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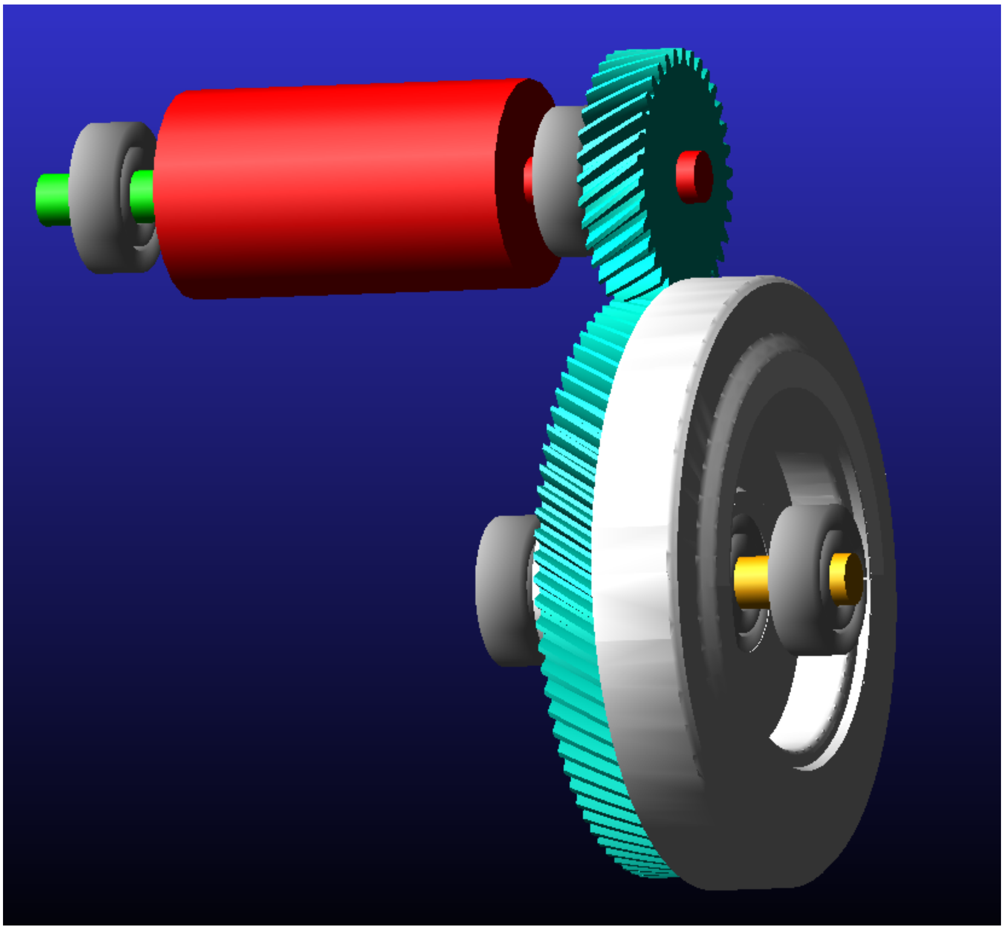

Connection of Electric Machine to a P2-X Transmission

Master's thesis in Automotive Engineering

DENNIS NORMAN

Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2019

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Examiner: Sven B Andersson, Mechanics and Maritime Sciences, Chalmers
Supervisor: Lucien Koopmans, Mechanics and Maritime Sciences, Chalmers &
Shabbir Adil, Transmission System and Simulations, CEVT

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Department of Mechanics and Maritime Sciences
Division of Combustion and Propulsion Systems
Chalmers University of Technology
SE-412 96 Gothenburg
Telephone +46 31 772 1000

Cover: The gear drive model in MSC Adams, visualizing the rotor axle, the rotor, bearings, the gears, the clutch and the clutch axle.

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Abstract

This thesis aims at investigating two different mechanical connections by simulating the performance regarding bearing force vibrations. The chosen mechanical connections are a direct, helical, involute gear drive and a silent chain. The methodology can be explained in four steps. First, mechanical transmission systems are studied in order to choose the two mechanical connections to model and simulate. Second, two models were built in the software MSC Adams. Third, the models were simulated in the same software, and simulation results were produced. Finally, the simulation results were analyzed, discussed and based on the analysis, the conclusion that gear drive is more suitable based on the defined criterion's' compared to chain drive. However, the results can be questioned by the fact that many assumptions were made regarding the chain drive model. Additional, more granular models and simulations of the chain drive are needed in order to be fully certain of the conclusion.

Keywords: Transmission, DHT, Dedicated Hybrid Transmission, Mechanical Connection, P2-X, Gearbox, Machine Element, Simulation, Engineering

Preface

For almost all my life I have had a great fascination with technical solutions and how they have improved people's everyday lives. From an early age I was determined to become an engineer; even before I even knew what the word engineer really meant.

Today, because of this fascination, I have a great interest in vehicles and technology. It all started when I began my studies at Mechanical Engineering at Chalmers. I chose this field of study because of my interest in problem solving and applied math. The step towards Automotive Engineering felt natural because of its complexity and the depth of advanced mechanical engineering, which this thesis work also touches upon.

Acknowledgements

First of all, I wish to express my sincere gratitude to Alejandro Martinez, Vice President for Mechanical Propulsion System at CEVT and Shabbir Adil, Simulation Engineer at CEVT for providing me an opportunity to write my thesis work at CEVT.

The friendliest gratitude to my very supportive supervisors, both at CEVT, Shabbir Adil and at Chalmers University of Technology, Sven B Andersson as well as Lucien Koopmans. Great thanks to you all, for all the guidance, patience and support throughout this thesis work.

I would also like to point out the meaning of my supportive friends and family, who never stopped believing in me, and encouraged me to continue fulfilling my dream.

Dennis Norman, Gothenburg, May 2019

Nomenclature

Chain Terminology

P	Chain pitch
P_d	Pitch diameter
D_t	Tip diameter
op_d	Over pin diameter
G	Maximum guide groove diameter
Z_i	Number of teeth on gear number i
B	Diameter to base of working face
R_d	Root diameter
D_p	Pin diameter
a	Shaft center distance in pitches
X	Number of links in the chain
L	Chain length in pitches

Other Symbols

ω	Angular Velocity
r	Radius
v	Speed
M	Momentum
F	Force

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1
Introduction

1.1 Background and Motivation

All products in the field of automotive are being challenged, in order to constantly reach new goals toward a more environmentally friendly transportation system. One major challenge is electrification. Many systems in the powertrain are being electrified, and that creates a need for newly engineered solutions when it comes to designed components. Two examples of older, conventional hybrid system are Integrated Motor Assist (IMA) and power split technology. In IMA systems, the engine and the electric machine (e-machine) can run simultaneously all the time, such as in Honda's application of this system. Power split device is another wide-spread technology that is slightly newer than IMA. It frees the electric machine from the engine. However, some applications of this technology, such as Toyota's, require an extra motor, thus adding to its cost and weight.

Development of the next generation of electrification of the powertrain, building on the legacy of systems such as IMA and power split device, are carried out at the Powertrain Engineering department at CEVT. One new concept is a Dedicated Hybrid Transmission (DHT). These transmissions are specially developed for hybrid drives. The DHT uses at least two sources of propulsion, i.e. an internal combustion engine (ICE) and an e-machine. The DHT systems offer an integrated solution of hybrid technology that is providing reduced costs, weight and emissions, as well as increased fuel efficiency.[2] The concept idea of a DHT was first presented 2015 at the International CTI Symposium in Berlin [2]. The DHT triple clutch, integrated electric motor system created an uprising debate about the the future of hybrid powertrain technology.

The hybrid P2 technology stands for a parallel two clutch system. This technology will reduce the cost compared to conventional transmission systems, and it is thus well known for the cost effective engineering. The only things this hybrid solution requires are a single motor, a standard transmission and two clutches. The clutches enable the gas engine and the e-machine to operate separately, and due to this simplicity, and due to the use of few parts, this solution is able to be a cost effective one. Other advantages except its low cost, are that the P2 technology gives better regenerative braking. In other hybrid solutions, a torque converter is needed. However, this is not required when using a hybrid P2, or P3 and P4, because the e-machine and engine disconnects through a clutch.[12] The total vehicle load is reduced with P2 technology, because it has less components compared to other hybrid solutions.[11]

The difference between the different hybrid architectures P0-P4 is where the e-machine is connected to the powertrain. In the P2 system, as well as the P3 and P4 system, the clutch enables the system to transfer full load on the e-machine and can therefore shut down the engine. This make the systems, P2, P3 and P4 more efficient compare to P0 and P1 where mechanical disconnection of the e-machine from the engine is not possible. Figure 1.1 displays where the e-machine is connected to the powertrain in the different hybrid architectures. In P0, the e-machine is connected to the engine. In P1, the connection is established before the clutch. In P2, which this thesis is about, the connection is on the input to the transmission. In P3, the e-machine is connected to the transmission. In the hybrid architecture P4, the e-machine is connected to the output from the transmission, before the differential.[12]

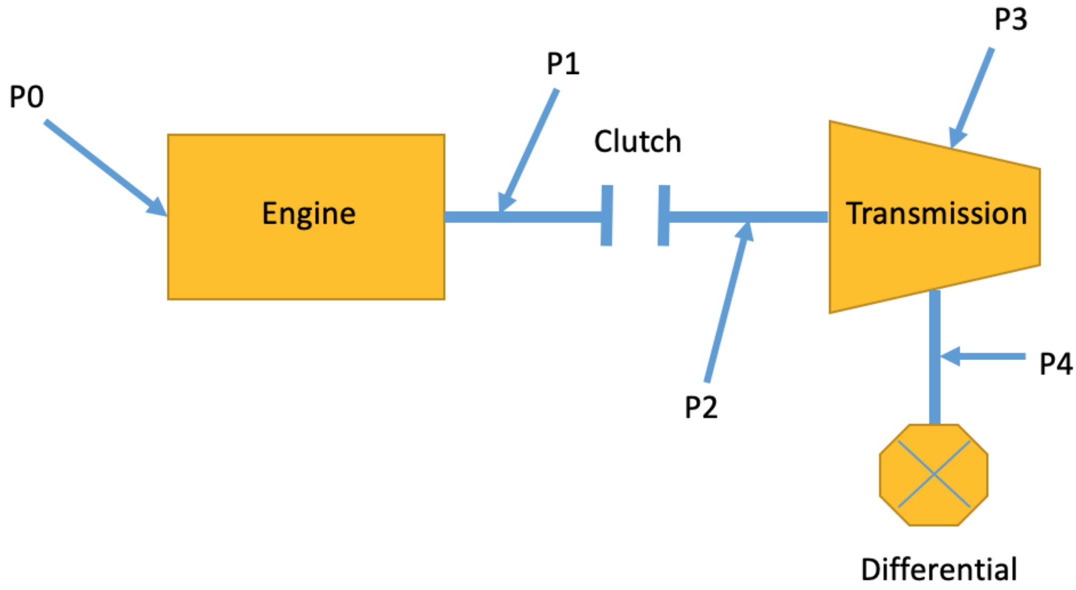

Figure 1.1: Schematic display of hybrid architecture, P0-P4.

The motivation for the thesis is that a DHT with P2-X technology is a new type of transmission, where the research about the effectiveness of different mechanical connections has great potential to improve the powertrain. However, the exact mechanics of these connections has not yet been fully researched, hence the need for this thesis work.

1.2 Project Description and Outline

The thesis work treats what type of connection to be used between the output axle of the e-machine and the input axle of the transmission. As alluded to above, the definition of a P2-X transmission is an off-axis mounted e-machine to the input of the transmission, where P2 stands for e-machine connection to input shaft and X that it is mounted off-axis. Examples of mechanical connections that can be used are belt, chain and gears.[12]

First, a theory section about the different possible mechanical connections are presented. This leads to a decision that gear drive and chain drive will be chosen for further investigation by modelling and simulation. Then, the models of the chosen drives are described, followed by an explanation of the project methodology. Finally, the simulation results are presented followed by a discussion and conclusion based on the simulation results.

1.3 Objectives

The objective of this work is to compare the vibration forces for direct gear connection with chain drive connection between e-machine and transmission. The comparison is based on simulations of the force vibrations while in operation, using the software MSC Adams.

1.4 Limitations

This thesis does not include the physical construction of prototypes, and no specification of the e-machine or the transmission is treated. Specification refers to e.g. torque requirement, required power factor, etc. Two different connection types have been simulated. The connection types are restricted to be mechanical connections, and more specifically positive transmission. There is a fixed ratio between the e-machine and the transmission. Limitations in MSC Adams makes it possible only to look at bearing vibrations in one plane. Axial forces and axial force vibrations are neglected in this thesis work.

Furthermore, only the software program MSC Adams was used for the simulations. Not more than one test script was developed and tested. The requirements for the optimal choice are limited to force vibrations in the bearings. The connection type with best performance regarding bearing force vibration is the optimal choice. The material of the mechanical connections will be assumed and specified in the simulation software.

The thesis work does not include and aspect of the assembly of the transmission. For example, the complexity of the assembly in the production line, where the transmissions are being built.

2

Mechanical Transmission Systems

This chapter describes a mechanical transmission system, and explains and compares different kinds of mechanical transmission systems. In the end, the reasons for using direct, helical, involute gear drive and silent chain drive for further investigation and model building are presented.

2.1 Description of Mechanical Transmission Systems

Mechanical transmission systems use normal forces or friction forces between mechanical components. If normal forces create the movement, then it is called a positive transmission. Conversely, it is called a non-positive transmission if friction forces create movement. Typical positive transmissions are gear and chain drive, and the characteristics of a positive transmission is that there are no speed losses and slip does not occur [6]. These requirements are necessary for the connection types treated in this thesis. Because of this, only positive transmission types will be investigated.

The concept DHT might be a relatively new type of transmission. However, in the field of power transmissions, parallel shaft involute gears is the dominant drive. Other types of typical parallel shaft drives are chains and toothed belts.[9]

In figure 2.1, the basics of gear drive mechanics are displayed. The figure is a demonstration of gear rotational direction, as well as correlation between angular velocity and velocity. If the driver gear is rotating clockwise, then the driven gear rotates counter-clockwise. However, momentum is either clockwise or counter-clockwise for both gears while in operation. Where the individual gears intertwine the velocity has to be the same. Angular velocity is dependent on the gear radius.

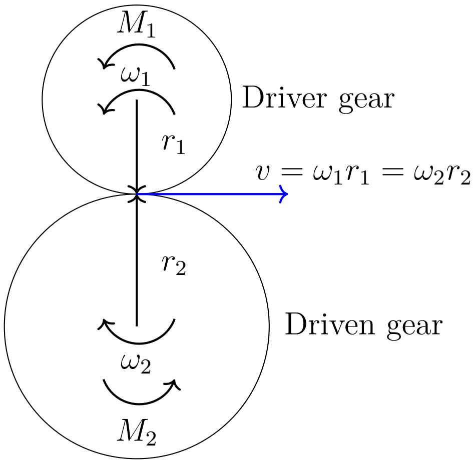

Figure 2.1: Schematic display of gears, visualizing angular velocity, gear radius, velocity and how it is correlated for gears, as well as direction of momentum.

The notations displayed in figure 2.1 are described below,

$$\begin{aligned}
\omega_{1\&2} &= \text{Angular velocity of respective gear} \\
r_{1\&2} &= \text{Radius of respective gear} \\
M_{1\&2} &= \text{Momentum of respective gear} \\
v &= \text{Velocity}
\end{aligned}$$

In figure 2.2 the basics of chain drive is displayed. It is a free body diagram where the chain forces and momentum are shown. Both sprockets have the same rotational direction while in operation. However, if the driver sprocket momentum as well as the rotational direction is clockwise then the momentum on the driven sprocket is counter-clockwise. The velocity of the chain is equal to the radius of either sprocket times its angular velocity.

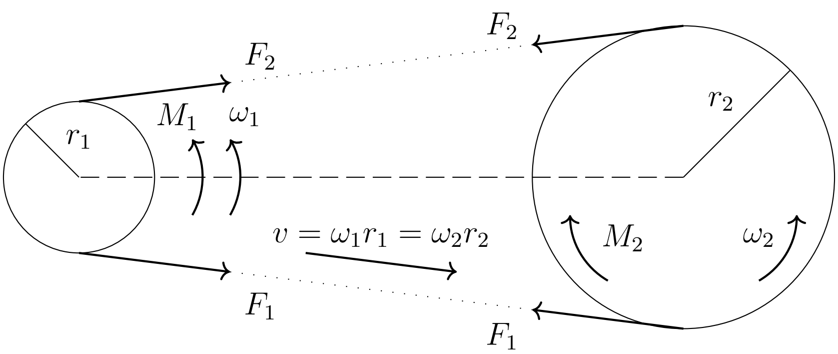

Figure 2.2: Schematic display of chain drive, visualizing angular velocity of the sprockets, sprocket radius, chain velocity and how these three parameters are correlated. The sprocket momentum as well as chain forces are also presented in the figure.

The notations displayed in figure 2.2 are described below,

$M_{1\&2}$ = Torque of respective sprocket
$\omega_{1\&2}$ = Angular velocity of respective sprocket
$r_{1\&2}$ = Radius of respective sprocket
v = Velocity
$F_{1\&2}$ = Chain force

2.2 Gears

This chapter provides a general description of gears, its geometry and the most important elements. Different types of gears will be presented and compared with each other, as well as the meaning and influence of the transmission error.

The definition of a gear is taken from Gear Metrology by Scoles, C. A. and Kirk, R. and is "any toothed member designed to transmit motion to another toothed member by means of successively engaging teeth"[8]. An example of a simple gear drive consists of two gears, where the gear wheels' teeth meshes with each other. One of the gear is a driver wheel and the other is a driven wheel. The gear with the smaller number of teeth is called pinion, and the gear with the larger number of teeth is called wheel.[4]

The gear teeth are designed to ensure that a uniform angular rotation is maintained during tooth engagement. A constant speed ratio between the driver and the driven gear is also maintained, because the gear teeth prevent slippage from occurring. The speed ratio i, also called gear ratio, is defined with equation 2.1.[1]

$$i = \frac{\omega_1}{\omega_2} = \frac{r_2}{r_1} = \text{const} \tag{2.1}$$

The difference between input and output torque is constant, because the speed ratio is constant. The torque equation 2.2 can be obtained, assuming no losses.[1]

$$\frac{M_1}{r_1} = \frac{M_2}{r_2} \Rightarrow \frac{r_2}{r_1} = \frac{M_2}{M_1} = i \tag{2.2}$$

2.2.1 Spur Gears

The definition of spur gears are cylindrical gears that have straight gear teeth cut, parallel to the axis of the gear. Spur gears are used to connect parallel shafts and the movement makes the connected gears rotate in opposite directions. Spur gears are often used to transmit a large amount of power. On the other hand, the gear teeth get into full contact with each other when meshing, and that creates a significant amount of noise and stress on the gear teeth. Therefore, spur gears are disadvantaged for high speed transmission and when quiet operation is required.[10]

2.2.2 Helical Gears

The purpose of helical gears are the same as the purpose of spur gears: to connect parallel shafts. However, during operation the helical gears can handle higher rotational speed and are also quieter compared to spur gears. The reason for this is the main point that separates the gears from each other. For spur gears the teeth are cut so that the teeth are parallel to the axis of the gear. Helical gear teeth, however, are cut at an angle to the axis of the gear. Consequently, the teeth engage more gradually for helical gears and also create axial forces. The reason for the gradual engagement is because of the inclination of the contact path, meaning that contact starts at one end of the tooth and when the gear rotates the contact gradually spreads. The load is thereby gradually distributed and helical gears operate more smoothly and quietly compared to spur gears.[10]

2.2.3 Involute Gears

Involute gears have a so called involute gear tooth profile. The meaning of involute is that the tooth profile is design in such a way that when the gear teeth meshes the direction and position of the normal force is always on the so called pressure line.[9] The pressure line, which can be seen as the red line in figure 2.3, is a theoretical string that is tangent to both the driver gear and driven gear's base circle, which is defined as a circle from which involute tooth profiles are derived. Except from the obvious fact that involute gears have to be involute, there are two more relevant criteria. First, that the contact ratio has to be greater than 1.0, and second, that the gears must have the same base pitch.[8]

The meaning of contact ratio is the average number of teeth that are in contact with each other when the gears are rotating. A contact ratio greater than 1.0 means that before one pair of teeth finish their contact the next pair has to start contact with each other.[5] On a gear, the base pitch is the distance from one tooth's flank to the adjacent tooth's flank, measured around the base circle.[7]

In figure 2.3 a gear drive is displayed and the pressure line is visible in red. The gear at the top of the figure is the driver gear and it is rotating counter-clockwise. The gear at the bottom of the figure is the driven gear and it is rotating clockwise.

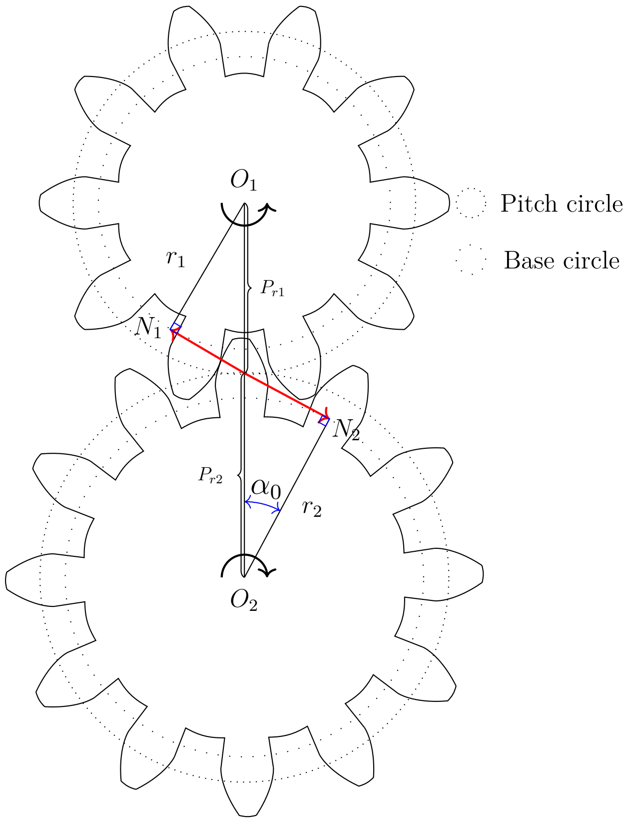

Figure 2.3: Gear drive where the pressure line is marked in red. The figure demonstrates contact ratio, pressure line gear teeth meshing, gear rotational direction, base and pitch circle of involute gears, as well as the angle with which the pressure line is tilting.

Where,

$O_{1\&2}$ = Rotation of gear
$P_{r1\&2}$ = Pitch radius
$N_{1\&2}$ = Point where the pressure line is tangent to the respective gear's base circle
$r_{1\&2}$ = Radius from gear centre to base circle
α_0 = The angle with which the pressure line is tilting

2.2.4 Transmission Error

In theory, it would be possible to build a vibration free power transmission. There would be no vibration created by the gear teeth meshing if gears were perfectly involute, absolutely rigid and accurately positioned. However, it is not possible to build a vibration-free transmission because of the so called transmission error (T.E.).[9] By a single flank engagement, it can be said that the transmission error is the driven gears difference between the geometrically ideal rotation and the actual rotation. If the driver gear was rotating at an absolute steady angular velocity, the driven gear would too, if the T.E. did not exist. There will be a difference, or error, between the theoretical exact position and the real position of a gear drive in operation, and this error is the transmission error.[3]

2.3 Chains

This chapter provides a general description of a chain, and presents the two common types of chain drive.

Chain drive consists of an endless chain and sprockets. The chain's links mesh with the sprockets. The sprocket can be described as toothed wheels. One sprocket is the driver sprocket and the other is the driven sprocket. The sprockets are connected to individual shafts.[4]

The superior characteristics of chain drives are:
1. High efficiency over the drive's whole life time
2. There is a constant speed ratio between the driver and driven sprocket
3. The load of the shaft bearings are relatively low
4. Better regarding shock absorption than gear drive
5. Long lift time expectancy
6. Large speed ratios are possible [4]

2.3.1 Roller Chain

Roller Chains use cylindrical pins for contact with the sprocket. When the roller chain is engaged with the sprocket it has a freedom of joint action. This means that between the rollers and the sprocket teeth there is no rubbing action.[4]

2.3.2 Silent Chain

Silent chains use involute teeth profiles for contact with the sprocket. The joint action is similar between roller chain and silent chain. The chain links engage simultaneously on both sides of the sprocket teeth, when the silent chain is wrapped around the sprocket.[4]

2.4 Belt

In this section, belts are described in general terms, and positive transmission and non-positive transmission with belt drive are also covered.

In a belt drive, two pulleys, or wheels, are connected by an endless flexible belt. As mentioned in Chapter 2, power transmission can be either positive or non-positive. Most of the belt drives depend on friction and are therefore a non-positive transmission type. However, there is a type of belt drive, called trapezoidal toothed belt drive, where the belt has teeth on the inside of the belt that engage with the pulleys' teeth. Consequently, this belt drive is similar to chain drive.[4]

2.5 Comparison of Mechanical Power Drives

When a connection is needed between rotating shafts that cannot be directly coupled, chain drive, gear drive, or belt drive (among others) can be used. In this chapter, these three different types will be compared to each other and advantages and disadvantages will be covered for all three types.

2.5.1 Chain Drive Compared with Gear Drive

In this chapter a comparison between chain drive and gear drive will be presented, by first presenting the advantages of chain drive followed by advantages of gear drive.

2.5.1.1 Advantages of Chain drive

For chain drive, the distance between the center points of the two shafts can be much longer compared to gear drive. This is because for gear drive the distance between the shafts' center points must be in such a way that the pitch surfaces of the gears are tangent to each other. If the distance between the shafts' center points is relatively large, the usage of chain drive can result in a simpler, less costly and more practical design. Conversely, for gear drive, this would require enlarging the two wheels, rendering an impractical size.[4]

The installation of chains is easier than the installation of gears, because chain drives can have less strict tolerances in the mechanics compared to those for gears. The ease in installing chain drives is not taken in to account in this report. However, it is worth mentioning that it is a great advantage, both from a production point of view but also from a maintenance point of view.[4]

Compared to a gear drive where one or two gear teeth are in contact with each other, chain drive provides better shock absorption. This is because chain drive offers a larger cushioning effect of the lubricant in the chain joints, and because there is normal elasticity in tension in a chain.[4]

Another difference between the two gear types is the types of actions when the gear is engaged; when helical gears are driven there are two types of actions ongoing: a sliding and a rolling one between the surfaces of the gear teeth. Chain drive on the other hand, uses only a rolling action when the rollers engage the sprocket teeth.[4]

When a chain drive is used, the load is more distributed compared to when gear drive is used. This is because for chain drive, the load is distributed over multiple sprocket teeth simultaneously, and in the case of gear drive only one up to a couple of teeth are meshing with each other simultaneously, thus creating a smaller area of contact. [4]

2.5.1.2 Advantages of Gear drive

When there are space limitations to consider, gears have an advantage. Because the pitch surfaces of gears have to be tangent, the shortest possible distance between shaft centers is required, which in turn makes the gearing option more compact. This compactness is of course an advantage when space in the application of the drive is a limiting factor. One such case is when gears are used in a transmission, which is the application this thesis is concerned with.

In addition, gears can operate at a higher rotational speed and can usually have a higher maximum speed ratio than chain drives can handle. Furthermore, when both extremely high speed and high horsepower are the operation requirements, gears are the preferable option.[4]

2.5.2 Chain Drive Compared with Toothed Belt Drive

In this chapter a comparison between chain drive and toothed belt drive (positive gear) will be presented, by first noting the advantages of chain drive followed by the advantages of toothed belt drive.

2.5.2.1 Advantages of Chain Drive

When comparing chain drive and belt drive, the chain drive will occupy less space than the belt drive. This is because the chain will be narrower and the sprockets will be smaller compared to belt and pulleys.[4]

Furthermore, it is relatively easier to install a chain. The chain can be wrapped around the sprockets and then mounted together. The belt, on the other hand, is often already laced, which means it is more difficult to get the belt over the pulleys than it is to get the chain over the sprockets. Additionally, when operating in a dusty atmosphere or environment, belt drives have an inherent fire hazard because of generation of static electricity. Conversely, the fire hazard is eliminated if chain drive is used. Furthermore, chains can operate at higher temperatures and chains are not affected by sun, oil or grease in the same manner that belts are. Also, belts can become progressively worse with age, but this does not happen to chain to the same extent. [4]

2.5.2.2 Advantages of Toothed Belt Drive

Lubrication is not required when using belt because there is no contact between metal and metal between belt and pulley. However, the belt might need periodic application of belt dressing in order to make sure that the belt's flexibility is maintained, especially when the belt ages. Belt drive is usually less noisy during operation compared to chain drive operation.[4]

2.6 Mechanical Transmission Systems Selection

The comparison between different mechanical transmission systems is behind the decision-making of what mechanical connections were used for further investigation. For this thesis project the three most relevant criteria for the mechanical connections are:

1. The bearing force vibration is as low as possible
2. It can handle high speed and torque without slippage
3. The space required is reasonable for use with a transmission

The first criterion is the most important, since it is the objective of the thesis. The second and third criteria are necessary for the application of the connection. I.e. any gear type that incurs slippage at high speed and torque is unsuitable for use with a vehicle transmission, and any gear that takes up too much space is similarly unsuitable to place in a vehicle. Because of this, positive connection types are absolutely necessary, which rules out belt drive. Conversely, silent chain and involute helical direct gear drive are chosen for further investigation. Helical gears run smoother and more quietly than spur gear, and is more suited for high speed. The silent chain drive was chosen because it takes up less space than a toothed belt drive as well as, because of the aging aspect. The models for these chosen connections are described in the next chapter.

In summary, the comparison between the two selected types of gears can be observed in table 2.1 for chain drive, and table 2.2 for gear drive.

Table 2.1: Summary of chain drive advantages and disadvantages

Connection	Pros	Cons
Chain drive	More distributed load Better shock absorption Large center distance possible	Takes up more space Worse for high speed

Table 2.2: Summary of gear drive advantages and disadvantages

Connection	Pros	Cons
Gear drive	Takes up limited space High rotational speed possible Better for high horsepower	Less cushioning effect

3
Models

This chapter describes the two models that have been built in the software MSC Adams. One model is with gear and the other model is with chain. Both models have the same common structure, which will be described first. The differences between the models is the connection between the clutch and the electric motor axle as well as the center distance. The center distance is the distance between the center of the clutch and the center of the electric motor axle.

3.1 Common Model Structure

The two different axles of the transmission model are the axle where the electric motor is attached and the axle connected to the ICE and the gear train, called clutch axle.

3.1.1 Electric Motor Axle

This part of the model consists of two bearings, one rotor and two rigid axle segments connecting the rotor with fixed joints in each end of the rotor. Only the rotor part of the electric motor is modeled, this with a rigid cylinder. The bearings are modeled with help of MSC Adams Machinery. Both bearings are connected to ground on the outer ring and to each axle segment on the inner ring. Connections are established by fixed joints. A torque is applied around the rotor's center of mass in order to create the rotation of the axle.

3.1.2 Clutch Axle

The clutch axle consists of four bearings, one clutch shell and one rigid axle, referred to as center axle. The center axle is supported by two bearings, where the inner ring is connected to the axle and the outer ring is connected to ground with fixed joints.

The clutch shell model was provided by CEVT and no inner parts are modeled. Instead, the connection between the center axle and the clutch is made by a fixed joint in the clutch center of mass. Further, the clutch is supported by two bearings. One four point contact ball bearing on the ICE side which connects to the center axle on the inner ring and the clutch on the outer ring. The other bearing, a needle roller bearing, connects the clutch on the inner ring and the ground on the outer ring. This bearing is located on the gear-train side of the clutch.

3.2 Gear Model

In the gear model, as can be seen in figure 3.1, the drive gear is connected to the electric motor axle and the driven gear is connected to the center axle. The connections are established by fixed joints. The gear tooth profile, gear design and tooth contact patch between the gears are made by MSC Adams Machinery built-in functionality for gears.

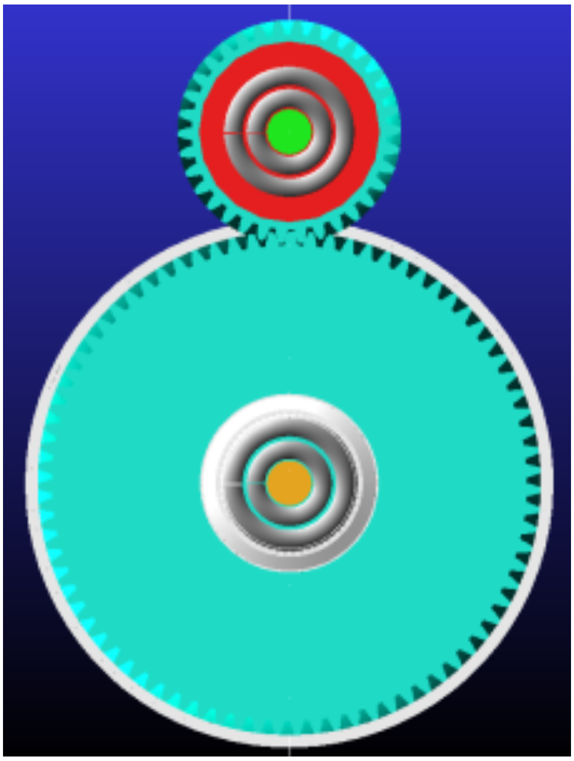

Figure 3.1: Gear model from Adams View, as seen from the transmission.

The gear design parameter values were predefined by CEVT, and are displayed in the table A.1. The exact values are a company secret. No other values than the ones provided by CEVT were investigated during this project. The given parameters were:

- Normal module
- Number of teeth
- Pressure angle
- Helix angle
- Hand of helix
- Profile shift coefficient
- Face width
- Rim diameter

3.3 Chain Model

The chain model, as can be seen in figure 3.2 has the driver sprocket connected to the electric motor axle and the driven sprocket connected to the center axle. The chain is wrapped around sprockets. Chain and sprockets are built with help of the MSC Adams Machinery chain module, and an additional script was implemented to facilitate modification of the chain tooth profile and sprocket tooth design. The extra code used can be found in Appendix D. There is an important and major limitation with chain drive models in Adams Machinery. The chain links are only possible to model in 2D.

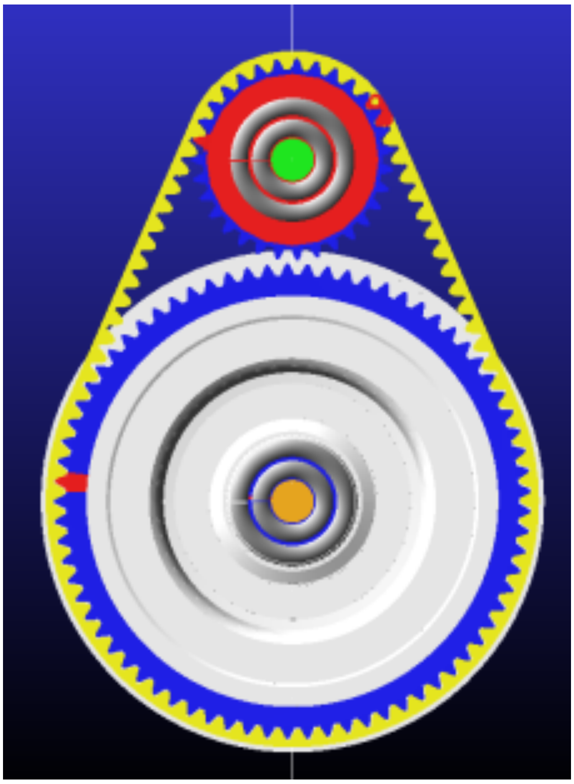

Figure 3.2: Chain model from Adams View, as seen from the transmission.

Given parameters by CEVT were:

- Number of sprocket teeth
- Chain pitch
- Number of links in the chain
- Pretension of the chain

The values of the given chain parameters can be found in table A.2.

3.3.1 Sprocket Equations

All the equations in this section are from Jackson and Moreland's work on design manual for roller and silent chain drives [4]. The MATLAB-script created to calculate the sprocket design is found in Appendix C.

Outside diameter for square shaped teeth, also called tip diameter for sprocket number i:

$$D_t = P\left(\cot\frac{\pi}{Z_i}\right) \tag{3.1}$$

where,
P = chain pitch in inches
Z_i = Number of teeth for sprocket number i Pitch diameter for sprocket number i:

$$P_d = \frac{P}{\sin\frac{\pi}{Z_i}} \tag{3.2}$$

Over pin diameter for odd number of teeth:

$$op_d = \cos\left(\frac{\pi/2}{Z_i}\right)\left[P_d - 0.125P\csc\left(\frac{\pi}{6} - \frac{\pi}{Z_i}\right)\right] + 0.625P \tag{3.3}$$

The root diameter is between the maximum guide groove diameter and the diameter to base of working face.

Maximum guide groove diameter:

$$G = P\left(\cot\left(\frac{\pi}{Z_i}\right) - 1.16\right) \tag{3.4}$$

Diameter to base of working face:

$$B = P\sqrt{1.515 + \left(\cot\left(\frac{\pi}{Z_i}\right) - 1.1\right)} \tag{3.5}$$

Root diameter is:

$$R_d = \frac{G+B}{2} \tag{3.6}$$

Pin diameter is:

$$D_p = 0.625P \tag{3.7}$$

3.3.2 Chain calculations

Center distance, a ,is specified by the equation:

$$a = \frac{p}{4}\left(X - \frac{z_2 + z_1}{2} + \sqrt{X - \left(\frac{z_2 + z_1}{2}\right)^2 - 8\left(\frac{z_2 - z_1}{2\pi}\right)^2}\right) \tag{3.8}$$

where,
X = number of links in the chain
p = chain pitch in inches

The length of the chain is calculated with the equation:

$$L = 2a + \frac{z_2 + z_1}{2} + \frac{\left(z_2 - z_1\right)^2}{4\pi^2 a} \tag{3.9}$$

where,
L = chain length in pitches
a = shaft center distance in pitches
z_1 = number of teeth in small sprocket
z_2 = number of teeth in large sprocket

Correction of center distance

$$a = \frac{L - \frac{z_2 + z_1}{2} + \sqrt{\left(L - \frac{z_2 + z_1}{2}\right)^2 - 8\frac{\left(z_2 - z_1\right)^2}{4\pi^2}}}{4} \tag{3.10}$$

4 Methodology

This chapter guides the reader through the methodology that has been used during the thesis project. The chapter is also meant to explain how the research, data collection, literature study etc. is carried out.

4.1 General Methodology

The literature review covered the kinds of machine elements or so called connection types that are possible to use between the e-machine and the transmission. The literature review is carried out by gather information from books, papers and articles, as well as from the Internet. Furthermore, the Bachelor course in Machine Elements at Chalmers University of Technology was referenced, as well as the course in CAD and MATLAB.

The simulation software that was used for simulations was new for the thesis worker and theory about the software itself had to be studied. During the literature study phase the thesis worker participated in a course on the software program MSC Adams View, the plugin MSC Adams Machinery and also consulted MSC Adams company experts.

After the literature review, it was decided together with the supervisor at CEVT and with supervisors at Chalmers which machine elements to investigate further. Thereafter, the software program MSC Adams View and the plugin MSC Adams Machinery was used to simulate the performance regarding bearing vibrations. The simulation results were the main arguments for the proposal of what connection type to be chosen as the optimal type.

The simulation preparation started at the same time the literature study was going on, because the study also involved getting familiar with the simulation software. This enabled the thesis worker to obtain the right qualifications to plan and perform the simulations in an analytic manner. Simulation, simulation investigation, and report writing was the major block in this thesis work.

Data was collected in different ways. The data collection in the literature study phase comes from, as previously mentioned, the study of academic papers, books, articles, publications etc. However, the main data collection comes from the simulation phase, where the thesis worker was carrying out the investigation by simulating two different cases. The data collection comes from results from simulations using the software program MSC Adams View.

In the planning report, it was specified what the conditions for the optimal choice of connection type were. Setting these up early enabled an overview of how the project was progressing towards its objective.

4.2 Simulation Description

In this part, the software used will be presented in greater detail, and the test script as well as the important time delay are presented. The chapter also covers how the simulations were performed.

4.2.1 MSC Adams Software

Adams was used because it is a powerful modeling and simulation tool, where it is possible to build and analyze any mechanical system. Adams is the acronym for Automatic Dynamic Analysis of Mechanical Systems. The software uses the Euler-Lagrange method to create equations of motions and then use predictor-corrector methods to solve those equations. The models are going to be built in the application Adams view, simulated in the application Adams solver and results will be reviewed using Adams post processor application. Adams MSC documentation (as part of the software) was used extensively.

4.2.2 Test Script

The reason for a test script is so that the two models can be run by the same script and evaluated accordingly. The test script tests two different torques. First maximum torque will be applied, for a specific time stamp. The torque will gradually increase from zero to maximum torque and speed will increase. When maximum torque is reached it will be constant for a certain time stamp and then gradually be decreased, until it is kept constant again. However, speed is increased when torque is decreased. When torque is kept constant, the speed should be constant. Details about the applied torque can be found in appendix A.2. Figure 4.1 displays a graph of the torque and speed variation over time. The graph is normalized to a scale from 0 to 1.

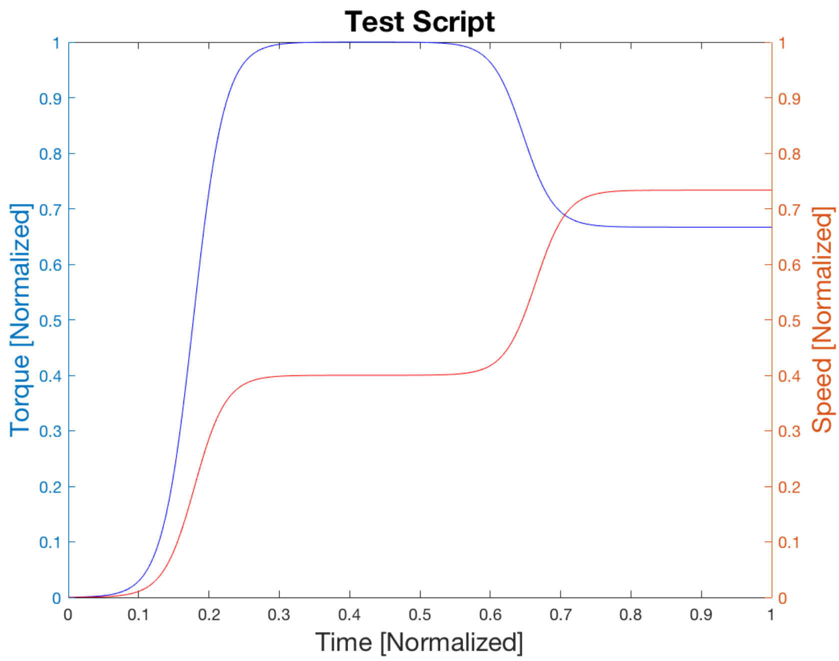

Figure 4.1: Speed and torque variation over time in the simulation test script

4.2.3 Time Delay

When torque is applied around the rotor's center of mass there has to be a reaction torque applied around the center axle. However, in order to create a rotation and to increase the rotational speed, a delay is added. Details about the time delay can be found in appendix A.2. The Δ_x in figure 4.2 is a time delay

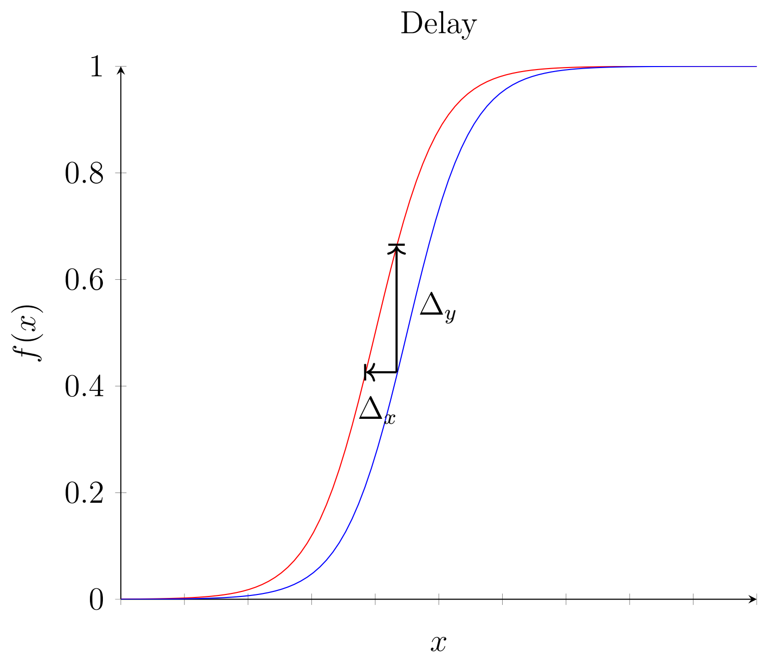

Figure 4.2: The time delay Δ_x and the difference between input torque and output torque, Δ_y, are visualized

4.2.4 Simulation Execution

The solver for the simulations was internal and the integrator HHT, with formulation I3 was used. The meaning of the integrator HHT is that Adams solver only uses C++. The meaning of the equation formulation I3 is that the solution has to satisfy all constraints. On the other hand, it does not ensure that velocities and accelerations calculated satisfy all first- and second time derivatives. The chain drive model simulation was run on MSC Adams computer, and took around 40 hours to finish. For the gear drive model, the simulation took around 20 hours and was run on CEVT computer.

5

Simulation Results

In this chapter, the simulation results are presented. All the figures display graphs where the values of the y- and x-axis have been removed, because the exact results are confidential and a company-owned secret.

Figure 5.1 displays the torque applied into the system by the test script. It can be seen in figure that applied input torque is the same for both the gear and the chain drive model, red and turquoise lines in the graph. Torque applied on rotor axle and clutch axle for both the gear drive model and the chain drive model are visible. The red line indicates the rotor axle for the gear drive model, and the blue line indicates the corresponding counter-torque. The gap between the curves is explained by the gear ratio, defined by equation 2.1. Conversely, the turquoise line indicates the rotor axle for the chain drive model and the black line is the corresponding counter-torque. The gap is explained by the gear ratio and it is positive because the axles have the same rotational direction, also explained by equation 2.1.

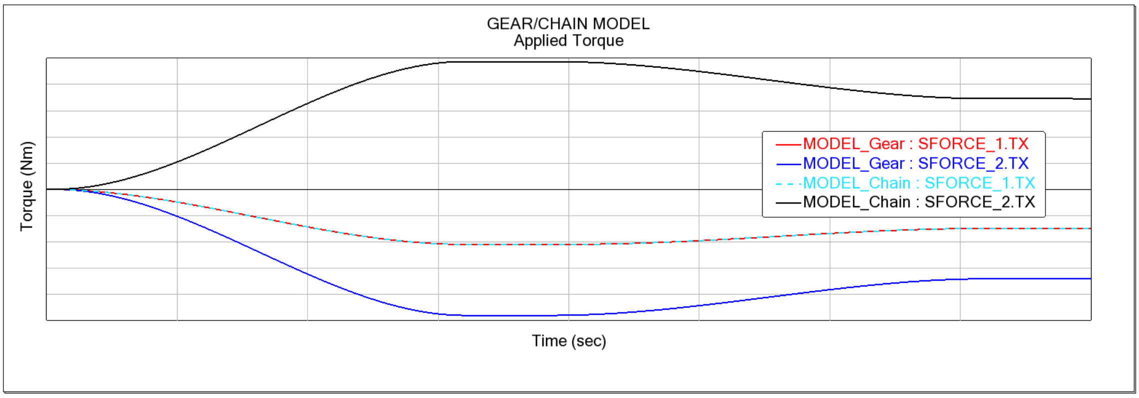

Figure 5.1: Applied torque on rotor axle (SFORCE_1.TX) and clutch axle (SFORCE_2.TX) for both the gear drive model and chain drive model

Figure 5.2 displays the angular velocity that occurs because of the applied torque. Both gear drive model (red and blue) and the chain drive model (turquoise and black) are visible in the graph. The rotation is defined around the x-axis in the model's global coordinate system. The angular velocity is zero at time zero for all curves. The red line is the only positive line. That is because it is defined that way; counter-clockwise rotational direction is positive and clockwise rotational direction is negative.

The angular velocity, figure 5.2, increases in magnitude because the torque is applied according to figure 5.1. However, angular velocity is accomplished with help from the time delay described in section 4.2.3. Without the time delay the angular velocity would remain zero throughout the whole test script, even if the torques are applied, because they would cancel each other out. Torque on the input axle is the same for both gear and clutch drive model, and the counter-torque on the clutch axle is symbolizing resistance from the powertrain. The counter-torque is the input torque multiplied with the gear ratio. The value is negative or positive depending on if the axle rotates clockwise or counter-clockwise, respectively.

At time zero, maximum torque specified by CEVT, is applied gradually and the angular velocity increases in magnitude for all cases. After approximately one third of the time, the torque is kept constant and therefore the curves of the angular velocities derivatives decreases, and moves toward zero. Thereafter, a change in torque happens again. This is the reason why the angular velocity for all cases increases again. The applied torque decreases to one third of its maximum. However, the time delay creates a torque difference between the rotor axle and clutch axle that according to Newton's second law of physics causes an angular acceleration, and therefore causes the angular velocity to increase. In the end of the time spectrum investigated, the torque is kept constant again, and the angular acceleration approaches zero.

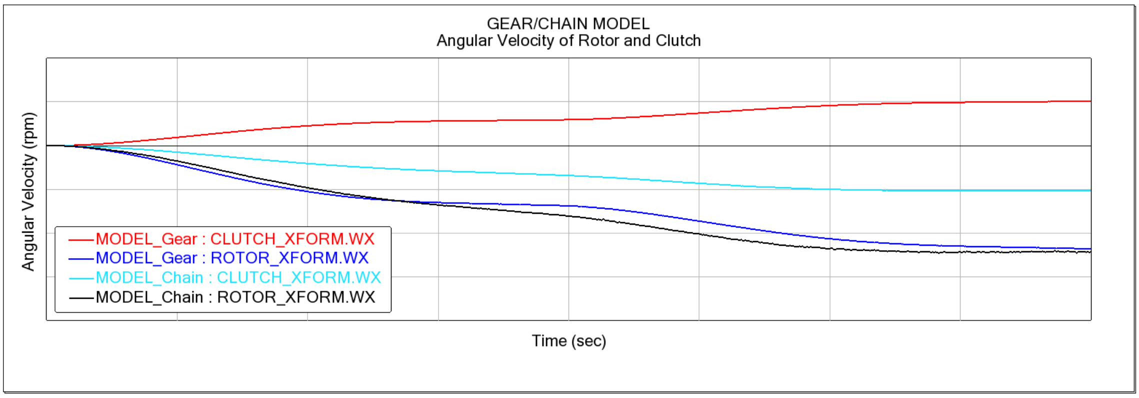

Figure 5.2: The angular velocity around x-axis for rotor and clutch in gear drive model and chain drive model

All figures 5.3, 5.4, 5.5, 5.6, 5.7 and 5.8 shows the total magnitude of the bearing forces for each bearing. All figures below have also been filtered using a Butterworth low pass filter of order 2 and scaled cutoff frequency 0.1.

The model has 6 bearings that were investigated by simulations. The first graph below shows the results of bearing number 1, the last graph presented in this chapter presents the results from bearing number 6. In between bearings with increasing number are presented.

The red curves in all figures below are the total magnitude of the bearing forces for the gear drive model. The blues curve in all figures below are the total magnitude of the bearing forces for the chain drive model. Torque is applied according to figure 5.1. It can be seen in all figures below that the bearing forces starts at zero force when time is zero. The red curve for all figures except 5.8 has the general behaviour like curves in figure 5.1. The bearing force magnitude increases until three eighths of the time has passed. After half of the time has passed, the force magnitude starts to decrease until it is kept constant again in the end of the time slot.

The blue curves do not have the same predictability as the red curves. The bearing force response to the applied torque does not in a single case follow the curve of the applied torque. In figure 5.3 and 5.7 the red and blue curves are of similar bearing force magnitude. Noteworthy regarding the chain drive model is that figure 5.6 shows that bearing number 4 has a higher magnitude compared to the bearing forces acting on the gear drive model.

However, the magnitude of the bearing force for gear drive is much larger on bearing number 3 (figure 5.5) than it is for gear drive on bearing number 4 (figure 5.6) (this cannot be directly observed in the graphs due to scaling of the y-axis). For chain drive, the load on bearing 3 and 4 (rotor axle) is similar in magnitude (figure 5.5 and figure 5.6).

For bearing 2, 3, and 6 the magnitude of the bearing forces for the chain drive model is lower compared to the magnitude of the bearing forces for the gear drive model, seen in figures 5.4, 5.5 and 5.8.

In figure 5.8 it is worth mentioning that the force magnitude is much lower compared to the other graphs displaying bearing forces because of the placement of this bearing inside the clutch. These forces are so small that they can be neglected. The results are included for completeness.

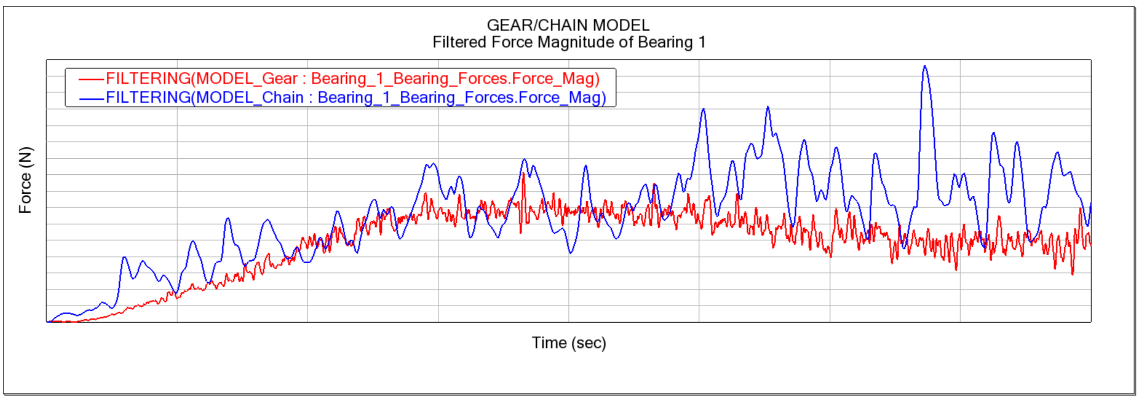

Figure 5.3: Filtered force magnitude of bearing 1 over the test cycle

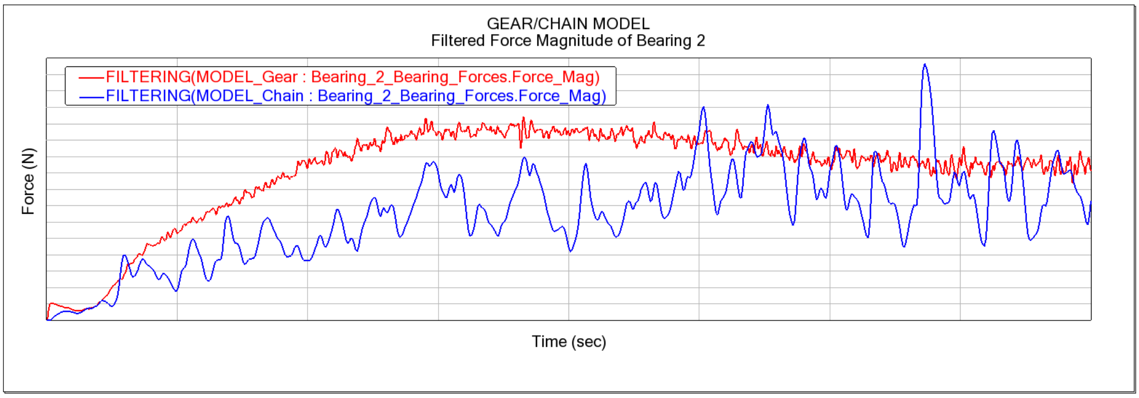

Figure 5.4: Filtered force magnitude of bearing 2 over the test cycle

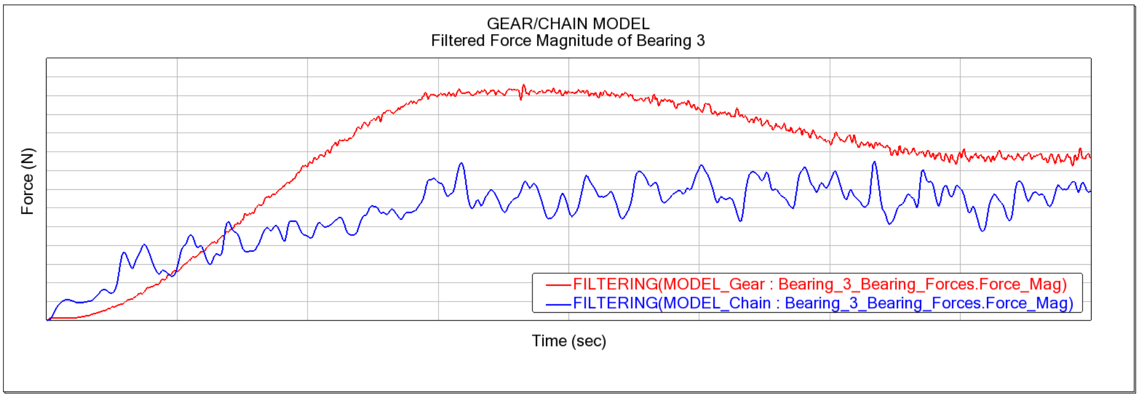

Figure 5.5: Filtered force magnitude of bearing 3 over the test cycle

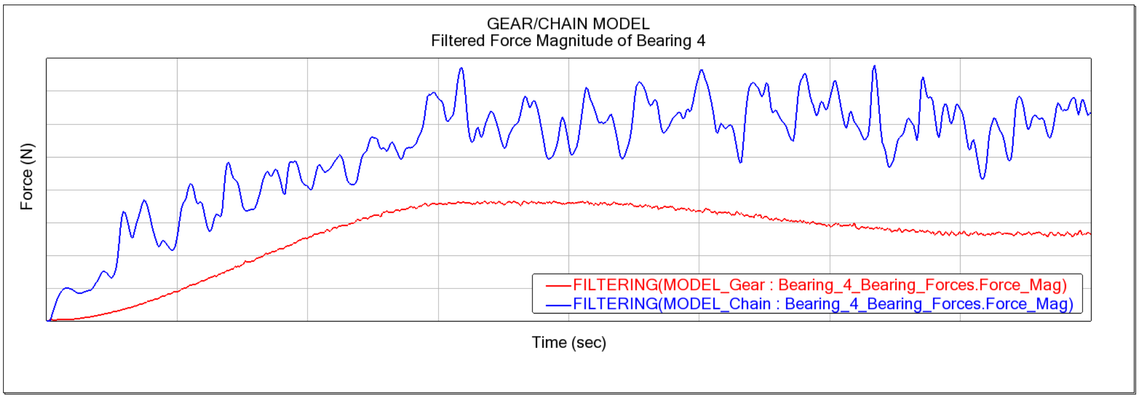

Figure 5.6: Filtered force magnitude of bearing 4 over the test cycle

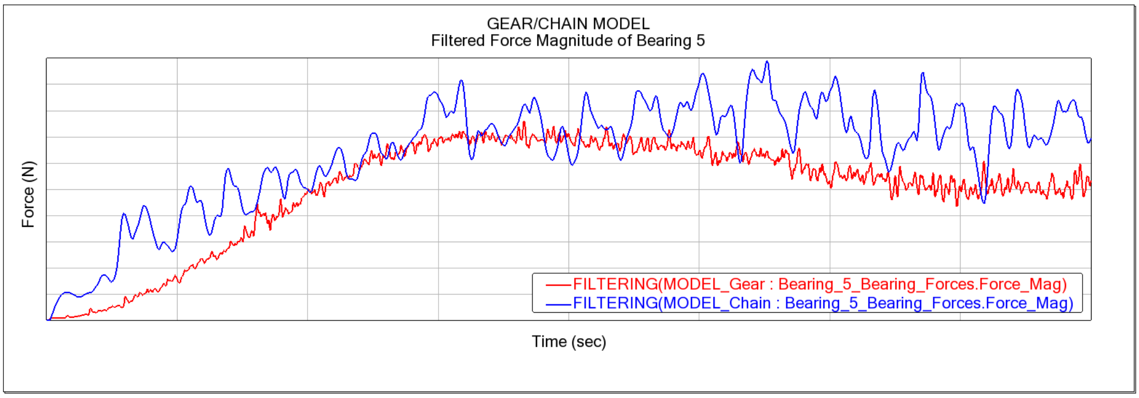

Figure 5.7: Filtered force magnitude of bearing 5 over the test cycle

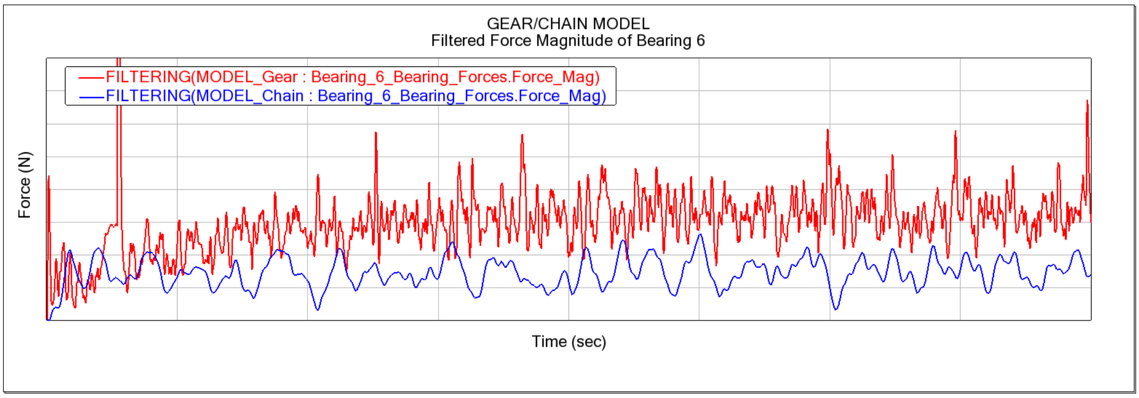

Figure 5.8: Filtered force magnitude of bearing 6 over the test cycle

6
Discussion and Conclusion

This chapter will first cover a comparison of the results in chapter 5 to the theory in chapter 2. Then, the discrepancies between the theory and the results will be discussed, and finally a conclusion will be made as to the purpose of the thesis; the best choice for connection type between e-machine and the clutch in a dedicated hybrid transmission.

6.1 Results Compared to Theory

This section will compare observed velocity, bearing forces, and vibrations to what could be expected from the corresponding theory.

6.1.1 Velocity Compared to Applied Torque

Comparing the angular velocity in figure 5.2 with the applied torque in figure 5.1, it can be seen that the gear drive model is behaving more similar to the applied torque than the chain drive model does. For the gear drive, the angular velocity stabilizes, and is almost kept constant when the applied torque is kept constant. In the chain drive model, the graph has a negative derivative until the very end of the test script. The derivative does not approach zero in the middle of the test script as it does for the gear drive model. This implies that the gear drive model has a more accurate response. In other words, it follows the torque applied to the system, according to the theory laid out in 2.1. This discrepancy between gear drive and chain drive can be explained by an insufficiently detailed model of the chain drive, to be further explored in section 6.2.2.

6.1.2 Bearing Forces

The observation that the behaviour of bearing force magnitude corresponds to the applied torque, displayed in figure 5.1, means that applied torque reaches its maximum when the bearing force is at its maximum in the gear drive model. However, this is not the case in the chain drive model. It cannot be said that the bearing force is at maximum when applied torque is at its maximum. This is most possibly explained by a deficient model, discussed in section 6.2.2.

Observing figure 5.5 and figure 5.6, chain drive has a more even distribution of load between the bearings, whereas gear drive has an uneven distribution (the magnitude of the force on bearing 3, closest to the rotor shaft, is much greater). This is in alignment with theory that chain drive should have better bearing load distribution properties (covered in section 2.5.1.1).

6.1.3 Bearing Force Vibration

It is clear that the amplitude of the vibrations are a lot larger in the chain drive model compared to the gear drive model. However, the theory supports that the chain drive has a better shock absorption compared to gear drive, as mentioned in section 2.5.1.1. This misalignment can be explained by a faulty transmission error, discussed in chapter 6.2.1 and that the model for chain drive is deficient, discussed in section 6.2.2.

6.2 Discussion

This section will cover the two most likely sources of discrepancies between expected and realized outcomes: transmission error and model deficiencies.

6.2.1 Transmission Error

From chapter 2.2.4 it is known that a power transmission cannot be absolutely perfect, an error in the teeth meshing creates vibrations. The transmission error directly affects the bearing force vibrations. The transmission error has not been measured or tested before the simulations, which means that it could be much smaller for the gear drive. In turn, this could mean that the vibrations should in fact be larger for the gear drive model, and that the large vibrations observed for the chain drive are not representative of a real chain drive system.

6.2.2 Model Deficiencies

The models have been made by specifications from CEVT and the design is described in section 3. The chain teeth profile and design of the sprocket teeth was complex to model. Assumptions had to be made and the gear drive model had to be downgraded from a more complex design in order to be comparable to the chain drive model. Specifically, these limitations were brought up in the section on limitations (1.4), as well in the section on the design of the chain model (3.3).

A more accurate model could potentially have entailed observing a constant velocity during maximum constant torque for the chain drive.

The design of the chain and the sprocket teeth can interfere with the results because they could cause the model to be misaligned with the corresponding theory. The design of the model could explain the observed variations in force for the chain drive, as a more advanced model for chain drive could have produced more accurate results where the vibrations would not have been as pronounced.

6.3 Conclusion

The three criteria used to make the decision for best connection type were: The bearing force vibration is as low as possible, it can handle high speed and torque without slippage, and the space required is reasonable for use with a transmission. The first criterion was the most important, and the second and third criteria were necessary for use of the connection with a transmission. Based on these criteria and the results of the simulation, the gear drive is the best connection type choice. However, it cannot be ignored that the model used in simulation was unable to accurately represent the chain drive, specifically with regards to vibrations. Therefore, this conclusion is limited, and must be complemented with a more detailed simulation of the chain drive before final conclusions can be made. The same logic must be applied to the fact that the transmission error was unknown, leading to potentially erroneous vibrations observed in the simulation.

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