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Tesi di Laurea Magistrale

Parametric Design of Automotive and Industrial Gearbox

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Abstract

The thesis proposed here is intended to carry out a parametric design of mechanical transmissions. The mechanical transmissions can be mainly defined in two categories:

• Automotive transmissions

Parametrically designed automotive transmission involves a 5-speed manual gearbox plus reverse-gear suitable for a front-wheel drive vehicle with a transverse-mounted front-mounted engine.

• Industrial transmissions

The parametrically designed industrial transmissions cover six possible configurations of an industrial speed reducer. The implemented configurations foresee a maximum number of reduction stages equal to four; they are described and explained more accurately in the foreseen section.

The whole procedure starts with some primary input such as torque, gear ratios, rotational speed, etc. After carrying the calculation through a MATLAB code, the final parameters are given to the NX software via an excel file.

In the NX software, every parameter from the input will be related to its proper expression. Then with a tool called PTS which stands for Product Template Studio, the possibility of altering the values is provided.

Acknowledgment

Firstly, I would like to thank my thesis advisor Professor Carlo Rosso from the mechanical engineering department at Politecnico di Torino. The door to Professor Rosso's office was always open whenever I ran into a trouble spot or had a question about my research or writing.

I would like to thank the expert who was involved in the validation of this thesis: Senior CAD Consultant, Mr. Domenico Giannetto Without his passionate participation and input, the thesis could not have been successfully conducted.

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Finally, I must express my very profound gratitude to my parents, family, and friends for providing me with unfailing support and continuous encouragement throughout my years of study and through the process of researching and writing this thesis. This accomplishment would not have been possible without them. Thank you.

Author

Mohammad Javad Amouei

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List of symbols

P_t	Transversal pitch
P_n	Normal Pitch
P_x	Axial pitch
m_t	Transversal module
m_n	Normal module
α_n	Normal pressure angle
α_t	Transversal pressure angle
z	Number of teeth
d_p	Pitch diameter
d_a	Addendum diameter
d_f	Root diameter
h_a	Addendum
h_f	Dedendum
h_t	Height tooth
τ	Transmission ratio
F_t	Tangential force
F_a	Axial force
F_r	Radial force
T	Torque
D_{ei}	The external outer diameter of the spline
D_{ii}	The external inner diameter of the spline
D_{ie}	The internal inner diameter of the spline
D_{ee}	The internal outer diameter of the spline
R_{p02}	Yield stress
ω	Rotational speed

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Introduction

In today's competitive world, time plays a crucial role in the mechanical product design market. Therefore, less time spends on designing the product, make it more valuable. Computers make the possibility to build a software prototype instead of the real one. These models exist only within computer's memory. Therefore, these models can be subjected to computer-based simulations for the principal analysis, and the results can be used in order to build a real prototype. The significant benefits of verifying the design within the computer are high speed, low cost, and more flexibility. Moreover, computer-based simulations can often make better representations of real-world conditions than those to which physical prototypes are subjected.

In recent years, the use of CAD software became so prevalent over the world and enterprises, depending on their product, prefer CAD software instead of using pencil, eraser, drafter, and sheets. Nowadays, the new concept is that industries start using more customized CAD software for their products. In order to sustain in this globally competitive world, it is mandatory for industries to introduce innovative products or the ability to change the existing products as quick as possible. Generally, in the whole product development cycle, 80% of the time spent in the design process. Therefore, a considerable amount of time can be saved in the design process. Design stage has a repetitive procedure such as modeling process, design. This repetitive task can be captured and standardized. Parameterization technique is suitable for this work. Gearbox is widely used for the transmission purpose because of the high efficiency of transmission, compactness, reliability. Globally a lot of gearbox manufacturer build the product for their customers according to the specific types and application. Each specification of gearbox assembly has its own modeling approach. Parametric modeling can be used for different configuration of the same assembly.

For the parametric sizing stage, a code has been implemented on the MATLAB Software. For the parametric modeling phase, the results from the MATLAB code was initially extracted and organized into Excel spreadsheets. From Excel sheets, each value will be connected to its proper expression. So, the NX Software imports the results and performs parametric modeling of the designed mechanical transmissions.

The components of the mechanical transmission were implemented in the NX Software in a parametric way. In other words, the parameters for dimensioning a single part of the assembly are linked to the values calculated during the sizing step performed with the MATLAB code. To achieve this parametric modeling, the dimensions were parameterized using the expression window in order to insert the parametric formulas. Furthermore, further dialogs have been implemented through the PTS (Product Template Studio) Tool. The PTS has given the possibility to introduce multiple selection menus that allow to manually change the characteristic parameters of the mechanical transmission (for example construction parameters of the gears).

1. Gear

The first part of the project and the starting point is Gears. In order to transmit different torque with the different speed, The gear ratios should be defined, and the gears will be designed compatible with dimension constraints.

1.1. Basic Geometry

1.1.1. Spur Gears

The simplest type of gears is spur gears hence the illustration of the fundamental of the gear meshing can be represented by the spur gears. The spur gears are widely used for almost every application.

The meshing of the spur gears is represented in figure 1.2.

There are some critical parameters that should be defined firstly.

1. Center distance

2. Modules

3. Pressure angle

4. number of teeth

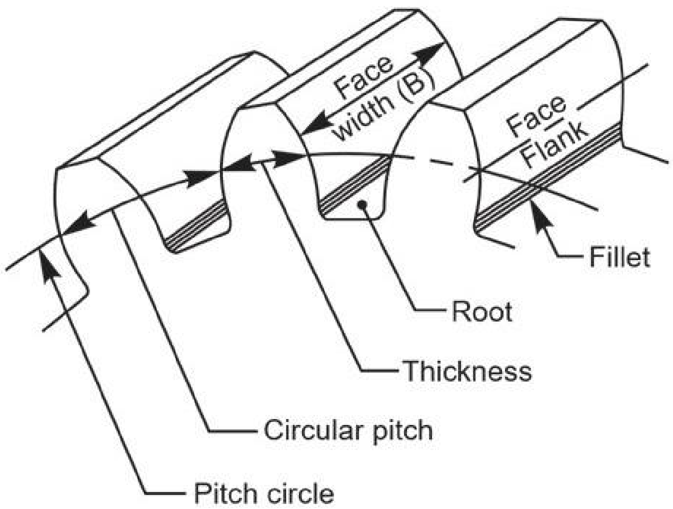


Figure 1.1 Gear basic geometry

The primary law of gearing is the constant angular velocities, so it can be defined as: "A common normal to the tooth profiles at their point of contact must, in all positions of the contacting teeth, pass through a fixed point on the line-of-centers called the pitch point."

There is no finite number of the curves that can satisfy this law of gearing. Hence modern gearing is based on an involute curve, as it has been used in this project.

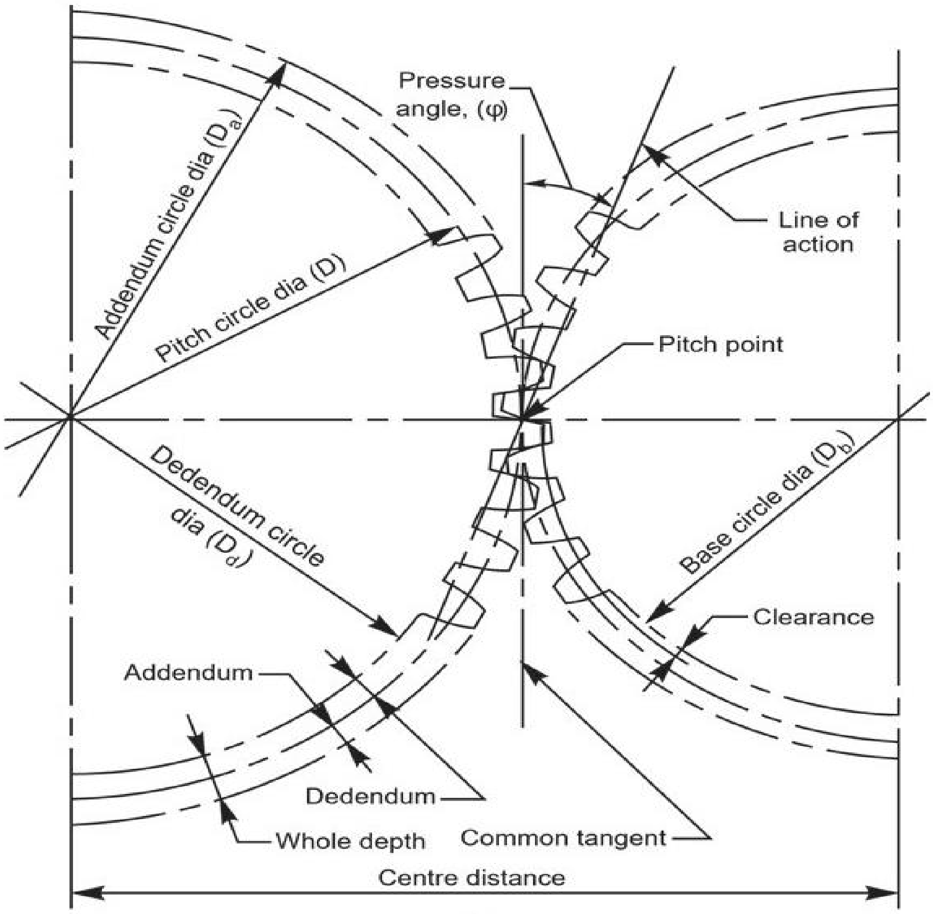


Figure 1.2 The meshing of 2 spur gear with related geometries

1.1.2. Helical gear

The difference between the spur gear and helical gear is that the teeth are twisted along a helical path. From the plane of rotation view, it looks like a spur gear, but in the axial direction, it seems like a set of staggered teeth.

The helical tooth profile is involute in the plane of rotation and can be developed in a manner similar to that of the spur gear. However, unlike the spur gear which can be viewed mainly as two dimensional, the helical gear must be portrayed in three dimensions to represent axial features.

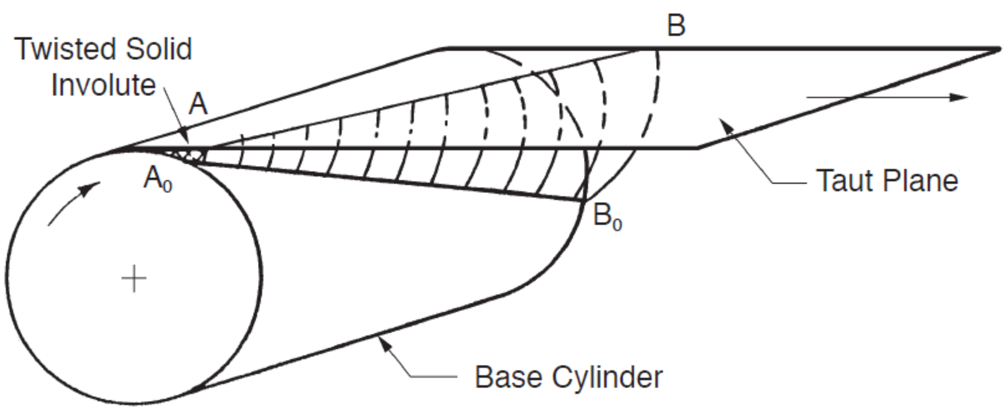


Figure 1.3 Representation for Helical Gear creation

1.2. Defining parameter of the gear

The type of gear which has mostly been used in this project is helical gear. Therefore all the relations are described for the helical gear; The spur gear can be defined as a helical gear with a helix angle equal to zero. Each gear has its own parameters, but in order to engage two gears, some of their parameters should be equal.

1.2.1. Helix angle (β)

The teeth of the helical gear, In the plane of rotation, are involute and all of the relationships governing spur gears apply to the helical gears as well. However, the primary definition of the helix angle is done by the axial twist of the teeth. Since the helix angle varies from the base of the tooth to the outside radius, the helix angle β is defined as the angle between the tangent to the helicoidal tooth at the intersection of the pitch cylinder and the tooth profile, and an element of the pitch cylinder. In the figure 1.4 helix angle is represented.

The helix angle range is defined from 0 to 45 degrees.

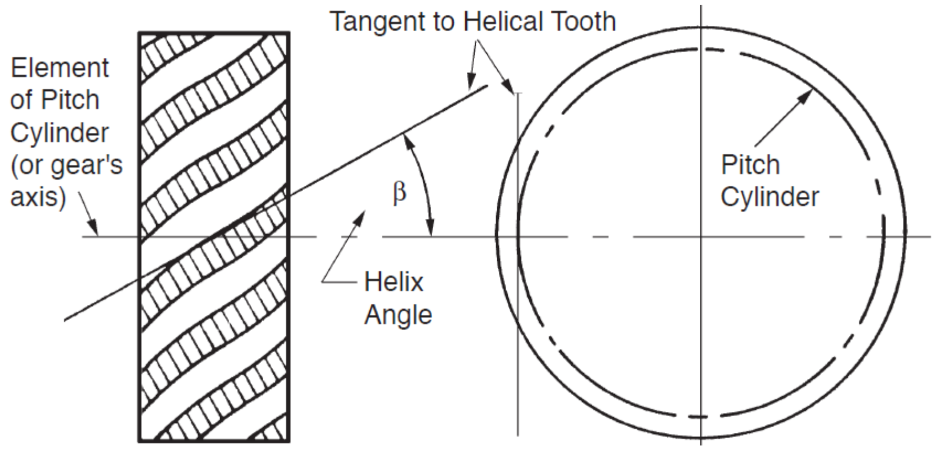


Figure 1.4 Helix angle representation

1.2.2. Pitch (P)

Pitch is a linear measurement along the pitch circle of the tooth spacing. It is the linear distance (measured along the pitch circle arc) between corresponding points of adjacent teeth. In the case of spur gear the pitch is equal to the pitch-circle circumference divided by the number of teeth:

$$P_t = \frac{\pi d}{z} \qquad (1.1)$$

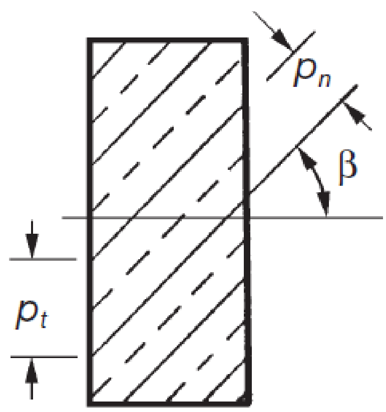


Figure 1.5 Normal and Transversal Pitch

In case of helical gear, there are two types of pitch that they are needed to be adequately defined; one is in the plane of rotation, and the other one is in a plane normal to the tooth.

$$P_n = P_t \cos\beta \qquad (1.2)$$

Another important parameter in order to define the geometry of the helical gear is the axial pitch and consequently the lead in an axial direction. The axial pitch of helical gear is the distance between the two consecutive points of the adjacent teeth parallel to the axis of the gear. Axial pitch can be expressed based on the circular pitch:

$$p_x = p_t \cot\beta \qquad (1.3)$$

A helical gear is a cylindrical gear in which the tooth face is helicoid. The displacement of one rotation is called the lead, and it is representing by L.

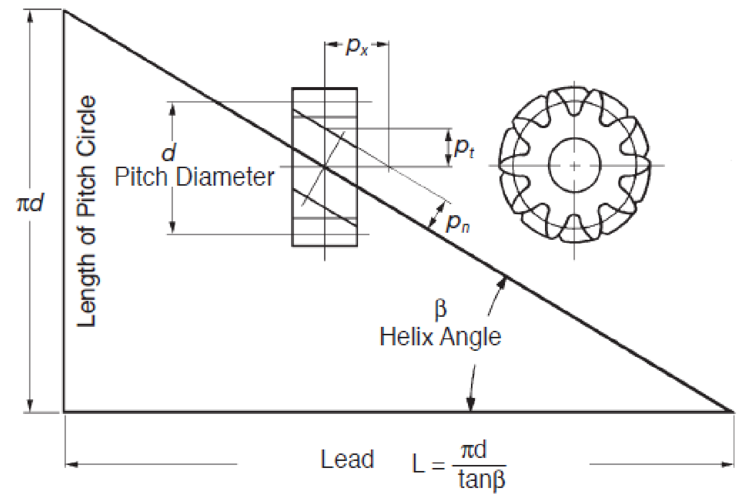


Figure 1.6 Lead and Axial Pitch

1.2.3. Module (m)

The module is the length of the pitch diameter per tooth, every type of module is the pitch divided by π, therefore:

$$m_t = \frac{P_t}{\pi} \qquad (1.4)$$

$$m_n = \frac{P_n}{\pi} \qquad (1.5)$$

$$m_n = m_t . cos\beta \qquad 1.6)$$

The normal modules are standard, and their value should be chosen from the following numbers.

Preferred normal module (m_n) [mm] : {1,1.25,1.5,2,2.5,3,4,5,6,8,10,12,16,20,25,32,40}

1.2.4. Pressure angle (α)

The pressure angle is defined as the angle between the line-of-action and perpendicular to the line-of-centers. From the geometry, it is evident that the pressure angle varies (slightly) as the center distance of a gear pair is altered.

Pressure angle can be defined regarding a single gear whereas, in case of mating helical gears, the normal pressure angle in the plane of rotation can be defined. Their relation can be expressed as:

$$\tan\alpha_t = \frac{\tan\alpha_n}{\cos\beta} \qquad (1.7)$$

Normal pressure angles are chosen from four predefined angles.

Normal pressure angle (α_n) [°]: {14.5,20,22.5,25}

There are four main circles which lead the overall shape of the gear.

Base circle: The base circle is the circle from which the involute curve of the teeth is generated.

Pitch or primitive circle: The tangent to the two base circles is the line of action. The intersection of this line with the line of center creates the pitch point. The distance between the center of the circles and the pitch point is the radius of the pitch circle. Using two inputs, i.e., the number of teeth and the module the pitch circle is equal to:

$$d = m_t z \qquad (1.8)$$

The base circle is related to the pressure angle and pitch diameter by the equation:

$$d_b = d\cos\alpha_t \qquad (1.9)$$

Addendum or tip circle: it is a circle that represents the outermost points of the teeth of a circular gear wheel. The addendum diameter with no undercut is equal to:

$$d_a = d + 2m_n \qquad (1.10)$$

Moreover, the addendum is:

$$h_a = m_n \qquad (1.11)$$

Dedendum or root circle: it is a circle touching the bottom of the spaces between the teeth of a gear wheel. Its diameter is equal to:

$$d_f = d - 2.5m_n \qquad (1.12)$$

Therefore, the dedendum is:

$$h_f = 1.25m_n \qquad (1.13)$$

So, the height of the tooth is equal to:

$$h_t = h_a + h_f = 2.25m_n \qquad (1.14)$$

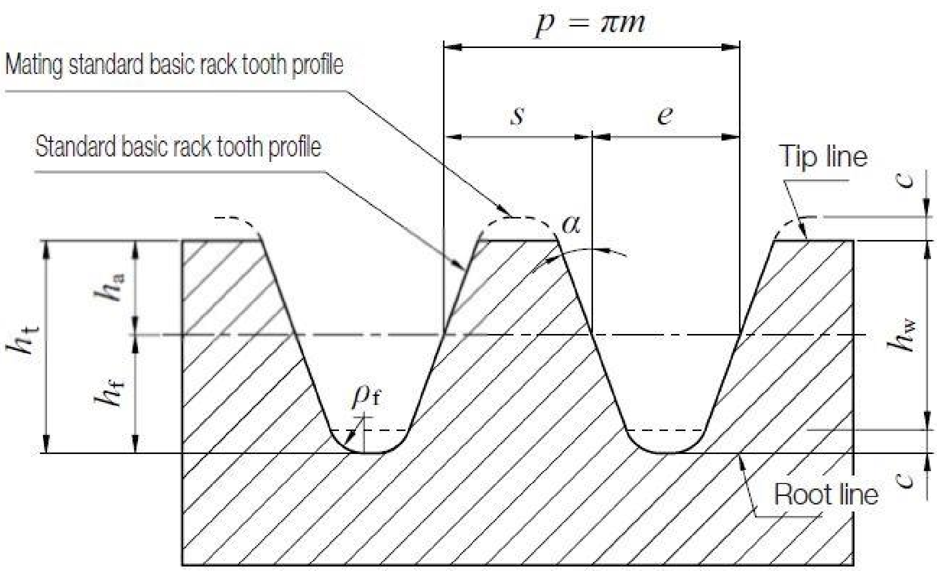


Figure 1.7 The representation for Pitch, Pressure angle, Addendum, and Dedendum

1.3. Verification of the Interaxis distance tolerance (The distance between input and output shaft)

The value of the inter-axis distance is extremely important. Due to the constructive parameters of the gears, it is difficult to obtain the exact value of the Inter-axis. For these reasons, a tolerance is introduced for the inter-axis distance within which the working distance of the mechanical transmission is allowed.

The formulas necessary for calculating the wheelbase of a gear composed of two engaged gear are as follows:

$$d = \frac{z.m_n}{\cos\beta} \quad (1.15)$$

$$i = \frac{d_{input} + d_{output}}{2} = \frac{m_n(z_{input} + z_{output})}{\cos\beta} \quad (1.16)$$

Minimum number of teeth

In the preliminary phase of design of the gears, it is crucial to verify the minimum allowed number of teeth in order to avoid the creation of the harmful phenomena of interference. It needs to be noted that there are at least two pairs of engaged teeth in the meshing gears. The formula used on the minimum number of teeth takes into account the actual ratio of transmission τ carried out by the gear. The formulas to carry out the verification of the minimum number of teeth are the following:

$$Z_{\min} = \frac{2(\sqrt{\sin^2\alpha_t \tau(\tau+2)+1}+1)}{\sin^2\alpha_t(\tau+2)} \quad (1.17)$$

note that:

$$\alpha_t = \tan^{-1}\left(\frac{\tan\alpha_n}{\cos\beta}\right) \quad (1.18)$$

$$\tau = \frac{z_{output}}{z_{input}} \quad (1.19)$$

1.4. The acting forces on the gears

In general, for Helical gears and therefore with non-zero helix angle the exchanged forces have three components: F_r Radial force component, Axial Force component F_a and F_t tangential force component. Obviously, the tangential force component F_t is responsible for the effective torque transmission between the gears. The formulas for calculating the forces exchanged between the gears are reported:

$$F_t = F_u = \frac{T}{r} \qquad (1.20)$$

$$F_r = \frac{F_t \tan \alpha_n}{\cos \beta} \qquad (1.21)$$

$$F_a = F_t \tan \beta \qquad (1.22)$$

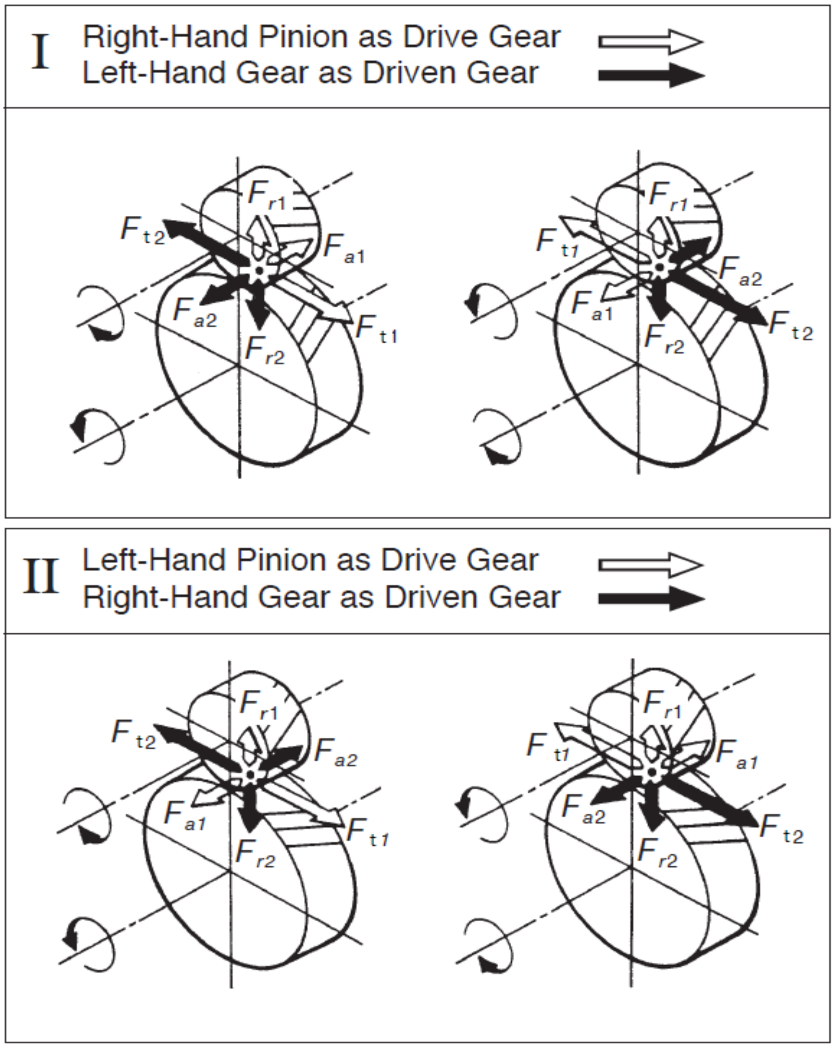


Figure 1.8 Representation of the imposed forces

For the part of the stress calculation and validation, all the formulas are taken from the ISO_06336 standard.

1.5. Design procedure of the Gear

The calculation reported in the previous section regarding the geometry needs to be reported and consequently design in the CAD software which is Nx Siemens. In order to have a parametric design, all the relation should be carefully added to the expression part.

The design procedure of the gear starts from the four fundamental circles, i.e., base circle, pitch circle, addendum, and dedendum circle. So, the sketch is created at the front face of the gear. Based on the formulas reported on the previous section the four diameters can be defined again:

$$d_p = \frac{m_n z}{\cos\beta} \qquad (1.23)$$

$$d_b = d_p.\arctan(\frac{\tan\alpha_n}{\cos\beta}) \qquad (1.24)$$

$$d_a = d_p + 2m_n \qquad (1.25)$$

$$d_f = d_p - 2.5m_n \qquad (1.26)$$

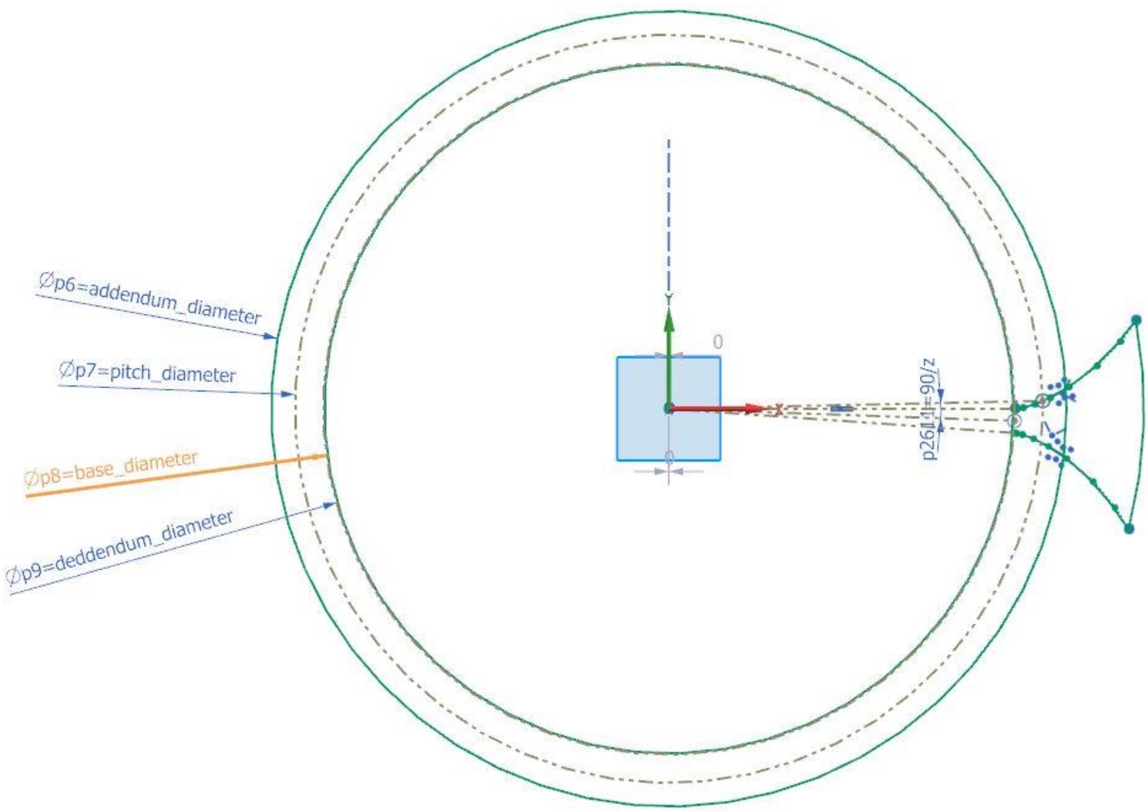


Figure 1.9 Gear Sketch

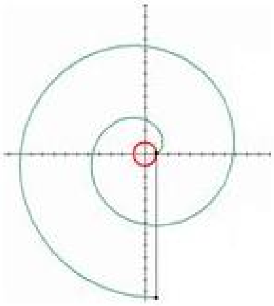
The two possible logic for creating the tooth is feasible, the teeth either can be added or can be subtracted. In order to make it similar to what happens in reality, the subtraction scenario is chosen.

Afterward, the involute curve of the tooth should be defined. To do so, there is a command called law curve in order to define the formulas for any desired curve. In the case of the involute curve, the following formulas are the required ones.

$$x(t)=r(\cos(360n.t)+2\pi.n.t.\sin(360n.t)) \quad (1.27)$$

$$y(t)=r(\sin(360n.t)-2\pi.n.t.\cos(360n.t)) \quad (1.28)$$

$$z(t)=0 \quad (1.29)$$



Where t is the function variable and n is the number of the turns.

In the next step, a circle with addendum diameter is extruded.

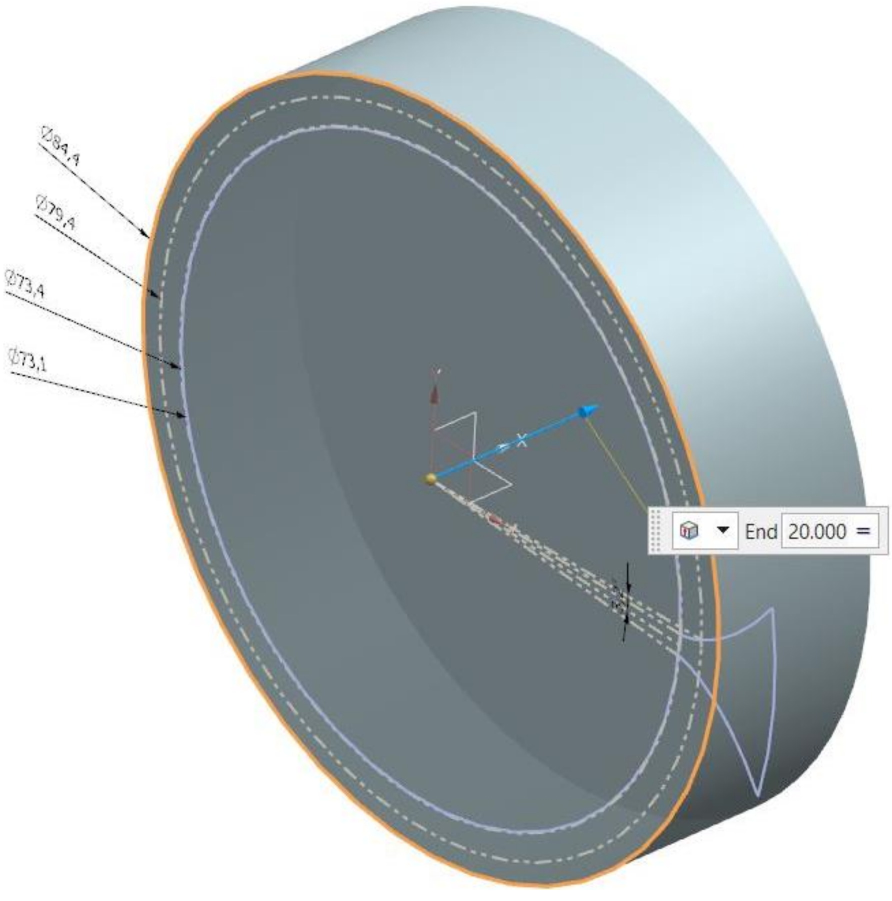


Figure 1.10 Extrusion of the gear sketch

Then an important step is to cut the gear with the cutting tool. In the case of the spur gear, the involute curve can be subtracted from the wheel along the wheel axis.

In the case of the helical gear the cutting path should be defined carefully, and with the helix function, it can be defined with two parameters which are the lead and the pitch diameter.

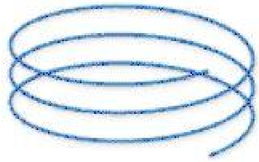
There are two ways to define the abovementioned parameters; they can be designed either by the pre-defined helix command in Nx or through the law curve.

In the case of the law curve the functions are:

$$x(t) = r\cos(n.t) \qquad (1.30)$$

$$y(t) = r\sin(n.t) \qquad (1.31)$$

$$z(t) = p.n.t \qquad (1.32)$$



Where p is the pitch (axial pitch) and r is the radius and n is the number of turns.

The second way of demonstration is by the helix command. The only limitation of this command is the hand of the helix, which can be only chosen inside the command, not through the sign change.

In the helix dialog box, the size is the diameter of the helix which is equal to the pitch diameter. The pitch is the axial pitch which is mentioned before. Its formula is:

$$Lead = \frac{\pi d_p}{\tan\beta} \qquad (1.33)$$

The next choice can be either the length of the helix or number of turns, where the length can be chosen equal to the width of the gear.

Finally, the direction of the helix can be either right or left. It is essential that two engaged gears have the opposite helix hand.

Then with a command called swept the cutting path can be assigned to the cutting tool in order to cut the wheel.

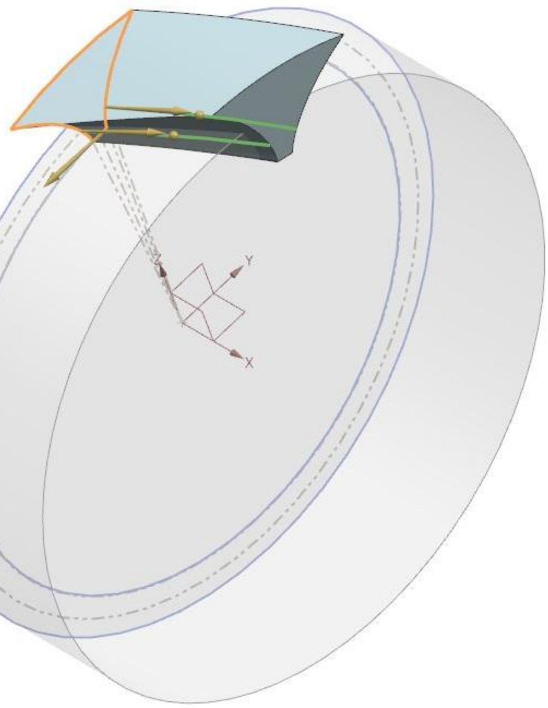


Figure 1.11 Extrusion of the cutting gear

Meanwhile, a fillet at the root radius should be applied; In order to maximize the life of a plastic gear and minimize the stress concentration, the root fillet radius should be large enough, consistent with conjugate gear action. The standard fillet radius is 0.38 of the modules. Sudden changes in cross-section and sharp corners should be avoided, especially given the possibility of additional residual stresses which may have occurred during the molding operation.

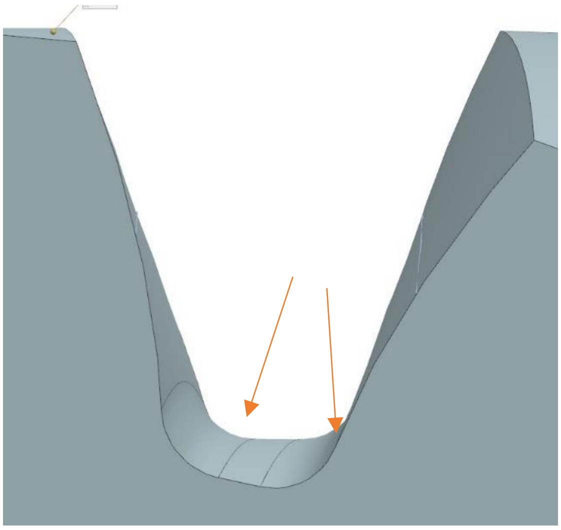


Figure 1.12 Fillet at the root of the teeth

Then the tool can subtract the tooth profile from the gear with the subtract command.

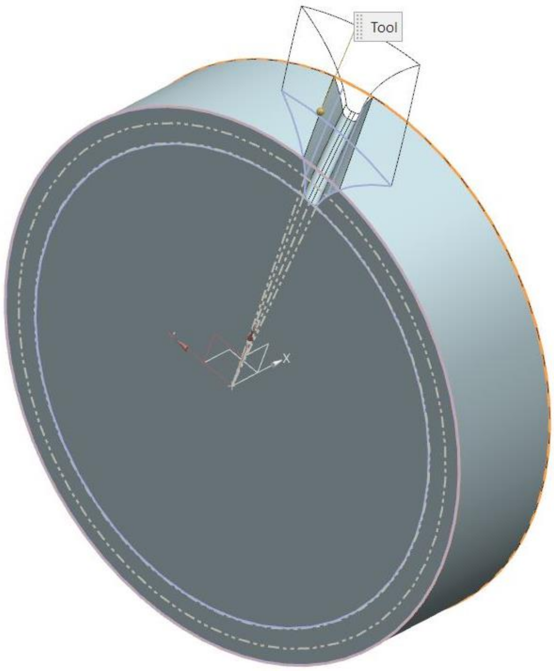


Figure 1.13 Cutting the gear

Finally, the cutting process should be replicated with the circular pattern feature, where the count is the number of teeth, and the span angle is equal to 360 degree which is one turn.

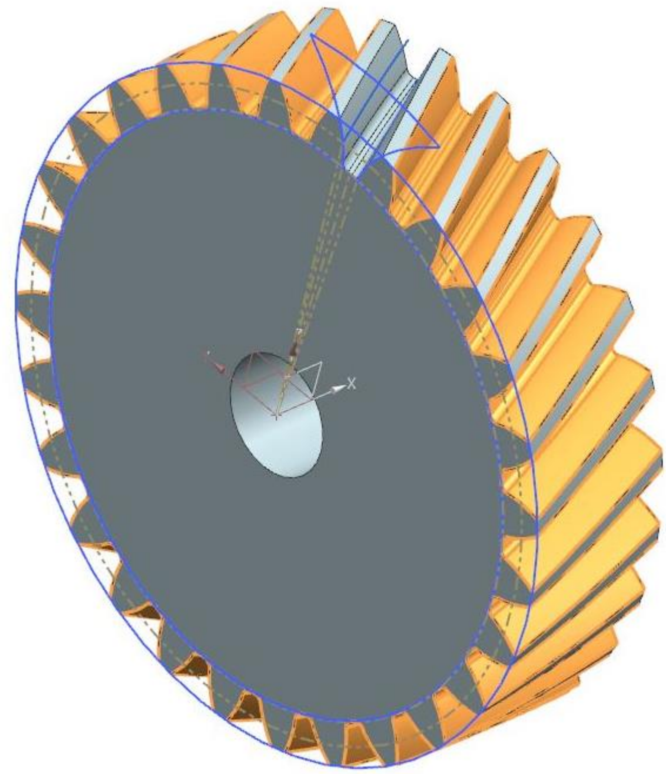


Figure 1.14 Final profile of the gear

2. Shafts

The design and specification of the shaft have high importance for either industrial or automotive gearbox. The shaft diameter is a crucial factor in order to determine the feasibility of the gearbox design. In the design process, the strength and the deformation should be calculated and considered carefully.

The details of the shaft itself that should be examined are the followings:

- Material selection
- Geometric layout
- Stress and strength analysis (static and fatigue strength)
- Deflection and rigidity

Deflection is not affected by strength, but rather by stiffness as represented by the modulus of elasticity, which is substantially constant for all steels. For that reason, rigidity cannot be controlled by material decisions, but only by geometric decisions.

Fatigue failure can be moderately reduced as the strength increase until a certain level, afterward, some unfavorable effects in notch sensitivity and endurance limit neutralize the benefit of higher strength.

If strength dominates the deflection, then a material with higher strength should be tried in order to allow the shaft size to be reduced until the high deflection becomes the issue.

The general layout of the shaft needs to be specified in order to place shaft elements such as gears and bearings.

This initial layout is mandatory in order to obtain the free body force analysis and extract the shear-moment diagram. The geometry of the shaft is usually is the stepped cylinder. The use of the shaft shoulders is an excellent means of axially locating the shaft elements and carrying any thrust loads.

Figure 2.1 Nissan 5SPD Billet Main Shaft

The axial positioning of the components usually depends on the geometry of the layout and other meshing components. Generally, it is better to have the massive and more load carrying components between bearings. In most cases, only two bearing should be used, but for extremely long shafts which are carrying several load-bearing components, it may be compulsory to provide more than two bearing supports. Since the distance between two bearing cause large bending moments and because of shoulders, grooves and bearing seat notches exist, the typical shaft configuration of a vehicle transmission is unfavorable and need to be carefully designed.

After the preliminary specification of the shaft layout and shaft diameters, the design based on the maximum anticipated loading should be done, so the maximum engine torque is used in the calculation. Transmission shaft are designed for finite service life based on established load profiles.

As it is mentioned above, in automotive gearboxes, shafts have a long distance between the two bearings and usually subjected to asymmetrical loads. This characteristic causes large deflection and large bending angles.

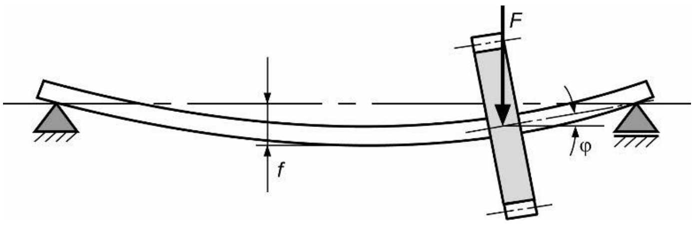


Figure 2.2 Deflection and bending angle in shafts with large distances between the bearing and asymmetrical loading

When designing a transmission shaft, the designer should work based on the maximum engine torque T_{max}. Therefore if the calculation relates to some other shafts rather than a transmission input shaft, the effective torque in the shaft should be calculated concerning the gear ratio of the respective gears.

The forces are considered as point loads.

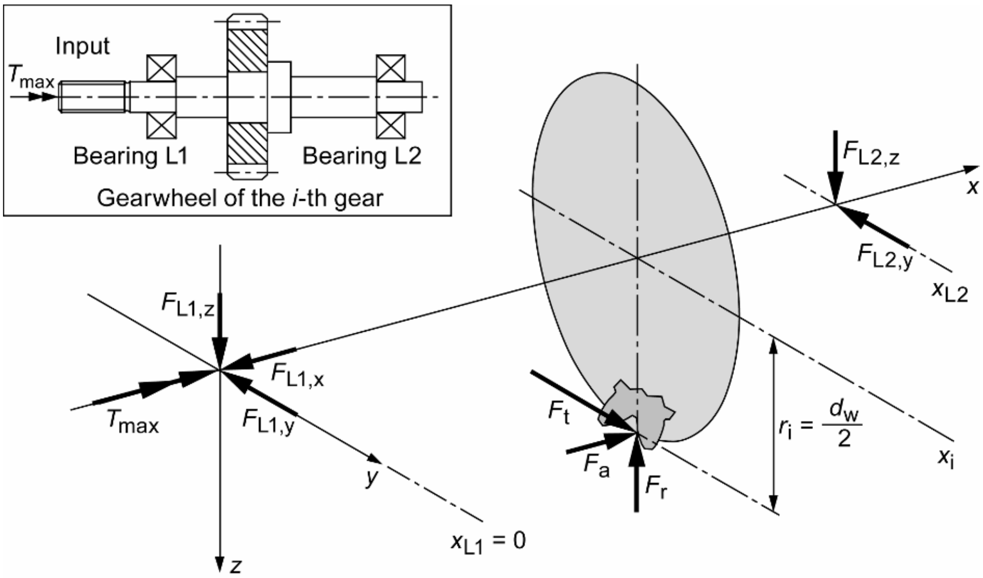


Figure 2.3 Loading of an input shaft for a single stage transmission

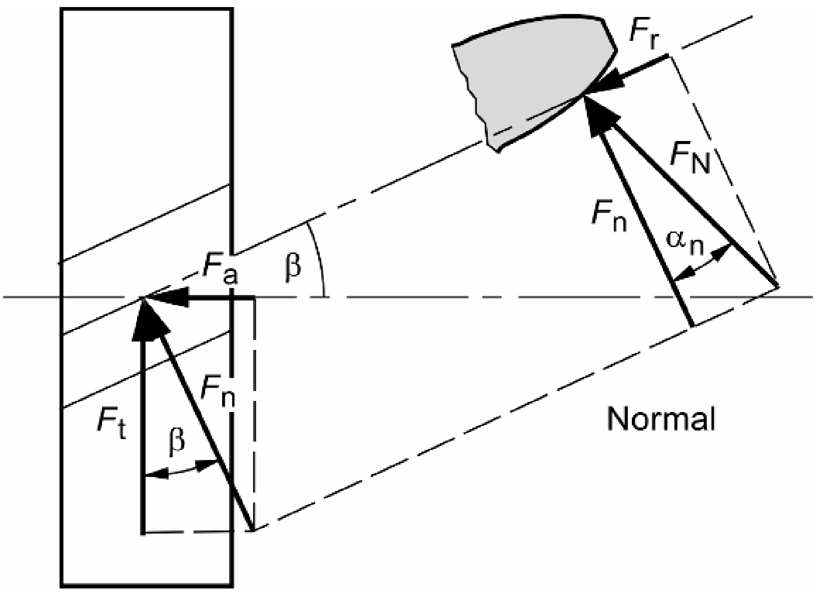


Figure 2.4 Forces acting at the tooth flanks of a helical gear

The analysis of the beam is done by the finite element. There are two beam theories which are usually used; the first and simpler one is Euler-Bernoulli, and the other one is the Timoshenko theory.

The Timoshenko beam model has achieved a significant improvement regarding computational results compared to the more straightforward and less accurate model of the Euler-Bernoulli beam. The Timoshenko beam model also considers the effects of the shear stress, while in the Euler-Bernoulli beam hypothesis this aspect is neglected. Moreover, since the Timoshenko beam model considers the shear stress acting on the beam member section, the cross sections do not remain normal to the axis after the deformation caused by the load. Hence, in the Timoshenko beam model, the cross sections are inclined and not normal to the center axis of the beam.

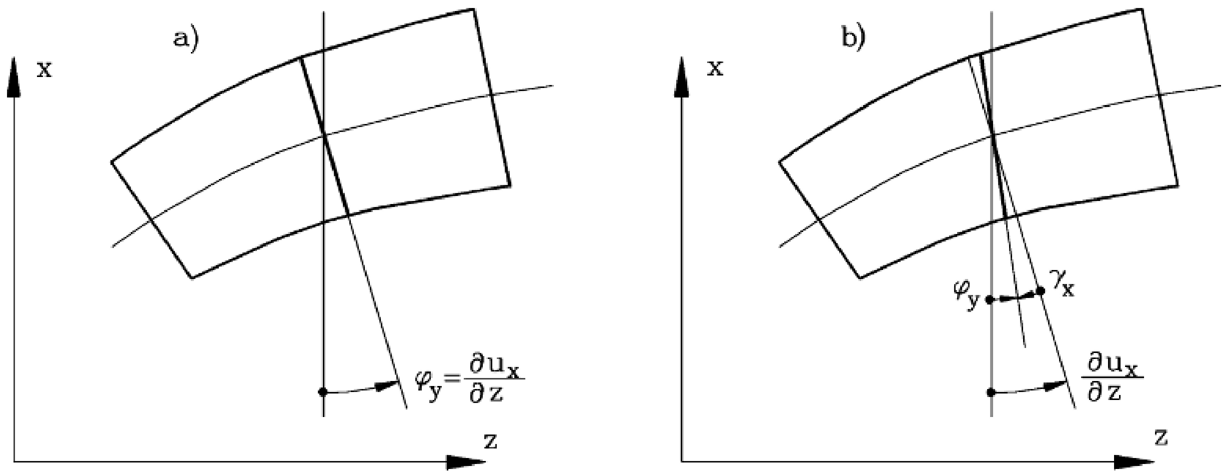


Figure 2.5 Effect of the shear deformation on beam bending. (a) Euler-Bernoulli beam; (b) Timoshenko Beam

2.1. Beam analysis with finite element

In order to perform beam analysis, the finite element method has been chosen. The finite element is one of the most common discretization methods. The FEM method is based on the subdivision of the geometry into the finite number of elements. Then many different element formulations have been developed. Then the elements need to be assembled therefore they usually developed using matrix form to obtain formulas easily into codes.

In the case of a Timoshenko beam, as it suggested before for this analysis, the simple Timoshenko beam has been used. It is a prismatic homogenous beam with a node at each end and six degrees of freedom per node.

The following figure shows a diagram of the Timoshenko beam, the directions and the positive verses for the displacements u, v, w and the rotations θ_x, θ_y and θ_z:

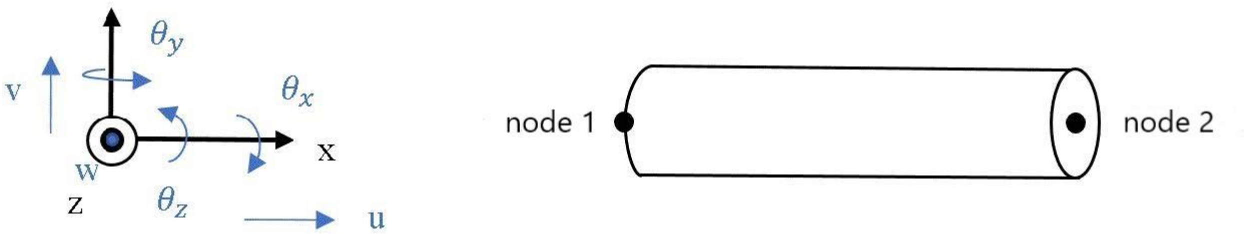


Figure 2.6 Reference system adopted for the Timoshenko beam model showing the positive verses of the axes, the translations, and the rotations

Each cross section has six degrees of freedom, three displacements, and three rotations. The total number of degrees of freedom of the element is thus 12. The generalized coordinates of the elements are:

$$q = [u_{x1}, u_{y1}, u_{z1}, \phi_{x1}, \phi_{y1}, \phi_{z1}, u_{x2}, u_{y2}, u_{z2}, \phi_{x2}, \phi_{y2}, \phi_{z2}]^T \quad (2.1)$$

Some geometric parameters should be calculated in order to form the desired matrices, the expressions for calculating the geometric properties of a non-hollow beam with a circular cross-section are as follows:

$$A = \frac{\pi d^2}{4} \qquad (2.2)$$

$$I_x = I_p = \frac{\pi d^4}{32} \quad (2.3)$$

$$I_y = I_z = \frac{\pi d^4}{64} \quad (2.4)$$

$$\phi_y = \frac{12 E I_z}{G A l^2} \qquad (2.5)$$

$$\phi_z = \frac{12 E I_y}{G A l^2} \qquad (2.6)$$

Where the description of the parameters are:

d: diameter of the beam element [m]

E: Young’s modulus [Pa]

G: Shear modulus [Pa]

A: Cross section area [m^2]

I_x, I_y, I_z: Moment of inertia

Φ_y, Φ_z: the ratio between the shear and the flexural flexibility of the beam

Then the stiffness matrix mass matrix and stress-strain matrix are:

$$[K]_e =$$

$$E\begin{bmatrix} \frac{A}{l} & 0 & 0 & 0 & 0 & 0 & -\frac{A}{l} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{12I_z}{(1+\phi_y)l^3} & 0 & 0 & 0 & \frac{6I_z}{(1+\phi_y)l^2} & 0 & -\frac{12I_z}{(1+\phi_y)l^3} & 0 & 0 & 0 & \frac{6I_z}{(1+\phi_y)l^2} \\ 0 & 0 & \frac{12I_y}{(1+\phi_z)l^3} & 0 & -\frac{6I_y}{(1+\phi_z)l^2} & 0 & 0 & 0 & -\frac{12I_y}{(1+\phi_z)l^3} & 0 & -\frac{6I_y}{(1+\phi_z)l^2} & 0 \\ 0 & 0 & 0 & \frac{GI_x}{lE} & 0 & 0 & 0 & 0 & 0 & -\frac{GI_x}{lE} & 0 & 0 \\ 0 & 0 & -\frac{6I_y}{(1+\phi_z)l^2} & 0 & \frac{(4+\phi_z)I_y}{(1+\phi_z)l} & 0 & 0 & 0 & \frac{6I_y}{(1+\phi_z)l^2} & 0 & \frac{(2-\phi_z)I_y}{(1+\phi_z)l} & 0 \\ 0 & \frac{6I_z}{(1+\phi_y)l^2} & 0 & 0 & 0 & \frac{(4+\phi_y)I_z}{(1+\phi_y)l} & 0 & -\frac{6I_z}{(1+\phi_y)l^2} & 0 & 0 & 0 & \frac{(2-\phi_y)I_z}{(1+\phi_y)l} \\ -\frac{A}{l} & 0 & 0 & 0 & 0 & 0 & \frac{A}{l} & 0 & 0 & 0 & 0 & 0 \\ 0 & -\frac{12I_z}{(1+\phi_y)l^3} & 0 & 0 & 0 & -\frac{6I_z}{(1+\phi_y)l^2} & 0 & \frac{12I_z}{(1+\phi_y)l^3} & 0 & 0 & 0 & -\frac{6I_z}{(1+\phi_y)l^2} \\ 0 & 0 & -\frac{12I_y}{(1+\phi_z)l^3} & 0 & \frac{6I_y}{(1+\phi_z)l^2} & 0 & 0 & 0 & \frac{12I_y}{(1+\phi_z)l^3} & 0 & \frac{6I_y}{(1+\phi_z)l^2} & 0 \\ 0 & 0 & 0 & -\frac{GI_x}{lE} & 0 & 0 & 0 & 0 & 0 & \frac{GI_x}{lE} & 0 & 0 \\ 0 & 0 & -\frac{6I_y}{(1+\phi_z)l^2} & 0 & \frac{(2-\phi_z)I_y}{(1+\phi_z)l} & 0 & 0 & 0 & \frac{6I_y}{(1+\phi_z)l^2} & 0 & \frac{(4+\phi_z)I_y}{(1+\phi_z)l} & 0 \\ 0 & \frac{6I_z}{(1+\phi_y)l^2} & 0 & 0 & 0 & \frac{(2-\phi_y)I_z}{(1+\phi_y)l} & 0 & -\frac{6I_z}{(1+\phi_y)l^2} & 0 & 0 & 0 & \frac{(4+\phi_y)I_z}{(1+\phi_y)l} \end{bmatrix}$$

$$[M]_e =$$

$$m_{el}\begin{bmatrix} \frac{1}{3} & 0 & 0 & 0 & 0 & 0 & \frac{1}{6} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{13}{35}+\frac{6I_z}{5Al^2} & 0 & 0 & 0 & \frac{11l}{210}+\frac{I_z}{10Al} & 0 & \frac{9}{70}-\frac{6I_z}{5Al^2} & 0 & 0 & 0 & -\frac{13l}{420}+\frac{I_z}{10Al} \\ 0 & 0 & \frac{13}{35}+\frac{6I_z}{5Al^2} & 0 & -\frac{11l}{210}-\frac{I_z}{10Al} & 0 & 0 & 0 & \frac{9}{70}-\frac{6I_z}{5Al^2} & 0 & \frac{13l}{420}-\frac{I_y}{10Al} & 0 \\ 0 & 0 & 0 & \frac{I_p}{3A} & 0 & 0 & 0 & 0 & 0 & \frac{I_p}{6A} & 0 & 0 \\ 0 & 0 & -\frac{11l}{210}-\frac{I_z}{10Al} & 0 & \frac{l^2}{105}+\frac{2I_y}{15A} & 0 & 0 & 0 & -\frac{13l}{420}+\frac{I_y}{10Al} & 0 & -\frac{l^2}{140}-\frac{2I_y}{30A} & 0 \\ 0 & \frac{11l}{210}+\frac{I_z}{10Al} & 0 & 0 & 0 & \frac{l^2}{105}+\frac{2I_z}{15A} & 0 & \frac{13l}{420}-\frac{I_z}{10Al} & 0 & 0 & 0 & -\frac{l^2}{140}-\frac{2I_z}{30A} \\ \frac{1}{6} & 0 & 0 & 0 & 0 & 0 & \frac{1}{3} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{9}{70}-\frac{6I_z}{5Al^2} & 0 & 0 & 0 & \frac{13l}{420}-\frac{I_z}{10Al} & 0 & \frac{13}{35}+\frac{6I_z}{5Al^2} & 0 & 0 & 0 & -\frac{11l}{210}-\frac{I_z}{10Al} \\ 0 & 0 & \frac{9}{70}-\frac{6I_z}{5Al^2} & 0 & -\frac{13l}{420}+\frac{I_y}{10Al} & 0 & 0 & 0 & \frac{13}{35}+\frac{6I_z}{5Al^2} & 0 & +\frac{11l}{210}+\frac{I_z}{10Al} & 0 \\ 0 & 0 & 0 & \frac{I_p}{6A} & 0 & 0 & 0 & 0 & 0 & \frac{I_p}{3A} & 0 & 0 \\ 0 & 0 & \frac{13l}{420}-\frac{I_y}{10Al} & 0 & -\frac{l^2}{140}-\frac{2I_y}{30A} & 0 & 0 & 0 & +\frac{11l}{210}+\frac{I_z}{10Al} & 0 & \frac{l^2}{105}+\frac{2I_y}{15A} & 0 \\ 0 & -\frac{13l}{420}+\frac{I_z}{10Al} & 0 & 0 & 0 & -\frac{l^2}{140}-\frac{2I_z}{30A} & 0 & -\frac{11l}{210}-\frac{I_z}{10Al} & 0 & 0 & 0 & \frac{l^2}{105}+\frac{2I_z}{15A} \end{bmatrix}$$

$$[D]_e = \frac{E(1-\nu)}{(1+\nu)(1-2\nu)} \begin{bmatrix} 1 & \frac{\nu}{1-\nu} & \frac{\nu}{1-\nu} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{\nu}{1-\nu} & 1 & \frac{\nu}{1-\nu} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{\nu}{1-\nu} & \frac{\nu}{1-\nu} & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1-2\nu}{2(1-\nu)} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1-2\nu}{2(1-\nu)} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1-2\nu}{2(1-\nu)} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & \frac{\nu}{1-\nu} & \frac{\nu}{1-\nu} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{\nu}{1-\nu} & 1 & \frac{\nu}{1-\nu} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{\nu}{1-\nu} & \frac{\nu}{1-\nu} & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{1-2\nu}{2(1-\nu)} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{1-2\nu}{2(1-\nu)} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{1-2\nu}{2(1-\nu)} \end{bmatrix}$$

2.2. Assembly of mass [$\boldsymbol{M}$], stiffness [$\boldsymbol{K}$] and stress-strain [$\boldsymbol{D}$] matrices

Once the mass, stiffness and stress-strain matrices of the single beam element composed of two nodes have been written, they must be inserted in the matrices of global mass [M], global stiffness [K] and global stress-strain [D] for the whole shaft. In fact, the shaft is decomposed into a series of beams elements by inserting the nodes at the points where there is a change of geometry, applied force or any constraints. The assembly of the matrices was automated with a calculation algorithm within the Matlab code.

2.3. Static analysis of the shaft:

Before checking the static deflection of the shaft, initial diameters must be chosen to calculate the geometric properties of the sections of the beam elements in which the shaft has been divided. In the developed code an iterative algorithm has been implemented on the diameters for the different beam elements of the shaft. The diameters of the elements have a considerable influence on the verification of the static deflection of the shaft; with larger diameters, there are greater stiffnesses and therefore less static deflection while with smaller diameters there are less stiffness and greater static deflections.

After assembling the global mass [M] and global stiffness [K] matrices, the constrained degrees of freedom must be eliminated. In the case of static deflection, the constrained degree of freedom are:

Bearing: blocks the translations u, v, w, and the rotations θy, θz allowing the rotation θx;

Gear: blocks rotation θx allowing the rotations θy, θz and the translations u, v, w.

Therefore, for the constrained degree of freedom, the lines and columns related to the respective degrees of freedom at the global matrices must be eliminated, and new obtaining the known matrices of mass [M], stiffness [K], damping [C], and the force vector {F}. The equation of motion of the shaft, considering the known matrices, is as follows:

$$[M]\{\ddot{X}\}+[C]\{\dot{X}\}+[K]\{X\}=\{F\} \quad (2.7)$$

Where in the case of the static deflection, neglecting the second derivative and the first derivative of vector displacements $\{X\}$, the equation of the motion of the shaft will be simplified in:

$$[K]\{X\}=\{F\} \quad 2.8)$$

Where with solving the equation of motion the displacement vector can be found.

$$\{X\}=[K]^{-1}\{F\} \quad (2.9)$$

Then from displacement's vector, the translational displacement should be less than the permitted tolerance.

$$Max\ (|\{X\}_{translational}|) \leq \text{maximum permitted deflection}$$

Maximum permitted deflection= min($h_{a,l,input}$-$h_{f,l,input}$, $h_{a,l,output}$-$h_{f,l,input}$)

To perform the static dimensioning of the shaft, subsequently, to calculate the static stresses that are imposed on the shaft, the static deflection displacement vector {X}, which was previously calculated to perform the static deflection check of the shaft, is used. Obviously, the diameters of the beam elements, chosen to check the static deflection of the shaft and in which the shaft has been divided, have a considerable influence on the static sizing of the shaft. From the displacement vector of the static deflection {X} the displacements u, v, w and the rotations θx, θy and θz of the N + 1 nodes in which the shaft has been subdivided and inserted into the following vectors are extracted:

$$\{u\}_{static}=\begin{Bmatrix} u_1 & u_2 & u_3 & \dots & u_{N-1} & u_N & u_{N+1} \end{Bmatrix}^T$$

$$\{v\}_{static}=\begin{Bmatrix} v_1 & v_2 & v_3 & \dots & v_{N-1} & v_N & v_{N+1} \end{Bmatrix}^T$$

$$\{w\}_{static}=\begin{Bmatrix} w_1 & w_2 & w_3 & \dots & w_{N-1} & w_N & w_{N+1} \end{Bmatrix}^T$$

$$\{\theta_x\}_{static}=\begin{Bmatrix} \theta_{x1} & \theta_{x2} & \theta_{x3} & \dots & \theta_{x,N-1} & \theta_{x,N} & \theta_{x,N+1} \end{Bmatrix}^T$$

$$\{\theta_z\}_{static}=\begin{Bmatrix} \theta_{z,1} & \theta_{z,2} & \theta_{z,3} & \dots & \theta_{z,N-1} & \theta_{z,N} & \theta_{z,N+1} \end{Bmatrix}^T$$

$$\{\theta_y\}_{static}=\begin{Bmatrix} \theta_{y1} & \theta_{y2} & \theta_{y3} & \dots & \theta_{y,N-1} & \theta_{y,N} & \theta_{y,N+1} \end{Bmatrix}^T$$

Based on the above vectors the strains ε_{xx}, ε_{yy} , ε_{zz} , γ_{xy} , γ_{yz}, γ_{xz} can be calculated:

$$du_i = u_i - u_{i-1} \quad (2.10)$$

$$dv_i = v_i - v_{i-1} \quad (2.11)$$

$$dw_i = w_i - w_{i-1} \quad (2.12)$$

$$dx = l\left(1-\frac{du}{\sqrt{(du)^2+(dv)^2+(dw)^2}}\right) \quad (2.13)$$

$$dy = l\left(1-\frac{dv}{\sqrt{(du)^2+(dv)^2+(dw)^2}}\right) \quad (2.14)$$

$$dz = l\left(1-\frac{dw}{\sqrt{(du)^2+(dv)^2+(dw)^2}}\right) \quad (2.15)$$

$$\varepsilon_{xx} = \frac{du}{dx} \qquad (2.16)$$

$$\varepsilon_{yy} = \frac{dv}{dy} \qquad (2.17)$$

$$\varepsilon_{zz} = \frac{dw}{dz} \qquad (2.18)$$

$$\gamma_{xy} = \frac{du}{dy} + \frac{dv}{dx} \quad (2.19)$$

$$\gamma_{yz} = \frac{du}{dy} + \frac{dv}{dx} \quad (2.20)$$

$$\gamma_{xz} = \frac{du}{dz} + \frac{dw}{dx} \quad (2.21)$$

Then all these parameters of strains can be collected in a matrix.

$$\{\varepsilon\}_{static} = \{\varepsilon_{xx} \quad \varepsilon_{yy,1} \quad \varepsilon_{zz,1} \quad \gamma_{xy,1} \quad \gamma_{yz,1} \quad \gamma_{xz,1} \quad \cdots \quad \varepsilon_{xx,N+1} \quad \varepsilon_{yy,N+1} \quad \varepsilon_{zz,N+1} \quad \gamma_{xy,N+1} \quad \gamma_{yz,N+1} \quad \gamma_{xz,N+1}\}^T$$

Then the static vector of the stresses {σ} acting in the nodes is calculated:

$$\{\sigma_{static}\} = [D]\{\varepsilon\}_{static} \quad (2.22)$$

Then the stress tensor for each node is defined as:

$$[\sigma]_i = \begin{bmatrix} \sigma_{xx,i} & \tau_{xy,i} & \tau_{xz,i} \\ \tau_{xy,i} & \sigma_{yy,i} & \tau_{yz,i} \\ \tau_{xz,i} & \tau_{yz,i} & \sigma_{zz,i} \end{bmatrix}$$

The tensor of the stresses is needed to be transformed to principal stresses. Therefore the calculation of the eigenproblems as the following should be done;

$$\det \begin{bmatrix} \sigma_{xx,i} - \sigma & \tau_{xy,i} & \tau_{xz,i} \\ \tau_{xy,i} & \sigma_{yy,i} - \sigma & \tau_{yz,i} \\ \tau_{xz,i} & \tau_{yz,i} & \sigma_{zz,i} - \sigma \end{bmatrix} = 0 \qquad (2.23)$$

The eigenvalues σ that solve the self-problem are the principal's tensions in each node are $\sigma_{1,i}$, $\sigma_{2,i}$ and $\sigma_{3,i}$.

The equivalent stresses $\sigma_{eq,i}$ in every single node is calculated by involving the principals tension in the each node $\sigma_{1,i}$, $\sigma_{2,i}$ and $\sigma_{3,i}$ and using the equation relative to the failure hypothesis of Von Mises which is based on the maximum energy of deformation:

$$\sigma_{eq,i} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_{1,i} - \sigma_{2,i})^2 + (\sigma_{2,i} - \sigma_{3,i})^2 + (\sigma_{1,i} - \sigma_{3,i})^2} \qquad (2.24)$$

The vector for equivalent tension is:

$$\{\sigma_{eq}\}_{static} = \left\{\sigma_{eq,1} \quad \sigma_{eq,2} \quad \cdots \quad \sigma_{eq,N} \quad \sigma_{eq,N+1}\right\}^T \qquad (2.25)$$

The expression for the static sizing is as following:

$$\sigma_{eq,\max} = \max\left(\left\{\sigma_{eq}\right\}_{static}\right) \qquad (2.26)$$

$$FoS = \frac{R_{p_{02}}}{\sigma_{eq,\max}} \qquad (2.27)$$

For the sizing of the Shafts in both automotive and industrial gearbox, the value has been used for the minimum safety coefficient to be guaranteed for the static sizing of the shaft considers as:

$$FoS \geq 2 \qquad (2.28)$$

2.4. Shaft Connection

Most shafts serve to transmit torque from an input gear or pulley, through the shaft, to an output gear or pulley. Of course, the shaft itself must be sized to support the torsional stress and torsional deflection. It is also necessary to provide a means of transmitting the torque between the shaft and the gears. In order to connect the gears to the shaft and provide the torque transmission, there are several common elements:

2.4.1. Keys

One of the most effective and economical means of transmitting moderate to high levels of torque is through a key that fits in a groove in the shaft and gear. Keyed components generally have a slip fit onto the shaft, so assembly and disassembly are easy.

The feather key is considered the least strong connection and is restricted to the simpler connections where load reversals are minimal and where the shear strength of the key is considered sufficient.

2.4.2. Splines

Splines are mainly short and thick gear teeth formed on the outside of the shaft and the inside of the hub of the load-transmitting component such as gears. Splines are generally much more expensive to manufacture than keys. They are typically used to transfer high torques. In the case of the automotive gearbox, usually, the splines are used. One feature of a spline is that it can be made with a reasonably loose slip fit to allow for large axial motion between the shaft and component while still transmitting torque. This is useful for connecting two shafts where relative motion between them is common. There are two types of spline that are usually put to work, straight spline and involute spline.

The **straight-sided spline** is able to carry reversing loads and is reasonably simple to manufacture. The load carrying capacity of the spline is better than the simple feather key, but due to pitch errors, not all the splines carry the full load, allowing other splines to become overloaded; therefore, considerable care is required in the calculation to ensure there is an adequate strength.

The **involute spline** is considered as the most reliable connection method as the number of teeth is increased with respect to the straight spline. They are also able to take high reversing loads. Because the tooth form is primarily based on the gear tooth, pitch errors are minimized, ensuring a far greater number of teeth able to carry the torsional load. In any shaft calculation the strength of the shaft will dictate the load carrying capacity, and therefore this will be the first requirement to be calculated.

2.4.3. Press or shrink fits

They are used both for torque transfer and for preserving axial location. The resulting stress-concentration factor is usually quite small.

The operating principle realizes the diameter of the shaft on which the hub is mounted with a diameter slightly larger than the diameter of the hub hole. In this way, it is possible to obtain an interference between the shaft and the hub that allows the hub to be mounted on the shaft, by constraining the torsional and axial displacement.

In this chapter, the spline connection that is used in this project will be explained in detail.

2.4.4. Spline design

In order to dimension the spline profile, a function implemented in the MATLAB code. The type of spline profile which is used in this project is the involute spline which is the most reliable type of the spline. Since the spline profile is under load due to the torque which should be transmitted, its profile needs to be dimensioned, and its reliability should be checked according to one of the standards. In this project, the verification is done considering the ISO 4156.

2.4.5. Spline parameters

2.4.5.1. Pressure angle

Similar to gears, there are some standard pressure angles for the splines. The standard pressure angles based on the ISO are 30°, 37.5°, and 45° and is represented by α.

2.4.5.2. Module

Another defining parameter is a module. The modules for each pressure angle are as the following:

Table 2.1 Normalized Module

		Module, m_n [mm]													
Pressure angle	30°	0.5	0.75	1	1.25	1.5	1.75	2	2.5	3	4	5	6	8	10
	37.5°	0.5	0.75	1	1.25	1.5	1.75	2	2.5	3	4	5	6	8	10
	45°	0,25	0.5	0.75	1	1.25	1.5	2	2.5						

2.4.5.3. Number of teeth

The number of teeth can be in the range of 6 to 100 teeth; Its symbol is z.

In order to have better understating about the spline profile, firstly, different diameters need to be clarified. The profile of the spline is quite similar to the profile of the gears, and they consist of internal and external teeth.

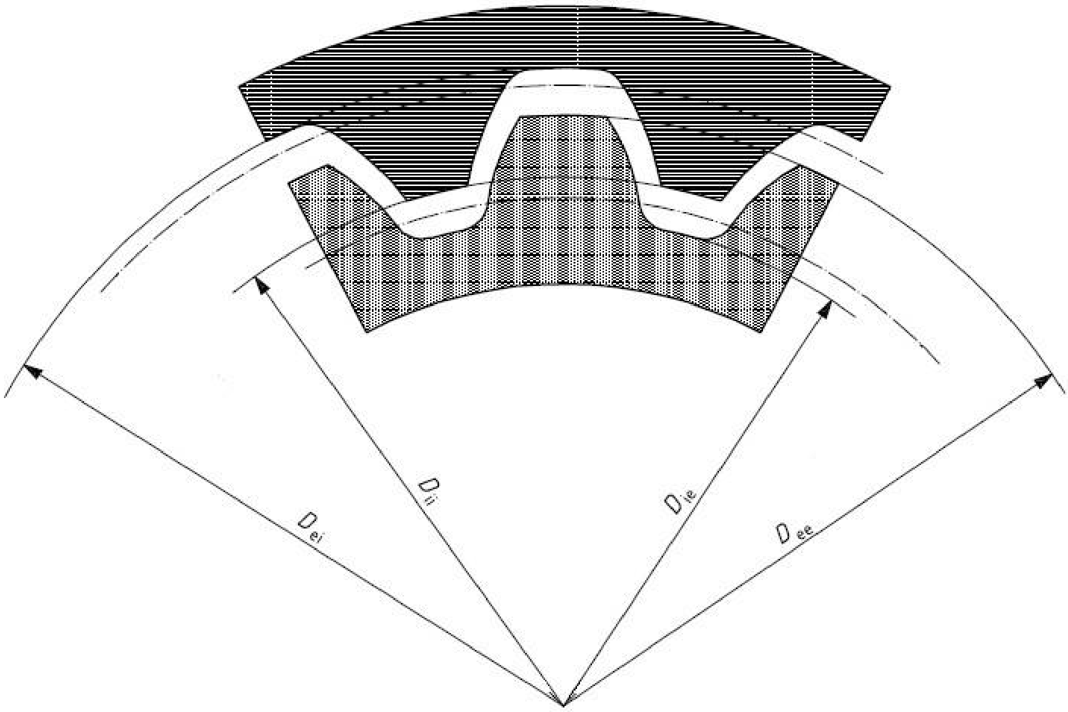


Figure 2.7 The diameters for the spline representation

Where the D_{ei} is the external outer diameter, D_{ii} is the external inner diameter. Whereas D_{ie} is the internal inner diameter and D_{ee} is the internal outer diameter. In the MATLAB code, they are defined as d_{o_e}, d_{i_e}, d_{i_i}, d_{o_i} respectively.

Based on the standard, the abovementioned diameters are all extracted from the pitch or mean diameter. The definition of the mean diameter is similar what has defined for gears.

$$D_m = m_n z \qquad (2.29)$$

For each pressure angle, the values for different diameters need to be defined according to the standard. In the following formula for the sake of simplicity, the tolerances have been neglected. Pressure angle equal to 30°:

$$D_{ie} = m_n(z-1.8) \qquad (2.30)$$

$$D_{ee} = m_n(z+1) \qquad (2.31)$$

$$D_{ei} = m_n(z+1.8) \qquad (2.32)$$

$$h_t = 0.9m_n \qquad (2.33)$$

Pressure angle equal to 37.5°:

$$D_{ie} = m_n(z-1.4) \qquad (2.34)$$

$$D_{ee} = m_n(z+0.9) \qquad (2.35)$$

$$D_{ei} = m_n(z+1.4) \qquad (2.36)$$

$$h_t = 0.7m_n \qquad (2.37)$$

Pressure angle equal to 45°:

$$D_{ie} = m_n(z-1.2) \qquad (2.38)$$

$$D_{ee} = m_n(z+0.8) \qquad (2.39)$$

$$D_{ei} = m_n(z+1.2) \qquad (2.40)$$

$$h_t = 0.6m_n \qquad (2.41)$$

Where in these formulas h_t is the height of each tooth and will be considered in order to calculate the cross-section area.

2.4.6. Strength Capacity of Iso Involute Spline

In order to calculate the strength capacity of the spline, the following formulas have been used.

The shear stress imposes on the mean diameter from the applied torque:

$$\tau_m = \frac{4Tk_s}{D_m z b h_t} \qquad (2.42)$$

Whereas the compressive stress in teeth due to the applied torque is:

$$\sigma_c = \frac{2Tk_s}{D_{ie} z b h_t} \qquad (2.43)$$

The expressions for the safety factor to carry out the strength's validation of the spline profile are as the following:

$$FoS_{shear} = \frac{0.577R_{p02}}{\tau_m} \qquad (2.44)$$

$$FoS_{von-mises} = \frac{R_{p02}}{\sqrt{\sigma_c^2 + 3\tau_m^2}} \qquad (2.45)$$

2.4.7. Effective factors

In order to have a better understanding of the above formulas, some effective factors are needed to be defined.

2.4.7.1. Application factor k_a

The application factor compensates for any uncertainties in loads and impacts, whereby $k_a \geq 1.0$.

Table 2.2 spline application factors K_a

	Type of Load			
	Uniform Generators, Fans	Light Shock Oscillating Pumps	Intermediate Shock Actuators	Heavy Shock Presses, Shears
Power Source	Application Factor (K_a)			
Uniform (turbine motor)	1.0	1.2	1.5	1.8
Light shock (hydraulic motor)	1.2	1.3	1.8	2.1
Medium shock (ICE)	2.0	2.2	2.4	2.8

In the MATLAB code, for the type of load the intermediate shock actuators and the power source the medium shock has been taken into account. Therefore the value for this factor is equal to 2.4.

2.4.7.2. Fatigue life Factor k_f

It is evident that fatigue life factor depends upon fatigue life and it depends on the number of cycles, and the type of the shaft rotation.

Table 2.3 The fatigue Life factor k_f

Number of Torque Cycles (Start/Stop Cycles)	Fatigue Life Factor (K_f) Unidirectional	Fully Reversed
1×10^3	1.8	1.8
1×10^4	1.0	1.0
1×10^5	0.5	0.4
1×10^6	0.4	0.3
1×10^7	0.3	0.2

In the code, the assumed number of cycles is $10^{5,}$ and the type of direction is unidirectional.

2.4.7.3. Service factor k_s

The final factor which is obtained by just dividing the two previous factors is service factor with this factor is used in case of fixed close fit/guided spline

2.5. Shaft CAD Design

Parametric design of a shaft with different parts and features is complicated. Therefore, here the idea is to provide a schematic of the overall dimension of the shaft.

This idea is to divide the shaft into different parts. Each part is related to its own components. Therefore, in this configuration, the shaft will be divided into eight parts, where six of them are related to gears, and the other two are related to bearings. Then each component for the gears divides into three parts where the first part and the last part are the lengths of the shaft before and after the gear, and the middle part is the width of gear. The middle part can be either spline in order to get connected to the gear a simple cylinder to represent the press/shrink fit connection. Also, in the case of the gear integrated with the shaft the simple cylinder will be taken into account.

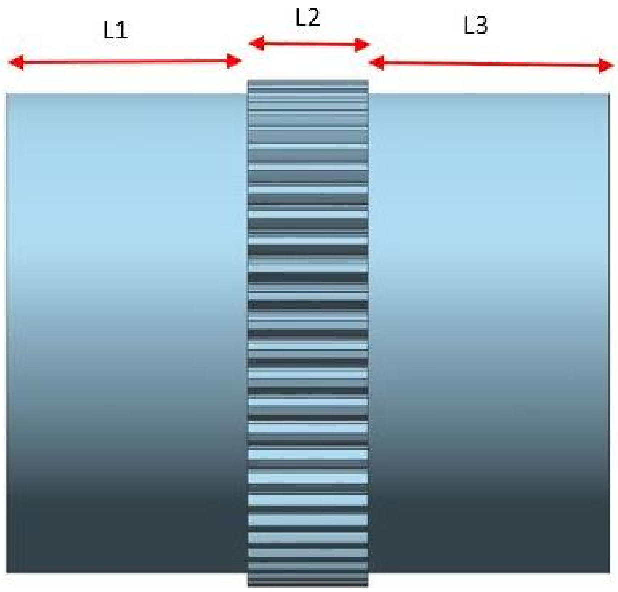


Figure 2.8 Shaft component

The separate components then will be connected and assembled in a file which will be implemented in the main assembly as a sub-assembly and represent the shaft.

3. Synchronizers

In a manual synchromesh transmission, friction clutches called synchronizers are used to synchronize the rotational speed of the transmission output shaft and the gear to be engaged to actualize and secure smooth and noiseless gear transition. The size and location of synchronizers in transmissions varies for passenger cars and trucks. In sizing and locating the synchronizers, it is vital to ensure minimizing the effect of system inertia and relative speeds of the rotating components. It must be noted that synchronizer endures many torques and forces, more than any other transmission components, and is expected to continue to work flawlessly for the life of the vehicle.

3.1. The gear change process

The gear change process in this project is synchromesh. It means that the gears are constantly meshed, and they are rotating as a driving-driven pair; except here the synchronizers are inserted between gears.

The gears rotate all the time freely and the intended gear engaged through the synchronizer, transmit the torque. The synchronizer acts as friction clutch to match the relative speed between the engine, gear and clutch disk. Therefore the synchronizer provides the mechanism for the gear change process.

The synchronization process always follows the same sequences. The sleeve is moved by the shift fork towards the gear to be engaged. As long as there is a speed difference between the sleeve/hub-system and the gear wheel the sleeve is blocked by the blocker ring and the synchronizer rings create a friction torque. When the speeds are synchronized, the sleeve can be moved further and engages into the spline of the engagement ring at the gear wheel.

3.2. Synchronizer components

The synchronizer assembly is selected and designed to meet the load requirements of a specific application. Typically, a synchronizer assembly will be inserted between two gears and with sliding engages the intended one. Synchronizer assembly consists of the following components, and respective design features are briefly described.

3.2.1. Hub

Hub has inner and outer splines; it is rotationally connected to the output shaft, spline-fitted with internal splines. Three equally spaced slots are designed on the outer diameter for housing the detent assemblies.

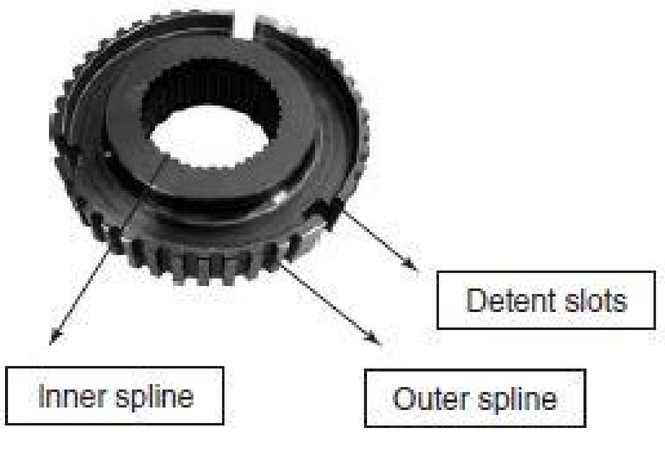


Figure 3.1 Hub

3.2.2. Sleeve

The sleeve has inner splines, and it is fitted to the external splines of the hub and travels during the synchronization to lock with the clutching teeth of the intended gear. It has three annular grooves on the internal splines, equally spaced, for housing the three detents. An annular recess around the outer surface is provided for shift fork legs that instrumentally slide the sleeve left or right when activated by the externally applied force at the gearshift lever for consequent gear engagement.

Figure 3.2 Sleeve

3.2.3. Strut detents

Three strut detent assemblies per synchronizer are designed; the detent strut bumps or balls are nested in the three annular grooves of the sleeve in neutral position. These are primary energizing elements and provide an initial indexing load for the friction (blocker) ring and set it ready for the oncoming sleeve.

3.2.4. Friction (blocker) ring

The friction ring made of Bronze with oil wiping threads, having three equally spaced slots for the strut detent to push on. Various design and manufacturing developments have occurred to tackle the higher loads. Powder metal rings are being used coated with different robust friction materials for the adequate and stable coefficient of friction at a specified shift force. Material friction durability and efficiency are also significant considerations. The inner side of the blocking ring is threaded in order to have better engagement with the cone and acts as a break.

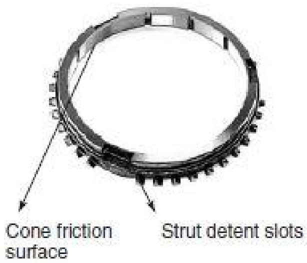


Figure 3.3 Friction ring

3.2.5. Friction cone

Friction cone is usually the integral part of the gear. The friction cone is responsible for generating the cone torque required for the synchronization when the friction ring surrounds it in contact.

3.2.6. Gear locking teeth

The sleeve initiates its move from the neutral position, and it pushes the friction ring, which generates the friction torque and subsequently cone torque when the synchronization process complete. Therefore, the sleeve can freely pass through the ring and complete the gear shifting process.

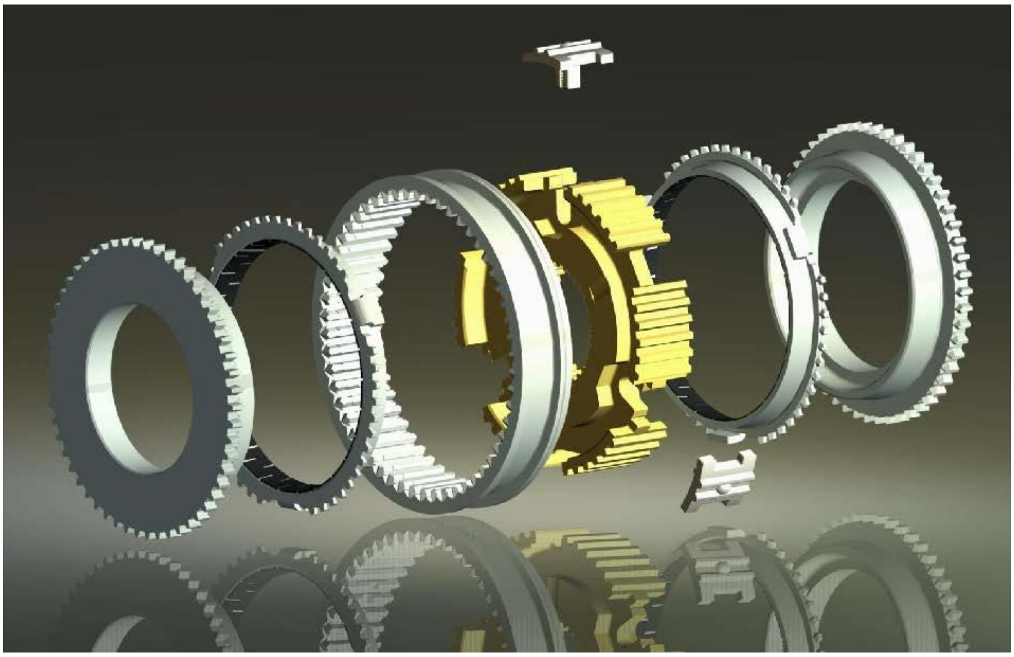


Figure 3.4 synchronizer parts

3.3. The synchronization process

The first position is the neutral position, and no gear is engaged as it can be seen in the picture.

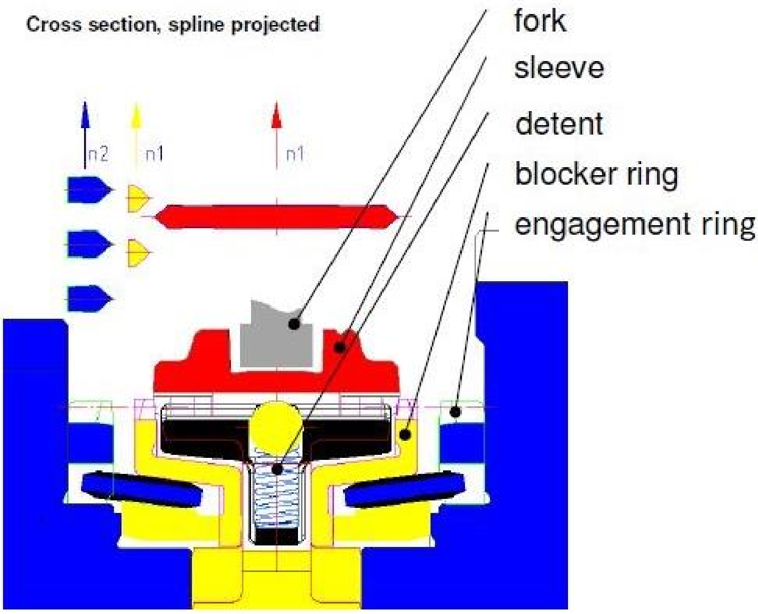


Figure 3.5 neutral position

3.3.1. Pre-synchronization process

The pre-synchronization process starts with the fork moving the sleeve in the axial direction until the detents make contact with the friction ring then the force creates a friction torque in the synchronizer. The friction torque places the blocker ring in the correct position.

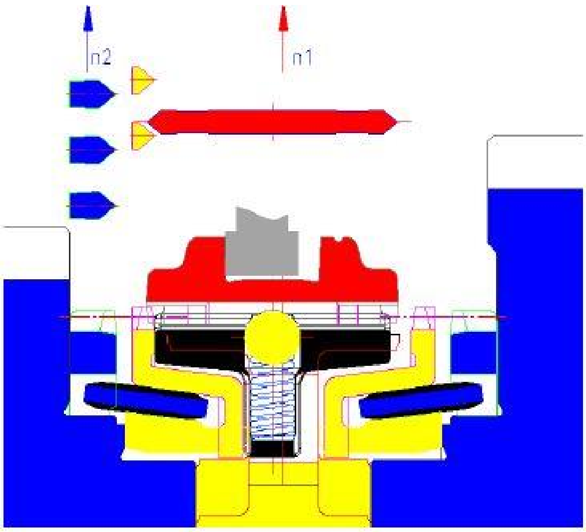


Figure 3.6 Pre-synchronization postion

3.3.2. Blocking position

As the force increases the sleeve move towards blocking teeth of the blocking ring, then the teeth of the sleeve push against the blocking teeth and speed difference reduce so the two speed n_1, n_2 becomes equal.

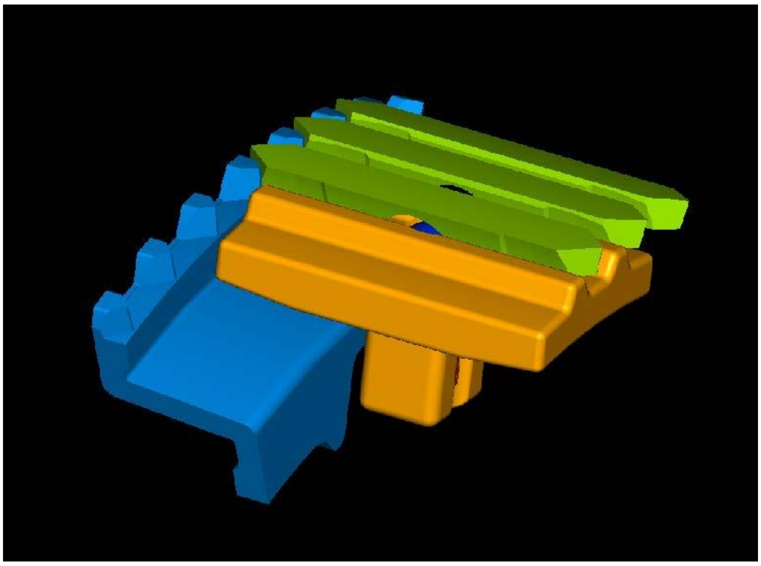


Figure 3.7 blocking position

3.3.3. Blocking release

When the speed difference is null, the sleeve can move forward through the spline of the blocking ring.

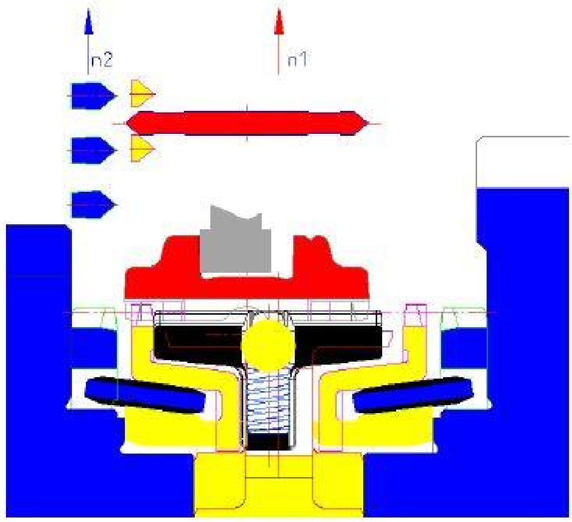


Figure 3.8 blocking release position

3.3.4. Free flight phase

The sleeve moves forward towards the spline of the engagement ring, and in this case, the new speed difference between two speeds can occur again.

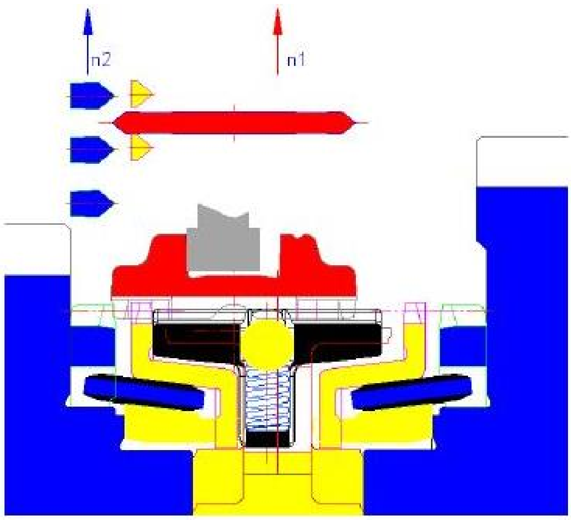


Figure 3.9 Free flight position

3.3.5. Engagement

The sleeve enters into the engagement ring, in this stage, the speed difference can cause bumps.

3.3.6. Gear shifting

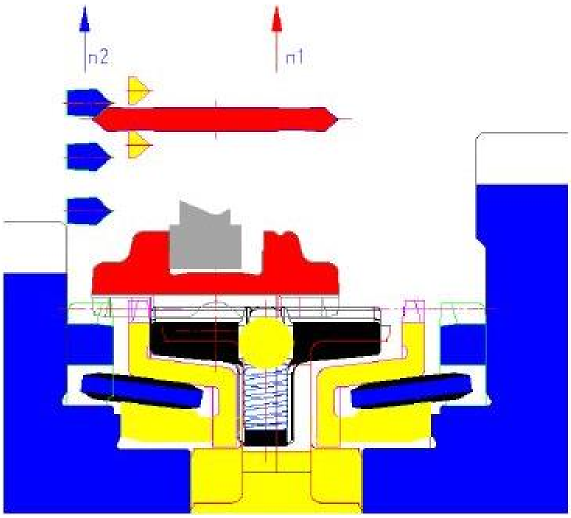


Figure 3.10 Gear shifting position

When the sleeve has completely moved into the engagement ring, the gear and the shifting process is considered as complete.

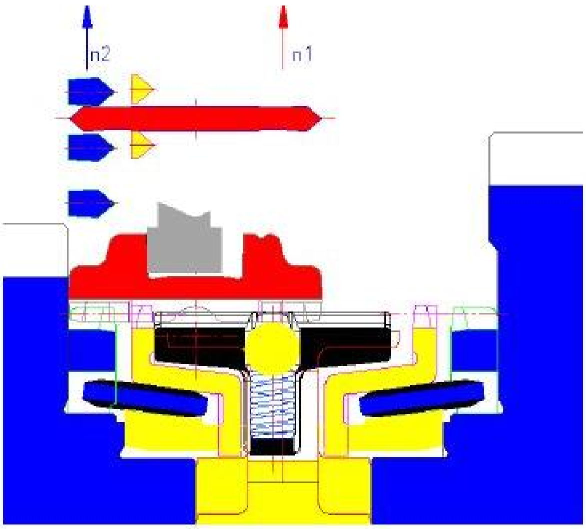


Figure 3.11 complete shifted position

The speed difference Δω between the rotation speed of the gear, the gear to be engaged, and the rotational speed of the shaft on which the gear is mounted is a critical design parameter for sizing the synchronizer and for determining the actual position of the synchronizer. In fact, the single synchronizer can be mounted on the primary shaft or the secondary shaft. In the first instance it can be stated that as the difference in rotational speed Δω increases, the friction losses increase within the synchronizer and the engagement time t_r tends to increase. As a result, as the difference in rotational speed Δω increases, the efficiency of the clutch deteriorates. Since the gear on which the synchronizer is mounted is not integral with the shaft, the synchronizer must adjust the rotation speeds ω of the shaft. For sizing the synchronizer conservatively, it is necessary to calculate the maximum difference in rotation speed $\Delta\omega_{max}$ that the synchronizer must cover during the shift phase. For these reasons, it is necessary to consider a gear engaging time and calculate the difference in rotation speed Δω which occurs in the following operating conditions: the condition of upshifting and condition of downshifting both with the synchronizer positioned on the primary shaft and with the synchronizer placed on the secondary shaft.

3.4. Calculation of the difference in rotation speed Δω during the upshifting from first to second gear

3.4.1. The position of the synchronizer on the first shaft

Before making the gear change, the first gear is the engaged one, so can be written as:

$$\tau_1 = \frac{\omega_{1,input}}{\omega_{1,output}} \qquad (3.1)$$

Where it has been assumed that the rotational speed of the gear mounted on the primary shaft of the first gear $\omega_{1,\ input}$ is equal to the rotational speed of the engine. Therefore, the rotational speed of the gear mounted on the secondary shaft of first gear $\omega_{1,\ output}$ is obtained. Since the synchronizer is mounted on the primary shaft then the rotational speed of the gear mounted on the secondary shaft of second gear $\omega_{2,\ output}$ is equal to the rotational speed of the gear mounted on the secondary shaft of the first gear $\omega_{1,\ output}$:

$$\omega_{1,output} = \omega_{2,output} = \frac{\omega_{engine}}{\tau_1} \qquad (3.2)$$

Consequently, the rotational speed of the gear mounted on the primary shaft of the second gear $\omega_{2,\ input}$ is:

$$\omega_{2,input} = \tau_2 \omega_{2,output} = \frac{\tau_2}{\tau_1}\omega_{engine} \qquad (3.3)$$

Thus, the rotational speed of the gear on the primary shaft for the second gear $\omega_{2,\ input}$ is different from the rotation speed of the primary shaft $\omega_{shaft,\ input}$ because it is not integrated with the shaft.

The difference in rotational speed which must become zero by the synchronizer mounted on the primary shaft during the engagement phase from the first to the second gear:

$$\Delta\omega_{synchronizer} = \left|\omega_{shaft,input} - \omega_{2,input}\right| = \left|\omega_{engine} - \frac{\tau_2}{\tau_1}\omega_{engine}\right| = \left|\omega_{engine}(1 - \frac{\tau_2}{\tau_1})\right|$$

(3.4)

3.4.2. The position of the synchronizer on the second shaft

Before making the gear change, the first gear is the engaged one, so it can be written as:

$$\tau_1 = \frac{\omega_{1,input}}{\omega_{1,output}} = \frac{\omega_{engine}}{\omega_{1,output}} \qquad (3.5)$$

Where it has been assumed that the rotational speed of the gear mounted on the primary shaft of the first gear $\omega_{1,\ input}$ is equal to the rotational speed of the engine ω_{engine}. Therefore, the rotational speed of the gear mounted on the secondary shaft of first gear $\omega_{1,\ output}$ is obtained:

$$\omega_{2,input} = \omega_{1,input} = \omega_{engine} \quad (3.6)$$

Consequently, the rotational speed of the gear mounted on the secondary shaft of the second gear $\omega_{2,output}$ is:

$$\omega_{2,output} = \frac{\omega_{1,input}}{\tau_2} = \frac{\omega_{engine}}{\tau_2} \quad (3.7)$$

The rest of the formula in order to find the difference between the rotational speed will be:

$$\omega_{shaft,input} = \omega_{2,input} = \omega_{1,input} = \omega_{engine} \qquad (3.8)$$

$$\omega_{shaft,ouput} = \omega_{1,output} = \frac{\omega_{engine}}{\tau_1} \quad (3.9)$$

$$\Delta\omega_{synchronizer} = \left|\omega_{shaft,output} - \omega_{2,output}\right| = \left|\frac{\omega_{engine}}{\tau_1} - \frac{\omega_{engine}}{\tau_2}\right| = \left|\omega_{engine}(\frac{1}{\tau_2} - \frac{1}{\tau_1})\right| \quad (3.10)$$

The above calculations are valid in case of the upshifting from first to second gear, in case of downshifting the same logic can be used and implemented in the code, and finally, the maximum between the two cases, the maximum for the upshifting and downshifting will be considered.

3.4.3. Determination of the position of the synchronizer

In a synchromesh, all gears are turning hence the inertia of all gears must be overcome in order to make a gear selection.

The synchronizer can be positioned on the primary shaft or the secondary shaft. Depending on the position of the synchronizer, the equivalent inertia of the transmission elements to be synchronized change. It should be remembered that during synchronization, the synchronizer adjusts the rotation speeds ω of the gear wheel and the shaft on which it is mounted until the rotation speed difference Δω becomes zero at the end of the engagement phase of the desired gear. Furthermore, it is important to remember that the equivalent inertia of the transmission elements which must be synchronized significantly affects the synchronizer's engagement performance as the increase of the equivalent inertia increases the engagement time t_r and consequently worsens the performance of the synchronizer. A good rule for determining the position of the synchronizer is to mount it on the shaft in which the maximum speed difference that will be covered and nullified by the synchronizer is minimized. However, in order to do this, it is necessary that the gears and shaft on which the synchronizer is mounted are not made by one piece but are mounted on a bearing, in this way the possibility of synchronizing and the correct engagement of the desired gear is guaranteed.

3.5. Design procedure

As it discussed before, synchronizer has five major parts to be designed; hub, sleeve, blocking ring, cone, struts. Four of this five parts has been designed and considered in the assembly.

3.5.1. Hub

The hub is a part which is connected to the shaft, and it is similar to external gear. In order to design the profile of the hub, the following steps have been done.

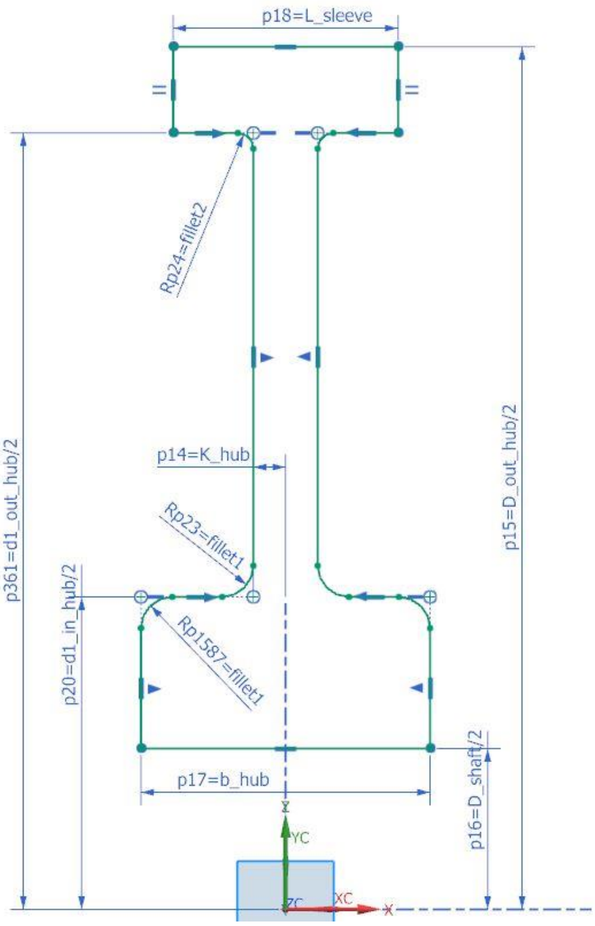


Figure 3.12 Hub sketch

Figure 3.12 is the sketch of the cross-section of the hub, all parameters such as different diameters, b_hub, k_hub, and L_sleeve have been calculated on the code, the only assumed parameters here are the diameter of the fillets which are considered as the 1-5% of the main diameters. This is just an approximation, in order to demonstrate the fillet on the profile and the numbers, can be changed to the desired one if it is necessary.

With the revolve command the profile will be shown in figure 3.13:

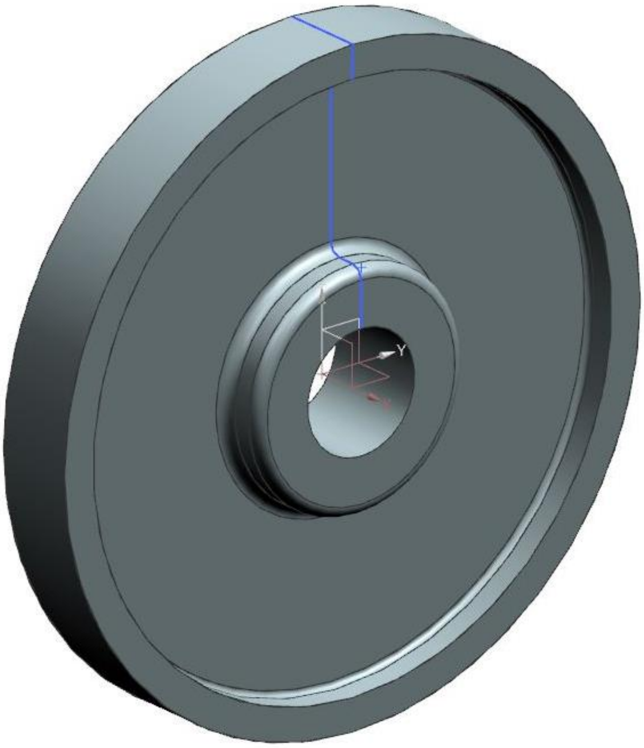


Figure 3.13 Hub extrusion

As it mentioned before the profile of the teeth of the hub can be either straight, trapezoidal or involute. In this project the trapezoidal profile has been taken into account.

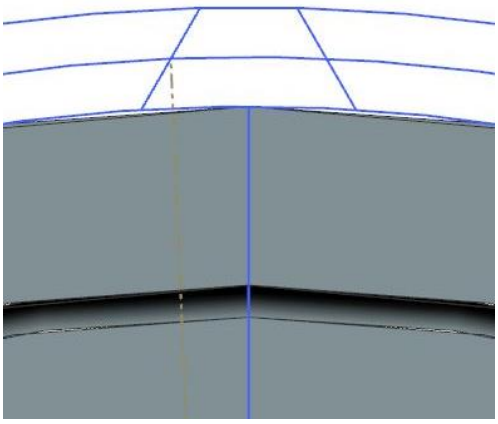


Figure 3.14 Hub's teeth profile

Then the extrusion and the adding the chamfer with the height of half of the module and then using the pattern feature will make the following result in figure 3.15:

Figure 3.15 hub's final Shape

3.5.2. Sleeve

The next component which has to be designed is the sleeve. The position of this component is above the hub and engaged with it, where the profile of the sleeve is similar to an internal gear in order to engage with the hub. Similar to the hub, the first sketch of the sleeve's profile is as following:

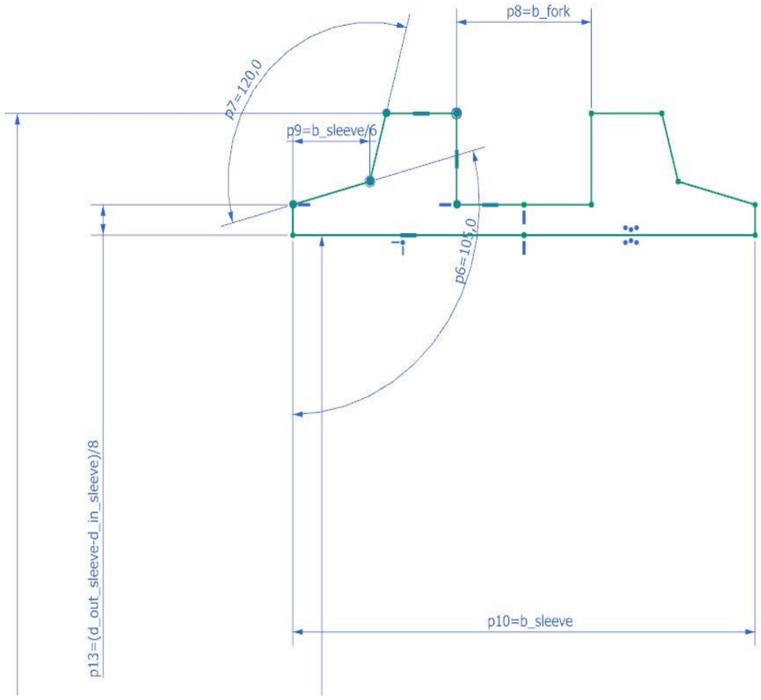


Figure 3.16 sleeve's sketch

The main characteristic of the sleeve is the annular recess shift fork. As it can be seen there are some parametrical values defined based on the main dimensions, the reason is since there was not any calculation for the exact profile, the hypothetical values have been chosen, and these values can be changed easily since they are defined parametrically.

Figure 3.17 sleeve final shape

Then this sketch will be revolved around the rotation axis, and the internal teeth with the trapezoidal profile will be added, and similar to the hub profile the above result will be obtained.

3.5.3. Blocking ring

Blocking ring is made of bronze and is the next component that has been considered to be designed in this project. Similar to the other component of the synchronizer, the sketch of the blocking ring without the teeth is drawn.

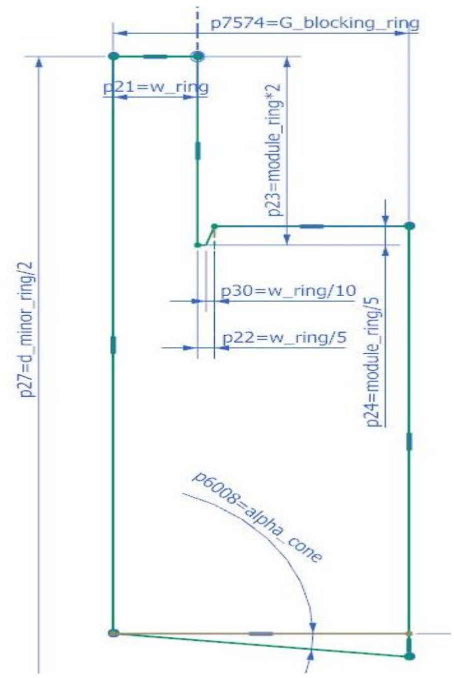


Figure 3.18 Blocking Ring's sketch

As it can be seen in the design, there is some vital parameter such as the external diameter, cone angle, and the width that defines the overall geometry. On the other hand, some other less crucial parameters are taken into account based on the main parameters.

After revolving the sketch, the trapezoidal profile of the teeth has been added, and in order to obtain the chamfer angle, two angled plane has been trimmed the tooth.

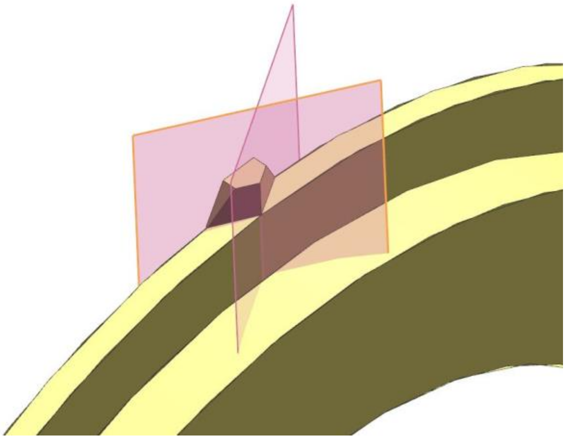


Figure 3.19 Blocking ring's teeth profile

Since the tooth and the ring shape of the blocking ring are two different objects during design, they need to be connected and united through the united command in order to have the possibility of blending edges.

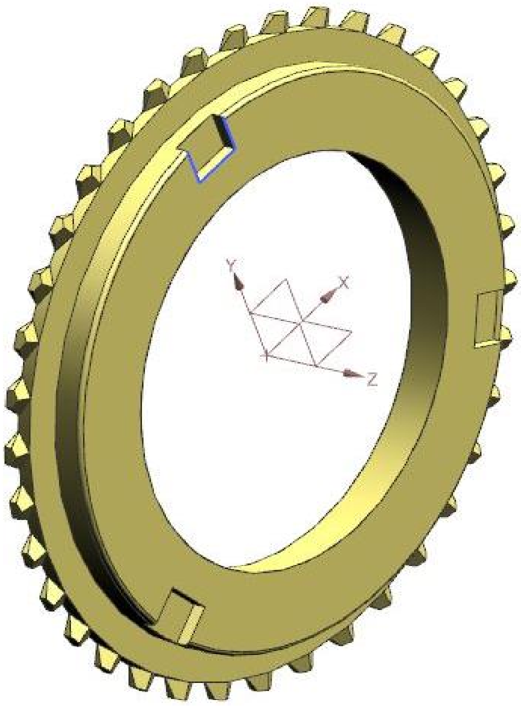


Figure 3.20 Blocking ring's final profile

As it can be seen, the final shape of the locking gear is shown above; three grooves are also subtracted from the feature in order to demonstrate the slot for the strut detents.

3.5.4. Friction Cone

As it said before, the friction cone can be either an integrated part of the gear or a separated part connected with the gear. In this project, the friction cone is an integral part of the gear. This component is made of a ring with a profile similar to the blocking ring and a cone which is the main part to create the required friction.

Since the friction is an integral part of the gear, therefore it should be noted that in the file regarding the design of the gear, three conditions have been implemented. The gear can be connected to the shaft with either the spline or the press fit (or even the integral part of the shaft where in this project its design is considered similar to press fit. If the synchronizer exists, then there is a symbolic needle bearing which connects loosely connects the gear to the shaft, and it can rotate freely.

The sketch of the cone is as following, there are some main values which are extracted from the Matlab code, and there are some other less crucial parameters which are assumed based on the main parameters.

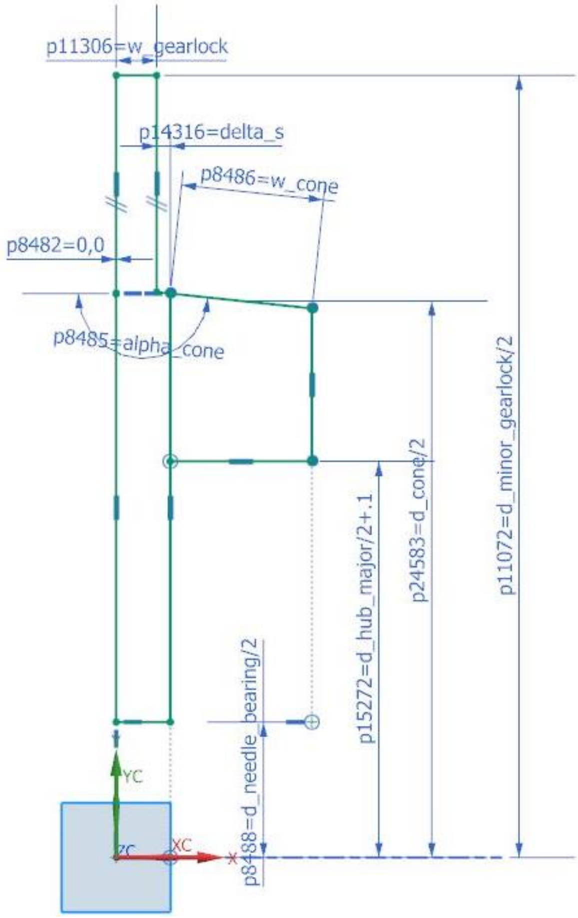


Figure 3.21 Gear lock sketch

With revolving and adding the profile of the locking teeth of similar to the profile of the blocking ring, the side view of the friction cone with locking teeth will be demonstrated as follows:

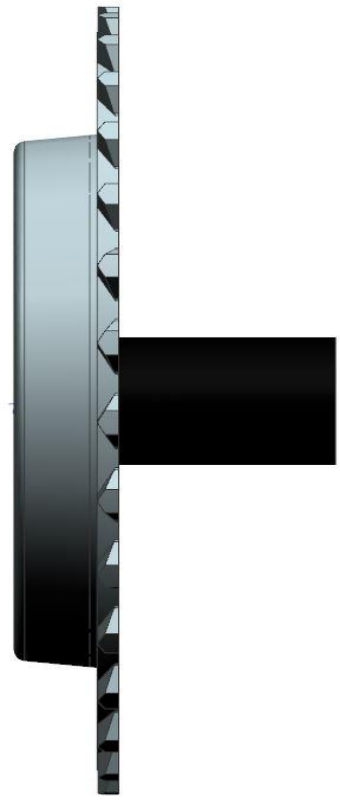


Figure 3.22 synchronizer's cone and needle bearing

Note that the cylinder part at the right part is a simple demonstration of the needle bearing. This part is an integrated part of the gear, in the results chapter the whole profile of the gear with the cone will be demonstrated.

4. Bearing

The role of the bearing is to support the moving components with respect to each other and transfer the forces to the housing. The most common used bearing for the automotive applications is rolling bearing. Shafts usually have one bearing at each end. Therefore two bearings in total are used.

Since rolling bearings are subjected to loads during operation, they must be dimensioned, and it must be checked that they are suitable for the application. If the gears mounted on the driving shaft are spur gear, then the bearings are subjected only to radial forces. On the other hand, if the gear is helical, then the bearings are subjected to both forces in the radial and the axial direction. For the sizing of the bearings that are mounted both on the automotive shafts and on the industrial transmission shafts, the catalog of the SKF Group, the world leader in the production of rolling bearings, has been utilized.

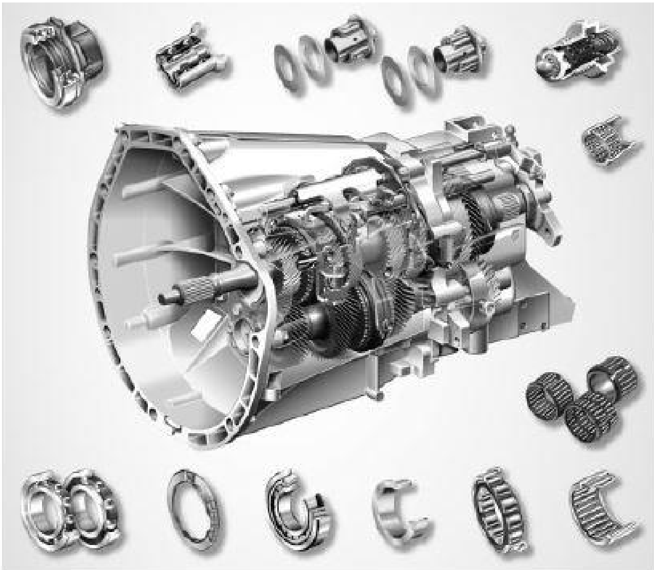


Figure 4.1 Types of bearing construction in a typical manual transmission

Transmission bearings are designed specifically for the respective transmission and the installation location. It is therefore beneficial if the transmission manufacturer—in some cases, the vehicle manufacturer themselves—works together with the bearing supplier from an early design stage. If the two parties define the space required for the bearing arrangement together, technically optimum solutions that also offer high efficiency are possible, for example, by planning the assembly flange as part of the bearing housing. The process of bearing development should always begin with a precise analysis of the entire transmission. In addition to the design conditions and the forces and torques at work, the oil supply in the transmission must also be considered. Only then should a specific bearing concept and its design implementation be selected. Therefore, it is not possible to assign specific bearing construction types to applications generally.

4.1. Different type of the rolling bearing

4.1.1. Ball bearing

Ball bearings use balls as the rolling element as their name indicate. Today, transmission manufacturers use angular contact and groove ball bearings. They are mostly different in the arrangement of the running surfaces.

4.1.1.1. Deep groove ball bearing

It is the most common and classic type of ball bearing. In this type of the ball bearings, the balls move in the groove between the outer ring and the inner ring. They are suitable for application where radial force is the dominated force. The axial force can be tolerated up to 10% of the radial force. So higher axial for can reduce the bearing life.

Figure 4.2 Structure of a deep groove ball bearing

Due to the small contact surface of the ball bearings, they consider suitable for guiding the shaft as locating bearings, but this fact can reduce the threshold of their load capacity. It means that deep groove ball bearing can be mainly used in smaller transmission with the input torque no more than 250 Nm.

4.1.1.2. Angular contact ball bearings

In this type of the bearings, the running surface of the outer and inner ring are diagonally angled with respect to the bearing axis. It means that a greater axial force can be tolerated. On the other hand, with a similar dimension, more balls can be positioned which causes the increase of the load capacity. The individual angular contact ball bearings can be used if the there is a minimum axial load exist. Therefore the contact ball bearings are usually mounted in pairs and mirror-inverted. This configuration can also increase the load capacity.

In transmission, angular contact ball bearings are often used in medium-sized load cases where the deep groove ball bearings are not suitable.

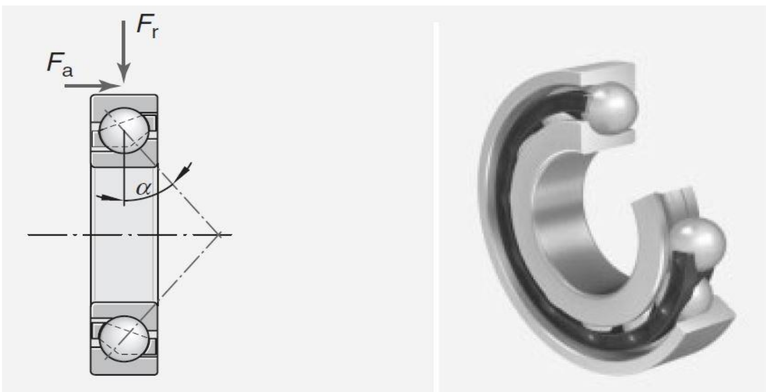


Figure 4.3 Structure of an angular contact ball bearing

4.1.2. Roller bearings

Unlike the ball bearing where the rolling elements have point-type contact to the running surface, the contact to the running surface in roller bearings is linear. This type of configuration leads to increase the static load capacity, and also the axial load capacity can be higher.

4.1.2.1. Cylindrical roller bearings

Cylindrical roller bearing concerning their main loading can be designed either radial or axial. With a limited range, also the axial force can be transferred by a radial bearing through the design of the end retainer in the inner and the outer ring. Radial cylindrical roller bearings are often used in the non-locating bearings arrangement of the pinion shaft in the transmission.

4.1.2.2. Radial and axial needle roller bearings

Needle roller bearings are useful where the height is limited. These are a type of the cylindrical bearing in which the length to diameter ratio is higher than 2.5.

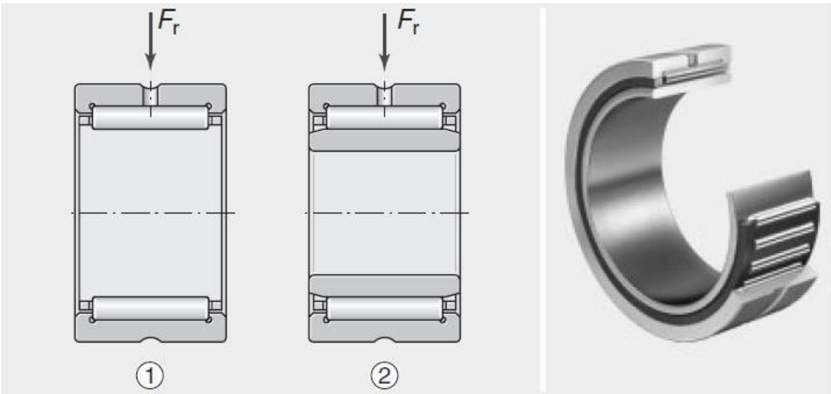


Figure 4.4 Structure of a radial needle roller bearing. 1- Needle bearing without inner race 2- Needle bearing with an inner race

4.1.2.3. Needle roller and cage assemblies

These kinds of bearing have no inner or outer ring and the running elements are placed directly on the surrounding component. The size of the rolling element can be very small compared to the diameter of the bearing. Needle roller bearings are mostly used for the bearing arrangement in constant mesh gears.

When gears are not engaged, the load imposed on the bearing is uneven due to the vibrations that happen. This feature causes the high load on the cage, and therefore proper calculation must be used to keep the cage protected against the fractures. It is worth mentioning when the gears are engaged; the static loads are very high since there is no relative motion between the shaft and the gear. Thus, with the suitable design of the clearance between the cage and the shaft, it is possible that the roller endures a slight relative motion, and so preventing further damage.

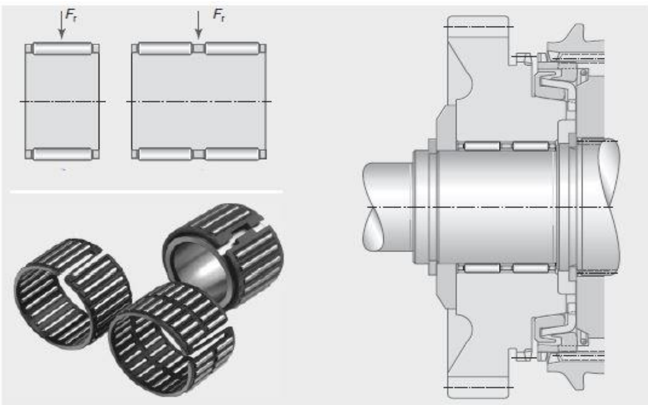


Figure 4.5 Different types of needle roller and cage assemblies as used in speed gears.

5. Assembly and layouts

In the previous chapters, the components that have been designed are introduced. In this chapter, the layout and the assembly that has been considered in this project will be defined.

5.1. Automotive layout

Generally, the passenger's cars layouts, depending on the position of the engine and its orientation can have different configurations.

In this project, the layout for the automotive gearbox is a single stage, 5-speed gearbox. Since the designing components are the ones exclusively inside the gearbox, not the external part, hence the orientation of the engine can be either transversal or longitudinal.

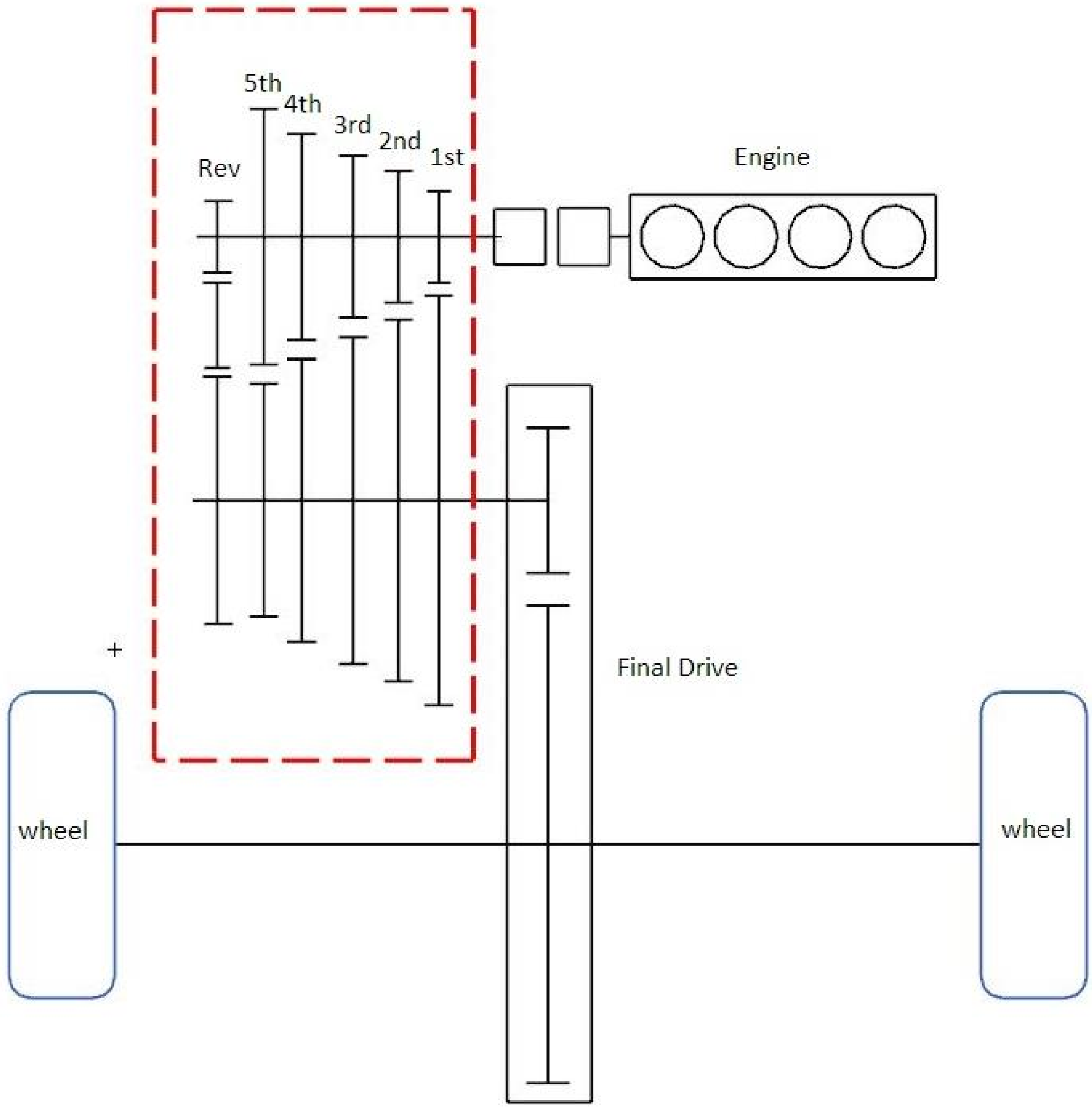


Figure 5.1 Automotive layout

5.1.1. The position of the reverse gear

There are various configurations for designing the reverse gear. The required reversal of the rotational direction of the output shaft is usually achieved by implementing an idle gear into the power flow. The general rule for toothed gearing is that increasing or reducing the number of gears by one, reverse the direction of rotation of the output shaft. In the following, there are six possible configurations that can be taken into consideration.

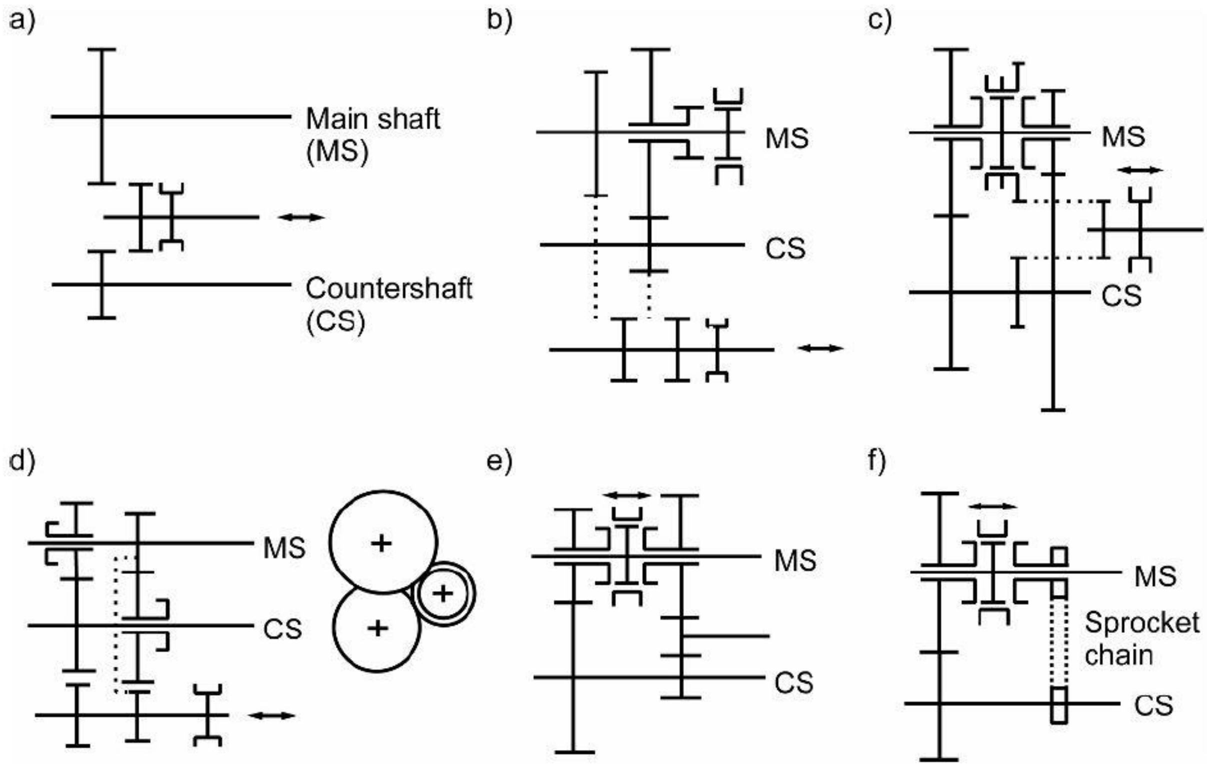


Figure 5.2 Six different configurations for reverse gear

a) An axial sliding gear is inserted between each fixed wheel of the main shaft and the countershaft
b) shiftable shaft with two pinions between a reverse gearwheel of the main shaft and a forward gearwheel of the countershaft
c) the sliding gear is inserted between a fixed wheel of the countershaft and a toothed sliding sleeve of a synchronizer on the main shaft
d) sliding shaft with two pinions between a forward gearwheel of the main shaft and a forward gearwheel of the countershaft
e) reverse gear with intermediate pinion constantly engaged, shifting with sliding sleeve
f) reverse gear using tooth-type chain, shifting with sliding sleeve

In this project, the configuration “e” has been considered. Since the idle shaft can be placed in an angle with respect to the input and output shaft, an algorithm has been defined in the code that collects all the possible geometrical options and then in the later stage. It will validate the permissible stresses that are imposed on the gear from the smallest possible configuration to the largest and chooses the configuration with the minimum required space.

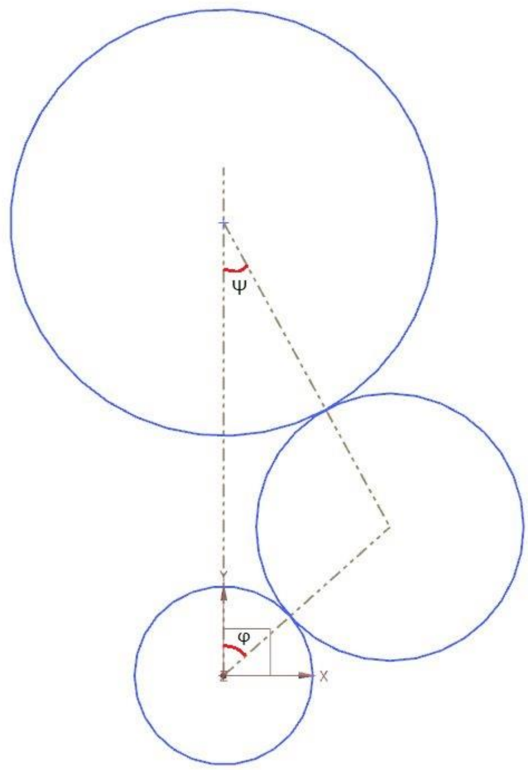


Figure 5.3 The configuration for the reverse gear

To achieve this goal, four for-cycle has been implemented inside the MATLAB code, the first two are related to the center of the idle gear. The third four cycle is related to the diameter of the pinion the fourth one represents the diameter of the idle gear. Since the center of the pinion is considered at the center of the coordinate system, then the following calculation can validate if the idle gear is positioned correctly on the reverse configuration or not.

$$\left|\sqrt{(x_{idle}-x_{pinion})^2+(y_{idle}-y_{pinion})^2}-(r_{pinion}+r_{idle})\right|\leq tolerance \quad (5.1)$$

$$\left|\sqrt{(x_{idle}-x_{gear})^2+(y_{idle}-y_{gear})^2}-(r_{pinion}+r_{gear})\right|\leq tolerance \quad (5.2)$$

$$r_{pinion}+r_{gear}\leq 0.85(\text{Shafts-distance}) \quad (5.3)$$

The last condition is a safe approximation in order to avoid the collision of the teeth of the pinion and the gear.

So if the above conditions are met, the results which are the diameters of the three gears and the two coordinates of the idle gear are stored and will be used to find the angle of φ and Ψ which define the correct position of the idle gear.

5.2. Industrial Layout

The parametrically designed industrial transmissions cover four possible configurations of an industrial speed reducer. The implemented configurations predict a maximum number of reduction stages equal to four. The diagram of a four-stage industrial transmission is shown in the following figure:

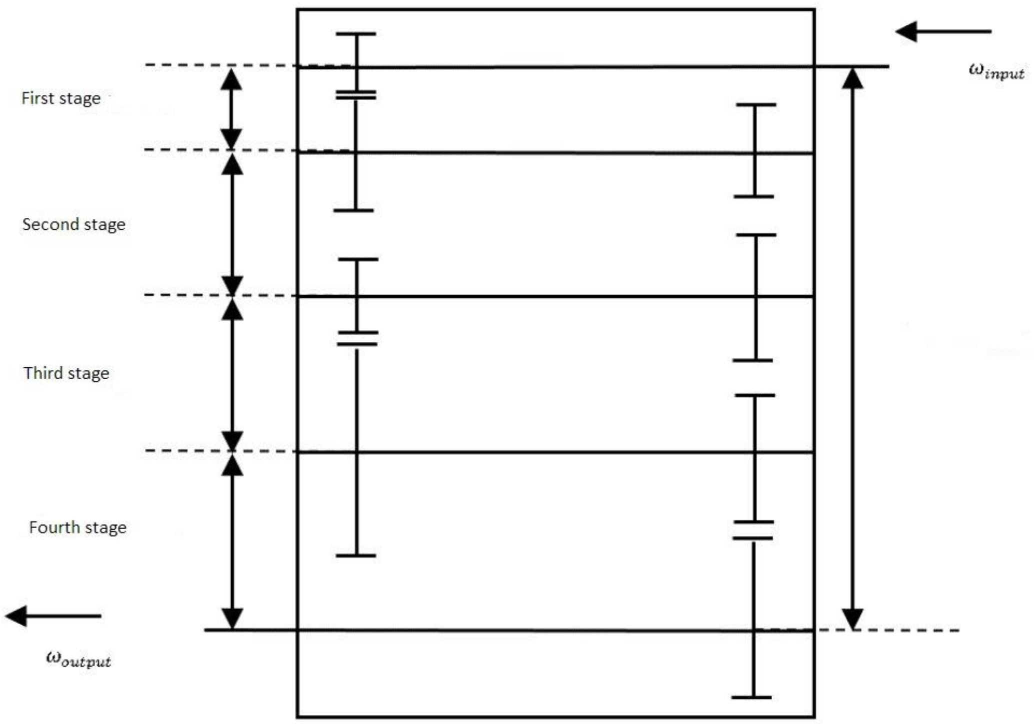


Figure 5.4 Industrial layout

In order to obtain other layouts with fewer stages, the upper stages will be suppressed. So the layout with one, two and three stages will be created.

5.3. Assembly

The important part of this chapter is assembling the designed component from the previous chapters as a layout. During the calculation phase, it is essential to consider the parts that are integrated with the shaft in order to avoid assembly problem in this stage.

5.3.1. Assembly on the NX

NX assembly is a part file that contains the induvial parts. They are added to the assembly in a way that the parts are virtually in the assembly file and at the same time they are linked to the original parts. So every change in the original part can be reflected in the assembly as well. This feature avoids the need for duplication of the parts, and so saves the memory.

5.3.2. Assembly approaches

There are three main ways to develop the assembly. NX can work with any of the following assembly modeling methodologies:

5.3.2.1. Top-down approach

In this approach, first the assembly part is created, and then the individual parts are modeled inside the assembly. The details of each individual parts are added as the project gets further along. This approach is usually used for larger projects.

5.3.2.2. Bottom-up approach

In this approach the components are created traditionally then they are added to the assembly section. This approach is typically used for a smaller project with a few team members.

5.3.2.3. The Middle-out approach

This modeling approach is a mixture of Top-down and the bottom-up approach. This type of the assembly is usually constructed with most of the parts already created, and additional parts are designed and created using the assembly for construction information.

5.3.2.4. Constraints

An important part of the assembly is the definition of the constraints. Particularly in the parametric modeling, constraints have a crucial role in order to maintain the configuration after any change in the components. The main constraints in the NX software are as following:

Touch/Align: Planar objects selected to align will be coplanar, but the normal to the planes will point in the same direction. Centerlines of cylindrical objects will be in line with each other.

Concentric: Constraints circular or elliptical edges of two components, so the centers are coincident, and the planes of the edges are coplanar.

Distance: This establishes a +/- distance (offset) value between two objects

Parallel: Objects selected will be parallel to each other.

Perpendicular: Objects selected will be perpendicular to each other.

Bond: Creates a weld and welds components together to move as a single object.

Center: Objects will be centered between other objects, i.e., locating a cylinder along a slot and centering the cylinder in the slot.

Angle: This fixes a constant angle between the two object entities chosen on the components to be assembled.

5.3.2.5. Interpart Expressions

Another vital part of the assembly in the parametric modeling is the interpart expression. As it is already noted before, in order to make parametric modeling it is important to define the relationship between the parameters in each individual component. Then these the input should be defined in the assembly, and they need to be connected through the interpart expression. It means that whatever changes in the input of the assembly, it will be reflected on the individual part as well.

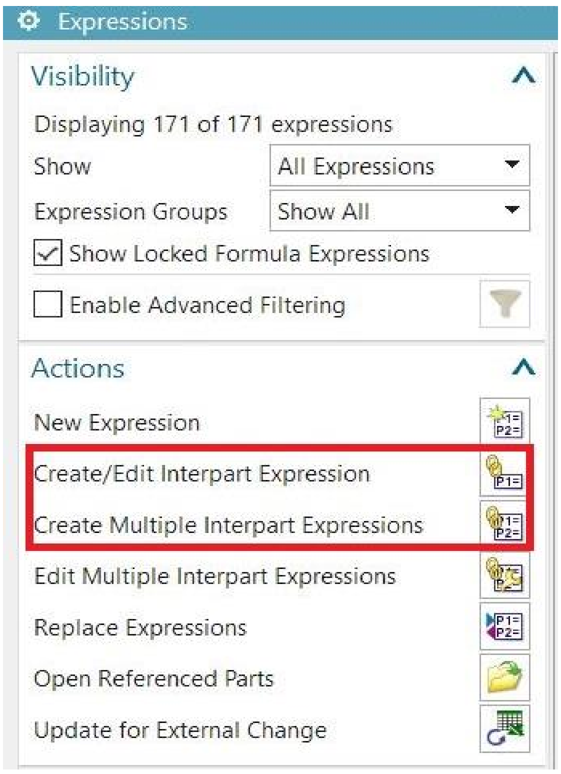


Figure 5.5 Create an Inter-part expression

The Interpart expressions can be defined from the expression windows as is shown above. Since in this project the aim is to have all the inputs at the same window, all the inputs will be added to the assembly expression window. Then each input will be connected to its proper expression in each component expression window.

	↑ Name	Formula	Value	Type	Source
1	Default Group				
2				Number	
3	helix_angle_1	(Interpart)	19.7	Number	"Gearbox"::helix_angle_1
4	Helix_hand_input_1	(Interpart)	"right"	String	"Gearbox"::Helix_hand_input_1
5	module_1	(Interpart)	2.5	Number	"Gearbox"::module_1
6	pressure_angle_1	(Interpart)	14.5	Number	"Gearbox"::pressure_angle_1
7	width_gear_1	(Interpart)	30	Number	"Gearbox"::width_gear_1
8	z_input_1	(Interpart)	19	Number	"Gearbox"::z_input_1

Figure 5.6 Interpart expression

This is an example of the interpart expressions belong to first input gear. As it can be seen in the Formula column is written “Interpart,” and this input cannot be modified directly from the part. In the source column, the source that the expression comes from is demonstrated.

5.3.3. The connection between MATLAB code and NX

The state of art of this project is the connection between a MATLAB code and the NX CAD. This possibility was feasible with an intermediate program, Excel, which is supported by both programs.

After doing all the calculation through the MATLAB, the corresponding output will be gathered in some excel sheets, and those stored data will be used as an input for the NX software.

NX has the possibility to read either an individual cell or a column/row of an excel. In case of the single data, the value of the cell can be stored either as number or string depending on its value.

In case of collecting a row or column, the type of the expression is a List. If the data are stored as a list, each member of the list need to be assumed to its own corresponding expression

↑ Name	Formula	Value	Type
synchronizer3_list_excel	ug_read_list(excel_source+"\GEARBOX_DATA_synchro.xlsx", "b15", ...	{6,120,77.2746885,109.549377,3,10.55783694,1,1.10359447,1...	List
Synchronizer3			
alpha_cone_ring_3	nth(1,synchronizer3_list_excel)	6	Number
b_hub_3	nth(21,synchronizer3_list_excel)	27	Number
Beta_ring_3	nth(2,synchronizer3_list_excel)	120	Number
d1_in_hub_3	nth(20,synchronizer3_list_excel)	45.12492426	Number
d_cone_ring_3	nth(3,synchronizer3_list_excel)	77.2746885	Number
d_needle_bearing_3	30	30	Number
d_pitch_hub_3	d_pitch_ring_3	109.549377	Number
d_pitch_ring_3	nth(4,synchronizer3_list_excel)	109.549377	Number
G_ring_3	nth(9,synchronizer3_list_excel)	10.5	Number
K_hub_3	nth(10,synchronizer3_list_excel)	3	Number
L_cone_ring_3	nth(6,synchronizer3_list_excel)	10.55783694	Number
L_hub_3	nth(22,synchronizer3_list_excel)	21	Number
w_cone_3	nth(6,synchronizer3_list_excel)	10.55783694	Number
w_ring_3	nth(16,synchronizer3_list_excel)	3	Number
z_hub_3	nth(15,synchronizer3_list_excel)	30	Number
z_ring_3	nth(15,synchronizer3_list_excel)	30	Number

Figure 5.7 Excel connection with NX

This table is an example for the data allocation from the list of parameters that are taken from the excel as it can be seen after reading the list of value from the excel on the first row they will be assigned to their corresponding parameter with “nth()” command.

5.3.4. Designing a figurative case

The case for this project is just a rough design in order to estimate the minimum overall space needed for the gearbox. Therefore, its size needs to be altered according to the other component in order to represent the minimum required space for the components.

5.3.5. Wave geometry linker

WAVE Geometry Linker is a means of associatively linking (copying)geometry from one part into another. These can be bodies, faces, curves, sketches, routing objects, etc. which can then be used in the receiving part to design from or refer to. In this project wave Geometry linker is used in order to create a parametric case. Since changing the dimension of each component can affect every other component then it is beneficial to use the wave geometry linker as a tool to simplify the problem. Hence, the best approach to model the case is using the Wave geometry linker.

5.3.6. Product template studio (PTS)

Product template studio modularizes parametric designs into easily reusable templates without dealing the complicated codes. A template starts with a good parametric model, and the author of the template adds a simple user interface that will help the end user change the parameters of the design. The input can be any type, either the numerical input or different choices that have been already defined in the design. The advantage is that the end user does not have to interact with the overall design environment and it only deals with the simplified and user-friendly interface. So, the simplified user interface allows for easy interaction with sophisticated design and promotes consistency and reuse.

In this project product template, the studio has been used in order to personalize the geometry of the gears. Since the calculation that has been made on the MATLAB code is not the unique solution and it is just the first solution that satisfies the requirements, therefore it is beneficial to give the possibility to the end user to change the design parameters as they have been calculated by himself.

In NX 11 the user template studio is integrated inside the software, and it can be reached from the NX menu. This is where product template studio window can be open.

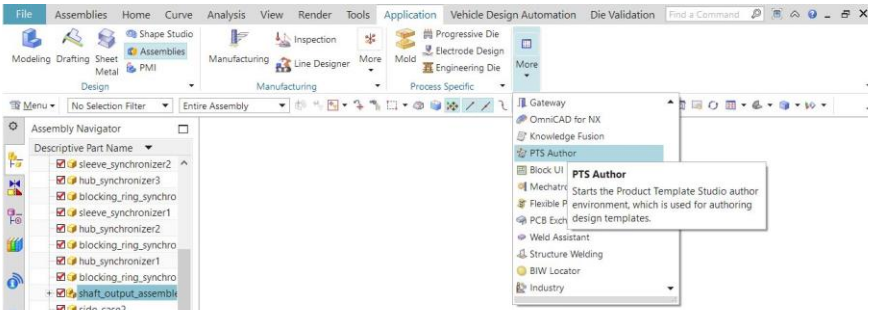


Figure 5.8 product template studio

After opening the PTS, there are some options available to create the user interface. The Controls and the Expression are the most used options to make the user interface.

In the Controls section, there are some basic options like separator and an action button are placed in this section. In the Expression section, the list of all the expressions used in the assembly is present so the desired expressions which are the input of can be selected in the user interface.

For every single speed gear, the following data are present, and they can be changed. There is a button defined at the end of each gear parameters in order to reset the values to default values that are calculated with the MATLAB code.

The same Tab is available for the Reverse gear with some additional parameters which are the number of the teeth of the idle gear and angles of φ and Ψ that help to define the position of the idle gear with respect to the input and output shaft.

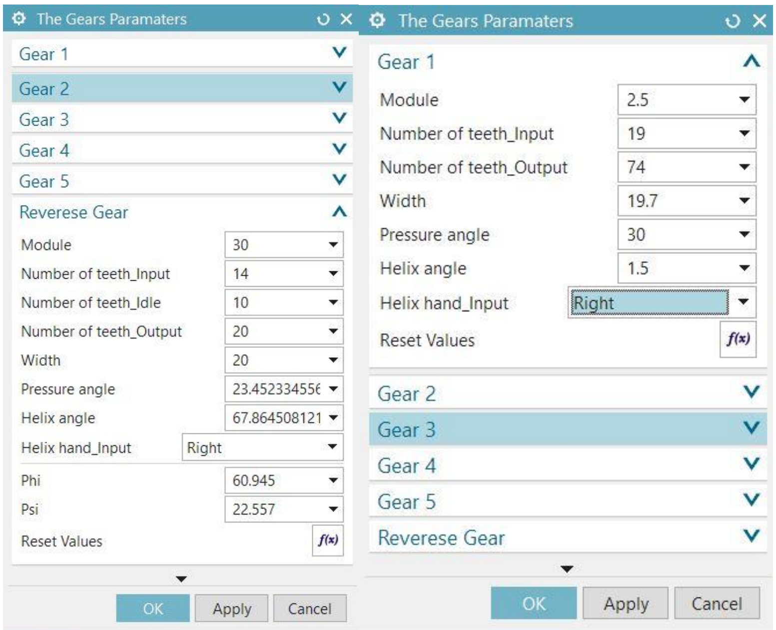


Figure 5.9 User interface for the PTS

5.3.6.1. Visual rules

In this section of the PTS, the different rules can be defined in order to make a change in the parts or assembly. Some of the rules that can be defined are consist of adding, removing or replacing components, set expressions, perform a numeric calculation or even run another visual rule.

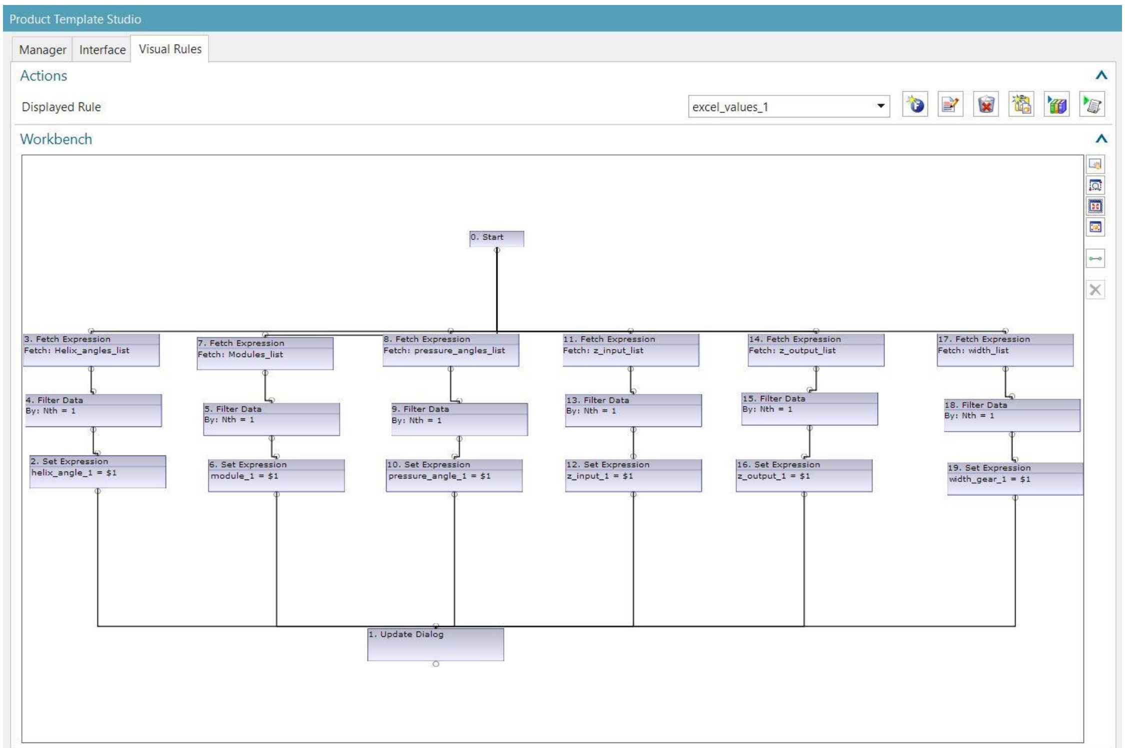


Figure 5.10 An example of Visual rules

6. Results and discussion

In this chapter, a case study for each automotive and industrial gear was taken into account. There are some limited inputs that are given to the code, and the required output is obtained in order to dimension the gearboxes.

6.1. Automotive gearbox

The case of the automotive gearbox is 5-speed gearbox plus the reverse. The following inputs are given to the Automotive gearbox code:

6.1.1. Input data

Table 6.1 Gear data

Gear Ratio	First Gear	3.909
	Second Gear	2.158
	Third Gear	1.345
	Fourth Gear	0.974
	Fifth Gear	0.829
	Reverse Gear	3.818
	The distance between the input and output shaft [mm]	120
	Tolerance for the shaft distances	0.1%
	Tolerance for the gear ratios	1%
Material	Material	17CrNiMo6
	Mechanical treatment	Tempered Case-hardening

Table 6.2 Engine data

Maximum torque [Nm]	300
Rotational speed at the maximum torque [rpm]	2000
Maximum rotational speed [rpm]	5000
Minimum rotational speed [rpm]	600

Table 6.3 Shaft data

Material	C45
Damping factor	5%
Mechanical Treatment	Surface Hardening

Table 6.4 Synchronizer data

Minimum service life [year]	15
Average daily use hour [hour]	5

6.1.2. Output data

Table 6.5 Gear data

	First	Second	Third	Fourth	Fifth	Reverse	Idle
Module [mm]	2.5	2.5	2.5	2.5	2.5	2.5	2.5
Number of teeth_input	19	30	40	48	52	14	17
Number of teeth_output	74	65	54	47	43	53	-
Width [mm]	30	30	30	30	30	30	30
Pressure angle [°]	14.5	14.5	14.5	14.5	14.5	20	20
Helix angle [°]	14.3	11	10.2	10.1	11.1	34.2	34.2

Table 6.6 Synchronizers data

	First	Second	Third
Number of cones	3	3	3
The angle of cone [°]	6	6	6
Chamfer angle of the teeth [°]	120	120	120
Average diameter [mm]	118	91.1	102.6
Maximum diameter [mm]	190.9	137.2	160.2
Friction torque [Nm]	152.3	117.6	132.5
Engaging time [s]	186	75	593
Number of teeth	40	34	37
Approximate width [mm]	36	36	36

Table 6.7 Splines data

	First Input	Second Input	Third Input	Fourth Input	Fifth Input
Number of Teeth	62	62	62	62	62
Pressure Angle [°]	30	30	30	30	30
Module[mm]	0.5	0.5	0.5	0.5	0.5

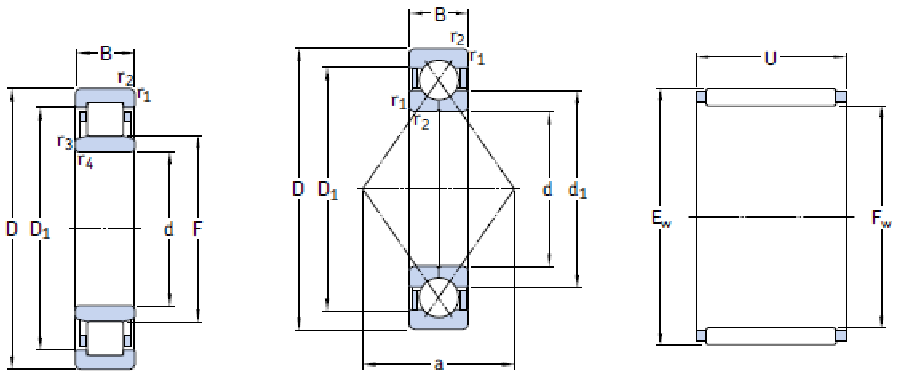


Table 6.8 Bearing for the gear connected to the synchronizers data

	First	Second	Third	Fourth	Fifth	Reverse
Type of bearing	Single row cylindrical roller bearings	Single row cylindrical roller bearings	Single row cylindrical roller bearings	Four-point contact ball bearings	Four-point contact ball bearings	Single row cylindrical roller bearings
d [mm]	65	65	45	45	45	45
D [mm]	100	100	75	85	85	85
B [mm]	18	18	16	19	19	23
d_1 [mm]	-	-	-	54.5	54.5	-

D_1 [mm]	88.5	88.5	65.3	72	72	73
Dynamic load rating [kN]	62.7	62.7	44.6	63	63	85
Static load rating [kN]	81.5	81.5	52	56	56	81.5
Fatigue Load limit [kN]	9.8	9.8	6.3	2.36	2.36	10.6
Designations	NU1013ECP	NU1013ECP	NU1009ECP	*QJ209MA	*QJ209MA	*NU2209ECP

Table 6.9 Bearing for the shafts data

	Left bearing on the input shaft	Right bearing on the input shaft	Left bearing on the output shaft	Right bearing on the output shaft	Left bearing on the idle shaft	Right bearing on the idle shaft
Type of bearing	Needle roller and cage assemblies	Single row cylindrical roller bearings	Single row cylindrical roller bearings	Single row cylindrical roller bearings	Needle roller and cage assemblies	Needle roller and cage assemblies
d (F_w) [mm]	28	20	45	40	40	42
D (E_w) [mm]	33	52	75	90	48	47
B [mm]	13	15	16	23	20	13
d_1 [mm]	-	31.2	-	57.5	-	-
D_1 [mm]	-	42.4	65.3	75.6	-	-
Dynamic load rating [kN]	14.7	35.5	44.6	93	34.7	17.2
Static load rating [kN]	24.5	26	52	78	58.5	33.5
Fatigue Load limit [kN]	2.85	3.25	6.3	10.2	7.35	4
Designations	K28x33x13	*NU304ECP	NU1009ECP	*NU308ECP	K40x48x20	K42x47x13

It needs to be mentioned that some of the chosen bearings are not able to tolerate the axial force, this limitation needs to be taken into account and the involved bearings should be replaced with another bearing with similar characteristics.

6.1.3. Static analysis of the shafts

Rigid shafts do not bend or deflect too much under the bending moments. These shafts should be designed based on the permissible lateral deflection. When a shaft that supporting gear is deflected, it affects the meshing of the gear teeth, and it is not proper. Additionally, the misalignment between the journal and the bearing causes wear at the bearing and gear surface at the early stages. So the allowable deflection for the transmission shaft is between 0.1% to 0.3% of the length of the shaft. In this case of study, the shaft length is 0.34 m, so the maximum acceptable deflection should be between $34 \times 10^{-4} and\ 10^{-3}$.

The following results are:

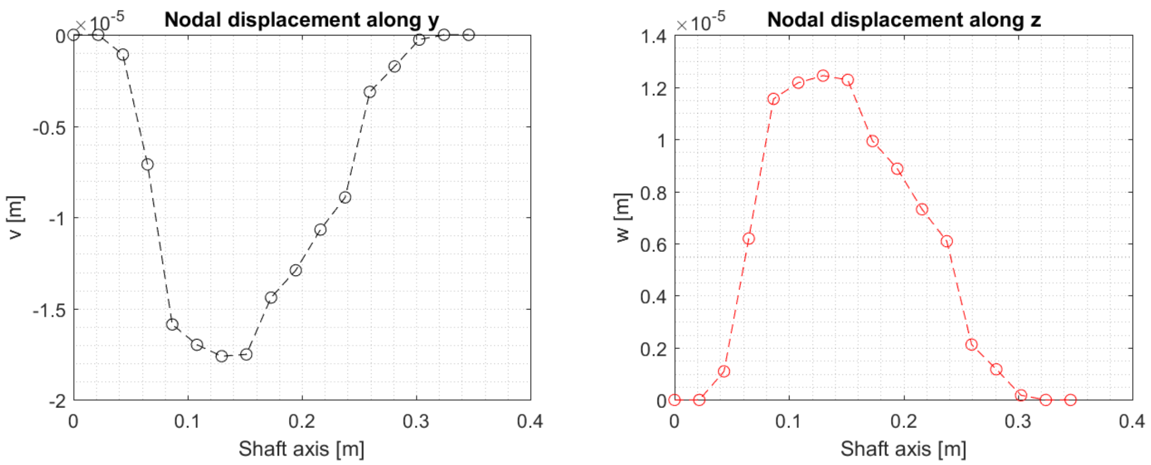


Figure 6.1 First shaft deflection

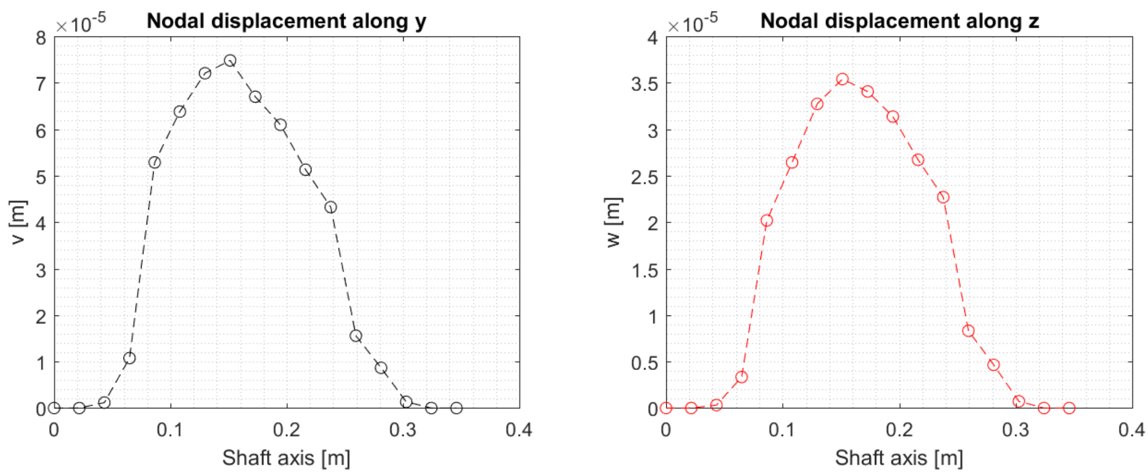


Figure 6.2 Second shaft deflection

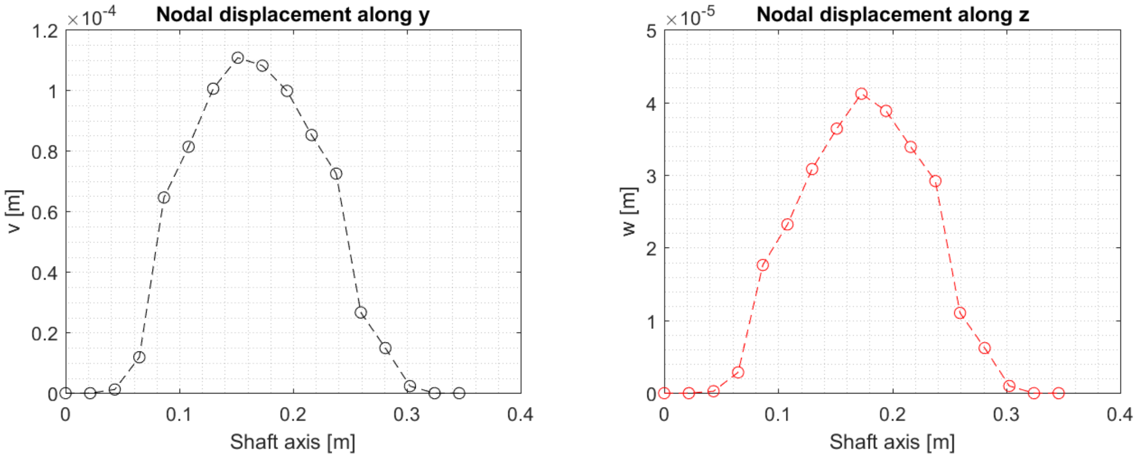


Figure 6.3 Third shaft deflection

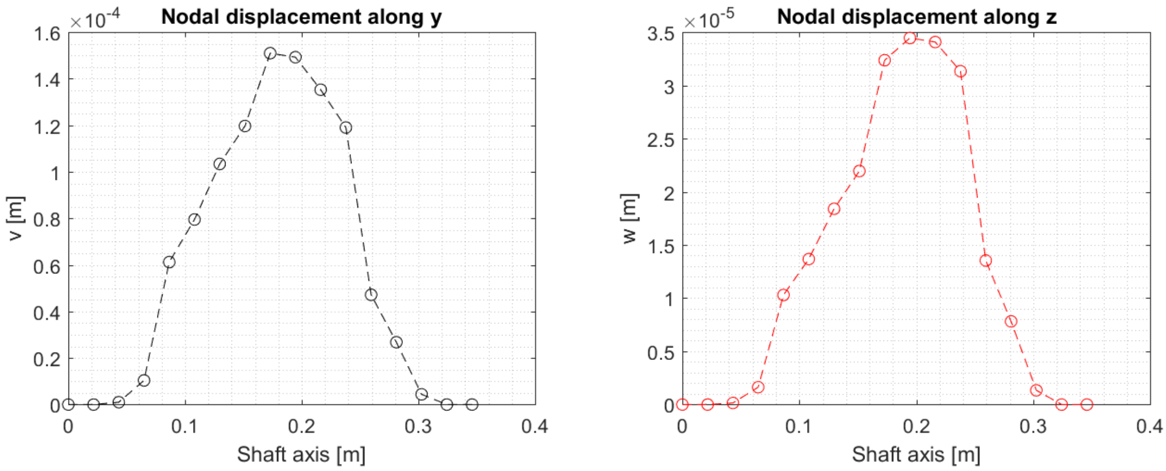


Figure 6.4 Fourth shaft deflection

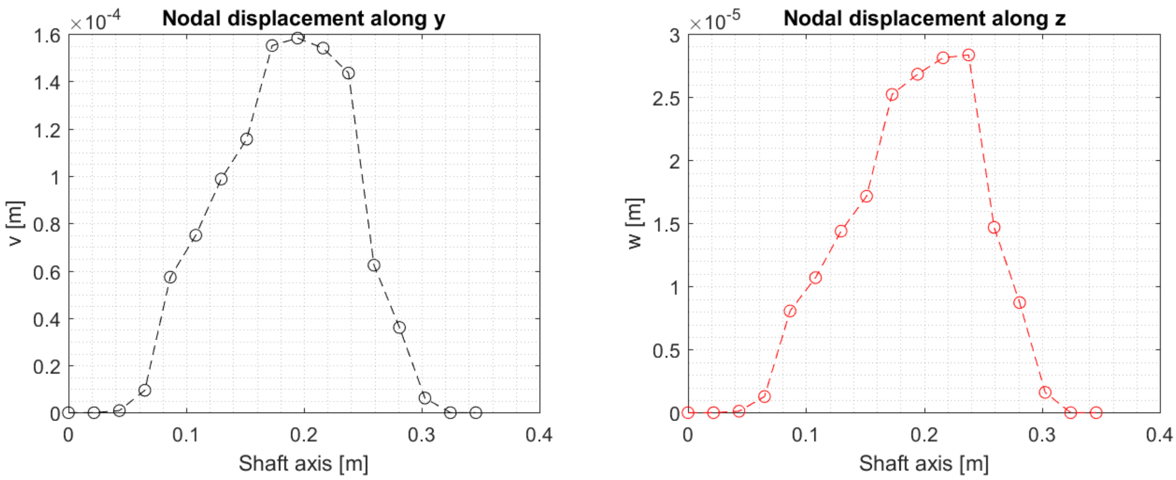


Figure 6.5 Fifth shaft deflection

6.2. Industrial gearbox

The case of the industrial gearbox is considered such a way that there are four stages gearbox which will reduce the speed of the motor and increase the torque.

6.2.1. Input data

Table 6.10 Engine data

Maximum input torque [Nm]	300
Rotational speed at the maximum torque [rpm]	3500
Maximum rotational speed [rpm]	3500
Minimum rotational speed [rpm]	0

Table 6.11 User Input Parameters

Force along the x axis [N]	10000
Force along the y axis [N]	10000
Force along the z axis [N]	10000
Torque around the x axis [Nm]	0
Torque around the y axis [Nm]	0
Torque around the z axis [Nm]	0

Table 6.12 Gear data

Overall gear ratio	70
Inter-axis distance between first and last shaft	3
Inter-axis tolerance	0.1%
Gear ratio tolerance	1%
Material	17CrNiMo6
Mechanical treatment	Tempered /case-hardening

Table 6.13 Shaft data

Material	C45
Mechanical treatment	Surface hardening
Damping factor	5%

6.2.2. Output data

Table 6.14 Gear data

	First	Second	Third	Fourth
Module [mm]	2.5	10	20	20
Number of teeth_input	31	25	40	48
Number of teeth_output	211	93	29	51
Width [mm]	50	50	73	57
Pressure angle [°]	14.5	14.5	14.5	14.5
Helix angle [°]	10.7	0	0	0

Table 6.15 Spline data

	First Stage	Second stage		Third stage		Fourth stage	
	output	Input	output	Input	output	Input	output
Number of Teeth	87	87	91	100	90	90	97
Pressure Angle [°]	30	30	30	30	30	30	30
Module[mm]	1.3	1.3	1.8	2	1.3	1.3	1.5

Table 6.16 Bearing data

	Left bearing on first shaft	Right bearing on first shaft	Left bearing on second shaft	Right bearing on second shaft	Left bearing on third shaft	Right bearing on third shaft	Left bearing on fourth shaft	Right bearing on fourth shaft	Left bearing on fifth shaft	Right bearing on fifth shaft
Type of bearing	Needle roller and cage assembly	Four-point contact ball bearing	Needle roller and cage assembly	Needle roller and cage assembly	Single row deep groove ball bearing	Needle roller and cage assembly	Single row cylindrical roller bearing	Single row cylindrical roller bearing	Single row cylindrical roller bearing	Single row cylindrical roller bearings
d (F_w) [mm]	65	65	90	100	190	145	90	90	65	80
D (E_w) [mm]	73	120	97	107	240	153	125	125	90	140
B (U) [mm]	23	23	20	21	24	26	22	22	16	26
d_1 [mm]	-	78.5	-	-	206	-	102	102	75.5	101
D_1 [mm]	-	101	-	-	224	-	111	111	81	123
Dynamic load rating [kN]	44	110	42.9	45.7	76.1	70.4	105	105	58.3	160
Static load rating [kN]	95	112	114	127	98	224	176	176	88	166
Fatigue Load limit [kN]	11.8	4.75	13.7	15.3	2.8	25	20.8	20.8	10.2	21.2
Designations	K65x73x23	QJ213N2MA	K90x97x20	K100x107x21	61838	K145x153x26	NCF2918CV	NCF2918CV	NCF2913CV	NU216ECP

6.3. Final Design

6.3.1. Gear

In order to verify the final design of the gear, it is compared to the gear created by KissSoft software which is design software for mechanical engineering applications.

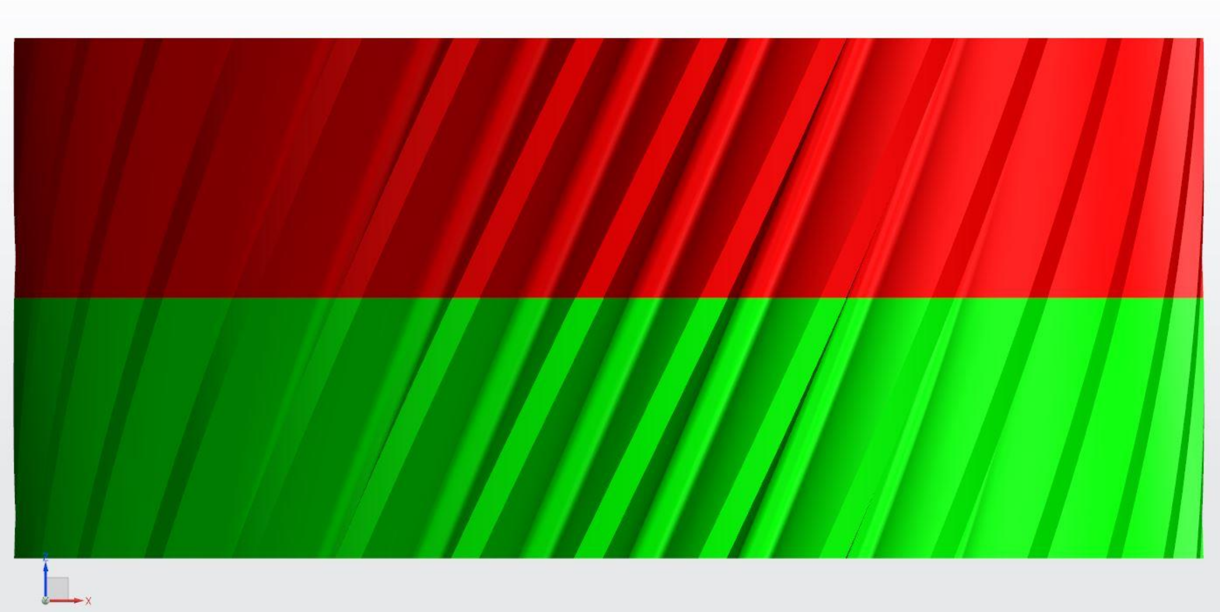


Figure 6.6 comparison between KissSoft and NX gear

The top image is extracted from the KissSoft, and the lower image is the design that has been done in this project with NX software.

As it can be seen in figure 6.6, the two profiles are quite identical, and the created profile is acceptable.

Gear option selection

In the case of the automotive gearbox, there are three types of gear design/connection to the shaft. The three option are:

1. A simple hole in order to represent the press fit or a gear integrated with the shaft.
2. The spline profile inside the gear hole
3. Synchronizer cone and needle bearing

In the case of the Industrial gearbox, the first two options are available to be selected.

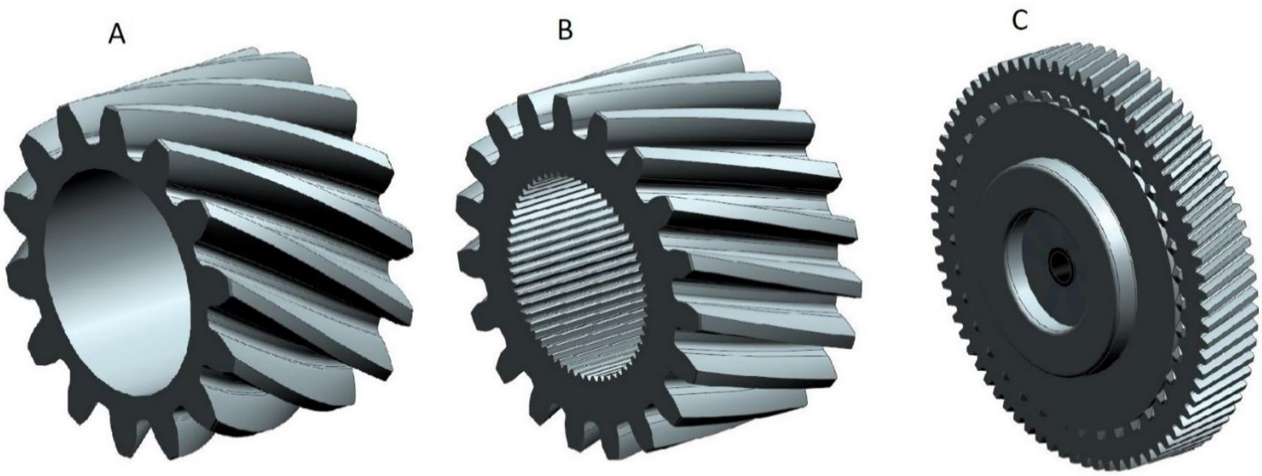


Figure 6.7 A. Input reverse gear B. input first gear C. Output first gear

6.3.2. Shaft

By assembling the different parts of the shafts, the final shape of the input and output shaft will be as following:

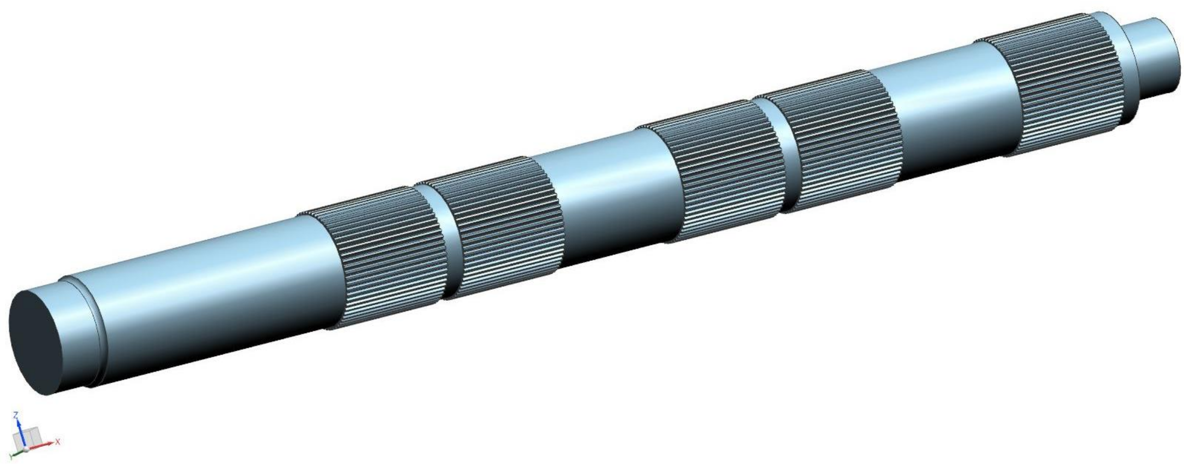


Figure 6.8 Input Shaft

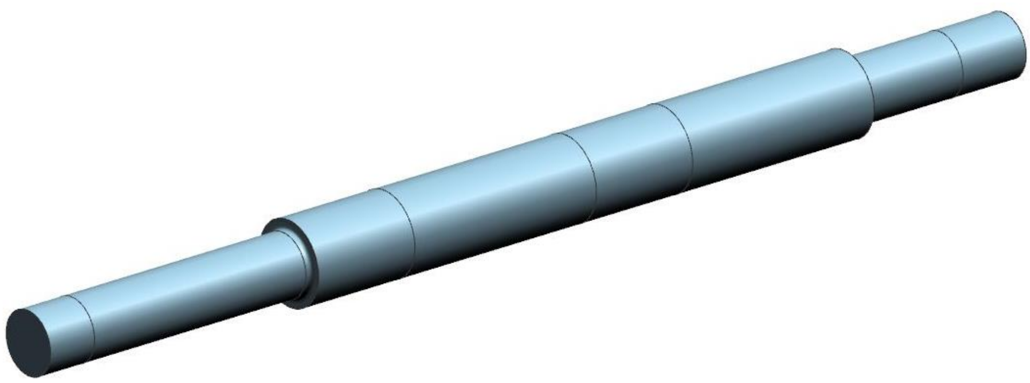


Figure 6.9 Output Shaft

6.3.3. Synchronizer

The assembly of the components of the first synchronizer is represented below.

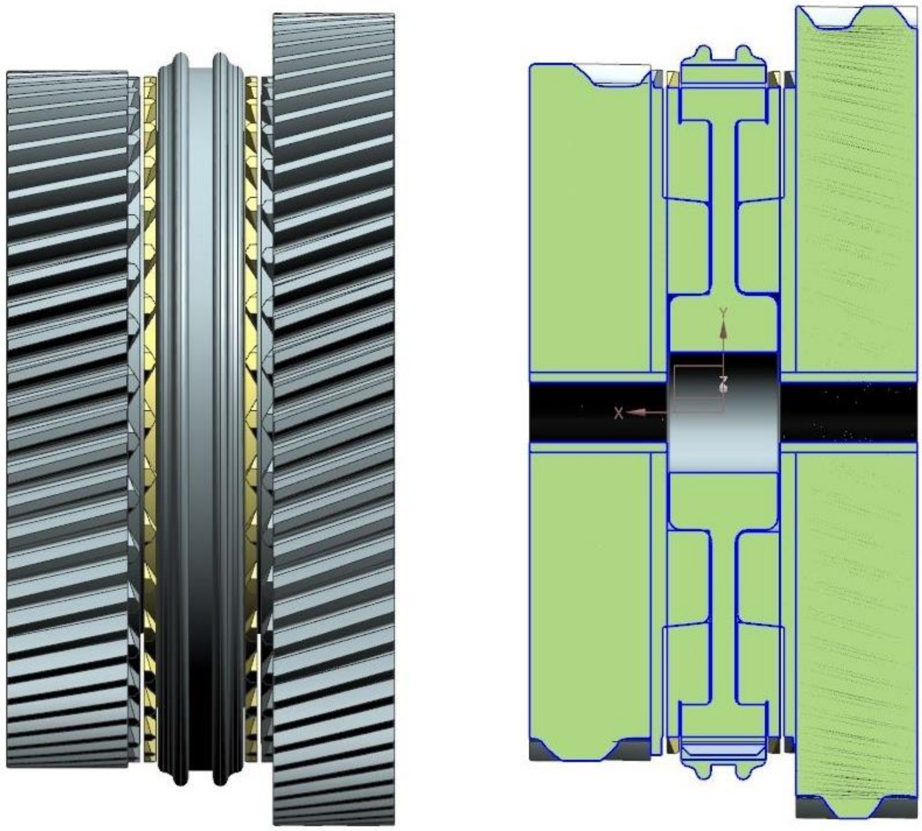


Figure 6.10 Side view and cross section of for the synchronizer

6.4. Final assembly

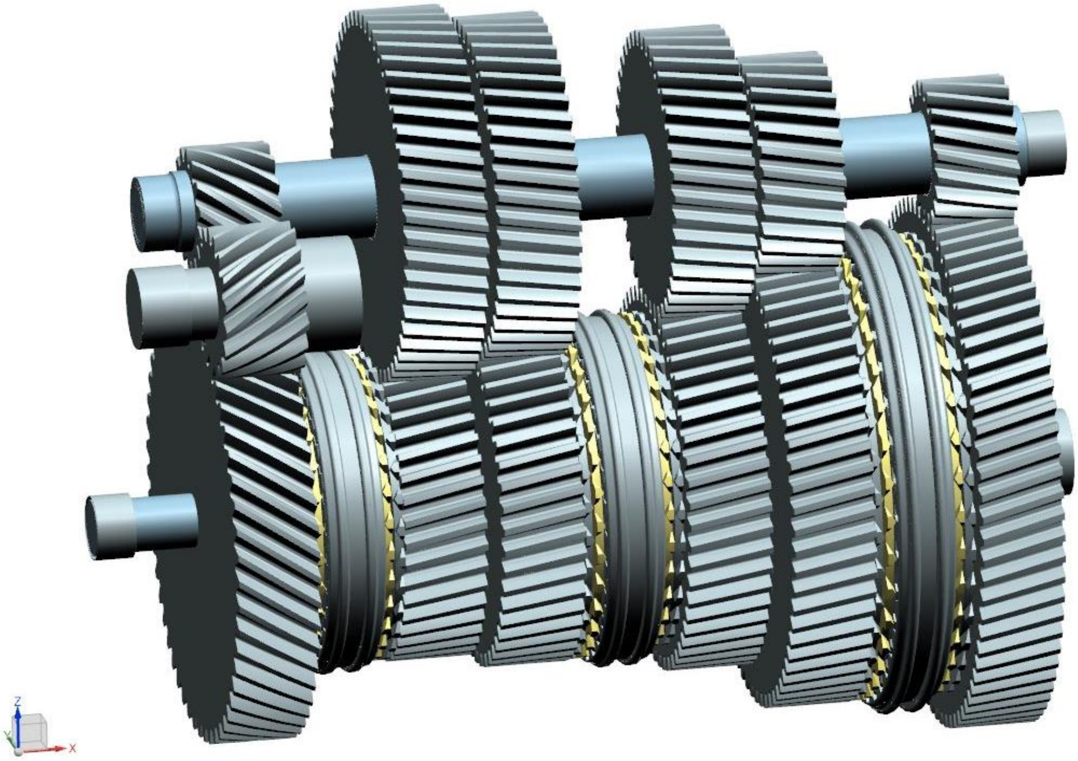


Figure 6.11 Automotive gearbox assembly

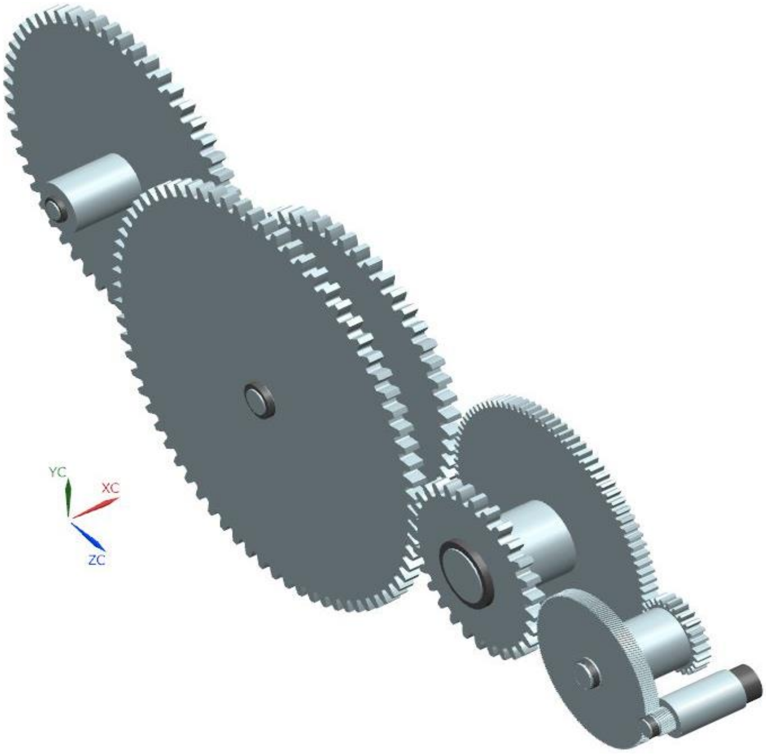


Figure 6.12 Industrial gearbox assembly

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