

Design of a cycloid reducer

**—Planetary stage design, shaft design, bearing
selection design, and design of shaft related parts**

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Summary

The RV reducer is one type of two stage cycloid reducers which are widely used in many fields of engineering. This project has designed the first stage of the RV reducer, as well as the related components. The details contain design of input shaft, planetary gears, output shaft, common bearings and eccentric bearings. The fatigue analysis is mostly used in the calculation process because the fatigue failures are frequent in this type of rotation machine. In the same time, the general bearings designs are based on the SKF General Catalogue and the eccentric bearings design are based on the Chinese standard.

All the design components in this project have been dimensioned and achieved good safety factors. They can be seen in the result part in details.

Acknowledgement

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1 Introduction

In this chapter, the design of a specific cycloid reducer is briefly explained. The RV reducer is one type of cycloid reducer that contains two stages .Its planetary stage in particular is discussed in greater detail.

1.1 Background

This bachelor degree project which is about dimensioning and designing the cycloid gearbox has been requested by Swepart Transmission AB. Swepart is a famous competitive manufacturer of customer unique gearboxes, precision-grounded gearwheels, and transmission-parts for vehicles in Sweden. Cycloid reducers are a type of gear transmission. Their main task is to decrease the rotational speed for motors. In many situations, because of the cycloid reducer's unique stable and compact structure, it is more suitable than the spur gear reducer and worm gear reducer; as a result, nowadays it has become widely used in many fields like industrial robots, and wind turbine generators. The widespread usage makes the company desire to develop this kind of gearbox of its own.

Now Swepart has a plan to design and produce such a gearbox that can compete with the Nabtesco (one of the precision cycloid gearbox manufacturer in the world) type cycloid gearbox. For that reason, Swepart needs some general information about the cycloid gearbox to start the design.

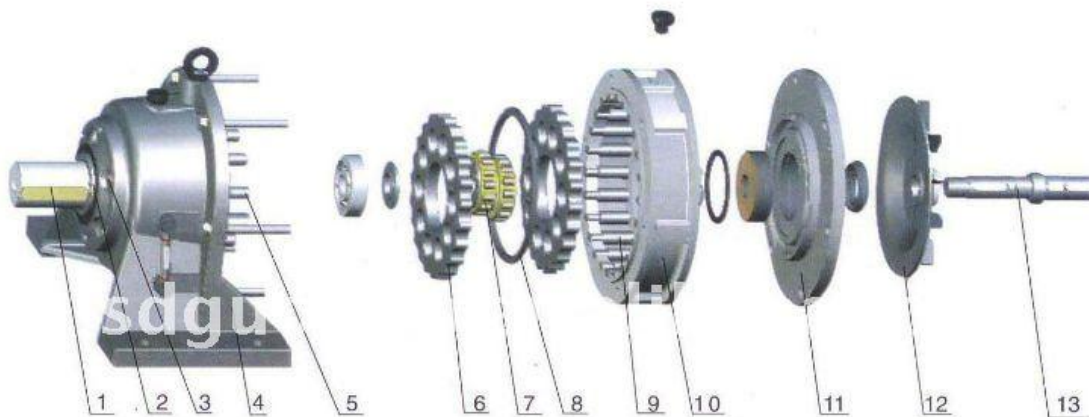


Figure 1 detail view of general cycloid reducer

(<http://www.seekpart.com/company/54008/products/20118462911927.html>) [24 May 2012]

Figure 1 shows a general type of the cycloid reducer where 1 is the output shaft, 2 is the rolling element bearing, 3 is the housing, 4 is the base, 5 is the carrier pins, 6 is the cycloid gear, 7 is the eccentric bearing, 8 is the washer, 9 is the needle tooth pin, 10 is the needle tooth

sleeve, 11 is the housing, 12 the is hub, and 13 is the input shaft. It only has one stage with one input shaft for the cycloid gear which is different from the RV reducer that has two stages.

There are different types of cycloid reduces. RV is one type of the cycloid reducer which has two stages. They are the planetary spur gear stage and the cycloid gear stage. It has several input shafts for the cycloid stage. The RV type is shown on figure 2 below:



Figure 2 nabtesco RV reducer

(<http://www.nabtescomotioncontrol.com/products/rv-e-series-gearbox/>) [20 May 2012]

1.2 Purpose

The purpose of the project is to dimension and design the first stage of the cycloid reducer which includes input shaft, output shaft, key, spline and bearings based on the knowledge from existing cycloid reducer. The arrangement of the cycloid reducer's components follows the RV design shown on figure 2. A lot of dimensions and equations have been determined in this project. In order to obtain a reliable design, a failure criterion of the RV reducer has been fulfilled. Also fatigue analysis is performed to achieve safety factors of the design components. The project provides the design process and working principle for the company as a reference when they manufacture a real RV cycloid gearbox.

1.3 Limitations

Since lacking of enough experience, it is difficult to analyze everything about the gearbox in a limited time. So, the total design is divided into two groups, this project deals with first stage, shafts, shaft related parts and bearings. There is another group who works with the cycloid part in the second stage. And the material heat treatment is unknown.

2 Theory

In this chapter, the RV cycloid gearbox will be generally presented, as well as the first stage of the gearbox which is a planetary spur gear, together with shaft and bearing design consist of how to dimension them and achieve failure analysis.

2.1RV cycloid gear construction

The detail design of RV type reducer can be seen on the figure 3 below:

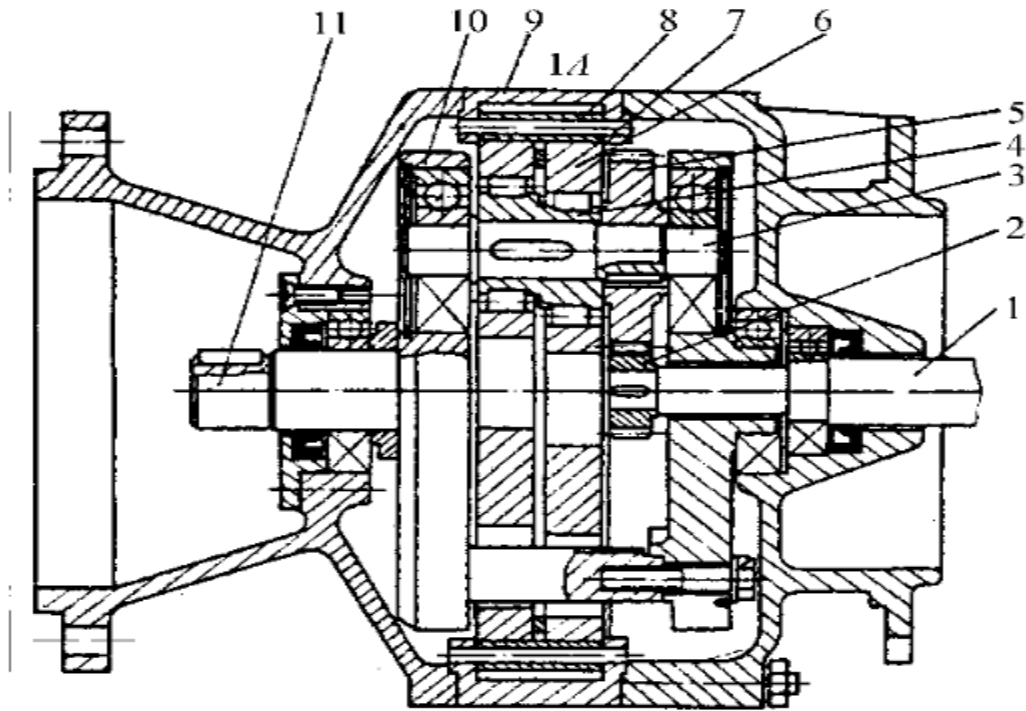


Figure 3 RV gearbox structure scheme (Rao, 1994)

Where 1 stands for input (ingoing) shaft; 2 is input (sun) gear; 3 means crankshaft (carrier); 4 is eccentric sleeve; 5 represent planet pinion; 6 is cycloid gear (RV gear); 7 is needle tooth pin; 8 is needle tooth sleeve; 9 is frame; 10 is support disk; 11 output (outgoing) shaft.

The sun gear usually connects the input shaft by a spine to transmit power from a motor. The several planet pinions allocate equally in a circle to distribute power into the cycloid gear stage. Crankshaft is the rotational shaft of the cycloid gear. It is the connection between the planet gear and support disk. The crankshaft's rotation results in the revolution and rotation of the cycloid gear. In order to accomplish the equilibrium in the radial direction, usually, it has two same cycloid gears arranged on crankshaft by the eccentric sleeve, also the angle between them is 180° . Needle tooth pins fix in the frame and output shaft connect with the support disk as a whole to transmit power. Furthermore the bearing bores which have the same number of crankshaft are set on the support disk.

2.1.1 RV cycloid gearbox divided in two stages

Nowadays, cycloid gearbox has many types of reducers which are widely used in many areas in daily lives such as transportations, high load manufacturing and so on since it has big range of transmission ratio, large load capacity, high torsion stiffness on the shaft and excellent performance on the efficiency. RV cycloid gearbox which belongs to the closed (encased) type planet gear is one member of them. It contains two main stages: the planetary spur gear and the cycloid gear. The details can be observed on the picture below, the first stage is a simple planetary gear and the second stage is the cycloid gear.

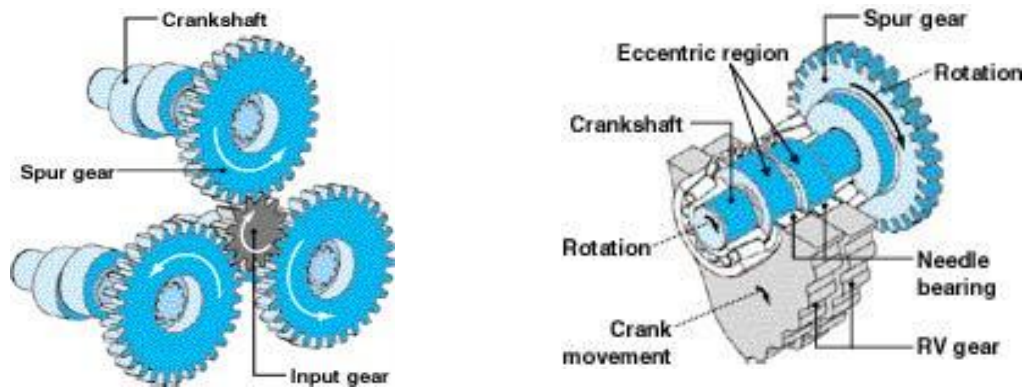


Figure 4 Two stage of the RV cycloid gearbox
(<http://www.nabtescomotioncontrol.com/technology.php>) [20 May 2012]

As mentioned above, this project is to design the first stage of the RV reducer which is shown in the left of figure 4.

2.1.2 The spur gear transmission

Spur gears are one of the simplest and widest used transmission parts around the world. Usually they are used in parallel shaft to transmit power from the input parts. They have a lot of advantages such as high efficiency, relatively low noise, heavy load capacity, smooth transmission, long and service life when high precise gear are manufactured. Such a transmission is shown figure 5 below:



Figure 5 Spur gear
(<http://www.winchbin.com/43/winch-gearing-types-explained/>) [21 May 2012]

There are a lot of dimensions in the gear transmission, the calculations of those dimensions are shown in appendix 3. And always it is used to transmit power. The driving part is called pinion and driven part is called gear. The detail dimensions are shown on figure 6 below as following:

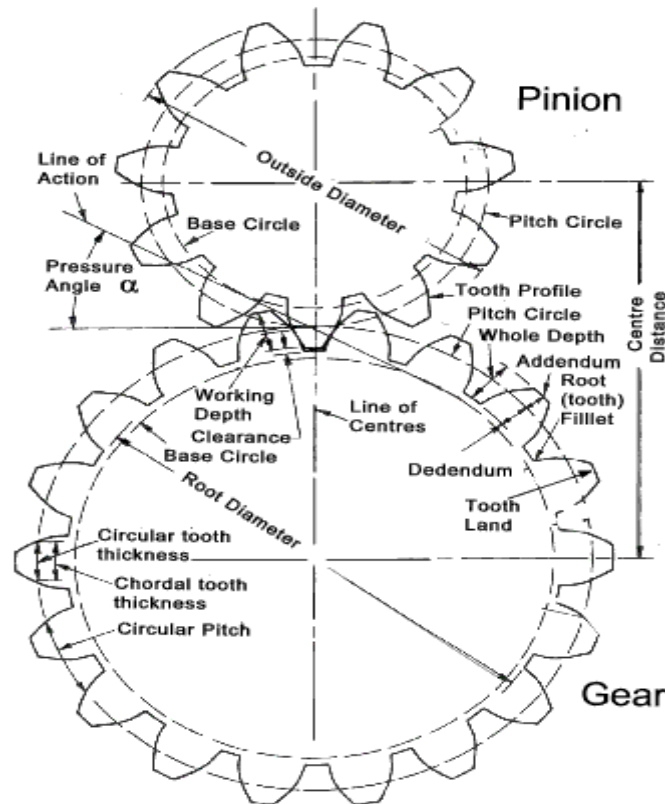


Figure 6 Gear nomenclatures

(http://www.roytech.co.uk/Useful_Tables/Drive/Gears.html) [15 Jan 2011]

Some rules need to be followed when designing the gears transmission. The teeth number should be more than 13 to keep a uniform transmission speed and avoid undercutting (interference) when manufacturing them. Furthermore, the teeth face width should be less than 10 times of module considering hard condition of shaft and bearing will be arranged. What is more, the contact ratio should be between 1 and 2 and tooth thickness on the top should not be smaller than 0.4 times of module.

According to some limitations of the teeth number, they should be integer and prime number. Usually it is very hard to accomplish exactly the same centre distance we need. Thus some addendum modifications need to be set to compromise the centre distance and also it will strengthen the contact gears at the same time.

From figure 4, there are three planet gears in the first stage, so some relations must be achieved to have a perfect assembly.

1) **Adjacency condition**

Since there are three planet gears distribute equally in a circle, the restriction that the addendum circle should not collide each other needs to follow. The equation need to be fulfilled can be seen in the appendix 3.

2) **Concentric condition**

First the centre distance between the sun (centre) gear and several planet gears should be the same. This is due to smooth movement. Secondly, the centre distance between the sun gear α_1 (figure 12 in page 11) and planet gear g_1 (figure 12 in page 11) should be the same with centre distance between the crankshaft and support disk (figure 3 in page 3) which is the radius of the distribution circle for the crankshaft on the support disk to guarantee the input (sun) gear have coaxial relation with the support disk and outgoing shaft.

3) **Crankshaft arrangement**

Usually, there are two cycloid gears in the second stage which are symmetric installation distributed in a circle. When the crankshaft number is odd, the profile of the cycloid gear must stagger $1/2$ tooth ($1/4$ pitch) according to the crankshaft to make sure the two cycloid gear contact the needle pin simultaneously.

The calculation process can be found in the appendix 3.

There are so many failures such as breakage, surface fatigue failure; plastic flow failure and wear failure during the gears perform. Therefore, it is very significant to check whether they have high surface stress and bending stress to find out the safety factor to make sure the gears will perform well during the service life. All the calculation process is presented in the appendix 3.

2.1.3 Shaft

The shaft is one of the most important machine components. The shape of a shaft is commonly like a long cylindrical rod while sometimes it changes depending on different functions. Usually shaft is applied to transmit rotating motion, power and give support to machine elements on it. Gears, pulleys, sprockets, cams and other machine parts can be connected with a shaft to drive it or driven by its rotation.

Shafts can be classified by different forms. In this part will introduce two types which are used to sort shaft. Based on the load subjected to a shaft, it can be three types such as axle, spindle and shaft.

According to the shape of shaft it can be classified into two types which are straight shaft and non straight shaft. Straight shaft is applied during the design process. Straight shaft has only

one straight axis for each part like stepped shaft, smooth shaft, hollow shaft and camshaft etc.

Shafts used in first stage of the designed RV gearbox are belongs to the type which can be subjected to both bending moment and torque. The first stage input shaft is a common used stepped shaft. The output shaft in first stage is also a stepped shaft but with two eccentric regions (on bearings). If consider the shaft and eccentric bearing as an entirety, it is the non-straight type that often being called crankshaft because the axis of bearing is not coincide with the shaft's axis (the rotational axis). Details can be found in 2.1.6 (eccentric bearing).

Stepped shaft is the most commonly used shaft. Like its name interpreted, stepped shaft has a shape that each part on the stepped shaft is concentric but does not have same diameter. Because of that reason, different machine components can be mounted on a same shaft without diameter limitation.

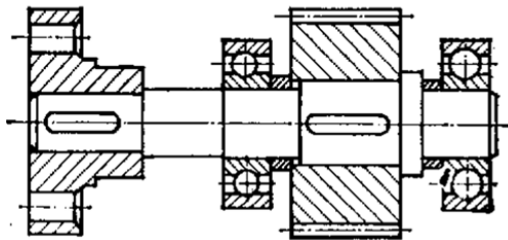


Figure 7 Stepped shaft (<http://www.nly.cn/zhou0.htm>) [15 May 2012]

Basically a stepped shaft consists of three parts. In figure 7, the parts which mounted with hubs are named shaft heads; the parts that mounted with bearings are shaft necks; the last is shaft body which links shaft head and shaft neck together.

2.1.4 Shaft related parts

An individual shaft cannot perform its functionality which is to support and transport motion to machine components mounted on it. The mounted machine component in contact with shaft is named hub. A series of associated parts are needed to fix hubs to shaft and eliminate the possibility of sliding can be called shaft-hub joints.

Shaft related parts also called shaft-hub joints can be separated into two groups.

The first group of shaft related parts is used to avoid the relative axial motion between shaft and hub. These parts are usually shaft shoulder, sleeve and snap ring.

A shaft shoulder is simply the cross section changing parts on a stepped shaft; sleeve is generally used between two hubs which have a small distance; snap ring is a compact and low-cost part to retain hubs.

The second group of shaft related parts is to avoid relative rotation between hubs and shaft. Key and spline are commonly used in this group (figure 8).

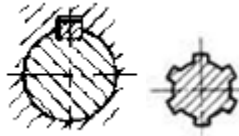


Figure 8 Flat key and 6-teeth spline (Cheng, 2004)

There are a variety of keys to transmit torque, like square key, flat key, round key, etc. They can be chosen depending on specific situation.

Spline can be seen as an integral of several keys on one shaft. Teeth number of four, six, ten and sixteen are common. It can be internal ring which is inside a shaft and external ring which is outside. The teeth shape of a spline can be subdivided into straight-sided and involutes' type. Spline provides more precision, better oriented and can withstand heavy load. This becomes the reason that in the designed RV reducer, the input shaft uses a spline to connect with the input gear.

2.1.5 General Bearing

Bearing is the machine element which is used for constraining relative motion, carrying the shaft loads. Apparently, bearing is indispensable to a shaft to keep shaft's rotating precision, reduce friction and withstand loads.

Basically, according to the motion types of the contact surface, bearing can be separated into two groups: sliding bearing and rolling bearing. Bearings in the designed RV reducer are all belong to rolling element bearing mainly because of the high precise positioning requirement.

According to the shape of roller, rolling-element bearing can be also divided into two types which are ball bearing and roller bearing. Generally, the ball bearing is often for higher speeds while the roller bearing can carry higher loads.

Mostly rolling-element bearing has another classification based on the primarily load act on it. The first category can mainly carry radial load; the second type is named thrust which mainly withstand axial load; the last is angular-contact for combining both radial and axial loads.

Furthermore roller bearing could still be classified again due to different roller configurations such as: cylindrical, spherical, tapered, needle.

On the basis of analysis above, the support bearings on the input shaft of designed RV reducer are chosen as deep groove bearing. Selecting ball bearing is due to the highest rotational speed of this reducer is on the input shaft. The second reason is that the sun gear on the input shaft is a

spur gear, so the axial load should almost not be produced.

A deep groove ball bearing can fulfill all requirements. In figure 9, the bearing construction has briefly introduced. Here 1 means the inner ring, 2 is outer ring, 3 is the ball element between them, and 4 is the retainer which is known as additional part to keep each ball separated.

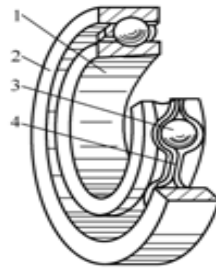


Figure 9 Deep groove ball bearing (Wang, 2007)

Bearing used to support the first stage output shaft of the designed RV reducer is selected to be tapered roller bearing (figure 10). The reason to choose this type of bearings is that the rotational speed on this shaft is already decreased by the first stage but the load on it will be very high that need a roller bearing. The conical elements can carry a certain axial load.

It can be chosen in the SKF General Catalogue according to the equivalent dynamic load and shaft diameter. All the calculation process will follow the SKF General Catalogue which is shown in appendix 6.



Figure 10 Tapered roller bearing

(<http://www.bearingbuy.cn/shop/bearingproduct/taperedrollerbearing/1492.html>) [20 May 2012]

2.1.6 Eccentric bearing

The next bearing named eccentric bearing (figure 11) on the first stage output shaft is always the most important part of the whole reducer. People normally called this bearing to be the heart of the RV cycloid reducer.

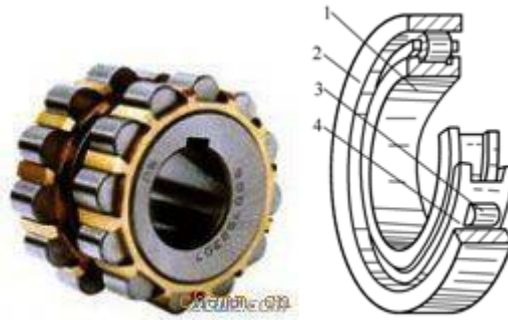


Figure 11 Eccentric bearing and cylindrical roller bearing

(<http://www.zcwz.com/gonghuo/?cpleiid=333>) [15 May 2012]

This bearing type is similar with the cylindrical roller bearing which can withstand a high radial load. The difference between these two types is obvious in figure 11 that is the eccentric region inside and lack of outer ring.

Two eccentric bearings are respectively connected with two cycloid gears. They are staggered 180° when mounted together. The eccentric region is used to be an eccentric sleeve but now it is manufactured as an integral with bearing.

This eccentric part makes the whole shaft could be treated as a crankshaft. The function of the eccentric region is not only to support and constrain the shaft but also to drive and support the cycloid gear. When shaft rotates, eccentric bearings drive cycloid gear revolution, while the eccentric bearings support the cycloid gear to contact with pins. This contact with pins makes cycloid gear rotate in the opposite direction to its revolution.

The series motions from crankshaft (stepped shaft with eccentric bearing) to the cycloid gear are the most important part in the RV reducer. This is the reason why people considered the eccentric bearing as the heart of RV reducer.

Lacking of outer ring is for the purpose of compact design and it can increase sizes of other parts in the eccentric bearing.

2.2 The transmission principle

Generally, it will be more apparent and perceptible to make a drive scheme to have a clear understanding about how the power transmits.

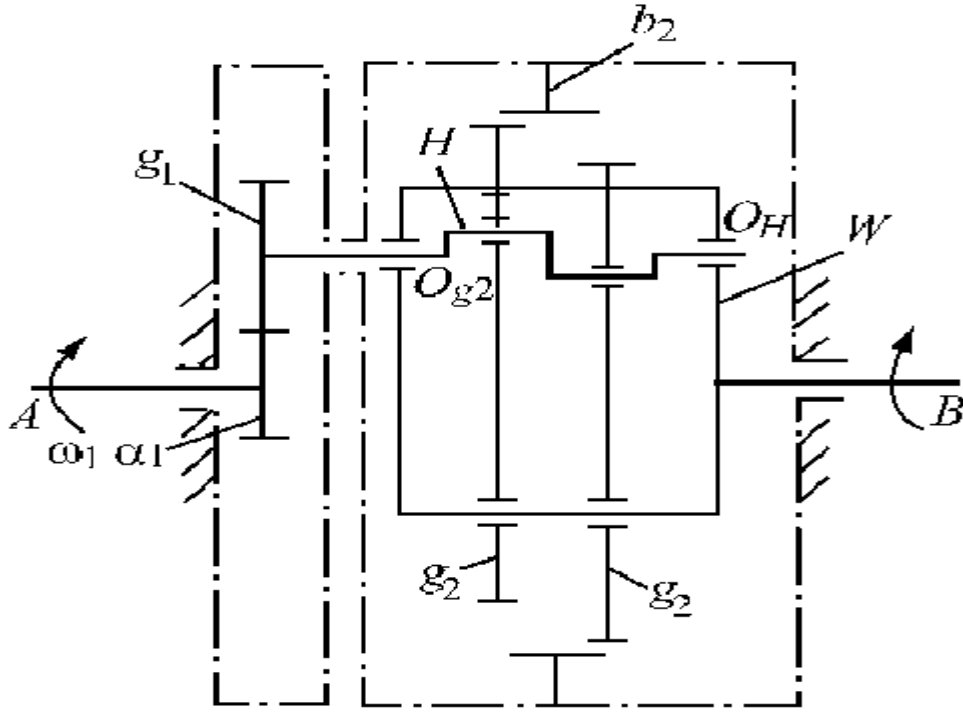


Figure 12 RV gearbox drive scheme (Rao, 1994)

The input gear (α_1) transmits power from motor to the planet gear (g_1) which is the first stage of the RV reducer. Then the rotation of carrier (crankshaft) H will bring about the eccentric motion of the cycloid gear g_2 . When the needle pins are fixed with the frame, the cycloid gear will follow the crankshaft to revolute, at the same moment, it will rotate around the shaft O_{g2} . Through the bearings function on the support disk (figure 3 in page 3), the cycloid gear rotation speed will transmit to the output shaft which means $\omega_v = \omega_{g2}$. It is like the double-crank mechanism which also has the same characteristic.

2.2.1 Calculation of the total ratio

The ratio of this gearbox always equals to the angular speed of input shaft divided by the angular speed of output shaft. Generally, this kind of gearbox has very big ratio which means the output shaft rotational speed will be much lower than the input rotational speed. At the same time, the output will have higher torque. That is why this kind of gearbox can perform well in the precise conditions and also heavy work.

After accomplishing some research for the drive scheme (figure 13 in page 12), it is recognized that there are two stages of gear reducer combined together. The first stage is the simple planet gear mechanism which is the differential mechanism in this gearbox; the second stage is cycloid gear which is the closed (encased) mechanism. Differential mechanism usually has 2 degree of freedom. Closed mechanism means it enclose the centre gear and the crankshaft to have only 1 degree of freedom to make the output certainly.

According to (Rao, 1994), the drive scheme of RV gearbox can be replaced by the drive structure scheme which is shown below.

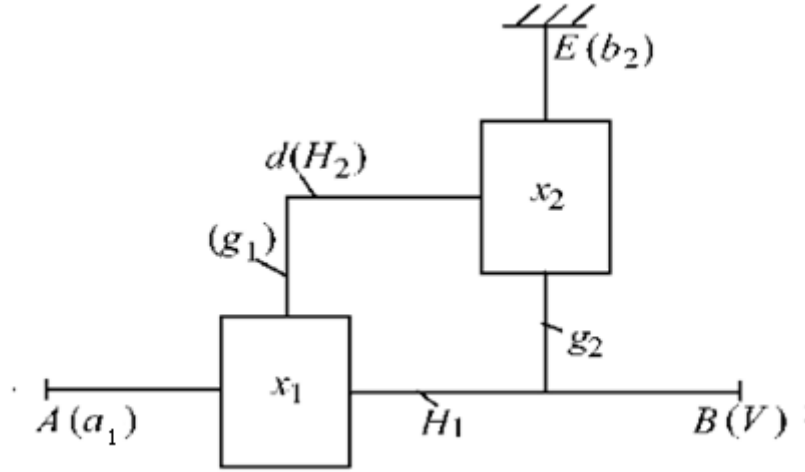


Figure 13 RV gearbox drive structure scheme (Rao, 1994)

It can be noticed from figure 13 above that there are two planetary mechanisms in this kind of gearbox: x_1 and x_2 . The input part is A which stands for the sun gear and the outgoing part is B (V) and output shaft in the figure 13. Also as mentioned before it has the angular speed relation: $\omega_v = \omega_{g2}$. The needle pin is the supporting part E. Planet gear g_1 and crankshaft H_2 are the auxiliary components. And g_2 is the cycloid gear.

In the planetary mechanisms ratio calculation is not as same as usual. It is normally been treated as relative ratio such as i_{AB}^E which is the ratio of angular velocity or rotation speed of input gear A (a_1) and output shaft B (V) relative to pin E. As the pin is fixed $\omega_E = 0$ this ratio is also the total ratio of the RV reducer.

$$i_{\text{tot}} = i_{AB}^E = \frac{\omega_A - \omega_E}{\omega_B - \omega_E} = \frac{\omega_A}{\omega_B} (\omega_E = 0) \quad (\text{Eq 2.1})$$

According to the type of relative ratio there exists some common relationship of it.

For example, $i_{XY}^Z = \frac{\omega_X - \omega_Z}{\omega_Y - \omega_Z}$; $i_{YX}^Z = \frac{\omega_Y - \omega_Z}{\omega_X - \omega_Z}$ the relationship between these two ratios is:

$$i_{XY}^Z = \frac{1}{i_{YX}^Z} \quad (\text{Eq 2.2})$$

$i_{XZ}^Y = \frac{\omega_X - \omega_Y}{\omega_Z - \omega_Y}$, the relationship between this ratio and i_{XY}^Z is:

$$i_{XY}^Z = 1 - i_{XZ}^Y \quad (\text{Eq 2.3})$$

This structure scheme (figure 13 in page 12) consists of two planetary mechanisms x_1 and x_2 . Each mechanism includes three parts. It is used to generate the total ratio.

Mechanism x_1 (first stage):

Input gear A (a_1), $Z_{a_1} = Z_1$ is teeth number of pinion in first stage.

Planet gear g_1 , $Z_{g_1} = Z_2$ is teeth number of gear in first stage.

Planetary frame H_1 is connected with the cycloid gear.

The relative ratio in mechanism x_1 is:

$$i_{a_1 g_1}^{H_1} = -\frac{Z_{g_1}}{Z_{a_1}} = -\frac{Z_2}{Z_1} \quad (\text{negative means opposite rotating direction}) \quad (\text{Eq 2.4})$$

Mechanism x_2 (second stage):

Crankshaft d(H_2) is the planetary frame of this mechanism. And it is also the first stage output shaft and input shaft of second stage. The angular velocity or rotation speed must be the same: $\omega_{g_1} = \omega_{H_2}$

Pin E (b_2), which is fixed, Z_{b_2} is the teeth number of pins. $\omega_{b_2} = 0$

Cycloid gear g_2 : Z_{g_2} is the teeth number of cycloid gear.

The relative ratio in mechanism x_2 is:

$$i_{b_2 g_2}^{H_2} = \frac{Z_{g_2}}{Z_{b_2}} \quad (\text{Eq 2.5})$$

Total output shaft B (V). As mentioned before the angular velocity of cycloid gear is equal to the angular velocity of total output shaft. The cycloid gear is connected with planetary frame H_1 ; the angular velocity is also the same.

$$\omega_{g_2} = \omega_V = \omega_{H_1}$$

As mentioned in (Eq 2.1) of this thesis:

$$i_{tot} = i_{AB}^E$$

According to (Eq 2.3) $i_{XY}^Z = 1 - i_{XZ}^Y$

$$\rightarrow i_{tot} = i_{AB}^E = 1 - i_{AE}^B$$

In figure 13 in page 12, two planetary mechanisms formed the structure scheme that:

$$i_{AE}^B = i_{Ad}^B \times i_{dE}^B$$

$$\rightarrow i_{tot} = 1 - i_{Ad}^B \times i_{dE}^B \quad (\text{Eq 2.6})$$

From figure 13 in page 12 and according to relationship of angular velocity in previous section:

$$i_{Ad}^B = \frac{\omega_{a_1} - \omega_V}{\omega_{H_2} - \omega_V} = \frac{\omega_{a_1} - \omega_{H_1}}{\omega_{g_1} - \omega_{H_1}} = i_{a_1g_1}^{H_1}$$

According to (Eq 2.4) $i_{a_1g_1}^{H_1} = -\frac{Z_{g_1}}{Z_{a_1}} = -\frac{Z_2}{Z_1}$

$$\rightarrow i_{Ad}^B = -\frac{Z_2}{Z_1} \quad (\text{Eq 2.7})$$

From figure 13 in page 12 and according to relationship of angular velocity in previous section:

$$i_{dE}^B = \frac{\omega_{H_2} - \omega_V}{\omega_{b_2} - \omega_V} = \frac{\omega_{H_2} - \omega_{g_2}}{\omega_{b_2} - \omega_{g_2}} = i_{H_2b_2}^{g_2}$$

According to (Eq 2.2) $i_{XY}^Z = \frac{1}{i_{YX}^Z}$

$$\rightarrow i_{dE}^B = i_{H_2b_2}^{g_2} = \frac{1}{i_{b_2H_2}^{g_2}}$$

According to (Eq 2.3) $i_{XY}^Z = 1 - i_{XZ}^Y$

$$\rightarrow i_{dE}^B = \frac{1}{i_{b_2H_2}^{g_2}} = \frac{1}{1 - i_{b_2g_2}^{H_2}}$$

According to (Eq 2.5) $i_{b_2g_2}^{H_2} = \frac{Z_{g_2}}{Z_{b_2}}$

$$\rightarrow i_{dE}^B = \frac{1}{1 - \frac{Z_{g_2}}{Z_{b_2}}} = Z_{b_2} \quad (\text{Eq 2.8})$$

Plug equation 2.4 and 2.8 into equation into equation 2.6

$$i_{\text{tot}} = 1 - \left(-\frac{Z_2}{Z_1}\right) \times Z_{b_2}$$

$$\rightarrow i_{\text{tot}} = 1 + \frac{Z_2}{Z_1} \times Z_{b_2} \quad (\text{Eq 2.9})$$

2.2.2 Efficiency of RV reducer

The designed RV reducer definitely has its own efficiency which is the approximate ratio of the real value and the ideal number. During the transmission process of the RV reducer, energy loss can mainly result from gear contacting, rotating rolling-element bearings and hydraulic loss. The total efficiency is $\eta = \eta_{AE}^B \cdot \eta_n \cdot \eta_M$.

The symbol η_{AE}^B is presented as the gear contacting efficiency. It contains spur gears' contacting in the first stage and contacting of cycloid gear with pins in the second stage. Symbol η_n is the efficiency of all used rolling-element bearings especially for eccentric bearings on the first stage output shaft. Efficiency of the hydraulic is displayed as η_M .

Calculations about the efficiency of the designed RV reducer can be found in Appendix 3.

2.2.3 RV cycloid gear main characteristic

RV cycloid gear has several advantages when comparing to other reducers:

1) Big range of transmission ratio

As the RV reducer has a spur gear level at first, it is convenient to replace the spur gears with different teeth numbers in order to change the transmission ratios. Moreover, the company can make a series of this kind of gearbox without changing a lot of components.

2) Large load capacity

The RV reducer always has several planetary gears and shafts in the first stage which are used to distribute the power from the motor. Furthermore, the output has a supporting disk which also results in high load capacity.

3) **High efficiency**

All of the bearings are rolling bearings except needle teeth pin support which lead to high efficiency during the service life.

4) **Long service life**

Because the RV reducer has first stage reducer, the rotation speed of the output part will not reach very high speed, additionally, the friction and clearance between components is very small. Therefore it usually has very long service life.

2.3 Fatigue analysis

Since all the rotating components are subjected to cycle loading during their service life, it is important to analyze the fatigue strength on each part.

2.3.1 Basic knowledge of fatigue loading

Many machine components suffer fatigue. As a result, it has to be taken into account in machine design. Fatigue failures always come from repeated loading such as bending moment and torsion on the shaft even though the stress level has not reached the yielding strength. It has high stress in some area such as keys, splines, fillets, shoulders that will have more possibility of fatigue failures. Particularly, the initial fatigue crack can easily develop and cause failure because of high stress concentration.

Usually the S-N curve are applied to analyze the material characteristic, it shows the relationship between service life and strength limit after a lot of tests. The S-N curve is shown as following in figure 14.

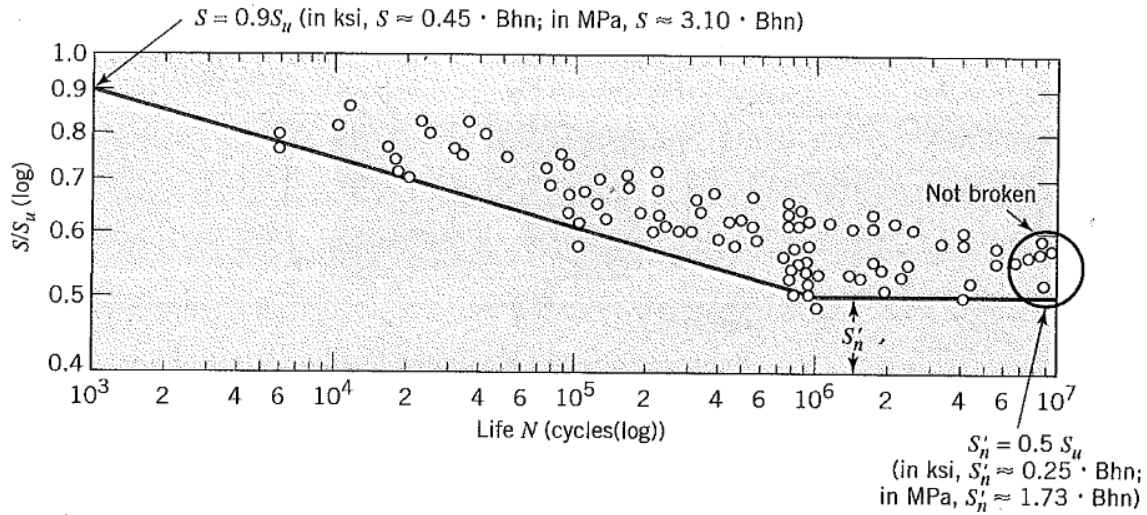


Figure 14 S-N curve (Juvinall and Marshek, 2012)

Where S_u is ultimate strength, S'_n is standard fatigue strength for rotating bending, N is service life. With increasing service life, the strength will decrease until the standard fatigue strength S'_n is achieved. Strengths on two typical life points (10^3 and 10^6) are always used to reflect the reduction of strength. When the life is 10^3 cycles, the strength is 0.9 times ultimate strength. While the life is 10^6 cycles, the strength will decline to 0.5 times ultimate strength.

2.3.2 Spur gears analysis

Fatigue failure of gear (especially on the gear teeth) occurs because of the repeated contacting. This motion generates iterations of loads and stresses that result in fatigue failure on gear surface firstly. Then the failure can propagate from the surface (usually from the pitch circle) to other parts of gear. For this reason, strength limit of gear reduces over time like the general S-N curves (figure 14 in page 16) shows.

It is necessary to analyze the fatigue when designing the spur gears in the first stage of RV reducer. Detail of the spur gear design and fatigue analysis can be found in Appendix 3.

The design process is as same as the usual method according to the Swedish Standard for spur gears. The core of this fatigue calculation is to compare calculated fatigue stress with the calculated fatigue strength. If fatigue stress is higher, the design can be proved to meet the requirement of fatigue.

Based on the Swedish Standard, fatigue stress (σ_H) of spur gears is calculated from this equation (page 8 in Standards and Equations for Gear Design) below:

$$\sigma_H = Z_H \times Z_M \times Z_E \times \sqrt{\frac{F_{ber} \times K_{H\alpha} \times K_{H\beta} \times (u + 1)}{b \times d_1 \times u}} \quad (\text{Eq 2.10})$$

In this equation: Z_H is stress concentration factor in the rolling point; Z_M is material factor; Z_ϵ is meshing factor; F_{ber} is calculation load; $K_{H\alpha}$ is load distribution factor; $K_{H\beta}$ is load propagation factor; b is gear contacting width; d_1 is pitch diameter; u is the speed ratio of gear transmission.

The spur gear fatigue strength (σ_{HP}) is also calculated from Swedish Standard by using the equation (page 10 in Standards and Equations for Gear Design) below:

$$\sigma_{HP} = \frac{\sigma_{Hlim} \times K_L \times Z_R \times Z_v \times K_{HX} \times K_{HN} \times K_{HK}}{S_H} \quad (\text{Eq 2.11})$$

In this equation: σ_{Hlim} is the surface fatigue stress limit; K_L , Z_R , Z_v are factors usually equal to one; K_{HX} is reduction factor due to volume; K_{HN} is life length factor due to surface pressure; K_{HK} is hardness combination factor; S_H is surface fatigue safety factor.

2.3.3 Stepped Shaft Analysis

The stepped shape causes stress concentration. Notches which are the cross sections changing parts could decrease the strength of the shafts. As mentioned before, the usage of stepped shaft is to transmit rotational motion and supporting machine components on it. During its rotating it will bear both bending moment and torque, which will become sources of failure. The stress concentration places are definitely the most dangerous regions.

The stepped shafts in this design withstand bending and torsion, so equivalent bending stress will be generated from the bending and torsion stress in order to calculate the safety factor.

$$\sigma_{ea} = \sqrt{\sigma_a^2 + \tau_a^2} \quad (\text{Eq 2.12})$$

$$\sigma_{em} = \frac{\sigma_m}{2} + \sqrt{\tau_m^2 + \frac{\sigma_m^2}{2}} \quad (\text{Eq 2.13})$$

Symbols: σ_{ea} is equivalent alternating bending stress, σ_a is alternating “normal” stress, τ_a is alternating torsion stress, σ_{em} is equivalent mean bending stress, σ_m is mean “normal” stress, τ_m is mean torsion stress. From the ratio of σ_{ea} and σ_{em} , the possible design overload point can be generated from the figure 15 below.

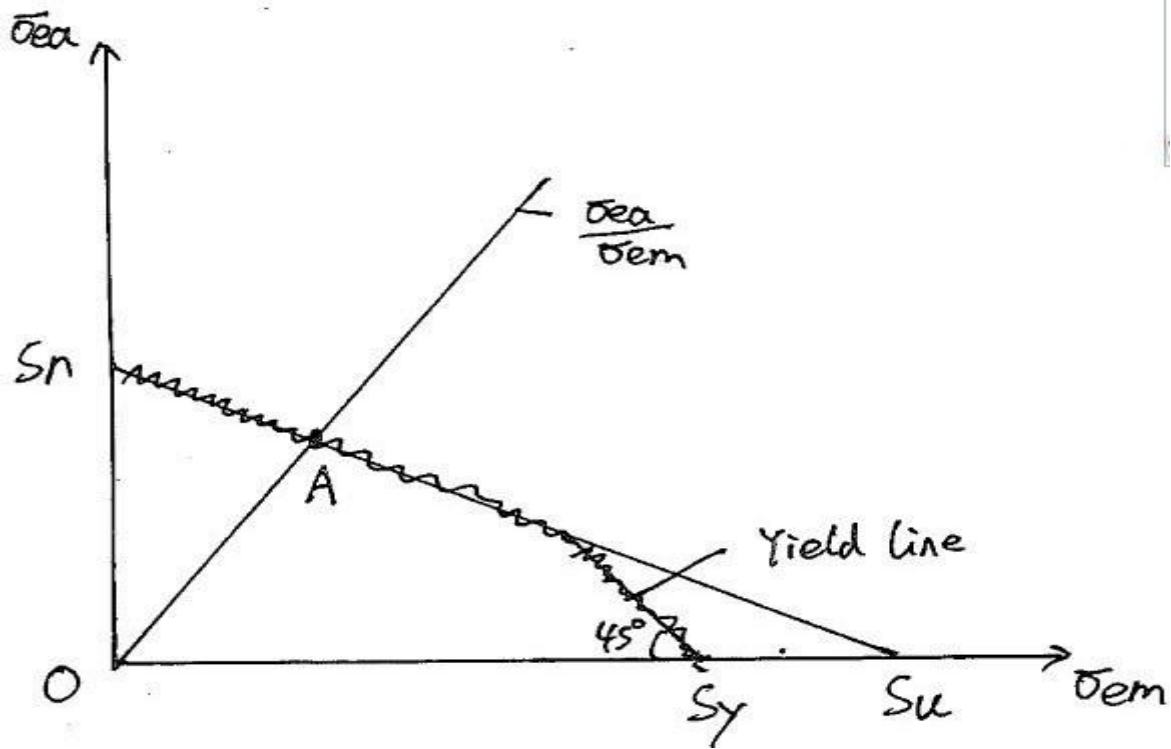


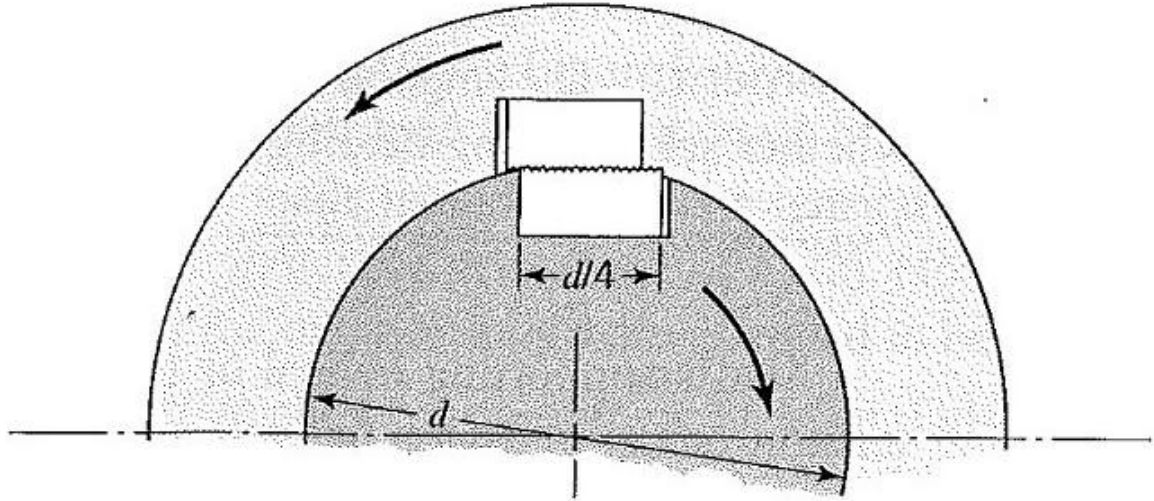
Figure 15 fatigue strength diagram

Where S_n is endurance limit, S_y is yield strength, S_u is ultimate strength. Point A is the design overload point. The fatigue calculation process will be presented in appendix 5.

Based on the failure analysis, the most important part in stepped shaft design is to figure out all types of loads on it. From the force analysis, strength should be checked on all the dangerous regions. According to those analyses the suitable dimensions can be chosen to avoid failures and verify a required working life.

2.3.4 Shaft related parts analysis

It is apparently necessary to take shaft related parts into account in the design process. The way is also to analyze failure on key and spline by checking strength.



(c) Shear failure of a tightly fitted key

Figure 16 Failure on a tightly fitted square key (Juvinal and Marshek, 2012)

Figure 16 shows failure on a tightly fitted square key caused by shear force. For the key itself, shear force is uniformly distributed over the side of key surface. The torque produces shear force which can be treated as same as torque capacity of the shaft. Based on this relation, an eligible key length can be chosen. For the shaft design, it is important to consider stress concentration at the keyway which is the fitted place (like a groove) for the key. Usually engineer can find a fatigue stress concentration factor for keyway in the standard. Then by estimating shaft strength via this factor they can verify shaft diameter. The stress concentration factor can be calculated from the equation below:

$$K_f = 1 + (K_t - 1) \cdot q \quad (\text{Eq 2.14})$$

Where K_f is fatigue stress concentration factor, K_t is static stress concentration factor, q is notch sensitive factor. The calculation process is shown in appendix 5.

Spline can be seen as the part on a shaft with many keys around it. Since that shape feature, strength on a spline is commonly considered to be equal to the strength on a shaft which has the minor spline diameter. Generally during the spline manufacturing some treatments has been made to increase its strength like cold working and residual stresses. These processes make it possible to have almost the same strength with a shaft without spline.

2.3.5 Rolling-element bearing analysis

The calculation process is based on the service life. And also it is generated from fatigue analysis. SKF General Catalogue will be used to calculate the normal bearings. The equation

for service life is:

$$L_{nmh} = a_1 \cdot a_{SKF} \cdot \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P}\right)^p \quad (\text{Eq 2.15})$$

Where L_{nmh} is SKF rating life at $(100 - n)\%$ reliability (according to SKF Catalogue), a_1 is life adjustment factor for reliability, a_{SKF} is SKF life modification factor, n is rotation speed, C is basic dynamic load rating, P is equivalent dynamic bearing load, p is exponent of the life equation.

2.3.6 Eccentric bearing analysis

The eccentric bearing is one of the most important components in RV reducer because it needs to stand a lot of radial forces from the pins. And usually it fails very frequently, so in the eccentric bearing design the service life is based on 5000h. The calculation is based on Chinese standard. The equation for service life is:

$$C = F \sqrt[10/3]{\frac{60nL_h}{10^6}} \quad (\text{Eq 2.16})$$

Where C is selected dynamic load, F equivalent dynamic load, n is rotation speed, L_h

The force analysis on the eccentric bearing is not the same with the normal bearings, so it is necessary to interpret here to make it easier to understand.

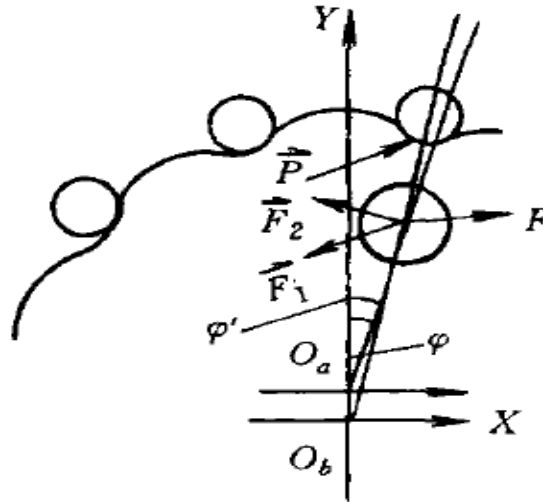


Figure17 Force analysis on the cycloid gear(Yao and Zhang, 1997)

In the figure 17, P is the force from pins to the cycloid gear, and F is the force from bearing to

cycloid gear. F can be divided into F_1 which is equal to $-\frac{1}{n}F$ (n is the number of the bearing) and F_2 which is perpendicular to radius of cycloid gear. The total torque of F_1 to the centre of cycloid gear (O_a) is 0. The total torque of F_2 to the centre of cycloid gear is half of the output torque. The detail calculations are presented in the appendix 4.

The most important factors, when choosing the eccentric bearing, are the eccentric distance and equivalent dynamic load.

3 Method

The methods used to fulfill the project are required data collection by contacting the company, applying design tool such as SolidWorks and Matlab and literature review.

3.1 Required design data

From the company, some input data are given for calculation and design:

- 1) Output torque is 6300Nm
- 2) Output rotation speed is 4.3rpm
- 3) Total ratio is 200-300(best for 240)

And also a plastic model is provided during the design to give us a primary reorganization of this kind of reducer. Furthermore, a real RV reducer as a reference is submitted to make the design more clearly to understand.

3.2 Design tool

As this RV reducer is very complex, sometimes it is advised to apply SolidWorks to make 3D drawing. And also Matlab is used to solve some complicate equations which is hard to calculate by hand, when it comes to the eccentric bearing design.

3.3 Literature review

Calculations about shafts, rolling bearings and gears are based on the Sweden standards. The design process of this part is come from Machine Components Design (Juvinall and Marshek, 2012).

At the same time, it is hard to find some calculation and design process about calculating ratio, efficiency and eccentric bearing in English book and it is easy for us to get some Chinese articles and thesis to accomplish our project task.

4 Results

After a lot of calculations shown in the appendix, the results of each component are generated which will be listed as follows:

4.1 The ratio

The total ratio recommended is 240, so the design will be based on that. There are two stages of mechanism. The first stage is planetary gear and the second stage is cycloid gear. The first stage ratio is 4.06 and the second stage ratio is 58. Then, the total ratio becomes 240.5 (approximately 240 as recommended). The calculation process is in appendix 1.

4.2 Efficiency

As mentioned before in the theory, there are 3 kinds of losses that will affect the efficiency of RV reducer which are meshing friction loss, bearing loss and hydraulic loss:

- Meshing friction efficiency: 93.34%
- Bearing efficiency: 99%
- Hydraulic efficiency: 99%

So the total efficiency is 91.5%. The process is shown in appendix 2.

4.3 First stage of spur gear

The gears in the transmission must be able to withstand the loads that occur while driving the reducer; these loads give both bending stress and surface stress on the gear teeth. As a result, the gears need to be proved to have enough strength or have a safety factor. The gear specifications are following:

Table 4.1 gear specification

	Pinion	Gear
Reference profile	SMS1871	SMS1871
Module	2mm	
Pressure angle	20°	
Teeth number	17	69
Helix angle	0°	

Pitch diameter	24mm	138mm
Addendum modification	+0.7	+0.71
Addendum diameter	40.28mm	144.32mm
Dedendum	1.1mm	1.08mm
Dedendum diameter	31.8mm	135.84mm
Base diameter	32mm	130mm
Center distance	86mm	
Total contact ratio	1.24	
Face width	20mm	18mm

Calculations of these specifications are supported in appendix 3, after we had found out all the specifications for the gears we calculated the strength of the gears from both bending stress and surface pressure stress. And the strength is enough when safety is 2. Also the calculation process is shown in appendix 3. The drawings of the gears are showing in appendix 11.

4.4 Eccentric bearing

The force will vary a lot when the gears rotate, and the equation for the force is very complicate. So, this equation is calculated it by using Matlab which is show in appendix 4. And then the maximum force is generated. And then the service life also can be calculated.

- Dynamic load: 49452N
- Service life: 10230h
- Inner diameter of bearing: 22mm
- Outer diameter of bearing: 53.5mm
- Bearing designation: 180752904

All the calculation process can be found in appendix 4.

4.5 Output shaft of first stage

Three stepped shafts are the output shafts of first stage. They are placed in a circle with same angle to each other. Every shaft is manufactured to same shape by same material, mounted with

same components on it. Based on the above reasons, it only needs to design one shaft in the process. All the design processes can be found in appendix 5.

Two selected eccentric bearings (appendix 4), two tapered roller bearings (appendix 6) and one designed gear are mounted on this shaft.

Forces loaded on the shaft are from gear and eccentric bearings which have been calculated in appendix 5.

- Forces from eccentric bearing: $F_x = 10355\text{N}$ (horizontal view), $F_y = 4360\text{N}$ (vertical view).
- Forces from gear: $F_{r2} = 200\text{N}$ (radial force), $F_{t2} = 550\text{N}$ (tangent force).

According to these forces, the maximum bending moment and the maximum torque can be generated. The maximum bending moment and the maximum torque is at section E-E (figure 18) where the right eccentric bearing mounted. The axial load is zero.

- Maximum bending moment: $M = 72484\text{N} \cdot \text{mm}$ (at section E-E)
- Maximum torque: $T = 37950\text{N} \cdot \text{mm}$ (at section E-E)

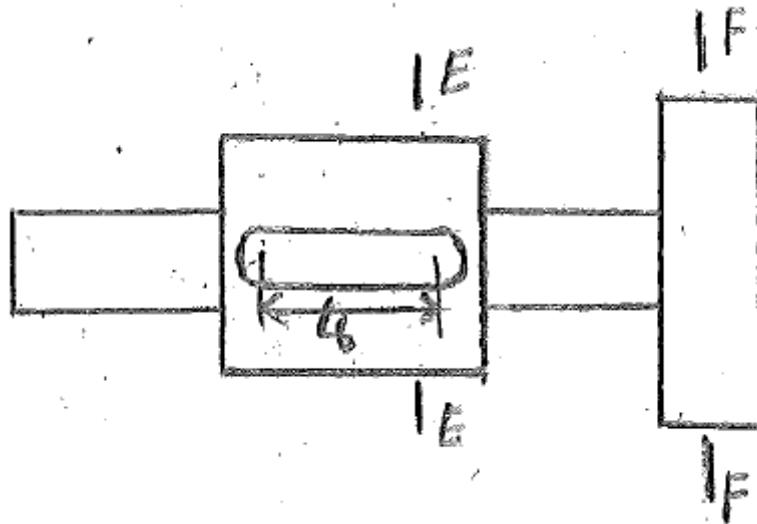


Figure 18 First stage output shaft

The equivalent alternating bending stress and equivalent mean bending stress are calculated when safety factor is one.

- Equivalent alternating bending stress: $\sigma_{ea} = 104\text{MPa}$

-
- Equivalent mean bending stress: $\sigma_{em} = 63.2\text{MPa}$

From the fatigue strength diagram, the allowable equivalent alternating and mean bending stress can be found. Then the safety factor at section E-E is generated.

- Allowable equivalent alternating bending stress: $\sigma_{ea} = 300\text{MPa}$
- Allowable equivalent mean bending stress: $\sigma_{em} = 71\text{MPa}$
- Safety factor at section E-E: $SF_1 = 2.88$

The other safety factor from section F-F is also generated. This part of shaft which has minimum diameter is mounted with gear.

- The torsion stress when safety factor is one: $\tau_m = 115\text{MPa}$
- Shear yield strength: $S_{ys} = 435\text{MPa}$
- Safety factor at section F-F: $SF_2 = \frac{435}{115} = 3.8$

By comparing those two safeties factor the final safety factor for whole shaft is $SF = 2.88$. The drawing of the first stage of output shaft is presented in appendix 11.

4.6 Tapered roller bearing

Two tapered roller bearing has been used on the first stage output shaft. From Appendix 5 loads on bearing A and bearing B are only radial forces that they can be treated as the Equivalent dynamic bearing loads. Compared value of load on bearing A to bearing B: $B_r = 2258\text{N} < A_r = 2842\text{N}$, as they are selected in the same type, it is enough to analyze only bearing A.

The required reliability for all bearings on the first stage output shaft is 90%. As the number of bearings is 4 (2 tapered roller bearings and 2 eccentric bearings), the reliability of one tapered roller bearing is:

$$\text{Reliability} \approx \sqrt[4]{90\%} = 97.4\%$$

It is finally chosen that to calculate the SKF rating life at 98% reliability.

The type of tapered roller bearing is 30204J2/Q.

The SKF rating life (appendix 6) is calculated to be: $L_{2mh} = 55348\text{h}$

4.7 Input shaft design

Because there are three planet spur gears meshing with the center gear, the tangent forces and radial force counteract with each other. The resultant force on the shaft is 0. There is only torsion on the shaft in the place where it has a key. Also a safety factor is generated by comparing the yield torsion stress. The calculation process is in the appendix 7.

- Shaft diameter: 20mm
- Safety factor: 6.86.

The drawing of the input shaft is shown in appendix 11.

4.8 Check strength of spline and key

In the calculation process of the first stage output shaft, the part where mounted with the gear is treated as flat key instead of spline. In reality the spline is used to connect the shaft and gear. It is necessary to select the type of spline and check the shear stress on it. The calculation process can be found in Appendix 8.

- Splines number: $N = 6$
- Inner diameter of spline: $d_5 = 18\text{mm}$
- Outer diameter of spline: $D_1 = 20\text{mm}$
- Spline width: $W = 5\text{mm}$
- Height of spline: $h = 1\text{mm}$
- The shear stress of the spline is 44.4MPa that is lower than the allowable stress $[\sigma_p]_1 = 120\text{MPa}$.

Keys on the output and input shaft of the first stage should also be selected. Shear stresses of them are calculated to compare with allowable shear stress.

Key on the first stage output shaft:

- Key thickness $H = 6\text{mm}$.
- Working length of the key $l = 20\text{mm}$.
- Key width $B = 6\text{mm}$

-
- The shear stress of this key is 52.3MPa that is lower than the allowable shear stress $[\sigma_p]_2 = 110\text{MPa}$

Key on the first stage input shaft:

- Key thickness $H = 6\text{mm}$.
- Working length of the key $l = 22\text{mm}$.
- Key width $B = 6\text{mm}$
- The shear stress of this key is 43.3MPa that is lower than the allowable shear stress $[\sigma_p]_2 = 110\text{MPa}$

4.9 Output shaft

On the output shaft, there are support disk and roller pins. The diameter of the output shaft is 90mm. Since it needs bearings to carry the shaft and the bending moment from the roller pins, a bigger diameter of the output shaft is recommended. So the output shaft is also a stepped shaft, where is shown in the appendix 9. The bigger diameter that has two bearings is 110mm. And the safety factor for output shaft is 4.9. The drawing of the output shaft is shown in appendix 11.

4.10 Bearing on the output shaft

Two deep groove bearing will be selected. From appendix 10, the reaction force is calculated and the bigger force on the bearing is 118628N.

The required reliability for all bearings on the first stage output shaft is 90%. As the number of bearings is 2. The reliability of one tapered roller bearing is:

$$\text{Reliability} \approx \sqrt[2]{90\%} = 94.8\%$$

It is finally chosen that to calculate the SKF rating life at 95%.

The type of deep groove bearing is: *6022-2Z

The SKF rating life (appendix 10) is calculated to be: $L_{5mh} = 4461\text{h}$

5 Analyses of results

The results from last chapter need to be analyzed so that the main design problems can be found

and suggestion for improvement can be proposed.

5.1 Ratio

The total ratio is 240.5 not exactly the same with 240 which is recommended. Because the gear teeth should be complete number and they should be mutually prime numbers, it is very difficult to achieve 240. So, only an approximate number is reached and it also meets the requirement.

During the calculation process of this kind of planetary gear transmission, the relative rotation speed is used to get absolute rotation speed such as the rotation speed of the first stage output shaft. So, it is very important to distinguish the relative and absolute speed correctly when calculating the ratio.

5.2 Efficiency

Normally, RV reducer has very high efficiency when it works. According to the calculations and results, lots of factors should be taken into consideration to gain the efficiency such as the number of gear and pinion teeth, the number pin and cycloid gear teeth, short width coefficient and some other coefficient.

5.3 First stage gears

There are three same planet gears meshing with one sun gear to distribute the input torque in the RV reducer. As a result, only one planet gear and the sun gear are chosen to accomplish the calculation. The simplified picture of the first stage gear arrangement is shown as follows.

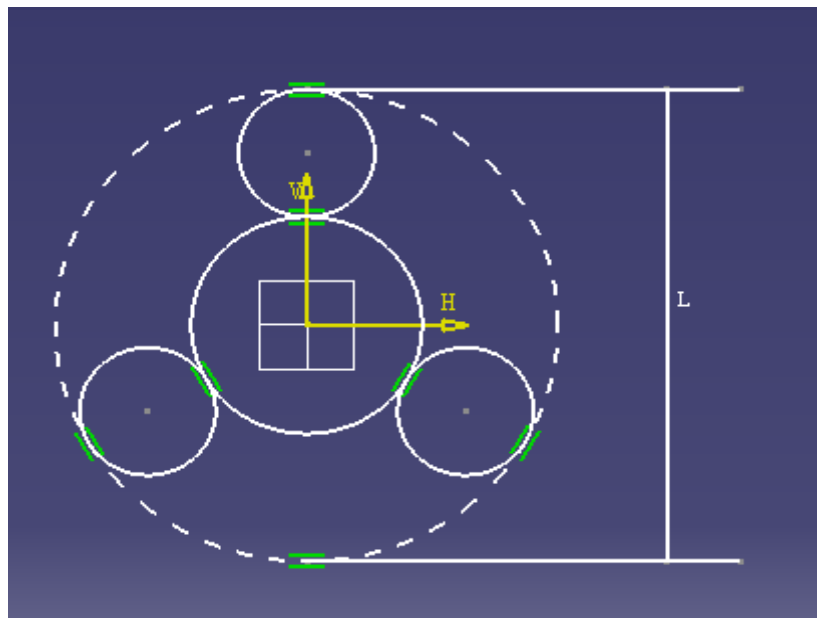


Figure 19 The simplified picture of the first stage

The length L in figure 19 should approximate to the diameter of cycloid gear in the second stage

according to considering the total arrangement.

The material of gears is 20MnCr6 as recommended. It can meet the requirements of bending strength and surface fatigue strength when safety factor is two. To achieve a bigger safety factor, the company can choose better way of heat treatment to increase the strength such as carburizing hardening.

5.4 Eccentric bearing

As mentioned before in the theory part, the eccentric bearing is one of the most important parts of the RV reducer. And it should withstand a lot of loads so its service life is only 10230h which is shorter than other kinds of normal bearings in the RV reducer. As a result, these eccentric bearings need careful maintenance and precise installation. The calculation approach is based on the Chinese standard.

From figure 2 in the appendix 4 page 2, it is obvious that the force in the Y direction from the pins to the eccentric bearing is periodic variation and its period is 2π . The service life is based on the maximum force during one period.

5.5 Output shaft of the first stage

Spline is manufactured on the shaft to connect with the gears. In the process of shaft design, the method to calculate the strength is used to treat the spline as a flat key to ensure better safety, because the spline is stronger than flat key.

The shaft diameter is based on the selecting eccentric bearing because of its standardization. The biggest diameter is equal to the inner diameter of the eccentric bearing.

The gears on the shaft are all spur gears and the force from the pins to the bearing is radial force, so there are bending and torsion stress on the shaft without any axial loads. Comparing the bending and torsion stress to equivalent alternating bending stress from fatigue strength diagram to generate the safety factor.

The material is the same with gears in the first stage. In order to increase safety factor, the better heat treatment can be applied to get higher strength.

5.6 Tapered roller bearing

Although there are no axial loads in the shaft, the tapered roller bearings are chosen because they are used in the model given from the company. According to the SKF General Catalogue, the calculated service life is 55348h. It has longer life than eccentric bearing because its load is smaller. All the forces are maximum reaction force from appendix 5.

5.7 Input shaft

As mentioned in the result, the forces counteract with each other. Consequently, the stress on the shaft is only from the torque, and it can guarantee a higher safety factor because of lower

stress. And it is an advantage of this kind of planetary gear transmission design.

5.8 Spline and key

Spline is used to connect the gears with first stage output shaft. Keys are used in first stage output shaft to connect eccentric bearing and in the input shaft to mount pinion. It is shown in the result all the shear stresses on the spline and keys can meet the safety requirement. This proved that it is strong enough to use a key on the input shaft instead of spline.

5.9 Output shaft

Since it needs bearings on the output shaft and the bearing will carry very high load, it is selected that the shaft where has bearings has diameter of 110mm because the big bearing are needed to have more service life. And the material is also the same with the gear. In order to increase the safety factor, also better heat treatment can be applied to enlarge the strength of the material.

5.10 Deep groove bearing

As the rotation speed is so low and the force on the bearing is very high, the SKF rating life is 4461h. There are two bearings on the output shaft to carry the shaft because it needs high load capacity. The service life is not so high is mainly because the rotation speed is very low and the force on the bearing is very high because of the output torque.

6 Conclusions

Through the first stage spur gear's design, the total ratio becomes much bigger. In the same time it will increase the service life because the planetary gear will reduce the output shaft of first stage (crankshaft) rotation speed. The two stage arrangement results in that the rotation speed of cycloid gear would not be so high. Furthermore, three planetary gears are used to insure the high load capacity. In this design, the first stage won't have lots of spaces, which can keep the whole reducer in compact.

After the force analysis and fatigue analysis on the input shaft, the planetary gear, output shaft of the first stage and the bearings, all the requirements can be achieved. The first stage of RV reducer's design has been accomplished. Totally, this project has reached the needs according to the company.

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Appendix 1 ratio calculation

Symbol	Explanation	value	Other
i_{tot}	Total ratio	240.5	Calculated
Z_{b_2}	Pin number	59	Given
Z_{g_2}	Cycloid gear teeth number	58	Given
Z_2	Gear teeth number	69	Selected
Z_1	Pinion teeth number	17	Selected
i	First stage ratio	4.06	Calculated
n_v	Total output rotation speed	4.3rpm	Given
n_{a_1}	Total input gear rotation speed	1034.15rpm	Calculated
n_{g_1}	First stage output shaft rotation speed	249.4rpm	Calculated

Given requirement:

- Total ratio of RV reducer ≈ 240
- Pin number $Z_{b_2} = 59$
- Cycloid gear teeth number $Z_{g_2} = 58$. ($Z_{b_2} - Z_{g_2} = 1$)
- $Z_1 \geq 13$ (To design a combination drive)
- Total output rotation speed $n_v = 4.3\text{rpm}$

Selection of first stage gears teeth

According to (Eq 2.9) in the theory 2.2.1

$$\text{Equation } i_{tot} = 1 + \frac{Z_2}{Z_1} \times Z_{b_2}$$

$$\rightarrow \frac{Z_2}{Z_1} = \frac{i_{tot} - 1}{Z_{b_2}} \approx \frac{240 - 1}{59}$$

$$\rightarrow \frac{Z_2}{Z_1} \approx 4.05$$

Select:

$$Z_1 = 17$$

$$Z_2 = 69$$

$$\rightarrow \text{Ratio of first stage } i = \frac{z_2}{z_1} = \frac{69}{17}$$

$$\rightarrow i = 4.06$$

The new total ratio of RV reducer:

$$i_{\text{tot}} = 1 + \frac{69}{17} \times 59 = 240.5$$

Find rotation speed of first stage output shaft

The total output shaft rotation speed is $n_v = 4.3\text{rpm}$

The total input gear rotation speed

$$n_{a_1} = n_v \times i_{\text{tot}} = 4.3 \times 240.5$$

$$\rightarrow n_{a_1} = 1034.15\text{rpm}$$

$$\text{Equation (1) } i_{a_1g_1}^{H_1} = -\frac{z_{g_1}}{z_{a_1}} = -\frac{z_2}{z_1}$$

$$i_{a_1g_1}^{H_1} = \frac{\omega_{a_1} - \omega_{H_1}}{\omega_{g_1} - \omega_{H_1}} = \frac{\omega_{a_1} - \omega_v}{\omega_{g_1} - \omega_v} = -\frac{z_2}{z_1}$$

$$\frac{n_{a_1} - n_v}{n_{g_1} - n_v} = -\frac{69}{17}$$

$$\frac{1034.15 - 4.3}{n_{g_1} - 4.3} = -4.06$$

$$\rightarrow n_{g_1} = -249.4\text{rpm} \text{ (Negative means rotate in opposite direction)}$$

Appendix 2 Calculation of efficiency

Symbol	Explanation	Value	Other
Z_{a_1}	Sun gear teeth number	17	Selected
Z_{g_1}	Planet gear teeth number	69	Selected
Z_{b_2}	Pin number	59	Selected
Z_{g_2}	Cycloid gear teeth number	58	Selected
$i_{a_1g_1}^{H_1}$	Ratio when a_1 is input, g_1 is output and H_1 is the relative component	-4.06	Calculated
$i_{H_2b_2}^{g_2}$	Ratio when H_2 is input, b_2 is output and g_2 is the relative component	59	Calculated
$\Psi_{a_1g_1}^{H_1}$	Lost coefficient for the first stage	0.000675	Calculated
Ψ^{H_2}	Lost coefficient for the cycloid gear	0.000849	Calculated
$\eta_{a_1g_1}^{H_1}$	Efficiency when a_1 is input, g_1 is output and H_1 is the relative component	0.98	Calculated
$\eta_{H_2b_2}^{g_2}$	Efficiency when H_2 is input, b_2 is output and g_2 is the relative component	0.952	Calculated
$\eta_{a_1g_2}^{b_2}$	Meshing efficiency	0.9334	Calculated
η_n	Bearing efficiency	0.99	Calculated
η_d	Efficiency when considering hydraulic loss	0.99	Selected
η	Total efficiency	91.5%	Calculated

Calculate the ratio when a_1 is input, g_1 is output and H_1 is the relative component.

$$i_{a_1g_1}^{H_1} = -\frac{Z_{g_1}}{Z_{a_1}} = -\frac{69}{17} = -4.06$$

Calculate the ratio when H_2 is input, b_2 is output and g_2 is the relative component.

$$i_{H_2b_2}^{g_2} = \frac{Z_{b_2}}{Z_{b_2} - Z_{g_2}} = \frac{59}{59 - 58} = 59$$

Lost coefficient for the first stage:

$$\Psi_{a_1g_1}^{H_1} = 2.3f_{z1} \left(\frac{1}{Z_{a_1}} + \frac{1}{Z_{g_1}} \right) = 2.3 * 0.04 * \left(\frac{1}{17} + \frac{1}{69} \right) = 0.000675$$

Lost coefficient for the cycloid gear:

$$\Psi^{H_2} = \frac{K_z * f_{z2}}{Z_{b_2}} = \frac{1.67 * 0.03}{59} = 0.000849$$

The efficiency when a_1 is input, g_1 is output and H_1 is the relative component.

$$\eta_{a_1 g_1}^{H_1} = (1 - \Psi_{a_1 g_1}^{H_1})^n = (1 - 0.00675)^3 = 0.98$$

The efficiency when H_2 is input, b_2 is output and g_2 is the relative component.

$$\eta_{H_2 b_2}^{g_2} = \frac{1 - \Psi^{H_2}}{1 + Z_{g_2} \Psi^{H_2}} = \frac{1 - 0.000849}{1 + 58 * 0.000849} = 0.952$$

So the meshing efficiency when a_1 is input, g_2 (B) is output and b_2 is relative component is:

$$\eta_{a_1 g_2}^{b_2} = \frac{1 - i_{a_1 g_1}^{H_1} * i_{H_2 b_2}^{g_2} (\eta_{a_1 g_1}^{H_1} \eta_{H_2 b_2}^{g_2})}{i_{AB}^E} = \frac{1 - (-4.06) * 59 * 0.98 * 0.952}{240.5} = 0.9334$$

Bearing efficiency:

$$\eta_n = 1 - 0.012 \eta_{a_1 g_2}^{b_2} = 1 - 0.012 * 0.9334 = 0.99$$

Efficiency when considering hydraulic loss (η_d) is approximately 99%

So the total efficiency:

$$\eta = \eta_{a_1 g_2}^{b_2} \eta_d \eta_n = 0.9334 * 0.99 * 0.99 = 0.915 = 91.5\%$$

Appendix 3 Calculation of first stage gear

Symbol	Explanation	Value	Other
z_2	Gear teeth number	69	From appendix 1
z_1	Pinion teeth number	17	From appendix 1
m	Module	2mm	Selected
x_1	Addendum modification of pinion	0.7	Selected
x_2	Addendum modification of gear	0.71	SMS 1871 diagram 8.5.2 (reference figure 3.1)
Y_{F1}	Stress concentration factor of pinion	2.02	SMS 1871 diagram 8.5.2 (reference figure 3.1)
Y_{F2}	Stress concentration factor of gear	2.02	SMS 1871 diagram 8.5.2 (reference figure 3.1)
d_1	Pitch diameter of pinion	34mm	Calculated
d_2	Pitch diameter of gear	138mm	Calculated
α	Pressure angle	20°	Selected
d_{b1}	Base diameter of pinion	32mm	Calculated
d_{b2}	Base diameter of gear	130mm	Calculated
a	Reference center distance	86mm	Calculated
α_w	Pressure angle at rolling circle	24.15°	Calculated
a_w	Centre distance with addendum modification	88.56mm	Calculated
Δh_a	Addendum reduction	0.26mm	Calculated
h_{a1}	Addendum of pinion	3.14mm	Calculated
h_{a2}	Addendum of gear	3.16mm	Calculated
d_{a1}	Addendum diameter of pinion	40.28mm	Calculated
d_{a2}	Addendum diameter of gear	144.32mm	Calculated
h_{f1}	Dedendum of pinion	1.1mm	Calculated
h_{f2}	Dedendum of gear	1.08mm	Calculated
d_{f1}	Dedendum diameter of pinion	31.8mm	Calculated
d_{f2}	Dedendum diameter of gear	135.84mm	Calculated
p_b	Base pitch	5.9mm	Calculated
ε_α	Contact ratio	1.24	Calculated
T_{tot}	Total output torque	6300N · mm	Given
η	Total efficiency	91.5%	From appendix 2
T_d	Design output torque	6885N · mm	Calculated
n_{tot}	Total output rotational speed	4.3rpm	Given
K_1	Load factor	1	Selected
n	Input rotational speed	1034.15rpm	Calculated
v	Pitch line velocity	1.841 m/s	Calculated

K_v	Dynamic factor	1.38	Calculated
T_1	Input torque for one planet gear	$9.54\text{N} \cdot \text{m}$	Calculated
F_{ber}	Calculation load	774N	Calculated
Y_F	Bending stress concentration factor	2.02	SMS 1871 diagram 8.5.2 (reference figure 3.1)
b	Contact face width of pinion	20mm	Selected
Y_β	Helix angle factor	1	Calculated
Y_ε	Contact ratio factor	0.81	Calculated
$K_{F\alpha}$	Load distribution	1	Calculated
$K_{F\beta}$	Load propagation factor	1.3	Selected
$K_{H\beta}$	Load propagation factor	1.3	Selected
σ_F	Bending stress	41.1MPa	Calculated
Y_s	Stress concentration factor	1	Selected
S_F	Bending safety factor	2	Selected
S_H	Surface fatigue safety factor	1.41	Calculated
K_{FX}	Reduction factor due to volume	1	Selected
K_{FN}	Life length factor	1	Selected
σ_{Flim}	Bending stress limit	280MPa	Figure 10.15d (reference figure 3.5)
σ_{FP}	Bending strength	140MPa	Calculated
β_b	Helix angle	0°	Given
Z_H	Stress concentration factor for surface fatigue in the rolling point	1.59	Calculated
E	Young's module	206000MPa	Table 2 (reference figure 3.6)
Z_M	Material factor	$268\sqrt{\text{N}/\text{mm}^2}$	Calculated
Z_ε	Meshing factor	0.96	Calculated
$K_{H\alpha}$	Load distribution factor	1	Selected
σ_H	Surface fatigue stress	555.5MPa	Calculated
K_{HX}	Reduction factor due to volume	1	Selected
K_{HN}	Life length factor due to surface pressure	1	Selected
K_{HK}	Hardness combination factor	1	Selected
$\sigma_{H\text{lim}}$	Surface fatigue stress limit	830MPa	Figure 10.14d (reference figure 3.7)
σ_{HP}	Surface fatigue strength	589MPa	Calculated

Given requirements and analysis

From appendix 2, $z_1 = 17$, $z_2 = 69$

$$b \leq 10 \cdot m \text{ (To design a combination drive)}$$

$$0.5 \leq x_1 \leq 1 \text{ (To design a combination drive)}$$

Calculation of first stage gears

Select modules:

$$m = 2\text{mm (Swedish Standard SMS52)}$$

Find addendum modification of gears

$$\text{Select } x_1 = 0.7 \text{ (} 0.5 \leq x_1 \leq 1 \text{)}$$

In SS1871 Diagram 8.5.2 (reference figure 3.2), assume fillet radius $r = 0.38 \cdot m$

$$\left. \begin{array}{l} z_1 = 17 \\ z_2 = 69 \\ x_1 = 0.7 \end{array} \right\} \rightarrow Y_{F1} = Y_{F2} = 2.02; x_2 = 0.71$$

Find pitch diameters of gears

$$\text{SS1863 (2.1) } d = m \cdot z$$

$$\begin{cases} d_1 = m \times z_1 = 2 \times 17 \\ d_2 = m \times z_2 = 2 \times 69 \end{cases}$$

$$\rightarrow \begin{cases} d_1 = 34\text{mm} \\ d_2 = 138\text{mm} \end{cases}$$

Find base diameters of gears

$$\text{SS1863 (2.2) } d_b = d \cdot \cos \alpha$$

$$\text{Select pressure angle } \alpha = 20^\circ$$

$$\begin{cases} d_{b1} = d_1 \times \cos 20^\circ = 34 \times \cos 20^\circ \\ d_{b2} = d_2 \times \cos 20^\circ = 138 \times \cos 20^\circ \end{cases}$$

$$\rightarrow \begin{cases} d_{b1} = 32\text{mm} \\ d_{b2} = 130\text{mm} \end{cases}$$

Find reference center distances of gears

$$\text{SS 1863 (3.8) } a = \frac{m \cdot (z_1 + z_2)}{2}$$

$$a = \frac{2 \times (34 + 138)}{2} = 86\text{mm}$$

Find center distance with addendum modification of gears

$$\text{SS1863 (3.11) } \text{inv}\alpha_w = \text{inv}\alpha + \frac{2 \cdot (x_1 + x_2)}{z_1 + z_2}$$

$$\alpha = 20^\circ \rightarrow \text{inv}\alpha = 0.0149044 \text{ (Involute table in page 22 of Gear Design)}$$

$$\rightarrow \text{inv}\alpha_w = 0.0149044 + \frac{2 \times (0.7 + 0.71)}{17 + 69} = 0.02684$$

$$\rightarrow \alpha_w = 24.15^\circ \text{ (Involute table in page 22 of Gear Design) (Reference figure 3.2)}$$

$$\text{SS1863 (3.10) } a_w = \frac{a \cdot \cos \alpha}{\cos \alpha_w}$$

$$a_w = \frac{86 \times \cos 20^\circ}{\cos 24.15^\circ} = 88.56 \text{ mm}$$

Find addendum reduction

$$\text{SS 1863 (3.3) } \Delta h_a = m \cdot \left(\frac{z_1 + z_2}{2} + x_1 + x_2 \right) - a_w$$

$$\Delta h_a = 2 \times \left(\frac{17 + 69}{2} + 0.7 + 0.71 \right) - 88.56 = 0.26 \text{ mm}$$

Find addendums of gears

$$\text{SS1863 (2.5) } h_a = m \cdot (1 + x) - \Delta h_a$$

$$\begin{cases} h_{a1} = 2 \times (1 + x_1) = 2 \times (1 + 0.7) - 0.26 \\ h_{a2} = 2 \times (1 + x_2) = 2 \times (1 + 0.71) - 0.26 \end{cases}$$

$$\rightarrow \begin{cases} h_{a1} = 3.14 \text{ mm} \\ h_{a2} = 3.16 \text{ mm} \end{cases}$$

Find addendum diameters of gears

$$\text{SS1863 (2.7) } d_a = d + 2 \cdot h_a$$

$$\begin{cases} d_{a1} = d_1 + 2 \times h_{a1} = 34 + 2 \times 3.14 \\ d_{a2} = d_2 + 2 \times h_{a2} = 138 + 2 \times 3.16 \end{cases}$$

$$\rightarrow \begin{cases} d_{a1} = 40.28\text{mm} \\ d_{a2} = 144.32\text{mm} \end{cases}$$

Find dedendums of gears

$$\text{SS1863 (2.6)} \quad h_f = m \cdot (1.25 - x)$$

$$\begin{cases} h_{f1} = 2 \times (1.25 - x_1) = 2 \times (1.25 - 0.7) \\ h_{f2} = 2 \times (1.25 - x_2) = 2 \times (1.25 - 0.71) \end{cases}$$

$$\rightarrow \begin{cases} h_{f1} = 1.1\text{mm} \\ h_{f2} = 1.08\text{mm} \end{cases}$$

Find dedendum diameters of gears

$$\text{SS 1863 (2.8)} \quad d_f = d - 2 \cdot h_f$$

$$\begin{cases} d_{f1} = d_1 - 2 \times h_{f1} = 34 - 2 \times 1.1 \\ d_{f2} = d_2 - 2 \times h_{f2} = 138 - 2 \times 1.08 \end{cases}$$

$$\rightarrow \begin{cases} d_{f1} = 31.8\text{mm} \\ d_{f2} = 135.84\text{mm} \end{cases}$$

Check the adjoining requirement

Since there are 3 planetary gears in the first stage with teeth number of $z_2 = 69$, their addendum diameter must meet the adjoining requirement. This means addendum circle of three planetary gears must not interact when rotating.

The requirement is equation (17-8) in page 433 of 《行星传动机构设计》 (reference figure 3.3):

$$d_{ag1} < 2 \cdot a'_1 \cdot \sin \frac{180^\circ}{n_p}$$

The addendum diameter of planetary gear that $d_{ag1} = d_{a2} = 144.32\text{mm}$; a'_1 is center distance $a'_1 = a = 86\text{mm}$; n_p is number of planetary gears $n_p = 3$.

$$2 \times a \times \sin \frac{180^\circ}{3} = 149\text{mm} > d_{a2} = 144.32\text{mm}$$

→The calculated parameters of gear can fulfill the adjoining requirement.

Find base pitch of gears

$$\text{SS 1863 (3.4)} \quad p_b = \pi \cdot m \cdot \cos \alpha$$

$$p_b = \pi \times 2 \times \cos 20^\circ = 5.9\text{mm}$$

Find contact ratio

$$\text{SS 1863 (3.5)} \quad \varepsilon_{\alpha} = \frac{1}{p_b} \cdot \left(\frac{\sqrt{d_{a1}^2 - d_{b1}^2}}{2} + \frac{\sqrt{d_{a2}^2 - d_{b2}^2}}{2} - a_w \cdot \sin \alpha_w \right)$$

$$\rightarrow \varepsilon_{\alpha} = \frac{1}{5.9} \times \left(\frac{\sqrt{40.28^2 - 332^2}}{2} + \frac{\sqrt{144.32^2 - 130^2}}{2} - 88.56 \times \sin 24.15^\circ \right)$$

$$\rightarrow \varepsilon_{\alpha} = 1.24$$

Calculation of gears' strength

This part includes bending and strength calculation.

Given total output torque $T_{\text{tot}} = 6300 \text{ N} \cdot \text{mm}$.

Given total output rotational speed $n_{\text{tot}} = 4.3 \text{ rpm}$

The design output torque:

$$T_d = \frac{T_{\text{tot}}}{\eta} = \frac{6300}{91.5\%}$$

$$\rightarrow T_d = 6885 \text{ N} \cdot \text{mm}$$

Find calculation load

$$\text{SS 1871 (3.1)} \quad F_{\text{ber}} = \frac{2 \cdot T_1}{d_1} \times K_1 \times K_v$$

$$\text{SS 1871 (3.2)} \quad K_1 = 1 \text{ (uniform motion)}$$

$$\text{SS 1871 (3.3) equation 4} \quad K_v = \frac{50 + 14 \cdot \sqrt{v}}{50} \text{ (uses for other spur gears)}$$

The input rotational speed:

$$n = n_{\text{tot}} \times i_{\text{tot}} = 4.3 \times 240.5$$

$$\rightarrow n = 1034.15 \text{ rpm}$$

The pitch line velocity:

$$v = \frac{d_1 \cdot \pi \cdot n}{60} = \frac{34 \times \pi \times 1034.15}{60}$$

$$\rightarrow v = 1.841 \text{ m/s}$$

$$\rightarrow K_v = \frac{50 + 14 \times \sqrt{1.841}}{50} = 1.38$$

The input torque:

$$T_1 = \frac{T_d}{i_{\text{tot}}} \times \frac{1}{n} = \frac{6885}{240.5} \times \frac{1}{3}$$

$$\rightarrow T_1 = 9.54 \text{ N} \cdot \text{m}$$

$$\rightarrow F_{\text{ber}} = \frac{2 \times 9.54 \times 10^3}{34} \times 1 \times 1.38 = 774 \text{ N}$$

Find bending stress

From front part the bending stress concentration factor:

$$Y_F = Y_{F1} = Y_{F2} = 2.02$$

Select the contact face width of pinion

$$b = 10 \cdot m = 10 \times 2$$

$$\rightarrow b = 20 \text{ mm}$$

SS 1871 (6.3) $Y_\beta = 1$ (for spur gear)

SS 1871 (6.4) $Y_\epsilon = \frac{1}{\epsilon_\alpha}$

$$Y_\epsilon = \frac{1}{1.24} = 0.81$$

SS 1871 (6.5) $K_{F\alpha} = 1$ (normally)

SS 1871 (6.7) $K_{F\beta} = K_{H\beta}$ (normally)

$K_{F\beta} = K_{H\beta} = 1.3$ (No information)

SS 1871 (6.1) $\sigma_F = Y_F \times Y_\beta \times Y_\epsilon \times \frac{F_{\text{ber}} \times K_{F\alpha} \times K_{F\beta}}{b \times m}$

$$\rightarrow \sigma_F = 2.02 \times 1 \times 0.81 \times \frac{774 \times 1 \times 1.3}{20 \times 2} = 41.1 \text{ MPa}$$

Find bending strength

SS 1871 (7.2) $Y_s = 1$ ($r = 0.38 \cdot m$)

SS 1871 (7.3) $S_F = S_H^2$

Select $S_F = 2$

$$\rightarrow S_H = \sqrt{S_F} = \sqrt{2}$$

$$\rightarrow S_H = 1.41$$

SS 1871 (7.4) $K_{FX} = 1$ (normally)

SS 1871 (7.5) $K_{FN} = 1$ (assume the required life length is more than 10^7 cycles)

Since the material of gears is chosen as 20MnCr6-5. According to **ThyssenKrupp Steel**

Material Specifications (in reference figure 3.5) its hardness is more than 270HBW when the plate thickness $b \leq 20 \text{ mm}$.

$$\left\{ \begin{array}{l} \text{Choose hardness} = 350 \text{ HBW} \\ \text{Alloy steel} \\ \text{Figure 10.15d in page 197 of Machine Design (reference figure 3.5)} \end{array} \right.$$

$$\rightarrow \sigma_{Flim} = 280 \text{ MPa}$$

$$\text{SS 1871 (7.1) } \sigma_{FP} = \frac{\sigma_{Flim} \times Y_s \times K_{FX} \times K_{FN}}{S_F}$$

$$\sigma_{FP} = \frac{280 \times 1 \times 1 \times 1}{2} = 140 \text{ MPa}$$

$$\rightarrow \sigma_{FP} = 140 \text{ MPa} > \sigma_F = 41.1 \text{ MPa}$$

This analysis above means the gear design can fulfill the bending requirement.

Find surface fatigue stress

$$\text{SS 1871 (4.2) } Z_H = \sqrt{\frac{\cos \beta_b \times \cos \alpha_{\omega t}}{\cos \alpha_t^2 \times \sin \alpha_{\omega t}}}$$

$\beta_b = 0^\circ$ (Spur gear)

$$\alpha_t = \alpha = 20^\circ$$

$$\alpha_{\omega t} = \alpha = 24.15^\circ$$

$$\rightarrow Z_H = \sqrt{\frac{\cos 0^\circ \times \cos 24.15^\circ}{\cos 20^\circ \times \sin 24.15^\circ}} = 1.59$$

SS 1871 Table 2 (reference figure 3.7) $E = 206000 \text{ MPa}$ (steel)

SS 1871 (4.3) $Z_M = \sqrt{0.35 \cdot E}$ (same material in both pinion and gear)

$$\rightarrow Z_M = \sqrt{0.35 \times 206000} = 268 \sqrt{\text{N/mm}^2}$$

SS 1871 (4.4) $Z_\epsilon = \sqrt{\frac{4-\epsilon_\alpha}{3}}$ (for spur gear)

$$\rightarrow Z_\epsilon = \sqrt{\frac{4-1.24}{3}} = 0.96$$

SS 1871 (4.5) $K_{H\alpha} = 1$ (normally)

SS 1871 (4.6) $K_{H\beta} = 1.3$ (No information)

SS 1871 (4.1) $\sigma_H = Z_H \times Z_M \times Z_\epsilon \times \sqrt{\frac{F_{ber} \times K_{H\alpha} \times K_{H\beta} \times (u+1)}{b \times d_1 \times u}}$

$$u = i = 4.06$$

$$\rightarrow \sigma_H = 1.59 \times 268 \times 0.96 \times \sqrt{\frac{774 \times 1 \times 1.3 \times (4.06 + 1)}{20 \times 34 \times 4.06}} = 555.5 \text{ MPa}$$

Find surface fatigue strength

SS 1871 (5.1) $K_L = Z_R = Z_v = 1$ (lack of experience)

SS 1871 (5.2) $S_H = \sqrt{S_F} = 1.41$

SS 1871 (5.3) $K_{HX} = 1$ (normally)

SS 1871 (5.4) $K_{HN} = 1$ (assume the fatigue life length is more than 10^9 cycles)

SS 1871 (5.5) $K_{HK} = 1$ (normally)

$$\left\{ \begin{array}{l} \text{Hardness} = 350\text{HBW} \\ \text{Alloy steel} \\ \text{Figure 10.14d in page 196 of Machine Design (reference figure 3.7)} \end{array} \right.$$

$$\rightarrow \sigma_{\text{Hlim}} = 830\text{MPa}$$

$$\text{SS 1871 (5.1) } \sigma_{\text{HP}} = \frac{\sigma_{\text{Hlim}} \times K_L \times Z_R \times Z_V \times K_{\text{HX}} \times K_{\text{HN}} \times K_{\text{HK}}}{S_H}$$

$$\rightarrow \sigma_{\text{HP}} = \frac{830 \times 1 \times 1 \times 1 \times 1 \times 1 \times 1}{1.41} = 589\text{MPa}$$

$$\rightarrow \sigma_{\text{HP}} = 589\text{MPa} > \sigma_{\text{H}} = 555.5\text{MPa}$$

This analysis above means the gear design can fulfill the surface fatigue requirement.

Reference

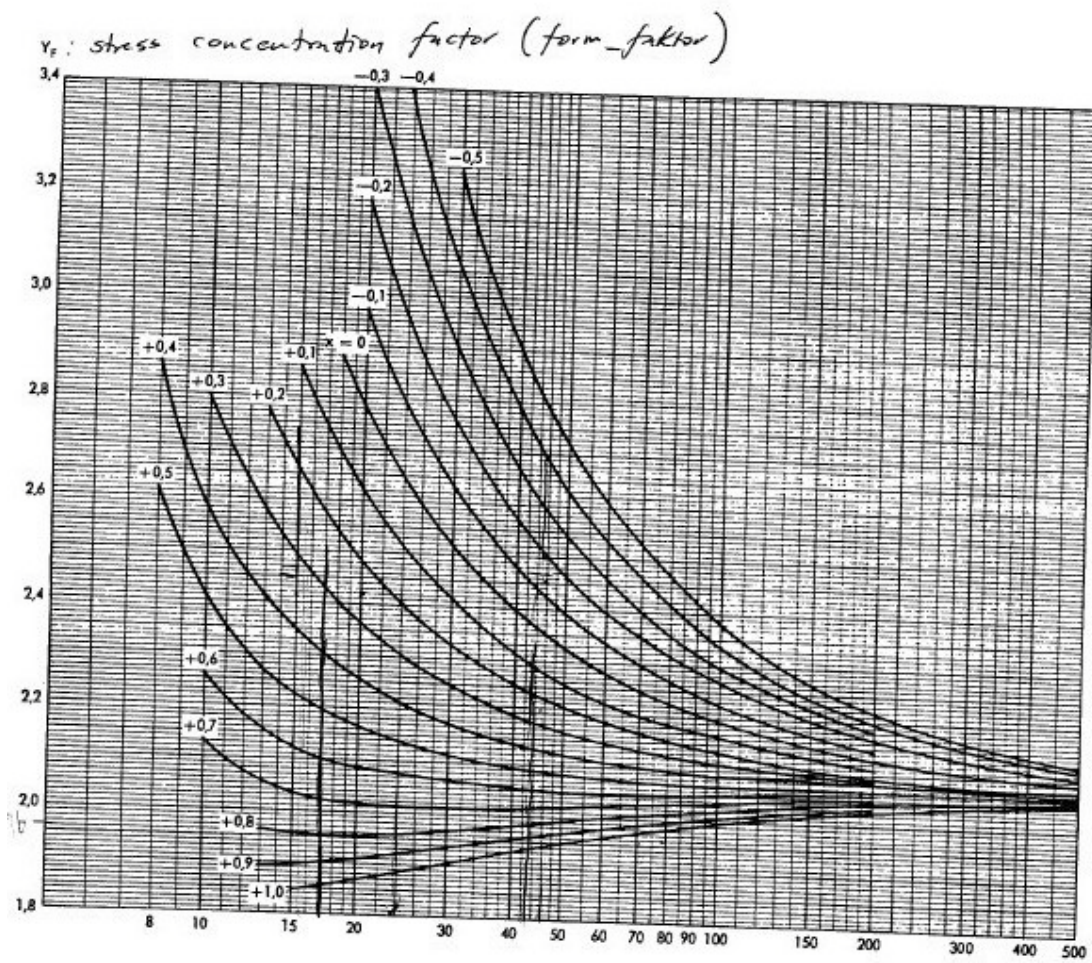


Figure 3.1 SS1871 Diagram 8.5.2

Tabellen visar $\tan(\alpha) = \tan(\alpha) \cdot \alpha = \text{tabellvärdet} \cdot 10^{-6}$
ten power to minus 6

α	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
15	6149,80	6275,99	6403,94	6533,67	6665,19	6798,51	6933,65	7070,63	7209,46	7350,14
16	7492,71	7637,16	7783,52	7931,80	8082,01	8234,17	8388,29	8544,39	8702,49	8862,59
17	9024,71	9188,87	9355,08	9523,36	9693,71	9866,17	10040,7	10217,4	10396,3	10577,3
18	10760,4	10945,8	11133,3	11323,1	11515,1	11709,4	11905,9	12104,8	12305,9	12509,3
19	12715,1	12923,2	13133,6	13346,5	13561,7	13779,4	13999,4	14222,0	14447,0	14674,4
20	14904,4	15136,9	15371,9	15609,4	15849,5	16092,2	16337,5	16585,4	16835,9	17089,1
21	17344,9	17603,4	17864,6	18128,6	18395,3	18664,7	18936,9	19211,9	19489,7	19770,3
22	20053,8	20340,1	20629,3	20921,5	21216,5	21514,5	21815,4	22119,3	22426,2	22736,1
23	23049,1	23365,1	23684,2	24006,3	24331,6	24660,0	24991,6	25326,3	25664,2	26005,3
24	26349,7	26697,3	27048,1	27402,3	27759,8	28120,6	28484,7	28852,3	29223,2	29597,6
25	29975,3	30356,6	30741,3	31129,5	31521,3	31916,6	32315,4	32717,9	33123,9	33533,6
26	33947,0	34364,0	34784,7	35209,2	35637,4	36069,4	36505,1	36944,7	37388,1	37835,4
27	38286,6	38741,6	39200,6	39663,6	40130,6	40601,5	41076,5	41555,5	42038,7	42525,9
28	43017,2	43512,8	44012,4	44516,3	45024,5	45536,9	46053,5	46574,5	47099,8	47629,5
29	48163,6	48702,0	49245,0	49792,4	50344,2	50900,6	51461,6	52027,1	52597,3	53172,1
30	53751,5	54335,6	54924,5	55518,1	56116,4	56719,6	57327,6	57940,5	58558,2	59180,9
31	59808,6	60441,2	61078,8	61721,5	62369,2	63022,1	63680,1	64343,2	65011,6	65685,1
32	66364,0	67048,1	67737,6	68432,4	69132,6	69838,3	70549,3	71265,9	71988,0	72715,7
33	73448,9	74187,8	74932,4	75682,6	76438,5	77200,3	77967,8	78741,1	79520,4	80305,5
34	81096,6	81893,6	82696,7	83505,8	84321,0	85142,4	85969,9	86803,6	87643,5	88489,8
35	89342,3	90201,2	91066,5	91938,2	92816,5	93701,2	94592,5	95490,4	96394,9	97306,1
36	98224,0	99148,7	100080	101019	101964	102916	103875	104841	105814	106795
37	107782	108777	109779	110788	111805	112829	113860	114899	115945	116999
38	118061	119130	120207	121291	122384	123484	124592	125709	126833	127965
39	129106	130254	131411	132576	133750	134931	136122	137320	138528	139743
40	140968	142201	143443	144694	145954	147222	148500	149787	151083	152388
41	153702	155025	156358	157700	159052	160414	161785	163165	164556	165956
42	167366	168786	170216	171656	173106	174566	176037	177518	179009	180511
43	182024	183547	185080	186625	188180	189746	191324	192912	194511	196122
44	197744	199377	201022	202678	204346	206026	207717	209420	211135	212863
45	214602	216353	218117	219893	221682	223483	225296	227123	228962	230814
46	232679	234557	236448	238353	240271	242202	244147	246105	248078	250064
47	252064	254078	256106	258149	260206	262277	264363	266464	268579	270709
48	272854	275015	277190	279381	281588	283810	286047	288301	290570	292856
49	295157	297475	299809	302160	304527	306912	309313	311731	314166	316619
50	319089	321577	324082	326605	329146	331706	334283	336879	339494	342127
51	344779	347450	350141	352850	355579	358328	361096	363884	366693	369521
52	372370	375240	378130	381042	383974	386928	389903	392899	395918	398958
53	402020	405105	408212	411342	414495	417671	420871	424093	427340	430610
54	433904	437223	440566	443933	447326	450744	454187	457656	461150	464670
55	468217	471790	475390	479016	482670	486351	490060	493797	497562	501355
56	505177	509027	512907	516816	520755	524724	528723	532753	536813	540905
57	545027	549182	553368	557586	561837	566121	570438	574789	579173	583591
58	588044	592531	597053	601611	606205	610834	615500	620203	624943	629720

Figure 3.2 Involute table

$$d_{a_{g_1}} < 2a_1' \sin \frac{180^\circ}{n_p} \quad (17-8)$$

Figure 3.3 Equation (17-8) in page 433 of 《行星传动机构设计》

ThyssenKrupp Steel

Material Specifications



Wear-resistant special structural steel	Steel grade		Material No.	Material Specification
	TKS-Short name	EN-Short name		
Heavy plate	XAR [®] 300	20MnCr6-5	1.8704	760 October 2006

Scope

This Material Specification applies to normalised or normalised rolled plates in thicknesses up to 50 mm made of the wear-resistant special structural steel XAR[®] 300.

Application

The steel may be used at the discretion of the purchaser for wear-exposed structures, e.g. excavating, mining and earth-moving machinery, truck dump bodies, conveying, crushing and pulverising equipment, scrap presses and other machinery.

The processing and application techniques as a whole are of fundamental importance for the successful use of the products fabricated of this steel. The processor/fabricator must assure himself that his design and work methods are appropriate for the material, are state-of-the-art and are suitable for the envisaged purpose.

The selection of the material is left up to the purchaser.

Chemical composition (heat analysis, %)

C	Si	Mn	P	S	Cr	Mo	B
≤ 0.21	≤ 0.65	≤ 1.50	≤ 0.025	≤ 0.025	≤ 1.20	≤ 0.30	≤ 0.005

The steel has a fine-grained microstructure. Nitrogen is absorbed to form nitrides.

Delivery condition: N (normalised)

Hardness at room temperature in the delivery condition:

Plate thickness ≤ 20 mm: Hardness ≥ 270 HBW

Plate thickness > 20 mm: Hardness ≥ 240 HBW

The Brinell hardness shall be determined in accordance with ISO 6506.

Figure 3.4 20MnCr6-5 in ThyssenKrupp Steel Material Specifications

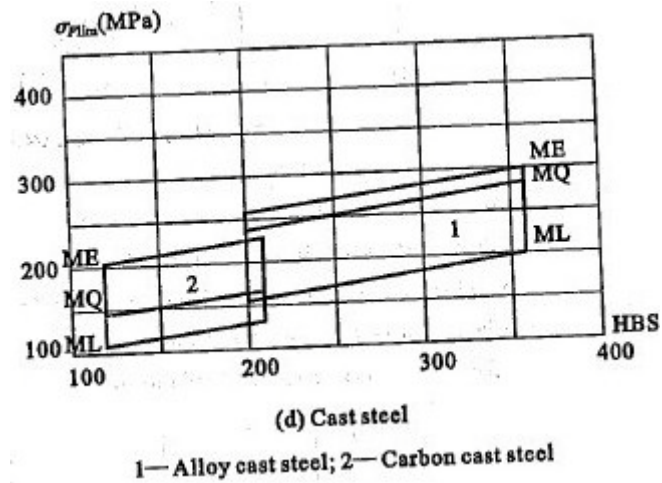


Figure 3.5 Endurance limit for bending strength

Table 2 Material factor
(Z_M)

Gear					Z_M	
Material		Material			$\sqrt{N/mm^2}$	$\sqrt{kp/mm^2}$
Steel	206 000 (21 000)	Steel	206 000	(21 000)	268	85,7
		Cast steel	201 000	(20 500)	267	85,2
		Nodular iron	173 000	(17 600)	257	81,9
			172 000	(17 500)	256	81,7
		Grey cast iron	103 000	(10 500)	219	70
			113 000	(11 500)	226	72,1
			126 000	(12 800)	234	74,6
Cast steel	201 000 (20 500)		118 000	(12 000)	229	73,1
		Cast steel	201 000	(20 500)	265	84,7
		Nodular iron	173 000	(17 600)	255	81,4
Nodular iron	173 000 (17 600)	Grey cast iron	118 000	(12 000)	228	72,8
		Nodular iron	172 000	(17 500)	246	78,4
Grey cast iron	126 000 (12 800) 118 000 (12 000)	Grey cast iron	118 000	(12 000)	221	70,7
		Grey cast iron	118 000	(12 000)	206	65,8
		Grey cast iron	118 000	(12 000)	203	64,8

Figure 3.6 SS 1871 Table 2

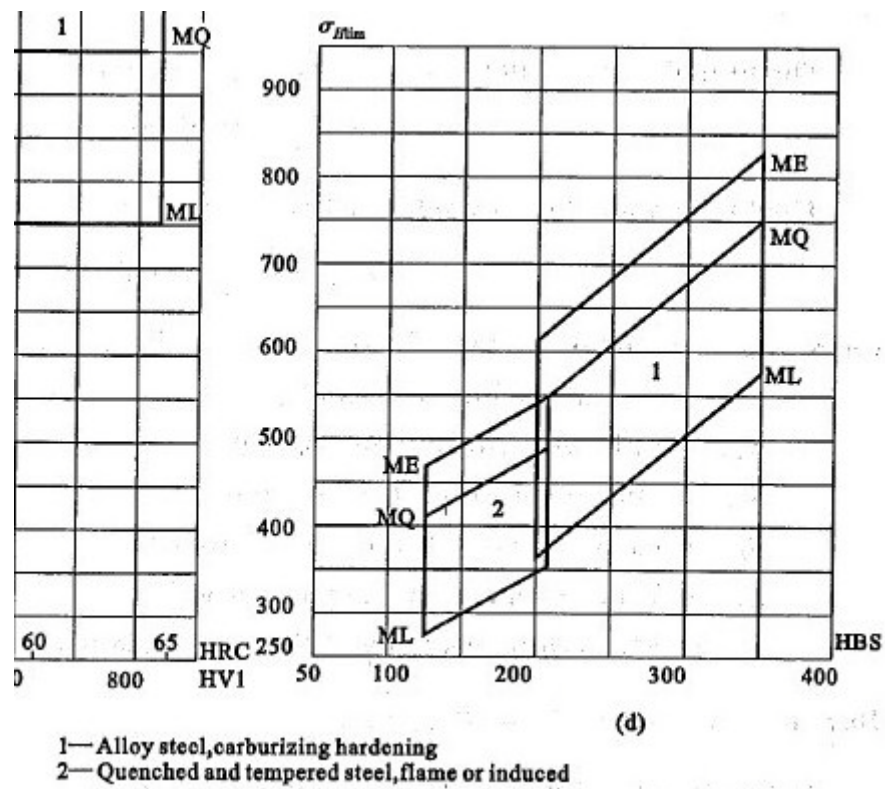


Figure 3.7 Endurance limits for contact strength

Appendix 4 Choose the eccentric bearing

Symbol	Explanation	Value	Other
P	Force from the pins to the cycloid gear	33686N	Calculated
P_x	Force from the pins to the cycloid gear in X direction	31034N	Calculated
P_{Ymax}	Maximum force from the pins to the cycloid gear in Y direction	13100N	Calculated
K_1	short width coefficient	0.662	Calculated
e	Eccentric distance	1.75mm	Selected
Z_4	Pin number	59	Selected
Rz	Pin wheel radius	147.5mm	Selected
M_v	Output torque	6300Nm	Given
Z_3	Cycloid gear teeth number	58	Selected
Z_1	Sun gear teeth number	17	Selected
Z_2	Planet gear teeth number	69	Selected
m	Module of gear	2	Selected
F_x	The force on the bearing in X direction	10355N	Calculated
F_y	The force on the bearing in Y direction	4360N	Calculated
R	Total reaction force on bearing	11235N	Calculated
n	Relatively rotation speed	253.7rpm	Calculated
C	Selected dynamic load	61300	Selected
C_1	Actual dynamic load	49452N	Calculated
L_h	Assumed service life	5000h	Selected
L_{h1}	Actual service life	10230h	Selected
d	Inner diameter of bearing	22mm	Selected
D	Outer diameter of bearing	53.5mm	Selected

Force from the pins to the cycloid gear:

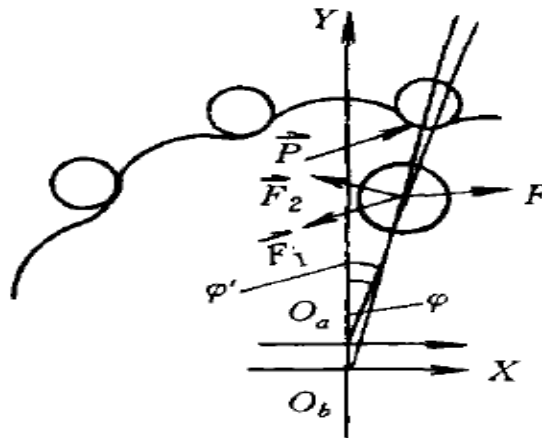


Figure4.1 force analysis on the cycloid gear(RV 传动机构中转臂轴承的动载荷分析)

$$P = \sqrt{(P_x^2 + P_y^2)}$$

Calculate short width coefficient:

$$K_1 = \frac{eZ_4}{R_z} = \frac{1.75 * 59}{155.95} = 0.662$$

The force on cycloid gear in X-direction:

$$P_x = \frac{Z_4 M_v}{2K_1 Z_3 R_z} = \frac{59 * 6300000}{2 * 0.662 * 58 * 155.95}$$

$$\rightarrow P_x = 31034N$$

$$P_y = \sum_i^{Z_4/2} \frac{2M_v [\cos(\varphi - \frac{2\pi i}{Z_4}) - K_1] \sin(\varphi - \frac{2\pi i}{Z_4})}{K_1 Z_3 R_z [1 + K_1^2 - 2K_1 \cos(\varphi - \frac{2\pi i}{Z_4})]}$$

This equation is so complex that it is suggested to use Matlab to solve it, the force is shown as follows:

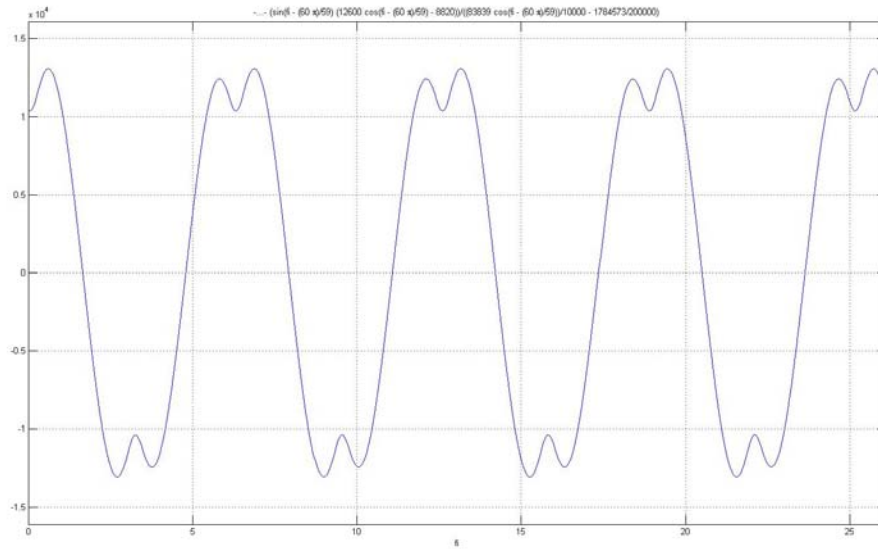


Figure 4.2 variable force on Y direction shown in Matlab

So the max force is $P_{Ymax} = 13100N$ when $\varphi = 0.61 + 2n\pi$ ($n = 0, 1, 2, \dots$)

When $n=0$, $\varphi = 0.61$

$$\varphi' = \frac{Z_4}{Z_3} \varphi = \frac{59}{58} 0.61$$

$$\varphi' = 0.62 \text{rad} = 35.5^\circ$$

Because the equilibrium of force, the force on the bearing is $F = \sqrt{F_x^2 + F_y^2}$

$$F_x = \frac{1}{n} \left(P_x + \frac{Mv}{m(Z_1 + Z_2)} \cos \varphi' \right) = \frac{1}{3} \left(31034 + \frac{6300}{2 * (17 + 69)} * \cos 35.5^\circ \right)$$

$$\rightarrow F_x = 10355 \text{N}$$

$$F_y = \frac{1}{n} \left(P_y - \frac{Mv}{m(Z_1 + Z_2)} \sin \varphi' \right) = \frac{1}{3} \left(13100 - \frac{6300}{2 * (17 + 69)} * \sin 35.5^\circ \right)$$

$$\rightarrow F_y = 4360 \text{N}$$

So the total reaction force from the picture above is $R = \sqrt{F_x^2 + F_y^2} = \sqrt{10355^2 + 4360^2}$

$$\rightarrow R = 11235 \text{N}$$

$$P = \sqrt{P_x^2 + P_y^2} = \sqrt{31034^2 + 13100^2}$$

$$\rightarrow P = 33686 \text{N}$$

Now we need to choose the bearing from the standard.

Assume the bearing service life is $L_h = 5000 \text{h}$

$$F = 1.2R = 1.2 * 11235$$

$$\rightarrow F = 13482 \text{N}$$

$$n = |n_H| + |n_v| = 249.4 + 4.3$$

$$\rightarrow n = 253.7 \text{rpm}$$

$$C_1 = F \sqrt[10/3]{\frac{60nL_h}{10^6}} = 13482 * \sqrt[10/3]{\frac{60 * 253.7 * 5000}{10^6}}$$

$$\rightarrow C_1 = 49452N$$

Because $e=1.75\text{mm}$ and $C=49452N$, so we choose the eccentric bearing is 180752904 which $d=22\text{mm}$, $D=53.5\text{mm}$, $C=61300N$

$$\text{So actual service live is } L_{h1} = \frac{10^6}{60n} \left(\frac{C}{F} \right)^{\frac{10}{3}} = \frac{10^6}{60 \times 253.7} \left(\frac{61300}{13482} \right)^{\frac{10}{3}}$$

$$\rightarrow L_{h1} = 10230h$$

Reference:

主要尺寸 (mm)					轴承 型号	额定负荷 (N)		极限转速 (rpm)		键槽 b t ₁		重量 (kg) (参考)
d	D	L	r	e		静动		脂润滑	油润滑			
19	70	36	1.1	1.5	150752904k1	7800	7800	7000	9500	6	2.8	0.82
	70	36	1.1	2.0	200752904k1	7800	7800	7000	9500	6	2.8	0.82
	70	36	1.1	3.0	300752904k1	7800	7800	7000	9500	6	2.8	0.82
22	53.5	32	1.1	0.65	70752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	0.75	80752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	1.0	100752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	1.25	130752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	1.5	150752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	1.75	180752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	2.0	200752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	2.5	250752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	3.0	300752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	3.5	350752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	4.0	400752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	4.25	430752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	4.5	450752904	50900	61300	8500	11000	6	2.8	0.35
	53.5	32	1.1	5.0	500752904	50900	61300	8500	11000	6	2.8	0.35
	61.8	34	1.1	0.75	80752904k	71000	83000	7500	9500	6	2.8	0.5
	61.8	34	1.1	1.0	100752904k	71000	83000	7500	9500	6	2.8	0.5
	61.8	34	1.1	1.25	130752904k	71000	83000	7500	9500	6	2.8	0.5
	61.8	34	1.1	1.5	150752904k	71000	83000	7500	9500	6	2.8	0.5
	61.8	34	1.1	2.0	200752904k	71000	83000	7500	9500	6	2.8	0.5

Figure 4.3 Chinese standard of eccentric bearing
(<http://www.jiansuji001.com/2007-8/200782191328.htm>)

Appendix 5 Calculation of first stage output shaft

Symbol	Explanation	Value	Other
F_Y	Force from eccentric bearing in vertical plane	4360N	From Appendix 4
F_X	Force from eccentric bearing in horizontal plane	10355N	From Appendix 4
T_1	Input torque	9.54N · m	From Appendix 3
d_1	Pinion pitch diameter	34mm	From Appendix 3
F_{t1}	Tangent force on pinion	561.2N	Calculated
α	Pressure angle	20°	From Appendix 3
F_{r1}	Radial force on pinion	204.3N	Calculated
η	Efficiency of first stage gears	98%	From Appendix 2
b	Gear width	20mm	From Appendix 3
L_1	Length of part 1 of shaft	18mm	Selected
L_2	Length of part 2 of shaft	16mm	Selected
L_3	Length of part 3 of shaft	16mm	Selected
L_4	Length of part 4 of shaft	18mm	Selected
L_5	Length of part 5 of shaft	24mm	Selected
L	Total length of shaft	92mm	Selected
R_{BV}	Reaction force on bearing B in vertical view	911.1N	Calculated
R_{AV}	Reaction force on bearing A in vertical view	-1111.1N	Calculated
S_V	Maximum shear force in vertical view	3248.9N	Calculated
M_V	Maximum bending moment in vertical view	27816N · m	Calculated
R_{BH}	Reaction force on bearing B in horizontal view	2065.8N	Calculated
R_{AH}	Reaction force on bearing A in horizontal view	-2615.8N	Calculated
S_H	Maximum shear force in horizontal view	7739.2N	Calculated
M_H	Maximum bending moment in horizontal view	66934N · m	Calculated
d_2	Gear pitch diameter	138mm	From Appendix 3
T	Torque on shaft	37950N · mm	Calculated
A_a	Axial load on bearing A	0N	Calculated
A_r	Radial load on bearing A	2842N	Calculated
B_a	Axial load on bearing B	0N	Calculated
B_r	Radial load on bearing B	2258N	Calculated
d_3	Shaft diameter (mounted with gear)	18mm	Selected

r_1	Shaft fillet radius	2mm	Selected
r_2	Fillet radius on the keyway	0.2mm	Selected
K_{t1}	Static stress concentration factor on keyway (for torsion, mounted with eccentric bearing)	3.7	Diagram 3 (reference figure 5.22)
S_u	Ultimate strength	923MPa (134ksi)	Selected
S_y	Yield strength	750MPa	Selected
q_1	Notch sensitive factor of shaft part (for torsion, mounted with eccentric bearing)	0.92	Figure 8.24 (reference figure 5.23)
K_{f1}	Fatigue stress concentration factor of shaft part (for torsion, mounted with eccentric bearing)	3.484	Calculated
τ_a	Torsional alternating stress	0MPa	Calculated
τ_m	Torsional mean stress	181MPa	Calculated
$\sigma_{a,a}$	Axial alternating stress	0MPa	Calculated
$\sigma_{a,m}$	Axial mean stress	0MPa	Calculated
M	Maximum bending moment	72484N · mm	Calculated
D	Shaft diameter (mounted with eccentric bearing)	22mm	Selected
d_4	Shaft diameter (mounted with tapered roller bearing)	20mm	Selected
q_2	Notch sensitive factor of shaft part (for bending, mounted with eccentric bearing)	0.87	Figure 8.24 (reference figure 5.23)
B	Key width	6mm	Table 1 (reference figure 5.21)
H	Key thickness	6mm	Table 1 (reference figure 5.21)
K_{t2}	Static stress concentration factor on shaft (for bending, mounted with eccentric bearing)	1.57	Figure 4.35(reference figure 5.24)
K_{f2}	Fatigue stress concentration factor of shaft part (for bending, mounted with eccentric bearing)	1,5	Calculated
$\sigma_{b,m}$	Bending mean stress	0MPa	Calculated
$\sigma_{b,a}$	Bending alternating stress	300Mpa	Calculated
σ_{ea}	Equivalent alternating bending stress	300Mpa	Calculated
σ_{em}	Equivalent mean bending stress	181Mpa	Calculated
C_T	Temperature factor	1	Calculated
C_R	Reliability factor	1	Calculated
S'_n	R.R.Moore endurance limit	461.5Mpa	Calculated
C_L	Load factor	1	Calculated

C_G	Gradient factor	0.9	Calculated
C_S	Surface limit	0.9	Calculated
S_n	Fatigue endurance limit	374Mpa	Calculated
SF_1	Safety factor from section E-E	2.88	Calculated
K_{t3}	Static stress concentration factor on keyway (for torsion, mounted with gear)	3.7	Diagram 3 (reference figure 5.22)
K_{f3}	Fatigue stress concentration factor of shaft part (for torsion, mounted with gear)	3.484	Calculated
q_3	Notch sensitive factor of shaft part (for torsion, mounted with gear)	0.92	Figure 8.24 (reference figure 5.23)
S_{ys}	Shear yield strength	435Mpa	Calculated
SF_2	Safety factor from section F-F	3.8	Calculated
SF	Final safety factor for whole shaft	2.88	Calculated

Find forces acted on shaft

Forces acted on this shaft are forces from two eccentric bearings and forces from the gear of first stage.

Find forces from eccentric bearings

Force in the vertical plane $F_Y = 4360\text{N}$ (from Appendix 4).

Force in the horizontal plane $F_X = 10355\text{N}$ (from Appendix 4).

Find forces on pinion

Since the pinion is spur gear there are only radial force and tangent force act on it.

The input torque is:

$$T_1 = 9.54\text{N} \cdot \text{m}.$$

Pinion pitch diameter:

$$d_1 = 34\text{mm}$$

Tangent force on pinion:

$$F_{t1} = \frac{T_1}{d_1/2} = \frac{9.54 \times 10^3}{17}$$

$$\rightarrow F_{t1} = 561.2\text{N}$$

Pressure angle is $\alpha = 20^\circ$ (from Appendix 1).

Radial force on pinion

$$F_{r1} = F_{t1} \times \tan \alpha = 561.2 \times \tan 20^\circ$$

$$\rightarrow F_{r1} = 204.3\text{N}$$

Find forces on gear

The efficiency of first stage gears is $\eta = 98\%$ (from Appendix 2 η is equal to $\eta_{a_1g_1}^{H_1}$).

Tangent force on gear:

$$F_{t2} = F_{t1} \times \eta = 561.2 \times 98\%$$

$$\rightarrow F_{t2} = 550\text{N}$$

Radial force on gear:

$$F_{r2} = F_{r1} \times \eta = 204.3 \times 98\%$$

$$\rightarrow F_{r2} = 200\text{N}$$

Geometry of the shaft

Figure 5.1 shows the geometry of the first stage output shaft. It is separated into 5 parts.

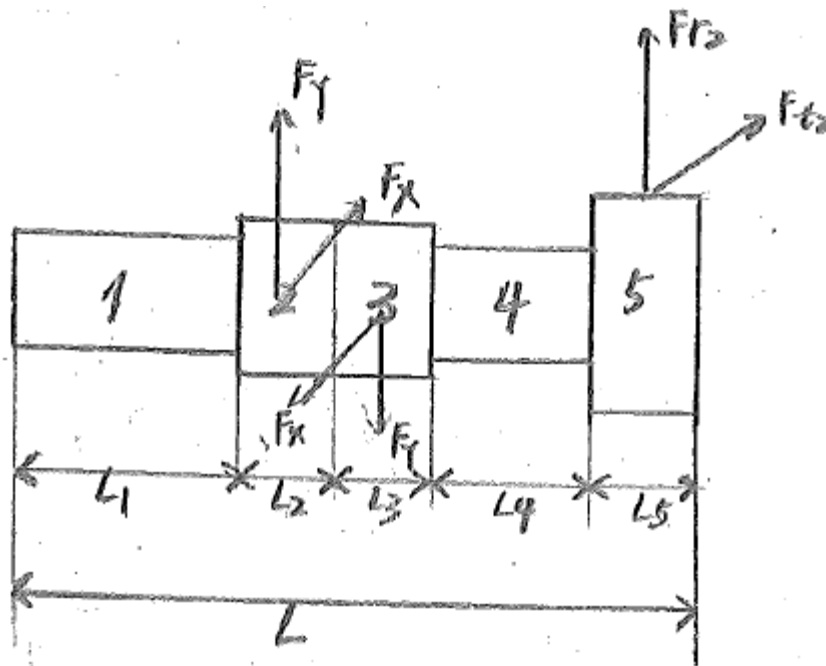


Figure 5.1 Geometry of the first stage output shaft

Part 1 is tapered roller bearing $L_1 = 18\text{mm}$. (According to the selected bearing type in Appendix 6)

Part 2 is eccentric bearing $L_2 = 16\text{mm}$. (According to the selected bearing type in Appendix 4)

Part 3 is eccentric bearing $L_3 = 16\text{mm}$. (According to the selected bearing type in Appendix 4)

Part 4 is tapered roller bearing $L_4 = 18\text{mm}$. (According to the selected bearing type in Appendix 6)

Part 5 is gear $L_5 = 24\text{mm}$. (According to the gear width $b = 20\text{mm}$ in Appendix 3)

The total length of shaft is $L = 92\text{mm}$.

Calculation of forces and bending moments on the shaft

The calculation can be divided into vertical plane and horizontal plane.

Vertical view (from side)

From the analysis of forces acted on the shaft and selected bearings and gear, figure 5.2 shows the vertical view of the shaft with forces on it.

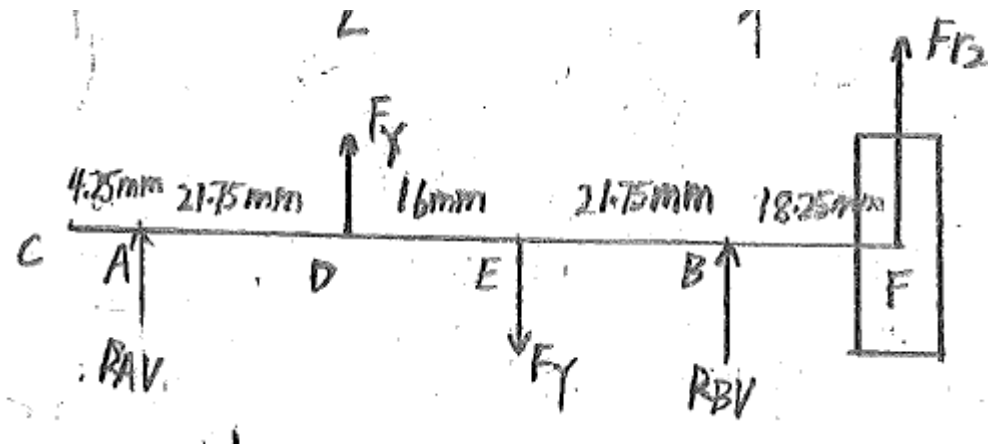


Figure 5.2 Vertical view of shaft with forces

In figure 5.2:

$$CA = 4.25\text{mm}$$

$$AD = 21.75\text{mm}$$

$$DE = 16\text{mm}$$

$$EB = 21.75\text{mm}$$

$$BF = 18.25\text{mm}$$

Find reaction forces

$$\uparrow: R_{AV} + F_Y + (-F_Y) + R_{BV} + F_{r2} = 0$$

$$\rightarrow R_{AV} + R_{BV} = -200 \quad (1)$$

$$A: F_Y \times 21.75 - F_Y \times (21.75 + 16) + R_{BV} \times (21.75 + 16 + 21.75) + F_{r2} \times (21.75 + 16 + 21.75 + 18.25) = 0$$

$$\rightarrow -69760 + 59.5 \times R_{BV} + 15550 = 0$$

$$\rightarrow R_{BV} = 911.1\text{N}$$

Plug into equation(1):

$$\rightarrow R_{AV} + 911.1 = -200$$

$$\rightarrow R_{AV} = -1111.1\text{N}$$

Calculation of shear forces and bending moments

Section AD: $0 \leq x \leq 21.75\text{mm}$ (see figure 5.3).



Figure 5.3 Section AD

$$\downarrow: S_{(x)} - R_{AV} = 0$$

$$\rightarrow S_{(x)} = -1111.1\text{N}$$

$$\text{AD: } M_{(x)} - R_{AV} \cdot x = 0$$

$$\rightarrow M_{(x)} = -1111.1 \cdot x$$

In this section when $x = 21.75\text{mm}$ (at point D) has the maximum moment:

$$M_D = -1111.1 \times 21.75 = -24166.43\text{N} \cdot \text{mm}$$

Section DE: $21.75 \leq x \leq 37.75\text{mm}$ (see figure 5.4).

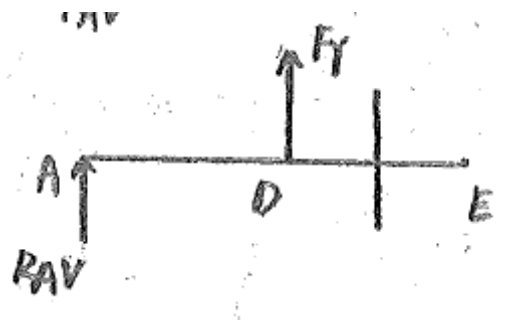


Figure 5.4 Section DE

$$\downarrow: S_{(x)} - F_Y - R_{AV} = 0$$

$$\rightarrow S_{(x)} = 4360 + (-1111.1) = 3248.9\text{N}$$

$$DE: M_{(x)} - R_{AV} \cdot x - F_Y \cdot (x - 21.75) = 0$$

$$M_{(x)} + 1111.1 \cdot x - 4360 \cdot x + 4360 \times 21.75 = 0$$

$$\rightarrow M_{(x)} = 3248.9 \cdot x - 94830$$

In this section when $x = 37.75\text{mm}$ (at point E) has the maximum moment:

$$M_E = 3248.9 \times 37.75 - 94830 = 27816\text{N} \cdot \text{mm}$$

Section EB: $37.75 \leq x \leq 59.5\text{mm}$ (see figure 5.5).

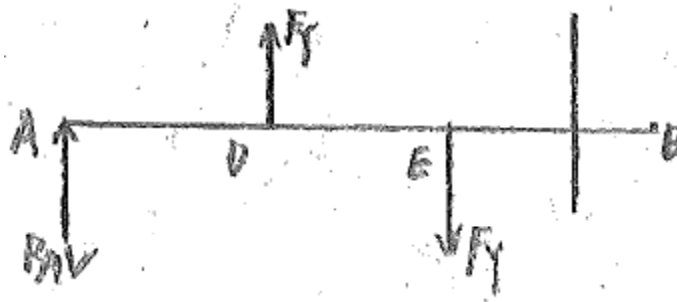


Figure 5.5 Section EB

$$\downarrow: S_{(x)} - R_{AV} - F_Y + F_Y = 0$$

$$\rightarrow S_{(x)} = -1111.1\text{N}$$

$$EB: M_{(x)} - R_{AV} \cdot x - F_Y \cdot (x - 21.75) + F_Y \cdot (x - 37.75) = 0$$

$$M_{(x)} + 1111.1 \cdot x + 4360 \times 21.75 - 4360 \times 37.75 = 0$$

$$\rightarrow M_{(x)} = -1111.1 \cdot x + 69760$$

In this section when $x = 35\text{mm}$ (at point E) has the maximum moment.

$$M_E = -1111.1 \times 37.75 + 69760 = 27816\text{N} \cdot \text{mm}$$

Section BF: $59.5 \leq x \leq 77.75\text{mm}$ (see figure 5.6).

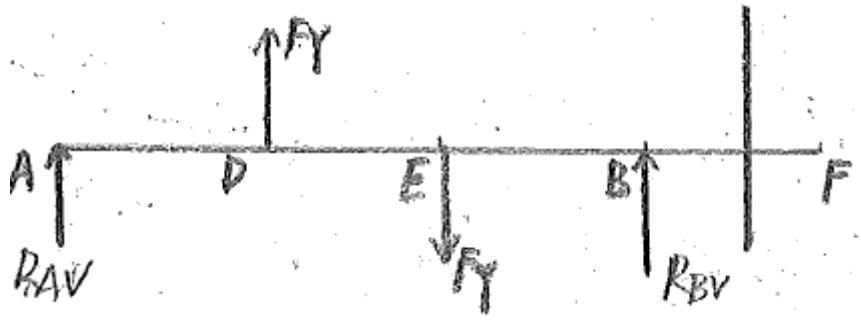


Figure 5.6 Section BF

$$\downarrow: S_{(x)} - R_{AV} - F_Y + F_Y - R_{BV} = 0$$

$$\rightarrow S_{(x)} = -200\text{N}$$

$$\text{BF: } M_{(x)} - R_{AV} \cdot x - F_Y \cdot (x - 21.75) + F_Y \cdot (x - 37.75) - R_{BV} \cdot (x - 59.5) = 0$$

$$M_{(x)} + 1111.1 \cdot x + 4360 \times 21.75 - 4360 \times 37.75 + 911.1 \times 59.5 - 911.1 \cdot x = 0$$

$$\rightarrow M_{(x)} = -200 \cdot x + 15550$$

In this section when $x = 54\text{mm}$ (at point B) has the maximum moment.

$$M_B = -200 \times 59.5 + 15550 = 3650\text{N} \cdot \text{mm}$$

Shear force diagram and bending moment diagram in vertical view

Figure 5.7 and 5.8 show shear force diagram and bending moment diagram in vertical view, respectively.

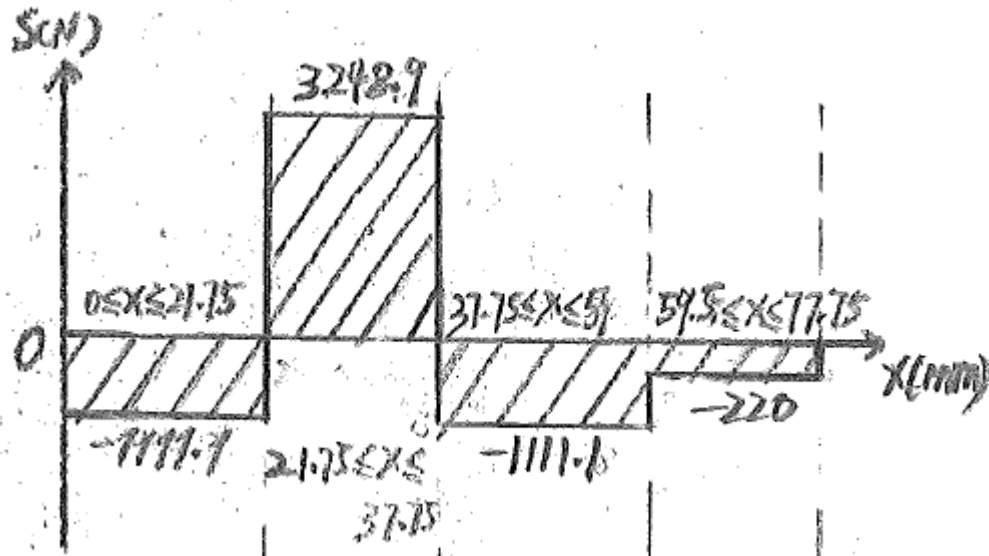


Figure 5.7 Shear force diagram in vertical view

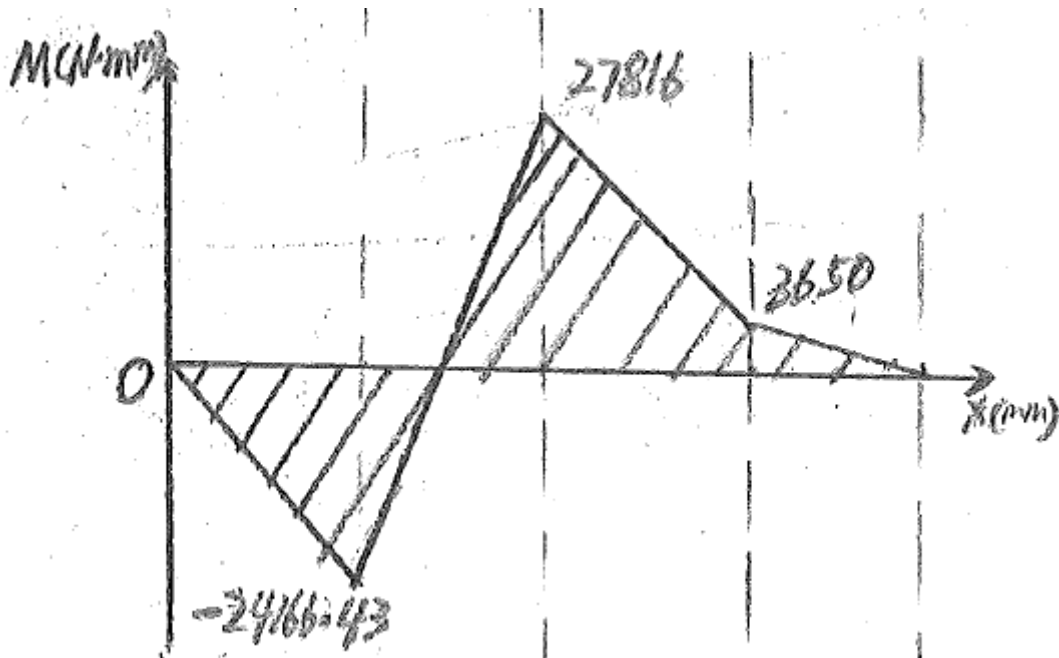


Figure 5.8 bending moment diagram in vertical view

From figure 5.7 and 5.8 the maximum shear force and bending moment in vertical view can be found.

The maximum shear force in vertical view:

$$S_v = 3248.9 \text{ N (At section DE)}$$

The maximum bending moment in vertical view:

$$M_v = 27816 \text{ N} \cdot \text{m (At section E-E)}$$

Horizontal view (from above)

From the analysis of forces acted on the shaft and selected bearings and gear, figure 5.9 shows the horizontal view of the shaft with forces on it.

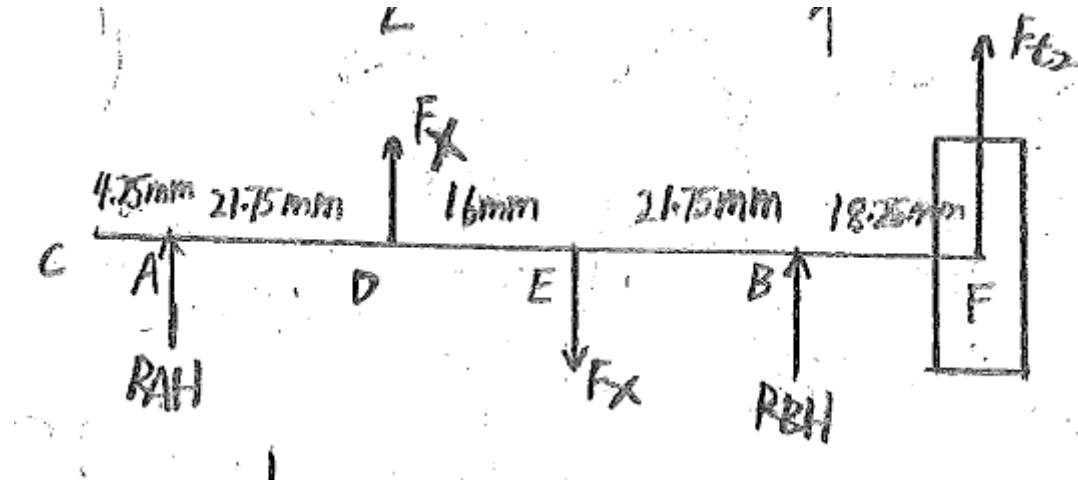


Figure 5.9 Horizontal view of the shaft with forces

Find reaction forces

$$\uparrow: R_{AH} + F_X + (-F_X) + R_{BH} + F_{t2} = 0$$

$$\rightarrow R_{AH} + R_{BH} = -550 \quad (1)$$

$$A: F_X \times 21.75 - F_X \times (21.75 + 16) + R_{BH} \times (21.75 + 16 + 21.75) + F_{t2} \times (21.75 + 16 + 21.75 + 18.25) = 0$$

$$\rightarrow -165680 + 59.5 \cdot R_{BH} + 42762.5 = 0$$

$$\rightarrow R_{BH} = 2065.8N \quad (2)$$

Plug into equation(2):

$$\rightarrow R_{AH} + 2065.8 = -550$$

$$\rightarrow R_{AH} = -2615.8N$$

Calculation of shear forces and bending moments

Section AD: $0 \leq x \leq 21.75\text{mm}$ (see figure 5.10).

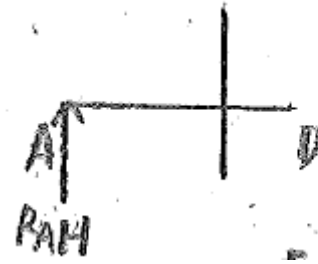


Figure 5.10 Section AD

$$\downarrow S_{(x)} - R_{AH} = 0$$

$$\rightarrow S_{(x)} = -2615.8\text{N}$$

$$AD: M_{(x)} - R_{AH} \cdot x = 0$$

$$\rightarrow M_{(x)} = -2615.8 \cdot x$$

In this section when $x = 19\text{mm}$ (at point D) has the maximum moment:

$$M_D = -2615.8 \times 21.75 = -56893.7\text{N} \cdot \text{mm}$$

Section DE: $21.75 \leq x \leq 37.75\text{mm}$ (see figure 5.11).

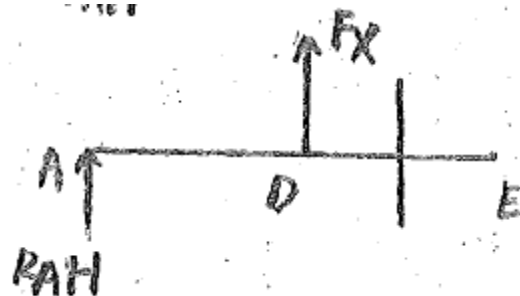


Figure 5.11 Section DE

$$\downarrow S_{(x)} - F_X - R_{AH} = 0$$

$$\rightarrow S_{(x)} = 10355 + (-2615.8) = 7739.2\text{N}$$

$$DE: M_{(x)} - R_{AH} \cdot x - F_X \cdot (x - 19) = 0$$

$$M_{(x)} + 2615.8 \cdot x - 10355 \cdot x + 10355 \times 21.75 = 0$$

$$\rightarrow M_{(x)} = 7739.2 \cdot x - 225221$$

In this section when $x = 37.75\text{mm}$ (at point E) has the maximum moment:

$$M_E = 7739.2 \times 37.75 - 225221 = 66934\text{N} \cdot \text{mm}$$

Section EB: $37.75 \leq x \leq 59.5\text{mm}$ (see figure 5.12).

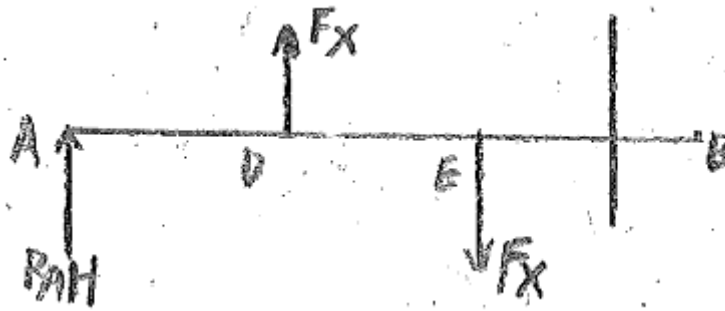


Figure 5.12 Section EB

$$\downarrow: S_{(x)} - R_{AH} - F_X + X = 0$$

$$\rightarrow S_{(x)} = -2615.8\text{N}$$

$$\text{EB: } M_{(x)} - R_{AH} \cdot x - F_X \cdot (x - 21.75) + F_X \cdot (x - 37.75) = 0$$

$$M_{(x)} + 2615.8 \cdot x + 10355 \times 21.75 - 10355 \times 37.75 = 0$$

$$\rightarrow M_{(x)} = -2615.8 \cdot x + 165680$$

In this section when $x = 37.75\text{mm}$ (at point E) has the maximum moment.

$$M_E = -2615.8 \times 37.75 + 165680 = 66934\text{N} \cdot \text{mm}$$

Section BF: $59.5 \leq x \leq 77.75\text{mm}$ (see figure 5.13).

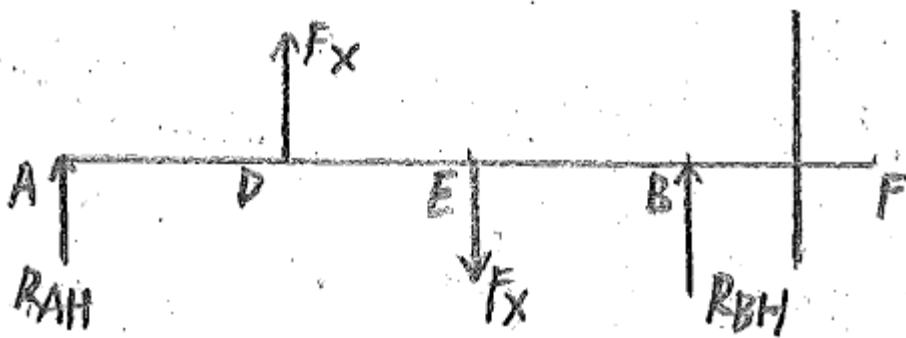


Figure 5.13 Section BF

$$\downarrow: S_{(x)} - R_{AH} - F_X + F_X - R_{BH} = 0$$

$$\rightarrow S_{(x)} = -550\text{N}$$

$$\text{BF: } M_{(x)} - R_{AH} \cdot x - F_X \cdot (x - 21.75) + F_X \cdot (x - 37.75) - R_{BH} \cdot (x - 59.5) = 0$$

$$M_{(x)} + 2615.8 \cdot x + 10355 \times 21.75 - 10355 \times 37.75 + 2065.8 \times 59.5 - 2065.8 \cdot x = 0$$

$$\rightarrow M_{(x)} = -550 \cdot X + 42762$$

In this section when $x = 54\text{mm}$ (at point B) has the maximum moment.

$$M_B = -550 \times 59.5 + 42762 = 10037\text{N} \cdot \text{mm}$$

Shear force diagram and bending moment diagram in horizontal view

Figure 5.14 and 5.15 show shear force diagram and bending moment diagram in horizontal view, respectively.

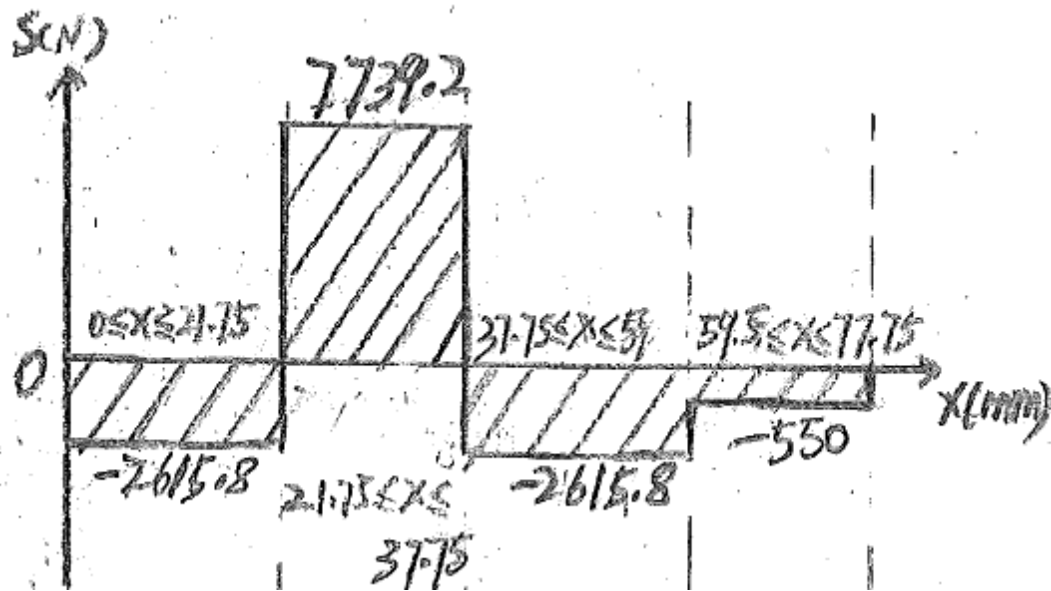


Figure 5.14 Shear force diagram in horizontal view

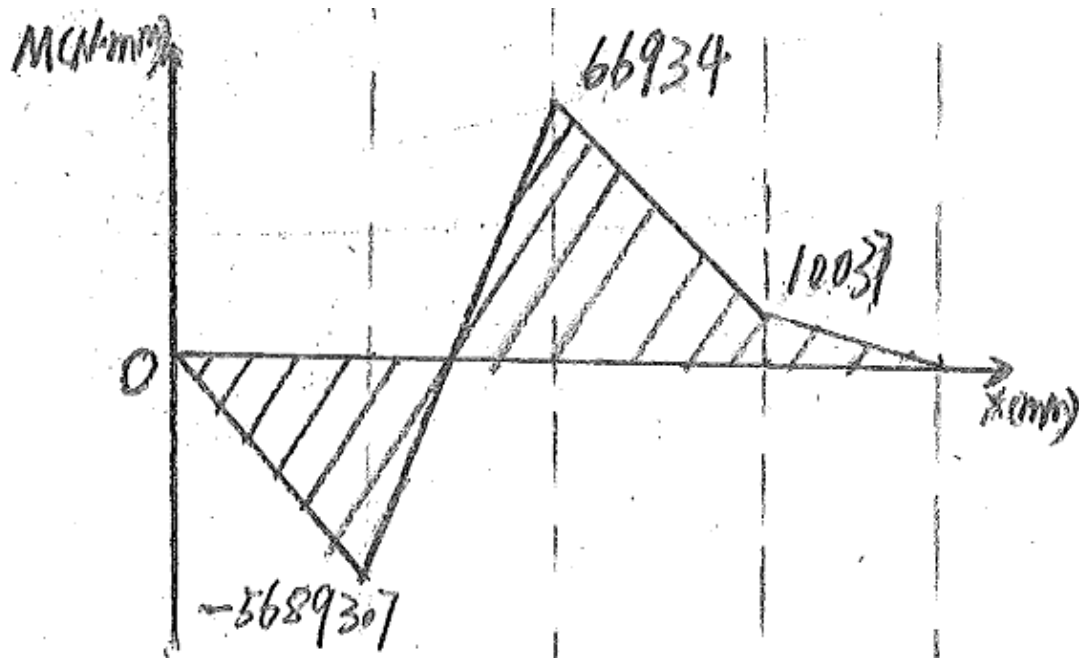


Figure 5.15 Bending moment in horizontal view

From figure 5.14 and 5.15 the maximum shear force and bending moment in horizontal view can be found.

The maximum shear force in horizontal view:

$$S_H = 7739.2\text{N (At section DE)}$$

The maximum bending moment in horizontal view:

$$M_H = 66934\text{N} \cdot \text{m (At section E-E)}$$

Torsion loading diagram

The torque is generated by the rotating gear.

$$T = F_{t2} \times \frac{d_2}{2} = 550 \times \frac{138}{2}$$

$$\rightarrow T = 37950\text{N} \cdot \text{mm (At section F-F)}$$

Figure 5.16 displays the torsion loading diagram.

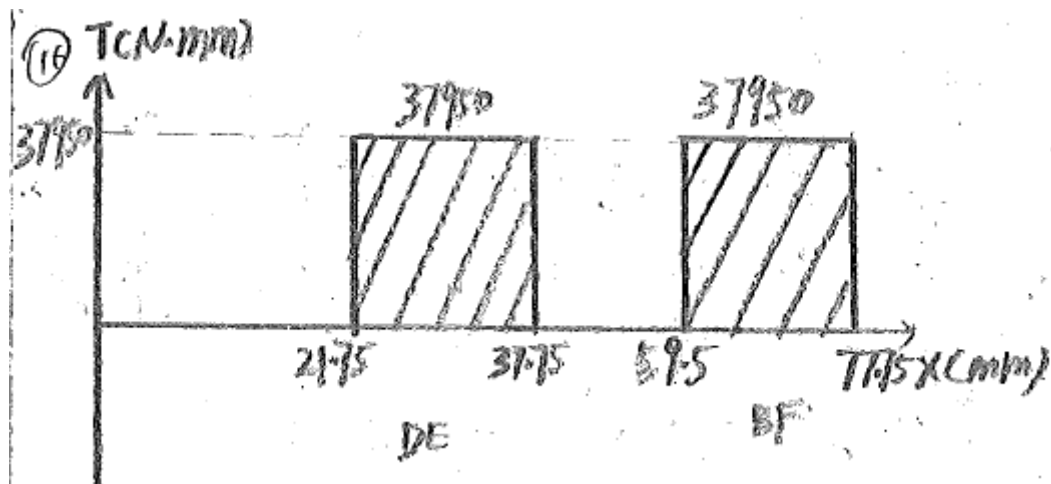


Figure 5.16 Torsional loading diagram

Bearing loads

Since loads on eccentric bearings have been calculated, this part analyzes the loads on bearing A and bearing B.

Find loads on bearing A

There is no axial load exist on the shaft that the axial load on bearing A is zero.

$$A_a = 0\text{N}$$

Figure 5.17 shows the radial load on bearing A.

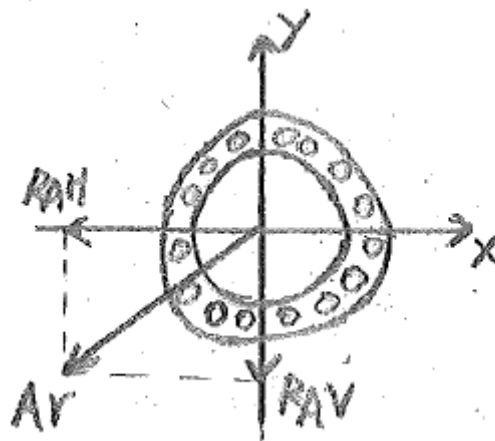


Figure 5.17 Radial load on bearing A

$$A_r = \sqrt{R_{AV}^2 + R_{AH}^2} = \sqrt{(-1111.1)^2 + (-2615.8)^2}$$

$$\rightarrow A_r = 2842\text{N}$$

Find loads on bearing B

There is no axial load exist on the shaft that the axial load on bearing B is zero.

$$B_a = 0\text{N}$$

Figure 5.18 shows the radial load on bearing B.

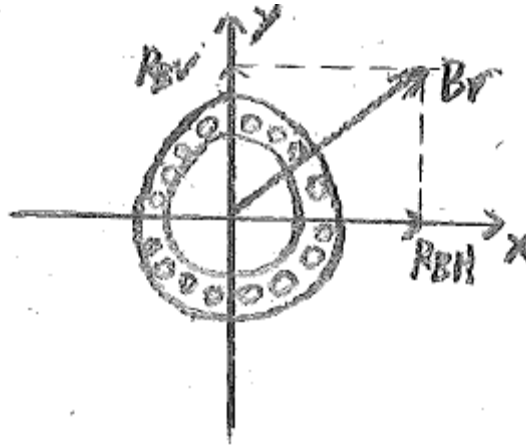


Figure 5.18 Radial load on bearing B

$$B_r = \sqrt{R_{BV}^2 + R_{BH}^2} = \sqrt{911.1^2 + 2065.8^2}$$

$$\rightarrow B_r = 2258\text{N}$$

Find safety factor for the shaft

In the previous part, the maximum bending moment occurs at section DE where mounted with one eccentric bearing. The maximum torque occurs on section BF ($d_3 = 18\text{mm}$) where located the gear and section DE where mounted with one eccentric bearing. It is obvious that both of the maximum bending moment and maximum torque will happen at a same section E-E (in the middle of section DE). As a result, the whole shaft safety factor will be calculated from the most dangerous section E-E.

Find torsional stress caused by the maximum torque

The maximum torque is $T = 37950\text{N} \cdot \text{mm}$ (at section E-E in figure 5.19).

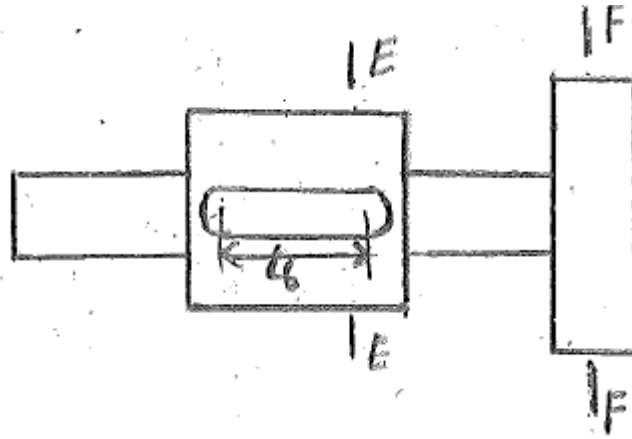


Figure 5.19 Section E-E

The shaft diameter in this section is $D = 22\text{mm}$.

The fillet radius on whole shaft is $r_1 = 2\text{mm}$.

$$\left\{ \begin{array}{l} D = 22\text{mm}. \\ \text{Table 1 (reference figure 5.21) in page 11 of Shaft design and shaft related parts} \end{array} \right.$$

$$\rightarrow \left\{ \begin{array}{l} B = 6\text{mm} \\ H = 6\text{mm} \end{array} \right.$$

The fillet radius on the key is $r_2 = 0.2\text{mm}$.

$$\rightarrow \frac{r_2}{B} = \frac{0.2}{6} = 0.033$$

$$\left\{ \begin{array}{l} \text{Diagram 3 (reference figure 5.22) in page 12 of Shaft design and shaft related parts} \\ \frac{r_2}{B} = 0.033 \end{array} \right.$$

$$\rightarrow K_{t1} = 3.7$$

From Appendix 3, the hardness of the shaft material (20MnCr6-5) is 350HBW.

$$\rightarrow \left\{ \begin{array}{l} S_u = 923\text{MPa} (134\text{ksi}) \\ S_y = 750\text{MPa} \end{array} \right.$$

$$\left\{ \begin{array}{l} \text{Figure 8.24 (reference figure 5.23) in page 28 of Handbook for Machine Design} \\ r_1 = 2\text{mm} \\ S_u = 134\text{ksi} \\ \text{Torsion} \end{array} \right.$$

$$\rightarrow q_1 = 0.92$$

Handbook for Machine Design (8.2) $K_f = 1 + (K_t - 1) \cdot q$

$$K_{f1} = 1 + (K_{t1} - 1) \cdot q_1 = 1 + (3.7 - 1) \times 0.92$$

$$\rightarrow K_{f1} = 3.484$$

Since the torque is constant, the torsion alternating stress $\tau_a = 0 \text{ MPa}$

Handbook for Machine Design (8.5) $\frac{\tau_m}{SF \cdot K_f} = \frac{16 \cdot T_m}{\pi \cdot d^3}$

$$\frac{\tau_m}{SF \cdot K_{f1}} = \frac{16 \cdot T}{\pi \cdot D^3}$$

$$\rightarrow \tau_m = \frac{16 \times 37950 \times 3.484}{\pi \times 22^3} SF = 63.2 \cdot SF$$

Find axial stress

There is no axial force on the shaft which means the axial stress is zero.

$$\sigma_{a,a} = \sigma_{a,m} = 0 \text{ MPa}$$

Find bending stress caused by the maximum bending moment

$$M_v = 27816 \text{ N} \cdot \text{mm}$$

$$M_H = 66934 \text{ N} \cdot \text{mm}$$

$$M = \sqrt{M_v^2 + M_H^2} = \sqrt{27816^2 + 66934^2}$$

$$\rightarrow M = 72484 \text{ N} \cdot \text{mm} \text{ At section E-E (figure 5.19)}$$

The shaft diameter (mounted with eccentric bearing) is $D = 22 \text{ mm}$.

The shaft diameter (mounted with tapered roller bearing) is $d_4 = 20 \text{ mm}$.

{ Figure 8.24 (reference figure 5.23) in page 28 of Handbook for Machine Design
 $r_1 = 2 \text{ mm}$
 $S_u = 134 \text{ ksi}$
 Bending

$$\rightarrow q_2 = 0.87$$

$$\left\{ \begin{array}{l} \text{Figure 4.35 (a) in page 14 of Handbook for Machine Design (reference figure 5.24)} \\ \frac{D}{d_4} = \frac{22}{20} = 1.1 \\ \frac{r_1}{d_4} = \frac{2}{20} = 0.1 \\ \text{Bending} \end{array} \right.$$

$$\rightarrow K_{t2} = 1.57$$

$$\text{Handbook for Machine Design (8.2) } K_f = 1 + (K_t - 1) \cdot q$$

$$K_{f2} = 1 + (K_{t2} - 1) \cdot q_2 = 1 + (1.57 - 1) \times 0.87$$

$$\rightarrow K_{f2} = 1.5$$

The rotating static bending moment results in alternating stress on the shaft and the bending mean stress is zero.

$$\sigma_{b,m} = 0 \text{ MPa}$$

$$\text{Handbook for Machine Design (8.6) } \frac{\sigma_a}{SF \cdot K_f} = \frac{32 \cdot M_a}{\pi \cdot d^3}$$

$$\frac{\sigma_{b,a}}{SF \cdot K_{f2}} = \frac{32 \cdot M}{\pi \cdot D^3}$$

$$\rightarrow \sigma_{b,a} = \frac{32 \times 72484 \times 1.5}{\pi \times 22^3} \times SF = 104SF$$

Find equivalent bending stress

$$\text{Equation (a) in page 25 of Handbook for Machine Design: } \sigma_{ea} = \sqrt{\sigma_a^2 + \tau_a^2}$$

$$\sigma_a = \sqrt{\sigma_{a,a}^2 + \sigma_{b,a}^2} = \sqrt{0^2 + (104SF)^2}$$

$$\rightarrow \sigma_a = 104SF$$

$$\tau_a = 0 \text{ MPa}$$

$$\rightarrow \sigma_{ea} = \sqrt{0^2 + (104SF)^2} = 104SF$$

Equation (b) in page 25 of Handbook for Machine Design: $\sigma_{em} = \frac{\sigma_m}{2} + \sqrt{\tau_m^2 + \frac{\sigma_m^2}{2}}$

$$\sigma_m = \sqrt{\sigma_{a,m}^2 + \sigma_{b,m}^2} = \sqrt{0^2 + 0^2}$$

$$\sigma_m = 0 \text{ MPa}$$

$$\tau_m = 63.2 \text{ SF}$$

$$\sigma_{em} = 0 + \sqrt{(63.2 \text{ SF})^2 + 0^2} = 63.2 \cdot \text{SF}$$

Find fatigue endurance limit of shaft

Table 8.1a (reference figure 5.25) in page 27 of Handbook for Machine Design, assume $N > 10^6$.

$$S_n = S'_n \cdot C_L \cdot C_G \cdot C_S \cdot C_T \cdot C_R$$

$$C_T = C_R = 1 \text{ (No information)}$$

$$S'_n = 0.5 \cdot S_u \text{ (No information, steel)}$$

$$\rightarrow S'_n = 0.5 \times 923 = 461.5 \text{ MPa}$$

$$C_L = 1 \text{ (Bending)}$$

$$C_G = 0.9 \text{ (Bending, } 10 < d < 50 \text{ mm)}$$

$$\left\{ \begin{array}{l} S_u = 134 \text{ ksi} \\ \text{Fine - ground} \end{array} \right. \text{ Figure 8.13 (reference figure 5.26) in page 26 of Handbook for Machine Design}$$

$$\rightarrow C_S = 0.9$$

$$\rightarrow S_n = 461.5 \times 1 \times 0.9 \times 0.9 \times 1 \times 1 = 374 \text{ MPa}$$

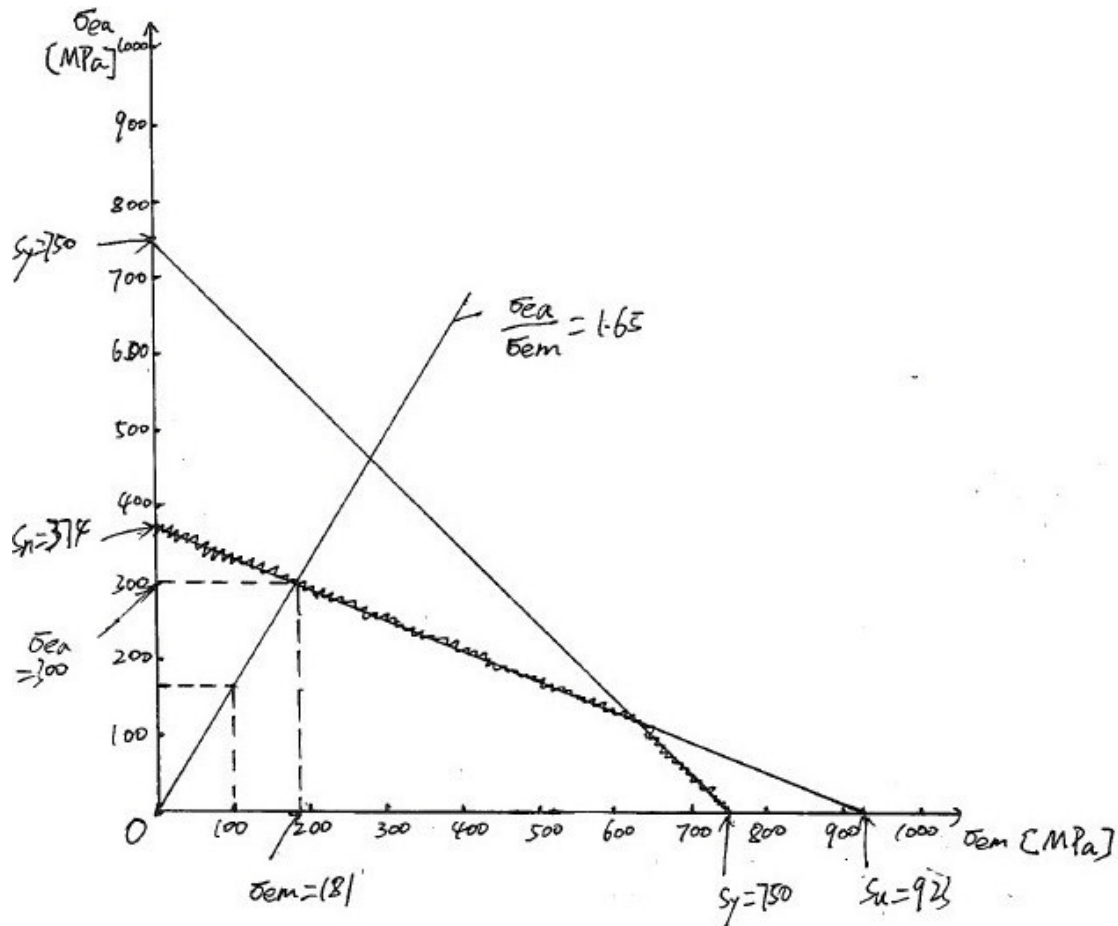


Diagram 5.1 Fatigue strength diagram

$$\left\{ \begin{array}{l} \frac{\sigma_{ea}}{\sigma_{em}} = \frac{104SF}{63.2SF} = 1.65 \\ \text{Diagram 5.1 (fatigue strength diagram)} \end{array} \right.$$

$$\rightarrow \sigma_{ea} = 300 \text{ MPa}$$

$$\sigma_{em} = 181 \text{ MPa}$$

$$\rightarrow SF_1 = \frac{300}{104} = 2.88$$

Find another safety factor on section F-F

Although there is only torsional stress on section F-F (figure 5.20), shaft diameter on this part is the minimum. It is necessary to check this part by generating another safety factor.

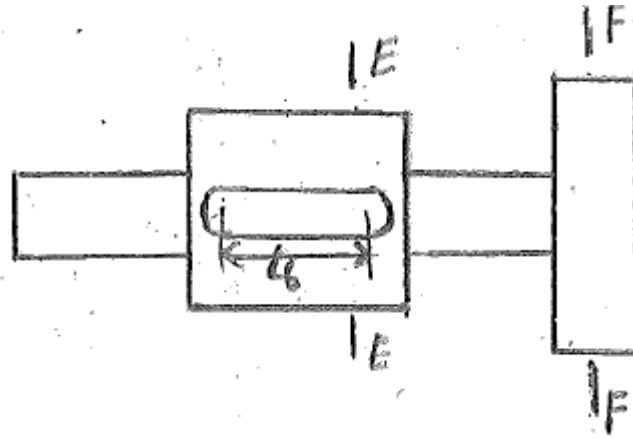


Figure 5.20 First stage output shaft

Spline is manufactured on this section to connect gear with the shaft. A flat key which has approximate dimension with the selected spline (in Appendix 8) will be used in the calculation of the shaft. Because spline is usually stronger than flat key, if the flat key can fulfill the requirement, the spline will also be safe.

The shaft diameter is $d_3 = 18\text{mm}$ where mounted with the gear

The fillet radius on whole shaft is $r_1 = 2\text{mm}$.

$$\left\{ \begin{array}{l} d_3 = 18\text{mm} \\ \text{Table 1 (reference figure 5.1) in page 11 of Shaft design and shaft related parts} \end{array} \right.$$

$$\rightarrow \begin{cases} B = 6\text{mm} \\ H = 6\text{mm} \end{cases}$$

The fillet radius on the key is $r_2 = 0.2\text{mm}$.

$$\rightarrow \frac{r_2}{B} = \frac{0.2}{6} = 0.033$$

$$\left\{ \begin{array}{l} \text{Diagram 3 (reference figure 5.2) in page 12 of Shaft design and shaft related parts} \\ \frac{r_2}{B} = 0.033 \end{array} \right.$$

$$\rightarrow K_{t3} = 3.7$$

$$\left\{ \begin{array}{l} \text{Figure 8.24 (reference figure 5.3) in page 28 of Handbook for Machine Design} \\ r_1 = 2\text{mm} \\ S_u = 134\text{ksi} \\ \text{Torsion} \end{array} \right.$$

$$\rightarrow q_3 = 0.92$$

Handbook for Machine Design (8.2) $K_f = 1 + (K_t - 1) \cdot q$

$$K_{f3} = 1 + (K_{t3} - 1) \cdot q_3 = 1 + (3.7 - 1) \times 0.92$$

$$\rightarrow K_{f3} = 3.484$$

Handbook for Machine Design (8.5) $\frac{\tau_m}{SF \cdot K_f} = \frac{16 \cdot T_m}{\pi \cdot d^3}$

$$\frac{\tau_m}{SF \cdot K_{f1}} = \frac{16 \cdot T}{\pi \cdot d_3^3}$$

$$\rightarrow \tau_m = \frac{16 \times 37950 \times 3.484}{\pi \times 18^3} SF = 115 \cdot SF$$

Equation (8.4) in page 25 of Handbook for Machine Design $S_{ys} = 0.58 \cdot S_y$

$$\rightarrow S_{ys} = 0.58 \times 750 = 435 \text{ MPa}$$

Only torsional stress on this section

$$S_{ys} = 115 \cdot SF$$

$$\rightarrow SF_2 = \frac{435}{115} = 3.8$$

Select the final safety factor

Compare two generated safety factor

$$SF_1 = 2.88 < SF_2 = 3.8$$

Select the final safety factor is:

$$SF = SF_1 = 2.88$$

Reference

table from Swedish standard SMS 2305.

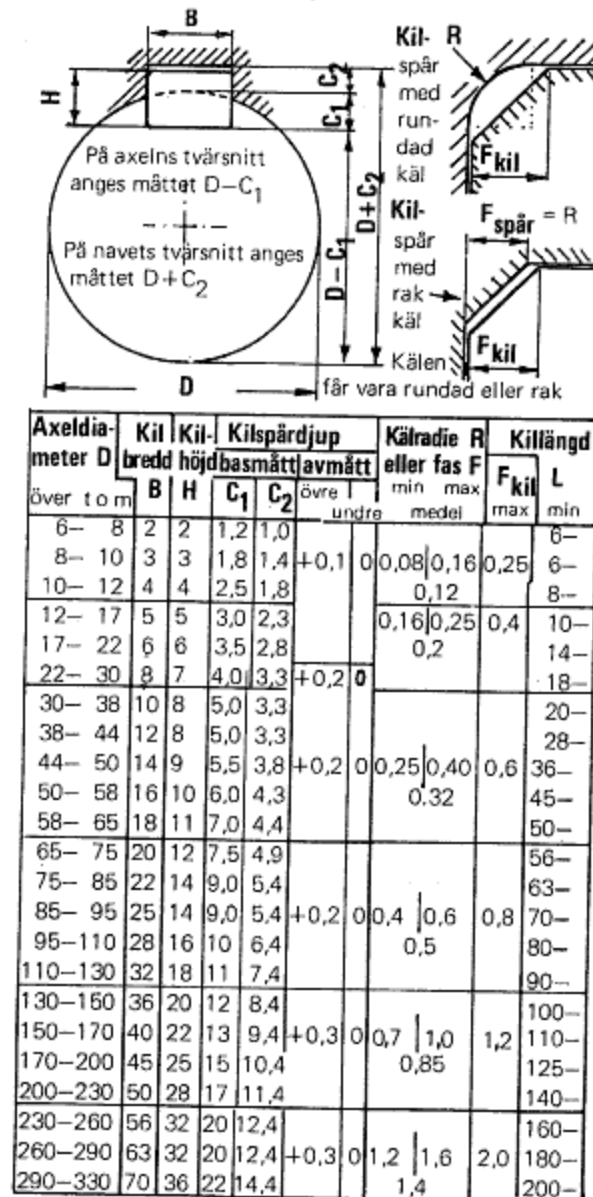


Figure 5.21 Table 1 key standard

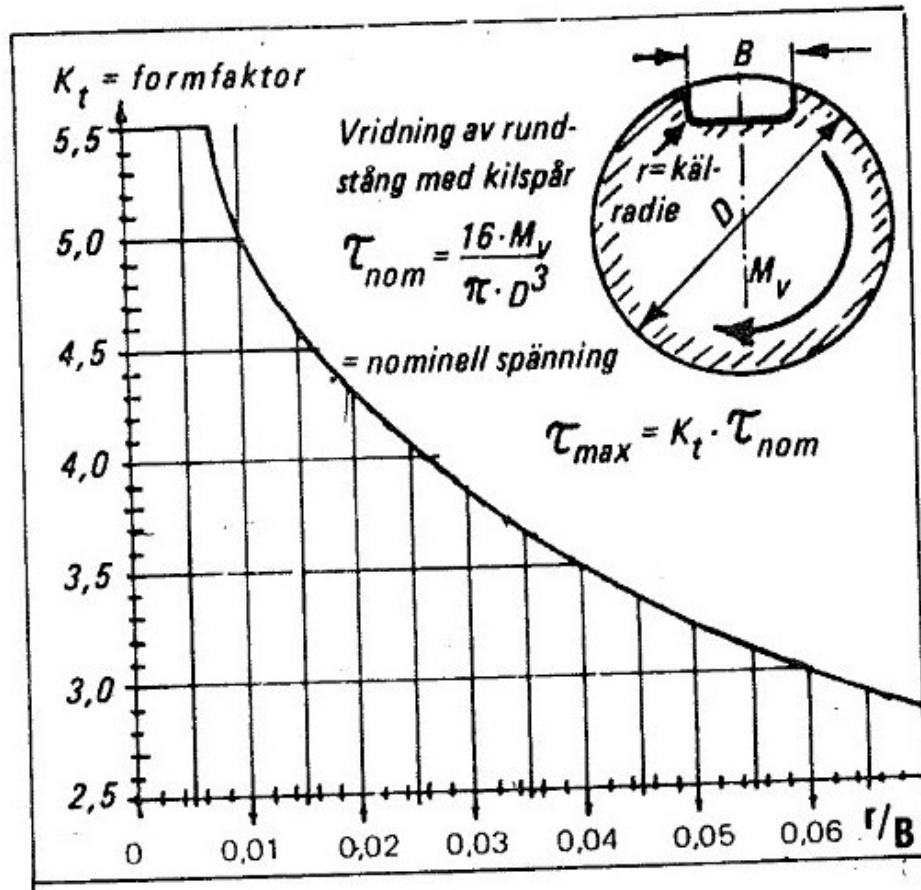


Diagram 3. For calculation of static stress concentration factor in keyways

Figure 5.22 diagram for static stress concentration factor in keyways

FIGURE 8.24
Notch sensitivity curves
(after [9]). Note: (1) Here r
is the radius at the point
where the potential fatigue
crack originates. (2) For
 $r > 0.16$ in., extrapolate or
use $q \approx 1$.

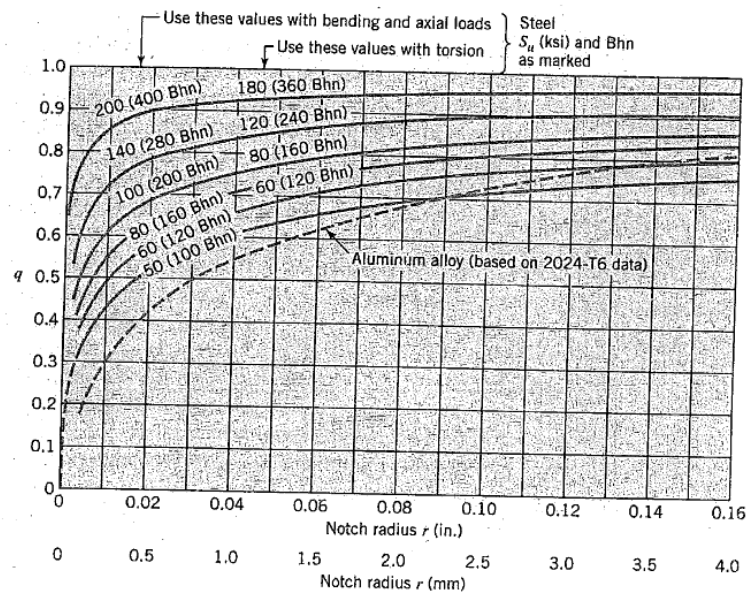


Figure 5.23 Notch sensitivity curves

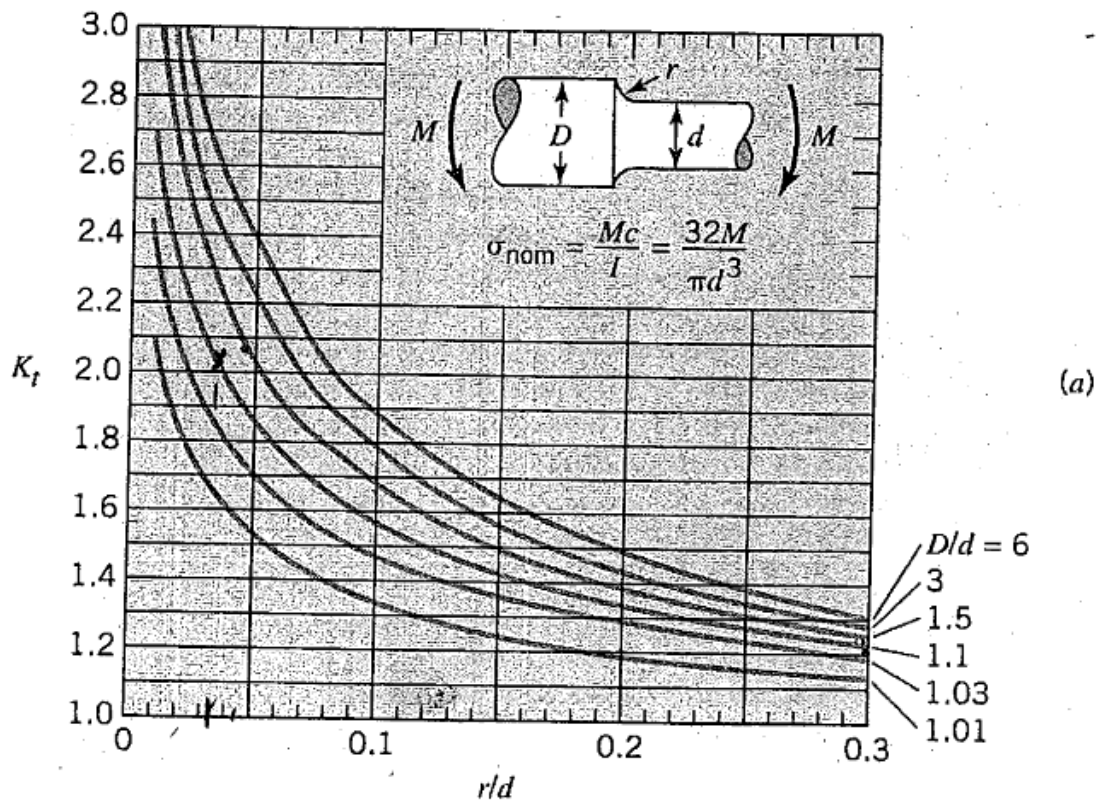


Figure 5.24 Figure 4.35(a) in page 14 of Handbook for Machine Design

TABLE 8.1 Generalized Fatigue Strength Factors for Ductile Materials (*S-N* curves)**a. 10^6 -cycle strength (endurance limit)^a**Bending loads: $S_n = S'_n C_L C_G C_S C_T C_R$ Axial loads: $S_n = S'_n C_L C_G C_S C_T C_R$ Torsional loads: $S_n = S'_n C_L C_G C_S C_T C_R$ where S'_n is the R.R. Moore, endurance limit,^b and

	Bending	Axial	Torsion
C_L (load factor)	1.0	1.0	0.58
C_G (gradient factor): diameter < (0.4 in. or 10 mm)	1.0	0.7 to 0.9	1.0
(0.4 in. or 10 mm) < diameter < (2 in. or 50 mm) ^c	0.9	0.7 to 0.9	0.9
C_S (surface factor)	see Figure 8.13		
C_T (temperature factor)	Values are only for steel		
$T \leq 840^\circ\text{F}$	1.0	1.0	1.0
$840^\circ\text{F} < T \leq 1020^\circ\text{F}$	$1 - (0.0032T - 2.688)$		
C_R (reliability factor): ^d			
50% reliability	1.000	"	"
90% "	0.897	"	"
95% "	0.868	"	"
99% "	0.814	"	"
99.9% "	0.753	"	"

b. 10^3 -cycle strength^{e, f, g}Bending loads: $S_f = 0.9S_u C_T$ Axial loads: $S_f = 0.75S_u C_T$ Torsional loads: $S_f = 0.9S_{us} C_T$ where S_u is the ultimate tensile strength and S_{us} is the ultimate shear strength.^aFor materials not having the endurance limit, apply the factors to the 10^8 or 5×10^8 -cycle strength.^b $S'_n = 0.5S_u$ for steel, lacking better data.^cFor (2 in. or 50 mm) < diameter < (4 in. or 100 mm) reduce these factors by about 0.1. For (4 in. or 100 mm) < diameter < (6 in. or 150 mm), reduce these factors by about 0.2.^dThe factor, C_R , corresponds to an 8 percent standard deviation of the endurance limit. For example, for 99% reliability we shift -2.326 standard deviations, and $C_R = 1 - 2.326(0.08) = 0.814$.^eNo corrections for gradient or surface are normally made, but the experimental value of S_u or S_{us} should pertain to sizes reasonably close to those involved.^fNo correction is usually made for reliability at 10^3 cycle strength.^g $S_{us} \approx 0.8S_u$ for steel; $S_{us} \approx 0.7S_u$ for other ductile metals.*Figure 5.25 Generalized fatigue strength factors for ductile materials*

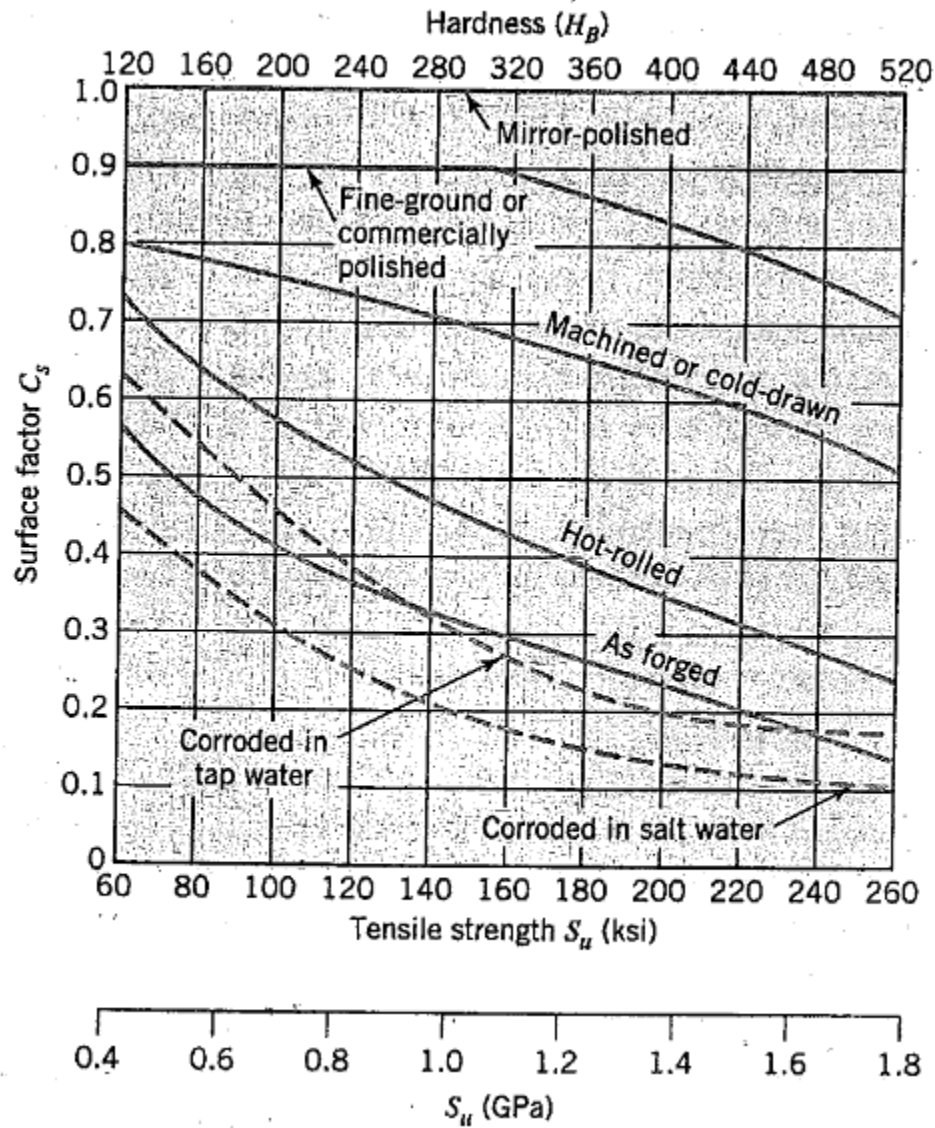


Figure 5.26 Reduction in endurance limit owing to surface finish-steel parts

Appendix 6 Calculation of tapered roller bearings

Symbol	Explanation	Value	Other
A_a	Axial load on bearing A	0N	From Appendix 5
A_r	Radial load on bearing A	2842N	From Appendix 5
B_a	Axial load on bearing B	0N	From Appendix 5
B_r	Radial load on bearing B	2258N	From Appendix 5
P	Equivalent dynamic bearing load	2842N	Calculated
a_1	Life adjustment factor for reliability	0.33	Table 1 (page 53 of SKF General Catalog)
p	Exponent of the life equation	$\frac{10}{3}$	Given
C	Basic dynamic bearing load	27.5kN	Page 618 of SKF General Catalog
n	Bearing rotational speed	249.4rpm	From Appendix 1
d	Minimum bearing diameter	20mm	Page 618 of SKF General Catalog
D	Maximum bearing diameter	47mm	Page 618 of SKF General Catalog
d_m	Mean bearing diameter	33.5mm	Calculated
v_1	Required viscosity	70 mm ² /s	Diagram 5 in page 60 of SKF General Catalog
v	Actual operating viscosity	60 mm ² /s	Diagram 6 in page 61 of SKF General Catalog
k	Viscosity ratio	0.86	Calculated
η_c	Degree of contamination	0.55	Table 4 in page 62 of SKF General Catalog
P_u	Fatigue load limit	3kN	Page 618 of SKF General Catalog
a_{SKF}	SKF life modification factor	1.3	Diagram 2 in page 55 of SKF General Catalog
L_{2mh}	SKF rating life at 98% reliability	55348h	Calculated

Loads on bearings

Two tapered roller bearing (bearing A and B) have been mounted on the output shaft of first stage. In Appendix 5 loads act on them have been calculated. Only radial forces are acted on those two bearings, the axial forces are zero.

Axial load on bearing A: $A_a = 0\text{N}$

Radial load on bearing A: $A_r = 2842\text{N}$

Axial load on bearing B: $B_a = 0\text{N}$

Radial load on bearing B: $B_r = 2258\text{N} < A_r = 2842\text{N}$

The magnitude of force on bearing A is bigger than bearing B. Since bearing A and B is selected to the same type of standard tapered roller bearing on SKF catalog the calculation can only treat bearing A.

Because of $A_a = 0\text{N}$, the equivalent dynamic bearing load is $P = A_r = 2842\text{N}$.

Find bearing type in SKF General Catalog

The bearing type is chosen in Tapered Roller bearing SKF General Catalog. According to its diameters, the type is selected as: **30204J2/Q** in page 618 of SKF General Catalog

Since bearings are chosen in the SKF General Catalog the calculation process and equations must follow the process given by SKF.

Find SKF rating life

Equation in page 52 of SKF General Catalog:

$$L_{nmh} = a_1 \cdot a_{SKF} \cdot \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P}\right)^p$$

Symbol L_{nmh} is the SKF rating life at $100 - n\%$. This n in the symbol represent the failure probability of the bearing. The total reliability of bearings on one shaft is selected as 90%. As the number of bearings is 4 (2 tapered roller bearings and 2 eccentric bearings), the reliability of one tapered roller bearing is:

$$\text{Reliability} \approx \sqrt[4]{90\%} = 97.4\%$$

Select the reliability of this bearing is 98%. The failure probability is 2% that means the SKF rating life can be written as L_{2mh} .

The exponent of the life equation is $p = \frac{10}{3}$ for roller bearings.

- **Find life adjustment factor for reliability a_1**

$$\begin{cases} \text{Reliability} = 98\% \\ \text{Table 1 in page 53 of SKF General Catalog} \end{cases}$$

$$\rightarrow a_1 = 0.33$$

- **Find basic dynamic load rating and rotational speed**

From the bearing type **30204J2/Q** and the SKF General Catalog, the basic dynamic load rating is:

$$C = 27.5\text{kN}$$

From Appendix 1 the rotational speed of bearing which has the same value of the first stage output shaft is:

$$n = n_{g_1} = 249.4\text{rpm}$$

- **Find SKF life modification factor**

From the bearing type **30204J2/Q** and the SKF General Catalog:

$$d = 20\text{mm}$$

$$D = 47\text{mm}$$

The mean diameter is:

$$d_m = \frac{d + D}{2} = \frac{20 + 47}{2}$$

$$\rightarrow d_m = 33.5\text{mm}$$

$$\begin{cases} \text{Diagram 5 in page 60 of SKF General Catalog} \\ d_m = 33.5\text{mm} \\ n = 249.4\text{rpm} \end{cases}$$

$$\rightarrow v_1 = 70\text{ mm}^2/\text{s}$$

$$\begin{cases} \text{Table 2 in page 246 of SKF General Catalog} \\ \text{Diagram 6 in page 61 of SKF General Catalog} \end{cases}$$

$$\rightarrow v = 60\text{ mm}^2/\text{s} \text{ (At } 55^\circ\text{C, operating temperature)}$$

The viscosity ratio is:

$$k = \frac{v}{v_1} = \frac{60}{70}$$

$$\rightarrow k = 0.86$$

$$\left\{ \begin{array}{l} \text{Table 4 in page 62 of SKF General Catalog} \\ \text{Normal cleanliness} \\ d_m = 33.5\text{mm} < 100\text{mm} \end{array} \right.$$

$$\rightarrow \eta_c = 0.55$$

From the bearing type **30204J2/Q** and the SKF General Catalog, the fatigue load limit is:

$$P_u = 3\text{kN}$$

$$\eta_c \cdot \frac{P_u}{p} = 0.55 \times \frac{3000}{3171}$$

$$\rightarrow \eta_c \cdot \frac{P_u}{p} = 0.52$$

$$\left\{ \begin{array}{l} \text{Diagram 2 in page 55 of SKF General Catalog} \\ \eta_c \cdot \frac{P_u}{p} = 0.52 \\ k = 0.86 \end{array} \right.$$

$$\rightarrow a_{\text{SKF}} = 1.3$$

- **Find SKF rating life**

$$L_{2\text{mh}} = a_1 \cdot a_{\text{SKF}} \cdot \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P} \right)^p = 0.33 \times 1.3 \times \frac{10^6}{60 \times 249.4} \times \left(\frac{27.5 \times 10^3}{2842} \right)^{\frac{10}{3}}$$

$$\rightarrow L_{2\text{mh}} = 55348\text{h}$$

Reference

SKF General Catalogue

Appendix 7 Input shaft design

Symbol	Explanation	Value	Other
T_{out}	Output torque	6300Nm	Given
η	Total efficiency	0.915	From Appendix 2
i	Total ratio	240.5	From Appendix 1
T	Input torque	28.6Nm	Calculated
d	Input shaft diameter	20mm	Calculated
B	Spline thickness	6mm	Selected
r	Fillet radius	0.2mm	Selected
K_t	Static stress concentration on keyway	3.7	Selected
S_y	Yield strength	750Mpa	From Appendix 3
q	Notch sensitive factor	0.92	Calculated
S_{ys}	Yield strength when torsion are applied	435Mpa	Calculated
K_f	Fatigue stress concentration factor	3.484	Calculated
τ_a	Alternating torsion stress	0	Calculated
τ_m	Mean torsion stress	63.4Mpa	Calculated
SF	Safety factor	6.86	Calculated

Input torque:

$$T = \frac{T_{out}}{\eta i} = \frac{6300}{0.915 * 240.5} = 28.6\text{Nm}$$

The shaft diameter is 20mm

Spline is manufactured on this part; in the calculation process will be treated as flat key. By using this way the result can ensure more safety to the shaft.

$$\left\{ \begin{array}{l} d = 20\text{mm} \\ \text{Table 1 (reference figure 7.1) in page 11 of Shaft design and shaft related parts} \end{array} \right.$$

$$\rightarrow \left\{ \begin{array}{l} B = 6\text{mm} \\ r = 0.2\text{mm} \end{array} \right.$$

$$\rightarrow \frac{r}{B} = \frac{0.2}{6} = 0.033$$

$$\left\{ \begin{array}{l} \text{Diagram 3 (reference figure 7.2) in page 12 of Shaft design and shaft related parts} \\ \frac{r}{B} = 0.033 \end{array} \right.$$

$$\rightarrow K_t = 3.7$$

From Appendix 3, the hardness of the shaft material (20MnCr6-5) is 350HBW.

$$\rightarrow S_y = 750 \text{Mpa}$$

$$\rightarrow S_{ys} = 0.58S_y = 0.58 * 750$$

$$\rightarrow S_{ys} = 435 \text{Mpa}$$

$$\left\{ \begin{array}{l} \text{Figure 8.24 (reference figure 7.3) in page 28 of Handbook for Machine Design} \\ r = 2\text{mm} \\ S_u = 134\text{ksi} \\ \text{Torsion} \end{array} \right.$$

$$\rightarrow q = 0.92$$

Handbook for Machine Design (8.2) $K_f = 1 + (K_t - 1) * q$

$$K_f = 1 + (K_t - 1) * q = 1 + (3.7 - 1) * 0.92$$

$$\rightarrow K_f = 3.484$$

Since the torque is constant, the torsion alternating stress $\tau_a = 0 \text{MPa}$

Handbook for Machine Design (8.5) $\frac{\tau_m}{SF \cdot K_f} = \frac{16 \cdot T_m}{\pi \cdot d^3}$

$$\frac{\tau_m}{SF \cdot K_f} = \frac{16 \cdot T}{\pi \cdot d^3}$$

$$\rightarrow \tau_m = \frac{16 * 28.6 * 1000 * 3.484}{\pi * 10^3} SF = 63.4 \cdot SF$$

There is no alternating torsion stress, so make $\tau_m = S_{ys}$.

$$\rightarrow 63.4 \cdot SF = 435$$

$$\rightarrow SF = 6.86$$

So the input shaft is strong enough when it performs.

Reference

Figure 7.1 (refer to figure 5.21 in appendix 5) Table 1 key standard

Figure 7.2 (refer to figure 5.22 in appendix 5) diagram for static stress concentration factor in keyways

Figure 7.3 (refer to figure 5.23 in appendix 5) Notch sensitivity curves

Appendix 8 Check strength of spline and key

Symbol	Explanation	Value	Other
d_3	Shaft diameter (mounted with gear)	18mm	From Appendix 5
N	Number of splines	6	Selected
d_5	Inner diameter of spline	18mm	Selected
D_1	Outer diameter of spline	20mm	Selected
W	Spline width	5mm	Selected
h	Height of spline	1mm	Selected
b	Gear width	20mm	From Appendix 3
k	Coefficient of load with no equal apportionment between teeth	0.75	Selected
l	Working length of spline tooth	20mm	Selected
d_m	Average diameter of spline	19mm	Calculated
$[\sigma_p]_1$	Allowable crushing stress for spline	120MPa	Selected from table 6.2 in page 103 of Machine Design (reference figure 8.3)
T	Torque on first stage output shaft	37950N · mm	From Appendix 5
D	Shaft diameter (mounted with eccentric bearing)	22mm	From Appendix 5
H	Key thickness	6mm	From Appendix 5
L_6	Working length of the key	22mm	Selected
$[\sigma_p]_2$	Allowable crushing stress for key on first stage output shaft	110MPa	Selected from table 6.1 in page 102 of Machine Design (reference figure 8.4)
T_1	Input torque	28600N · mm	From Appendix 7
d_6	Input shaft diameter	20mm	From Appendix 7

Check the strength of spline

As mentioned in Appendix 5 the part where mounted with gear on the shaft is calculated as a flat key from Swedish Standard. This is because that the result safety factor calculated from flat key will be lower than from spline. If the flat key can fulfill the requirement the spline with approximate dimensions to the flat key will definitely achieve too.

This part is going to check the shear stress on the selected spline to ensure safety again.

From the SAE Straight Tooth Splines Standard, the spline mounted on the shaft has been selected.

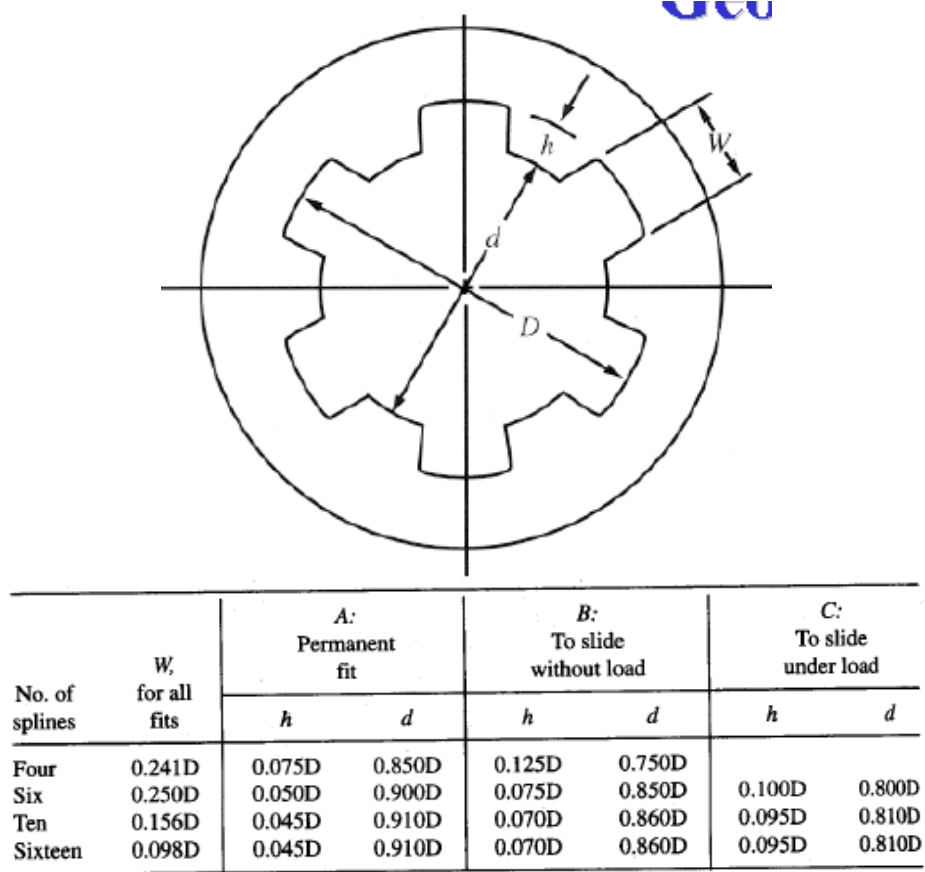


Figure 8.1 SAE Straight Tooth Splines Standard
 (<http://mech.sharif.ir/~durali/design/Shafting/details/Lecture%2020.pdf>)

In figure 8.1, the parameters of the selected permanent fit spline are:

$$\text{Splines number: } N = 6$$

$$\text{Inner diameter of spline: } d_5 = d_3 = 18\text{mm}$$

$$\text{Outer diameter of spline: } D = \frac{d}{0.9}$$

$$D_1 = \frac{d_5}{0.9} = \frac{18}{0.9}$$

$$\rightarrow D_1 = 20\text{mm}$$

$$\text{Spline width: } W = 0.25 \cdot D$$

$$W = 0.25 \times D_1 = 0.25 \times 20$$

$$\rightarrow W = 5\text{mm}$$

$$\text{Height of spline: } h = 0.05 \cdot D$$

$$h = 0.05 \times D_1 = 0.05 \times 20$$

$$\rightarrow h = 1\text{mm}$$

The spline is used to mount gear on the first stage output shaft. The gear is required to be fixed joint.

Equation (6.3) in page 103 of Machine Design about shear stress on fixed spline joint:

$$\frac{2T}{k \cdot z \cdot h \cdot l \cdot d_m} \leq [\sigma_p] \quad (1)$$

In this equation:

- k is coefficient of load with no equal apportionment between teeth; its value depends on making precision mainly, generally $k = 0.7 \sim 0.8$. Select $k = 0.75$.
- z is the number of spline tooth.

$$\rightarrow z = N = 6$$

- h is the working height of spline tooth, that $h = 1\text{mm}$.
- l is the working length of spline tooth. In this case it will be equal to the width of the gear.

$$\rightarrow l = b = 20\text{mm}$$

- d_m is the average diameter of spline.

$$d_m = \frac{D + d}{2} = \frac{D_1 + d_5}{2}$$

$$\rightarrow d_m = \frac{20 + 18}{2} = 19\text{mm}$$

- $[\sigma_p]$ is the allowable crushing stress on spline that can be chosen from table 6.2 in page 105 of Machine Design (reference figure 8.3). From the table 6.2 select the spline: withstand dead load, working condition is middling, surface with heat treatment.

$$\rightarrow [\sigma_p]_1 = 120\text{MPa}$$

- Torque on the first stage output shaft $T = 37950\text{N} \cdot \text{mm}$.

Plug those values into equation (1):

$$\frac{2T}{k \cdot z \cdot h \cdot l \cdot d_m} = \frac{2T}{k \cdot N \cdot h \cdot b \cdot d_m}$$

$$\rightarrow \frac{2 \times 37950}{0.75 \times 6 \times 1 \times 20 \times 19} = 44.4\text{MPa} < [\sigma_p]_1 = 120\text{MPa}$$

From the shear stress analysis, the selected spline is proved to be able to carry the torque and meet the safety requirement.

Check the strength of the key on first stage output shaft

Selected flat key on the first stage output shaft is used for eccentric bearing.

Equation (6.2) in page 101 of Machine Design for flat key is:

$$\frac{4 \cdot T}{d \cdot h \cdot l} \leq [\sigma_p] \quad (2)$$

- T is the torque on first stage output shaft $T = 37950 \text{ N} \cdot \text{mm}$.
- d is the shaft diameter. In this case it is the shaft diameter where mounted eccentric bearing.

$$\rightarrow d = D = 22 \text{ mm (From Appendix 5)}$$

- h is the thickness of key.

$$\rightarrow h = H = 6 \text{ mm}$$

- l is working length of the key. It is equal to the keyway length on first stage output shaft (in figure 8.2).

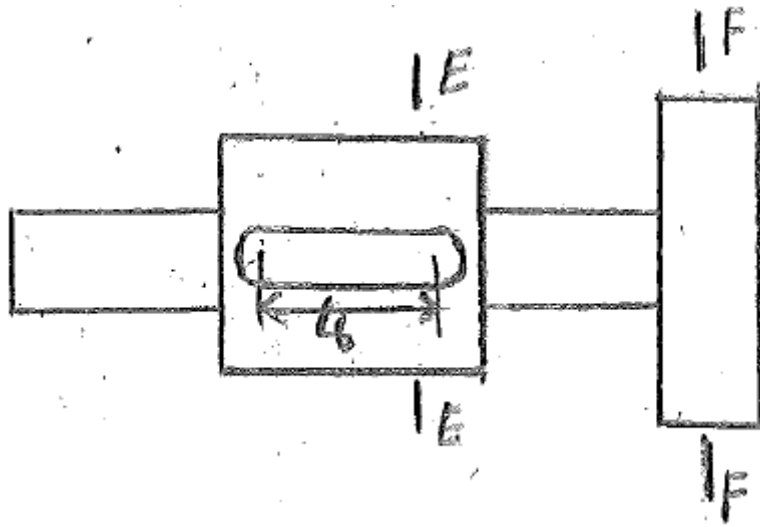


Figure 8.2 Length of key on the first stage output shaft

$$l = L_6 = 22 \text{ mm}$$

- $[\sigma_p]$ is the allowable crushing stress on flat key that can be chosen from table 6.1 in page 102 of Machine Design (reference figure 8.4). From table 6.1 select the key: fixed joint, material is steel, withstand light impact load.

$$\rightarrow [\sigma_p]_2 = 110 \text{ MPa}$$

Plug those values into equation (2):

$$\frac{4 \cdot T}{d \cdot h \cdot l} = \frac{4 \cdot T}{D \cdot H \cdot L_6}$$

$$\rightarrow \frac{4 \times 37950}{22 \times 6 \times 22} = 52.3 \text{ MPa} < [\sigma_p]_2 = 110 \text{ MPa}$$

From the shear stress analysis, the key on first stage output shaft is proved to be able to carry the torque and meet the safety requirement.

Check the strength of the key on input shaft

- The input torque $T_1 = 28600 \text{ N} \cdot \text{mm}$. (From Appendix 7)
- The input shaft diameter $d_6 = 20 \text{ mm}$. (From Appendix 7)
- Key thickness $H = 6 \text{ mm}$.
- Working length of the key $l = 22 \text{ mm}$.
- Allowable crushing stress $[\sigma_p]_2 = 110 \text{ MPa}$.

Plug those values into equation (2):

$$\frac{4 \cdot T}{d \cdot h \cdot l} = \frac{4 \cdot T_1}{d_6 \cdot H \cdot l}$$

$$\rightarrow \frac{4 \times 28600}{20 \times 6 \times 22} = 43.3 \text{ MPa} < [\sigma_p]_2 = 110 \text{ MPa}$$

From the shear stress analysis, the key on first stage input shaft is proved to be able to carry the torque and meet the safety requirement.

Reference

Table 6.2 Allowable crushing stress and allowable pressure of spline joints (MPa)

Working way of joint	Allowable value	Working condition	Surface without heat treatment	Surface with heat treatment
Dead load	$[\sigma_p]$	Badness	35~55	40~70
		Middling	60~100	100~140
		Well	80~120	120~200
Sliding joints without load	$[p]$	Badness	15~20	20~35
		Middling	20~30	30~60
		Well	25~40	40~70
Sliding joints with load	$[p]$	Badness	—	3~10
		Middling	—	5~15
		Well	—	10~20

Figure 8.3 Table 6.2 in page 105 of Machine Design

Table 6.1 Allowable crushing stress and allowable pressure of keys joints (MPa)

Working way of joint	Material of weaker element	Dead load	Light impact load	Impact load
$[\sigma_p]$ (Fixed joint)	Steel	125~150	100~120	60~90
	Cast iron	70~80	50~60	30~45
$[p]$ (Sliding joint)	Steel	50	40	30

Figure 8.4 Table 6.1 in page 102 of Machine Design

Appendix 9 Output shaft design

Symbol	Explanation	Value	Other
T_v	Output torque	6300Nm	Given
T_g	Torque on each cycloid gear	3150Nm	Calculated
σ_b	Tensile strength	923Mpa	Selected
σ_{FP}	Bending strength	396.89Mpa	Calculated
d_p'	Pin diameter	45mm	Selected
e	Eccentric distance	1.75mm	Selected
b_g	Thickness of cycloid gear	15mm	Selected
δ	Distance between two cycloid gear	0.000849	Calculated
Z_w	Pin hole number	3	Selected
R_w	Distributed circle of pin and bearing radius	86mm	Selected
d_p	Roller diameter	60mm	Selected
d_w	Pin hole diameter	63.5mm	Calculated
d_b	Outer ring diameter	53.5	From appendix 4
D_f	Dedendum diameter of cycloid gear	286.5	Calculated
L_1	Length of part 1 of shaft	50mm	Selected
L_2	Length of part 2 of shaft	100mm	Selected
L_3	Length of part 3 of shaft	20mm	Selected
L_4	Length of part 4 of shaft	24.5mm	Calculated
L_5	Length of part 5 of shaft	7.5mm	Calculated
Q_{max}	Maximum force on the pin	58605N	Calculated
R_B	Reaction force on bearing B	118628N	Calculated
R_A	Reaction force on bearing A	-60023N	Calculated
S_{max}	Maximum shear force	60023N	Calculated
M_{max}	Maximum bending moment	3721426N · m	Calculated
d	Diameter of part 1 of shaft	90mm	Selected
D	Diameter of part 2 of shaft	110mm	Selected
S_u	Ultimate strength	923MPa (134ksi)	From Appendix 5
S_y	Yield strength	750MPa	From Appendix 5
S_{ys}	Shear yield strength	435Mpa	From Appendix 5
K_{t1}	Static stress concentration factor on shaft (for bending, section B-B)	2.05	Figure 4.35(reference figure 9.12)
q_1	Notch sensitive factor of shaft part (for bending, section B-B)	0.89	Figure 8.24 (reference figure 9.13)
K_{f1}	Fatigue stress concentration factor of shaft part (for bending, section B-B)	1.93	Calculated
K_{t2}	Static stress concentration factor on shaft (for torsion, section B-B)	1.75	Figure 4.35(reference figure 9.14)
q_2	Notch sensitive factor of shaft part (for	0.94	Figure 8.24 (reference

	torsion, section B-B)		figure 9.15)
K_{f2}	Fatigue stress concentration factor of shaft part (for torsion, section B-B)	1.7	Calculated
S_n	Fatigue endurance limit	332MPa	Calculated
τ_a	Torsional alternating stress	0MPa	Calculated
τ_m	Torsional mean stress	201MPa	Calculated
$\sigma_{a,a}$	Axial alternating stress	0MPa	Calculated
$\sigma_{a,m}$	Axial mean stress	0MPa	Calculated
$\sigma_{b,m}$	Bending mean stress	0MPa	Calculated
$\sigma_{b,a}$	Bending alternating stress	270Mpa	Calculated
σ_{ea}	Equivalent alternating bending stress	270Mpa	Calculated
σ_{em}	Equivalent mean bending stress	201Mpa	Calculated
SF_1	Safety factor from section B-B	4.9	Calculated
K_{f3}	Fatigue stress concentration factor of shaft part (for torsion, section A-A)	1.7	Calculated
SF_2	Safety factor from section A-A	5.8	Calculated
SF	Final safety factor for whole shaft	4.9	Calculated

The output shaft of the whole RV reducer is connected with a disk. There are three pins on the disk to transmit the rotation from the second stage (cycloid gear). In this appendix, the first step is to determine the dimensions and on the pin of the disk; then start to design the output shaft by calculating according to fatigue analysis.

Calculate the pin diameter according to the bending stress:

The acting force on the pins is shown in figure 9.1

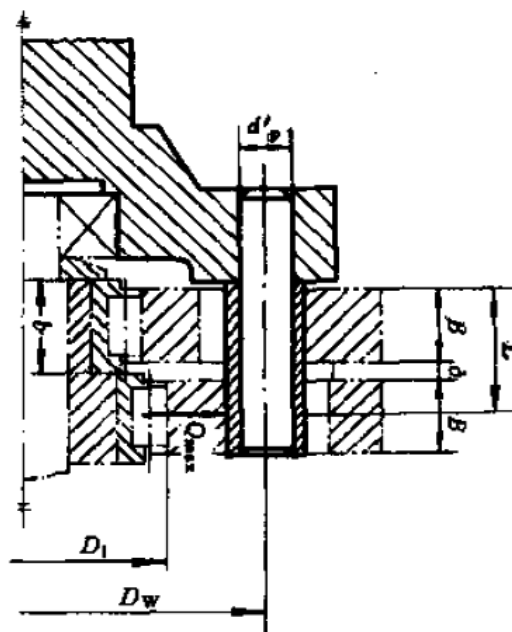


Figure 9.1 force acting on the pin (Rao, 1994)

The maximum bending stress is:

$$\sigma_F = \frac{Q_{\max} L}{W}$$

The maximum force on the pin:

$$Q_{\max} = \frac{4.8T_g}{Z_w R_w}$$

And then:

$$W = \frac{\pi d'^3}{32} = 0.1d'^3 \text{ (W is the section module)}$$

$$L = 1.5b_g + \delta$$

Because of $\sigma_F \leq \sigma_{FP}$, from those equations it can be simplified:

$$\sigma_F = \frac{4.89 * 10^4 T_g (1.5b_g + \delta)}{Z_w R_w d_p'^3} \leq \sigma_{FP}$$

$$\rightarrow d_p' \geq 36.6 * \sqrt[3]{\frac{T_g (1.5b_g + \delta)}{Z_w R_w \sigma_{FP}}}$$

$$T_g = \frac{T_v}{2} = \frac{6300}{2} = 3150 \text{ Nm}$$

$$\sigma_{FP} = 0.43\sigma_b = 0.43 * 923 = 396.89 \text{ Mpa}$$

R_w should be equal to the center distance of the first stage gear.

$$\rightarrow d_p' \geq 36.6 \times \sqrt[3]{\frac{T_g (1.5b_g + \delta)}{Z_w R_w \sigma_{FP}}} = 36.6 \times \sqrt[3]{\frac{3150(1.5 * 15 + 2)}{3 * 86 * 396.89}}$$

$$\rightarrow d_p' \geq 33.3 \text{ mm}$$

So from the reference diameter of the pins from reference figure 9.1, choose 45mm and then the roller diameter d_p is 60mm.

$$\rightarrow \begin{cases} d'_p = 45\text{mm} \\ d_p = 60\text{mm} \end{cases}$$

And now they also need to fulfill the geometry relation shown in reference figure 9.11

$$\frac{d_w}{2} + \frac{d_b}{2} < R_w$$

$$\frac{d_w}{2} + R_w < D_f$$

$$\frac{d_b}{2} + R_w < D_f$$

Calculate the output shaft

A simple picture of the output shaft with forces is shown in figure 9.2. Point A and B (figure 9.3) are mounted with two bearings which result in reaction forces. The detail of the dimensions:

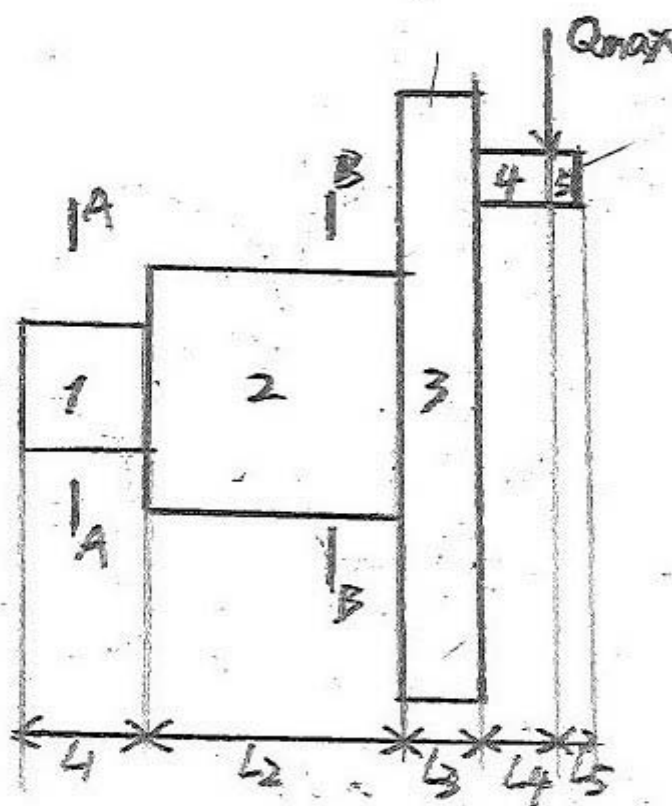


Figure 9.2 Simple picture of output shaft

$$L_1 = 50\text{mm}$$

$$L_2 = 100\text{mm}$$

$$L_3 = 20\text{mm}$$

L_4 is the distance from force Q_{\max} acted point to the disk. In reference figure 9.10, distance $B = 15\text{mm}$ and $\delta = 2\text{mm}$ are all given.

$$L_4 = B + \delta + \frac{B}{2} = 15 + 2 + \frac{15}{2}$$

$$\rightarrow L_4 = 24.5\text{mm}$$

$$L_5 = \frac{B}{2} = \frac{15}{2}$$

$$\rightarrow L_5 = 7.5\text{mm}$$

Find reaction forces

Figure 9.3 shows the forces on the shaft, according to the selected bearing of A and B (Appendix 10), the dimensions in Figure 9.3 are:

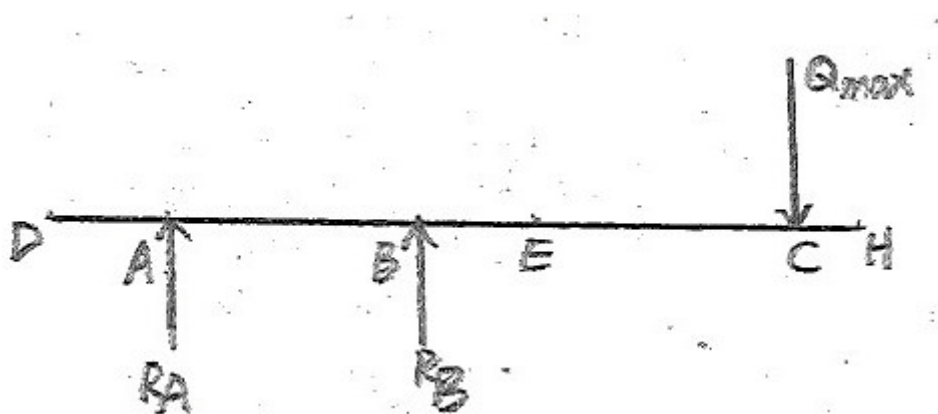


Figure 9.3 Forces on output shaft

$$DA = 19\text{mm}$$

$$AB = 62\text{mm}$$

$$BE = 19\text{mm}$$

$$EC = 44.5\text{mm}$$

$$CH = 7.5\text{mm}$$

The maximum force on the pin:

$$Q_{\max} = \frac{4.8T_g}{Z_w R_w} = \frac{4.8 \times 3150 \times 10^3}{3 \times 86}$$

$$\rightarrow Q_{\max} = 58605\text{N}$$

$$\uparrow: R_A + R_B - Q_{\max} = 0 \quad (1)$$

$$\rightarrow R_A + R_B = 58605$$

$$A: R_B \times 62 - Q_{\max} \times (44.5 + 19 + 62) = 0$$

$$\rightarrow R_B = 118628\text{N}$$

Plug $R_B = 118628\text{N}$ into equation (1)

$$\rightarrow R_A = -60023\text{N}$$

Calculation of shear forces and bending moments

Section AB: $0 \leq x \leq 62\text{mm}$ (see figure 9.4)

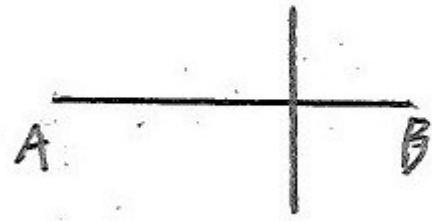


Figure 9.4 Section AB

$$\downarrow: S_{(x)} - R_A = 0$$

$$\rightarrow S_{(x)} = -60023\text{N}$$

$$AB: M_{(x)} - R_A \cdot x = 0$$

$$\rightarrow M_{(x)} = -60023 \cdot x$$

In this section when $x = 62\text{mm}$ (at point B) has the maximum moment:

$$M_B = -60023 \times 62 = -3721426\text{N} \cdot \text{mm}$$

Section BC: $62\text{mm} \leq x \leq 125.5\text{mm}$ (see figure 9.5)



Figure 9.5 Section BC

$$\downarrow: S_{(x)} - R_B - R_A = 0$$

$$\rightarrow S_{(x)} = 118628 + (-60023) = 58605\text{N}$$

$$\text{BE: } M_{(x)} - R_A \cdot x - R_B \cdot (x - 62) = 0$$

$$M_{(x)} + 60023 \cdot x - 118628 \cdot x + 118628 \times 62 = 0$$

$$\rightarrow M_{(x)} = 58605 \cdot x - 7354936$$

In this section when $x = 62\text{mm}$ (at point B) has the maximum moment:

$$M_D = 58605 \times 62 - 7354936 = -3721426\text{N} \cdot \text{mm}$$

The bending moment at point E is:

$$M_E = 58605 \times 81 - 7354936 = -2607931\text{N} \cdot \text{mm}$$

Shear force diagram and bending moment diagram

Figure 9.6 and 9.7 show shear force diagram and bending moment diagram respectively.

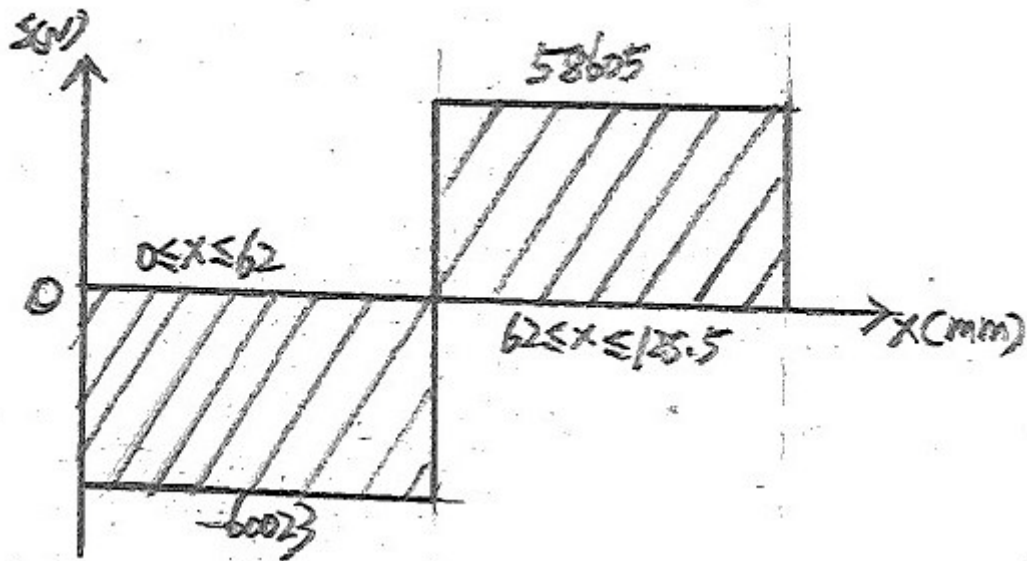


Figure 9.6 Shear force diagram

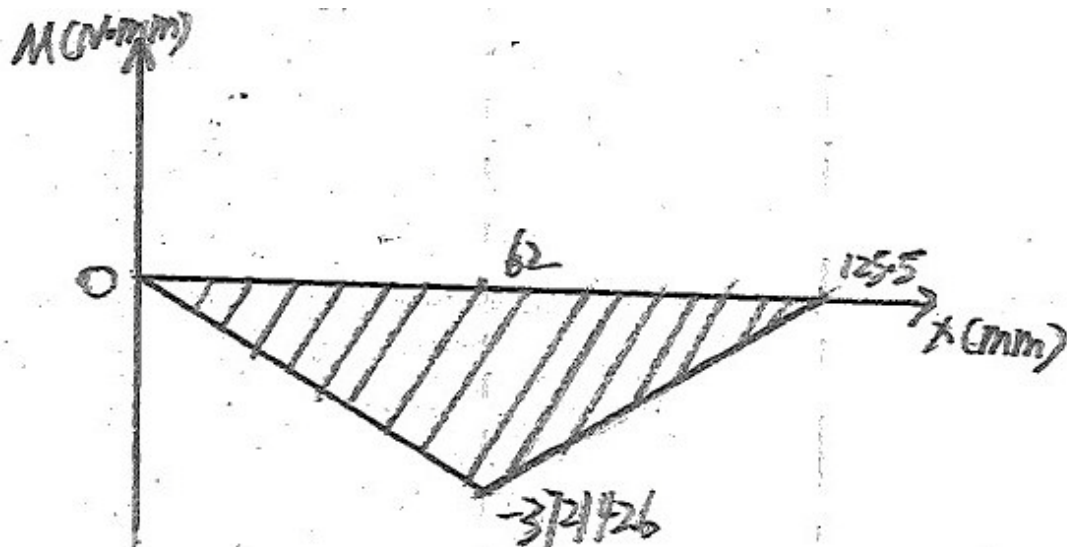


Figure 9.7 Bending moment diagram

From figure 9.6 and 9.7 the maximum shear force and bending moment can be found.

The maximum shear force:

$$S_{\max} = 60023 \text{ N (At section AB)}$$

The maximum bending moment:

$$M_{\max} = 3721426 \text{ N} \cdot \text{m (At section B-B)}$$

Torsion loading diagram

The torque on all parts of output shaft is the output torque.

$$T = T_v = 6300\text{N} \cdot \text{m}$$

$$\rightarrow T = 6300000\text{N} \cdot \text{mm}$$

Figure 9.8 displays the torsion loading diagram.

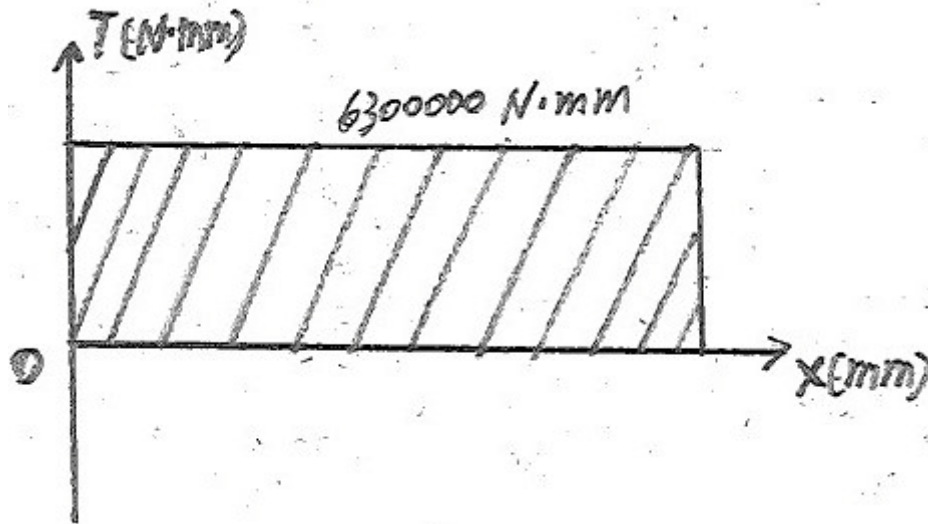


Figure 9.8 Torsion loading diagram

Bearing loads

As there are no axial forces on bearing, the bearing force is equal to the reaction force on it.

$$R_B = 118628\text{N}$$

$$R_A = -60023\text{N}$$

Find safety factor for the shaft

From the analysis above, there are two critical sections on the output shaft. They are section B-B with bending moment and torque on it; section A-A with torque on it. Figure 9.9 has shown these two sections. Diameters on section A-A, section B-B are selected as 90mm and 110mm respectively.

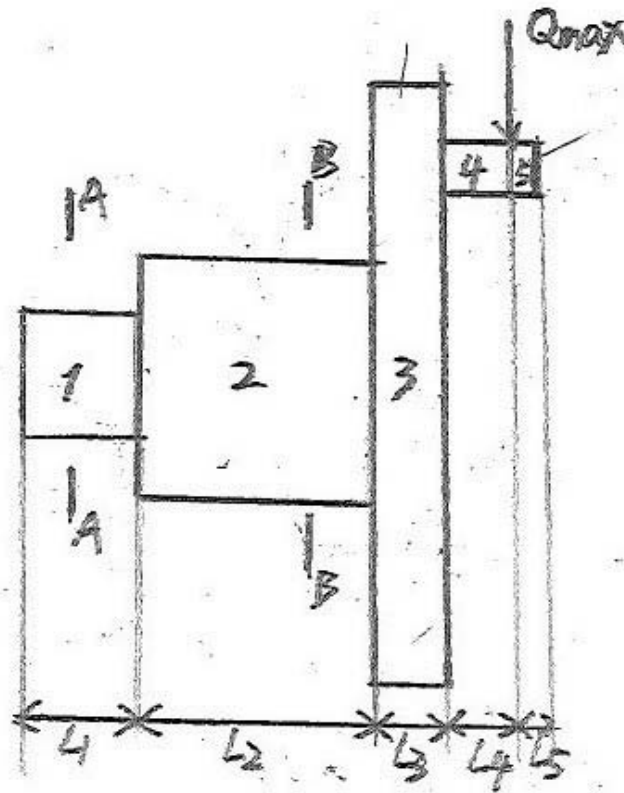


Figure 9.9 Section A-A and Section B-B in output shaft

$$\rightarrow \begin{cases} d = 90\text{mm} \\ D = 110\text{mm} \end{cases}$$

Find safety factor from section B-B

Firstly is to find the bending stress.

Select the fillet radius on the shaft is $r = 3\text{mm}$

$$\left\{ \begin{array}{l} \frac{D}{d} = \frac{110}{90} = 1.22 \\ \frac{r}{d} = \frac{3}{90} = 0.033 \\ \text{Bending} \end{array} \right. \text{Figure 4.35(a) in page 14 of Handbook for Machine Design}$$

$$\rightarrow K_{t1} = 2.05$$

$$\left\{ \begin{array}{l} \text{Figure 8.24 in page 28 of Handbook for Machine Design} \\ r = 3\text{mm} \\ S_u = 134\text{ksi} \\ \text{Bending} \end{array} \right.$$

$$\rightarrow q_1 = 0.89$$

Handbook for Machine Design (8.2) $K_f = 1 + (K_t - 1) \cdot q$

$$K_{f1} = 1 + (K_{t1} - 1) \cdot q_1 = 1 + (2.05 - 1) \times 0.89$$

$$\rightarrow K_{f1} = 1.93$$

The rotating static bending moment results in alternating stress on the shaft and the bending mean stress is zero.

$$\sigma_{b,m} = 0 \text{ MPa}$$

Handbook for Machine Design (8.6) $\frac{\sigma_a}{SF \cdot K_f} = \frac{32 \cdot M_a}{\pi \cdot d^3}$

$$\frac{\sigma_{b,a}}{SF \cdot K_{f2}} = \frac{32 \cdot M_{\max}}{\pi \cdot D^3}$$

$$\rightarrow \sigma_{b,a} = \frac{32 \times 3721426 \times 1.93}{\pi \times 110^3} \times SF = 55SF$$

Secondly is to find torsional stress.

$$\left\{ \begin{array}{l} \frac{D}{d} = \frac{110}{90} = 1.22 \\ \frac{r}{d} = \frac{3}{90} = 0.033 \\ \text{Torsion} \end{array} \right. \text{Figure 4.35(c) in page 14 of Handbook for Machine Design}$$

$$\rightarrow K_{t2} = 1.75$$

$$\left\{ \begin{array}{l} \text{Figure 8.24 in page 28 of Handbook for Machine Design} \\ r = 3 \text{ mm} \\ S_u = 134 \text{ ksi} \\ \text{Torsion} \end{array} \right.$$

$$\rightarrow q_2 = 0.94$$

Handbook for Machine Design (8.2) $K_f = 1 + (K_t - 1) \cdot q$

$$K_{f2} = 1 + (K_{t2} - 1) \cdot q_2 = 1 + (1.75 - 1) \times 0.94$$

$$\rightarrow K_{f2} = 1.7$$

Since the torque is constant, the torsion alternating stress $\tau_a = 0 \text{ MPa}$

Handbook for Machine Design (8.5) $\frac{\tau_m}{SF \cdot K_f} = \frac{16 \cdot T_m}{\pi \cdot d^3}$

$$\frac{\tau_m}{SF \cdot K_{f2}} = \frac{16 \cdot T}{\pi \cdot D^3}$$

$$\rightarrow \tau_m = \frac{16 \times 6300000 \times 1.7}{\pi \times 110^3} SF = 41 \cdot SF$$

The third step is to find equivalent bending stresses.

There is no axial load.

$$\sigma_{a,a} = \sigma_{a,m} = 0 \text{ MPa}$$

Equation (a) in page 25 of Handbook for Machine Design: $\sigma_{ea} = \sqrt{\sigma_a^2 + \tau_a^2}$

$$\sigma_a = \sqrt{\sigma_{a,a}^2 + \sigma_{b,a}^2} = \sqrt{0^2 + (55SF)^2}$$

$$\rightarrow \sigma_a = 55SF$$

$$\tau_a = 0 \text{ MPa}$$

$$\rightarrow \sigma_{ea} = \sqrt{0^2 + (55SF)^2} = 55SF$$

Equation (b) in page 25 of Handbook for Machine Design: $\sigma_{em} = \frac{\sigma_m}{2} + \sqrt{\tau_m^2 + \frac{\sigma_m^2}{2}}$

$$\sigma_m = \sqrt{\sigma_{a,m}^2 + \sigma_{b,m}^2} = \sqrt{0^2 + 0^2}$$

$$\sigma_m = 0 \text{ MPa}$$

$$\tau_m = 41SF$$

$$\sigma_{em} = 0 + \sqrt{(41SF)^2 + 0^2} = 41 \cdot SF$$

$$\rightarrow \frac{\sigma_{ea}}{\sigma_{em}} = \frac{55SF}{41SF} = 1.34$$

The next step is to find fatigue endurance limit

Table 8.1a (reference figure 9.15) in page 27 of Handbook for Machine Design, assume $N > 10^6$.

$$S_n = S'_n \cdot C_L \cdot C_G \cdot C_S \cdot C_T \cdot C_R$$

$$C_T = C_R = 1 \text{ (No information)}$$

$$S'_n = 0.5 \cdot S_u \text{ (No information, steel)}$$

$$\rightarrow S'_n = 0.5 \times 923 = 461.5 \text{ MPa}$$

$$C_L = 1 \text{ (Bending)}$$

$$C_G = 0.8 \text{ (Bending, } 100 < d < 150 \text{ mm)}$$

$$\left\{ \begin{array}{l} S_u = 134 \text{ ksi} \\ \text{Fine - ground} \\ \text{Figure 8.13 in page 26 of Handbook for Machine Design} \end{array} \right.$$

$$\rightarrow C_S = 0.9$$

$$\rightarrow S_n = 461.5 \times 1 \times 0.9 \times 0.8 \times 1 \times 1 = 332 \text{ MPa}$$

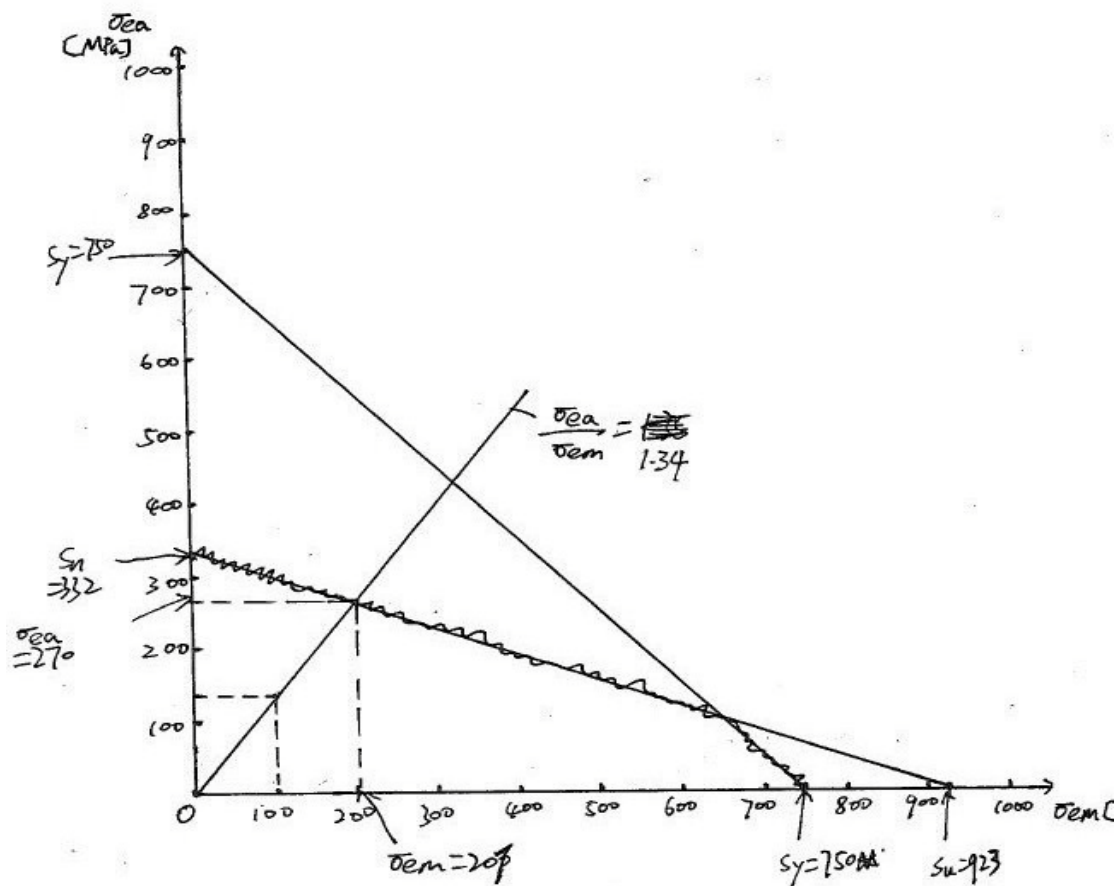


Diagram 9.1 Fatigue strength diagram

$$\left\{ \begin{array}{l} \frac{\sigma_{ea}}{\sigma_{em}} = \frac{55SF}{41SF} = 1.34 \\ \text{Diagram 9.1 (fatigue strength diagram)} \end{array} \right.$$

$$\rightarrow \sigma_{ea} = 270 \text{ MPa}$$

$$\sigma_{em} = 201 \text{ MPa}$$

$$\rightarrow SF_1 = \frac{270}{55} = 4.9$$

Find safety factor from section A-A

Section A-A only has torsional stress, the diameter of this shaft part is $d = 60 \text{ mm}$.

The fillet radius and torque are the same with section B-B.

$$\rightarrow K_{f3} = K_{f2} = 1.7$$

Handbook for Machine Design (8.5) $\frac{\tau_m}{SF \cdot K_f} = \frac{16 \cdot T_m}{\pi \cdot d^3}$

$$\frac{\tau_m}{SF \cdot K_{f3}} = \frac{16 \cdot T}{\pi \cdot d^3}$$

$$\rightarrow \tau_{m1} = \frac{16 \times 6300000 \times 1.7}{\pi \times 90^3} SF = 75 \cdot SF$$

Let:

$$\tau_{m1} = S_{ys} = 435 \text{ MPa}$$

$$\rightarrow SF_2 = \frac{435}{75} = 5.8$$

Select the final safety factor

Compare two generated safety factor

$$SF_2 = 5.8 > SF_1 = 4.9$$

Select the final safety factor is:

$$SF = SF_1 = 4.9$$

Reference

柱销直径 d_p	12	14	17	22	26	32	35	45	55
柱销套直径 d_s	17	20	26	32	38	45	50	60	75

Figure 9.10 reference pin diameter (行星传动机构设计)

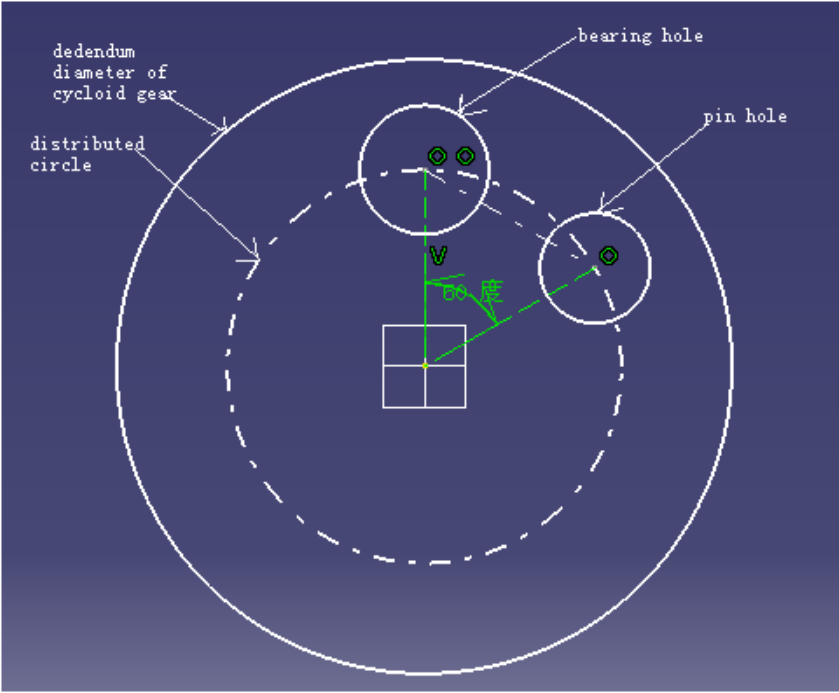
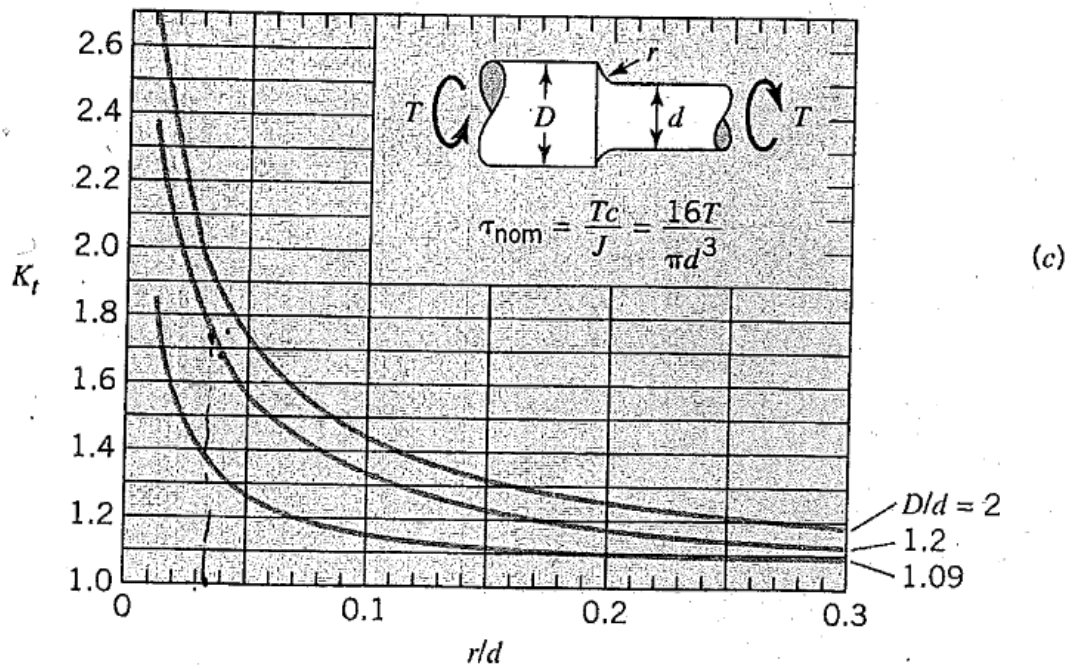


Figure 9.11 geometry relations between the pin holes and bearing hole

Figure9.12(refer to figure 5.24in appendix 5 page14 of Handbook for Machine Design)

Figure9.13(refer to figure 5,23 in appendix 5 Notch sensitivity curves)

**FIGURE 4.35**

Shaft with fillet (a) bending; (b) axial load; (c) torsion [7].

Figure 9.14figure4.3c in page 14 of Handbook for Machine Design

Figure 9.15 (refer to figure 5.25in appendix 5) Generalized fatigue strength factors for ductile materials

Figure 9.16 (refer to figure 5.26 in appendix 5) Reduction in endurance limit owing to surface finish-steel parts

Appendix 10 Calculation of bearings on output shaft

Symbol	Explanation	Value	Other
R_B	Reaction force on bearing B	118628N	From Appendix 9
R_A	Reaction force on bearing A	60023N	From Appendix 9
P	Equivalent dynamic bearing load	118628N	Calculated
a_1	Life adjustment factor for reliability	0.62	Table 1 (page 53 of SKF General Catalog)
p	Exponent of the life equation	3	Given
C	Basic dynamic bearing load	151kN	Page 346 of SKF General Catalog
n	Bearing rotational speed	4.3rpm	Given
d	Minimum bearing diameter	110mm	Page 346 of SKF General Catalog
D	Maximum bearing diameter	200mm	Page 346 of SKF General Catalog
d_m	Mean bearing diameter	155mm	Calculated
v_1	Required viscosity	1000 mm ² /s	Diagram 5 in page 60 of SKF General Catalog
v	Actual operating viscosity	1000 mm ² /s	Table 2 in page 246 of SKF General Catalog
k	Viscosity ratio	1.0	Calculated
η_c	Degree of contamination	0.9	Table 4 in page 62 of SKF General Catalog
P_u	Fatigue load limit	4kN	Page 346 of SKF General Catalog
a_{SKF}	SKF life modification factor	0.9	Diagram 1 in page 54 of SKF General Catalog
L_{2mh}	SKF rating life at 95% reliability	4461h	Calculated

Loads on bearings

Two sealed single row deep groove ball bearings (bearing A and B) have been mounted on the output shaft of RV reducer. In Appendix 9 loads act on them have been calculated. Only radial forces are acted on those two bearings, the axial forces are zero.

Radial load on bearing A: $R_A = 60023\text{N}$

Radial load on bearing B: $R_B = 118628\text{N} > R_A = 60023\text{N}$

The magnitude of force on bearing B is bigger than bearing A. Since bearing A and B is selected to the same type of standard sealed single row deep groove ball bearing on SKF catalog the calculation can only treat bearing B.

The equivalent dynamic bearing load is $P = R_B = 118628\text{N}$.

Find bearing type in SKF General Catalog

The bearing type is chosen in sealed single row deep groove ball bearing of SKF General Catalog. According to its diameters, the type is selected as: ***6222-2Z** in page 346 of SKF General Catalog.

Since bearings are chosen in the SKF General Catalog the calculation process and equations must follow the process given by SKF.

Find SKF rating life

Equation in page 52 of SKF General Catalog:

$$L_{nmh} = a_1 \cdot a_{SKF} \cdot \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P}\right)^p$$

Symbol L_{nmh} is the SKF rating life at $100 - n\%$. This n in the symbol represent the failure probability of the bearing. The total reliability of bearings on one shaft is selected as 90%. As the number of bearings is 2, the reliability of one tapered roller bearing is:

$$\text{Reliability} \approx \sqrt{90\%} = 95\%$$

Select the reliability of this bearing is 95%. The failure probability is 2% that means the SKF rating life can be written as L_{5mh} .

The exponent of the life equation is $p = 3$ for ball bearings.

- **Find life adjustment factor for reliability a_1**

$$\begin{cases} \text{Reliability} = 95\% \\ \text{Table 1 in page 53 of SKF General Catalog} \end{cases}$$

$$\rightarrow a_1 = 0.62$$

- **Find SKF life modification factor**

From the bearing type *6222-2Z and the SKF General Catalog:

$$d = 110\text{mm}$$

$$D = 200\text{mm}$$

The mean diameter is:

$$d_m = \frac{d + D}{2} = \frac{110 + 200}{2}$$

$$\rightarrow d_m = 155\text{mm}$$

$$\left\{ \begin{array}{l} \text{Diagram 5 in page 60 of SKF General Catalog} \\ d_m = 155\text{mm} \\ n = 4.3\text{rpm} \end{array} \right.$$

$$\rightarrow v_1 = 1000 \text{ mm}^2/\text{s}$$

Since the bearing will withstand a huge load with a low rotational speed, select extreme high viscosity with solid lubrications in table 2.

Table 2 in page 246 of SKF General Catalog

$$\rightarrow v = 1000 \text{ mm}^2/\text{s} \text{ (At } 40^\circ\text{C, operating temperature)}$$

The viscosity ratio is:

$$k = \frac{v}{v_1} = \frac{1000}{1000}$$

$$\rightarrow k = 1.0$$

$$\left\{ \begin{array}{l} \text{Table 4 in page 62 of SKF General Catalog} \\ \text{High cleanliness} \\ d_m = 155\text{mm} > 100\text{mm} \end{array} \right.$$

$$\rightarrow \eta_c = 0.9$$

- **Find basic dynamic load rating and rotational speed**

From the bearing type *6222-2Z and the SKF General Catalog, the basic dynamic load rating is:

$$C = 151\text{kN}$$

The rotational speed of the bearing is equal to the given rotational speed of RV reducer output shaft:

$$n = 4.3 \text{ rpm}$$

From the bearing type ***6222-2Z** and the SKF General Catalog, the fatigue load limit is:

$$P_u = 4 \text{ kN}$$

$$\eta_c \cdot \frac{P_u}{p} = 0.9 \times \frac{4000}{118628}$$

$$\rightarrow \eta_c \cdot \frac{P_u}{p} = 0.03$$

$$\left\{ \begin{array}{l} \text{Diagram 1 in page 54 of SKF General Catalog} \\ \eta_c \cdot \frac{P_u}{p} = 0.03 \\ k = 1.0 \end{array} \right.$$

$$\rightarrow a_{\text{SKF}} = 0.9$$

- **Find SKF rating life**

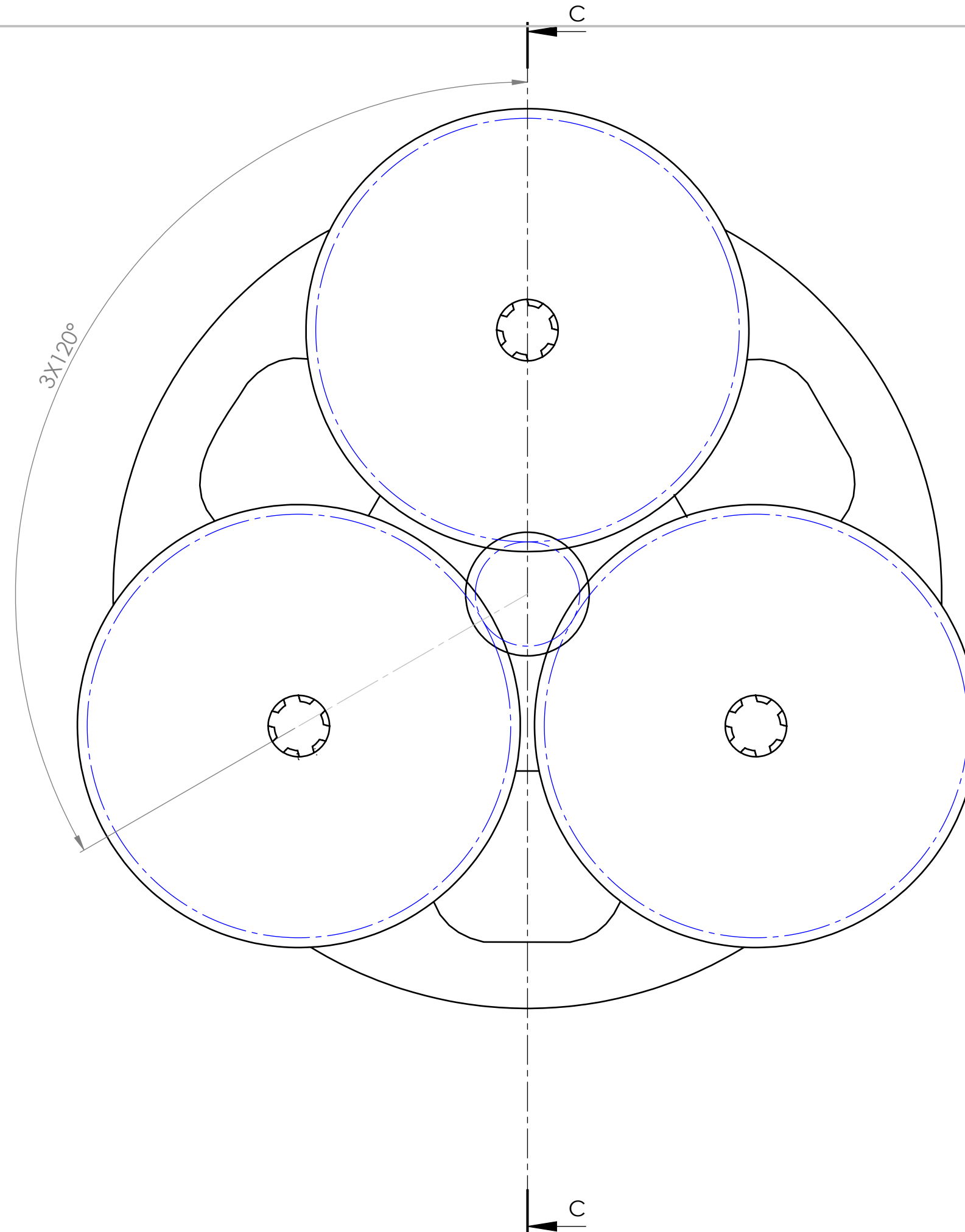
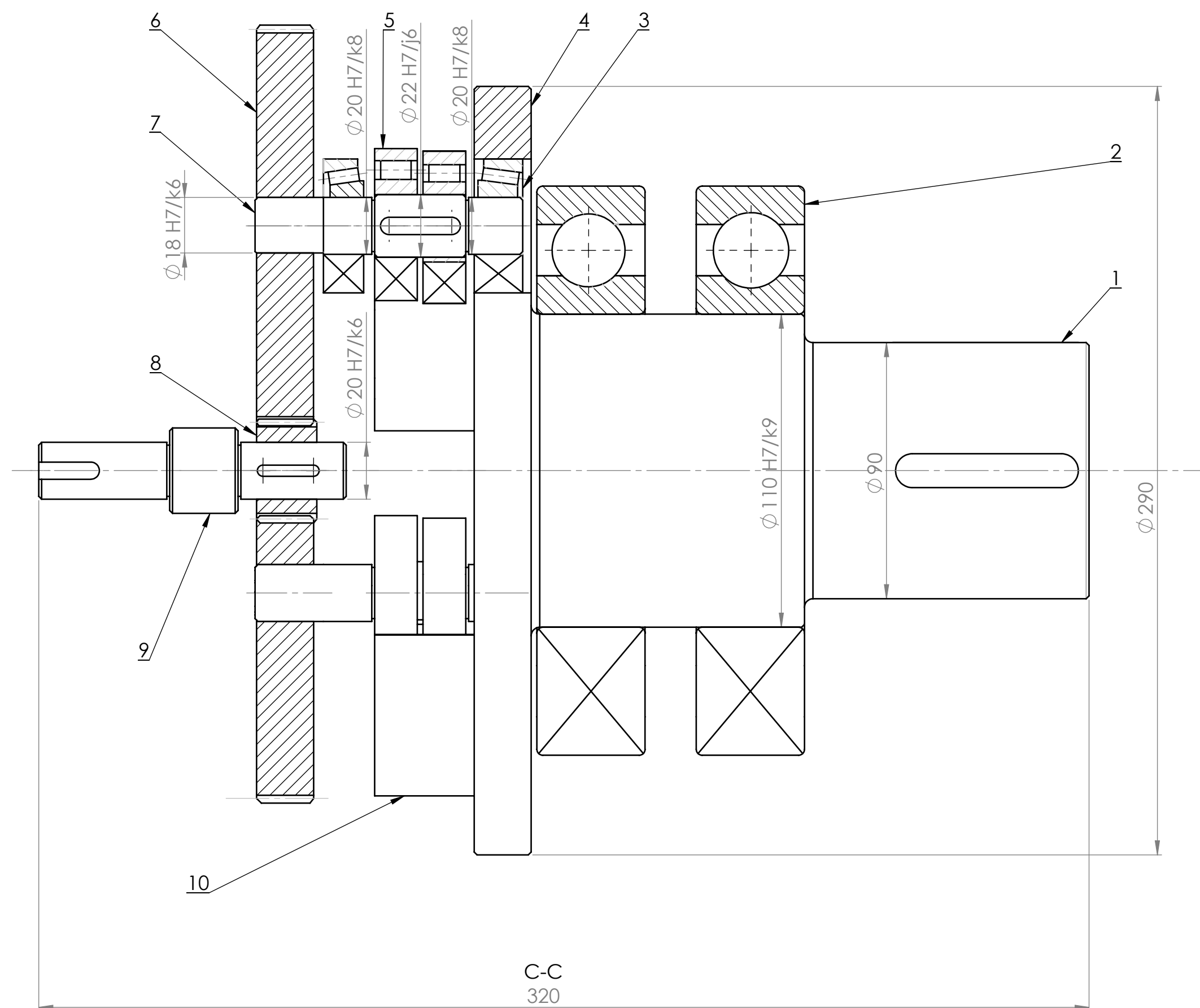
$$L_{2\text{mh}} = a_1 \cdot a_{\text{SKF}} \cdot \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P} \right)^p = 0.32 \times 0.9 \times \frac{10^6}{60 \times 4.3} \times \left(\frac{151 \times 10^3}{118628} \right)^3$$

$$\rightarrow L_{2\text{mh}} = 4461 \text{ h}$$

Reference

SKF General Catalogue

Appendix 11 Drawings



10	Roller pin	3	6	
9	Input shaft	1	2	
8	Sun gear	1	3	
7	Crankshaft	3	5	
6	Planetary gear	3	4	
5	Eccentric bearing	6		180752904
4	Support disk	1	6	
3	Tapped roller bearing	6		30204J2/Q
2	Deep groove bearing	2		*6222-2Z
1	Output shaft	1	6	
Number	Component	Amount	Drawing number	Other

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

NAME

SIGNATURE

DATE

DRAWN

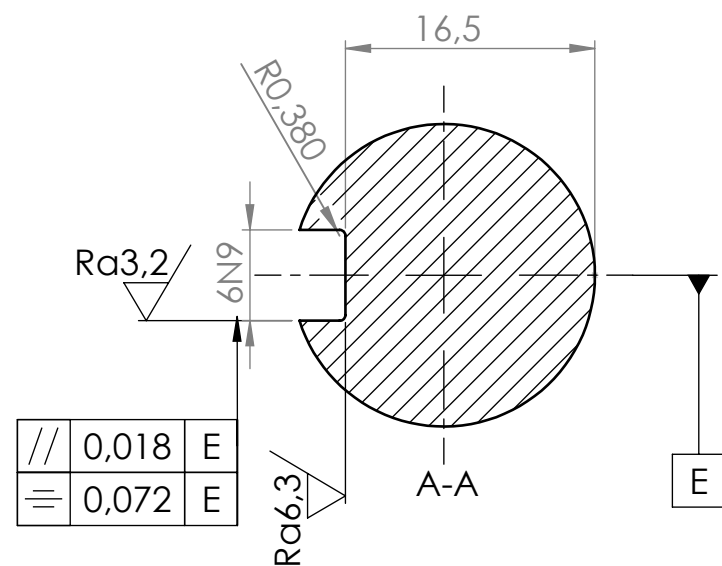
CHKD

APP'VD

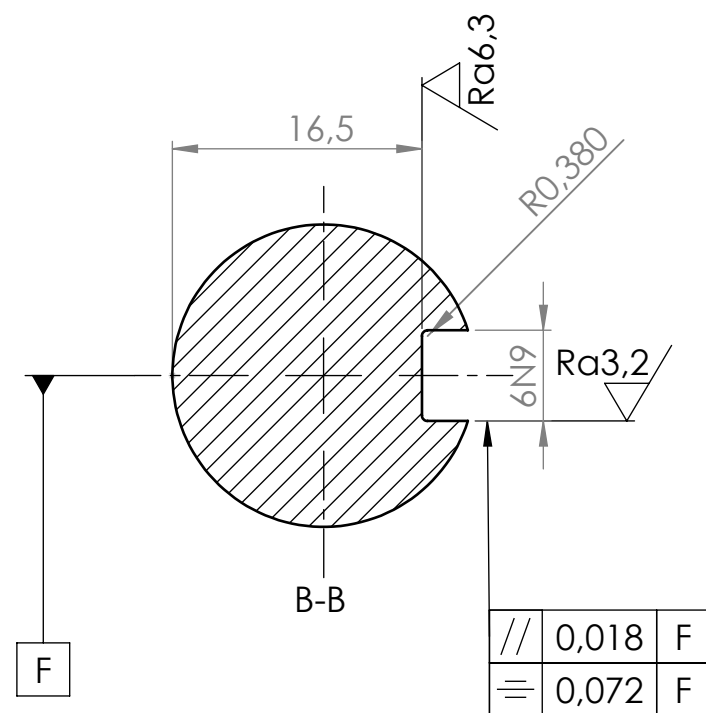
MFG

Q.A

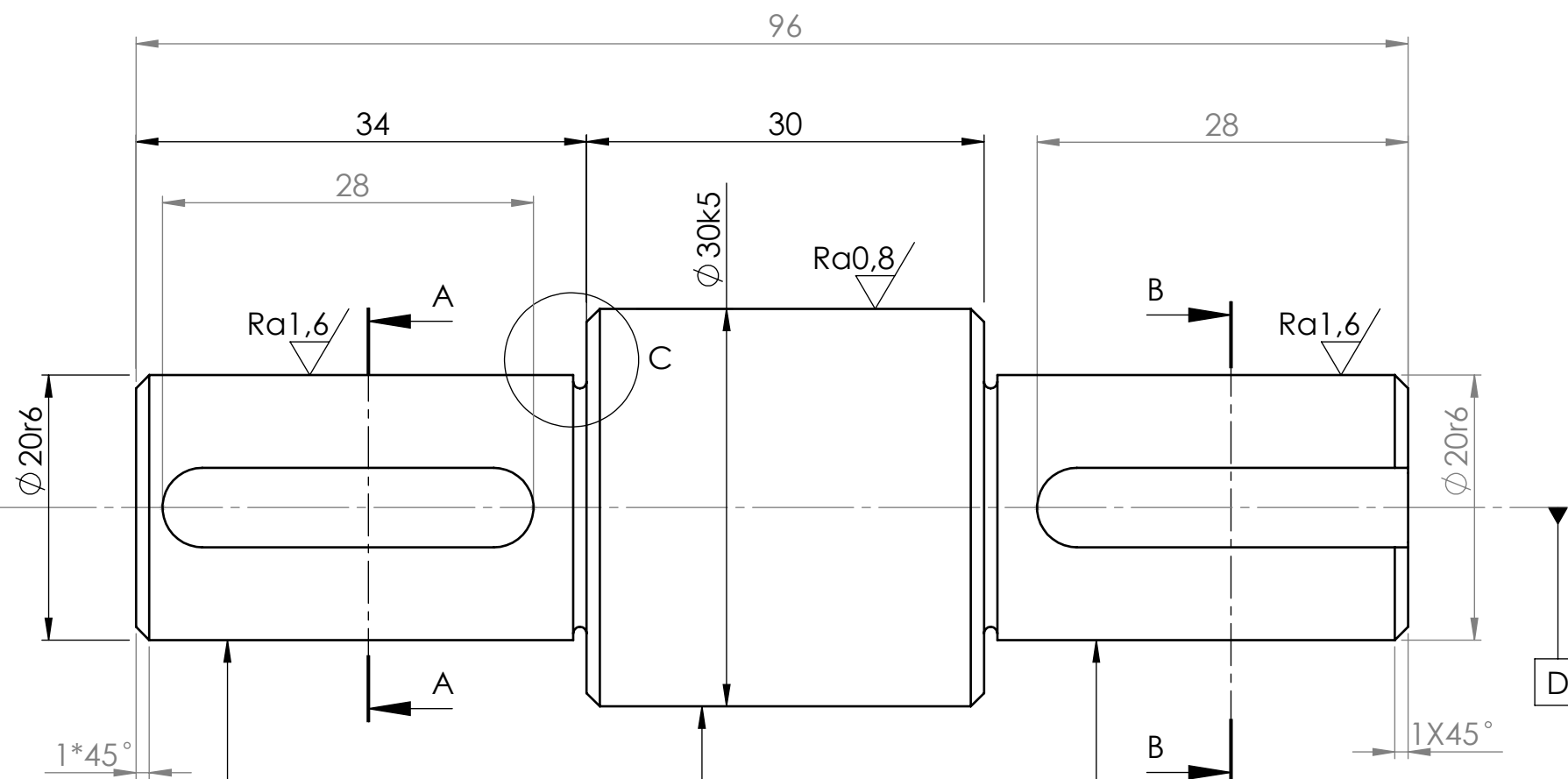
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=	0,072	E



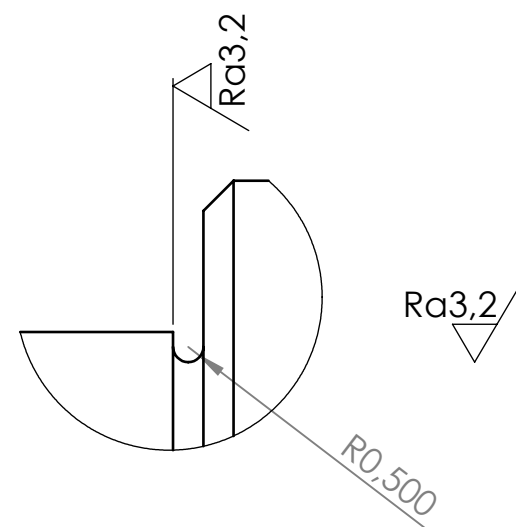
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⊙	φ 0,03	D
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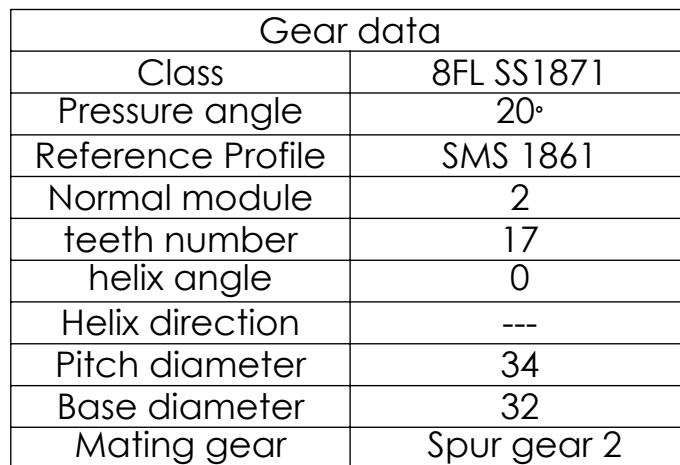
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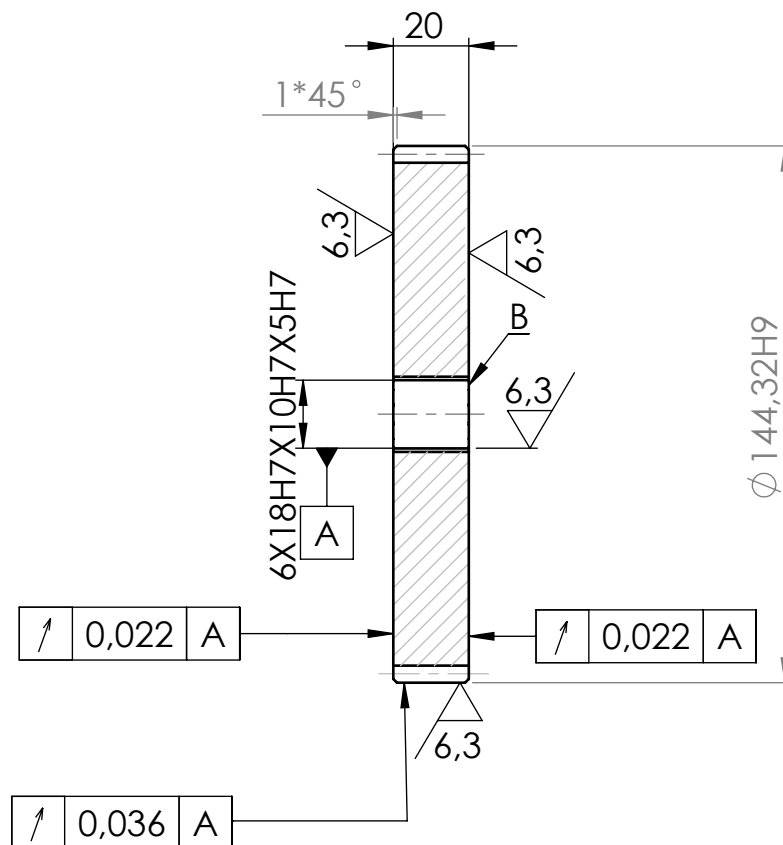
C (4 : 1)

All the chamfers' radius is 1mm

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES:ISO2768-mH LINEAR: ANGULAR:					FINISH: <div>6.3/</div>		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION			
		NAME		SIGNATURE		DATE				TITLE: Ingoing shaft				
DRAWN														
CHK'D														
APPV'D														
MFG														
Q.A										DWG NO. 2			A3	
										SCALE:2:1			SHEET 2 OF 6	



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES:ISO2768-mH LINEAR: ANGULAR:						FINISH: <div>6,3 ▽</div>		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
		NAME		SIGNATURE		DATE						TITLE:	
DRAWN												Spur gear 1	
CHK'D													
APPV'D													
MFG													
Q.A													
								MATERIAL: 20MnCr6		DWG NO. 3		A4	
								WEIGHT:		SCALE:2:1		SHEET 3 OF 6	



Gear data	
Class	8FL SS18
Pressure angle	20°
Reference profile	SMS 18
Normal module	2
Teeth number	69
Helix angle	0
Helix direction	----
Pitch diameter	138
Base diamter	130
Mating gear	spur gear 1

UNLESS OTHERWISE SPECIFIED:
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SURFACE FINISH:
TOLERANCES: ISO2768-mH
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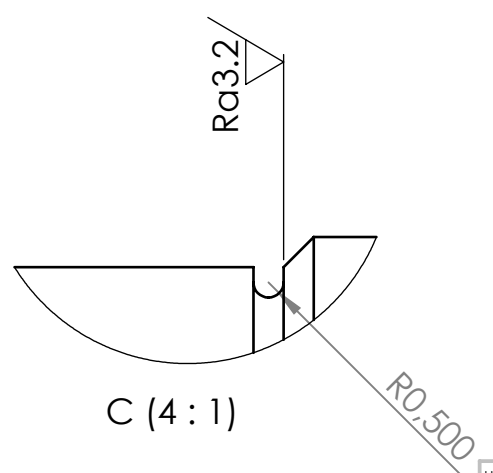
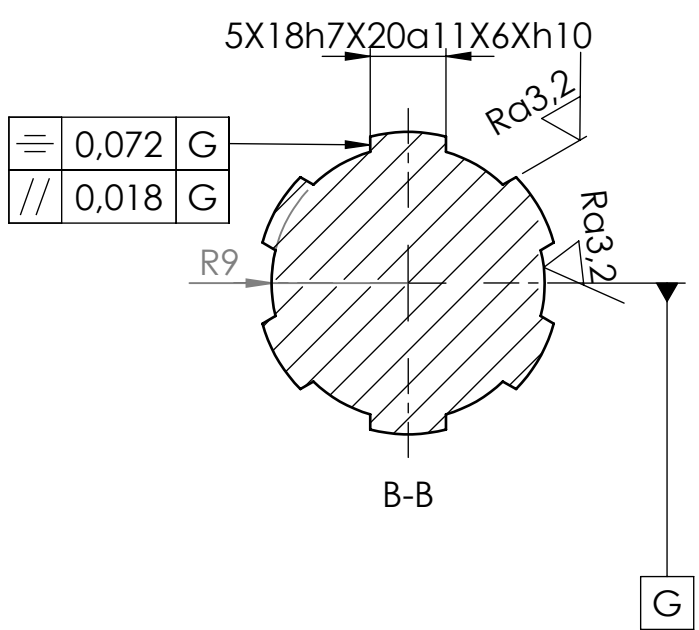
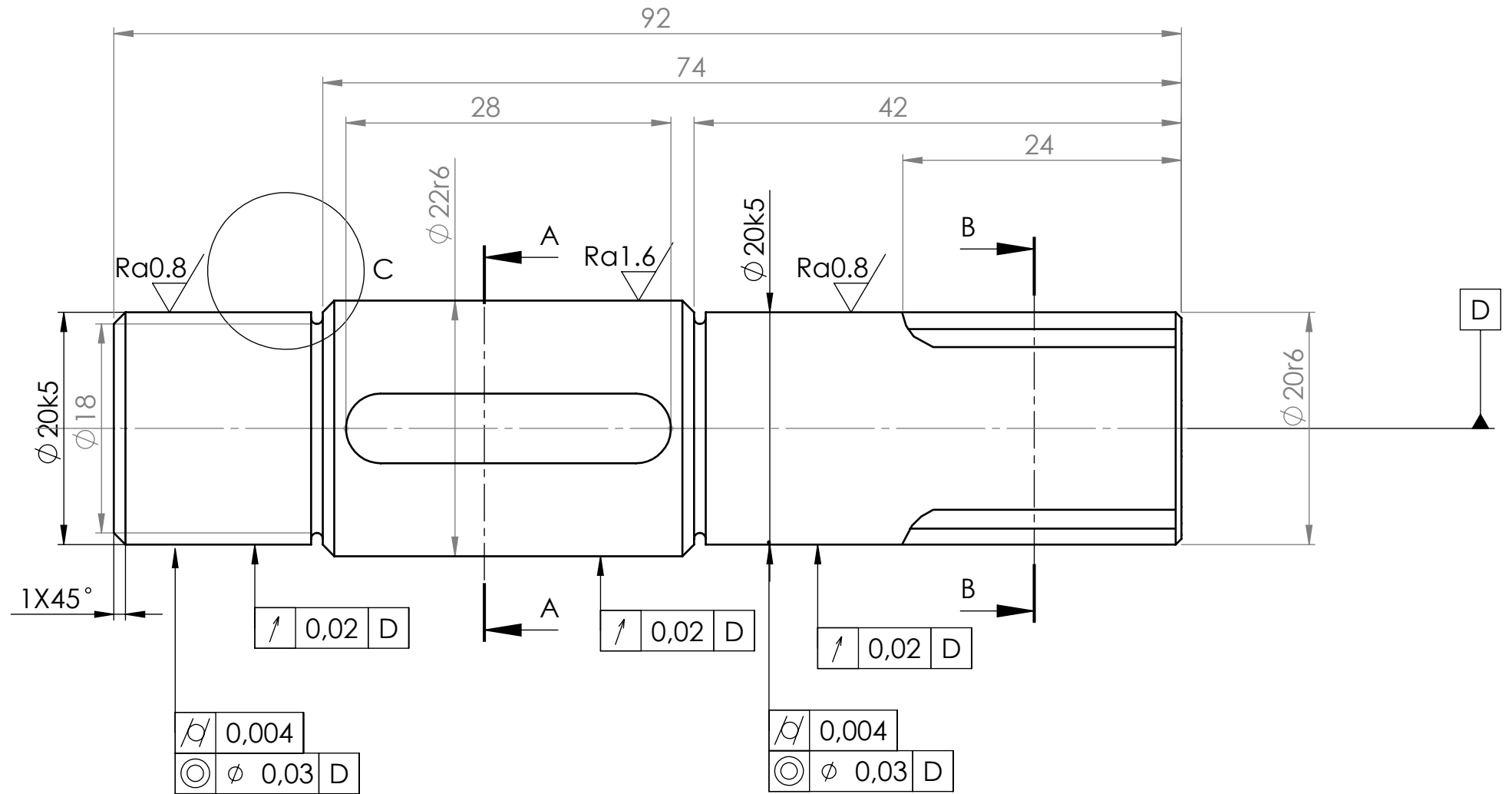
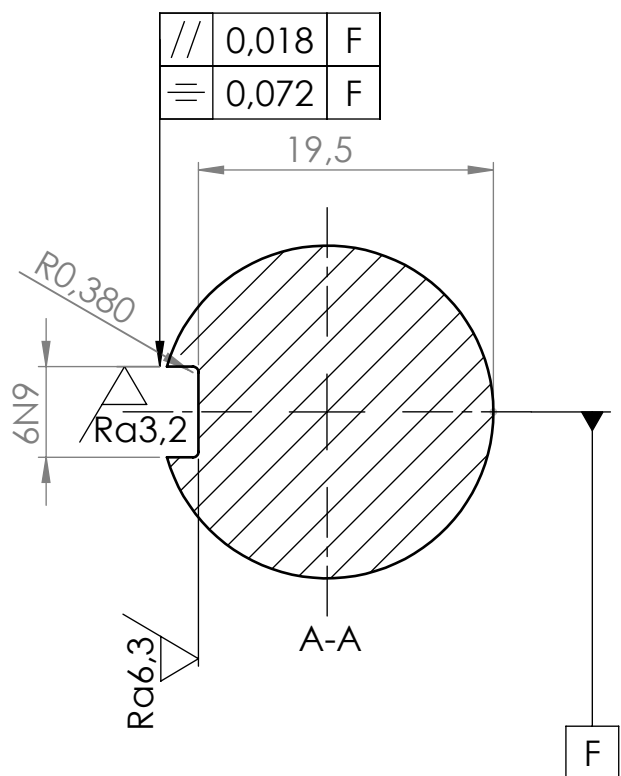
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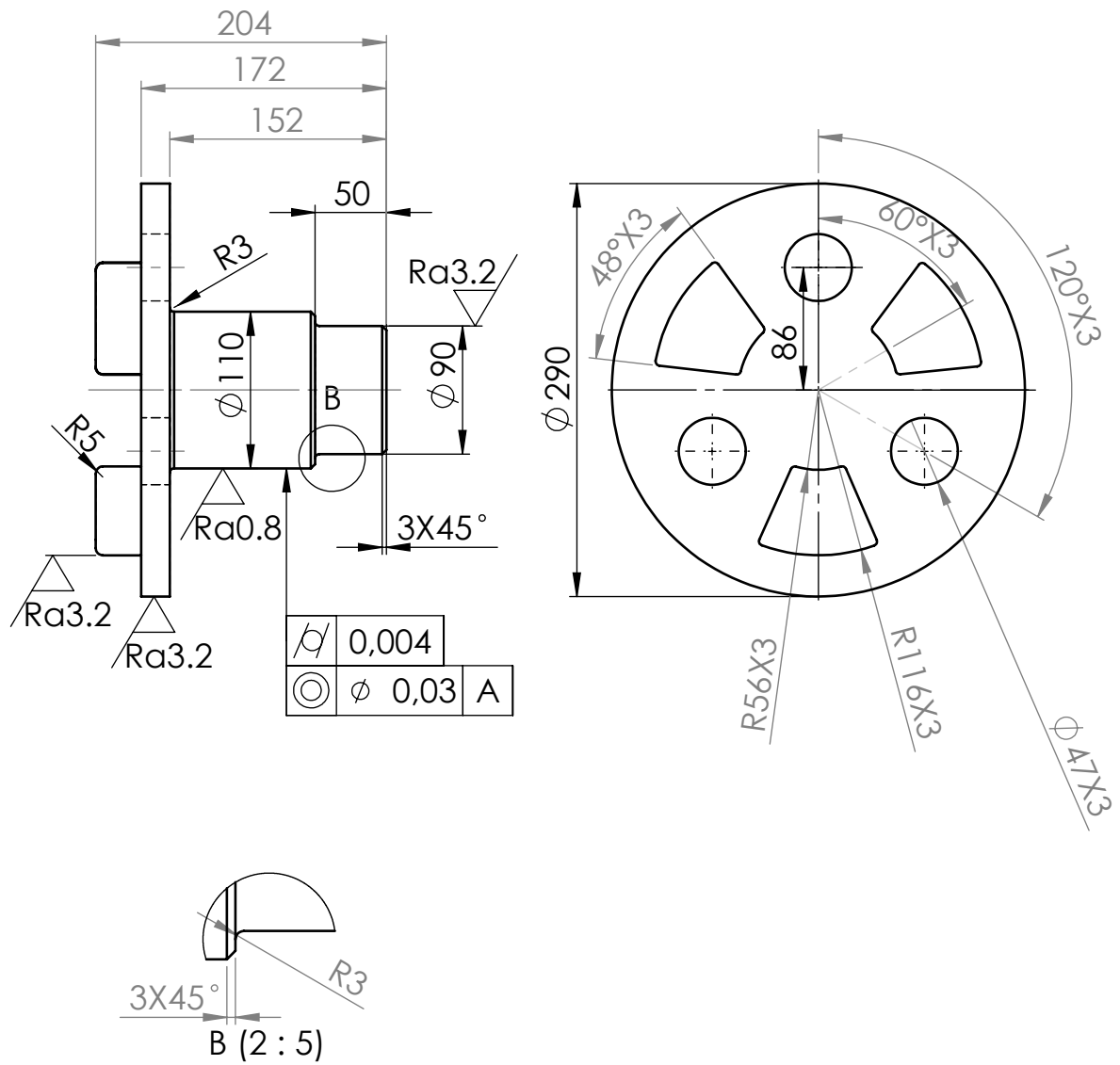
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DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A					
			MATERIAL: 20MnCr6		
			WEIGHT:		

TITLE:		Spur gear 2	
DWG NO.		4	A4
SCALE:1:2		SHEET 4 OF 6	



All the chamfers' radius is 1mm

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		NAME		SIGNATURE		DATE				TITLE: Crankshaft			
DRAWN													
CHK'D													
APPV'D													
MFG													
Q.A										DWG NO. 5			A3
										SCALE:2:1			SHEET 5 OF 6
								WEIGHT:					



All the chamfers' radius is 3mm

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES: ISO2768-mH
LINEAR:
ANGULAR:

FINISH:

6.3

DEBUR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

NAME	SIGNATURE	DATE		
DRAWN				
CHK'D				
APPV'D				
MFG				
Q.A				
			MATERIAL:	20MnCr6
			WEIGHT:	

TITLE:		Outgoing shaft	
DWG NO.		6	A4
SCALE:1:5		SHEET 6 OF 6	