Optimization of a passive preload actuator for bearing units.
Technical feasibility of variable preload in a pinion unit
Summary

Environmental protection plays an important role today. The automobile industry in particular is constantly criticized when it comes to pollutant emissions. Not only the automotive industry, but also the automotive suppliers are committed to deliver more energy efficient solutions. The rolling bearing industry in particular is committed to contribute towards a reduction of CO₂ emission by improving the efficiency rate of vehicle drive lines. SKF has launched a special Environmental Vehicle program (ENVI), dedicated to the development of energy efficient solutions. Part of the program is the development of a variable preload system for bearing units to reduce friction losses. The actuator for this project is developed in a previous stage at SKF Automotive Development Centre. The next stage, that is considered in this report is to investigate the influences and potential benefits of applying such a system on a bearing unit. Additionally, the system is optimized to meet the highest friction reduction feasible with acceptable bearing life and stiffness. The Hybrid Pinion Unit, developed by SKF for a rear axle differential is used as a case study.

First the influence of preload on bearing life, stiffness and friction is investigated using the calculation program SKF Bearing Beacon. This is an internally developed program used to calculate advanced bearing systems. It shows that only bearing life has a specific pre-displacement where the bearing life is optimal. Bearing stiffness has no absolute optimum, but significant stiffness improvements are feasible with only a slight amount of pre-displacement. Some stiffness is required for accurate gear mesh. The friction in the bearing increases when more pre-displacement is applied. Low friction is therefore only feasible by lowering the pre-displacement.

The reference case for comparing different solutions is the SKF Hybrid Pinion Unit. Before optimizing the variable preload, first the ultimate achievements are calculated for each individual parameter by using the ideal pre-displacement for each parameter neglecting the other parameters. The results of this theoretical study showed that the energy losses of the bearing, during the New European Driving Cycle, can be lowered with 48.7% (from 53.2 kJ to 27.3 kJ), though in practice the reduction will be less favourable. The maximum CO₂ reduction is 0.11 g/km. The system life is evaluated for a customer load cycle. This load cycle is used by a customer for the acceptance tests on the Hybrid Pinion Unit. The results showed that the system life could only be increased from 1404 hours to 1406 hours. This is within the error margin in the results and can be considered as unchanged. The stiffness is evaluated by measuring the deflection of the shaft at gear contact using the highest load condition in the customer load cycle (1331 Nm). The results showed that the deflection could be lowered with 11% in radial direction and 20% in axial direction.

The optimum curve for variable preload as function of torque is developed in two different stages. First, a theoretical optimum solution is found whereby the pre-displacement is independent of the applied torque. The second solution is derived from the first solution by curve fitting a linear polynomial. This is the more practical solution that meets the required linear relationship of the given actuator. The solutions are optimized for maximum energy loss reduction for the NEDC cycle, maximum life for the customer load cycle and minimum deflection at gear contact. The resulting energy loss reduction is 33.2% for the theoretical solution and 34.5% for the linear solution. The CO₂ reduction for both solutions is 0.08 g/km. The system life for both, the theoretical solution and linear solution, is not changed considering the error margin in the life prediction method. The deflection at gear contact for both solutions is reduced with 10.1% in radial direction and 18.8% in axial direction.

Finally, the design of the actuator is optimized with the Finite Element Program Pro/Mechanica and the nonlinear solver MSC Marc to meet the optimum curve characteristics with minimum amount of material stresses. The best design found, has a maximum Von Mises stress of 169 Mpa with 1000 Nm torque applied. The maximum Von Mises stress with 1500 Nm torque applied is 255 Mpa. The range of allowable stress is approximately 185 –to 360 MPA, based on the yield strength of construction steel. The fatigue limit however, is lower and should be investigated in more detail.

Only the influence of pre-displacement on bearing properties is considered in this research. It is recommended to investigate the influence on the total differential as well, including the gear mesh efficiency and the temperature effects.
Contents

1. INTRODUCTION ........................................................................................................... 5
   1.1 BACKGROUND ......................................................................................................... 5
   1.2 PREVIOUS RESEARCH ......................................................................................... 5
   1.3 THESIS OBJECTIVES ......................................................................................... 6
   1.4 ABOUT THIS REPORT ......................................................................................... 6

2. THE PINION UNIT AND ITS SURROUNDINGS .............................................................. 8
   2.1 INTRODUCTION ..................................................................................................... 8
   2.2 REAR AXLE DIFFERENTIAL .................................................................................. 8
   2.3 HYPOID GEARS .................................................................................................... 8
   2.3.1 Gear geometry .................................................................................................. 8
   2.3.2 Gear loads ....................................................................................................... 9
   2.3.3 Gear performance and efficiency ..................................................................... 9
   2.4 DESIGN REQUIREMENTS FOR PINION UNITS .................................................... 9
   2.4.1 Stiffness requirements ..................................................................................... 9
   2.4.2 System life requirements ............................................................................... 10
   2.5 DYNAMICS OF THE SYSTEM ............................................................................... 10
   2.5.1 Vibrations in the pinion unit .......................................................................... 10
   2.5.2 Vibrations in the driveline ............................................................................. 11
   2.6 CONCLUSIONS .................................................................................................... 11

3. EFFECTS OF BEARING PRELOAD ON THE SYSTEM PARAMETERS .......................... 12
   3.1 INTRODUCTION ..................................................................................................... 12
   3.2 BEARING TERMINOLOGY ..................................................................................... 12
   3.3 EFFECTS OF BEARING PRELOAD ON THE STIFFNESS ....................................... 12
   3.3.1 Stiffness of a single ACBB ............................................................................ 13
   3.3.2 Stiffness of a single TRB ............................................................................. 13
   3.3.3 Stiffness of a duplex TRB/ACBB .................................................................. 14
   3.4 EFFECTS OF BEARING PRELOAD ON THE LIFE ................................................ 15
   3.4.1 Effects of preload on the internal load distribution ...................................... 15
   3.4.2 Life prediction calculations .......................................................................... 16
   3.4.3 Life calculations with SKF bearing beacon .................................................. 16
   3.5 EFFECTS OF BEARING PRELOAD ON THE FRICTION LOSSES .......................... 17
   3.5.1 Case study: constant reaction force ............................................................... 18
   3.6 CONCLUSIONS .................................................................................................... 19

4. ANALYSIS OF THE PINION UNIT WITH STATIC BEARING PRELOAD ...................... 20
   4.1 INTRODUCTION ..................................................................................................... 20
   4.2 ENERGY LOSSES IN THE SYSTEM ...................................................................... 20
   4.2.1 Calculation of energy losses with VEP ............................................................ 20
   4.2.2 Power losses .................................................................................................... 20
   4.2.3 Energy losses .................................................................................................. 21
   4.3 SYSTEM LIFE ....................................................................................................... 21
   4.4 SYSTEM STIFFNESS ............................................................................................ 22
   4.4.1 Deflection at maximum load ......................................................................... 22
   4.4.2 Deflection at gear contact .............................................................................. 22
   4.5 CONCLUSIONS .................................................................................................... 23

5. ULTIMATE ACHIEVEMENTS WITH VARIABLE PRELOAD ........................................... 24
   5.1 INTRODUCTION ..................................................................................................... 24
   5.2 MAXIMUM ACHIEVABLE ENERGY LOSS REDUCTION ........................................ 24
   5.2.1 Power loss reduction ....................................................................................... 24
   5.2.2 Energy loss reduction ...................................................................................... 25

© SKF Automotive Development Centre | Confidential
5.3 Maximum achievable bearing life ................................................................. 25
  5.3.1 Optimum preload for life ........................................................................ 25
  5.3.2 Bearing life for the customer load cycle ............................................... 26
5.4 Maximum achievable stiffness .................................................................... 27
  5.4.1 Deflection at highest load condition ..................................................... 27
  5.4.2 Deflection at gear contact ..................................................................... 29
5.5 Conclusions ............................................................................................... 29
6. Finding the optimum pre displacement ........................................................ 30
  6.1 Introduction ............................................................................................... 30
  6.2 Limits for optimization ............................................................................ 30
  6.3 Optimum preload for friction ................................................................. 30
  6.4 Optimum pre displacement for life ......................................................... 32
  6.5 Optimum preload combined ................................................................... 33
  6.6 Linearization ............................................................................................. 33
  6.7 Conclusions ............................................................................................. 34
7. Analysis of the pinion unit with variable preload ........................................... 35
  7.1 Introduction ............................................................................................... 35
  7.2 Energy losses in the system ..................................................................... 35
  7.3 System life ................................................................................................ 35
  7.4 System stiffness ....................................................................................... 36
  7.4.1 Highest load condition ........................................................................ 36
  7.4.2 Deflection at gear contact .................................................................... 37
  7.5 Conclusions ............................................................................................. 37
8. Geometric optimization of the actuator .......................................................... 38
  8.1 Introduction ............................................................................................... 38
  8.2 Behaviour of the actuator ........................................................................ 38
  8.3 Optimization of the actuator ................................................................... 39
  8.3.1 FEM model with parameters .............................................................. 39
  8.3.2 Optimization study .............................................................................. 40
  8.3.3 Optimization results ............................................................................ 40
  8.3.4 Final check of the optimized design ..................................................... 41
  8.4 Conclusions ............................................................................................. 42
9. Conclusions and recommendations .............................................................. 43
  9.1 Conclusions ............................................................................................. 43
  9.2 Recommendations .................................................................................... 43
REFERENCES ....................................................................................................... 44
LIST OF FIGURES ................................................................................................ 45
  FIGURES ......................................................................................................... 45
  TABLES ............................................................................................................ 46
APPENDIX A ....................................................................................................... 47
  A1. Symbols ..................................................................................................... 47
  A2. Abbreviations .......................................................................................... 47
APPENDIX B ....................................................................................................... 48
APPENDIX C ....................................................................................................... 50
APPENDIX D ....................................................................................................... 54
APPENDIX E ....................................................................................................... 55
APPENDIX F ....................................................................................................... 56
1. Introduction

1.1 Background

Environmental protection plays an important role today. The automobile industry in particular is constantly criticized when it comes to pollutant emissions. Some of the key points in the regulation for passenger cars according to The European Automobile Manufacturers Association ACEA [7] are:

- A reduction in average CO2 emissions from new cars to 120 g/km.
- Average new car CO2 emissions should fall to 95 g/km in 2020.
- Penalties will be imposed on a sliding scale; manufacturers exceeding their target by more than 3 g/km will pay €95 per excess gramme. Smaller charges between €5 and €25 for excesses of 1 – 3 g/km.

Figure 1 shows that in 2008 only 16% percent of the new cars meet the requirement of 120 g/km. This means that a lot more measures have to be taken to go from 16% to 100% in the nearby future.

To reach these requirements, also the rolling bearing industry is committed to contribute towards a reduction of CO2 emission by improving the efficiency rate of vehicle drive lines. Depending on the driveline layout, the different gearbox variants present a potential for reducing the mechanical power loss in the transfer of engine power. From rolling bearing point of view, due to the given load conditions, axle drives offer a larger scope for improvement than standard transmission. The potentials for different parts of the driveline are given in Figure 2.

SKF started the ENVI program to work on these potential reductions. Part of this program is friction reduction in bearings by removing the initial preload when it is not needed. In a previous research an actuator is developed that meets this requirement.

1.2 Previous research

Some actuators for the variable preload project where developed in a previous research. In this research, the feasibility of multiple actuation principles for preload applications are investigated. Active actuation principles, like piezoelectric actuation, were not considered in this research due to their complexity and the associated costs.

The results of this research showed that there are multiple promising actuation principles for this application. A very interesting principle is the torque activated torsion beam, shown in Figure 3. This actuator is designed to expand under the influence of torque. Here, the higher the torque applied, the higher the expansion and the higher the preload. Therefore, the preload becomes a function of the torque. This is a very promising characteristic of this actuator.

The actuation principle is further developed to a basic concept shown in Figure 4. Here, the concept is applied on a pinion unit but it can be applied to other applications as well. The idea is to
replace the U-joint, that normally fixes the prop shaft with the pinion unit, with the actuator. The torque flow from the prop shaft is directed through the actuator causing the actuator to expand. More details about this project can be found in [4].

The scope of the project is to determine the influence of preload on bearing units whereby the pinion bearing unit is used as reference.

1.3 Thesis objectives
Although there were good reasons for developing a variable preload system, the exact effects of variable preload are still unknown. Therefore, a second research is performed to investigate the effects of variable preload with regard to the application of the torque activated torsion tube on a pinion unit of a differential.
The objectives for this research are divided into a main objective and a number of sub objectives:

Main objective
To optimize a given passive variable preload actuator with respect to the friction losses, and to consider the potential benefits for application in a differential of a rear- or four wheel drive vehicle.

Sub objectives
a) To perform a literature study about bearing preload
b) To perform a literature study about the pinion unit and its surroundings
c) To assess the potential advantages when applying variable preload on bearing units
d) To develop a method for optimizing preload for bearing units.
e) To optimize a given passive variable preload actuator for application in a rear- or four wheel drive vehicle.

Scope of the project
For the calculations two load cycles are used. The customer load cycle, that is used during the development of the Hybrid Pinion Unit (HyPU). This load cycle consist of 27 load cases, each with a prescribed time share. This load cycle is used to calculate the bearing life. A detailed description is appended in Appendix B. The other load cycle is the New European Driving Cycle (NEDC). This is the European standard for measuring the pollutant emissions and fuel consumption of a vehicle. This driving cycle consist of a predefined velocity pattern that the driver has to follow. The load conditions corresponding with this driving cycle are calculated using Vehicle Environmental Performance (VEP), a SKF software tool. A detailed description of the driving cycle and VEP are appended in Appendix C.
The project starts with a pre study on the pinion unit of a rear axle differential. During this study, essential information about pinion units is gathered from literature. This is converted to form a basis for understanding the effects of changing the preload on the bearings for a pinion unit. The study is described in detail in chapter 2. Besides investigating the application, a detailed analysis is performed on the influence of preload on rolling bearings in general. The main parameters investigated are life, stiffness and friction. Details about this study can be found in chapter 3.

After this pre study, chapter 4 is used to analyze the current configuration of the pinion unit with constant preload. This is the baseline, that is used for comparing the potential benefits of variable preload.

Next, in chapter 5, the ultimate achievements are calculated for life, stiffness and friction. That is, assigning for each parameter the best preload, and calculating subsequently the benefits for that particular parameter.

The data gathered in the previous described chapters forms a basis for the optimization of the preload characteristics in chapter 6. There are two solutions here. The theoretical optimum solution, that doesn’t considers the linear relationship of pre displacement with the applied torque.

This curve cannot be constructed with the given actuator because the actuator couples the pre displacement linearly with the applied torque. Therefore, a second solution exist that is the linear variant of the theoretical solution. Furthermore, comparing both curves gives the potential negative consequence of using an actuator that is restricted to this linear relationship.

This comparison is made in chapter 7. Here, the two curves are compared with each other and with the reference pre displacement. The linear curve is used as input for the geometric optimization of the actuator in chapter 8. The optimization is performed using the Finite Element Method (FEM) program Pro/Mechanica and the nonlinear program MSC Marc.

The main conclusions and recommendations of this research can be found in chapter 9.

At the end of the report six appendices have been added:

A Symbols and abbreviations
B Bearing Beacon model and parameters
C Workflow for calculating energy losses with VEP
D Magnified figures
E Optimization results
F Project description
2. The pinion unit and its surroundings

2.1 Introduction
The pinion unit of a rear axle differential is a potential application where variable preload can be applied. First, the reader is introduced to this type of differential and its key components. Because variable preload is expected to affect the properties of the bearings significantly, the design requirements of the pinion unit need to be known. The information provided in this chapter is useful to determine the required bearing characteristics, that are examined in chapter 3.

2.2 Rear axle differential
A rear axle differential transmits the power from the gearbox to the rear wheels. The power flow to the wheels has to be rotated 90°. Therefore, the differential consists of a crown gear that can be spiral or hypoid. The crown gear requires very rigid support bearings to achieve proper tooth contact and uniform load distribution across the width of the teeth [8]. To achieve a compact differential design, the pinion shaft is mounted with overhung bearings (see Figure 6). These bearings are normally two Tapered Roller Bearings (TRB) arranged in a X-arrangement with a certain offset to increase the bending stiffness. TRB’s are used because only they can bear the heavy loads from resulting from the tooth contact.

To reduce friction, studies have been performed to replace the TRB’s with Angular Contact Ball Bearings (ACBB) [8]. However, the fatigue life, stiffness and loading capacity of ball bearings are inferior in comparison with roller bearings. A compromise is made by SKF with the Hybrid Pinion Unit (HyPU). This bearing assembly uses a TRB for the head bearing and a ACBB for the tail bearing (see Figure 6). The outer rings of both bearings are merged which increases the stiffness of the system. To increase the stiffness even more, a preload is applied to the bearings of the crown wheel and the pinion unit. The preload on the crown gear is generally lower compared with the pinion unit and the bearing speeds are much lower due to the final drive ratio. Therefore, the power losses are much lower for the crown gear bearings, and becomes variable preload for the pinion unit more interesting. Power loss reduction is interesting for reducing CO₂ emissions as well as reducing temperatures caused by the heat generation in the bearings. Hayashi et al. have already examined the influence of temperature and the effect on the preload in [9]. Although it isn’t part of the project to calculate the influence on the temperatures. It's an important effect caused by the power losses and gives a second reason for using variable preload.

2.3 Hypoid gears
2.3.1 Gear geometry
The differential consists of a crown gear and a pinion gear. The axis of the pinion gear is set to a certain offset with regard to the axis of the crown gear (see Figure 8). The pinion, which is connected to the drive shaft, can be lowered this way. This means that the drive shaft doesn’t intrude into the passenger compartment. The offset and the geometry of the tooth profile result in significant reaction forces that are discussed in the next paragraph.

Figure 7 HyPU unit

Figure 6 Rear axle differential

© SKF Automotive Development Centre | Confidential 8
2.3.2 Gear loads

The forces generated in the tooth contact zone during the transfer of torque from the pinion unit to the crown gear can be calculated when the gear dimensions are known. However, the gear loads are already known from the load cycle provided by a customer (Appendix B). Therefore it’s more meaningful to use these values than to calculate these values again using the gear geometry. A linear relationship is found between the torque and the three force components in the contact zone. The equations are given by:

\[ F_t = 0.1638 \cdot T \]  \hspace{1cm} (2.1)
\[ F_a = 30.475 \cdot T \]  \hspace{1cm} (2.2)
\[ F_i = \frac{1}{0.034} \cdot T \]  \hspace{1cm} (2.3)

Normally these reaction forces are calculated using the complete differential in SKF Bearing Beacon (SKF BB), a SKF software tool. However, the pinion unit is entering the differential under an angle, which is not feasible in Bearing Beacon. Therefore only the pinion unit is considered, using the tooth reaction forces from the load cycle. More details about the Bearing Beacon model are described in Appendix B.

2.3.3 Gear performance and efficiency

Next to friction in the bearing, there is also friction between the tooth-tooth contacts of the gears. There is an indirect relationship between the rigidity of the pinion shaft and the friction behaviour in the contact area. The models that are used for this type of calculation consist of very complex calculations. An example of such a model is developed by Xu et al. in [14]. To append these effects into the calculations on variable preload is too elaborate and is outside the scope of this project. Especially, because the values need to be optimized and therefore made parametric. For that reason, the gear efficiency is not examined.

2.4 Design requirements for pinion units

2.4.1 Stiffness requirements

The main requirement for pinion units is high stiffness. In literature, it is most of the time referred as system rigidity. The rigidity of the system is defined by the amount of deflection during loading. It is possible to compare different designs for rigidity by measuring the deflection. High rigidity is needed to have proper tooth contact resulting in smooth running and low noise production. Two rigidity requirements were found in literature. The first one, found in [1] gives deflection limits for the deflection and the deflection angle. The deflection of the pinion shaft is calculated with:

\[ f = \sqrt{f_x^2 + f_y^2} \]  \hspace{1cm} (2.4)

The deflection angle at the end of the shaft is calculated with:

\[ \tan \varphi = \sqrt{\tan^2 \varphi_x + \tan^2 \varphi_y} \]  \hspace{1cm} (2.5)

Then Lechner and Naunheimer state that the permissible deflection is
The permissible deflection angle is:
\[
\tan \phi_{\text{perm}} \leq 0.001
\]

The second one, found in [2] gives separate deflection limits for the bearings and shaft. These deflections with their limits are given in Figure 10. Here, the deformation of the pinion shaft is separated from the radial deflection of the bearings.

![Figure 10 Rigidty requirements (Fenton, 1998)](image)

**2.4.2 System life requirements**

The standard system life requirement for automotive components, used by automotive suppliers and manufacturers, is 1500 h. This is the life that the component must have for passing the acceptance tests. The output of the simulations however, gives not an exact prediction of the life because the output is depending on the accuracy of the input. That means that if the output of the calculations is lower than 1500 h, the design can still be good enough. Instead of using an absolute minimum life requirement it is better to take a good look at the individual results and estimate the possibilities for tuning the design when system life is indeed too low in practice.

**2.5 Dynamics of the system**

**2.5.1 Vibrations in the pinion unit**

Vibration problems are a large issue in dynamical systems. Whether these problems occur depends on the stiffness, damping and masses inside the system. Since the bearings in the pinion unit have the property to act like springs with very high stiffness, the pinion unit becomes an interesting dynamical system. Especially, because variable preload will influence the stiffness of the bearings. Similar analyses are already performed by Aini et al. [13] and Alfares et al. [12]. These analyses were for grinding spindles but the geometry of these spindles is similar to those of a pinion unit. Figure 11 shows the dynamic model of a grinding system.

![Figure 11 Dynamic model of a grinding system](image)

The main differences between a pinion unit and a grinding spindle are given in Table 1.

<table>
<thead>
<tr>
<th></th>
<th>Grinding spindle</th>
<th>Pinion unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearings</td>
<td>2 ACBB</td>
<td>2 TRB ACBB + TRB</td>
</tr>
<tr>
<td>Reaction forces</td>
<td>Fx,Fy</td>
<td>Fx,Fy,Fz</td>
</tr>
<tr>
<td>Moments</td>
<td>Low</td>
<td>High</td>
</tr>
<tr>
<td>Speeds</td>
<td>High</td>
<td>Low</td>
</tr>
</tbody>
</table>

Due to the significant differences between the two models, it is not feasible to use the results found in these technical papers. However, the method that is used can be used for pinion units as well. The derivation of this model requires the time of a second project and is therefore out of the scope of this project.

Although the results from the analysis performed by Alfares et al. [12] cannot be used directly, there are some interesting conclusions found in this study:

- The larger the initial preload applied, the less vibration amplitudes are generated.
- As the initial preload increases, i.e., the stiffness of the bearing increases, the dominant frequencies of the system shift to higher values.
- As the preload increases up to a certain value, the peak to - peak amplitude decreases. Beyond this value the reduction in vibration amplitude is
insignificant which indicates that larger values of preloading will not further reduce the vibration levels of the machine spindle. Therefore, this analysis can be used to calculate the optimum initial axial preload.

2.5.2 Vibrations in the driveline

At one end, the pinion unit is connected with the drive shaft with a universal joint connection. The basic principle of the variable preload concept is to replace this universal joint with the passive preload actuator. The connection however, will be less rigid and it is possible that this causes unwanted vibrations in the driveline. A distinction is made between torsional and bending vibrations. Bending vibrations don’t occur in the universal joint because of the limited length and good radial rigidity. The torsional stiffness however, is less and should therefore be examined. Normally, the total driveline would be calculated for torsional stiffness and vibrations. This takes a lot of time and requires a lot more unknown input. Therefore a more relative approach is chosen. The torsional stiffness is compared for a regular drive shaft and for the same drive shaft including the actuator. Depending on the decrease in stiffness it is determined whether further analyzing is necessary.

The torsional stiffness of a shaft is given by:

\[ K_{shaft} = \frac{G \cdot J}{L} \]  

(2.6)

with

\[ L = \text{length of the shaft} \ [m] \]
\[ G = \text{modulus of rigidity steel} = 79300 \ [N/m^2] \]
\[ J = \text{polar moment of inertia} = \frac{\pi \times (r^4 - r_i^4)}{2} \ [m^4] \]

Considering a regular solid prop shaft with a outer diameter of 25 mm and a length of 800 mm, the torsional stiffness for this shaft is 3041 Nm/rad.

The torsional stiffness of the actuator is calculated with the Finite Element Method, using Pro/Mechanica. The results show that the torsional stiffness of the actuator is 1.01 \( \cdot \) \( 10^5 \) Nm/rad.

The torsional stiffness of the prop shaft without actuator is equal to the torsional stiffness of the prop shaft; 3041 Nm/rad.

The torsional stiffness of the propshaft with actuator is given by:

\[ K_{total} = \frac{1}{K_{shaft}} + \frac{1}{K_{actuator}} = 2952 \frac{Nm}{rad} \]

This means that the actuator decreases the torsional stiffness of the propshaft by 2.9 %.

2.6 Conclusions

In this chapter, the basics and main aspects of the differential and the pinion unit are discussed. The preload is very high for the pinion unit and therefore interesting for a study about variable preload. The HyPU is used as reference case whereby the loads from the load cycle, provided by a customer, are used as boundary conditions. The main parameters that should be investigated when changing the preload are life, stiffness and friction losses. The friction losses are not only interesting because of the reduction in CO\(_2\) emissions, but also for the reduction of the temperature increase.

A quick investigation on the dynamic properties of the system is done. From analysis on a grinding spindle, found in literature, it can be concluded that the vibrations are not a problem if the preload is not too low. In case of doubt a five degree of freedom model is found in literature to calculate these effects. Furthermore, the influence of the actuator within the driveline is investigated. The results show that the torsional stiffness of the prop shaft decreases with 2.9% when the actuator is added. This is within the error of the calculation. Therefore, the actuator doesn’t cause additional problems when it is added to the driveline.
3. Effects of bearing preload on the system parameters

3.1 Introduction
In chapter 2 it is defined that, with regard to the pinion unit requirements, the main bearing criteria are stiffness and bearing life. Stiffness needs to be high enough to meet the design requirements, bearing life needs to be according to the life standards of vehicles. Because friction reduction is the field of interest, the parameter friction is also very important. These effects will be discussed in this chapter. Starting with a short introduction about bearing terminology, followed by the effects of preload on stiffness, life and friction.

3.2 Bearing terminology
Before starting to evaluate the effects of bearing preload, a short introduction is made for the reader to get familiar to some basic bearing macro geometry.

Figure 12 shows a cross section of an ACBB. The main dimensions are:

\[ d_o = \text{outer diameter} \]
\[ d_i = \text{inner diameter} \]
\[ d_m \approx \frac{d_o + d_i}{2} = \text{mean diameter} \]
\[ D = \text{ball diameter} \]
\[ \alpha = \text{contact angle} \]

Because of clearance, the mean diameter is not exactly the mean between the outer and inner diameter. Though, for our calculation the mean diameter can be calculated this way. The contact angle determines the capability of an ACBB to resist axial loads. The contact angle usually doesn’t exceed 40°. The inner and outer raceways are mostly referred as inner and outer ring respectively. In general, the balls (or rollers for a TRB) are referred as rolling elements.

Figure 13 shows a cross section of a TRB. The main dimensions are:

\[ \alpha_o = \text{cup contact angle} \]
\[ \alpha_c = \text{cone contact angle} \]
\[ \alpha_r = \text{roller angle} \]
\[ l = \text{length of roller} \]
\[ T = \text{Bearing width} \]
\[ B = \text{Cone width} \]

The terminology differs somewhat from other bearings. The outer raceway of a TRB is usually called the cup, and the inner raceway is called the cone. Because of the difference between the cone and cup contact angles, there is a force component that drives the tapered rollers against the rib. This causes a significant increase of friction.

3.3 Effects of bearing preload on the stiffness
Preload plays an important role when determining the stiffness properties of a bearing. To show the effects of preloading, some calculations were done with the deflection equations given by Harris and Kotzales in [3]. The equation for the axial deflection of an ACBB is given by:

\[ \delta_a = 4.36 \cdot 10^{-4} \left( \frac{F_a}{Z \cdot \sin(\alpha)} \right)^{\frac{2}{3}} \frac{D^{\frac{1}{3}} \cdot \sin(\alpha)}{D^{\frac{1}{3}}} \]  \( (3.1) \)

with
\[ F_a = \text{axial force} \]
\[ Z = \text{number of rolling elements} \]
\[ \alpha = \text{contact angle} \]
\[ D = \text{ball diameter} \]
The equation for the axial deflection of a TRB is given by:

$$\delta_a = 7.68 \cdot 10^{-5} \left( \frac{F_a}{Z \cdot \sin(\alpha)} \right)^{0.9} \left( \frac{1}{l^{0.8} \cdot \sin(\alpha)} \right)$$ (3.2)

with

- $F_a$ = axial force
- $Z$ = number of rolling elements
- $\alpha$ = contact angle
- $l$ = roller length

The limitation of these equations is that the radial deflection is zero, when the axial deflection is calculated. Therefore, the influence of preloading on the radial stiffness can’t be evaluated. This isn’t a very big issue because these calculations are used to get a feeling about the effects of preload. In a later stadium, SKF Bearing Beacon will be used to evaluate the total stiffness properties. In this section, stiffness refers to axial stiffness only. The calculations are divided into three different steps. First, the deflection and stiffness of a single ACBB is calculated. Next, the same is calculated for a TRB. These two calculations are then used to evaluate the deflection and stiffness of a duplex bearing arrangement, containing one ACBB and one TRB.

### 3.3.1 Stiffness of a single ACBB

The ACBB in the HyPU has 13 balls with a ball diameter of 12.7 mm and a contact angle of 40°. These values are used to calculate the deflection of the ACBB for different axial loads. Normally, preload is only valid for duplex bearing arrangements. In this case, a preload would just be an additional axial force. Therefore, the zero deflection point will move to the point corresponding to the preload force. For example: When a preload of 2000 N is applied, the curve start not at 0 mm but at 0.01 mm (see Figure 14). In this figure, the axial deflection is calculated corresponding with an axial force. The results are checked in SKF Bearing Beacon. This plot shows that the deflection rate at the beginning is much larger than at higher axial loads. This so-called knee in the curve can be excluded by preloading the bearing. The stiffness of a bearing is given by the change in force divided by the change in deflection. This is done in Figure 15. This graph shows that the stiffness increases over the total loading range, but the rate is much higher at lower loads than at higher loads. For example, the stiffness increase for 100N – 2000N is 171 %. For 5100N to 7000 N is it 11 %.

![Figure 14: Axial deflection of a single ACBB subjected to an axial load](image)

![Figure 15: Axial stiffness of an ACBB subjected to an axial load](image)

### 3.3.2 Stiffness of a single TRB

The TRB in the HyPU has 19 rollers with a length of 22 mm and a contact angle of 17.5°. These values are used to calculate the deflection and stiffness of the TRB. The results are shown in Figure 16. The ‘knee’ in the curve of an ACBB almost disappeared for a TRB. The results are again evaluated with SKF Bearing Beacon. There are some differences between them. This is likely because the complexity of TRB’s is much higher, leaving more room for errors. The error however is still acceptable. Comparing the results with the ACBB shows that the deflection of a TRB is much lower than for an ACBB, subjected to the same load. Figure 17 shows the stiffness of a TRB as function of the axial load. Here, the increase in stiffness is 35 % for 100-2000 N and 3.2 % for
5100-7000 N. From this, we can conclude that preloading a TRB gives less improvement than preloading an ACBB.

3.3.3 Stiffness of a duplex TRB/ACBB

In the next calculation the ACBB and TRB are assembled together, similar to the HyPU. An axial load is applied at the TRB side of the shaft. For simplicity reasons, the range of the axial force is only chosen positive representing driving forward. Therefore, driving in reverse is not considered at the moment. Figure 18 shows the reaction forces acting on the model. \( F_a \) is the external axial force, \( F_1 \) is the reaction force resulting from the ACBB, \( F_2 \) is the reaction force resulting from the TRB.

Two equations need to be solved to determine the values of \( F_1 \) and \( F_2 \):

\[
F_a = F_2 - F_1 \quad (3.3)
\]

\[
\delta_{trb} = \delta_{acbb} \quad (3.4)
\]

The last one states that the increase in deflection of the TRB must be equal to the decrease of deflection of the ACBB. The above equations are calculated for different values of \( F_a \), with an initial preload of 5000 N. This is shown in Figure 19. It needs to be mentioned, that preload is not the same as axial force anymore, this was only valid for calculations with only one bearing. In this figure \( F_1 \) and \( F_2 \) are both equal to 5000 N at zero axial load corresponding with the preload. Increasing the axial load, increases \( F_2 \) and decreases \( F_1 \). \( F_1 \) will eventually become zero, depending on the initial preload.
that the single TRB deflects almost twice as much as the duplex bearing.

Figure 20 shows the deflection at one particular preload, i.e. 5000 N. The same calculation is done for several amounts of preload. This is shown in Figure 21. It’s clear that the deflection decreases with increased preload. From Figure 21 the stiffness for each preload is calculated by dividing the maximum axial load by the deflection corresponding with that load. This results in Figure 22. Here, the preload is varied from 2000 N to 10000 N. The stiffness increases with 39.5%. Therefore, the preload increases stiffness significantly where the rate is somewhat larger for lower preloads than for high preloads.

3.4 Effects of bearing preload on the life

3.4.1 Effects of preload on the internal load distribution

The improved life properties of a bearing result from the change in internal load distribution. Figure 23 shows the different load distributions for

A: zero clearance
B: preload
C: clearance

It shows that for a preloaded bearing, more elements are loaded resulting lower load per rolling element. This is why the life increases. For a bearing with clearance the amount of elements loaded decreases resulting in a higher load per rolling element. Therefore, life decreases.
3.4.2 Life prediction calculations

There are three categories of fatigue life:
- Equivalent load method
- Rolling element load method
- Stress integration method

Any life method can be placed in one of these categories.

Equivalent load method
The equivalent load method can normally be found in handbooks. The most known standard method is the ISO 281 addendum 4. The life is calculated using:

\[ L_{10} = \left( \frac{C}{P} \right)^n \]  

(3.5)

Where
- \( C \) = dynamic load rating (found in product tables)
- \( P \) = Equivalent dynamic bearing load
- \( n \) = 3 for ball bearings, 3.33 for roller bearings

This method however, only considers the external loads on the bearing. Therefore, the effect of preload is neglected.

Rolling element method
The rolling element load method uses the same starting point as the equivalent load method. The difference is that this theory is based on the load on each rolling element. Therefore, preload can be taken into account. The complexity of this method makes it not useable for quick calculations on the effect of variable preload.

Stress integration method
The stress integration method is the most recent evaluation method. It is based on the subsurface stresses in the contact area. This makes it a lot more advanced than the other methods and also a lot more complex. On the other hand, any type of loads and influence factors can be taken into account. The equations derived in this theory can't be used for a quick calculation. Though, SKF Bearing Beacon can be used instead.

3.4.3 Life calculations with SKF bearing beacon

Three different analyses are performed in SKF Bearing Beacon to determine the influence of preload on the bearing life. First, the bearing life of a model containing only one ACBB is calculated for different radial loads, see rectangle 1 in Figure 24. A single ACBB isn't designed to withstand moment loads. Therefore, the radial load is applied in the center of the bearing. Secondly, the bearing life of a TRB is calculated under the same loading conditions (rectangle 2 in Figure 24). A TRB can withstand moment loads to a certain level but for uniformity it's chosen to apply the radial force in the center. Third, the system life of the model with the ACBB and the TRB combined is evaluated (rectangle 3 in Figure 24). The distance between the bearings L1 is 50 mm according to the HyPU unit. Two different loading conditions are used. First, a centric radial load is applied resulting in a pure radial load in both bearings. Secondly, an analysis is performed with a radial force with an offset of 20 mm. For all three analyses, the pre-displacement is varied from preload to clearance. Other influences on the internal clearance like temperature, shaft/housing interferences and bearing speeds are not taken into account during these analyses.

Results
The result from the first analysis is shown in Figure 25. The axial pre-displacement is varied from -40 \( \mu \text{m} \) preload to 20 \( \mu \text{m} \) clearance. The life is plotted for 7, 8 and 16 kN radial load. All three curves show a optimum when the bearing is slightly preloaded. It's worth noticing that the optimum moves to the left when higher radial loads are applied. This phenomenon can be explained with the earlier discussed internal load distribution. Because the radial load compresses only one side of the bearing, the internal load distribution is becoming like Figure 23 C. A higher preload is then needed to achieve the load distribution given in Figure 23 B.

Another interesting phenomenon is that for small radial loads, the influence of preload on the life is much higher than for high radial loads. Therefore, maintaining the preload at the optimum is more important for low radial loads than for high radial loads.

Another result from the study, which isn't shown in the graph, is that for very low radial loads the optimum can even move to clearance.
3.5 Effects of bearing preload on the friction losses

During operation friction losses occur in the bearing due to rolling, sliding, drag losses and losses from the seal. The frictional losses are strongly dependent on the bearing load and the environment of the bearing. In this section these losses are calculated using the mathematical model from SKF for calculating frictional moments [16]. This is done for the ACBB and for the TRB. Figure 29 shows the frictional moment of the ACBB as function of the radial and axial load. It shows that the frictional moment is almost linear for both radial and axial loads. Therefore, radial load and axial load needs to be as small as possible. For a TRB however, Figure 30 shows that the frictional moment is strongly nonlinear. Low frictional moments occur only at very low loads. This is likely because the friction caused by sliding between the roller and the rib starts to play a significant role when the bearing is loaded. This figure also shows that for a TRB, the axial load has far more influence than radial load, except for zero axial loads.
The above plots show the influence of loading on the bearing frictional moment but conclusions about preload can't be made easily. To check the influence of preload, a case study is performed in the next section.

3.5.1 Case study: constant reaction force

The reaction forces resulting from the ACBB and the TRB are influenced by the external axial force and the preload. Figure 19 shows these reaction forces as function of the axial load with a static preload of 5000 N. This load case is used as a reference for determining the efficiency improvements. Next, a variable preload function needs to be defined to recalculate the new reaction forces. The preload function can be anything. For now, it's chosen to change the preload in such a manner that the reaction force at the ACBB stays constant. This is plotted in Figure 31.

Therefore, the axial reaction force for both bearings is calculated as function of the external axial force. The radial force is somehow related with this external axial force but for now this force is detached and varied from 0 to 15000N. For both situations (constant and variable preload) the frictional moment can be calculated as function of the external axial force and the radial force. This results in two 3D plots similar to those given in Figure 29 and Figure 30, except the axial force is not the one acting on the bearing but the one acting on the total system. A comparison is made between both plots resulting in the improvement caused by variable preload. This is plotted for the ACBB in Figure 32. Here −40% means that the frictional moment for variable preload is 40% lower than for the static preload. The pivot point is at an axial load of 12.600 N. Then the variable preload becomes higher than the static preload resulting in an increase of friction.

The steps made for the ACBB, can also be taken for the TRB. The plot is given in Figure 33. The surface looks similar that for an ACBB, except for the influence of the radial load. Because this influence is less, the surface becomes more even in the direction of the radial force.
The improvements given above look rather promising. The overall efficiency depends however, on the load cycle that is used. If for example the axial load never exceeds the 12,600 N limit, the overall efficiency would become very good. If not, the overall efficiency could turn out pretty bad. It has to be mentioned that the previous results strongly depend on the preload function that is used. The preload function that is used for now is not optimized and other negative influence on stiffness and life are not considered.

### 3.6 Conclusions

In this chapter, the influence of bearing preload on the stiffness, life and friction is described. The results show that preloading single bearings increases their axial stiffness significantly, especially ACBB’s are very sensitive for preload. Next, the stiffness of a system with two bearings, i.e. one ACBB and one TRB, is examined. The results show that stiffness increases significantly with 39.5% when the preload is increased from 2000N to 10,000N.

The influence of preload on life is examined with the software program SKF Bearing Beacon. The results show that the optimum life strongly depends on the amount of radial load that is applied. High radial loads require high amounts of preload to maintain optimum life. Low radial loads however, show an optimum at very low preload or even at clearance. In general the influence of preload is much lower for high radial loads than for low radial loads. Therefore, it’s more important to maintain the optimum preload at low loads than at high loads.

The influence on friction losses is determined using the equations given by the SKF calculation manual [16]. The frictional moments are calculated for a situation with static preload of 5000N and a situation with variable preload. The results show that a significant decrease in friction losses is feasible. These results however, are strongly dependent of the algorithm that is used for varying the preload and the external loading conditions. For the algorithm used in this chapter, the improvement in frictional moments vary from -45% to 22% for the ACBB and from -30% to 19% for the TRB. Where minus means decrease and plus means increase of frictional moment.

Resultantly, preload needs to be as high as possible for stiffness and as low as possible for low friction, and somewhere in between lies the optimum life. Therefore, a real optimum for all parameters is not likely to exist. In the next chapters, an ideal compromise is established.
4. Analysis of the pinion unit with static bearing preload.

4.1 Introduction

In this chapter, the friction losses, system life and system stiffness are calculated for the static preloaded hybrid pinion unit. The results are used to quantify the effect of variable preload on the pinion unit by comparing these results with other configurations. SKF Bearing Beacon is used for the calculations on the pinion unit. For detailed information about the model and the inputs, see Appendix B. Two different load cycles are used in this chapter. The customer load cycle and the NEDC driving cycle. The customer load cycle is more representative for realistic operation conditions. It consists of 27 load cases with a specific time share. This load cycle is used for quantifying life and stiffness of the system. The NEDC driving cycle is the European standard for measuring the fuel consumption and pollutant emissions of a vehicle. This cycle is used to determine the energy losses in the system. A detailed description of both cycles can be found in Appendix B and Appendix C.

4.2 Energy losses in the system

4.2.1 Calculation of energy losses with VEP

The NEDC cycle is used for calculation of the energy losses in the system. This driving cycle consist of a predefined velocity pattern that the driver has to follow. The loads on the HyPU, corresponding with this driving cycle are deducted from the Vehicle Environmental Performance tool (VEP). VEP is based on a Matlab/Simulink model, that includes a driver model for throttle and brake controls, an engine map, a vehicle model and a driveline model. The individual models are connected in a closed loop. During simulation it compares the output velocity with the velocity pattern of the driving cycle and corrects correspondingly the input. The result of the simulation is the fuel consumption, CO₂ emissions and the loads on the individual components.

The car used for the VEP simulations, is a Front Wheel Driven (FWD) vehicle that normally doesn’t have a differential with a pinion unit. A rear wheel driven vehicle is not available yet. To solve this problem the pinion unit is added to the driveline of the FWD vehicle. This means that the energy losses are slightly higher, but the influence of decreasing the friction in the pinion unit can still be investigated. The car that is used is the GM Insignia 1.6 turbo.

Normally, the energy losses are directly given by VEP. Though, running a new simulation for each new variant of preload is not preferable. Instead, the energy losses are calculated for each situation using the loads with timeshares given from VEP. The method consists out of calculating the power losses for each load condition with SKF Bearing Beacon and multiplying the result with the time that the load condition occurs. Unfortunately, there are approximately 5900 different load conditions that make it not feasible to calculate the power losses for each condition. Instead, the torque and speed range is divided into a matrix of sub ranges where each condition is assigned to a certain range. The average power loss and the timeshare for each range can be calculated. The time shares from the VEP calculations are given in Figure 34. Each bar represents a small sub range of the total torque and speed range. The height of each bar corresponds with the time that the load conditions occur in this sub-range, during the driving cycle. It’s interesting to notice that the loads from the driving cycle are very low compared with the loads from the customer cycle. That means that when the preload is properly varied, high life and stiffness in the customer cycle, as well as low friction losses in the NEDC cycle can be achieved.
speed and torque. Some areas of the graph however, will never occur in practice. For example, 1500 Nm in combination with 6000 rpm.

The VEP results give the torque and speed input of the differential for the NEDC cycle. The speed profile given in Appendix C shows that the speed is constant for some time at 50, 70, 100 and 120 km/h. This is interesting because the speeds can now be linked to a particular torque-speed combination. These combinations are used to determine the power losses for different speeds. These values are very useful because they are independent of the driving cycle. The power losses for the HyPU with constant preload are given in Figure 36.

The energy losses are the product of the power losses given in Figure 35 and the time that the power losses occur given in Figure 34. The time shares however, are given for a range and the power losses for a specific point. Therefore, for each bar the average torque and speed is used as reference for calculation of the power losses. The resultant energy losses for each bar are given in Figure 37.

The total energy loss in the HyPU during the driving cycle is the summation of all the individual energy losses. The results are:

- Total energy loss: 53.2 kJ.
- Average power loss: 75.8 W.
- CO₂ emissions (VEP): 161.51 g/km.

The total energy losses for this situation are calculated with VEP and with the simplified method using timeshares and power losses. The results from VEP give a total energy loss of 52.7 kJ. The error is very small using the simplified method.

The life of the system is calculated for the customer load cycle given in Appendix B. The results are shown in Figure 38, the life is plotted on a log scale to visualize the life at high loading conditions as well. The life is calculated for each load cycle separated into the life of the ACBB, the life of the TRB and the life of the system. The combined life is calculated using the time fraction of each load cycle. The combined life is then

- ACBB life: 1532 h
- TRB life: 5669 h
- System life: 1404 h

The life is lower than the required 1500 h. However, this is only a simulation and the
acceptance tests turned out that the life was enough.

![Image of life results graph]

**Figure 38 Life results of the HyPU with normal preload**

### 4.4 System stiffness

The system stiffness is quantified using the deflection of the pinion shaft. The deflections are calculated using the program SKF Bearing Beacon. First the deflection of the total shaft is analyzed with the maximum load form the customer load cycle. The results are given in section 4.4.1. In section 4.4.2 the deflection at gear contact is calculated as function of the applied torque.

#### 4.4.1 Deflection at maximum load

Maximum deflection of the shaft will occur when the highest load is applied. Therefore, this loading point is often used to quantify the stiffness of the system. The highest load is 1331 Nm given from the load cycle data in Appendix B. This value is converted into an axial, radial and tangential load with the equations given in paragraph 2.3.2.

The results are given in Figure 39. Here, the radial displacement at the ACBB is 27 μm, at the TRB 59 μm and at gear contact, where the tooth engages, the displacement is 172 μm. These values are rather high compared with the stiffness requirements given in paragraph 2.4.1. However, according to the designer of the HyPU, Thomas Wolf, the deflection isn’t a very large issue. This is because the manufacturer of the gears can compensate these deflections with modifying the gear geometry. It doesn’t mean that every deflection is permissible. It’s more likely that the bearing can’t carry the load than that the deflection becomes too high. The same is true for the axial deflection which is 84 μm at the ACBB, 96 μm at the TRB and 99 μm at gear contact.

![Image of shaft deflection graph]

**Figure 39 Shaft deflection for 20 μm preload**

#### 4.4.2 Deflection at gear contact

It’s not feasible to show for each torque the deflection of the total shaft as given in the previous section. To compensate that, only the deflection at gear contact is plotted as function of the torque. The result is given in Figure 40. The torque range is chosen -500 to 1000 Nm. Notice that the rate of
deflection for positive torques decreases with higher torques. This effect for negative torques is much smaller. That's because the axial load is mainly taken by the ACBB with positive torques. With negative torques, the axial load is mainly taken by the TRB. The TRB stiffness is less influenced by external loading than the ACBB (section 3.3). The effect is that the rate of deflection is lower for negative torques. Furthermore, the amount of deflection is also lower for negative torques due to the higher stiffness of a TRB.

![Figure 40 Deflection at gear contact as function of the applied torque](image)

4.5 Conclusions

The energy losses in the HyPU unit with normal preload using the NEDC driving cycle are 53.2 kJ. It's interesting, that the loads from this driving cycle are very low compared to the loads from the customer cycle. This makes it feasible for optimizing the bearing properties for energy losses as well as stiffness and life. The life of the current design is 1404 hours which is lower than the required 1500 hours, and therefore a matter of attention. In a later stage it should be investigated, what parameters can improve the life in the real product, e.g. lubrication. Reducing bearing life even more is certainly not preferable.

The radial deflection is 172 μm and the axial deflection 99 μm. This is higher than the requirements found in literature. New developments however, show that by changing the gear geometry the deflections can be compensated. Though, minimizing these deflections should always be the goal of the design optimization.
5. Ultimate achievements with variable preload

5.1 Introduction
Chapter 3 showed that there isn’t one single preload where all the system parameters are optimal. This chapter describes the effects on the system parameters when the preload is optimal for one parameter, neglecting the other parameters. Then these results are compared with the current system performance of chapter 4. Resulting in the ultimate achievements of applying variable preload on a pinion unit.

5.2 Maximum achievable energy loss reduction

5.2.1 Power loss reduction
The friction generated inside a bearing is largely dependent on the load applied to the bearing. Preload is just an extra axial load applied to the bearing to improve its properties. Therefore, the lower the preload, the lower the friction losses. The absolute maximum friction reduction is achieved when the preload is set to zero for every loading condition. Higher friction reductions are not feasible with variable preload.

The power losses generated within the HyPU unit with -20um preload were already calculated in section 4.2. The same method is used to calculate the power losses within the HyPU unit for zero preload. The results are given in Figure 41.

Both situations can be compared with each other resulting in the maximum feasible friction reduction for every loading condition (Moment, speed). The results are shown in Figure 42. Here, the results show that the speed has little influence on the friction reduction. Therefore, a 2D picture of the graph given in Figure 43, gives a lot better picture of the friction reductions.

<table>
<thead>
<tr>
<th>Moment (Nm)</th>
<th>Max speed</th>
<th>Min speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 Nm</td>
<td>-2.83 %</td>
<td>-4.37 %</td>
</tr>
<tr>
<td>300 Nm</td>
<td>-1.25 %</td>
<td>-1.47 %</td>
</tr>
<tr>
<td>500 Nm</td>
<td>-0.9 %</td>
<td>-1.05 %</td>
</tr>
</tbody>
</table>

Table 2 Values for friction reduction at low torques
In Figure 44, the power losses are given for different speeds. The blue bars represent the power losses with normal preload. The red bars represent the losses with zero preload. The reductions at constant speed are significant varying from 41% up to 71%. The main reason for these high reductions is that the torque, needed for driving at constant speed, is very low. For example, only 80 Nm is needed for driving 120 km/h. Therefore, the complete range for driving at constant speed is captured within the area where high reductions are feasible.

5.2.2 Energy loss reduction
The energy losses can be calculated using the same time shares as in paragraph 4.2.1. But now with the new power losses calculated for zero preload. The new energy losses are given in Figure 45. The summation of the individual losses gives the total energy loss during the NEDC cycle. The total energy losses are 27.3 kJ. The average power loss is 38.8 W. Notice that the energy losses for normal preload are 53.2 kJ. The maximum achievable reduction is then given by.

\[
\frac{53.2 - 27.3}{53.2} \times 100 = 48.7\% 
\]

This is a significant improvement of the energy losses. The value is high because the loads for this driving cycle are 1 to 3 times higher than the axial load caused by preload. Therefore, the influence of preload is significant. For higher load conditions the loads can become up to 15 times the preload value. The influence of preload at these loads is much smaller.

The pollutant emissions are also given by the VEP calculation. The CO₂ emission is 161.40 g/km. This is a CO₂ reduction of 0.11 g/km. This reduction seems rather small but it’s the results of just one bearing. Furthermore, the vehicle is driving only 60 % of the total cycle time. 16% of the time the vehicle is braking and 24 % of the time it’s stationary. That means that reducing friction in a bearing has only effect on 60 % of the total driving time.

5.3 Maximum achievable bearing life
Chapter 3 shows, that bearing life is the only parameter that has a real optimal pre displacement (bearing life is maximal). This optimum pre displacement is depending on the load condition. Theoretically it is possible to find for each load condition the optimal pre displacement. This is done in section 5.3.1 where the load range is divided in smaller sub ranges and for each sub range the optimal pre displacement is calculated. In section 5.3.2, the customer cycle is used to determine the effect of using the optimal pre displacements on bearing life.

5.3.1 Optimum preload for life
To calculate the optimum pre displacements, a matrix is defined with a range of -500 to 1000 Nm. For every combination, the optimal pre displacement for life is calculated. The optimal pre displacement values are plotted in Figure 46. It’s
interesting to notice that both, speed and torque have an influence on the optimum pre displacement values.

Figure 46 Optimum preload values for life (3D bar plot)

The absolute bearing life calculated for each combination is not meaningful because some conditions never occur and the time fraction for each condition is different. It’s more interesting to look at the change of life when the pre displacement is increased or decreased with 5 μm using the optimum preload as the initial preload. This results in the sensitivity of each condition to the change in pre displacement. Figure 47 gives the results for a larger pre displacement than the optimum. For example, a value of 0.2 shows that the life is 20% lower compared with the life at optimal pre displacement. It’s interesting that increasing the pre displacement has more influence on life when the torques are negative than with positive torques. And, in accordance with the results found in section 3.4, the influence of pre displacement decreases significantly when the torques are high.

Figure 47 Sensitivity for larger pre displacements than the optimum pre displacement

The result of a lower pre displacement than the optimum is shown in Figure 48. Notice the different legend that is used. The influence on life is approximately 6 times higher for increased pre displacement than for decreased pre displacement which is already shown in section 3.4. Again lower preloads have more influence for negative torques than for positive torques. Whereby low torques are more influenced by pre displacement.

Figure 48 Sensitivity for lower preloads than optimum preload

5.3.2 Bearing life for the customer load cycle

Theoretically, the pre displacement could be set to the optimum value for each load case given in Appendix B using variable preload. This results in the maximum feasible life for that load cycle. The optimum preload for each load case is determined using SKF Bearing Beacon. The results are given in Figure 49. The preload is varied from -70 μm preload to 30 μm clearance. For most load cases, the graph shows an optimum at ± 20 μm. For the low numbered load cases, the life is so small that the influence of preload can be neglected. The maximum difference there is about an hour between maximum and minimum life.

Figure 49 optimum system life for each load case
Bearing life of the system with a static preload of -20 µm is already calculated in chapter 4. The system life for this situation is 1404 hours. Bearing life is also calculated using the optimum pre-displacements found in Figure 49. The system life for this situation is 1406 hours. This means that life can theoretically be improved with a maximum of 2 hours. Practically, it’s unchanged considering the error margin in the life prediction method. Therefore, the effect of variable preload on life for this load cycle is very low. This is likely because the static preload is very good for the lower loading conditions where the influence of preload is the highest. Changing the preload for the high load conditions results in such a small improvement that the effect on the total life is wasted.

The previous results are based on the system life of the HyPU unit, considering both bearings. To complete the understanding of the effects of variable preload, a life calculation is made for both bearings separately. The results for the ACBB are given in Figure 50. The optimal preload for every load case is zero preload. Therefore, variable preload isn’t useful for optimizing the life of the ACBB. The maximum achievable life for the ACBB, using zero preload, is 1558 hours. The life of the TRB under these conditions becomes then 4865 hours. The system life however, decreases to 1394 hours.

The optimum life of the TRB is between -40 µm for low loads and -70 µm for high loads. Maintaining the optimum preload for all load cases results in a total life of 8448 hours for the TRB and 1391 hours for the ACBB. The system life however, decreases to 1332 hours.

5.4 Maximum achievable stiffness

Preload increases the stiffness of the bearings significantly. High stiffness is needed to reduce the deflection of the tooth to ensure proper tooth contact between the pinion gear and crown gear. There is however a maximum on the influence of preload. The similar approach used in the previous chapter is also applied here. First the deflection is calculated for the highest load condition. Followed by the deflection at gear contact as function of the applied torque.

5.4.1 Deflection at highest load condition

The deflection is measured for the highest load with different pre displacements. The results are shown in Figure 52 and Figure 53. The first figure shows the radial deflections for 0, 20 and 380 µm pre displacement. 380 µm is not realistic but it shows that even with very high preloads, a minimum deflection occurs. A more realistic maximum value is 60 µm preload (10 kN). It’s interesting to notice that the reduced deflection is mainly caused by the reduced deflection at the TRB. The deflection at the ACBB remains almost constant.
More interesting is the displacement reduction at different points when the pre displacement is increased. The points of interest are the location of the ACBB, the TRB and the point where the tooth engages. The reduction in radial displacement is given in Figure 54. The figure shows that if the maximum feasible preload is set to 60 μm, the reduction in radial displacement at the tooth contact is about 11%. For the TRB, the reduction is 22%. The radial displacement at the ACBB increases slightly (4%). This is the result of the changed force balance between the TRB and the ACBB caused by the increased preload.

Looking at the reductions in axial displacement given in Figure 55, the results show much greater reductions and the difference between the different points is much smaller. Reductions up to 20% are feasible at tooth contact.
5.4.2 Deflection at gear contact

Figure 56 shows the axial deflection at the tooth contact point for different torques values.

Notice that the pre-displacement only is influenced at low torques. The deflection is lower at high torques but then the curves remain parallel. This indicates that the pre-displacement has no influence anymore. The axial pre-displacement has the effect of keeping both bearings axially loaded up to a certain amount of external axial force caused by the torque. The higher the pre-displacement, the longer both bearings are axially loaded. As a result the deflection is lower when both bearings are loaded instead of only one bearing (see section 3.3). This explains the ‘pivot point’ in the graphs that is best shown at 60 µm. The same phenomenon is shown for the radial deflections given in Figure 57.

5.5 Conclusions

The most interesting with regard to energy loss reduction is the range of 0 - 500 Nm. Fortunately the loads from the NEDC cycle are within that range. This results into a energy loss reduction of 48.7 % for the zero preloaded HyPU compared with the 20 µm preloaded HyPU. The customer cycle has been used for calculating the optimal life. Applying the optimum preload for every load case of the customer cycle has less effect. The maximum gained system life is 2 hours. This is within the error of the life prediction method and should not be considered as an actual improvement. Therefore, optimizing preload for life is not interesting for this load cycle. The reduction in displacement caused by high preloads is significant. A maximum preload of 60 µm is used to review the maximum achievable displacement reduction. The reduction at the tooth contact point is then 11 % in radial direction and 20 % in axial direction. The optimum preload actuator developed in the next chapter should keep the preload as low as possible for the torques below 500 Nm and as high as possible for the torques above 500 Nm, without influencing the bearing life too much.
6. Finding the optimum pre displacement

6.1 Introduction
In the previous chapter is calculated that the friction and stiffness of the HyPU can be significantly improved by optimizing the preload. The goal of variable preload is to optimize the preload with respect to the friction losses, with an acceptable system life and stiffness. Friction losses can’t be reduced over the total load range due to the minimum required pre displacement for system life and stiffness in some parts of this range. Therefore the total torque range is split into the torque range where the main energy losses occur in the NEDC cycle and a range with the remaining torques. It makes sense to use the NEDC cycle for this calculation, since it is a standard in the automotive sector to determine fuel consumption. The first range, marked with red in Figure 58, is optimized to meet the largest friction reduction. The second range, marked with blue, is optimized to meet the best life and stiffness properties. In the previous chapters pre displacement is always mentioned as function of torque and speed. To reduce the complexity of the optimization method only the torque is used as variable. First the optimum preload is determined without considering the characteristics of the given actuator. This gives a more theoretical solution for optimal preload. At the end of this chapter these results are used to establish the optimum pre displacement curve when considering the passive preload actuator.

Figure 58 Split torque range

6.2 Limits for optimization
The amount of possible solutions is large. Solutions that are practically not feasible have to be skipped. Less solutions will reduce the optimization time significantly. It’s even practically not feasible due to memory limitations to calculate all the solutions. During the friction optimization two limits are used.

1. Minimum pre displacement of 10 µm
Theoretically it’s feasible to apply a very low pre displacement close to zero. However, bearings and rollers are subjected to tolerances and to ‘push out’ initial clearances a minimum preload is required. If the preload is too close to the tolerance of the initial rollers or raceways, the performance of the different bearings might vary. Furthermore, the higher the initial preload, the more accurately it can be applied, since it can be measured more accurately. It is estimated, that a 10µm preload is enough to overcome this problem during the friction optimization.

2. The pre displacement should always be higher or remain equal at increased torque
There are solutions thinkable that first increase the pre-displacement, then reduce it, and then increase it again, and so on. This means that during acceleration the stiffness is shifting constantly. This is not preferable. These solutions are excluded by excluding all the solutions with decreased pre-displacement with increased torques. The pre-displacements are allowed to remain equal.

6.3 Optimum preload for friction
The maximum reduction of 48.7 % found earlier is the summation of the reductions for each speed-load combination. It’s more interesting to have the reduction for different torques which gives a indication which range is interesting for optimizing the pre displacement. Figure 59 shows these reductions for the NEDC cycle with increments of 15 Nm. There are torques within the NEDC cycle that are higher than 120 Nm but the feasible reduction is there too small (lower than 0.1 %). Therefore, reducing the friction losses is only interesting for torques up to 120 Nm.
A pre-displacement must be assigned to each of the sub range given in Figure 59. Whereby the pre-displacement is limited to a range of 10 to 20 µm to reduce the friction losses. For the optimization increments of 1 µm are used. The results is a 8 x 11 matrix as shown in Figure 60. Each solution is a vector with 8 pre displacements. The coloured elements are examples of possible solutions.

Normally, that would results in $11^8$ combinations which is way out of limits. Applying the second optimization limit given in section 6.2, reduced the number to 44000 combinations and for each combination the potential energy loss reduction is calculated as well as the mean deflection at gear contact, which is correlated to the stiffness of the system. Bearing life is not considered here because for these loads the bearing life is always better if the pre displacement is lower than 20 µm. All the combinations are plotted in Figure 61. An enlargement is appended in Appendix D. The mean deflection is on the x-axis and the reduction on the y-axis. Each blue dot represents a single combination that has a mean deflection and a reduction. The red line indicates the optimal solutions where deflection is minimal for the given reduction. If maximum reduction is required without considering the deflection, the most right point is the best. The result is that the pre displacement remains equal to the minimum required pre displacement of 10 µm, which is obvious. The opposite is zero reduction with minimum deflection, which is equal to constant 20 µm pre displacement. However, it's interesting to examine the shallow part of the red line. Notice that, following the red line from the right to the left, with only 2 % less reduction, the mean deflection can be decreased with 20 %.(The value of reduced deflection is not examined but it is likely that the efficiency of the gears will be improved). Below the 35% line the slope starts to increase and the energy loss reduction quickly becomes less. The 35% line is therefore an interesting point for finding the best solutions. The solutions between 34.5 % and 35.5 % reduction and their corresponding mean deflection values are regarded. This is indicated with the green rectangle.
The curve is not very smooth due to the increments in torque and pre-displacement. To find the solutions for the other loads, a polynomial fit is added to it. The individual points and the polynomial are given in Figure 63. The corresponding equation is:

\[
\delta = -9 \times 10^{-6} \cdot T^3 + 1.8 \times 10^{-3} \cdot T^2 - 8.4 \times 10^{-3} \cdot T + 10 \tag{6.1}
\]

The optimum pre displacement is disastrous for bearing life. Therefore, in practice the initial pre displacement is always lower than optimal to avoid the risk of adding too much pre displacement. This means that the optimum pre displacement found for life is also the optimum for deflection, i.e. higher pre displacements are simply not preferable. In the previous chapter the optimum pre displacement is given as a function of both speed and torque. In the previous section speed is excluded to reduce the complexity of the problem. Therefore, it’s not interesting to use the 3D results found in the previous section here. Instead the average of the optimum pre displacements is taken over the speed range. Because some speeds occur more often during operation, it is preferable to use a weighted average. The customer load cycle, which is used to determine the life characteristics, is very suitable for this purpose. Figure 64 gives the time fraction for each speed range. The fractions are used to convert the pre-displacements given in Figure 46 (section 5.3.1) for speed and torque to new pre displacements independent of the speed. The converting procedure is basically multiplying the vector with time fractions with the matrix with optimum pre-displacements.

The optimum pre displacements independent of the speed are plotted in Figure 65. Two polynomials are used to find the solutions for other loads. The blue polynomial is used for the negative torques and the red polynomial for the positive torques.

The equation for the blue polynomial is:

\[
\delta = -1.04 \times 10^{-4} \cdot T^2 - 0.13 \cdot T + 1.68 \tag{6.2}
\]
The equation for the red polynomial is:

$$\delta = -2.9 \cdot 10^{-10} \cdot T^4 + 7.6 \cdot 10^{-10} \cdot T^3 - 6.8 \cdot 10^{-4} \cdot T^2 + 0.27 \cdot T - 14.9 \quad (6.3)$$

6.5 Optimum preload combined

The combination of the optimum preload for friction and for life is the optimal solution for variable preload. This is shown in Figure 66. The optimum pre-displacement curve is given by the outer line. There are two coupling areas (green) needed to complete the curve. Left from the friction area (red) the optimum pre-displacement for life is lower than 10 µm. Because a minimum of 10 µm is required these points are corrected to 10 µm. Right from the friction area the optimum pre-displacement for life is lower than for friction and stiffness. Optimization limit 2 stated that the curve shouldn’t decrease with increased torque. Therefore the pre-displacement is kept constant up to the moment where it meets the curve for optimum life. The pre-displacement never exceeds 20 µm in the coupling areas. Therefore the life properties are always equal or better than the original case. The curve looks rather strange but that’s caused by the concentration of the NEDC cycle on a very small load area. The curve becomes more continuous when the potential reductions (timeshares) are more evenly spread and over a larger range.

6.6 Linearization

The pre-displacement should be a linear function of the applied torque to meet the specifications of the actuator. There is a distinction here between pre-displacement and preload. Whereby pre-displacement is given in µm and preload in N. Due to the nonlinear bearing stiffness the preload is nonlinearly related to the pre-displacement. In the previous research a linear spring is used in the FEM analysis to simulate the bearing. This resulted into a linear relationship between torque and preload. However, this relationship becomes nonlinear when a nonlinear spring is used. This means that the actuator is linear with the pre-displacement and not with preload as is suggested in the previous research. This behaviour is explained in more detail in section 8.2. The result is that the curve found in the previous section must be used to create the linear curve without converting the values back to Newton. The linear polynomial is given in Figure 67. The best computed fit found started with 13.5 µm at zero torque. However, this would increase the friction dramatically at the most occurring torques. To solve this problem a boundary is added to keep the pre displacement for zero torque 10 µm.

The polynomial is given by:

$$\delta = 0.0327 \cdot T + 10 \quad (6.4)$$

Notice that the pre-displacement is decreasing with negative torques. There are however feasible concepts that keep the pre-displacement constant, when torques get negative. This is preferable.
Therefore, the linear curve is only valid for positive torques and the pre-displacement is 10 µm for negative torques.

![Figure 67 Linear polynomial for pre displacement](image)

### 6.7 Conclusions

Two optimum solutions are developed in this chapter. A theoretical optimum solution and a linear solution corresponding with the specifications of the actuator.

**Theoretical optimum solution.**

Energy loss reduction during the NEDC cycle is only valid for torques between 0 and 100 Nm. This is just a small part of the total load range of the HyPU. Therefore is the load range split into two ranges whereby the first one is optimized for the NEDC cycle with respect to friction reduction and stiffness. The other range is optimized for life and stiffness. Both results are combined into a single curve.

**Linear solution.**

The linear solution is found by fitting a linear polynomial through the theoretical optimum solution.

Both solutions are recalculated for stiffness, life and friction in the next chapter. The linear solution is used in chapter 8 as input for the geometric optimization of the actuator.
7. Analysis of the pinion unit with variable preload

7.1 Introduction
In chapter 6, two curves are developed. The nonlinear curve is the more theoretical solution when the pre displacement is independent of the applied torque. The linear curve is the linear approach of nonlinear curve. The influence of these curves on the friction, bearing life and stiffness is already known to a certain extent. In this chapter, the exact influences are calculated. The friction reduction is calculated again for the NEDC cycle and the life is calculated for the customer load cycle.

7.2 Energy losses in the system
The new power losses are calculated for both curves. This results into two new power loss matrices as given in Figure 41. These are returned into the VEP calculation tool, which gives the new energy losses and CO₂ emissions. The results are given below.

Linear curve:
Energy losses: 34.87 kJ
Energy loss reduction: 34.5 %
Average power loss: 49.6 W
CO₂ emission: 161.43 g/km
CO₂ reduction: 0.08 g/km

Nonlinear curve:
Energy losses: 35.55 kJ
Energy loss reduction: 33.2 %
Average power loss: 50.6 W
CO₂ emission: 161.43 g/km
CO₂ reduction: 0.08 g/km

The reduction differs a bit from the theoretical reduction that is found in the previous chapter i.e. 35.1 %. The differences are caused by the polynomial fits that are used to interpolate the results. Though, the deviation is not very high and in practice it’s even impossible to measure these deflections. The influence on the CO₂ emissions is, as expected, rather low. The main reason for applying variable preload should therefore not be reduction of the CO₂ emissions but rather the reduction of the power losses. These losses can be significantly reduced especially for driving at constant speed. The power losses are given in Figure 68. The difference between the nonlinear and linear curve is very small. The reductions are given for the linear solution.

7.3 System life
The system life is calculated again for the customer load cycle using variable pre-displacement. The curves developed in the previous chapter are only valid for the range of -500 to 1000 Nm. Not all the loads of the load cycle are within this range. The linear curve can just be extended to find the right values for the positive torques. The pre-displacement is constant for negative torques. The nonlinear solution cannot be easily extrapolated because the polynomials are higher order polynomials. The pre-displacement for the nonlinear curve is therefore kept constant outside the boundaries. The results of the life calculation in SKF BB are:

Linear curve
Life ACBB: 1515 h
Life TRB: 6213 h
Life system: 1404 h

Nonlinear curve
Life ACBB: 1515 h
Life TRB: 6233 h
Life system: 1405 h

The differences are very small compared to the original life. This is expected because the bearing life could at the most be increased with two hours, which is already within the error margin of the results. Even the nonlinear curve doesn’t meet this result due to the polynomial interpolation.
However, it’s positive that at least the life is not decreased with respect to the original. Of course, this is for the customer load cycle. In general the bearing life is better at lower loads (0 – 250 Nm).

### 7.4 System stiffness

Finally the system stiffness is checked for both solutions. First the deflection is measured for the highest load condition of the customer load cycle. And in section 7.4.2 the deflection at gear contact is evaluated.

#### 7.4.1 Highest load condition

The deflection at the highest load condition i.e. 1331 Nm is a bit complicated because this point is completely out of the boundaries used for optimization. The linear solution is linear interpolated and the nonlinear solution is kept constant. The resulting pre-displacement for the linear solution is 53.7 µm and for the nonlinear solution 43.5 µm. The difference is caused by the interpolation method. Therefore, the results of the linear solution are a lot more reliable and can be applied for the nonlinear solution as well. The results for both solutions are given in Figure 69. This figure shows the radial deflection of the shaft along the length of the axis. The deflections at the location of the ACBB, TRB and gear contact are given in Table 3. The reductions compared with static pre-displacement are also given.

The deflection at the ACBB is even slightly higher. That’s the result of the new load distribution between the two bearings caused by the pre-displacement. The load on the TRB is therefore much higher and the deflection decreases significantly. The resultant reduced deflection at the gear contact is 10.1 %, which comes close to the maximum feasible reduction given in section 5.4.1, i.e. 11%.

The axial deflections are given in Figure 70 and Table 4. As expected, pre-displacement has more influence on the deflection in axial direction.

#### Table 3 Radial deflections (µm)

<table>
<thead>
<tr>
<th></th>
<th>ACBB</th>
<th>TRB</th>
<th>Gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal</td>
<td>27</td>
<td>58.8</td>
<td>172.5</td>
</tr>
<tr>
<td>Linear</td>
<td>27.9</td>
<td>46.7</td>
<td>155</td>
</tr>
<tr>
<td></td>
<td>3.3 %</td>
<td>-20.6%</td>
<td>-10.1%</td>
</tr>
<tr>
<td>Nonlinear</td>
<td>27.5</td>
<td>50.3</td>
<td>160.3</td>
</tr>
<tr>
<td></td>
<td>1.9 %</td>
<td>-14.5%</td>
<td>-7.1%</td>
</tr>
</tbody>
</table>

#### Table 4 Axial deflections (µm)

<table>
<thead>
<tr>
<th></th>
<th>ACBB</th>
<th>TRB</th>
<th>Gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal</td>
<td>84.6</td>
<td>95.7</td>
<td>98.9</td>
</tr>
<tr>
<td>Linear</td>
<td>67.1</td>
<td>77.1</td>
<td>80.3</td>
</tr>
<tr>
<td></td>
<td>-20.7 %</td>
<td>-19.4 %</td>
<td>-18.8 %</td>
</tr>
<tr>
<td>Nonlinear</td>
<td>72.2</td>
<td>82.5</td>
<td>85.7</td>
</tr>
<tr>
<td></td>
<td>-14.7 %</td>
<td>-13.8 %</td>
<td>-13.4 %</td>
</tr>
</tbody>
</table>
7.4.2 Deflection at gear contact

The deflection at gear contact gives a lot better ground for comparing both solutions because no interpolation is needed. The radial deflection at gear contact is given in Figure 71. The difference between the three curves is very small for low torques. At high torques the deflection is much smaller for the variable solutions compared with the static solution. The difference between the two variable solutions is very small. Larger differences are found for the negative torques. There, the pre-displacement remains constant 10 µm for the linear solutions, which is a lot smaller than the pre-displacement for the nonlinear curve. Though, significant differences are found from torques of -250 Nm and higher, which don’t occur that often in practice.

![Figure 71 Radial deflection at gear contact](image)

The axial deflections are given in Figure 72. The difference between the curves is larger compared with radial deflection. The main differences are found for negative torques. Significant differences are already found at -150 Nm instead of 250 Nm for radial deflections.

![Figure 72 Axial deflection at gear contact](image)

7.5 Conclusions

Both solutions are compared with the reference pre-displacement of 20 µm. The energy loss calculations with VEP show that significant reductions are feasible on the NEDC cycle. 34.5% using the linear solution and 33.2% using the nonlinear solution. Reductions for driving at constant speed are varying between 33% at 120 km/h and 50% at 70 km/h. The influence on the CO₂ emissions are however, very small. Only 0.08 g/km on the total NEDC cycle. As expected, the influence on life is very small. The system life for both, the theoretical solution and linear solution, is not changed considering the error margin in the life prediction method. The stiffness at high torques is significantly improved using the variable solutions, whereby the difference between the two solutions is very small. The deflection at highest load condition i.e. 1331 Nm is reduced with 10.1% in radial direction and 18.8% in axial direction. The deflection at low torques is higher for the variable solutions. Though the absolute difference is rather low compared with the difference at high torques. For negative torques the nonlinear solutions is better than the linear solution.
8. Geometric optimization of the actuator

8.1 Introduction
The ideal linear curve is developed in chapter 6. In chapter 7 it is shown that significant improvements are obtained with this curve. In this chapter the actuator is optimized to meet the given curve. First the actuator is examined in more detail using a nonlinear FEM analysis. The actuator is then optimized to meet the curve requirements with the lowest material stresses. It's not part of the project to do an extensive search to all possible designs. The goal is to come up with one design that is feasible on more accounts.

8.2 Behaviour of the actuator
The behaviour of the actuator is already established in the previous research, described in [4]. Here, it is specified that the preload is linearly related to the applied torque. There is however, a distinction between preload and pre-displacement. Preload is the force that is a function of the pre-displacement and the stiffness of the bearing. The stiffness of the bearing is in the previous study linearized because Pro/Mechanica can only handle linear springs. It's now the question whether the preload – torque relationship is linear or the pre-displacement – torque relationship. To investigate this, a study is done using the Marc solver from MSC. This software package can handle non-linear stiffness of a spring. The FEM model of the actuator in Marc is shown in Figure 73. This model is only used to define the behaviour more exactly. The development of a model that has the preferred torque-pre displacement is developed in the next paragraphs.

In the Marc model, one end is fixed to the ground and the other end fixed with a spring. The torque is increased to 1000 Nm with increments of 20 Nm. Two simulations are performed. The first with a linear spring with a stiffness of 170,000 N/mm. This stiffness is given by fitting a linear polynomial into the graph of Figure 74. The second with a non-linear spring. The displacement-force characteristic of the non-linear spring is equal to the one of the HyPU that is shown in Figure 74.

The results of the analysis are given in Figure 75 and Figure 76. It's interesting to notice that the preload is not linear with the applied moment as was assumed in the previous research. But, pre-displacement is almost linear with the applied moment. This means that for the optimization of the model, the pre-displacement has to be used as reference and not the preload.
8.3 Optimization of the actuator

The optimization of the actuator is performed with the linear FEM program Pro/MEchanica. The model and the analysis are explained in more detail in section 8.3.1. This analysis is the reference for the optimization study in section 8.3.2 and 8.3.3. The results of the optimized model are discussed in detail in section 8.3.4.

8.3.1 FEM model with parameters

Model with dimensions

Figure 77 shows the main dimensions of the actuator. Here the beams are straight for easy dimensioning. Figure 78 shows the same model but twisted with an angle $\alpha$. One end section is fixed and the other end section is rotated with the angle $\alpha$ to deform the beams. These variables are used for the optimization study in section 8.3.2. A last parameter that isn’t showed is the number of beams, which can be changed as well. The thickness of the end sections is 15 mm.

Loads and constraints

The torque is applied at the red coloured surface of the actuator (see Figure 79). A value of 1000 Nm is used in consistency with the value used in the previous chapters. The other side of the actuator is fixed in all directions. The torque causes the actuator to expand, which normally is restricted by the bearing. For the simulation a linear spring is used to simulate this bearing. This spring is connected with a rigid link to the red coloured surface.

Linear spring characteristics

The previous section didn’t only show that the pre-displacement is linear with the applied torque. It also showed that there are two points where the result with a linear spring and with a nonlinear spring is the same. This is at zero torque and at a second point that is determined by the linear spring stiffness. The best results are achieved when the first point is equal to the initial pre-displacement i.e. 10 $\mu$m. And the second point at maximum torque, i.e. 1000 Nm. Then, the results are equal for both simulations at start and end position, which is very useful for comparing both simulations. The result is shown in Figure 80.
Here the first point is at 10 \( \mu \text{m} \) (706 N), the second point is at 42.7 \( \mu \text{m} \) (6500 N). The linear spring stiffness is then given by:

\[
C_{\text{spring}} = \frac{6500 - 706}{0.0427 - 0.01} = 177,200 \frac{N}{\text{mm}}
\]

![Figure 80 Stiffness of the linear spring](image)

**Material properties**
The actuator is made of steel with a Young’s modulus of 200 GPa and a Poisson ratio of 0.27

8.3.2 Optimization study

Optimization is performed with Pro/Mechanica using the optimization tool provided with it. This tool uses Sequential Quadratic Programming (SQP) to optimize the model. It basically searches for solutions that meet the given constraints and minimizes the given goal. The input of the analysis is the goal of the optimization study, the design limits and the limits of the variables used to dimension the geometry.

**Optimization goal**

Material stresses are the greatest challenge with this design. On one hand the actuator must be very stiff to accommodate the loads, on the other hand it should be less stiff to allow torsional deformation. Therefore the optimization goal is to minimize the material stresses in the actuator. The stress that is used is the Von Mises stress. It’s actually not a real stress but more a value that combines the three principal stresses in x, y and z direction to a single value. This value is compared with the yield strength of steel. There are also methods to check the fatigue strength of the design using the VM stress.

**Design limits**
The design limit is given by the curve characteristics given in section 6.6. The initial pre-displacement has very little influence on expansion of the actuator. In other words, if the actuator expands 30 \( \mu \text{m} \) without initial pre-displacement, it will expand 40 \( \mu \text{m} \) with 10 \( \mu \text{m} \) initial pre-displacement. This is very useful because the initial pre-displacement can initially be set to zero, which makes the analysis a lot less complicated. Therefore, the actuator is optimized to increase the pre-displacement from 0 \( \mu \text{m} \) at zero torque and 32.7 \( \mu \text{m} \) (42.7-10) at 1000 Nm.

**Optimization limits**
The optimization limits are given for the variables shown in Figure 77 and Figure 78. The outer and inner diameter of the bearing are used to determine the limits of the outer diameter and thickness of the actuator. The other variables are chosen within a range where the optimum solution should be. The limits of these variables are iteratively calculated. The problem of the optimization limits is that a large range is preferable but some combinations are geometrically not feasible. That’s the reason why the number of beams and the radii are not taken into account initially. Instead, first the optimal design is established for different number of beams with radii of 2 mm. Then, the best number of beams is chosen. Finally, the radii are increased until the best solution is found.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Min</th>
<th>Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>Twist angle ( \alpha )</td>
<td>100°</td>
<td>200°</td>
</tr>
<tr>
<td>Outer diameter Do</td>
<td>60 mm</td>
<td>80 mm</td>
</tr>
<tr>
<td>Thickness S</td>
<td>2 mm</td>
<td>20 mm</td>
</tr>
<tr>
<td>Beam length L</td>
<td>30 mm</td>
<td>80 mm</td>
</tr>
<tr>
<td>Beam thickness Sd</td>
<td>30°</td>
<td>80°</td>
</tr>
</tbody>
</table>

*Table 5 Optimization limits*

8.3.3 Optimization results

**Best number of beams**
The influence of the number of beams is investigated. For each number of beams, the optimal solution is found within the limits given in the previous section. The results are given in Table 6. It’s interesting to notice that the Von Mises stress isn’t influenced anymore with using more than three beams. The mass however is increasing with more beams. The reason is that more beams results automatically in lesser thickness. This results in higher material stresses. The best number of beams is three, which is also likely the best option for manufacturing purposes.

<table>
<thead>
<tr>
<th>Nr. Beams</th>
<th>Max VM stress (Mpa)</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>302</td>
<td>1.09</td>
</tr>
<tr>
<td>3</td>
<td>259</td>
<td>1.54</td>
</tr>
<tr>
<td>4</td>
<td>259</td>
<td>1.60</td>
</tr>
<tr>
<td>5</td>
<td>259</td>
<td>1.76</td>
</tr>
<tr>
<td>6</td>
<td>259</td>
<td>1.67</td>
</tr>
</tbody>
</table>

*Table 6 Influence of the number of beams*
Optimized design
The best design with radii of 2 mm and three beams has a maximum Von Mises stress of 259 MPa. The coloured stress plot given in Appendix E show that this maximum occurs in radius 1. In iteration 2 the radius is increased to 4 mm and again optimized to meet the given design limits. The maximum Von Mises stress is reduced to 195 MPa. Due to the change in geometry it’s feasible to increase the radius again up to 5 mm. Optimizing the actuator again results in a maximum Von Mises stress of 183 MPa. In iteration 4 the radius increased to 6.5 mm. The Von Mises stress is further reduced to 170 MPa. The geometry doesn’t allow larger radii anymore so the optimum design is achieved. Furthermore, the stresses on other locations as the radii are increasing rapidly and coming close to the maximum stress in the radii. Due to the manual optimization steps involved with the process it’s feasible that there are better solutions. Though, the optimization showed that there are solutions that meet the design limits and within the range of acceptable stresses. The design parameters of the optimal design are:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of beams</td>
<td>3</td>
</tr>
<tr>
<td>Twist angle α</td>
<td>186.8°</td>
</tr>
<tr>
<td>Outer diameter Do</td>
<td>80 mm</td>
</tr>
<tr>
<td>Thickness S</td>
<td>20 mm</td>
</tr>
<tr>
<td>Beam length L</td>
<td>41.5 mm</td>
</tr>
<tr>
<td>Beam thickness Sd</td>
<td>62.9°</td>
</tr>
<tr>
<td>Radius 1</td>
<td>6.5 mm</td>
</tr>
<tr>
<td>Radius 2</td>
<td>2 mm</td>
</tr>
</tbody>
</table>

Table 7 Optimum design parameters

8.3.4 Final check of the optimized design
The optimization study didn’t take into account the initial pre-displacement of 10 µm. A final analysis is performed with an initial pre-displacement of 10 µm. The results are given in Figure 81. The preload at 1000 Nm is 6533 N, which is very close to the required 6500 N. The maximum Von Mises stress is 169 MPa. It’s interesting to notice that it’s even lower than without pre-displacement. Though, the difference is very small and likely due to the stress error in the results. The yield strength of basic construction steel is according to [15] between 185 and 360 MPa. It’s good to have a margin here because the actuator must resist higher torques as well. The maximum stress with 1500 Nm torque applied is 255 MPa. This is still within the range of allowable stresses using the yield strength. For fatigue however it’s rather high. According to a rule of thumb, the maximum stress for fatigue must be lower than 50% of the ultimate strength. The ultimate strength for construction steel is between 235 and 510 MPa. It’s recommended to investigate the fatigue in more detail.

![Figure 81 Von Mises stress in the optimal design at 1000 Nm](image1)

A final check is performed using MSC Marc with a nonlinear spring. The Von Mises stress are given in Figure 82. The rounded maximum Von Mises stress is equal to the stress found with the linear spring, i.e. 169 MPa.

![Figure 82 Von Mises stress in the optimal design using a nonlinear spring](image2)

More interesting is the pre displacement as function of the torque. This is shown in Figure 83. Here, an initial pre-displacement of 10 µm is applied. The actual pre-displacement is a bit lower due to the contraction of the actuator. The pre-displacement at 1000 Nm is 41.7 µm. This close enough to the required 42.7 µm considering the potential errors due to simulation features. Finally the mass of the actuator is measured. The total mass of the optimized actuator is 1.65 kg.
8.4 Conclusions
In this chapter the design of the actuator is optimized to meet the curve characteristics established in section 6.6. The best design that is found consists of three beams. The size of the actuator is Ø 80 x 71.5 mm using end section thickness of 15 mm. The maximum VM stress in the model with 1000 Nm torque applied is 169 Mpa. The maximum VM stress with 1500 Nm torque applied is 255 Mpa. The range of allowable stress is approximately 185 – 360 MPa based on the yield strength of construction steel. According to a rule of thumb the maximum stress for fatigue must be lower than 50% of the ultimate strength. The ultimate strength for construction steel is between 235 and 600 MPa. It’s recommended to investigate the fatigue in more detail. The mass of the actuator is 1.65 kg.
9. Conclusions and recommendations

9.1 Conclusions
The effects and potential benefits of using variable preload on a pinion unit of a rear axle differential are investigated in this thesis. The reference is an actuator found in a previous research [4]. This actuator needs to be optimized to meet maximum friction reduction with acceptable bearing life and stiffness.

First, the influence of preload on general bearing units is investigated. It showed that only bearing life has one specific pre-displacement where the bearing life is optimal. The location of this optimal point is dependent on the load applied. Bearing stiffness has no absolute optimum but significant stiffness improvements are feasible with only a slight amount of pre-displacement. Some stiffness is required for accurate gear mesh. The friction in the bearing increases when more pre-displacement is applied. Low friction is therefore only feasible by lowering the pre-displacement.

The reference case for comparing different solutions is the SKF Hybrid Pinion Unit. Before optimizing the variable preload, the first ultimate achievements are calculated for each individual parameter by using the ideal pre-displacement for each parameter neglecting the other parameters. The results of this theoretical study showed that the energy losses of the bearing during the NEDC cycle can be lowered with 48.7% (from 53.2 kJ to 27.3 kJ), though in practice the reduction will be less favourable. The maximum CO2 reduction is 0.11 g/km. The system life is evaluated for a customer load cycle. This load cycle is used by a customer for the acceptance tests on the Hybrid Pinion Unit. The results showed that the system life could only be increased from 1404 hours to 1406 hours. This is within the error margin of the results and can be considered as unchanged. The stiffness is evaluated by measuring the deflection of the shaft at gear contact using the highest load condition in the customer load cycle (1331 Nm). The results showed that the deflection could be lowered with 11% in radial direction and 20% in axial direction.

The optimum curve for variable preload as function of torque is developed in two different stages. First, a theoretical optimum solution is found whereby the pre-displacement is independent of the applied torque. The second solution is derived from the first solution by curve fitting a linear polynomial. This is the more practical solution that meets the required linear relationship of the given actuator. The solutions are optimized for maximum energy loss reduction for the NEDC cycle, maximum life for the customer load cycle and minimum deflection at gear contact. The resulting energy loss reduction is 33.2% for the theoretical solution and 34.5% for the linear solution. The CO2 reduction for both solutions is 0.08 g/km. The system life for both, the theoretical solution and linear solution, is not changed considering the error margin in the life prediction method. The deflection at gear contact for both solutions is reduced with 10.1% in radial direction and 18.8% in axial direction.

Finally, the design of the actuator is optimized to meet the curve characteristics with minimum amount of material stresses. The best design found has a maximum VM stress of 169 Mpa with 1000 Nm torque applied. The maximum VM stress with 1500 Nm torque applied is 255 Mpa. The range of allowable stress is approximately 185 – 360 MPa based on the yield strength of construction steel. The mass of the actuator is 1.65 kg.

9.2 Recommendations
The stiffness of the system is evaluated using the deflection at gear contact. Another interesting feature related to the stiffness is vibration. It’s expected that the influence of preload on vibration is limited to a certain amount of pre displacement. This can be used as additional input for determining the minimum required pre-displacement.

The effects of bearing preload are calculated for the bearings only. Influence on gear mesh efficiency and other parts of the differential are excluded. This is worth researching because the efficiency of the differential could be increased due to the lower deflection at higher load conditions. This could be an additional benefit of variable preload.

The Hybrid Pinion Unit is used as reference for this research. The pre-displacement used for this system is rather low compared with a general pinion unit with 2 Tapered Roller Bearings (20 µm instead of 40 µm). It’s expected that the potential benefits are larger for a general pinion unit than for a HyPU due to the higher preload.

In this research the NEDC cycle is used for calculating the energy losses in the system. The loads in this cycle are very concentrated on a small torque area. It’s interesting for further analysis to use another load cycle as well. For example the FTP cycle that is used in the USA for emission testing.

The optimized design of the actuator found in this research proved that the variable preload is feasible with the current actuator. Though, a more detailed optimization study is recommended taking the manufacturability into account as well.

The Von Mises stress in the optimized actuator are within the range of allowable stresses given the yield strength of construction steel. The fatigue limit however, is lower and should be investigated in more detail.
References


[16] SKF homepage. Cited 12-02-2010
   URL: <http://www.skf.com/portal/skf/home/products?maincatalogue=1&lang=en&newlink=1_0_37>
List of figures

Figures
Figure 1 Progress in fuel efficiency. Source ACEA ......................................................... 5
Figure 2 Potential for CO₂ reduction. Source SKF ............................................................ 5
Figure 3 Torque activated torsion tube ............................................................................. 5
Figure 4 Actuation principle applied on the pinion unit ..................................................... 6
Figure 5 General inputs and outputs during the project ...................................................... 6
Figure 6 Rear axle differential ........................................................................................... 8
Figure 7 HyPU unit ............................................................................................................ 8
Figure 8 Gear geometry of a differential .......................................................................... 9
Figure 9 Force components resulting from the tooth contact ............................................ 9
Figure 10 Rigidity requirements (Fenton, 1998) ................................................................. 10
Figure 11 Dynamic model of a grinding system ................................................................. 10
Figure 12 ACBB dimensions (Harris and Kotzales, 2007) .................................................. 12
Figure 13 TRB dimensions and terminology (Harris and Kotzales, 2007) ......................... 12
Figure 14 Axial deflection of a single ACBB subjected to an axial load ......................... 13
Figure 15 Axial stiffness of an ACBB subjected to an axial load ...................................... 13
Figure 16 Axial deflection of a TRB subjected to an axial load ......................................... 14
Figure 17 Axial stiffness of a TRB subjected to an axial load ............................................ 14
Figure 18 Loading conditions duplex bearing arrangement .............................................. 14
Figure 19 Reaction forces as function of the axial load ...................................................... 14
Figure 20 deflection of a duplex bearing arrangement with 5000 N preload .................... 15
Figure 21 Deflection for different amounts of preload ...................................................... 15
Figure 22 Duplex bearing stiffness as function of preload ............................................... 15
Figure 23 Rolling element load distribution for different amounts of clearance (Harris & Kotzales, 2007) 15
Figure 24 Analysis definitions in SKF Bearing Beacon ..................................................... 16
Figure 25 life of an ACBB for 7, 8 and 16 kN radial load ................................................... 17
Figure 26 TRB life ............................................................................................................. 17
Figure 27 duplex life (no moment) ................................................................................... 17
Figure 28 duplex life (with moment) ................................................................................ 17
Figure 29 Frictional moments of an ACBB as function of radial and axial load ............... 17
Figure 30 Frictional moment of a TRB as function of radial and axial loading ................ 18
Figure 31 New reaction forces with variable preload ...................................................... 18
Figure 32 change in frictional moment of the ACBB ....................................................... 18
Figure 33 Change in frictional moment of the TRB ........................................................... 19
Figure 34 Time share for each load case ......................................................................... 20
Figure 35 Power losses for the HyPU unit with 20 µm preload ....................................... 21
Figure 36 Power losses at different speeds ...................................................................... 21
Figure 37 Energy losses in the HyPU unit ......................................................................... 21
Figure 38 Life results of the HyPU with normal preload .................................................. 22
Figure 39 Shaft deflection for 20 µm preload .................................................................. 22
Figure 40 Deflection at gear contact as function of the applied torque ............................. 23
Figure 41 Power losses for zero preload .......................................................................... 24
Figure 42 Power loss reduction ....................................................................................... 24
Figure 43 Power loss reduction at maximum and minimum speed ................................... 24
Figure 44 Power losses at different speeds for normal and zero preload ........................ 25
Figure 45 Energy losses with zero preload ...................................................................... 25
Figure 46 Optimum preload values for life (3D bar plot) ................................................... 26
Figure 47 Sensitivity for larger pre displacements than the optimum pre displacement ...... 26
Figure 48 Sensitivity for lower preloads than optimum preload ....................................... 26
Figure 49 optimum system life for each load case ............................................................. 26
Figure 50 Life of the ACBB for each load case ................................................................. 27
Figure 51 Life of the TRB for each load case .................................................................... 27
Figure 52 Radial displacement along the length of the shaft ............................................. 28
Figure 53 Axial displacement along the length of the shaft ............................................... 28
Figure 54 Radial displacement reductions as function of the preload ............................. 28
Figure 55 Axial displacement reductions as functions of the preload ............................. 28
Tables

Table 1 Comparison between a grinding spindle and a pinion unit ................................................. 10
Table 2 Values for friction reduction at low torques ................................................................. 24
Table 3 Radial deflections (µm) ................................................................................................. 36
Table 4 Axial deflections (µm) .................................................................................................. 36
Table 5 Optimization limits ......................................................................................................... 40
Table 6 Influence of the number of beams .................................................................................. 40
Table 7 Optimum design parameters ......................................................................................... 41
Table 8 load cycle data from customer ..................................................................................... 49
## Appendix A

### List of symbols and abbreviations

#### A1. Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Quantity</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>F</td>
<td>Force</td>
<td>N</td>
</tr>
<tr>
<td>T</td>
<td>Torque</td>
<td>Nm</td>
</tr>
<tr>
<td>n</td>
<td>Speed</td>
<td>1/min</td>
</tr>
<tr>
<td>ω</td>
<td>Speed</td>
<td>rad/s</td>
</tr>
<tr>
<td>P</td>
<td>Power</td>
<td>W</td>
</tr>
<tr>
<td>δ</td>
<td>Deflection</td>
<td>μm</td>
</tr>
</tbody>
</table>

**Subscript**

- a: Axial
- r: Radial
- t: Tangential

#### A2. Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACBB</td>
<td>Angular Contact Ball Bearing</td>
</tr>
<tr>
<td>HyPU</td>
<td>Hybrid Pinion Unit</td>
</tr>
<tr>
<td>NEDC</td>
<td>New European Driving Cycle</td>
</tr>
<tr>
<td>SKF BB</td>
<td>SKF Bearing Beacon (simulation program for calculating bearing systems)</td>
</tr>
<tr>
<td>TRB</td>
<td>Tapered Roller Bearing</td>
</tr>
<tr>
<td>VEP</td>
<td>Vehicle Environmental Performance (simulation program for calculating CO₂ emissions)</td>
</tr>
</tbody>
</table>
Appendix B

Bearing Beacon model with parameters

Introduction
SKF bearing beacon is the new mainstream bearing application program used by SKF engineers to find the best solution for customers' bearing arrangements. The program is the successor of BEACON and its technology allows the modelling in a 3D graphic environment of flexible systems incorporating customer components. SKF bearing beacon combines the ability to model generic mechanical systems (using also shafts, gears, housings etc.) with a precise bearing model for an in-depth analysis of the system behaviour in a virtual environment. It also performs bearing rolling fatigue evaluation using the SKF rating life in particular. SKF bearing beacon is the result of several years of specific research and development within SKF.

SKF Bearing Beacon model of the Pinion Unit
Figure 84 shows the main data conditions that are used in the BB model of the pinion unit. Here, the HyPU is displayed but the model can be used for other bearing arrangements as well. The moment in the BB model is converted to three reaction forces at the tooth-tooth contact point with the equations given in section 2.3.2. The outer rings are fixed for all directions, which assumes that the housing is very rigid. For now, it's acceptable but keep in mind that in real this is not the case, especially not for the newer housings made of aluminium.

The following internal settings are used for the bearings:
- No clearance reduction caused by temperature, ring speeds or interference fits
- Lubricant: SAF-AG4
  - Viscosity at 40 °C: 113 mm²/s
  - Viscosity at 100 °C: 17.2 mm²/s
  - ηc = 0.7 (cleanliness factor)
- Inner and outer ring temperature: 100 °C

Figure 84 SKF BB model parameters
Loadcases
A customer provided load cycle data for the design of the HyPU. The same data is used for the research on variable preload. The data consists of 27 load cases from where three load cases are representing reverse driving. The data is shown in Table 8 and Figure 85.

<table>
<thead>
<tr>
<th>Load case</th>
<th>Moment</th>
<th>Speed</th>
<th>Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-1</td>
<td>1065</td>
<td>460</td>
<td>3</td>
</tr>
<tr>
<td>1-2</td>
<td>849</td>
<td>1013</td>
<td>0.2</td>
</tr>
<tr>
<td>1-3</td>
<td>1331</td>
<td>460</td>
<td>1</td>
</tr>
<tr>
<td>2-1</td>
<td>704</td>
<td>691</td>
<td>8</td>
</tr>
<tr>
<td>2-2</td>
<td>561</td>
<td>1519</td>
<td>1</td>
</tr>
<tr>
<td>2-3</td>
<td>880</td>
<td>691</td>
<td>3</td>
</tr>
<tr>
<td>3-1</td>
<td>474</td>
<td>1031</td>
<td>12</td>
</tr>
<tr>
<td>3-2</td>
<td>378</td>
<td>2267</td>
<td>1</td>
</tr>
<tr>
<td>3-3</td>
<td>593</td>
<td>1031</td>
<td>9</td>
</tr>
<tr>
<td>4-1</td>
<td>375</td>
<td>1302</td>
<td>9</td>
</tr>
<tr>
<td>4-2</td>
<td>299</td>
<td>2864</td>
<td>2</td>
</tr>
<tr>
<td>4-3</td>
<td>469</td>
<td>1302</td>
<td>16</td>
</tr>
<tr>
<td>5-1</td>
<td>290</td>
<td>1689</td>
<td>10</td>
</tr>
<tr>
<td>5-2</td>
<td>231</td>
<td>3716</td>
<td>3</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Load case</th>
<th>Moment</th>
<th>Speed</th>
<th>Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>5-3</td>
<td>363</td>
<td>1689</td>
<td>22</td>
</tr>
<tr>
<td>6-1</td>
<td>228</td>
<td>2170</td>
<td>11</td>
</tr>
<tr>
<td>6-2</td>
<td>182</td>
<td>4775</td>
<td>3</td>
</tr>
<tr>
<td>6-3</td>
<td>285</td>
<td>2170</td>
<td>29</td>
</tr>
<tr>
<td>7-1</td>
<td>189</td>
<td>2587</td>
<td>9</td>
</tr>
<tr>
<td>7-2</td>
<td>176</td>
<td>5184</td>
<td>2</td>
</tr>
<tr>
<td>7-3</td>
<td>237</td>
<td>2587</td>
<td>34</td>
</tr>
<tr>
<td>8-1</td>
<td>151</td>
<td>3254</td>
<td>26</td>
</tr>
<tr>
<td>8-2</td>
<td>171</td>
<td>5183</td>
<td>2</td>
</tr>
<tr>
<td>8-3</td>
<td>189</td>
<td>3254</td>
<td>31</td>
</tr>
<tr>
<td>R-1</td>
<td>-590</td>
<td>-818</td>
<td>0.21</td>
</tr>
<tr>
<td>R-2</td>
<td>-590</td>
<td>-818</td>
<td>0.02</td>
</tr>
<tr>
<td>R-3</td>
<td>-738</td>
<td>-818</td>
<td>0.04</td>
</tr>
</tbody>
</table>

Table 8 load cycle data from customer

Figure 85 Load cycle data from a customer
Appendix C

Workflow for calculating energy losses with VEP

Introduction to VEP (Vehicle Environmental Performance)
VEP is based on a Matlab/Simulink model, which includes a driver model for throttle and brake controls, an engine map, a vehicle model and a driveline model. All the models consist of predefined parameters depending on the components configurations. The model is designed as a closed loop, which means that the output is compared with the input. The input signal is then automatically corrected. The closed loop feature is used to follow the driving cycle that is given as input. Any cycle can be used, for the comparison of the different pinion unit configurations the NEDC cycle is used.

Based on the actual internal bearing geometries, the vehicle parameters (like mass, air drag coefficient, etc.) and the engine map the software VEP calculates the demand on power and thus the fuel consumption of a vehicle for the given driving cycle. Once the engine power required for a specific load case is known, the torque with the resulting gear mesh forces and the associated speed depending on the gear ratio can be deduced for each component of the driveline. Before we are able to calculate the CO2 emission of the vehicle, it is necessary to build up a model of each driveline component and its bearings. Loaded with the torque and speed evaluated before by VEP, it is possible to analyse precisely the friction torques of any bearing eventually contributing to CO2 emission. Multiplying the friction torque with bearing’s speed results into the corresponding power loss.

![VEP Diagram](image)

**NEDC driving cycle**
Test cycles consisting of standardized speeds and elevation profiles have been introduced to compare the pollutant emissions of different vehicles on the same basis. And it is used for comparing the fuel economy of different cars. The test is performed on a chassis dynamometer in controlled environments where a test driver controls the vehicle. There are several common used drive cycles. In the United States, the federal urban driving cycle (FUDS) represents a typical city driving cycle, while the federal highway driving cycle (FHDS) reflects the extra-urban driving conditions. In Europe, the urban driving cycle (ECE) consists of three start-and-stop manoeuvres. The combined cycle proposed by the motor and vehicle expert group in 1995 (MVEG-95) repeats the ECE four times and adds an extra-urban portion.
referred to as the EUDC. The MVEG cycle is also referred to the New European Driving Cycle (NEDC). Figure 87 show the speed profile for the European cycle.

Typical measurements performed during the NEDC are:
- Urban fuel economy (first 800 seconds)
- Extra-urban fuel economy (800 to 1200 seconds)
- Overall fuel economy (complete cycle)
- CO₂ emission (complete cycle)
- Emission of CO, HC, NOₓ and fine particles (complete cycle)

![Figure 87: New European Driving Cycle](image)

**Vehicle parameters**
The current models in VEP are only for Front Wheel driven vehicles. These vehicles have the differential integrated into the gearbox and don’t consist of a hypoid gear but with a spur gear. The spur gear doesn’t have a pinion unit and so it cannot be used for this study. Developing a new model of a rear wheel driven car in VEP is not an option for this project. Instead, is the pinion unit integrated as an additional component in the driveline of the front wheel drive. This means that the energy losses are slightly higher but the influence of decreasing the friction in the pinion unit can still be investigated. The car that is used is the GM Insignia petrol 1.6 with the GBX M32 series gearbox. The final drive ratio of this car is 3.9:1.

**Workflow for calculating the energy losses in the pinion unit**
The research on variable preload requires many different configurations. And every different configuration has a different friction map. Calculating the energy losses for each of this configuration with VEP can be quite time consuming. A method is developed that uses the output from VEP for one particular configuration. All the other configurations can then be calculated based on this specific output. There are some minor errors in the results, because the calculation is not using a closed loop anymore. However, for quick comparisons is this method ideal. The workflow of this method is explained using the picture given on page 45:
Output from VEP: VEP is used to calculate the input torques and speeds for the differential. These signals are for the complete cycle included the braking parts. There are energy losses during braking but that don’t result in additional energy consumption by the engine. Therefore, the torques and speeds where the vehicle is braking are set to zero. This is referred as masking the signals. The new signals are only positive torques and speeds.

Binning process
Calculating the energy losses for every speed-torque combination is not an option because there are about 5900 combinations. Instead, the torque and speed range is divided into a predefined amount of bins. Each torque-speed combination is now assigned to a certain bin. The time of all these separate combinations are summed up resulting into the total time for each bin.

Power losses
The power losses are calculated for each bin by using the average torque and speed of the combinations that are assigned to that bin. The power losses are calculated using SKF BB.

Energy losses
The energy losses are calculated by multiplying the time shares for each bin with the power losses for each bin. Due to the binning process is a slight error in the results but that is well compensate by the time that is saved with this method.

CO₂ emissions
The CO₂ emissions are calculated for three different configurations in VEP.
1) Zero energy losses in the pinion unit
2) Energy losses in the pinion unit with 20 μm preload
3) Energy losses in the pinion unit with 50 μm preload
The energy losses for the pinion unit with variable preload should be somewhere between configuration 1 and 2. Instead of using the VEP tool again, the results are linear interpolated that results in a linear function to calculate the CO₂ emissions based on the energy losses in the HyPU. The equation is:

\[ CO₂ = 0.0043 \cdot E_{\text{loss}} + 161.28 \]  

\[ \text{[g/km]} \]  

(6.5)
Appendix D

Magnified figures
Appendix E

Optimization results

Legend

<table>
<thead>
<tr>
<th>Color</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Red</td>
<td>2.400e+02</td>
</tr>
<tr>
<td>Orange</td>
<td>2.160e+02</td>
</tr>
<tr>
<td>Yellow</td>
<td>1.920e+02</td>
</tr>
<tr>
<td>Green</td>
<td>1.680e+02</td>
</tr>
<tr>
<td>Light Green</td>
<td>1.440e+02</td>
</tr>
<tr>
<td>Light Blue</td>
<td>1.200e+02</td>
</tr>
<tr>
<td>Blue</td>
<td>9.600e+01</td>
</tr>
<tr>
<td>Purple</td>
<td>7.200e+01</td>
</tr>
<tr>
<td>Dark Blue</td>
<td>4.800e+01</td>
</tr>
<tr>
<td>Dark Green</td>
<td>2.400e+01</td>
</tr>
</tbody>
</table>

Iteration 1

Iteration 2

Iteration 3

Iteration 4
Appendix F

Project description

Project Title
Variable Preloaded Pinion Unit (V2PU)

Distribution
R. Breuker, V. Haans, C. Romijn, C. Vissers

Created
Constantijn Romijn 20-02-2010 Nieuwegein

Background
Environmental protection plays an important role today. The automobile industry in particular is constantly criticized when it comes to pollutants emissions. Not only the automotive industry but also the automotive suppliers are committed to deliver more energy efficient solutions. The rolling bearing industry in particular is committed to contribute towards a reduction of CO$_2$ emission by improving the efficiency rate of vehicle drive lines. SKF has launched a special Environmental Vehicle program (ENVI) dedicated to the development of energy efficient solutions. Part of the program is the development of a variable preload system for bearing units to reduce friction losses. The actuator for this project is developed in a previous stage at SKF Automotive Development Centre. The next stage is to investigate the influences and potential benefits of applying such a system on a bearing unit. The pinion bearing unit of a rear axle differential is used as case study.

Project definition
- Perform an analytical study on the influence of bearing preload on bearing life, stiffness and friction
- Investigate the potential benefits of variable preload on the pinion unit
- Develop the optimal variable preload algorithm for the Hybrid Pinion Unit minimizing the friction losses in the bearing,
- Optimize the given actuator with respect to the optimum preload algorithm.

Not part of the project is
- modifying the geometry of the bearing or differential
- investigation of the influence on gear mesh efficiency
- investigation of the influence of friction losses on heat generation.

Objectives
To optimize a given passive variable preload actuator with respect to the friction losses and to consider the potential benefits for application in a differential of a rear- or four wheel drive vehicle.

Deliverables
Report with a(n)
- detailed description of the bearing unit and location in the rear axle differential
- analytical study of the Hybrid Pinion Unit with and without variable preload
- optimal design for the given actuator

Critical factors
- Access to:
  - PTC Wildfire Pro/Engineer 4.0
  - PTC Wildfire Pro/Mechanica 4.0
  - MS Word
  - MS Excel
  - SKF Bearing Beacon 2.0
  - Vehicle Environmental Performance tool
- MathWorks Matlab
- MSC Marc
  - Project must be finished before the 26th of Mai
  - Availability of Advanced Engineering for support on Vehicle Environmental Performance (VEP).

**Project organization**

Project Leader: I. Dorrestijn  
Student: C. Romijn  
Mentor: V. Haans, C. Vissers  
VEP simulation tool: N. van der Mei, V. Ondrak  
Support on the HyPU: T. Wolf

**Critical milestones**

Finished literature study: before the 19th of February  
Basic analyses of the pinion unit: before the 8th of March  
Study on the effects of preload: before the 23th of March  
Optimized passive preload actuator: before the 7th of Mai  
Final report: before the 26th of Mai