 4 / M 3	35	11	5.5	10 / 11	1.5	51
GN 2240-30-K12-12-AL-*	30	12-12	M 4	35	11	5.5	10	1.5	51
GN 2240-30-K12-14-AL-*	30	12-14	M 4 / M 3	35	11	5.5	10 / 11	1.5	51
GN 2240-30-K14-14-AL-*	30	14-14	M 3	35	11	5.5	11	1.5	51
GN 2240-40-K12-12-AL-*	40	12-12	M 5	66	25	8.5	14	2	181
GN 2240-40-K12-14-AL-*	40	12-14	M 5	66	25	8.5	14	2	181
GN 2240-40-K12-15-AL-*	40	12-15	M 5	66	25	8.5	14	2	181
GN 2240-40-K12-16-AL-*	40	12-16	M 5	66	25	8.5	14	2	181
GN 2240-40-K14-14-AL-*	40	14-14	M 5	66	25	8.5	14	2	181
GN 2240-40-K14-15-AL-*	40	14-15	M 5	66	25	8.5	14	2	181
GN 2240-40-K14-16-AL-*	40	14-16	M 5	66	25	8.5	14	2	181
GN 2240-40-K15-15-AL-*	40	15-15	M 5	66	25	8.5	14	2	181
GN 2240-40-K15-16-AL-*	40	15-16	M 5	66	25	8.5	14	2	181
GN 2240-40-K16-16-AL-*	40	16-16	M 5	66	25	8.5	14	2	181
GN 2240-55-K18-18-AL-*	55	18-18	M 6	78	30	10.5	20	2	414
GN 2240-55-K18-19-AL-*	55	18-19	M 6	78	30	10.5	20	2	414
GN 2240-55-K18-20-AL-*	55	18-20	M 6	78	30	10.5	20	2	414
GN 2240-55-K18-25-AL-*	55	18-25	M 6	78	30	10.5	20	2	414
GN 2240-55-K19-19-AL-*	55	19-19	M 6	78	30	10.5	20	2	414
GN 2240-55-K19-20-AL-*	55	19-20	M 6	78	30	10.5	20	2	414
GN 2240-55-K19-25-AL-*	55	19-25	M 6	78	30	10.5	20	2	414
GN 2240-55-K20-20-AL-*	55	20-20	M 6	78	30	10.5	20	2	414
GN 2240-55-K20-25-AL-*	55	20-25	M 6	78	30	10.5	20	2	414
GN 2240-55-K25-25-AL-*	55	25-25	M 6	78	30	10.5	20	2	414

Weight type BS

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Joints, Couplings 10

D.7 Lock nuts SKF KMFE 4 and SKF KMFE 11



KMFE 4



KMFE metric lock nuts with an integral locking screw

KMFE lock nuts with integral locking screw are designed to locate CARB toroidal roller bearings, sealed spherical roller bearings and sealed self-aligning ball bearings axially on a shaft. They reduce the cost of the shaft as no keyway is required. Installation with an integral set screw is quick and easy and no separate locking device is necessary. Maximum axial run-out locating face and thread 0.02 to 0.03 mm.

- No keyway required
- Simple and robust locking for intended applications
- Reusable
- Cost-effective
- Available for thread M 20x1 to M 200x3 (sizes 4 to 40)

Overview

Dimensions

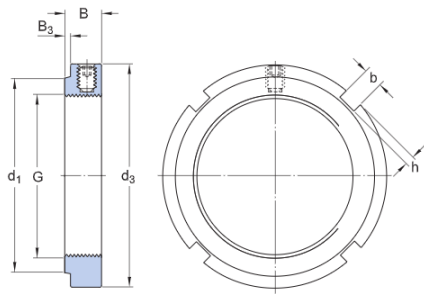
Bore diameter	20 mm
Outside diameter	32 mm
Width	9.5 mm

Properties

Associated mounting tool	HN 4
Locking device	Incorporated in the lock nut
Nut for hydraulic mounting	No
Thread designation	M20x1



Technical Specification



Dimensions

G	M20x1	Thread
d ₃	32 mm	Outside diameter
B	9.5 mm	Width
d ₁	26 mm	Diameter locating side face
B ₃	1 mm	Stand-out of locating face
b	4 mm	Width locating slot
h	2 mm	Depth locating slot

Calculation data

Axial static load carrying capacity	24 kN
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Mass

Mass lock nut	0.034 kg
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Mounting information

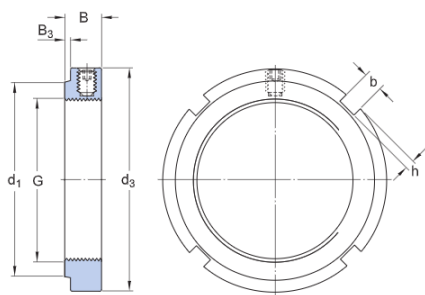
Associated spanner	HN 4
Set screw size	M5
Recommended grub screw tightening torque	4.5 Nm



KMFE 11 H

Porcas de fixação com fixação integral

Especificações técnicas



DIMENSÕES

G	M55x2	Rosca
d ₃	75 mm	Diâmetro externo
B	14 mm	Largura
d ₁	67 mm	Diâmetro da face lateral de fixação
B ₃	3 mm	Folga da face de fixação
b	7 mm	Largura da ranhura de fixação
h	3 mm	Profundidade da ranhura de fixação

DADOS DE CÁLCULO

Capacidade de carga estática básica axial	91.5 kN
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MASSA

Massa da porca de fixação	0.21 kg
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INFORMAÇÕES DE MONTAGEM

Chave associada	HN 10-11
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Tamanho do parafuso de fixação

M6

Torque de aperto recomendado para pino roscado

8 Nm

Mais informações

Detalhes do produto

[Designs e variantes](#)

[Dados do produto](#)

[Instalação e remoção](#)

[Sistema de designação](#)

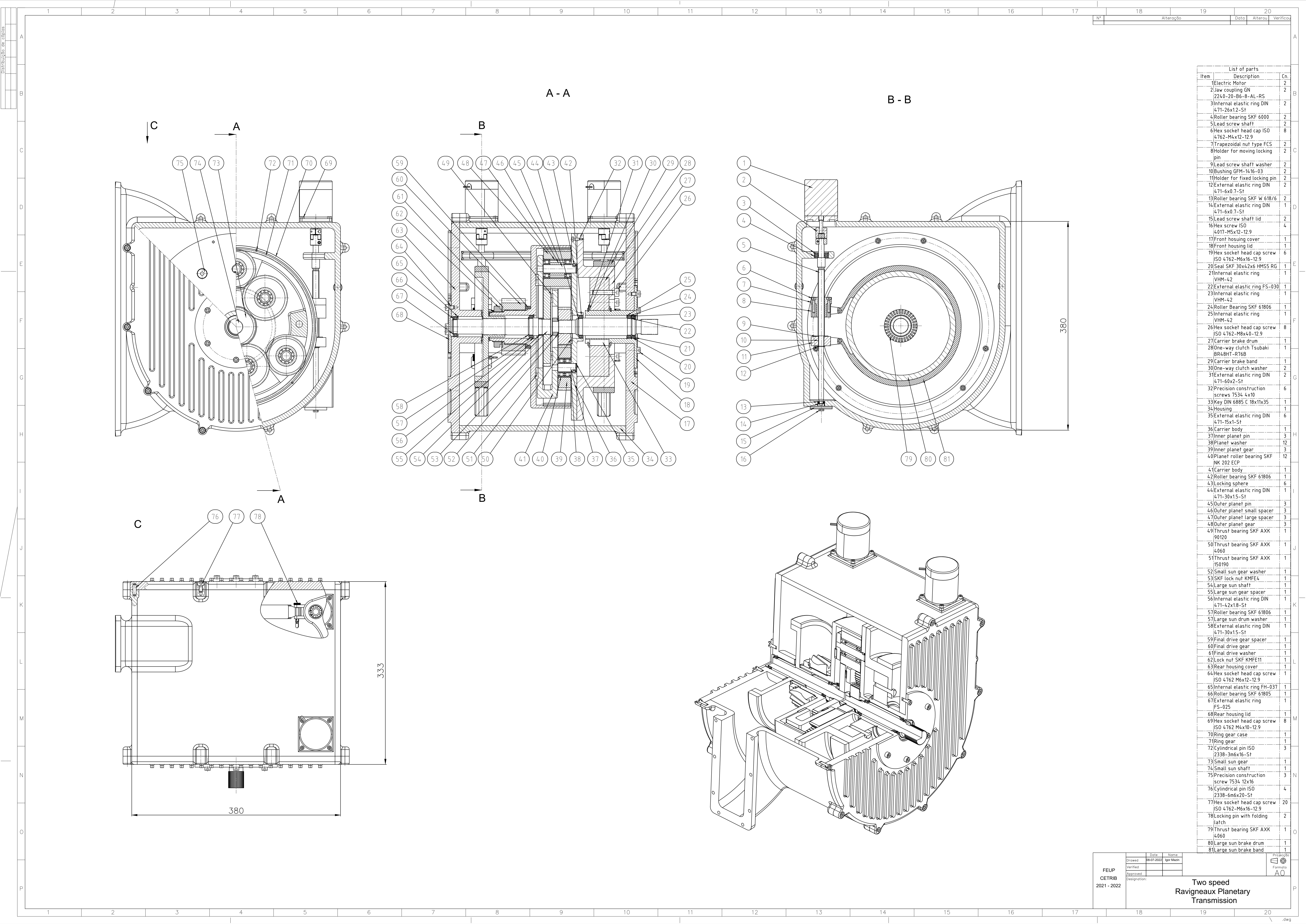
Ferramentas

[Ferramenta de engenharia](#)

Appendix E

Appendix - Technical Drawing

In this appendix, the technical 2D drawing of the transmission is provided.



List of parts		
Item	Description	Cn.
1	Electric Motor	2
2	Jaw coupling GN	2
3	Internal elastic ring DIN	2
4	71-26x12-St	2
5	Roller bearing SKF 6000	2
6	Lead screw shaft	2
7	Hex socket head cap ISO	8
8	4762-M4x12-12.9	8
9	Trapezoidal nut type FCS	2
10	Holder for moving locking pin	2
11	Lead screw shaft washer	2
12	Bushing GFM-14.16-03	2
13	Holder for fixed locking pin	2
14	External elastic ring DIN	2
15	471-6x0.7-St	2
16	Roller bearing SKF W 618/6	2
17	External elastic ring DIN	1
18	471-6x0.7-St	2
19	Lead screw shaft lid	2
20	Hex screw ISO	4
21	4.017-M5x12-12.9	4
22	Front housing cover	1
23	Front housing lid	1
24	Hex socket head cap screw ISO 4762-M6x16-12.9	6
25	Seal SKF 30x42x6 HMS5 RG	1
26	Internal elastic ring VHM-42	1
27	External elastic ring FS-030	1
28	Internal elastic ring VHM-42	1
29	Roller Bearing SKF 61806	1
30	Internal elastic ring VHM-42	1
31	Hex socket head cap screw ISO 4762-M8x40-12.9	8
32	Carrier brake drum BR48HT-R76B	1
33	One-way clutch Tsubaki	1
34	Carrier brake band	1
35	One-way clutch washer	2
36	External elastic ring DIN	2
37	471-60x2-St	6
38	Precision construction screws 7534 4x10	1
39	Key DIN 6885 C 18x11x35	1
40	Housing	1
41	External elastic ring DIN	6
42	471-15x1-St	1
43	Carrier body	1
44	Inner planet pin	3
45	Planet washer	12
46	Inner planet gear	3
47	Planet roller bearing SKF NK 202 ECP	12
48	Carrier body	1
49	Roller bearing SKF 61806	1
50	Locking sphere	6
51	External elastic ring DIN	1
52	471-30x15-St	1
53	Outer planet pin	3
54	Outer planet small spacer	3
55	Outer planet large spacer	3
56	Outer planet gear	3
57	Thrust bearing SKF AXK 90120	1
58	Thrust bearing SKF AXK 4060	1
59	Thrust bearing SKF AXK 150190	1
60	Small sun gear washer	1
61	SKF lock nut KMFE4	1
62	Large sun shaft	1
63	Large sun gear spacer	1
64	Internal elastic ring DIN	1
65	471-42x18-St	1
66	Roller bearing SKF 61806	1
67	Large sun drum washer	1
68	External elastic ring DIN	1
69	471-30x15-St	1
70	Final drive gear spacer	1
71	Final drive gear	1
72	Final drive washer	1
73	Lock nut SKF KMFE11	1
74	Rear housing cover	1
75	Hex socket head cap screw ISO 4762 M6x12-12.9	1
76	Internal elastic ring FH-037	1
77	Roller bearing SKF 61805	1
78	External elastic ring FS-025	1
79	Rear housing lid	1
80	Hex socket head cap screw ISO 4762 M4x10-12.9	8
81	Ring gear case	1
82	Ring gear	1
83	Cylindrical pin ISO 2338-3m6x16-St	3
84	Small sun gear	1
85	Small sun shaft	1
86	Precision construction screw 7534 12x16	3
87	Cylindrical pin ISO 2338-6m6x20-St	4
88	Hex socket head cap screw ISO 4762-M6x16-12.9	20
89	Locking pin with folding latch	2
90	Thrust bearing SKF AXK 4060	1
91	Large sun brake drum	1
92	Large sun brake band	1

FEUP
CETRIB
2021 - 2022

Drawn
18/07/2022
Verified
Igor Mac
Approved
Designation:

Two speed
Ravigneaux Planetary
Transmission

Project
Format
A0