Trabajo Fin de Máster
Máster Universitario en Ingeniería Industrial

Gearbox Transmission Error study using Multi Body Analysis software MSC Adams

Author: Antonio Roldán Sánchez
Tutors: Johan Wideberg
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Dpto. Ingeniería y Ciencia de los Materiales y del Transporte
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El tribunal nombrado para juzgar el Proyecto arriba indicado, compuesto por los siguientes miembros:

Presidente:

Vocales:

Secretario:

Acuerdan otorgarle la calificación de:

Sevilla, 2018

El Secretario del Tribunal

A mi familia

A mis maestros

Agradecimientos

Con este Trabajo de Fin de Máster se pone fin a una etapa muy importante de mi vida, una etapa que jamás olvidaré, una etapa llena de esfuerzos y sacrificios, una etapa de lecciones y aprendizaje, una etapa donde he cambiado, donde he madurado, una etapa que me ha convertido en la persona que soy hoy. Y me gustaría agradecer a todas las personas que han estado junto a mí en esta etapa de mi vida.

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Antonio Roldán Sánchez

Sevilla, 2018

Resumen

El error de transmisión es una de las variables de funcionamiento más importantes en un sistema de transmisión por engranajes. Se define como la diferencia entre el ángulo de giro teórico que debería tener el engranaje de salida en función del ángulo del engranaje de entrada y el valor real del ángulo de giro del engranaje de salida. Esta diferencia se debe principalmente a tres factores, la deformación elástica de los diferentes elementos del sistema debido a las cargas a las que están sometidos, errores en la fabricación o montaje de los engranajes que hacen que el perfil del engranaje no sea una evolvente perfecta o efectos dinámicos que pueden aparecer a altas velocidades. La gran importancia del error de transmisión reside en que está considerado por la literatura en este campo como el principal origen de las vibraciones producidas en un sistema con engranajes.

El objetivo de este proyecto será desarrollar un modelo numérico utilizando software de análisis MultiBody con el que podamos estudiar la evolución del error de transmisión en una caja de cambios de dos etapas en el que se introducirán además componentes modelados mediante software de elementos finitos para tener en cuenta la flexibilidad de los elementos del sistema como pueden ser los ejes o los rodamientos. El modelo se desarrollará utilizando un software comercial de análisis MultiBody de los más utilizados en la industria, en concreto, Adams 2017 de la marca MSC Software, aprovechando un nuevo módulo lanzado con esta versión para el modelado del contacto entre engranajes llamado *Advanced 3D Contact*. El modelo se simulará con distintos valores de las variables de entrada, principalmente velocidad de entrada y par resistente, para ver la evolución del error de transmisión en los distintos casos, así como se estudiarán distintos modelos donde se varía la rigidez de los distintos elementos del sistema y se analiza su impacto en el error de transmisión.

Abstract

Transmission error is one of the most important working variables in a geared transmission system. I can be defined as the difference between the theoretical rotation angle of the output gear for a given rotation of the input gear and the actual angle of rotation of the output gear in the real system. This difference between the theoretical angle and the real one has three main causes, the elastic deformation of the different components of the system due to forces they are subjected to, manufacturing and assembling errors that make the teeth profile differ from the perfect involute profile, and the dynamic effects that can appear at high speeds. The reason why the transmission error is so important is that it is considered to be one of the main causes of vibrations in geared systems.

The aim of this master thesis is to develop a numeric model using MultiBody analysis software that allows us to study the evolution of the transmission error in a two-stage gearbox, introducing also components modelled using finite elements software in order to account for the flexibility of different components of the system, such as the shafts of the bearings. The model will be developed using one of the most used tools in the industry for MultiBody analysis which is Adams 2017, from the company MSC Software taking advantage of their new module introduced in this version, *Advanced 3D Contact,* that allows us to model contact between gear teeth much more precisely than the previous version. The model will be simulated under different values for the input variables, mainly input speed and resisting torque, to study the evolution of the transmission error in the different scenarios, and we will also work with different models where we change the flexibility of the elements of the system to analyze their impact on the transmission error.

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Notation

N_i	Rotation speed of shaft i in rpm
ω_i	Rotation speed of shaft i in rad/s
Z_i	Number of teeth of gear i
T_i	Torque on gear i
r	Gear or Transmission ratio
D	Pitch diameter
D_b	Base diameter
α	Pressure angle
m	Module
ϑ_i	Angular position of gear i
β	Helix angle
TE	Transmission Error
MTE	Manufacturing Transmission Error
STE	Static Transmission Error
DTE	Dynamic Transmission Error
MBA	Multibody Analysis
FEM	Finite Element Model
MPC	Multi Point constraint
PPTE	Peak-to-peak Transmission Error

1 INTRODUCTION

Gears are one of the most used components in many engineering fields, but especially in mechanical engineering. They can easily be found in almost any machine or mechanism with moving parts. We use gears in everyday household items such as a clock or a hair-drier, but they are also a key component in way more technical and complex systems such as a wind turbine. And of course, it is of utmost importance their role in the automotive industry, where they are used in many subsystems but particulary they are the essential component in the gearbox, which allows the combustion engine to be used in a wide range of speeds, and therefore make cars and motorbikes usable in everyday tasks.

1.1 Use of gears

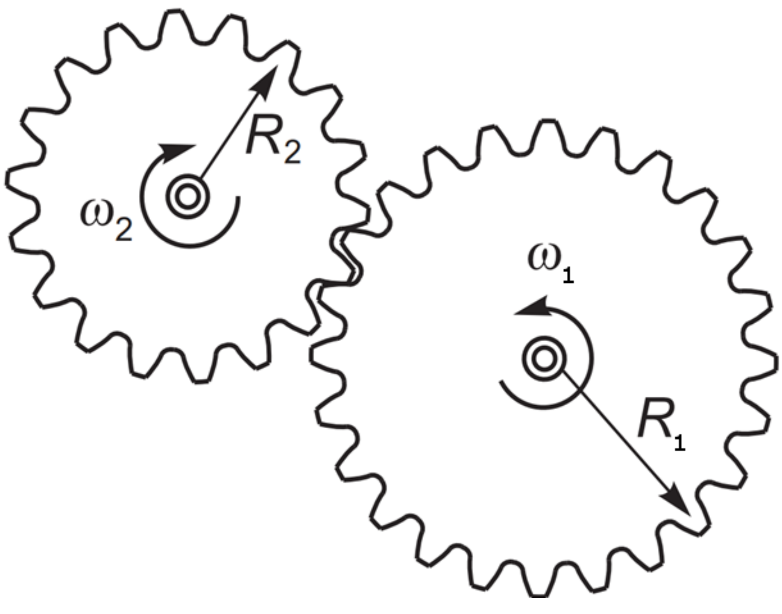


Figure 1-1 - Gear Pair [38]

The main use of gears is to transmit rotational power, usually by applying a transmission ratio that converts speed into torque, maintaining the power. To model this, we can use the basic equation for gears:

$$r = \frac{R_2}{R_1} = \frac{Z_2}{Z_1} = \frac{\omega_1}{\omega_2} = \frac{T_2}{T_1} \tag{1.1}$$

With this formula, we can see that the transmission ratio is a constant for the gear pair, which is the ratio between the number of teeth of each gear or the ratio between their radiuses and this ratio will also be the relation between the speeds of the gears and their torques. This equation derives also from the constant power condition:

$$P = \omega_1 T_1 = \omega_2 T_2 \tag{1.2}$$

Assuming that there are no power losses, the power will be the same for both gears, so torque and speed will be inversely proportional. For this reason, we use gears when we have a shaft rotating at a high speed, but low torque and we want to transform it into low speed but high torque, or vice versa. This property is the main reason that makes gears so useful in a wide range of cases because they allow you to transform an input motion into your desired motion. For example, electric motors usually have a very high speed but low torque so we use gears to reduce the speed and make it more usable for many specific tasks.

But this is not the only reason to use gears. There are many different typologies of gears that allow us to translate or change the axis of rotation, or also change the direction of the rotation.

1.2 Gear Vibration

Due to the wide use of gears in the industry, it is no wonder that gear technology is very well known and developed and that gears are the subject of many technical papers and scientific research studying different aspects inside the field of gears. One of the most important aspects when studying gears is gear vibration.

Gears are a source of vibration since they are moving parts rotating at high speed with many different surfaces coming into contact at even higher speeds. The teeth of the gears engage with high-frequency excitation, for example, a gear with 30 teeth rotating at 1000 rpm will have a tooth mating frequency of 500 Hz, and this excitation will cause the whole system to vibrate.

The American Gear Manufacturers Association (AGMA) has standards limits for gear vibrations, in terms of displacement, velocity, and acceleration. The standard is the ANSI/AGMA 6000-B96 which sets two different classes of gears depending on the limits they can hold, as we can see in Figure 1-2. The standard also sets procedures on how to measure vibrations, how to run the tests and what the limits of class A and B mean in terms of gear usability and life.

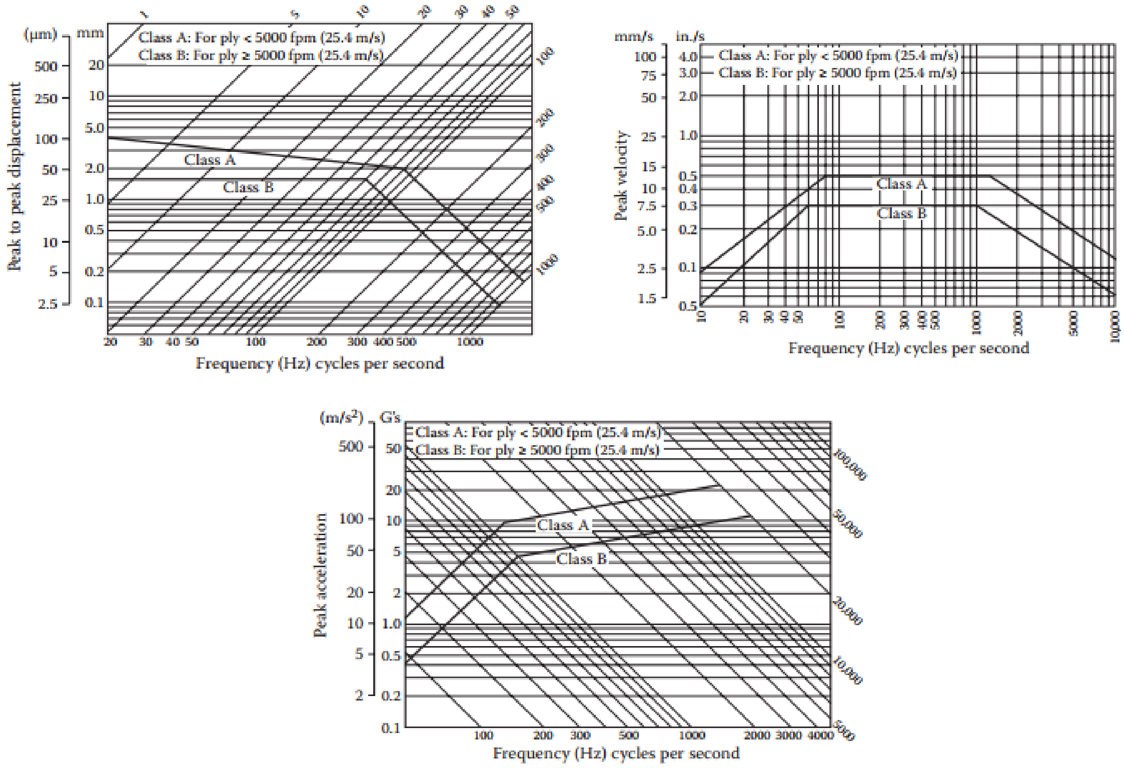


Figure 1-2 – Vibration limits in ANSI/AGMA 6000-B96

Vibration analysis is becoming nowadays a very important tool when diagnosing a gearbox, because we can detect if something is wrong with the transmission and locate a single bad tooth or even fatigue cracks growing before it becomes critical using just a vibration measurement, which is completely non-invasive and can even be carried out without disassembling the component. For this reason, vibration analysis is gaining importance in sectors where gears may play a critical role and a sudden failure can lead to significant consequences in terms of critical damage to the machine or even a safety hazard to the people such as in the automotive or aeronautic industries.

But in order to use vibration analysis, we must first understand what vibrations are normally generated due to the correct working of the gearbox and which vibration are abnormal and come from some malfunction. This is where the Transmission Error comes into play. Transmission Error is the difference between the theoretical movement that the output gear should follow when we impose a movement in the input gear and the real movement of the output. And this Transmission Error, hereafter TE, is considered by many authors to be the main source of gear noise and vibration [2] [3] as we will discuss at length in 3.3.

The aim of this master thesis is to study the viability of analyzing the Transmission Error of a geared transmission using commercial software MSC Adams based on Multi-Body Analysis, studying also the effect that the flexibility of different elements can have on the results of TE. This project is also a follow-up to a previous master thesis [4] developed in Politecnico di Torino that used MSC Adams too and a similar model to study the effect of profile modifications on the Transmission Error.

2 THEORY OF GEARS

In this chapter we will discuss the basic theory of gears, their main parts, the different types of gears we can find, some definitions, how they work, the main parameters needed to understand them and their geometry.

2.1 Basic principles

As we have already commented the main use for gears is to apply a transmission ratio to a rotating shaft to obtain some mechanical advantage. This gear ratio is defined as the relation between the speed of the input shaft and the speed of the output shaft. Although depending on the application the inverse relation might be used.

$$r = \frac{\omega_1}{\omega_2} \tag{2.1}$$

Since the linear speed at the point of contact must be the same, the gear ratio can be calculated as the relation between the diameters of the gears. And since the number of teeth is directly proportional to the diameter of the gear, the ratio can also be expressed as we see in the formula (2.2)

$$v = \omega_1 D_1 = \omega_2 D_2 \rightarrow r = \frac{\omega_1}{\omega_2} = \frac{D_2}{D_1} = \frac{Z_2}{Z_1} \tag{2.2}$$

Finally, assuming no power is lost in the transmission, which is a good assumption since the efficiency of gears is usually around 98%, we can obtain the relation between the torques and the gear ratio. The ratio of the torques is also known as Mechanical Advantage and it has the same value as the gear ratio

$$P = \omega_1 T_1 = \omega_2 T_2 \rightarrow r = \frac{\omega_1}{\omega_2} = \frac{T_2}{T_1} = MA \tag{2.3}$$

The importance of this formula should be emphasized, since what it is telling us is that, by using gears, we can obtain a mechanical advantage, and thus more force in the output than in the input, by increasing the number of rotations in the input with respect to the output. And this is the main objective when using gears.

2.2 Types of gears

There are many types of gears attending to several different classifications, such as the relative position to the axes, gear tooth profile, or other parameters. Each type of gear has certain characteristics that make them more suitable for certain tasks. We can find many books and handbooks on the topic of gears and their types such as [4, 5, 6, 7] where we can find more information about gear types, suitable applications, gear materials, and manufacturing. Here we will have an overview of the most important and used typologies, explaining their main characteristics.

2.2.1 Spur Gears

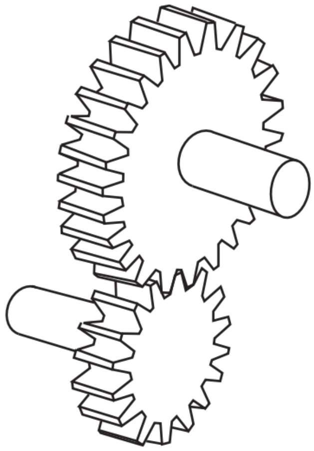


Figure 2-1 – Spur Gears [38]

It is the most basic type of gears and the most common and used one, this is because they are also the cheapest ones. They consist of two parallel axes connected with a couple of toothed wheels whose teeth mate with each other. The geometry of the wheel is that of the extrusion of the cross-section along the axis of rotation. They can transmit high torque, but they are not very smooth and should not be used in applications where vibrations are an issue. They only generate radial forces to the support bearings.

2.2.2 Helical Gears

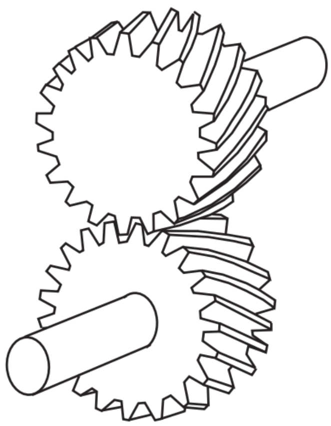


Figure 2-2 – Helical Gears [38]

The difference between spur gears and helical gears is that the teeth are not straight but they follow a spiral curve curving around a cylinder. The main advantage is that the meshing of the teeth is progressive, and this results in a smoother transmission of power, reducing the levels of noise and vibrations. They can withstand a slightly higher load than spur gears and have a longer life for the same load. On the other hand, they are less efficient because the friction is higher. Another difference with spur gears is that while working they also generate a thrust force component in the direction of the shafts, this must be considered when selecting bearings and use bearings that can withstand these forces, for example, tapered roller bearings. It should be noted that for a couple of helical gears to work one helix must be right-handed and the other must be left-handed.

A very similar kind of gears are the Double Helix gears, shown in the figure below.

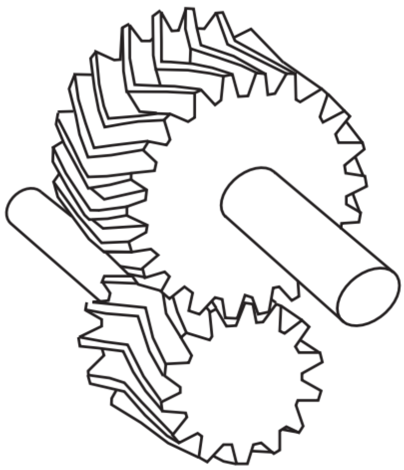


Figure 2-3 – Double Helix Gears [38]

Also known as herringbone gears, the working principle is the same as for helical gears, but instead each gear has two opposing helixes. They are used when a smoother action is necessary, but we want to avoid the problem of the thrust force produced by helical gears, because the two helixes counteract each other, and the net thrust force is zero.

Due to the fact that in helical gears the plane normal to the axis of the gear is not the same as the plane normal to the gear teeth, they have two different modules and two different pressure angles. The normal module is the one measured following the direction of the teeth and the transverse module is the one measured in the direction of the axis of the gear, as we can see in the figure below.

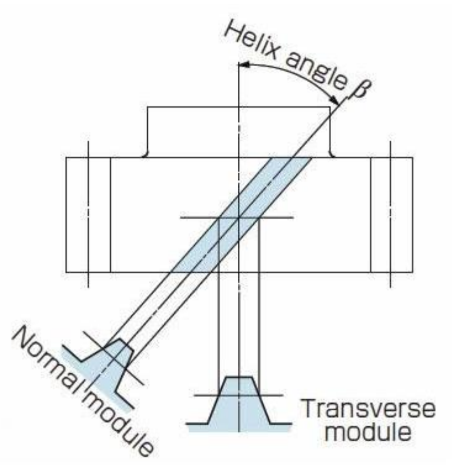


Figure 2-4 – Normal module vs Transverse module in helical gears [39]

The relation between normal and transversal variables is the following.

$$\begin{aligned} D_n &= D_t \cos(\beta) \\ m_n &= m_t \cos(\beta) \\ \alpha_n &= \alpha_t \cos(\beta) \end{aligned} \tag{2.4}$$

2.2.3 Internal Gears

All the gears seen so far are external gears because both gears have their teeth on their external faces. But if the teeth are on the internal face of the cylinder it is called an internal gear. There are internal gears with spur or helical teeth. For obvious reasons two internal gears can never mesh, but a pair internal-external is needed. One of the advantages of this kind of gears is that the center distance is smaller, making them suitable for a more compact design. They are also a bit more structurally sound and can withstand higher loads. They need a ratio of more than 5:1 to work. They found their main use is epicyclic gear trains

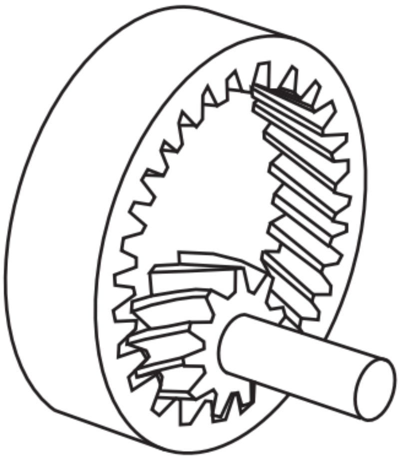


Figure 2-5- Internal Gears [38]

2.2.4 Bevel Gears

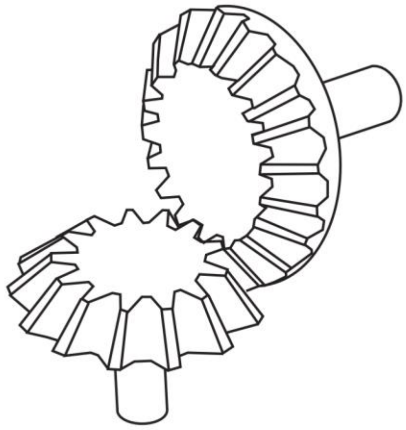


Figure 2-6 – Bevel Gears [38]

Bevel gears are characterized because their axes are not parallel, but they intersect in a point. They are also called coplanar gears. They are used when a change in the orientation of the axis of rotation is needed. They can be classified according to the shape of the teeth, the simplest shape would be straight teeth, which are the equivalent of spur gears; tapered teeth, also known as Zerol gears, imply a slight improvement in terms of vibrations, noise, and power capacity; and the most complicated version of bevel gears are the spiral gears, whose teeth follow a spiral path, and their relation with straight teeth is similar to that between spur and helical gears, they have a smoother power transmission and can carry higher loads.

2.2.5 Worm Gears

Worm gears are an example of gears whose axes are not parallel nor intersecting. The axis of the worm gear is coplanar with the other gear it is mating with. They are used when a high reduction is needed, up to 100:1, since the worm gear works as if it had only one or two teeth depending on the number of spirals. In addition, it can work as a lock, since the worm gear can drive the normal gear, but the normal gear cannot drive the worm gear, this also allows them to carry a higher load than almost any other type of gear. The problem with this type of gear is that due to the high friction between gears the efficiency is considerably lower than that of other gears, around 50 to 80 %

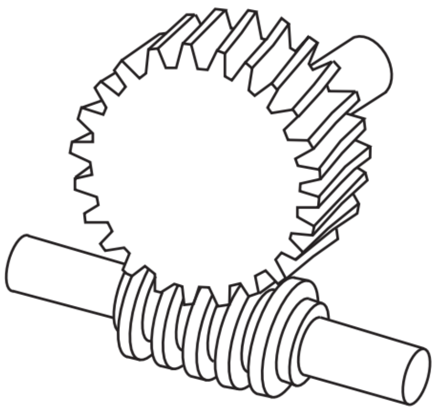


Figure 2-7 – Worm Gears [38]

2.3 Involute profile

Unlike pulleys that work due to friction, gears transmit power thanks to the normal force of the opposing faces of two mating teeth. In Figure 2-8 we can see the cross section of two gears mating, the point C is the contact point and the path it describes it is called the path of contact. The line connecting the point of contact and the point where the two imaginary pitch circles touch (P_{32}) is called pressure line also known as the line of action and it is the direction which the force will be applied. The angle it makes with the horizontal is the pressure angle α, and its value will change as the point of contact slides along the path of contact. It should be noted that when the path of contact is straight the path of contact and the action line are the same, so sometimes both terms are used interchangeably even though for the general case they do not mean the same.

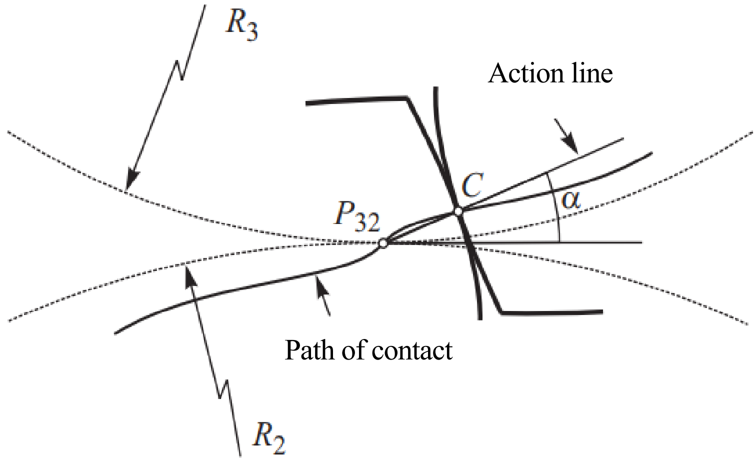
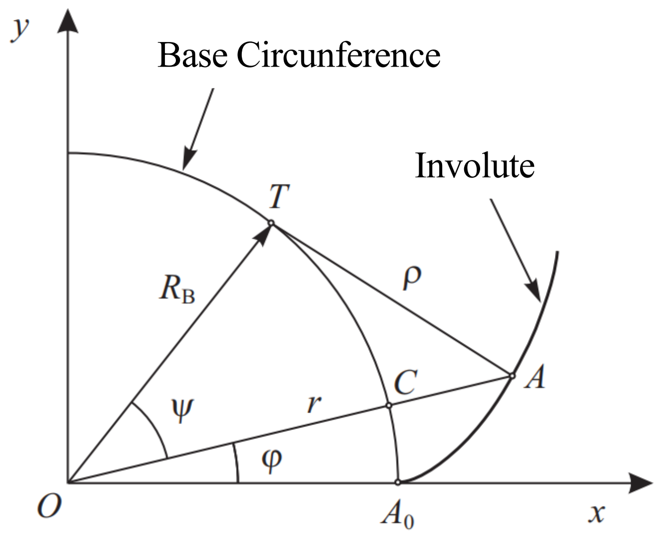


Figure 2-8 – Gear teeth mating [38]

The shape of the path of contact depends on the profile of the teeth. In order to reduce vibrations and ensure the transmission of power in the smoothest way the path of contact should be a straight line, so the pressure angle would be constant for the whole path and the direction of the force is always the same. The profile that achieves this straight path is the mathematical curve of the involute, and for this reason it is the profile that we can find in almost any gear but in some special cases.

The involute is the curve described by the end of a wire that unwinds from a spool keeping the wire straight under tension. We can see the curve in the figure below as well as the equations that define it. The segment TA is perpendicular to the segment OT, and its length is the same as that of the arc of circumference from A_0 to T. As a result, the curvature radius in the point A is ρ. The only parameter necessary to define a specific involute is the radius of the base circumference.



$$\left.\begin{array}{l} r\cos\psi = R_B \\ r\,sen\psi = \rho \end{array}\right\} \Longrightarrow \left\{\begin{array}{l} tg\psi = \frac{\rho}{R_B} \\ r = \frac{R_B}{\cos\psi} \end{array}\right.$$

Figure 2-9 – Involute definition [38]

2.4 Geometry and parameters

Gears have a very complex geometry with many different dimensions and parameters. In this section we will discuss the main dimensions of the gear, explaining them and how to calculate them and emphasize the few independent parameters we need in order to construct the whole geometry of the gear.

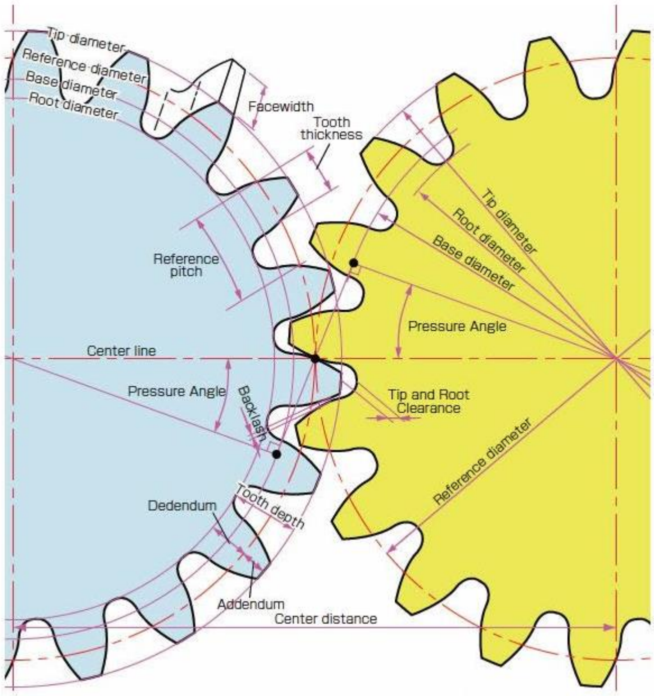


Figure 2-10 – Gear Geometry [39]

The main dimension of a gear is the pitch diameter (D), also known as reference diameter, which does not actually represent a physical dimension of the gear but represents the diameter of the imaginary circle that is centered axis of the gear and passes through the pitch point. The pitch point is the point of intersection between the line that connects the two centers, also known as the line of centers, and the line of action. This pitch diameter is the one we can use in the formula for the gear ratio.

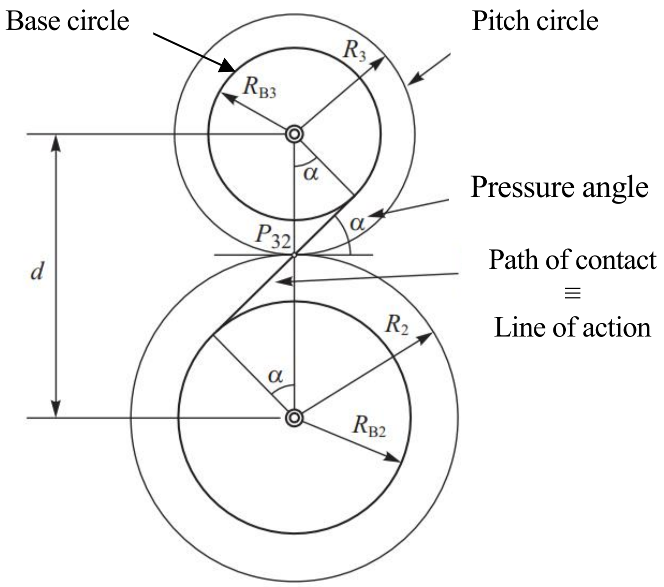
The first and most important parameter is the module (m) which is the relation between the pitch diameter and the number of teeth of the gear. It is the most important parameter because for two gears to work with each other they must have the same module. Its value is not an integer, but it is usually standardized to make gears compatible among different manufacturers. In the imperial system, however, the diametral pitch (DP) is used, which is the inverse of the module, the number of teeth divided by the pitch diameter.

$$m = \frac{D}{Z} \tag{2.5}$$

Another important dimension is the pitch (p) which is the distance between the faces of two adjacent teeth measured along the pitch diameter. It can be calculated as follows.

$$p = \frac{\pi D}{Z} \tag{2.6}$$

The base circle is the circle from which the involute profile is generated. In the figure below, we can see the geometric relation between the base and the pitch circle. And in order to calculate the base diameter (D_B), we use the expression (2.7)



$$D_B = D\cos(\alpha) \tag{2.7}$$

Figure 2-11 – Pitch and base circles [38]

The addendum and dedendum are the distances between the pitch circle and the outside circle and the root circle respectively. Their values are standardized and for most common applications are the same as the module.

$$D_{outside} = D + 2a \qquad\qquad D_{root} = D - 2d \tag{2.8}$$

The thickness of the tooth is measured in the pitch diameter, if it is measured along the circle it is called circular thickness, if it is measured in a straight line it is called cordal thickness. Its value is also standardized as half of the pitch, although it can be modified to tune the clearance between the gears according to the application.

In helical gears, we have also another parameter which is the helix angle, that defines the twist of the helix of the gear as shown in the figure below. For two helical gears to engage they must have the same helix angle.

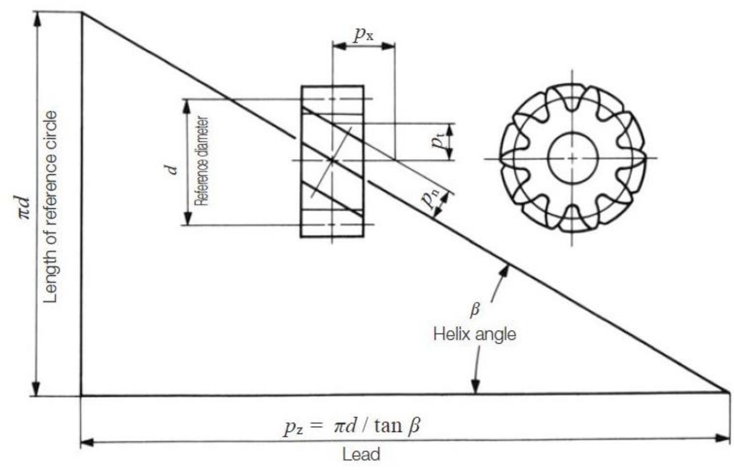


Figure 2-12 – Helix angle [39]

There are other more complex parameters such us tip and root relief, undercutting or crowning that can affect the working of the gears but those are not under the scope of this project.

In conclusion, the basic parameters that define the geometry of the gear are the module, which affects the size of the gear; the pressure angle, which affects the shape of the teeth; the number of teeth; and the helix angle for helical gears; which affects the twist of the helix.

3 TRANSMISSION ERROR

This project will be focused on studying Transmission Error and its evolution when using different operating parameters and flexible elements in the system. For this reason, we will dedicate this chapter to the TE, what it is, where it comes from, the different types there are, its relation with vibrations and the state of the art in the study of TE.

3.1 Definition

The idea of the Transmission Error was first defined by Harris in [8] in 1958 and Munro, his Ph.D. student at the time. The formal definition they gave was "**TE is the difference between the angular position that the output shaft of a drive would occupy if the drive were perfect and the actual position of the output**". In other words, the actual meshing of the gears is not perfect, due to the deformation of the teeth, error in the teeth profile which is not a perfect involute, and incorrect installation of the gear pair or other parameters, so the actual angular position of the output gear will not be the exact theoretical position, and the TE is defined as the difference between the theoretical one and the actual one.

$$TE = \theta_2 - \frac{Z_1}{Z_2}\theta_1 \tag{3.1}$$

The sign criterion is not widely stablished, some authors use this, and others use the opposite, but this is not really important because it only represents the direction of measurement of the angle. The important value is the magnitude of the angle.

As defined by Mark in [9] "A pair of meshing gears with rigid, perfect, uniformly spaced involute teeth would transmit exactly uniform angular motion" and thus the transmission error would be zero. But in the real world this never happens, every gear pair has some degree of irregularity in those aspects and this translates into a not uniform angular motion transmission. Another definition of TE given by Munro in [10] is **"The deviation in position of the driver gear (for any given position of the driving gear), relative to the position that the driven gear would occupy if both gears were geometrically perfect and undeformed"**

To measure the TE we just need to measure the angle at the input, the angle at the output, apply the gear ratio and subtract one from the other. As we can see this TE has a unit of angle, however, the results are rarely given as an angular measurement, but it is multiplied by the pitch radius of the gear to obtain a linear measurement since this is much more informative (usually below 5μm). The great advantage of specifying TE as a linear measurement is that it is a direct indication of the quality of the gears, two gears of the same quality will have the same TE regardless of their size, module or diameter, so we can compare different gears very easily [2]

In literature we can find the typical values of peak-to-peak TE (PPTE) obtained in gears with different levels of quality. For very large gears and low speed machinery we can see values of up to 20μm but this type of gears is only used in situations where vibrations and noise is not a concern at all. Values of 10μm are common on very rough and low-quality gears not suitable for many applications where vibrations must be kept under control. For medium and small, which are the most used ones in the industry, 3 to 5 μm of PPTE are typical of good quality gears. And for high precision gearing we can find values of PPTE around 1 μm which would be representative of extremely good quality and very precise gears [2].

$$TE = \frac{D_2}{2}\left[\theta_2 - \frac{Z_1}{Z_2}\theta_1\right] = \frac{1}{2}\left(D_2\theta_2 - D_1\theta_1\right) \tag{3.2}$$

Even though TE is a variable in time, it ussually follows a wave form, we do not need the whole evolution of the TE in time, but the single value of peak-to-peak TE is often reported since it gives information on the amplitude of the TE as Houser explains in [11]. If some further information of the vibration is needed Fourier analysis is also used to identify the most important frequencies of the signal in order to detect any problem in the transmission.

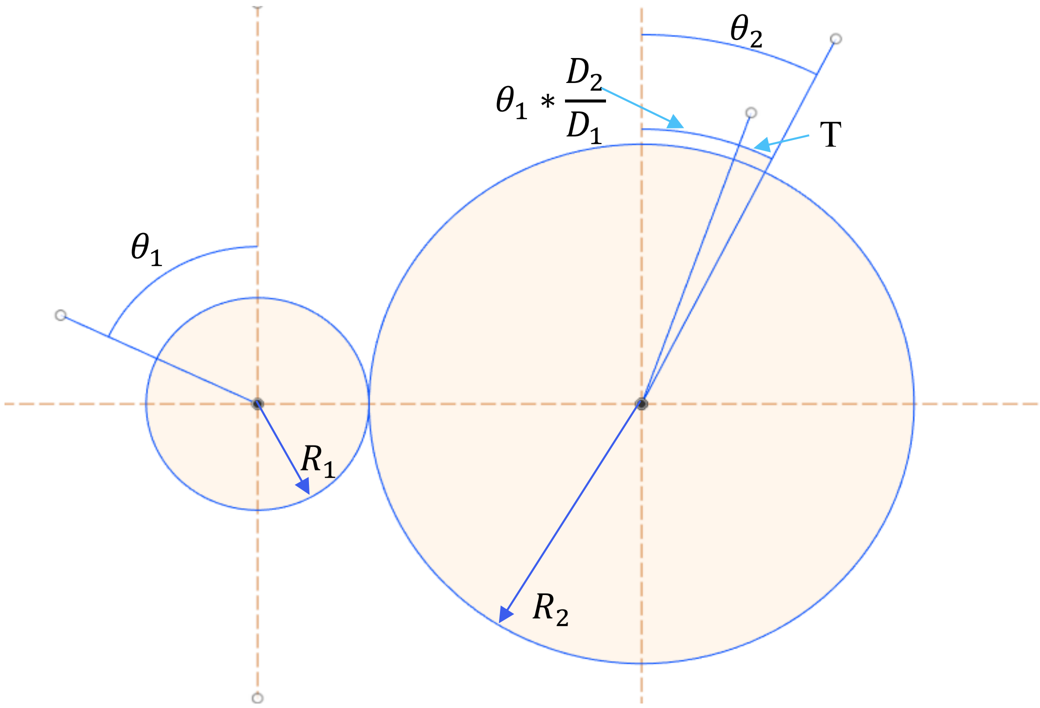


Figure 3-1 – Transmission error

3.2 Types and origins

There are three main types of TE that are often referred to in literature, and these types are related to the source and origin of the TE [12].

3.2.1 Manufacturing Transmission Error (MTE)

The simplest type of TE is the MTE which is caused by an error during the manufacturing of the gear that generally makes the tooth profile deviate from the perfect involute profile that it should follow. If the teeth don't have a perfect involute profile, this will make the transmission of torque not perfectly uniform. But MTE also accounts for the uneven spacing of the teeth along the pitch circle, errors during the assembly of the gearbox which may lead to the distance between gears not be the correct one, or even to a misalignment in the shafts which will make the axes not parallel to each other. This MTE is directly linked to the quality of the gears and its manufacturing process.

To measure the MTE we measure the TE of a gear pair under no load or a light load, in order to avoid any deformations that may occur under load, and at a low speed, to avoid any possible dynamic effects [3].

3.2.2 Static Transmission Error (STE)

Another source of TE can be found in the deflection and deformation of the system when it is under load. This deformation can be found in the gear teeth, but also in the bearings, shafts and even casing of the gearbox. Static Transmission Error (STE) accounts mainly for this source of TE which depends basically on the stiffness of the system.

In order to measure STE, we measure the TE of a gear pair under load, but in static conditions of low speed, to avoid dynamic effects. It should be noted that since the STE is measured by measuring the TE, it is impossible to get rid of the effect of the MTE, so the STE will include the MTE.

3.2.3 Dynamic Transmission Error (DTE)

The last kind of TE and probably the most important one is the Dynamic Transmission Error. It is the most important one because it considers all the variables that affect the system during its working life since it includes all manufacturing error, the static deformation of the system, but also the effects of the masses and inertial forces of the system, that generate also dynamic deformations.

To measure DTE, the TE is measured under load and at a high-speed condition, which is generally the working condition of many gears. So, it accounts for all the possible reasons of transmission error

3.3 Relation with vibrations

Vibrations and noise are an intrinsic response of almost any mechanical system with moving parts, and gears are not an exception to this rule. On the contrary, gears are more susceptible to generate vibrations and noise than many other mechanical components because they consist of many different teeth meshing and coming into contact at very high speeds. When the surfaces of these teeth contact with each other a contact force appears. This contact force is the responsible of the movement and working of the gears, but at the same time if this contact force is not constant it will become an excitation to the system leading to vibrations and noise. Thanks to the involute profile described in 2.3 we make sure that the direction of the contact force does not change and remains the same during the movement of the point of contact along the whole line of action. However, it is the variation in the amplitude of the force that gives the excitation for the vibration. According to Smith in [13] the source of this force variation is a variation in the smoothness in the contact and it is due to a combination of small variations of the form of the tooth and a varying elastic deflection of the teeth.

There are two main types of gear noise, [15] gear whine and gear rattle. The first one, **Gear Whine,** is a purely tonal noise, whose frequency changes in proportion to the speed. It is generated by the torsional vibration of the

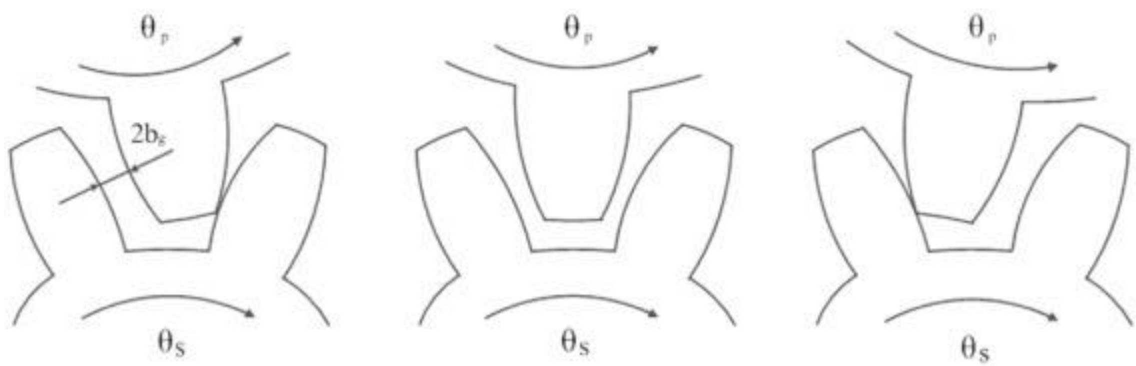


Figure 3-2 – Gear Rattle [15]

loaded gears due to the meshing stiffness variation during the teeth engagement and thus, the frequency of this noise is the meshing frequency of the teeth. And the second kind of noise, **Gear Rattle**, is not provoked by the deformation of the teeth but due to the clearance necessary for gears to work there is certain backlash when two gears are engaged. So, when the gears are not loaded the driven gear might vibrate back and forth within this backlash generating noise.

The main reason why Transmission Error is so important, and it is the subject of many studies nowadays, this one among them, is because there have been numerous studies that point out the correlation between Transmission Error and gear whine, and TE is widely considered as one of the main causes of gear noise, if not the most important one. In Figure 3-3 – Vibration excitation and transmission pathwe can see the overall path to noise, and how all the small deflections of the gears, distortions and deviations from the involute profile translate into Transmission Error and this TE will later derive into noise.

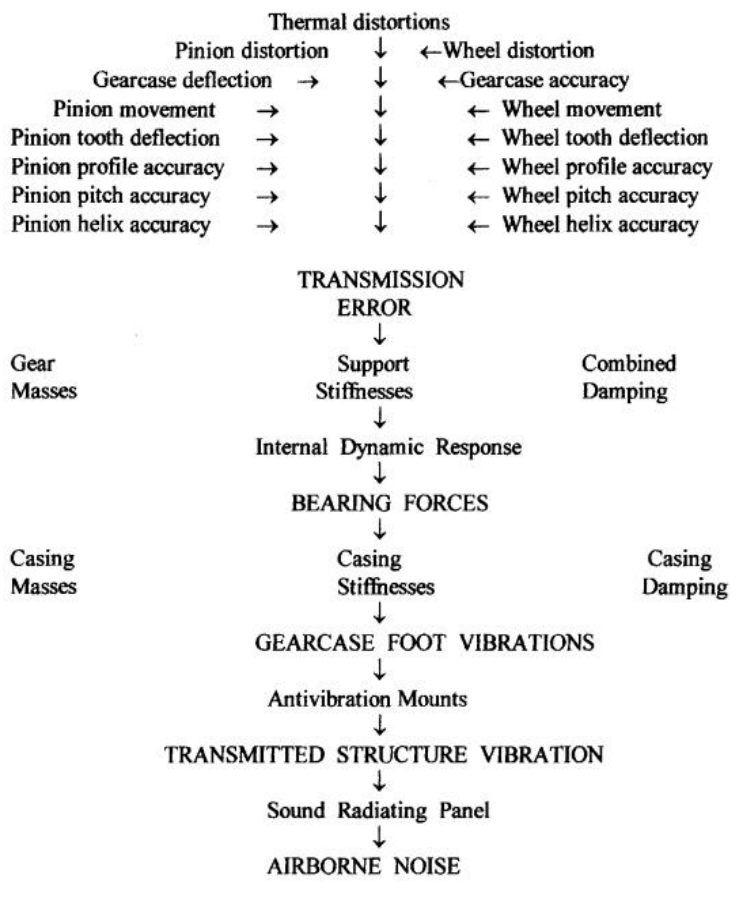


Figure 3-3 – Vibration excitation and transmission path [2]

3.4 State of the art

As we commented before the formal definition of the term Transmission Error was given by G. Harris in [8] in the late 50s, even though the basic idea of TE had been previously used. However, it wasn't until the main bulk of the classic studies on the subject had been carried out by Harris, Munro and Gregory [8, 10, 14, 15, 16, 17] that the term spread out and became widely accepted.

In [8] Harris also explained why tip relief is a good option to reduce TE. If the gear teeth have a perfect involute profile, to have the smoothest transmission possible the tips of the next pair of teeth must be coming into contact when the tips of the current pair are disengaging, so the distance between teeth is set with this rule in mind. The problem is that the teeth are flexible, and under load they will deflect, and this distance will change. If the distance is set to be exactly the needed one under deflection the teeth could not engage smoothly, so by adding tip relief this distance is increased and we ensure a smoother transition between teeth. To explain this and other phenomena he also developed what we know as "Harris Maps" which are graphs that plot the theoretic evolution of the TE along the roll distance for various loads. We can see an example of Harris maps in Figure 3-4, where the top curve is the unloaded case, and then as load is applied the double contact regime steadily expands around the changeover point, which is the point where the contact passes from a teeth pair to the next one.

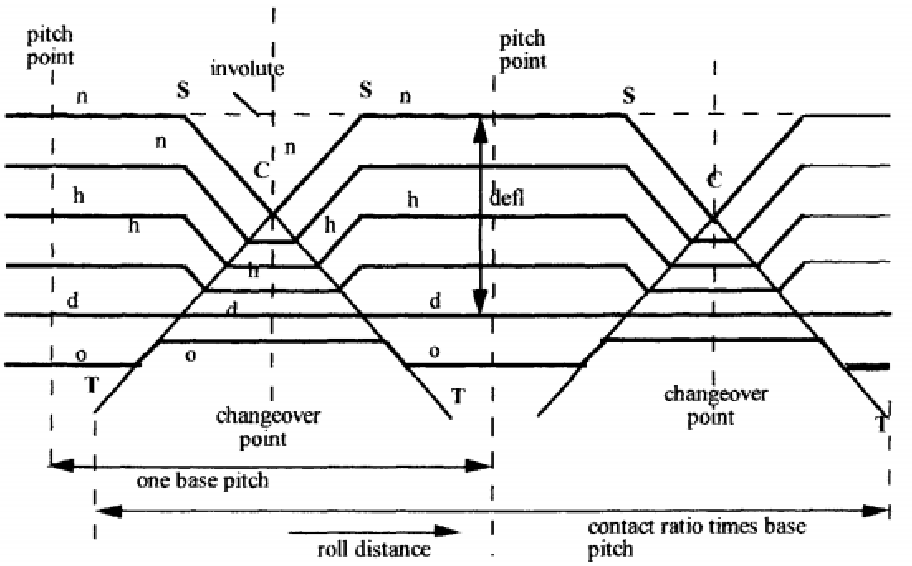


Figure 3-4 – Harris maps for Transmission Error

Since there is such a strong connection between transmission error and noise, most of the research carried out in the field of gear noise has also been related with TE. There are many good books and publications on gear noise that can be used as an introduction to the subject and explain the basic fundaments and more complicated aspects. Three of the best books on the subject of gear noise are [2] by J. Derek Smith, [18] by C.M Harris and [4] by Dudley.

Welbourn [3] collected an extensive research on the field of gear noise with many experiments in the field run by different researchers in the 70s. In its research he stablished the relation between transmission error and noise. The general consensus of the experiments pointed out that when PPTE of the gear pair doubled the noise generated by the system increased by 6 dB. The main factors that affect TE are power, tooth load and speed, when doubling these factors, they observed an increase of 3 dB in the sound generated and 6 dB for the case of speed. Welbourn also remarked the importance of tip relief for reducing the total TE.

Thanks to the investigations of Mark [9] and Munro [17] we can have a deeper understanding of the main sources of the TE which are the geometric errors in the tooth profile, the deflection of the teeth under load and the dynamics of the gears at high speeds. With their research they also formulated a general classification of the different types of TE depending on their origin and divided the TE into the three main categories seen in 3.2, Manufacturing TE (MTE), Static TE (STE) and Dynamic TE (DTE).

All of the above can be considered as the classical research work carried out in the field of transmission error, and this knowledge has been generally accepted and has formed the basic theory of the field. But the most recent works in this field are focusing on two main trends, the first and most important is related with the development of computer models that can simulate the gear contact and predict TE on a system without the need of real world tests, and the second trend is more related with TE and noise measurements in real world tests.

In [19] Fernandez del Rincon develops a model for studying the meshing stiffness of a gear pair. The meshing stiffness of the gear pair is a very important parameter in when studying TE because it is directly linked to the deflection of the teeth and thus to the STE, and this plays a fundamental role in the DTE. He uses FEM in order to test his model but the actual model is an analytical one, and it allows us to determine the STE as well as the meshing stiffness taking into account factors such as the deformation of the gear teeth, modifications on the working distance due to deformations of the supports, friction and the possibility of contacts on both flanks is also included. The deformation of the tooth has been approached by its decomposition in to local and global deformations, in this way the nonlinearity becomes linked only to the local contact model reducing considerably the computational cost

With [20] Lin and He present a method for analyzing the transmission error of helical gears with machining errors, assembly errors and tooth modifications. Just like Fernandez del Rincon they first use a FEM model the simulate the gear transmission system and then they establish the bending-torsional-axial coupling dynamic model of the transmission system based on the lumped mass method. With this method they try to evaluate the results of their model in terms of STE and DTE and compare them to the results obtained with the first FEM model.

Belingardi [1] uses a Multibody approach for the dynamic analysis of a gear transmission of an electric vehicle. He presents a transmission model, which is the same used in this very project and simulates it using an MBA program called RecurDyn to obtain measures of the contact forces and DTE values, and then analyzes the frequency response of the system. The results he obtains are very helpful in the evaluation of the dynamic loads applied to the shafts, the bearings, and the housing of the gearbox, in addition those results allow to make considerations on vibration and transmission noise and evaluate the presence of high loads that are generated under dynamic conditions.

There are many papers that study how TE can be reduced by introducing intentional geometric modifications to the gear teeth. One of these studies is [21] which is the base for this project, and it follows a MBA approach very similar to the one followed in this project. Using the *Advanced 3D contact* toolkit in Adams d'Addato generates gearbox model and run several tests to verify that teeth modifications such as tip and root relief have an effect on TE and the results obtained from Adams meet the expectations of the theory.

Also, Tharmakulasingam [12] for his Ph.D. dissertation developed a model to simulate and analyze the effects of tooth profile modifications on TE but following a FEM approach using Abaqus. For that he has created a Phyton script that makes the gear profile generation process automatic inside Abaqus, generates a FEM model which is able to account for the many nonlinearities of this particular problem, and with this model evaluates the STE as well as the DTE comparing them and checking the results. Finally, he uses a hybrid numerical/analytical method to optimize the gear profile that gives the best results of STE.

4 MULTIBODY ANALYSIS

The aim of this project is to study the feasibility of using a multibody dynamics software to try to model a gearbox, particularly focusing on the results of transmission error. Analyzing the effect of changing different values on the input, such as torque, speed, type of gear as well as different models introducing flexible elements to try to see if the results obtained are valid and correspond to what we could expect in a real situation.

4.1 Multibody dynamics

Multibody dynamics is a very powerful tool used in many engineering and science fields with the objective of analyzing the kinematic and dynamic behavior of a system composed of different interconnected bodies. It has many applications in various fields such as mechanical engineering, aerospace, biomechanics, robotics, particle simulation, vehicle dynamics and so on.

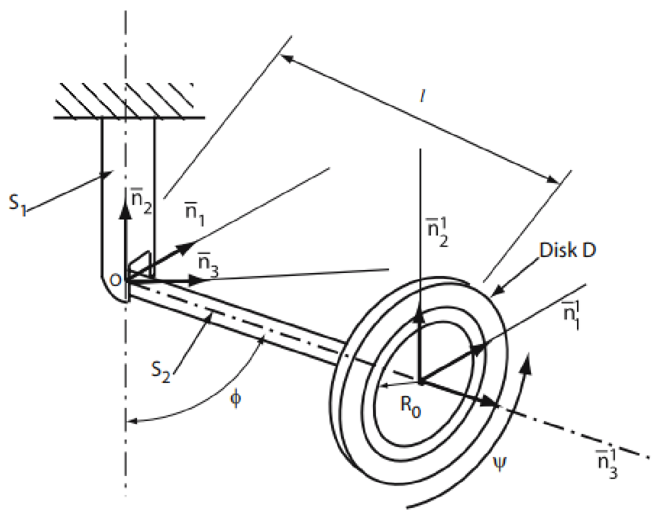


Figure 4-1 – Multibody dynamics problema [36]

The method consists of the formalization of a series of steps, where we first identify the bodies to be studied formulate them in a mathematical way, which is basically defined by their masses, their inertia matrixes, and the position of their centers of gravity. Then we must define the constraints between those bodies, which will limit their relative motions. The forces and reactions applied in each body must also be defined in the next step, and the last input to be defined would be the initial conditions in terms of position, speed, and acceleration. Once we have all the information we need to formulate the equations of motion, which are differential equations that describe the complete kinematic and dynamic behavior of the system. To obtain these equations we can use different methods, the most common ones are the Newton-Euler method and the Lagrange method. Once we solve those equations we will have the solution of the system knowing the equation of motion of each body and we can obtain its position, speed, and acceleration in each instant in time. This is the basic description of the Multibody Dynamics method, which could be applied even by hand calculating everything and solving all the equations, but the problem is that as soon as the number of bodies or degrees of freedom increases, the method becomes way too complicated to be used by hand. So, instead of that, we use computers and numeric methods to get rid of the most tedious tasks such as defining all the equations of motions and solving them.

The rise of computers and the increase of their computational power in the last decades have transformed the Multibody Analysis (MBA) into an incredible tool for designing, analyzing, prototyping and simulating complex mechanical systems. Now we can create very fast a digital prototype to simulate a system that 20 years ago would have been impossible even to imagine and obtain many more results for the designing stage of a project.

In the beginning, MBA was originally developed to model and analyze the dynamics of rigid bodies, but thanks again to the development in the computational power of microprocessors, nowadays we can find commercial software which is able to combine MBA with the Finite Elements Method to simulate a dynamic system composed not only by rigid bodies but also by flexible elements. This feature has made this kind of tool even more powerful since it is able to model almost any mechanical system imaginable.

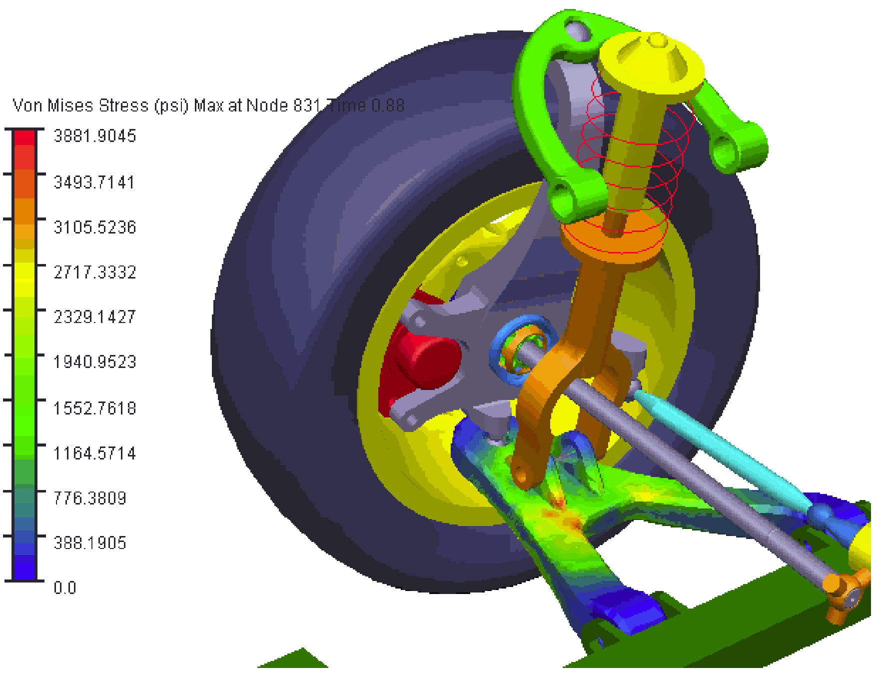


Figure 4-2 – Multibody Software with flexible elements [23]

There are many commercially available softwares based on Multibody dynamics simulation, such as RecurDyn, SimulationX, Amesim, EcosimPro, LMS Virtual.Lab Motion by Siemens, Simpack by Dassault Systems, and many more. In this project, we will be focusing particularly on ADAMS, developed by MSC Software.

4.2 MSC Adams

MSC Software is one of the global leaders in the developers of software for engineering simulation and tools, they develop many tools for different applications such as acoustics, additive manufacturing, computational fluid dynamics, but their main products are Nastran-Patran as a solution for Finite Element Analysis and Adams in the field of Multibody Analysis.

Adams (Acronym for Automated Dynamic Analysis of Mechanical Systems) is one of the most used software in the industry for MBA. First, because it is a strong and reliable product for solving MBA problems, but in addition to that Adams has many different modules that allow users to model very specific problems with high detail and precision.

One of the most important modules is Adams Cars companies in the automotive systems can simulate and predict many situations such as rollover prediction, vehicle drives, handling tests, loss of control evaluation, suspension analysis, and much more. This has made Adams the tool of choice for many big companies in this sector, such as GM, Nissan, Ford, Volkswagen, Audi, BMW, Renault, Porsche, and many more.

But the module in which this project will be focusing on is Adams Machinery. With this module, we can model all the most common components found in any mechanical systems, such as gears, bearings, pulleys, chains, cams, etc. We will use this module to model the gears of our system.

4.3 Advanced 3D Contact

One of the reasons for this project is that with one its latest releases, Adams 2017, they have introduced a new tool for the Adams Machinery module, called Advanced 3D Contact which simulates gear tooth flexibility much more realisticly. This feature allows you to define the gear part geometry and material properties of the gear, then Adams generates a finite element model and solves it in the background.

Based on this finite element model there are three options to define the contact behavior of the gear pair for the analysis. All three of them represent the contact force as a 6-component force [22]:

- The Run Time option computes the contact behavior of the gears during the analysis. This is the most accurate solution, but it is also the one with the highest computational cost since it must solve the FEM model for each tooth contact
- The Pre-Compute option runs first a setup analysis to predict contact behavior and during the simulation uses that setup analysis instead of solving each time. This is a trade-off in terms of accuracy to reduce simulation time.
- The Rigid option simplifies things further by treating the gear teeth rigidly. This, however, is different from the pre-existing rigid-body 3D Contact method. With Advanced 3D contact "Rigid" method smoother results are usually obtained because FEM-based methods are used to define fine meshes instead of traditional tessellation techniques for the contact detection. And with this tool, there are more gear modification options including commonly applied micro-geometry modifications.

The flexible tooth options here provide superior accuracy compared to using an Adams Flex representation of the gears. Adams Flex uses the modal superposition method, which assumes that the part's deformation can be captured by superimposing normal mode shapes. But in the case of gears, most of the deformation takes places in the teeth themselves, which is difficult to capture in the mode shapes.

5 DESCRIPTION OF THE RIGID MODEL

In this chapter we will describe the first model that will be used in the project. This model will be used as a first introduction to the software to establish a baseline for the capabilities of the program where we will see all the options to modify the different parameters available in Adams.

5.1 Gearbox

The gearbox that will be used in this study has been found in [1], it is a double stage gearbox developed for the differential of an electric vehicle and with the objective of keeping weight as low as possible. This transmission is constituted by an input ordinary gear system, able to transmit power from the electric motor to the wheels, followed by an epicyclic one named differential, although we will not be considering the differential for this study. We can see a model of the gearbox in the figure below.

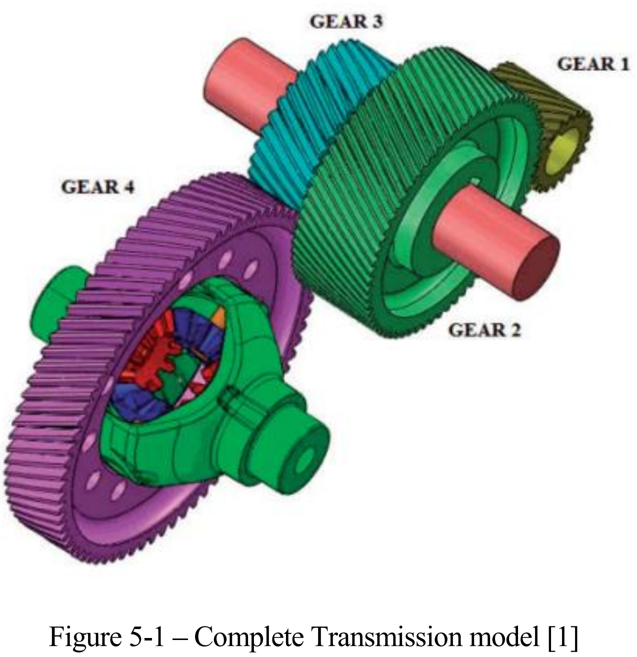


Figure 5-1 – Complete Transmission model [1]

The input of the electric motor would be connected to gear 1, and the output of gear 4 would connect to the wheels of the vehicle by means of the differential.

Even though this gearbox has both gear pairs composed by helical gears, we will have a model with spur gears and another model with helical gears. This will allow us to compare results between the two kinds of teeth.

We can find the main design dimensions of each gear in the following table

Table 5-1 – Main parameters of the gears

		Gear 1	Gear 2	Gear 3	Gear 4
Number of teeth	Z	22	68	34	75
Trans. Module	m_t	1.5mm	1.5mm	2mm	2mm
Trans. Press Angle	α_t	20º	20º	20º	20º
Helix angle	β	30º (L.H.)	30º (R.H)	30º (R.H.)	30º (L.H)
Normal Module	m_n	1.29mm	1.29mm	1.73mm	1.73mm
Normal Press. Angle	α_n	17.32º	17.32º	17.32º	17.32º
Face width	b	42mm	42mm	30mm	30mm

Due to some errors in the software, it wasn't possible to create a helical gear with those dimensions because the program crashe, so it was necessary to reduce the face width up to 10 mm for the helical gears. This should not affect the results of TE too much since the face width mainly affects the strength of the gear and its life.

Once we know the number of teeth of each gear we can calculate the transmission ratio of each stage, as well as the transmission ratio of the whole system.

$$r_1 = \frac{Z_2}{Z_1} \qquad r_2 = \frac{Z_4}{Z_3} \qquad r_T = \frac{Z_2 Z_4}{Z_1 Z_3} = r_1 r_2 \tag{5.1}$$

Table 5-2 – Transmission ratios

	Stage 1	Stage 2	Total
Transmission ratio	3.091	2.206	6.818

The position of the gears will be as shown in the figure Figure 5-2. A misalignment angle has been introduced between gears 1 and 2 and gears 1 and 4, the angle Φ has a value of 30º. The objective of such angle is to reduce the total length of the gearbox, which is its biggest dimension and thus we will reduce the total size to obtain a more compact transmission system.

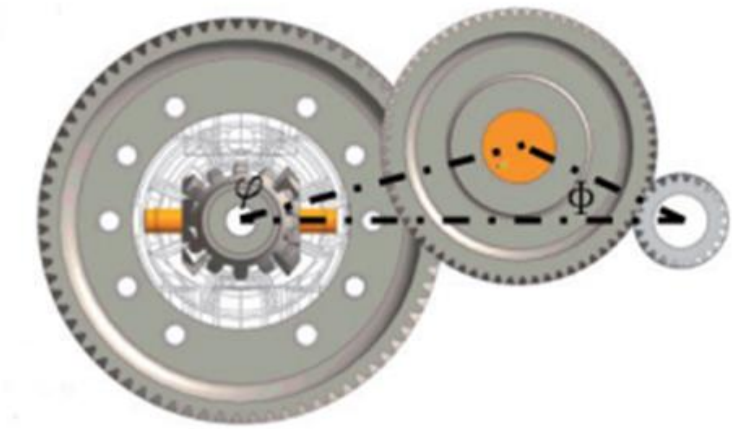


Figure 5-2 – Gearbox arquitectura [1]

With all the main parameters defined we can calculate the rest of the dimensions following the formulas described in chapter 2.4 and those dimensions are presented in the next table in mm.

Table 5-3 – Gears dimensions (mm)

		Gear 1	Gear 2	Gear 3	Gear 4
Pitch diameter	D	33	102	68	150
Base diameter	D_B	31.01	95.84	63.90	140.95
Addendum	h_a	1.5	1.5	2	2
Dedendum	h_f	1.875	1.875	2.5	2.5
Tip diameter	D_t	36	105	72	154
Root diameter	D_r	29.25	98.25	63	145
Center distance	d	67.5		109	
	x	0	-58.46	-58.46	-162.2
Center location	y	0	33.75	33.75	0
	z	0	0	-46	-46

5.2 Model

The first step to create our model would be to open Adams View, and we select *Create new model*, we have to select the name of the model, the working directory and the units of the model, in our case we will select MMKS (mm, kg, N, s, deg), The gravity can also be changed but for our model does not affect.

5.2.1 Advanced 3D Contact parameters

Once we have all the dimension we need to define the gears we open the *Create Gear Pair* in Adams inside the *Machinery* module. A wizard box will open where we will have to introduce all the parameters we need.

The first step is the type of gear, where we will select *Spur* or *Helical* depending on which model we are preparing.

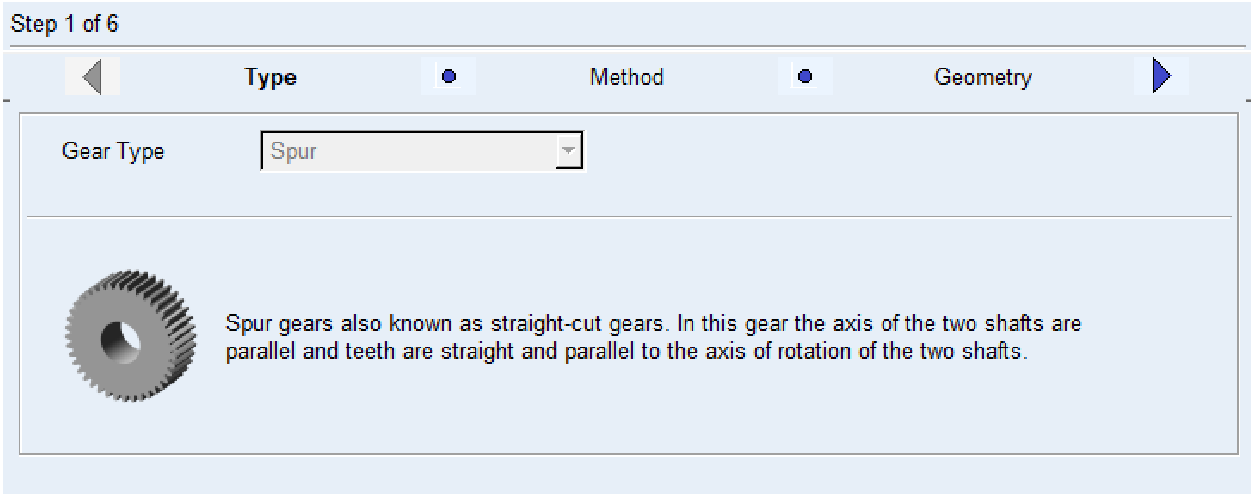


Figure 5-3 – Insert Gear Pair wizard - Type

In the second step, we must select the method for modeling the gear pair that Adams will use. Here we select *Advanced 3D Contact* as we have explained before.

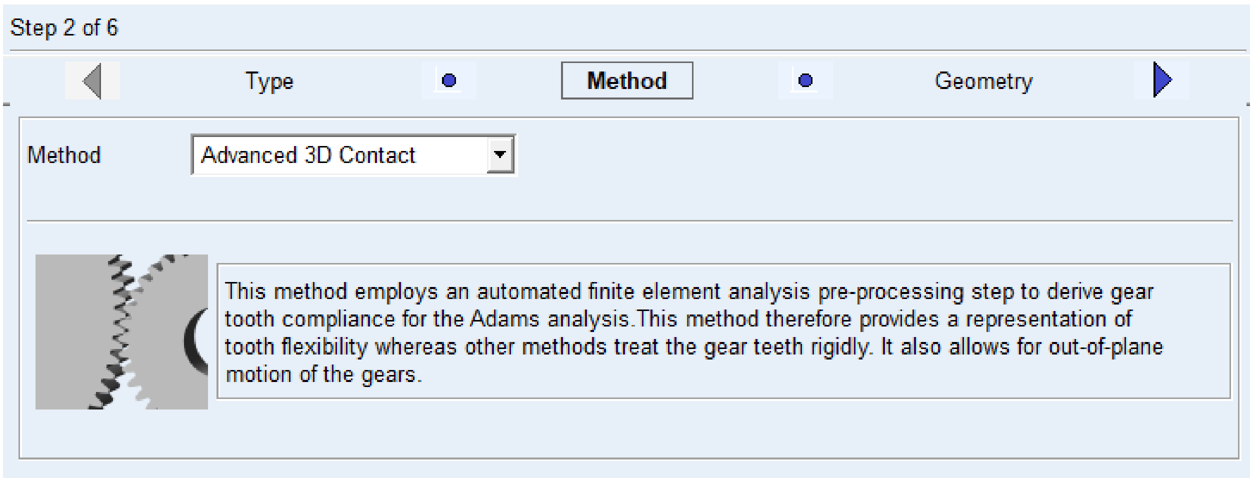


Figure 5-4 – Insert Gear Pair wizard - Method

The third step refers to the geometry of the gear pair, where we must introduce the module and the pressure angle, helix angle, center location, number of teeth and gear width of each diameter. It should be noted that for helical gears, the module that should be introduced is the Transverse module, not the normal one, and the same goes for the pressure angle. Rim/Bore Diameter refers to the internal diameter of the gear since Adams models only the crown of the gear as will be shown later, and this diameter must be within some limits. Here we can also select if one of the gears is internal, but in our case, all of them are external gears. There are also different tabs where we can modify parameters related to Tooth Profile and Tooth Modification, but those are not the object of this project so will be left as default. For a deeper understanding of those options and their effect on the Transmission Error, the thesis [21] can be consulted. There is a final tab to modify Mesh Properties, for the mesh of the FEM that Adams runs. This tab can be useful to tune in order to modify computational time and precision of the results.

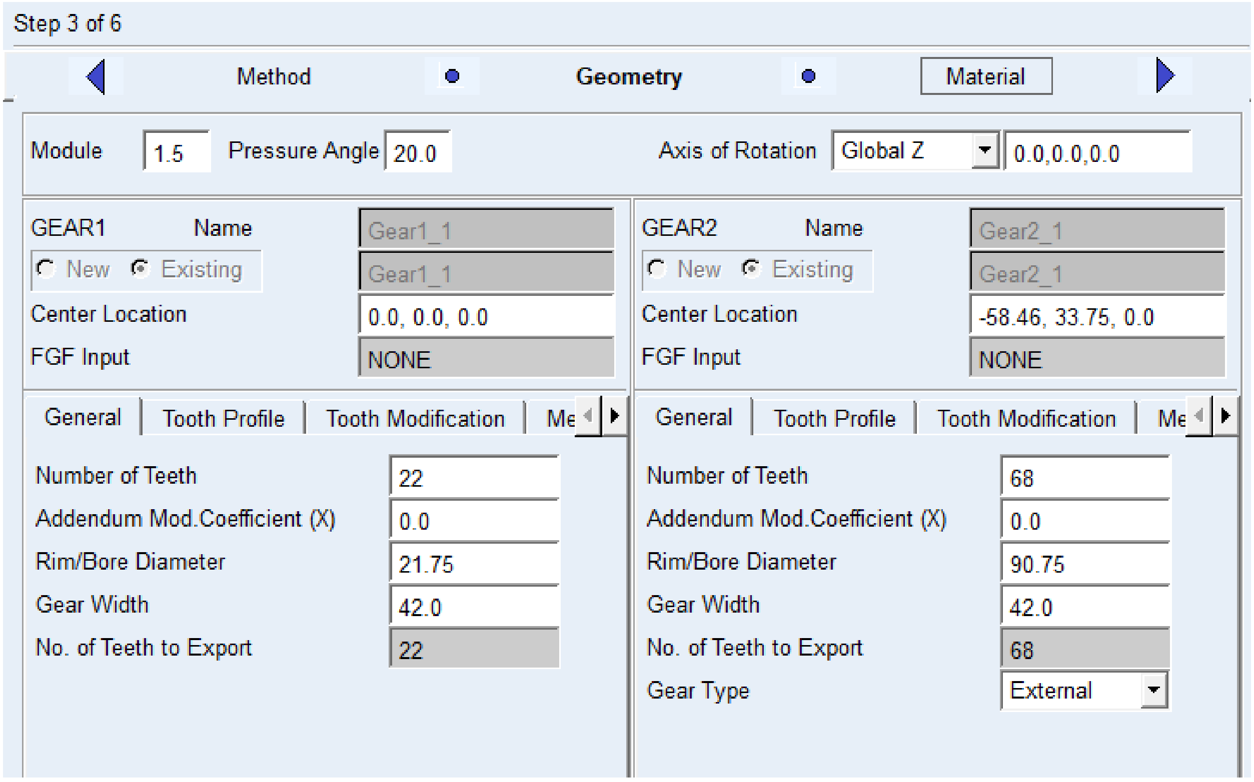


Figure 5-5 – Insert Gear Pair wizard – Geometry

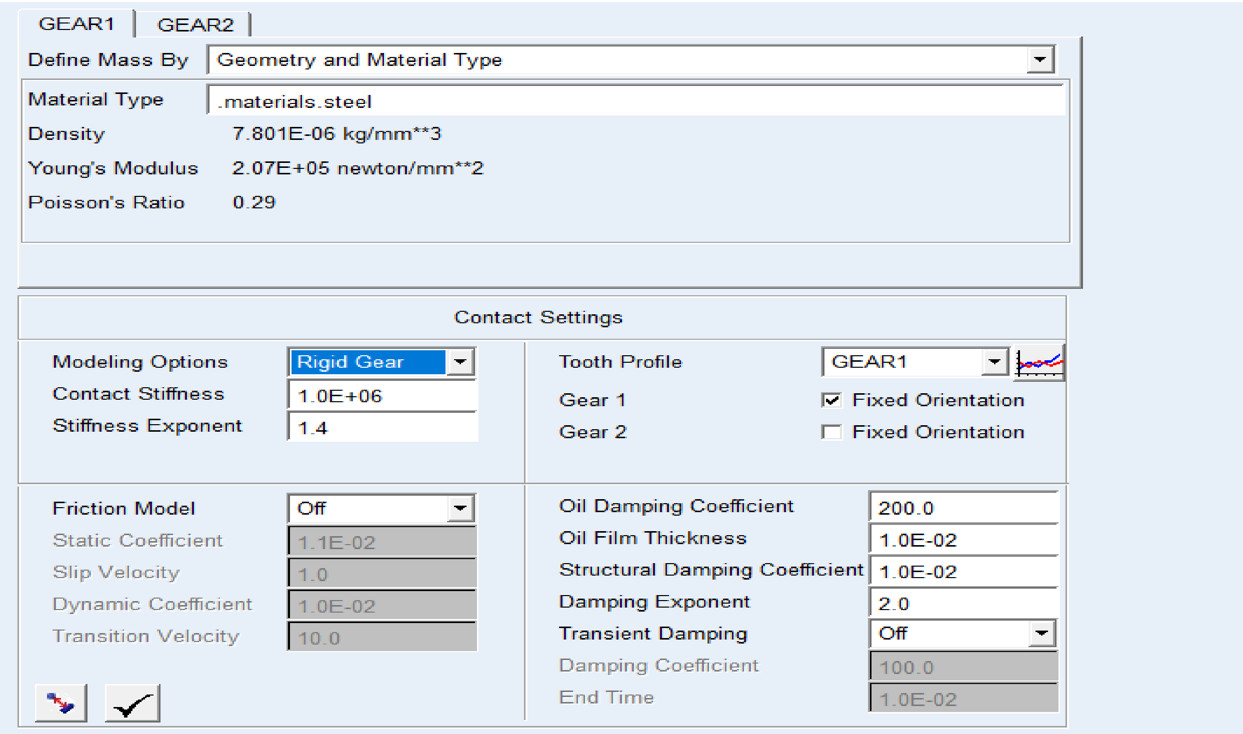


Figure 5-6 – Insert Gear Pair wizard - Material

Next step is related to the *Material.* Here we can select the material type of the gear, which will be steel, and the program will calculate its mass and inertia matrix according to the geometry previously defined, or we can also input the matrix number by number if we knew its values. In this step, we can also define the Contact Settings such as friction, oil damping and more, but these options will be left as default. Finally, we must select the Modelling option, among three choices RunTime, PreComputed and Rigid, as we explained in 4.3. For external reasons of this project, due to the fact that this one was a very recent release, there was a bug in the software which made impossible to use the RunTime option to model the gear contact, so all the studies have been carried out with the Rigid option, which may not be the best option in terms of precision but is quite a step forward compared with the previous option thanks to the Finite Element meshing of the gears when calculating the reaction forces.

The final step is *Connection*, where we have to define the constraints connecting the gears to the rest of the bodies of the system. In this first model since we are only modeling the meshing of the gears and not considering any deformation of other components the connection will be a perfect rotational constraint of the gear to the ground of the model which will allow them to rotate freely around their axis of rotation while restraining the rest of degrees of freedom. But for the gear 3, we must introduce a fix constraint to the gear 2, since both gears are mounted on the same shaft in the real gearbox, and they must have the same rotational speed.

Figure 5-7 – Insert Gear Pair Wizard - Connection

This process must be done for both gear pairs and the program will create the corresponding bodies for each gear as well as a *Gear System* for each gear pair where a lot of information about the gear contact will be calculated, such as contact force, distance and misalignment, friction, contact pressures, transmission error and many more variables.

With all the gears defined and generated by the program, there are still some details left to be defined so the model can be completed.

5.2.2 Measures

One of the strongest points of Multibody Simulation software is that once you have your model generated you can add *Measures* to monitor almost any variable you can imagine in the results, position, relative distance, speed, acceleration, force, torque, angular displacement, anything.

Even though by creating the gear pairs with the *Advanced 3D Contact* tool we can see the Transmission Error of each pair in the results, we have decided to monitor the value of TE with *measures*, since the one given by the program is not clearly defined, the units are not specified, and can only measure the TE of each pair, not the total TE of the whole system. For these reasons, *measures* for the TE will need to be created.

First of all, we need to create markers, which are construction points of reference that will be attached to the bodies and we will measure the movement of those markers. So, we create one marker for each gear in the center of the gear connected to each gear and another one in each center connected to the ground. Then with those markers, we create 3 *measures* for the angle of rotation of each axle. Once we have the angle of rotation of each axle we can calculate the TE as we have seen previously.

$$TE_{ang-12} = \theta_2 - \frac{Z_1}{Z_2}\theta_1 \qquad TE_{ang-34} = \theta_4 - \frac{Z_3}{Z_4}\theta_3 \qquad TE_{ang-total} = \theta_4 - \frac{Z_1}{Z_3}\frac{Z_3}{Z_4}\theta_1 \tag{5.2}$$

But those *measures* are in degrees, and we are interested in the linear TE so we need to convert them into first into radians and then into µm.

$$TE_{\mu m-12} = TE_{ang-12}\frac{D_2}{2}\frac{\pi}{180}1000 \qquad TE_{\mu m-34} = TE_{ang-34}\frac{D_4}{2}\frac{\pi}{180}1000$$

$$TE_{\mu m-total} = TE_{ang-total}\frac{D_4}{2}\frac{\pi}{180}1000 \tag{5.3}$$

5.2.3 Forces

There is only one force we need to introduce as an input in our model, and that force is a resistant torque that will be applied on the output gear (gear 4) opposing to the direction of motion. The value of this resistant torque will vary depending on the test we are running. We will have two different conditions, *loaded* and *unloaded*. On the loaded condition a torque of 10Nm will be applied which is the load resistance torque of the vehicle calculated in [1]. And for the unloaded condition, we will use a torque of 0.1Nm. Even the unloaded condition must have some small load because if the output wasn't loaded the gears would rattle too much and the values measured would not be a result of the transmission error but due to the excessive rattle of the gears.

5.2.4 Motion

Another condition we must impose in the model is the motion of the input gear. So, we go to the *Motions* tab in Adams, add a *Rotational Joint Motion* and we select the rotational joint between the first gear and the ground. The speed of this motion will also vary for the different simulations. We have two speed conditions, a *low speed* of 0.2 rad/s (1.9 rpm) that will be used to simulate static conditions, and a *high speed* of 945 rad/s (9024 rpm) to simulate dynamic conditions. These speed conditions are the same as the ones used in the previous thesis [4].

5.2.5 Simulation controls

Finally, we have the simulation controls, since this is a dynamic simulation of a system that it is evolving in time we need to specify the duration of the simulation as well as the number of steps or the size of the step. These values will vary from simulation to simulation, making sure that the *End Time* of the simulation is long enough so we have reached a permanent state of the system, and that the *Step size* is small enough that the sampling frequency can capture the real evolution of the variable without any antialiasing problems.

Having defined all the elements necessary and all the inputs required our model is completed and we can proceed with the analysis.

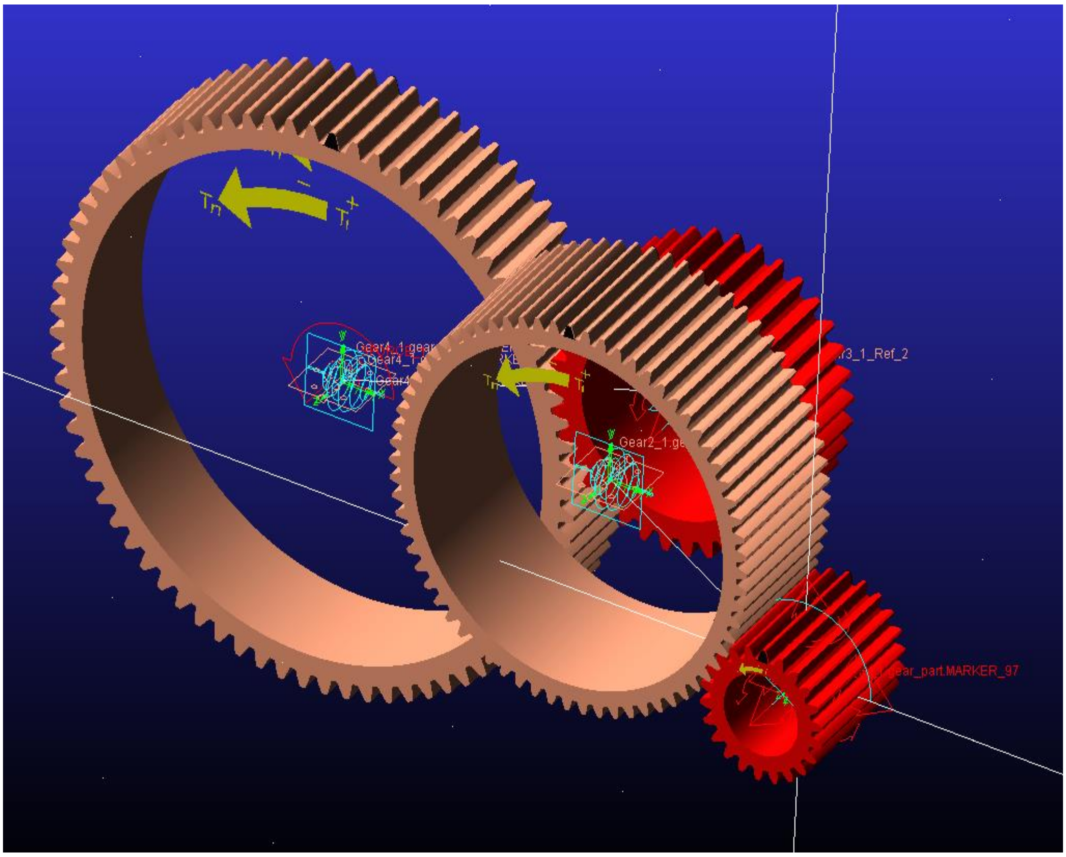


Figure 5-8 – Spur gear rigid model Adams

6 RESULTS FOR THE RIGID MODEL

This chapter will be dedicated to describing the analyses carried out on the first model as well as presenting and discussing the results obtained from those analyses.

The first model is an ideal model, we are not accounting for any flexibilities in the model, all the elements are completely rigid and the connections of the gears to the ground are also perfectly rigid. This means that no deformations will occur during the simulation, and the distance between the centers of engaged gears will remain constant and with its design value. These assumptions may make the model not suitable for real applications where a high precision is required but is more than enough for most applications trying to simulate gears and it will help us understand the effect of certain parameters, more precisely speed torque, and type of gears will have on the TE.

6.1 Parameters under study

As we have commented on this model we will run analyses to investigate the effect of the input speed, the resistant torque and the type of teeth on the value of TE. For this, we will run analyses with *helical* and *spur* gears, *loaded* and *unloaded* on the output, and with *low speed* and *high speed* on the input, as explained in 5.2.3 and 5.2.4. We have three different variables with two different values each, what gives us a total of 8 different analyses. In the table below, we can see the different analyses carried out with their corresponding conditions and the code they receive to named them.

Table 6-1 – Rigid model Analyses

Code	Teeth type	Input Speed (rad/s)	Resistant Torque (Nm)
Spur _S02_L100	Spur	0.2	0.1
Spur_S02_L10000	Spur	0.2	10
Spur_S945_L100	Spur	945	0.1
Spur_S945_L10000	Spur	945	10
Hel _S02_L100	Helical	0.2	0.1
Hel_S02_L10000	Helical	0.2	10
Hel_S945_L100	Helical	945	0.1
Hel_S945_L10000	Helical	945	10

The values obtained in the analyses with low speed will represent what we have called Static Transmission Error (STE) in chapter 3.2, and the values obtained from the analyses with high speed will give us the value of Dynamic Transmission Error (DTE) since the speed is high enough for dynamic effects to play an important role. For the Manufacturing Transmission Error (MTE) we can use the tab *Tooth profile* in the wizard when creating the geometry of the gears to modify some factors of the tooth flank, cutter rack or involute to tune the profile of the teeth, but the problem is that since the MTE is originated by manufacturing errors that are very difficult to measure in the real life, these parameters will be very difficult to tune perfectly to our exact gear. Anyways these modifications are out of the scope of this thesis and will be left as default. Another source for MTE in our model would be the FEM mesh that Adams generates to solve the contact. This error could be reduced by using a finer mesh if it was necessary.

6.2 Results

First of all, we should comment on the results of all the *Measures* taken for a single analysis and the relation between the variables. On Figure 6-1, we can see a comparison of the different values of the angular TE for a single analysis which corresponds to the code Spur_S02_L100, but the conclusion obtained from this particular analysis can be extrapolated to the rest of cases. It is important to point out that the values obtained are negative, so the vertical axis of the graph is negative too, but we do not care about the sign of the TE but its absolute value, so when we compare two different variables we will be speaking about their absolute values. The first thing we see in this graph is that the TE of the *Pair12* is lower than the *Pair34* in mean value due to the lower torque it is been subjected to, but the peak-to-peak value (PPTE) is higher, this is because it is rotating at a higher speed and has a higher meshing frequency than the *Pair34*. And the value of TE of the whole system, *Total*, seems to be almost like the summation of the two pairs but it is not the exact value. This makes sense since in a gear train the total TE between the input and the output should be an accumulation of all the TE of each pair it is made of, but not exactly the same value of the summation since the different gear ratios must affect the impact of each TE on the total one.

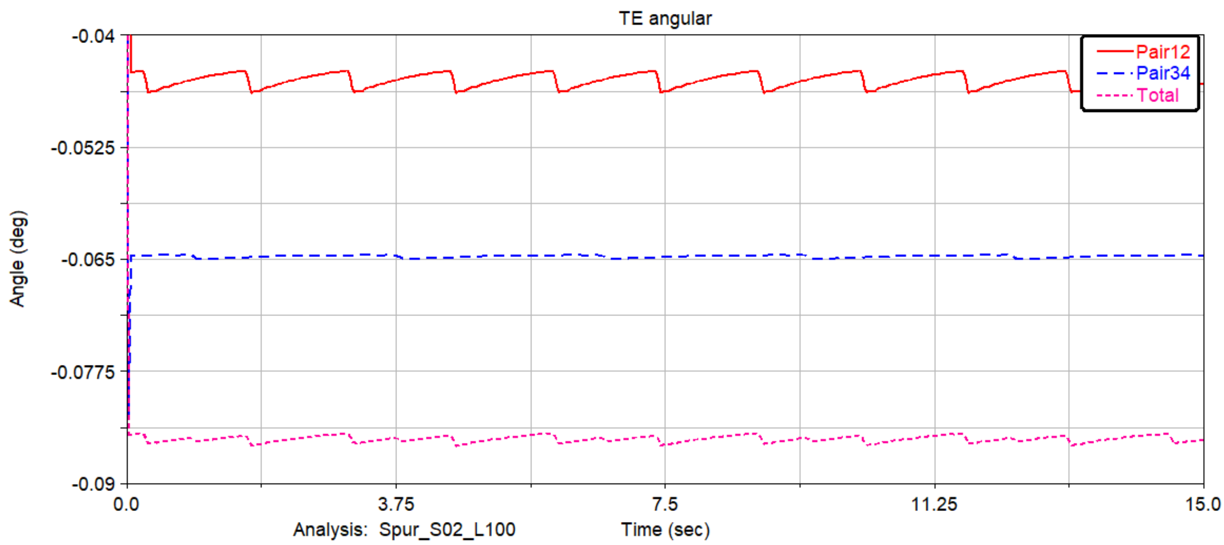


Figure 6-1 – Angular TE Comparison

On Figure 6-2, we have the values for the linear TE, which are same graphs as for the angular ones but each of them is scaled by the diameter of the corresponding gear to transform them from angular to linear displacements.

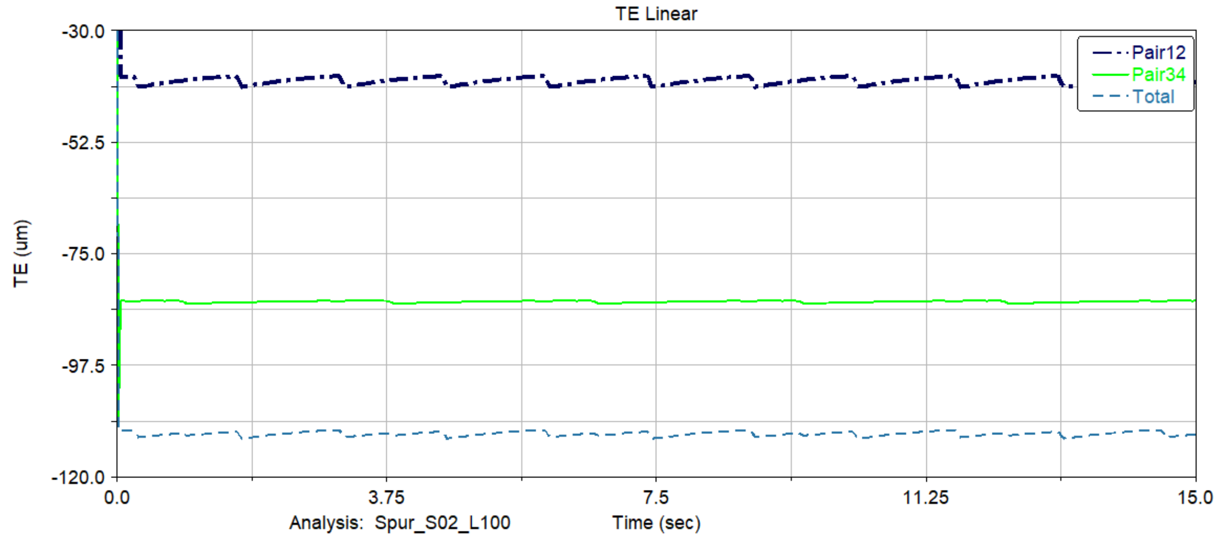


Figure 6-2 – Linear TE Comparison

Once we know the relation between the different TE inside a model, which is the same for all analyses, it is easier and gives more information to study only one of them when comparing two different analyses, for that reason from now on only the linear total TE will be studied and hereon when we speak about TE we will be referring to the same. In addition to that, the variable of TE is a continuous variable in time, which takes the form of a wave with the frequency corresponding to the meshing frequency of the specific gear pair, so it is fixed. Then to characterize these waves we only need two scalar variables which give us all the information we need about the curve. Those variables are: first the peak-to-peak TE (PPTE), the amplitude between the maximum and minimum values of the wave, and the mean value of the wave. Both variables are measured in µm. It is also worth remarking that the values we are obtaining are always negative, this is due to the signs when we defined the TE, but it doesn't really matter, only the absolute value is important. So, when reporting the values of mean TE, the absolute value will be used to help with clarity.

6.2.1 Effect of torque

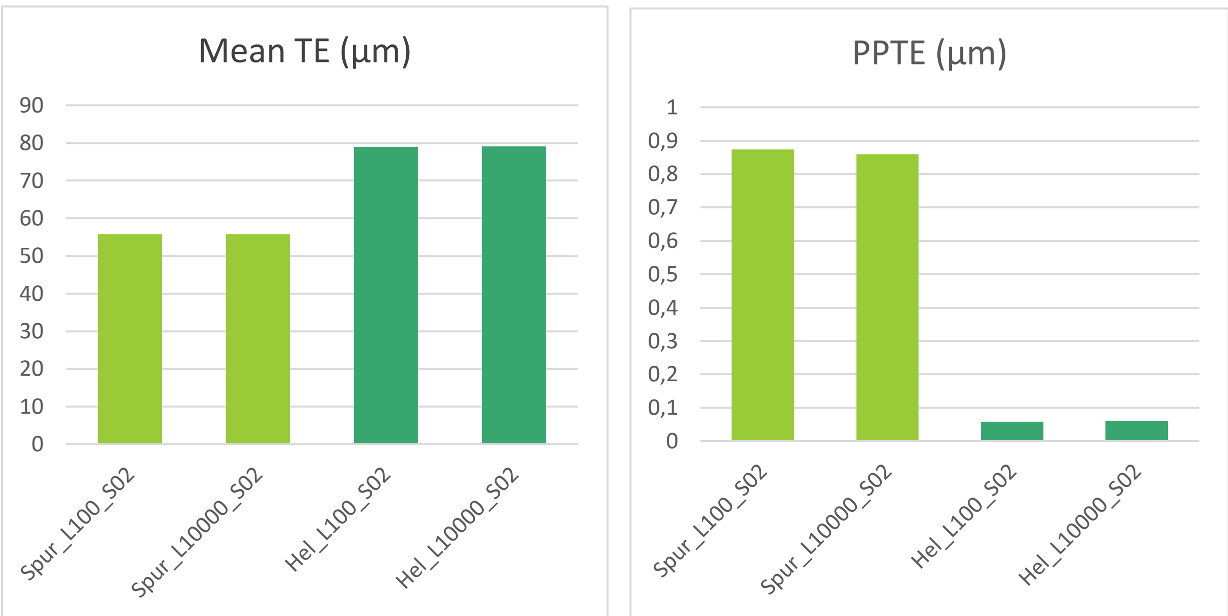


Figure 6-3 – Effect of torque at low speed

Looking at the results obtained from the analyses with low speed, which correspond to the STE, we can easily see that the increase in load does not affect the value of TE at all, not in peak-to-peak value nor in mean value. This is because our model is completely rigid, the gears are perfectly constrained to the ground, and the contact between gears is also modeled as *Rigid Gear* since the option *RunTime* could not be used. Using the *RunTime* option should give different results for different loads.

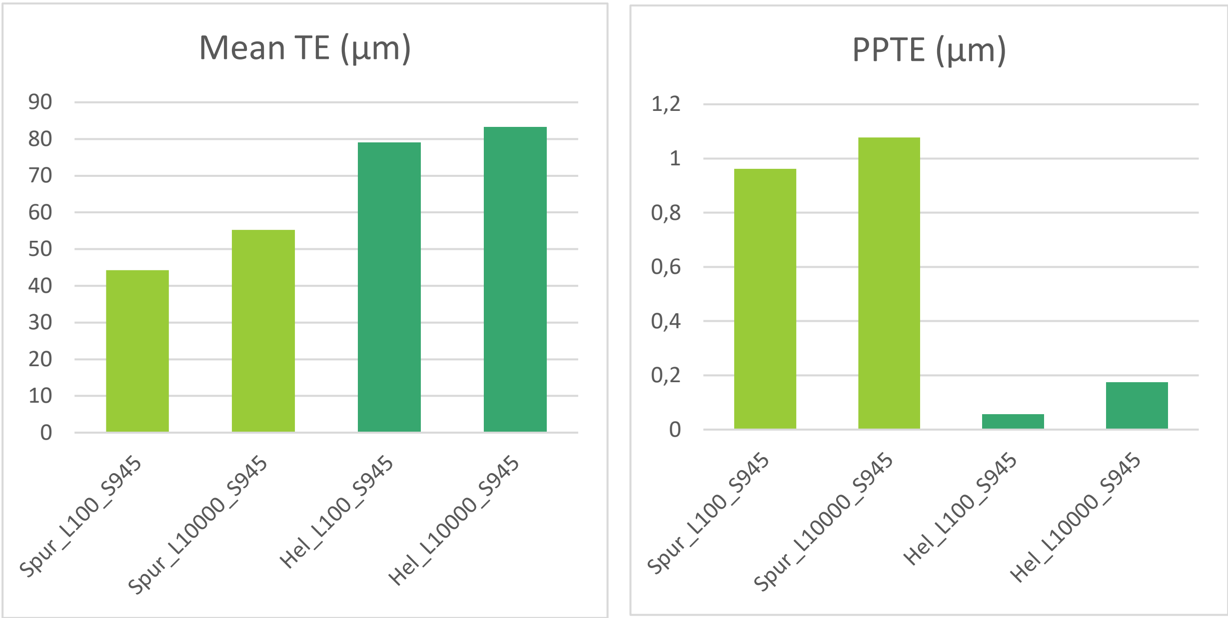


Figure 6-4 – Effect of torque at high speeds

However, at high speed the increase in torque it is notable. A higher load brings also a higher Mean TE, for both types of teeth, spur and helical which is to be expected. Regarding PPTE, it also increases considerably in both kind of gears when applying a higher torque.

6.2.2 Effect of speed

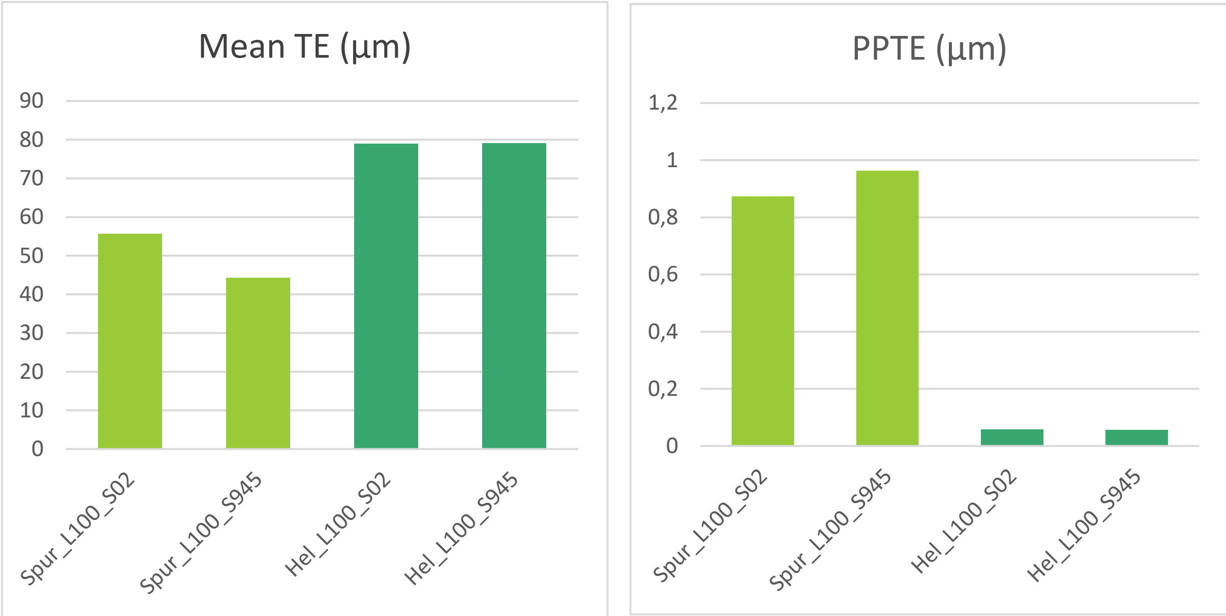


Figure 6-5 – Effect of speed with light load

When the gears are lightly loaded (Figure 6-5) going from low speed to high speed translates into a reduction in the mean value of TE of spur gears, probably due to the dynamics effects counteracting the effect of the load, but this effect does not appear on helical gears. The increase in speed brings also a slight increase in the value of PPTE. In the case of high load, there is no difference in terms of mean TE because the dynamic effects are not high enough to counteract the higher load. But the value of PPTE is considerably increased when going from low speed to high speed, which makes sense.

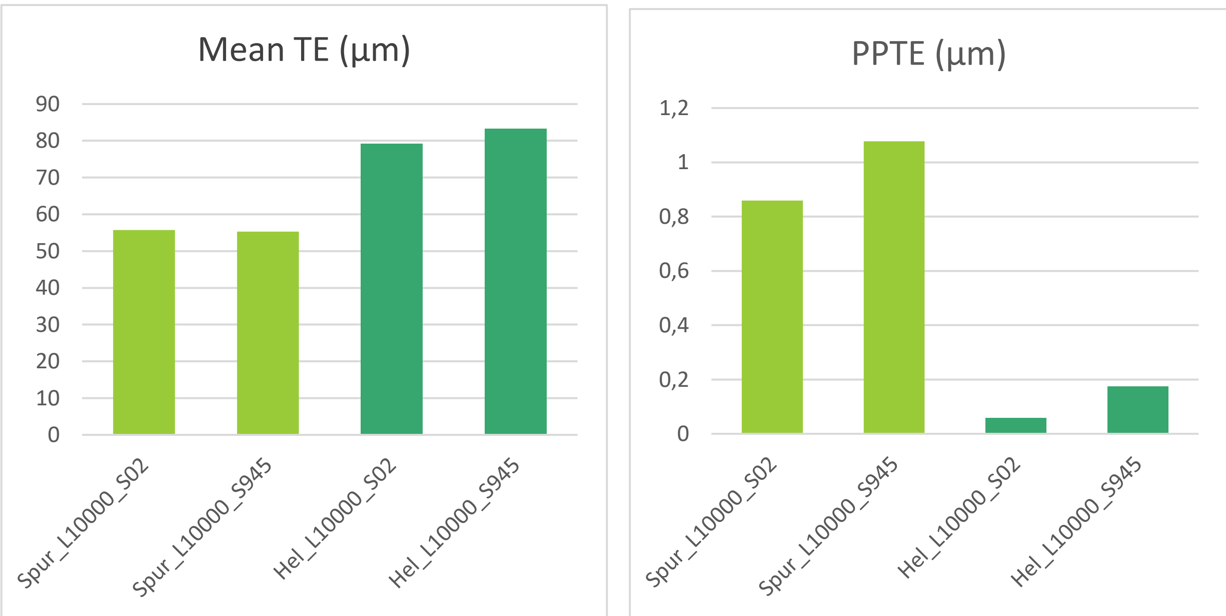


Figure 6-6 – Effect of speed with high load

It is also worth commenting the effect of the input speed on the computational time for the simulation since it had a very important impact. Most of the simulations for the *high-speed* cases took between 30 to 80 minutes or completing the analyses, which is a considerable amount of time, but it is reasonable for this kind of analyses. However, the simulations with *low-speed* conditions took much longer time, reaching up to 5 hours to obtain usable results. These longer computational times make the *low-speed* analyses not suitable for most situations.

6.2.3 Effect of the type of teeth

The effect of the type of teeth in the TE can be seen in any of the previous graphs but more accurately in Figure 6-7 and it is very clear. Helical gears have a cosiderable higher mean TE, due to the fact that the tooth is not engaged across all its length at the same time. But this reason is precisely why they also have a significantly lower PPTE because the teeth engage in a much smoother manner, reducing considerably the vibrations of the system.

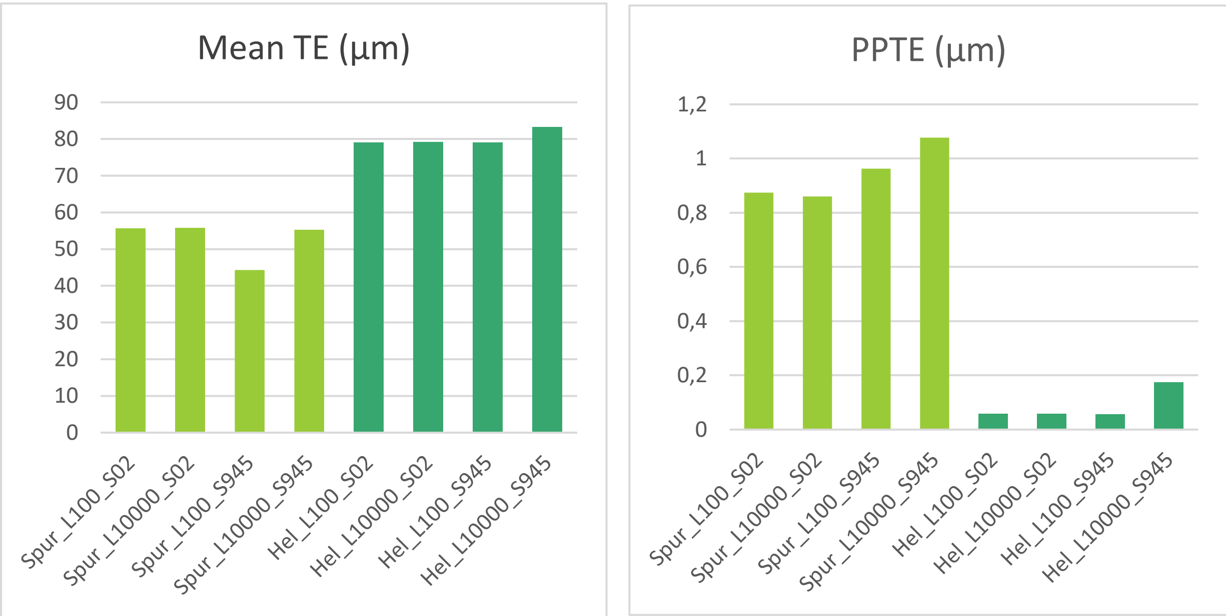


Figure 6-7 – Effect of the type of teeth

7 Description of the Flexible Model

After having studied the validity of the Rigid model we then proceed to a more complex model where we introduce flexible elements to study the effect they may have on the values of TE. In order to introduce those flexible elements, we are going to use Finite Elements (FEM) software, more precisely Simulia Abaqus, to model the flexible part and then that part must be converted into an MNF (Modal Neutral File) file that can be imported into Adams and work with it.

This chapter will be dedicated to describing the different FEM models that have been developed, the process of exporting that flexible part into an MNF file, the integration of the flexible parts into Adams and the rest of the model in Adams, as well as the problems and limitations that we have encountered during all the process.

First of all, it should be reminded that the flexible parts that we will be studying are the gear shafts, the gear webs, and the bearings. The crown of the gear cannot be made flexible because it is the part generated by Adams and it is where the contact is modeled. For the shafts and the webs, we will use FEM and for the bearings, Adams has a specific tool inside the *Machinery* module that allows us to model them with great precision. Another flexible element that could have been introduced is the gearbox case, but since we did not have a model of the case it will not be studied but anyway, the results drawn from these analyses can be extrapolated.

It should be noted that the dimensions of the shafts or the webs have not been calculated in any way since this is not a real case where we want to test a specific gearbox for a specific load scenario. The objective of this study is to test a particular methodology for estimating the TE on a general system. So, the dimensions of the elements have been estimated approximately according to the size of the gears.

7.1 Finite Elements Model

The parts we want to model using Finite Elements are the shafts and webs as we have commented. In the figure below, we have a CAD model of the shaft 2 which contains the webs for the gears 2 and 3. Looking at the structure of the part we can see it is constructed by a shaft which is a simple cylinder and two webs which are planar discs. The most direct option since it is a 3D model would be using 3D brick elements to model the whole part, but this would lead to a very high number of nodes and thus a high computational time. However, if we want to use a more efficient model, we should make simplifications that allow us to use simpler elements and the structure of the part is perfect for this. Such simplification would be to model the shaft with one-dimensional beam elements and the webs with two-dimensional shell elements rigidly connected in a point. This simplification would reduce considerably the number of elements and those elements would be simpler than in the 3D case, so it would result in a simpler and faster model.

We will model the shafts using the two options described and compare the results obtained from each of them, as well as comments on the advantages and disadvantages each model have.

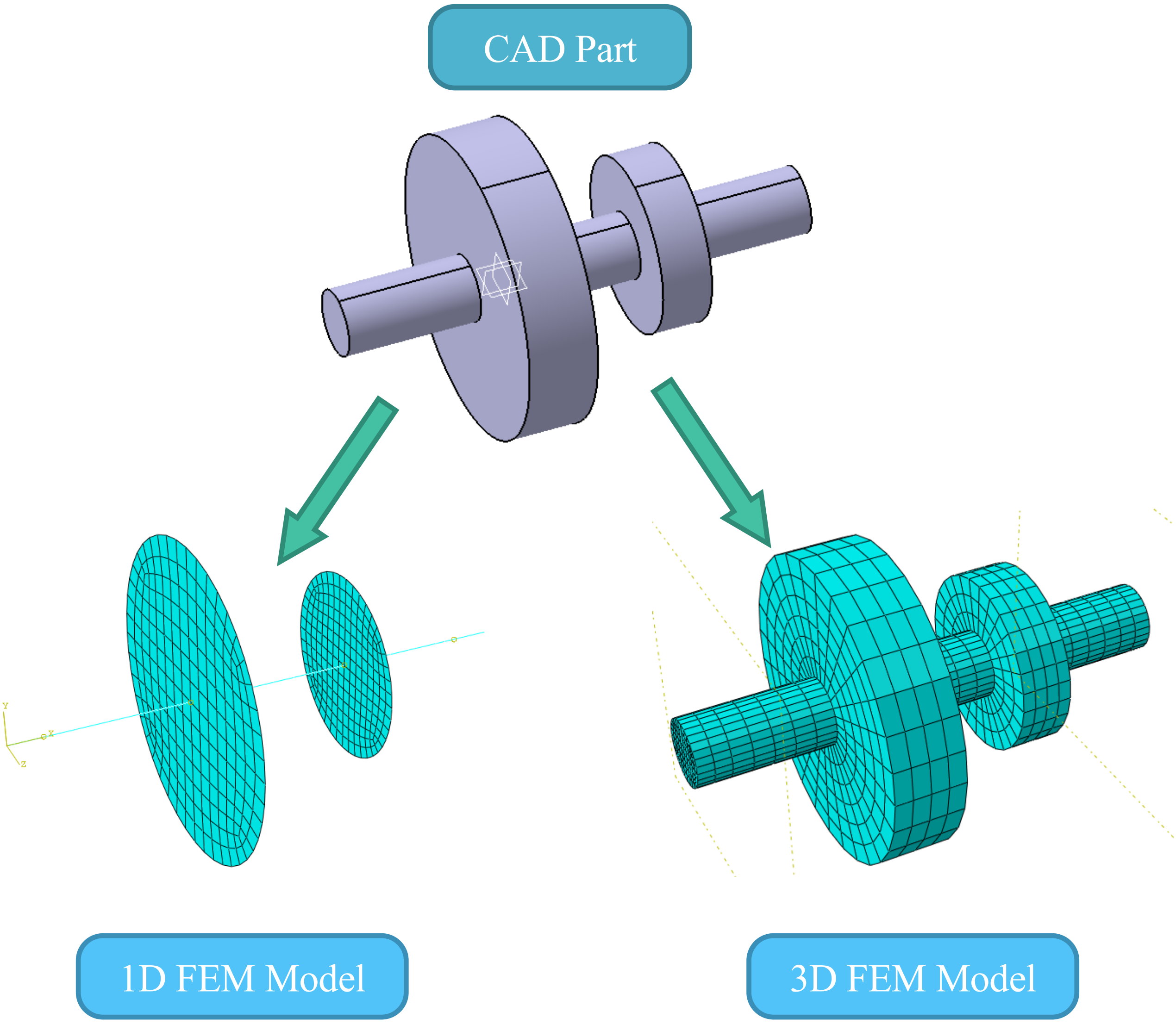


Figure 7-1 – FEM Models

The program used to generate the FEM models will be Simulia Abaqus by Dassault Systemes which is one the most powerful FEM software available nowadays and it is widely used in the industry. In addition, Abaqus has the advantage that allows us to export the models into an MNF file that can be later imported into Adams and used in our MBA model.

7.1.1 1D Model

For the 1D model, we will actually model the shafts and the webs as individual parts and then join them in Adams with constraints, this will allow us to make each individual part rigid or flexible for different analyses without having to create a new FEM model.

The first step when creating a FEM model is the geometry, which in this case is extremely simple. For the shafts, we only need to use the option *Create Part* select wire shape and draw a line with the length of our shaft. We also have divided the shaft in the points where the webs and the bearings will be, so when we mesh the part a node will be generated in that point. To do that we use *Create Datum Point* and enter the coordinates of those points, and then use *Partition Edge* to divide the geometry.

For the webs, we use again *Create Part* but select shell shape instead and draw a circle with the dimensions of the webs. And then we use *Partition face: Sketch* to divide the circle into 4 sectors so the mesh is uniform, and it generates a node in the center of the circle. With the geometries generated and divided we just have to go to the mesh module select the approximate size for the elements using *Seed part* with a size from 3 to 5 mm and then use *Mesh Part.* The meshed parts are presented below. The bigger parts will obviously have more elements.

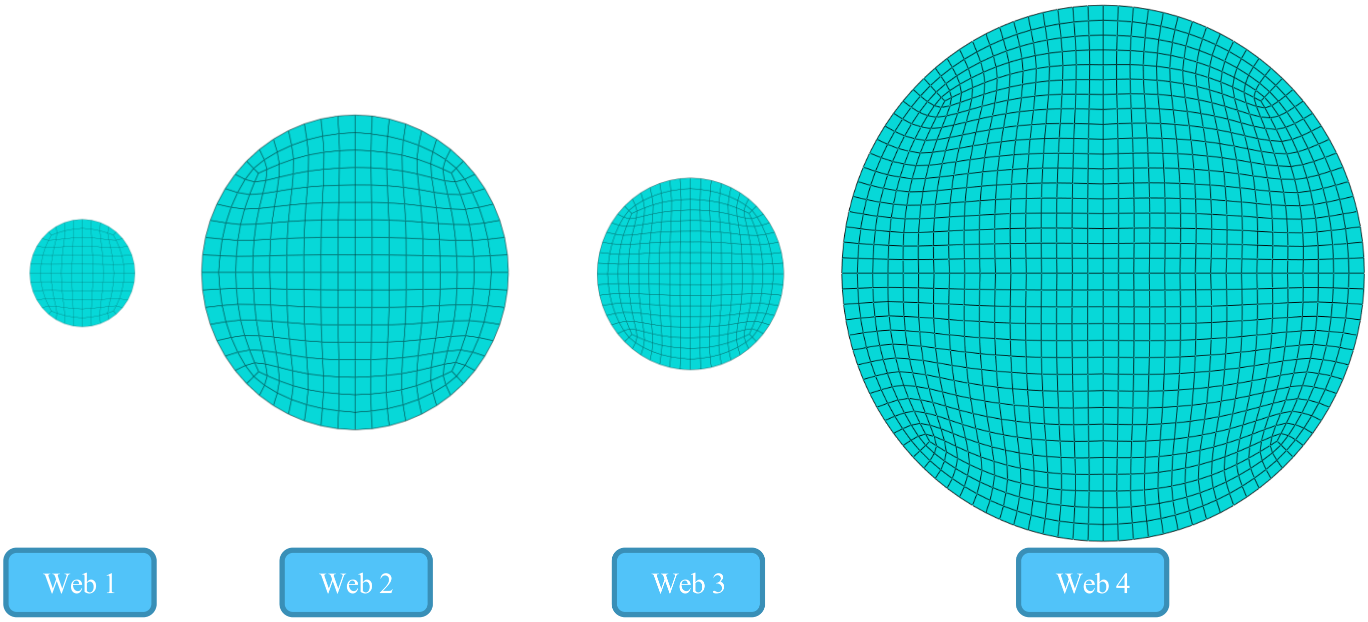


Figure 7-2 – Web Meshes

The next step would be to define the material, which will be steel. It is important to remember the units we are using, in this case, we are using Newtons (N) for the force and millimeters (mm) for distances, this gives us the stresses in Megapascals (MPa) and the mass must be in tonnes (T) to be consistent. So, the values needed to define the material are the Young's Modulus E=210e3 MPa, Poisson ratio υ=0.3 and density ρ=7.85e-9 T/mm^3. Then the sections are created for the webs as homogeneous and specify the material and the thickness of each web, after that those sections must be assigned to their corresponding webs. For the shaft, we must create a profile that will be circular and assign it to the shaft part.

To export the model into an MNF file we need to create a substructure in Abaqus and run a frequency analysis to obtain the modal shapes of vibrations, so to do that, we must create two different steps in Abaqus. *Create Step -> Linear perturbation -> Substructure generation* and *Create Step -> Linear perturbation -> Frequency*

The last step is the boundary conditions. For the step substructure we need to create a boundary condition called *Retained nodal DOFs,* this boundary condition retains the selected DOFs in the nodes that we choose, so when the substructure is created the only nodes that we will see are those retained ones, and those are the only nodes that will appear on Adams. So, we need to retain the nodes of the shaft where the webs and the bearings will be and also the ends of the shafts, and for the webs the nodes in the center of each web, and we will retain also 4 nodes distributed in the perimeter of the web to join them to the crown of the gears in Adams. And for the step frequency, we need to encastrate the parts in a single point, so they are not constrained to vibrate but we eliminate the movements as rigid bodies.

Once the whole model has been defined, the only thing left to do is create a job for each model and run the analyses. When we run the jobs the program will create some files that will be necessary to generate the MNF file.

7.1.2 3D Model

For the 3D model, the process is very similar. However, the geometry for the 3D model is more complex to generate in Abaqus, so it will be generated in CAD software, specifically Catia, and then imported to Abaqus. Once imported the geometry files, we have to divide the parts using *Partition Cell* in smaller, simpler cells that can be meshed with a uniform mesh and generating nodes in the points that we want as we did for the 1D model. Then the parts are meshed, and we obtain something the following meshes.

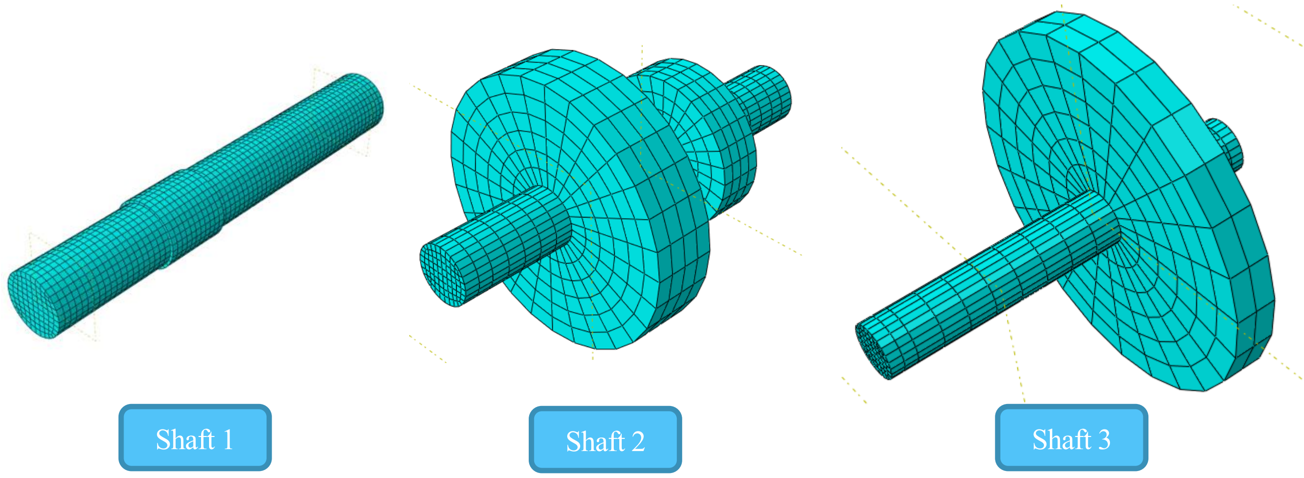


Figure 7-3 – 3D Model meshes

The material, the steps, and the boundary conditions are generated exactly as for the 1D model. The only thing that must be done in this model is that we have to generate Multi Point Constraints (MPC). An MPC is a kind of constraint that links the displacements and strains of a region or a number of nodes to a single node. This is necessary because when we use the *Retained Nodal DOFs* we select single nodes, and a node is a point in the part, it does not have dimensions, so if we apply a force in Adams to that dimensionless point it becomes a concentrated force creating a huge stress riser and the stresses that force produces are too high for the material. So, by using an MPC we connect that point to a section of the shaft and with the same force, the stresses are much lower. In addition, it is necessary to create MPCs linking the outer surfaces of the webs to the central point of the web, for reasons that will be discussed in chapter 7.2.

7.1.3 Abaqus/Adams Interface

With the FEM models completed, they now need to be exported into an MNF file so they can be used in Adams. This step was complicated to reach because even though Abaqus is capable of translating a substructure file into MNF thanks to an Abaqus/Adams Interface, such Interface cannot be found inside the main program, but must be executed through the command window, and the reference page inside Abaqus Help does not really explain the process very clearly. So, the process to transform the files obtained from Abaqus into an MNF will be described step by step.

The first thing to do is modify the INP file that Abaqus generates. The INP file is the input file that Abaqus sends to the solver which contains all the information from the model, but nothing has been calculated yet. The INP file is a text file that can even be written with a text editor such as Notepad. After generating our model in Abaqus we have to open the INP file it generates with a text editor and add the following lines at the end of the file inside the options for the step generating the substructure:

```
*SUBSTRUCTURE GENERATE, MASS MATRIX=YES, RECOVERY MATRIX=YES
*FLEXIBLE BODY, TYPE=ADAMS
```

After adding that to the model we save the modified file. Now we must run that modified file in Abaqus solver, to do that we can open Abaqus and run the job or we can do it with the Windows command line, here we will explain the second option since the next steps involve also the command line. To run the analysis, we need to open the Windows command line in the folder that contains the files, the easiest way to do that is to right-click the folder in question while holding the *Shift* key and choose *Run Command Prompt here…* and in the command line we must run the following commands:

```
ABAQUS JOB=job_name

ABAQUS ADAMS JOB=substructure_name UNITS=mmks MASS=Tonne
```

The first command will run the Abaqus analysis on the modified file. And the second command generates the MNF file, we have to input the name of the substructure file, which is usually the same as for the INP but followed by "_Z1", the units of our model are MMKS (mm, kg, N,s), but the mass is actually in tonnes as we saw in 7.1.1 so it must be modified with the last option. After running those commands, the MNF file should be ready to use and be imported into Adams.

7.2 Adams Model

The process of generating the Adams model will be exactly the same as followed in chapter 5.2 with the only exception of the last step when defining the gears when we have to choose the connection to the rest of the models we previously chose a rotational joint with the ground and now will be left unconnected.

Then we need to import the flexible MNFs generated, for that, we use the option *Flexible Bodies* → *Adams Flex: Create a flexible body* we specify the name of the part we are importing and chose the MNF file we want to import. And the part should appear in the Adams environment. With the parts in Adams, they need to be positioned with the *Rotate* and *Move* tools.

The next step is to connect all the parts using joints. For the 1D Model, the webs must be connected to the shafts with *Fixed Joint* at the center of each web, and they will also be connected to the crown of the gears with 4 *Fixed Joints* in the retained nodes that we selected in the substructure generation, we can see those joints in the figure below. For the 3D model, we only need to connect the crowns to the retained node in the center of the webs that was linked with an MPC to the outer surface of the webs.

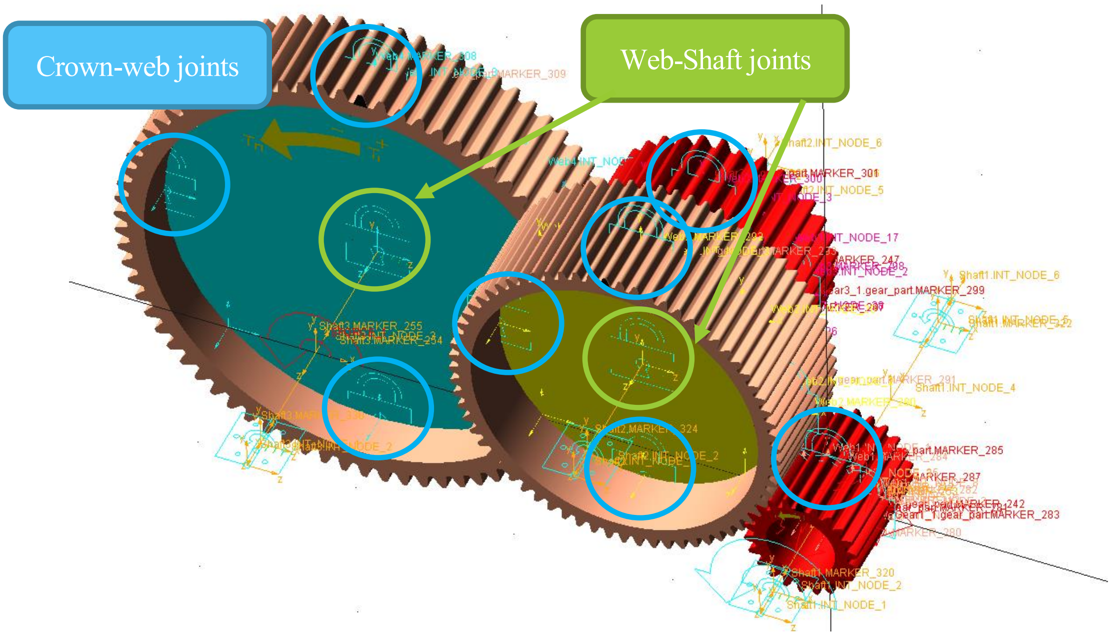


Figure 7-4 – Joints 1D Model

The connection to the ground will depend on the model we are trying to study. If the model does not account for bearings we just apply a *Revolute Joint* between the shaft and the ground, two joints per shaft in the corresponding locations.

If we want to study a model with bearings it is useful for us again the *Machinery* module inside Adams. We go to *Machinery* tab and open the tool *Create Bearing*. For this project we will use the method *Compliant* that allows us to generate a bearing with linear behavior and lets us specify the stiffness in each direction, but it should be mentioned that there is a more complex option, *Detailed*, that can be used if we need more precision when modelling the bearings you can specify the precise type of bearing and all its properties. In the geometry tab, we need to specify the point where the bearing will be created as well as its axis of rotation. In the connection tab, we must choose which part will be the shaft and which will be the housing (the ground in our case), we can also use the option *Impose Motion…* to impose the input motion to one of the gears of the first shaft. And finally, in this tab we can configure all the parameters of the bearing related to the stiffness and damping of the gears, those parameters will be left as default, except for the case of helical gears, where the gears in addition to the radial force also generate an axial force on the bearings, so the value of the axial stiffness must be increased. Since we are studying a general case we don't really care about the specific values, assuming that we are within a sensible range, if we were studying a specific case with specific bearings we would have to use the values provided by the supplier.

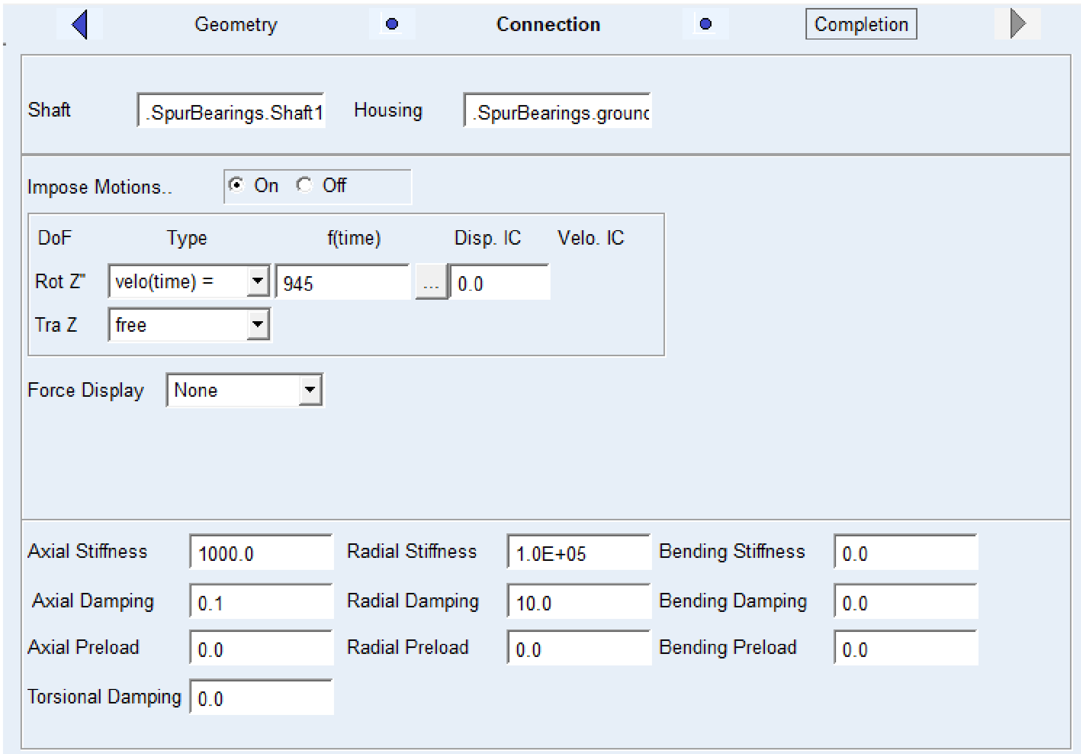


Figure 7-5 – Bearings Connection Details

After creating all the bearings necessary we have completed our model and the only thing left is to run the analyses. In the figures below, we can see some examples of the completed models.

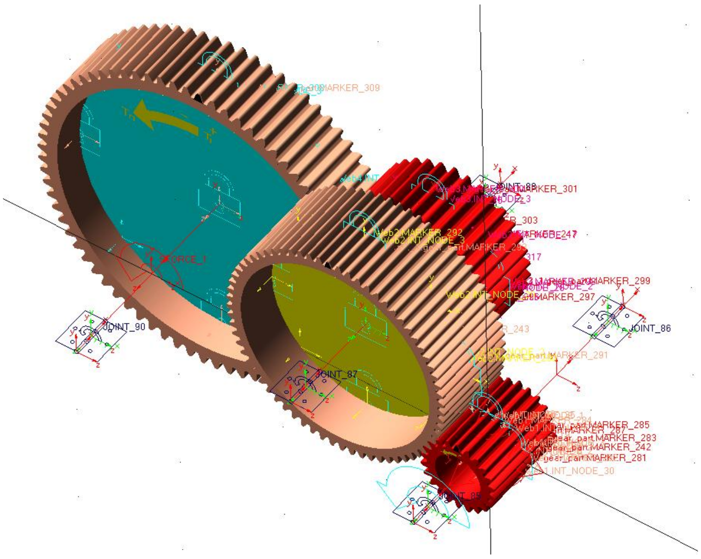


Figure 7-7 – 1D Model without Bearings

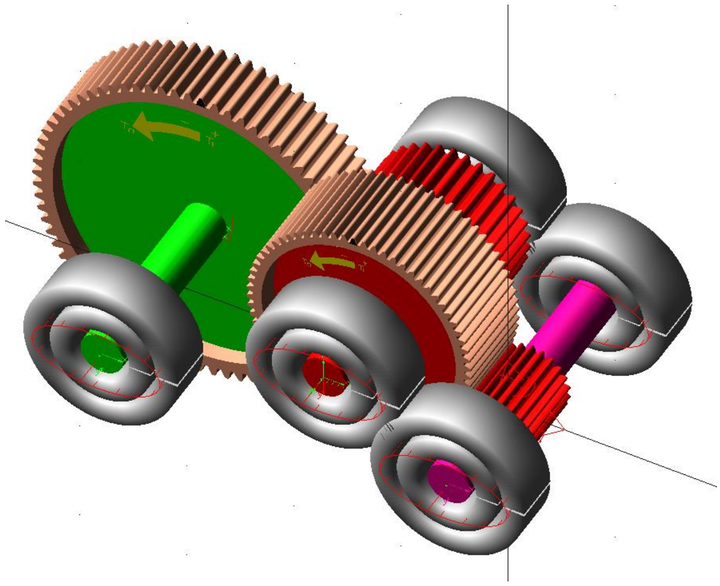


Figure 7-7 – 3D Model with Bearings

7.3 Limitations of this approach

This approach for simulating the dynamics of a gearbox taking into account the flexibility of some parts it is very versatile and has a huge potential, but however it also has certain limits that will be discussed in this section.

The first limit has already been discussed and it is related with the bug in the code that makes impossible the use of the *RunTime* model, so we have only been able to model the gears using the *Rigid Gear* option, and thus not considering the flexibility of the teeth. This is certainly not a limitation of this approach but some bug in our specific version of the program that we hope will be solved in future versions, however, it has been a limitation for this very project.

And another limitation when considering the deformations of the flexible parts is that Adams does not really solve the FEM problem for each part, but it tries to solve the deformed state by using Modal Superposition. Modal Superposition is the method used to analyze flexible parts in dynamic situations, and it uses the shape modes of the firsts natural frequencies of the part in a linear combination to try to obtain the actual deformed shape of the part. However, if the shape of the load is not simple, it may be difficult to calculate the deformed state using Modal Superposition. The more complicated the load the higher the number of modes that we will have to use to represent the deformed state. The number of modes we are using is a variable that can be selected when running the Abaqus analysis, and that information is later translated in the MNF file.

Finally, the last important limitation is that Adams is not designed to work with flexible parts but with rigid bodies, and this can be seen in the different *Joints* available within the program. Most *joints* are point joints, this means that they join two different bodies in one specific location. This is not a problem for rigid bodies, but for flexible parts, if we join two parts in a single point, all the forces will be transmitted from one body to the other through that single point. Resulting in a huge stress riser and much higher strains that they should be. There isn't a way to connect two surfaces in Adams as we could in a FEM program. So, in order to try to solve this problem, two approaches have been used. For the 1D model instead of connecting the gears to the webs in a single point, they have been connected through 4 different points and thus distributing the load at least in 4 points instead of only one. For the 3D model this approach could not be followed, because if we connect a flexible 3D part to a rigid body in more than one point the part becomes completely rigid and thus we lose all the information the flexible part can give us. So, to work around this problem what we did was modifying the FEM model, and inserting an MPC constraint that connects the outer surface of the web to the center point, and then in Adams connect that single point to the gear with a *Fixed Joint*, as we explained in 7.1.2.

8 RESULTS FOR THE FLEXIBLE MODEL

Just like chapter 6 was dedicated to describing the analyses and present the results obtained from the Rigid Model, this chapter will be dedicated to explaining the analyses carried out on the Flexible model, as well as their corresponding results

8.1 Parameters under study

On the Rigid Model we already examined the effect of the input speed and the resistant torque on the gearbox, so for this model, those parameters will be out of the scope of the study and will be set to a fixed value. More precisely all the analyses will be run under conditions of high speed (945 rad/s) and high load (10000 Nmm).

Instead of that, we will be analyzing the effects of introducing the flexibility of the different parts. To do that there is a very useful option in Adams, where we can select if a flexible body can behave as flexible or completely rigid. So, we will choose the specific flexible elements for each analysis.

We will also compare the results of the 1D Model against the 3D Model and those at the same time against the values obtained from the Rigid Model. And all those analyses will be carried out with both spur and helical gears.

In the table below, we can see an overview of the different analyses that will be run.

Table 8-1 – Flexible Model Analyses

Code	Teeth type	Flexible Bearings	Flexible Shafts	Flexible Webs
Flex1D_AllRigid	Spur	No	No	No
Flex1D_Webs	Spur	No	No	Yes
Flex1D_Shafts	Spur	No	Yes	No
Flex1D_ShaftsWebs	Spur	No	Yes	Yes
Flex1D_Bearings	Spur	Yes	No	No
Flex1D_BearingsWebs	Spur	Yes	No	Yes
Flex1D_BearingsShafts	Spur	Yes	Yes	No
Flex1D_AllFlexible	Spur	Yes	Yes	Yes
Flex3D_AllRigid	Spur	No	No	No
Flex3D_ShaftsWebs	Spur	No	Yes	Yes
Flex3D_Bearings	Spur	Yes	No	No
Flex3D_AllFlexible	Spur	Yes	Yes	Yes
Flex1DHel_AllRigid	Helical	No	No	No
Flex1DHel_Webs	Helical	No	No	Yes
Flex1DHel_Shafts	Helical	No	Yes	No
Flex1DHel_ShaftsWebs	Helical	No	Yes	Yes
Flex1DHel_Bearings	Helical	Yes	No	No
Flex1DHel_BearingsWebs	Helical	Yes	No	Yes
Flex1DHel_BearingsShafts	Helical	Yes	Yes	No
Flex1DHel_AllFlexible	Helical	Yes	Yes	Yes
Flex3DHel_AllRigid	Helical	No	No	No
Flex3DHel_ShaftsWebs	Helical	No	Yes	Yes
Flex3DHel_Bearings	Helical	Yes	No	No
Flex3DHel_AllFlexible	Helical	Yes	Yes	Yes

8.2 Results

After running all the simulations, we analyze the results obtained, starting with the 1D Model with spur gears. In Figure 8-1, we have the evolution of the total TE plotted against time, while in Figure 8-2 we can see the values of mean TE and PPTE for the different analyses so they can be compared. Both figures give us the same information, increasing the number of flexible elements in the system has two different effects on the total TE, it reduces the value of PPTE but increases the mean value of TE. Those were the expected effects since by introducing more flexible elements means that the total deformations will be higher, so the mean TE is higher, but at the same time those higher deformations reduce the peaks in the contact forces and thus reduce the amplitude of the vibrations and the PPTE. The impact of the flexibility of each part is not the same, introducing flexible bearings has the highest impact, followed by the impact of the flexible shaft, and in last place, we have the impact of the flexible webs, which has almost no noticeable effect. This is because the webs are 2D elements with the loads on the plane of the element which is much more rigid than the shafts which are 1D elements with the loads on the transversal plane.

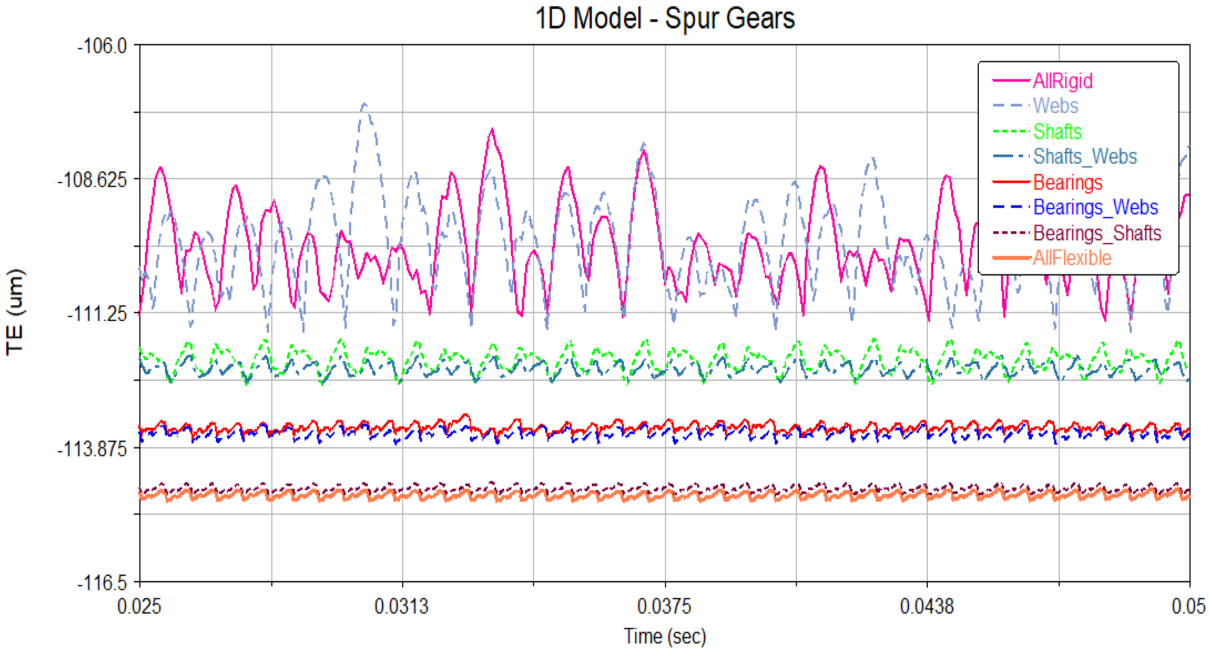
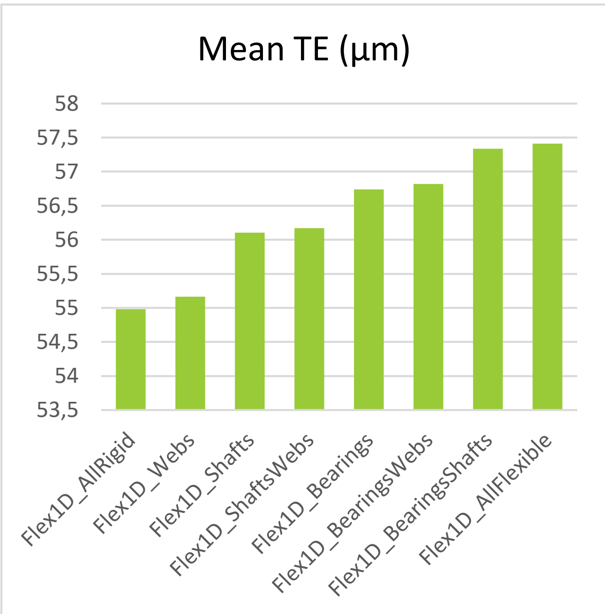


Figure 8-1 – TE on 1D Model – Spur gears (1)



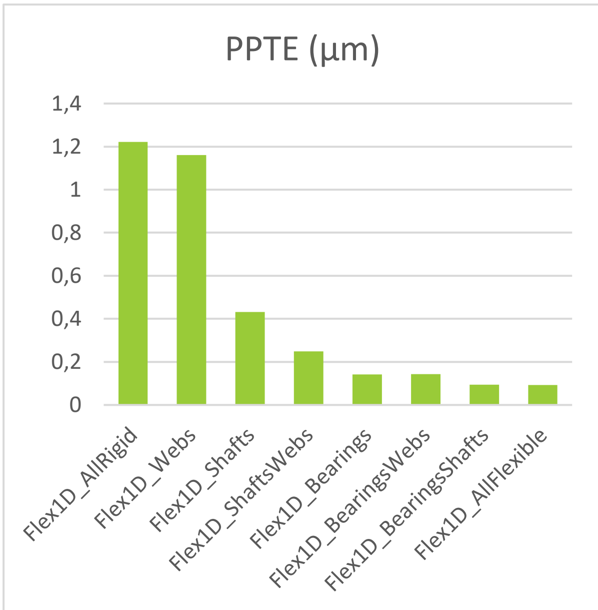


Figure 8-2 – TE on 1D Model – Spur gears (2)

With regards to the helical gears the results can be seen in Figure 8-3. The conclusions drawn from these results are the same as for the spur gears, with the exception that the values of mean TE are higher, and the ones of PPTE are lower, as explained in 6.2.3

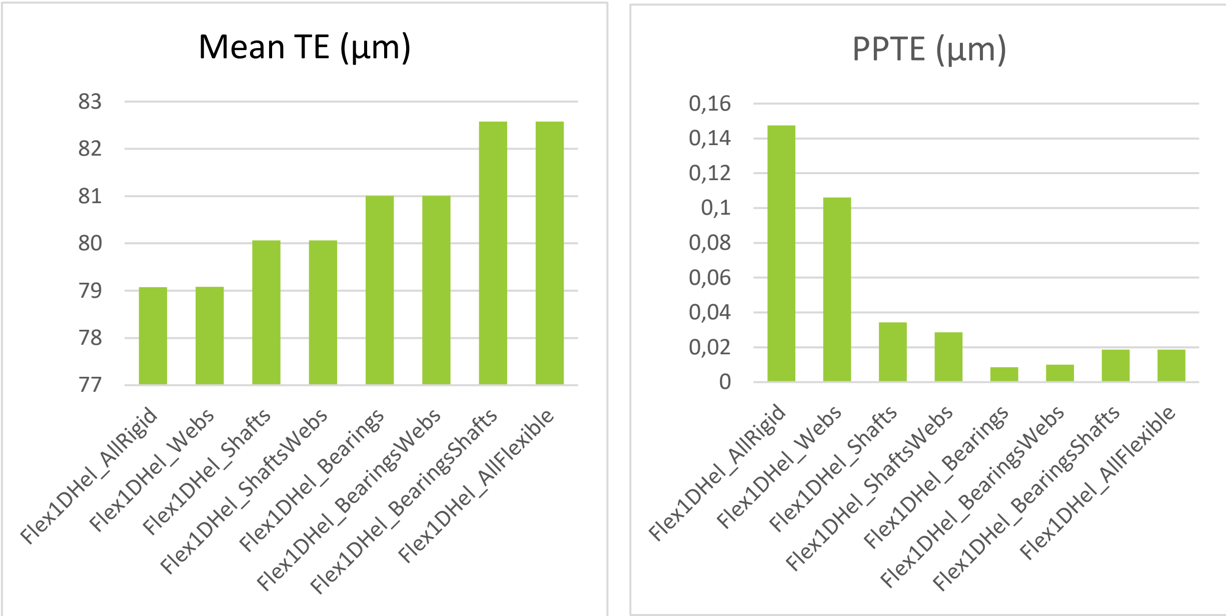


Figure 8-3 – TE on 1D Model – Helical gears

If we compare the results of the 1D Model against the 3D Model, shown in Figure 8-4, we can see that the results are really similar within a margin of error lower than 5% for both Mean TE and PPTE, with the exception of the Mean TE on both AllFlexible models, where the difference is 8.44%. Also, when comparing the results of the rigid analyses with the previous Rigid Model the difference is lower than 10%, although that difference can be due to the added masses of the shafts and webs that were not present in the Rigid Model.

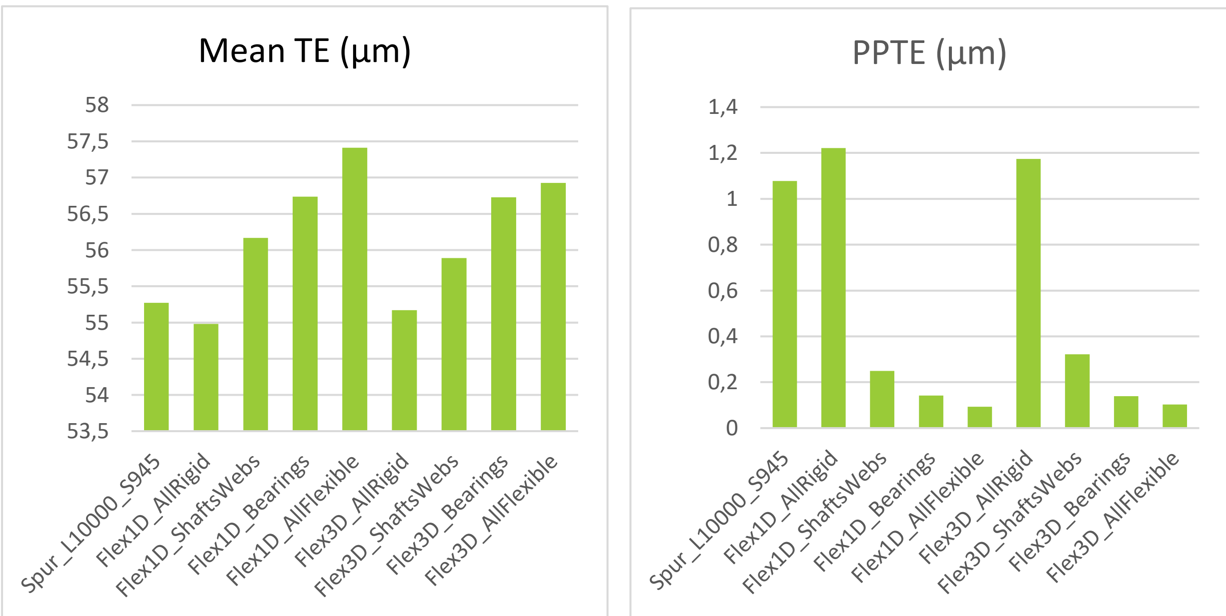


Figure 8-4 – 1D Model vs 3D Model – Spur gears

For the helical gears, we obtain similar results when comparing both models with helical gears, all the results have a difference lower than 5% except for the case of the Mean TE with all elements flexible where the difference is of 11.2%

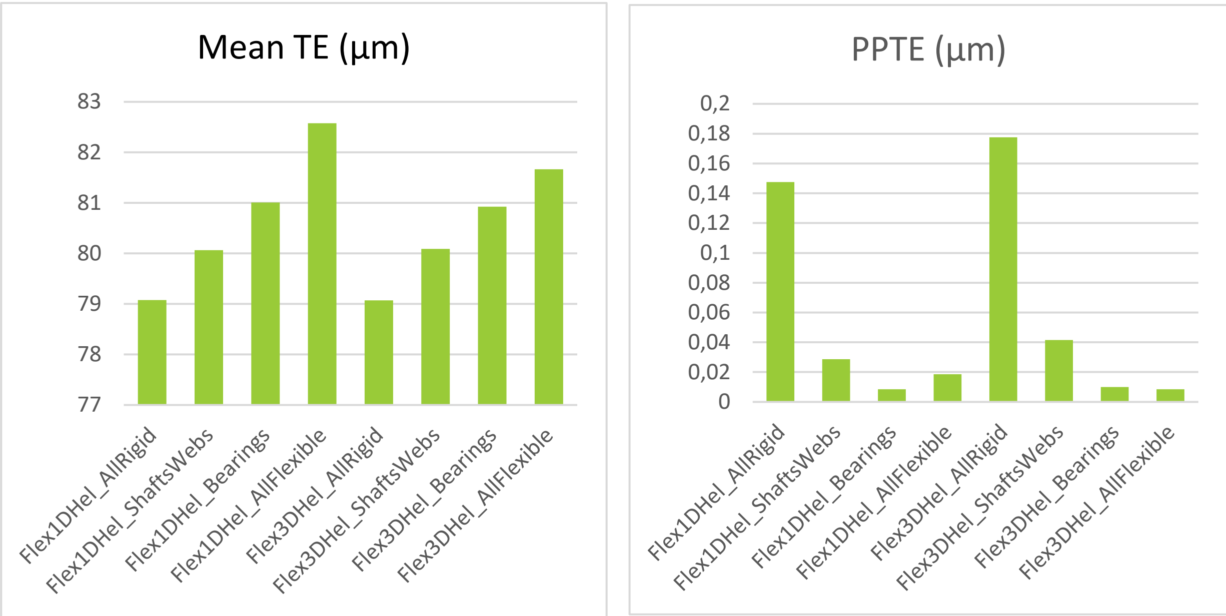


Figure 8-5 – 1D Model vs 3D Model – Helical gears

8.3 Problems encountered

After many tries and different approaches we managed to make the 3D Model work and could obtain some results from it, but the process to achieve a working 3D Model was difficult and several models had to be discarded. The main problems encountered will be described in this subsection.

The first problem was related to the concentrated forces. As we have already explained Adams is not designed to work with flexible bodies but with rigid ones, and the joints between bodies in Adams are located on single points. If we apply a joint, which will transmit forces, to a flexible body in a single node, this will result in very high concentrated forces and deformations and stresses much higher than they should be. So, to solve this problem the FEM model had to be modified to connect those single nodes to a region of the model with an MPC. Of course, this is a simplification and it does not represent the real behavior of the material, but it does not introduce too much error into the model and can be overlooked. This problem did not occur when working with the 1D Model, because in the 1D Model the single nodes represent a 3D space, and they have a thickness, or a cross-section assigned so the forces that appear in the flexible body are not concentrated forces.

The other big problem we found was that when using 3D flexible bodies, the simulation became very unstable due to numeric errors in the Solver of Adams. This meant that after setting up a model and start running the simulation it may start working without any problems and then after a certain time, it did not converge. This problem was solved by changing the Integrator inside the *Solver Settings* in Adams. The default integrator for dynamic simulations is called GSTIFF and had to be changed to WSTIFF. The main difference between them according to Adams Online Help [23] is that GSTIFF uses fixed coefficients for prediction and correction while WSTIFF uses variable coefficients. This change meant a more stable model and simulation but on the other hand, the time for the simulation was by a factor of 20% approximately.

9 CONCLUSIONS

Finally this last chapter will be dedicated to present the conclusions that have been obtained from the study as well as the strong and weak points of the approach followed.

The main objective of this project was to develop a computer model to analyze and predict the Transmission Error of a gearbox taking into account the flexibilities of the different elements of the system. For that, we used a combination of MBA and FEM software two of the strongest tools available nowadays in terms of physics and engineering simulations, the first is mostly used to model dynamic and kinematic problems with rigid bodies while the latter is the go-to tool when a structural analysis is needed (it can be also used for other problems such as heat transfer, fluid dynamics, electromagnetism, acoustics, etc). So, by combining both tools we are able to model dynamic problems with flexible elements in the system, these types of problems are very complex and complicated and would be practically impossible to model and solve with more traditional tools. But thanks to the strength and versatility of MBA and FEM we are able to solve these problems in a very reasonable time.

The first part of this project was dedicated to study and test the work of a new toolkit recently added to the latest versions of MSC Adams, the *Advanced 3D contact* toolkit. This toolkit allows users to model contact between gears with a very high precision, being able to modify many parameters of the gears and monitor many variables such as contact forces, displacements, misalignment and so on. For that the toolkit generates and solves a FEM model in the background to solve the contact of each pair of teeth. In order to test this toolkit, the Rigid Model was developed and with this model we tested that the effect of the input speed, the resistant torque and the type of teeth have on the TE. The results obtained from this model were coherent with what we should expect according to the theory but except for two details. First, since this is a digital model the gear generated could have a perfect involute profile and there would be no assembly errors, this level of precision is impossible to achieve in the real world, so our model is incapable of modelling the Manufacturing Transmission Error (MTE). The second detail is that due to some bug in the software we could not use the *RunTime* model for the contact, but we had to use the *Rigid Gear* which does not account for the deformation of the teeth, but we hope that this issue will be solved in future versions of Adams. For these two reasons the values obtained in terms of PPTE were quite smaller than we could expect in a real-world test, however the behavior of the model when changing speed, torque or teeth was the expected one and we could use the model to draw conclusions on the effect each of these parameters would have on a real gearbox.

The second part of the study was to try to introduce flexible elements into our previous Rigid Model. For that we followed two different approaches and generated two models to compare their results, those models were the 1D Model and the 3D Model, depending on the type of elements we use for the FEM model. We obtained good results from these two models, again meeting the expected results according to the theory, and obtaining very similar results with both models, the difference in the results were lower than 3% in most case and the biggest difference was below 8.5%. Seeing that the results of both models were practically the same we have decided to opt for the 1D Model as the best option for several reasons. First the 1D Model is much easier to generate in Abaqus, because it is a very simple geometry, the meshing process is automatic, there is no need to generate

MPC to eliminate stress risers, and the number of elements is much lower in the 1D case, leading to a considerable lower computational time to solve the model but the most important advantage of the 1D Model against the 3D Model is the stability of the simulation, in the first case there has been no problem in terms of stability while for the second case the system became very unstable leading to many failed simulations. In addition, when comparing the results for the different analyses of the 1D Model we can see that the effect of the flexible bearings and shafts is much more important than that of the flexible webs which is barely noticeable. The effect of introducing the flexibility of the webs is usually around 1%. For this reason, we consider that the best option to model the problem would be only with flexible shafts and bearings and the webs could be modelled in Adams as a rigid body, reducing the complexity of the model and the preprocessing time to generate it.

In conclusion we can consider this study a success. The *Advanced 3D Contact* toolkit has been proven to give good results coherent with what we should expect according to the literature, and with the second part of the project we have developed and refined a method to take into account the flexibility of the different elements in the system with good results too.

As recommendation for future work we leave the testing of the more complex contact model *RunTime* that could not be used in this project for errors in the software, and it can also be of interest a comparison the results obtained in a real-world test and a digital model following this approach tuned to model the specific gearbox of the real test rather than use a generic case as we followed for this project.

10 REFERENCES

[1] G. Belingardi, V. Cuffaro and F. Cura, "Multibody approach for the dynamic analysis of gears transmission for an electric vehicle," *Proceedings of the Institute of Mechanical Engineers,* vol. 232, pp. 57-65, 2018.

[2] J. D. Smith, Gear Noise and Vibration, New York: Marcel Dekker, 2003.

[3] D. B. Wellbourn, "Fundamental knowledge of gear noise," in *IMechE Conference on Noise and Vibrations of Engines and Transmissions 1979*, Cranfield, 1979.

[4] D. Dudley, Handbook of Practical Gear Design, McGraw Hill Inc., 1984.

[5] A. Lemanski, Gear Design S.A.E, Wharrendale, 1990.

[6] A. Stokes, Gear Handbook Design and Calculations, Oxford: Butterworth-Heinemann, 1992.

[7] S. P. Radzevich, Dudley's Handbook of practical gear design and manufacture, CRC Press, 2012.

[8] S. Harris, "Dynamic Loads on the Teeth of Spur Gears," *Proceedings of the Institute of Mechanical Engineers,* vol. 172, pp. 87-112, 1958.

[9] W. Mark, "Analysis of the vibratory excitation of gear systems: basic theory," *Journal of the Acoustical Society of America,* 1991.

[10] R. Munro, "A review of the Theory and Measurement of Gear Tranmission Error," *Proceedings of the First IMechE Conference on Gearbox Noise and Vibration,* pp. 3-10, 1990.

[11] D. Houser, "Gear Noise state of the art," *CETIM Internoise ,* vol. 2, no. 88, pp. 601-606, 1988.

[12] R. Tharmakulasingam, Transmission Error in Spur gears: Static and Dynamic Finite-element Modeling and Desing Optimization, Brunel University, 2009.

[13] J. D. Smith, Gears and theirs Vibrations, A Basic approach to understanding Gear Noise, The Macmillan Press LTD, 1983.

[14] R. W. Gregory, S. L. Harris and R. G. Munro, "Dynamic behaviour of spur gears," *Proceedings of the Institute of Mechanical Engineers,* vol. 178, no. 1, pp. 87-112, 1963-64.

[15] R. Gregory, S. Harris and R. Munro, "Torsional motion of a pair of spur gears," *Proceedings of the Institute of Mechanical Engineers,* 1963.

[16] R. Munro, "The D. C. Component of Gear Transmission Error," *Proceedings of the International Power Transmission Gearing Conference,* pp. 467-470, 1989.

[17] R. G. Munro, "The Effect of Geometrical Erros on the Transmission of Motion Between Gears," *Institute of Mechanical Engineers conference in Gearing,* pp. 79-90, 1970.

[18] C. Harris, Handbook of Acoustical Measurements and Noise Control, McGraw-Hill Inc., 1991.

[19] A. F. d. Rincon, F. Viadero, M. Iglesias, P. García, A. de-Juan and R. Sancibrian, "A model for the study of meshing stiffness in spur gear transmission," *Mechanism and Machine Theory,* vol. 61, pp. 30-58, 2013.

[20] T. Lin and Z. He, "Analytical Method for coupled transmission error of helical gear sysstem with machining error, assembly erros and tooth modification," *Mechanical Systems and Signal procesing,* vol. 91, pp. 167-182, 2017.

[21] F. d'Addato, Dynamic Simulation of power transmissions, Torino: Master thesis, Politecnico di Torino, 2016.

[22] MSC Software, Adams Machinery Gear Advanced 3D Contact Solution Brief, MSC Software, 2017.

[23] MSC Software, "Adams 2017.2 Online Help," 2017.

[24] H. Kohler, A. Pratt and A. Thompson, "Dynamics and noise of parallel axis gearing," *Institute of Mechanical Engineers Conference in Gearing,* pp. 111-121, 1970.

[25] R. Tharmakulasingam, G. Alfano and M. Atherton, "Static and dynamic transmission error of spur gear pair," *Journal of Sound and Vibration,* 2010.

[26] P. Velex and M. Ajmi, "On the modelling of excitations in geared systems by transmission errors," *Journal of Sound and Vibration,* no. 290, pp. 882-909, 2006.

[27] F. Kayama, The Dynamics of Parallel Axis Gears in an Automotive Transmission, PhD Thesis, University of Leeds, 2005.

[28] R. Parker, "Progress and problems in gear vibration and noise," in *Second International Conference on Damping Technologies*, Stellenbosch, South Africa, 2006.

[29] R. Kumar, N. Tiwari, D. Kunwar, R. V. Lakshmi and M. Chhetri, "Transimission Error on Spur Gears," *International Journal of Advanced Engineering Research and Studies,* vol. III, pp. 122-125, 2012.

[30] P. Davoli, C. Gorla, F. rosa, F. Rossi and G. Boni, "Transmission Eror and Noise Emission of Spur Gears," *Gear Technology,* 2007.

[31] M. Henriksson, On noise generation and dynamic transmission error of gears, Stockholm: Universitetsservice, 2009.

[32] E. Hiroaki and S. Nader, Gearbox Simulation Models with Gear and Bearing Faults, Mechanical Engineering, Dr. Murat Gokcek (Ed.), ISBN: 978-953-51-0505-3, InTech, Available from:, 2012.

[33] D. Lee and K. M. W. Lee, "Characteristics of Transmission Error and Vibration of broken tooth contact," *Journal of Mechanical Science and Technology,* vol. 12, no. 30, pp. 5547-5553, 2016.

[34] O. Bauchau, Flexible Multibody Dynamics, Springer, 2011.

[35] F. Amirouche, Fundamentals of Multibyody Dynamics. Theory and Applications, Brikhäuser, 2006.

[36] K. Arczewski, W. Blajer, J. Fraczek and M. Wojtyra, "Multibody Dynamics," *Computational Methods*

and Applications, vol. 23, pp. 195-215, 2001.

[37] M. Akerblom, Gear Noise and Vibrations - A Literature Survey, Eskilstuna, Sweden, 2016.

[38] J. Dominguez Abascal, Teoría de Máquinas y Mecanismos, Sevilla: Universidad de Sevilla, 2016.