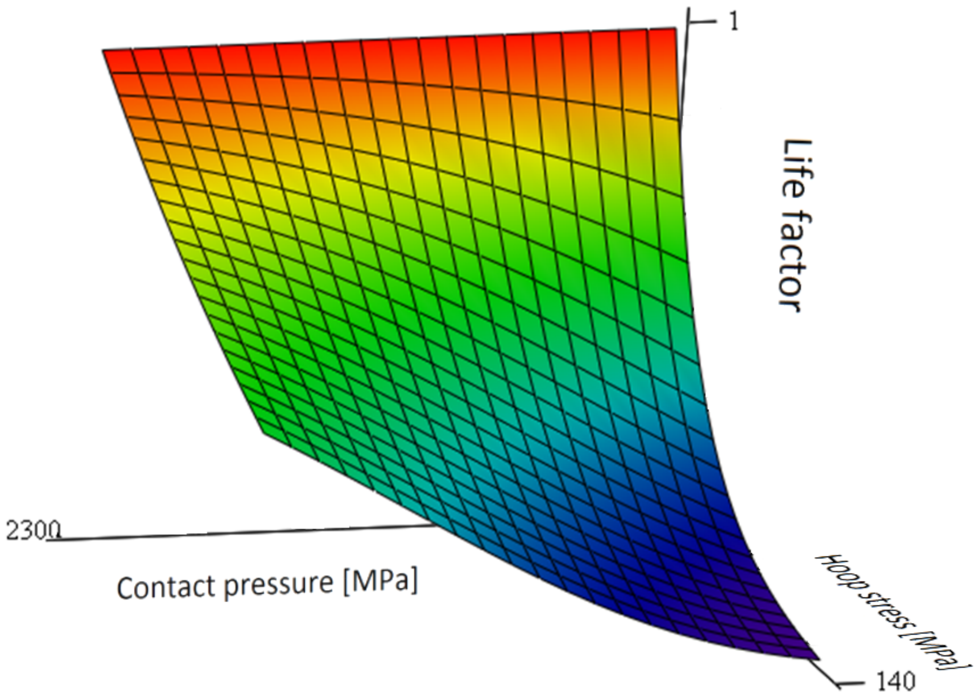

Master Thesis

Effect of Interference Fit on Rolling Bearing Performance

A Master Thesis in Product Development

Axel Qvick
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Supervisor: Helena Jacobsson Sandlund PhD, Department of Product Development of Self-aligning Roller Bearings, SKF
Examiner: Magnus Bengtsson Associate Professor, Department of Industrial and Materials Science, Chalmers

Department of Industrial and Materials Science
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2018

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Abstract

One may think that bearings cannot be developed more to exceed the performance of its precursors. In fact, the opposite is true. A lot of detailed development can be done to make the bearings more efficient. A great benefit in an efficient development phase is a good simulation tool, describing a certain behaviour to avoid expensive and long tests of bearings. With this as a foundation, this project was initiated to investigate whether it was possible to create a meta model of the hoop stress effect over the whole assortment of SKF's spherical roller bearings. In addition to this, it was of interest to investigate what parameters are affecting the hoop stress and in what degree they influence. When a bearing has been mounted onto a shaft, structural stress will be present in the inner ring. The circumferential stress component of these structural stresses in the inner ring is often called hoop stress. Observations have been done regarding a decrease of the life time due to hoop stresses. Hence, further investigations had to be done in order to quantify the properties of the phenomenon. This project did thus investigate and evaluate the effect of the hoop stress when such have been inflicted due to interference fit. The effect was then described by a created meta model to speed up the evaluation process compared to perform a longer simulation. The meta model describes the life time factor in regard to the hoop stress and the contact pressure, as a scientific model. An additional level of the meta model is the engineering model, describing the life time factor in regard to the interference and the load. An evaluation of the software, which was used to create the data for the meta model, was done along the way to assure the validity of the data and the model itself. The parameters affecting the hoop stress was identified early in the project as the interference, Δ, the ratio between the inner and outer diameter of the inner ring, k, and the material through Young's modulus. The final result of the project showed that it was possible to create a common meta model for the so called scientific model over the whole assortment of spherical roller bearings from the given data based on the life time model. Regarding the engineering model, a common model could not be created since no significant relation was identified. When evaluating the two models regarding the hoop stress effect, no difference between the bearings was seen in the scientific level meanwhile for the engineering level, it was realised that the most affected bearings were the ones with larger diameter and thinner inner ring.

Keywords: spherical roller bearing, interference fit, hoop stress, meta model, parameter study, performance model, product development.

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Axel Qvick and Axel Werner, Gothenburg, 1st of June 2018

Nomenclature

$A = Plane\ stress\ constant$ $[-]$
$B = Plane\ stress\ constant$ $[-]$
$a = Curve\ fit\ constant$ $[-]$
$b = Curve\ fit\ constant$ $[-]$
$c = Curve\ fit\ constant$ $[-]$
$d = Curve\ fit\ constant$ $[-]$
$C = Bearing\ dynamic\ load\ rating$ $[N]$
$D_w = Roller\ diameter$ $[m]$
$d = Bore\ diameter\ of\ the\ bearing$ $[m]$
$d_{ir,is} = Inner\ ring\ diameter, inner\ surface$ $[m]$
$d_{ir,os} = Inner\ ring\ diameter, outer\ surface$ $[m]$
$d_s = Shaft\ diameter$ $[m]$
$E = Young's\ modulus$ $[Pa]$
$E_{ir} = Young's\ modulus\ for\ the\ inner\ ring$ $[Pa]$
$E_s = Young's\ modulus\ for\ the\ shaft$ $[Pa]$
$I = Integral$ $[-]$
$F_a = Axial\ bearing\ load$ $[N]$
$F_r = Radial\ bearing\ load$ $[N]$
$k = Ratio\ between\ d_{ir,is} / d_{ir,os}$ $[-]$
$T = Temperature$ $[-]$
$u = Deflection$ $[m]$
$u_{ir} = Deflection\ of\ inner\ ring$ $[m]$
$u_s = Deflection\ of\ shaft$ $[m]$
$l_a = Effective\ roller\ length$ $[m]$
$P = Equivalent\ dynamic\ bearing\ load$ $[N]$
$p = Pressure\ between\ shaft\ and\ inner\ ring$ $[Pa]$
$p_{\max} = Maximum\ contact\ pressure\ between\ rollers\ and\ ring$ $[Pa]$
$X = Radial\ load\ factor\ for\ the\ bearing$ $[-]$
$Y = Axial\ load\ factor\ for\ the\ bearing$ $[-]$
$z = Number\ of\ rolling\ elements\ per\ row$ $[-]$
$\alpha = Coefficient\ of\ thermal\ expansion$ $[-]$
$\Delta = Interference$ $[m]$
$\nu = Poisson's\ number$ $[-]$
$\nu_{ir} = Poisson's\ number\ for\ the\ inner\ ring$ $[-]$
$\nu_s = Poisson's\ number\ for\ the\ shaft$ $[-]$

$\rho = Density$	$[kg/m^3]$
$\omega = Rotational\ \ velocity$	$[rad/s]$
$\sigma_r = Radial\ stress$	$[Pa]$
$\sigma_{r,ir} = Radial\ stress, inner\ ring$	$[Pa]$
$\sigma_{r,s} = Radial\ stress, shaft$	$[Pa]$
$\sigma_\varphi = Circumferential\ stress$	$[Pa]$
$\sigma_{\varphi,ir} = Circumferential\ stress, inner\ ring$	$[Pa]$
$\sigma_{\varphi,s} = Circumferential\ stress, shaft$	$[Pa]$
$\sigma_z = Axial\ stress$	$[Pa]$
$\sigma_{z,ir} = Axial\ stress, inner\ ring$	$[Pa]$
$\sigma_{z,s} = Axial\ stress, shaft$	$[Pa]$
$\sigma_i = Normal\ distribution$	$[-]$
$\chi = Least\ square$	$[-]$
$x_i = Independent\ variable$	$[-]$
$y_i = Dependent\ variable$	$[-]$
$y = Model\ function$	$[-]$
$\varepsilon = Least\ square\ error$	$[-]$

Contents

1 Introduction **1**
1.1 Background . . . 1
1.2 Aim and goals . . . 3
1.3 Limitations . . . 3
1.4 Specification of issue under investigation . . . 4
1.5 Schedule . . . 4

2 Methodology **5**
2.1 Documentation . . . 6
2.2 Define . . . 6
2.2.1 Stakeholders . . . 7
2.2.2 System boundaries . . . 7
2.2.3 Problem formulation . . . 7
2.3 Measure . . . 7
2.4 Analyse . . . 8
2.5 Improve . . . 8
2.6 Control . . . 8

3 Theory **9**
3.1 Rolling bearings . . . 9
3.1.1 Basic bearing designation system . . . 9
3.2 Bearing life time . . . 10
3.2.1 General bearing life time theory . . . 10
3.2.2 Hertz contacts theory . . . 12
3.2.3 General BEAST model theory . . . 12
3.3 Loads . . . 14
3.4 Mounting of bearings . . . 14
3.4.1 Ring creep . . . 15
3.4.2 Fretting . . . 15
3.4.3 Interference fit theory . . . 16
3.4.4 Interference fit practice . . . 20
3.5 Fatigue of the inner ring . . . 20
3.5.1 Contact pressure . . . 20
3.5.2 Hoop stress . . . 21
3.6 Curve fitting theory . . . 21

4 Procedure **23**
4.1 Define . . . 23
4.1.1 Stakeholders . . . 23
4.1.2 Customer needs . . . 24
4.1.3 System boundaries . . . 25
4.2 Measure . . . 26
4.2.1 Input . . . 26
4.2.2 Output . . . 28
4.2.3 Parametric study . . . 29
4.3 Analyse . . . 29
4.4 Improve . . . 30
4.4.1 Meta model - Scientific level . . . 31
4.4.2 Meta model - Engineering level . . . 31
4.5 Control . . . 31
4.6 Project process . . . 31

5 Results **33**
5.1 Pilot study . . . 33
5.1.1 Investigation of parameter levels . . . 34
5.1.2 BEAST evaluation . . . 35
5.1.3 Final configuration . . . 38
5.1.4 Curve fitting 1 . . . 42
5.1.5 Curve fitting 2 . . . 43
5.2 Minor study . . . 44
5.2.1 Measure . . . 44
5.2.2 Analyse - Effects within series . . . 45
5.2.3 Analyse - Effects over bore diameter . . . 47
5.2.4 Analyse - Effects of width . . . 48
5.2.5 Conclusion of minor study . . . 50
5.3 Major study . . . 50
5.3.1 Measure . . . 50
5.3.2 Analyse . . . 50
5.3.3 Conclusion of major study . . . 52
5.4 Final meta model - Scientific level . . . 53
5.4.1 Verification of the scientific model . . . 56
5.5 Transformation - Hoop stress to interference . . . 57
5.5.1 Measure . . . 58
5.5.2 Analyse . . . 58
5.6 Transformation - Contact pressure to load . . . 59
5.6.1 Measure . . . 59
5.6.2 Analyse . . . 60
5.7 Final meta model - Engineering level . . . 64
5.8 Summary of result . . . 66

6 Recommendations **67**

7 Discussion **69**

7.1 Methodology 69
7.2 Parameters 70
7.3 Modelling 71
7.4 BEAST model 71
7.5 Further investigation of parameters 72
7.6 Social, ethical and ecological aspects 73

8 Conclusion **75**

Bibliography **77**

A Appendix **I**

1 Introduction

SKF is a world leading bearing producer with multiple different variants of bearings for a wide range of applications in their product portfolio. In their facilities located in Gothenburg, they are producing and developing spherical roller bearings (SRBs) among others. The product line of SRBs consists of more than 500 main products, which means variants with unique outer dimensions. There is a large variation in dimensions, the outer diameter varies between 52 mm and 2.3 m, and the weight between 250 grams and 8.7 tones. Also, there are 12 ”series” corresponding to different proportions of bearing width compared to bearing thickness, for the same shaft diameter [1].

1.1 Background

The spherical roller bearing is designed to primarily accommodate radial loads and secondly axial loads with minimum occurrence of friction during rotational motion. The bearing is not designed to accommodate bending forces [2]. The bearing is self-aligning to compensate for misalignments that may occur on the shaft to prevent bending loads on the bearing. A SRB consists of barrel shaped rollers as seen in Figure 1.1. The barrel shape of the rollers gives a preferable load distribution in the bearing as seen in Figure 1.2 [3]. The bearing may be mounted onto the shaft in two ways, either with a tapered bore which is forced onto the shaft or with a cylindrical bore which is instead based on a temperature difference between the bearing and the shaft. Both of the mounting procedures conclude interference fit. The tapered bore introduces an asymmetric load distribution, resulting in higher inner ring structural stresses on the thinner ring side. To separate and guide the rollers, a guide ring and a cage are used for this purpose, mentioned in Figure 1.1. Both the inner and the outer ring have spherical raceways. The factors that influence the bearing performance are not only the load and the rotational speed of the bearing but also the geometry of the rollers, raceways, heat treatment and surface finish etc. The performance of the bearing is affected by all comprising components meanwhile for the calculated life time of the bearing, it is mainly affected by the rings and the rollers.

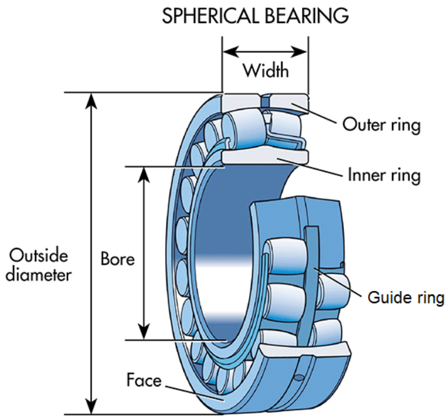

Figure 1.1: Shows the compound of a spherical roller bearing with included part denomination [1].

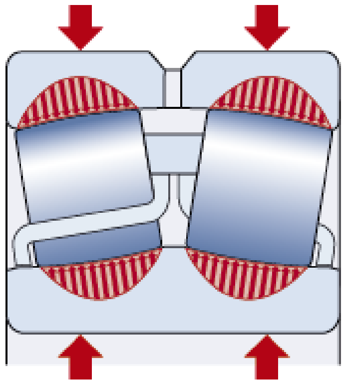

Figure 1.2: Shows the load distribution on the rollers during contact with the inner and outer ring [3].

Due to the interference fit of the bearing, the inner ring and the outer ring are exposed to structural stresses, in addition to the stresses caused by external bearing load. The stress in the inner ring will be tensile meanwhile for the outer ring, the stress will be compressive. Tensile stress in the inner ring is of most interest in a fatigue point of view. The circumferential stress component of these structural stresses in the inner ring has its maximum value between the inner ring and the shaft. It is further on decreasing in the radial direction but from a life time point of view, it is the circumferential stress component between the roller and the inner ring which is of interest, since it affects the life time due to fatigue of the sub-surface. The circumferential tensile stress of the inner ring is often called "hoop stress".

SKF are using an internal software called BEAST to simulate the dynamics of a bearing in a virtual environment. A specific model in BEAST have been developed to investigate how different SRB designs, hoop stress levels and load conditions are affecting the bearing life time. This life time calculation involves not only stresses from the bearing load but also structural stresses (e.g. hoop stress) due to, for example the mounting. At this point, the life time calculation model only allowed a scalar value of the hoop stress as an input. This was a disadvantage since the hoop stress is not a constant in reality but is varying over the geometry.

The question was now, based on the previous background, if the hoop stress have a substantial effect on bearing life time for SRBs at running conditions representative for customers and users. This included investigations of how different bearing load levels and hoop stress levels under normal running conditions affect the bearing life time in combination with a mounting procedure, according to SKF's recommendations.

1.2 Aim and goals

Since the hoop stress seemed to have an effect on the bearing performance and life, the underlying variables affecting the hoop stress were to be analysed more in detail to determine their relationships and degree of potential effect. The aim of this project was thus to identify the main variables influencing the hoop stress due to interference fit. Another aim was to assess if the hoop stress and its parameters in combination with the bearing load have a substantial effect on the bearing life time. Further on, the goal was to map the hoop stress effect in relation to the life time with respect to the included parameters and produce a model which describe this, a so called meta model. Such a meta model could then be used as a help to see if some bearings or parameters are more sensitive than others to the hoop stress but also new designs can be evaluated at a concept stage. The meta model was aimed to be based on existing theories and optimised to include as many bearings of the assortment as possible. The project as an entirety was done to generate more knowledge of spherical roller bearings and their relation and sensitivity to the hoop stress, during normal circumstances, in regard to their life time.

1.3 Limitations

The project was performed within 20 weeks at SKF's facilites in Gothenburg and more specifically during the period of 2018-01-15 to 2018-06-01. During this time, deliverables such as a planning report and a final report was presented accordingly to the aims and goals. No particular budget was assigned to the project. A limitation to only analyse the product segment of SRBs was set and hence did the study not include other product segments provided by SKF. The study focused on the effects of varying designs, dimensions, interference pressures and radial loads in relation to the hoop stress. The study also focused on evaluating the hoop stress effect for realistic user scenarios according to application area. The project was continuously aimed to be based on theoretical investigations of stress equations and life time simulations.

1.4 Specification of issue under investigation

Based on the information from previous sections, relevant questions that were needed to be answered during the lapse of the project are posed further on. These questions were heavily dependent on the information connected to the project's aims and goals but also with respect to the limitations presented earlier.

- What are the main parameters affecting the hoop stress and by which degree do they influence the hoop stress?
- What are reasonable limits for the hoop stress to give representative results from customer's point of view and normal running conditions?
- What hoop stress levels for different designs, dimensions, interferences and load conditions, might have a negative influence on the bearing life time?

1.5 Schedule

To have a dedicated time schedule for a project is of good help to not overshoot the length of the project but also not exceed the resources assigned to it. Therefore, a Gantt-chart was created at the start of the project to help the team to proceed and keep the time limits. The schedule was divided into smaller key parts which was thought to be carried out throughout the project. The available and dedicated time was 8 hours a day, 5 days a week and 20 weeks, resulting in 800 hours per person. An important note to mention is that the schedule was constantly updated since a Gantt-chart should be a living document and reflect the project's decisions timewise. The final Gantt-chart can be seen in Appendix A.1.

2

Methodology

In order to manage and perform a project with a result as good as possible, it is important to be methodical and precise in the tasks and work procedures. As a support to this foundation, multiple methodologies have been presented throughout the years as a help for the modern engineer to achieve the result one is aiming for. For the case of this project, an identification of the methodology DMAIC from the basics of Six Sigma was done through a brief investigation process of methods applicable for data based projects. DMAIC is a data-driven methodology which helps the user to carry out an improving and optimising project by using the five pillars the method is resting on and also, are an acronym of [4]. The five pillars are listed briefly below.

- Define – Define the project
- Measure – Collect and measure relevant data
- Analyse – Analyse the collected data
- Improve – Identify improvements from the analysed data
- Control – Standardise and normalise the improved state

An important note is that the methodology was not followed strictly compared to the original format but was instead adapted to suit the project in detail. The methodology worked as a general support, helping and aiming the project in the right direction. Hereby follows a detailed description for each of the included steps, which have been deconstructed into smaller parts. The project process as a whole is visualized in Figure 2.1.

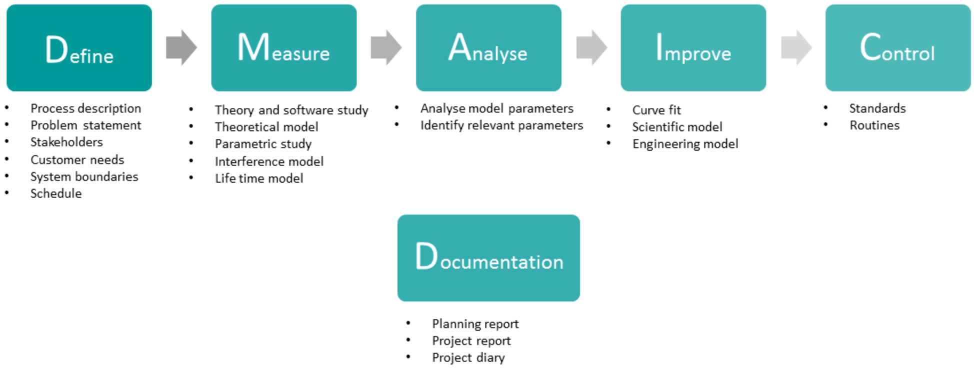

Figure 2.1: Shows the process of the project with DMAIC as a foundation.

2.1 Documentation

Documentation is not a main step within DMAIC but rather an underlying basics of the methodology. The documentation is among the most important in a project. Good documentation improves the efficiency of the project, minimising the risk of redoing work someone else already have been or are doing, both in the project itself and future projects. All steps, choices and decisions of larger extent should be documented well to allow for a good communication both inside projects but also to others, reading the results and reports. The documentation's main purpose is to spread the knowledge created inside a project and share it to others which will increase the knowledge overall. Such procedure can hopefully help a future project or individual to be more efficient and effective in their own project.

2.2 Define

During the define phase according to the methodology, the objective is to identify and understand the problem but also describe relevant relations to the project such as the customer needs, stakeholders, problem formulation and system boundaries.

2.2.1 Stakeholders

The stakeholders need to be identified to be able to explore the customer needs and establish their requirements. This is important, else there is a risk of creating something the stakeholders do not want. Additionally, if the requirements are captured and understood at an early stage, it increases the chances of doing what the stakeholders want in the first place which reduces waste of resources. Once the stakeholders have been identified, they should therefore be interviewed to gain knowledge about what expectations and requirements they have on the project. But not only the stakeholders should be interviewed. Individuals or teams that are experienced within the field and can add an important fact or consideration to the topic should be discussed with. This is all done to frontload the project with knowledge and build a good base to continue the project on.

2.2.2 System boundaries

All projects have to be limited to not continue forever and endlessly subtract resources. In projects over a longer time period, the goals and aims tend to be neglected or forgotten which results most often in a failed project. The limitations and system boundaries need to be stated early in the project in order to allow for distinct and understandable goals. They should also be stated and described together with the identified stakeholders in order to define accurate goals. Before settled, they also need to be agreed and accepted by all stakeholders having a saying in the project.

2.2.3 Problem formulation

When all of the previous steps have been performed with responsiveness and with precision, a clear problem formulation can be stated. The problem formulation should be easy to understand for all stakeholders, which then can agree or disagree to the formulation, as for the aims, goals and system boundaries. If the stakeholders disagree to the formulation, the process has to be iterated.

2.3 Measure

According to the methodology of DMAIC, it is important to collect the data correctly and set a baseline which the project can build and expand upon. This should also be done to enable for a comparison to the previous state and to see if improvements have been achieved. To do a reliable investigation and be able to construct a reasonable result, the practitioners need to have enough knowledge and understanding of the subject. Enhanced knowledge within the field will further on increase the chances of a successful project even beyond. Therefore one should always investigate relevant theories and similar projects to see what can be used for the own project and avoid possible pitfalls someone else already have done.

2.4 Analyse

If proper definition and measurement phases have been made, an easier analyse phase is to be expected. For the analyse phase a multitude of tools are available, both as softwares but also on paper. The purpose of this segment is to identify correlations and connections between parameters which then hopefully can be evaluated to either be improved or reduced. The analyse phase will accumulate the result which, at the end of this stage, should be compared to the previous state. If the result has reached an acceptable rate, the project can continue to the next phase. If not, the process must be iterated in order to achieve a result acceptable by all parties.

2.5 Improve

If there seem to be an opportunity of improvement based on the analysis, the improvement phase can be initiated where the identified improvement should be implemented. Such an improvement needs to be validated to verify if it reaches the defined requirements and expectations. This can be done by a plausibility analysis to see if the result is reasonable. A convergence analysis can also be performed to verify the result to be valid and to be within the accepted error limits, which should be discussed with the different stakeholders. The convergence analysis will further on tell how far away from the true value the result is. If the result is not approved by the stakeholders or a problem has been identified, the process must be iterated which starts with a decomposition of the identified problem to find the root cause and will hopefully lead to a solution of the particular problem.

2.6 Control

The control phase is about keeping and maintaining the improvement once such has been implemented. It is also about not stepping back to the original state, which often is easy due to tradition and culture. This can be avoided by implementing new standards or new routines which helps the user to follow the new improved solution as a normalised state.

3

Theory

This chapter will conclude and summarise the required theory and equations that were needed in order to proceed with the project. All information in this chapter were considered to be known and were not developed within this project. This chapter should rather be considered as a pre-requisite and a foundation which the project was built upon.

3.1 Rolling bearings

Bearings can be seen in all kind of applications but their main function is to constrain relative motion to only the desired direction. With other words, this function support applications rotating in some manner. More specific applications for a rolling bearing can be anything in a range from dentist drills to large crushers or windmills. Depending on the loading conditions and the size, different types of bearings are available to perform the task. There are a lot of other factors affecting the performance and life time of the bearing during usage. These factors were investigated through extensive theory studies and interviews with experienced personnel. The following chapters will in detail describe and evaluate how certain parameters affect and in what degree they affect the performance and life time of a bearing.

3.1.1 Basic bearing designation system

The designation of a bearing is standardised by ISO, but in short, it is depending on three main factors which further on consists of five to seven digits. The three main factors are the bearing type, bearing design and boundary dimensions [3]. The first digit indicates the bearing type, which is 2 for SRBs. The second and third digit indicate the relative width and outer diameter respectively. The last two digits multiplied by 5 give the bore diameter. This is the general case, exceptions are made for bearings with a smaller bore diameter than 10 mm and larger than 500 mm. An overview of how the different series of spherical roller bearings are varying with the same bore diameter is shown in Figure 3.1.

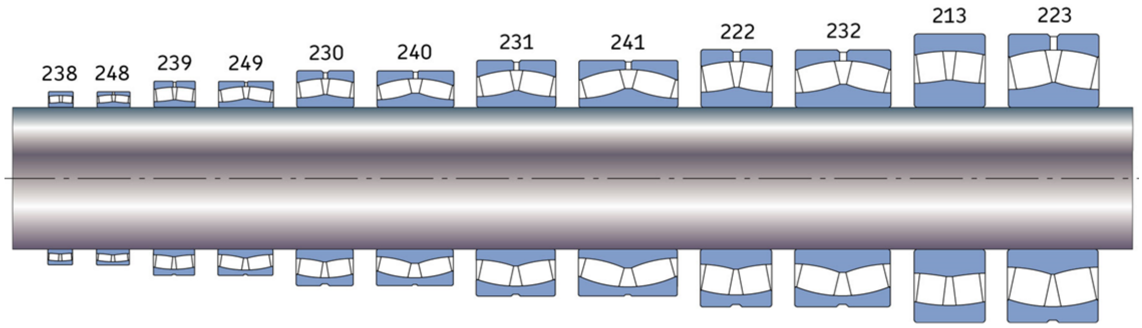

Figure 3.1: The figure shows how the series are varying for the same bore diameter.

The subsequent letters of the bearing designation represent the internal design of the bearing. These letters can also be combined with numbers to denominate the different internal designs. When comparing bearings, it would have been ideal to compare bearings with the same type of internal design and cage to avoid the potential influence of differences between the internal design. As described in previous chapters, it is known that the internal design only has a minor effect, if any effect, on the life time and was due to this not a restricting constraint.

3.2 Bearing life time

As a help to choose a suitable bearing for a certain application, the bearing life time is most often used. This is because the life time can be transformed into the number of revolutions from which the customer can choose a bearing which reaches the specifications of the application. Calculating the life time can basically be done in two ways, either by mathematical estimations or by computer simulations. These two alternatives will next be described in more detail to show advantages and disadvantages of them respectively.

3.2.1 General bearing life time theory

To calculate the approximate life time of a bearing, a general equation has been developed by SKF for this purpose, Equation 3.1. This equation is designed to give a fairly accurate result quickly. The equation has been extended and developed over time to be more accurate. These optimised equations are more specifically including other factors to improve the accuracy but with the disadvantage of more input data required from the user.

$$L_{10} = \left(\frac{C}{P}\right)^{p} \tag{3.1}$$

In Equation 3.1 the factor P is, a bit simplified described, the load. The exponent p is dependent on the bearing type, if it is a ball or roller bearing, the p value is 3 or 10/3 respectively. An important note to mention in this context is that the estimated life time, L_{10}, is at 90% reliability. The general equation is taking a number of correction factors into account to achieve a life time as accurate as possible but does not consider all of them. These have been compiled to the bearing specific factor C. The correction factors that need to be considered to calculate an even more accurate bearing life time may be described as a function of the raceways, rolling elements, cage, lubricant and sealing, as seen in Figure 3.2.

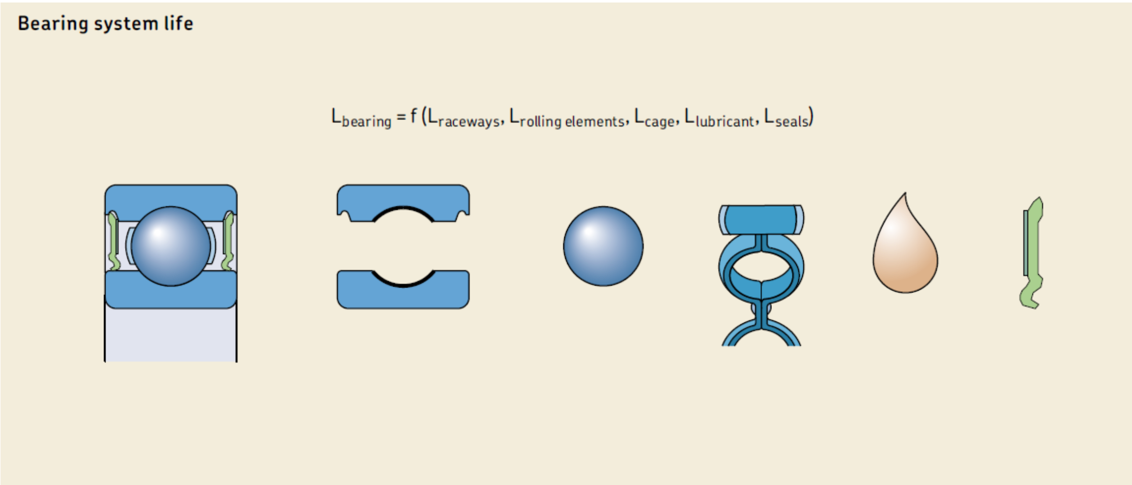

Figure 3.2: Shows the factors affecting the life time of a bearing as a system [3].

Dynamic load ratings

The dynamic load rating, C, is used for life calculations involving dynamically stressed bearings [3]. The ratings are based on an ISO standard and is an expression of the load which will result in a basic rating life of one million revolutions. It is assumed that the load is constant in magnitude and the load is pure radial for radial bearings. The dynamic load rating is according to ISO, proportional to a number of parameters, including three key factors according to Equation 3.2 [5]. The equation shows that the rating can be optimised in different ways, either by minimising the number of rollers or the effective roller length else by maximising the diameter of the rollers. This is due to the value of the exponent.

$$C_{ISO} \sim z^{3/4} \cdot D_w^{29/27} \cdot l_a^{7/9} \tag{3.2}$$

3.2.2 Hertz contacts theory

The Hertz contact theory is based upon an elliptical contact in the general case and includes a special case of line contact and another of point contact. Hertz contact theory is derived from the analytic solution of elasticity theory equations. The concentrating contact stresses are assumed to be independent of the shape and the boundary conditions of the bodies that are in contact. One qualification is that the contact area is much smaller than the body's size and radius in the contact point [6]. This Hertz theory is implemented in many programs within SKF. More specifically for this study, the contacts are referring to the contacts between the rollers and the both rings of the bearing.

3.2.3 General BEAST model theory

To evaluate the bearing performance without the use of costly real time testing, computer simulations can be of good use to both save time and resources. For this purpose, SKF are using a software called BEAST (BEAring Simulation Tools). BEAST is a state-of-the-art bearing simulation software developed inhouse by SKF and solely used within SKF. BEAST is a software for dynamic simulations of rolling bearings and other machine elements with contacts. It is a MBS (MultiBody Simulation) tool with similarities with for example the commercial software ADAMS. The software is able to consider all of the affecting factors in detail, which burdens the calculation time heavily. The simulations can be performed as dynamic or as so called quasi-static, which means that it is at internal equilibrium through the whole simulation [7].

Within BEAST, there are models tailored to analyse different cases, so called turnkey models. The turnkey model related to this project evaluates the roller bearing life with consideration to the hoop stress in the inner ring. The hoop stress affects the sub-surface fatigue in the sub-surface of the bearing together with the shear stresses due to the loading of the bearing, hence only the sub-surface fatigue of the inner ring was evaluated. The influence from the outer ring, the rollers and the surface fatigue were disabled in this model but can be handled through other models in BEAST. The model uses a quasi-static configuration to perform the calculations and simulations. The model have been developed in such way that the calculated life time of the inner ring is based on the shear stresses in the material, which are calculated in BEAST by evaluating the local contact deformations at the rollers and the inner ring. In addition to this, the BEAST model is considering the structural stresses such as hoop stress which contributes to the shear stresses in the inner ring. The inputs needed to run this type of analysis in the BEAST model, except for the hoop stress and radial load, are primarily dimensional parameters but also the rotational speed and the bearing clearance. Figure 3.3 shows how the interface of BEAST looks like when a complete bearing has been assembled together with the intimate parts in an exploded view next to it.

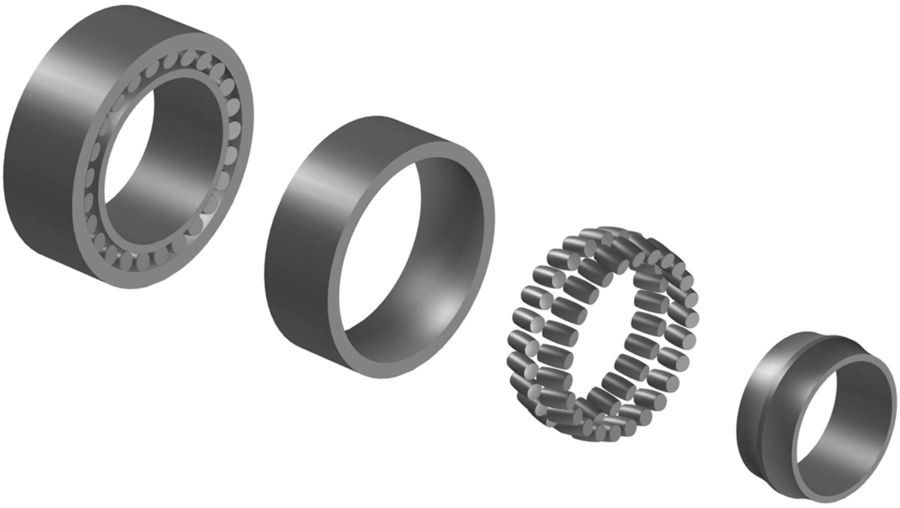

Figure 3.3: Shows the interface of BEAST when a complete bearing has been assembled together with an exploded view of the same.

The life time calculation within BEAST is stress based, which allows for a consideration of the hoop stress and other structural stresses. The calculation is basically an integral which takes the stresses into consideration in the sub-surface of the bearing. Both local stress from the contact between the rollers and the ring but also structural stresses as for example the hoop stress are hence included. This integral is performed over the whole contact volume for all the contacts of the bearing. For each contact, a predicted life time is calculated and the total life time of the inner ring is obtained by Palmgren-Miner's rule state [2]. The local contact stresses are in BEAST calculated with a modified Hertz theory which takes into consideration that the contact has boundaries as for example the contact is limited to the roller length, which original Hertz theory does not handle. So basically the difference between BEAST's way of calculating life time from the general L_{10} formula, seen as Equation 3.1, is that BEAST is stress based and L_{10} is load based. Most stress based life time calculation methods handle only shear stresses. However, a newly implemented life time calculation method in BEAST can handle all structural stresses including circumferential stresses. Hence this method was used for this study.

3.3 Loads

As mentioned previously, one of the main functions for a bearing is to be able to carry loads. The loads can be divided in two types depending on the load's direction. Depending on if the load is applied onto the outer ring or on the side of the bearing, it is called radial or axial load respectively. Radial load is the type of load the bearing is designed to withstand the best. Most commonly is that bearings are subjected to loads in radial and axial direction at the same time. If the resulting load is constant in relation to magnitude and direction, the equivalent dynamic load, P, can be determined by Equation 3.3.

$$P = XF_r + YF_a \tag{3.3}$$

In the equation, F_r and F_a represent the radial and axial load respectively. As SRBs are radial bearings, F_r is most often larger than F_a but the load can also be pure axial. X and Y are both tabulated bearing specific calculation factors to estimate the equivalent dynamic load P. For SRBs, the radial load calculation factor X is always 1 meanwhile the axial load calculation factor Y is varying depending on the bearing. To minimise the number of varying parameters, radial load was the only load condition considered for this study and thus not the axial load. This means for Equation 3.3, that the axial load F_a is 0 and X is 1. This further on results in the equivalent dynamic load being equal to the radial load, P = F_r. SKF have in the SKF Catalogue defined what can be considered as a light, normal, heavy and very heavy load. The magnitude of the load is proportional to the bearing specific load rating, C, which is defined for each bearing by SKF. The different load cases is listed below [3].

- Light load: $P \leq 0.05$ C
- Normal load: $0.05\ C < P \leq 0.1$ C
- Heavy load: $0.1\ C < P \leq 0.15$ C
- Very heavy load: $P > 0.15$ C

3.4 Mounting of bearings

One of the factors affecting the bearing's life time is the mounting procedure, or more specifically, the interference. This is due to the pressure which is inflicted by the interference between the inner ring and the shaft. This is needed in order to keep the bearing fixated onto the shaft first and foremost. The inner diameter of the inner ring needs to be a bit smaller than the diameter of the shaft in order to create the pressure which will keep the bearing at the desired place. To mount a bearing with a smaller inner ring diameter than the shaft's may sound difficult but several methods have been engineered to suit for this purpose. One method is to create a temperature difference between the shaft and the bearing, which makes the bearing expand or the shaft shrink depending on if the bearing is heated or the shaft is cooled. Another common method is to have a tapered shaft with a very small angle to make the inner ring fit at the edge. By putting an axial load on the bearing, it slides along the shaft to the desired, pre-calculated spot on the shaft. A disadvantage with this method is that the pressure will not be symmetrical over the inner ring and will be more difficult to calculate. The required interference is heavily dependent on the load. For a smaller load, a smaller interference is accepted meanwhile for a larger load, a larger interference is required.

When a bearing has been mounted and the application is running, there are still some problems that need to be considered due to the interference fit. More specifically, there are mainly two types of movements which may occur in the seating between the bearing's inner ring and the shaft during usage. These movements may have a damaging effect on the bearing if not payed attention to. The first type of the two is on macro level and is called ring creep. The second type is on micro level and is called fretting.

3.4.1 Ring creep

In a general case where a ring has a larger diameter than a shaft and they both rotates together, the ring will follow the rotation of the shaft but at a lower velocity. For each revolution the shaft does, the ring revolves a bit slower and will slowly creep from its original position. This phenomenon is called ring creep and can occur during usage in the case when a bearing has been mounted onto a shaft with interference fit. For each revolution, the displacement is equal to the difference between the inner ring's and the shaft's circumference. The recommended interference is dependent, among many other factors, on the magnitude of the load. The higher the load is, the larger interference is needed. The only truly effective way of locking the bearing onto the shaft in order to avoid ring creep is to mount the bearing with sufficient interference, which has a cost of increased stresses in the inner ring [8].

3.4.2 Fretting

If a force is applied to a disc, shear and normal stresses will appear in the disc. In the case when a bearing has been mounted onto a shaft, local sliding may occur between the ring and the shaft if the shear stress is greater than the frictional resistance between the surfaces. This local sliding may occur without ring creep and is resulting in a corroded surface. The phenomenon arise more easily when the inner ring is thin and the load is great. The corrosion is more likely to develop at the end parts of the bearing where the inner ring is thinner on a SRB. As for the ring creep, the fretting corrosion is prevented by having sufficient interference between the ring and the shaft. By having a larger interference, the stresses will also increase in the inner ring [8].

3.4.3 Interference fit theory

Based on the books Maskinelement [2] written by Melkersson K. and Mägi, M., Handbok och formelsamling i Hållfasthetslära [9] written by Bengt Sundström, and the SKF Catalogue [3], the primal theory regarding the interference, the hoop stress and related mathematical equations will be presented together with a description of the essential theory.

As a summary, the literature describes how the hoop stress is affected by the interference which is further on dependent on three main factors, involving several parameters each. The first of the factors is the pressure due to the interference fit between the inner ring and the shaft. Secondly, stresses inflicted by a difference in temperature during usage will also have an effect on the hoop stress. The last main factor affecting the hoop stress is the rotational speed. These factors are affecting the hoop stress in the case of plane stress, when the inner ring is assumed to be thin. However, for the case of plane deformation, when the shaft is assumed to be both long and thick, similar results are obtained but through more complicated equations. The inner ring and the shaft can be seen as elastic and cylindrical bodies. Therefore, the equations describing the stresses can be gathered directly from the elasticity theory. The equations in Equation 3.4 are describing the general plane stress case.

$$\begin{cases} \sigma_r(r) = -\dfrac{3+\nu}{8}\cdot\rho\cdot\omega^2\cdot r^2 + A - \dfrac{B}{r^2} - E\cdot\alpha\cdot\dfrac{I(r)}{r^2} \\ \sigma_\varphi(r) = -\dfrac{1+3\nu}{8}\cdot\rho\cdot\omega^2\cdot r^2 + A + \dfrac{B}{r^2} + E\cdot\alpha\cdot(\dfrac{I(r)}{r^2} - T(r)) \\ \sigma_z = 0 \\ I(r) = \displaystyle\int_{r0}^{r} r\cdot T(r)\cdot dr \\ u(r) = \dfrac{r}{E}\cdot(\sigma_\varphi - \nu\cdot\sigma_r) + \alpha\cdot r\cdot T(r) \end{cases} \tag{3.4}$$

An important note is that the equations are only valid when assuming a plane stress case. The equations are also considering an eventual temperature gradient, α and a rotational speed factor ω. A and B are two separate integration constants, defined from boundary conditions respectively. The boundary conditions for the inner ring and the solid shaft can be seen in Equation 3.5 and Equation 3.6 respectively.

$$\begin{cases} \sigma_{r,ir}\left(\dfrac{d_{ir.is}}{2}\right) = -p \\ \sigma_{r,ir}\left(\dfrac{d_{ir,os}}{2}\right) = 0 \end{cases} \tag{3.5}$$

$$\begin{cases} u_r(0) = 0 \\ \sigma_{r,s}\left(\dfrac{d_s}{2}\right) = -p \end{cases} \tag{3.6}$$

By inserting the boundary conditions into the equations in Equation 3.4, the stresses are obtained for the inner ring as in Equation 3.7 and for the shaft as in Equation 3.8. In Figure 3.4, the stress distribution of the different stresses is illustrated together with other important parameters.

$$\begin{cases} \sigma_{r,ir}(r) = -p \cdot \dfrac{\left(\dfrac{d_{ir,is}}{2r}\right)^2 - k^2}{1-k^2} \\ \sigma_{\varphi,ir}(r) = p \cdot \dfrac{\left(\dfrac{d_{ir,is}}{2r}\right)^2 + k^2}{1-k^2} \\ u_{ir}(r) = \dfrac{pr}{E_{ir}} \cdot \dfrac{(1-\nu_{ir})k^2 + (1+\nu_{ir})\left(\dfrac{d_{ir,is}}{2r}\right)^2}{1-k^2} \\ k = \dfrac{d_{ir,is}}{d_{ir,os}} < 1 \end{cases} \tag{3.7}$$

$$\begin{cases} \sigma_{r,s}(r) = -p \\ \sigma_{\varphi,s}(r) = -p \\ u_s(r) = -\dfrac{pr}{E_s}(1-\nu_s) \end{cases} \tag{3.8}$$

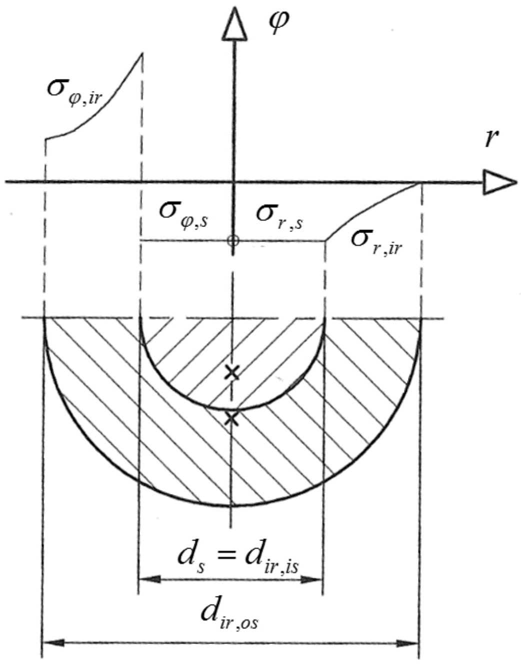

Figure 3.4: Shows the stress distribution through the ring and the shaft [2].

The sub-surface fatigue of the inner ring will most likely appear near the outer surface of the inner ring since the maximum stress due to the interaction between the rollers and the ring is located there. When looking at the outer diameter of the inner ring, one gets the following equations for the stresses in the different directions as stated in Equation 3.9. As can be seen by the equations, the hoop stress, σ_φ, is dependent on the relation between the inner and outer diameter, k, and of the pressure between the shaft and the inner ring, p.

$$\begin{cases} \sigma_{r,ir}\left(\frac{d_{ir,os}}{2}\right) = 0 \\ \sigma_{\varphi,ir}\left(\frac{d_{ir,os}}{2}\right) = \frac{2k^2}{1-k^2} \cdot p \\ \sigma_{z,ir}\left(\frac{d_{ir,os}}{2}\right) = 0 \end{cases} \tag{3.9}$$

The interference, Δ, can be described as the difference in displacement between the inner ring's inner diameter and the shaft's outer diameter, which can be seen in Equation 3.10. Figure 3.5 illustrates how the equation is obtained [2].

$$\Delta = 2(u_{ir} - u_s) \tag{3.10}$$

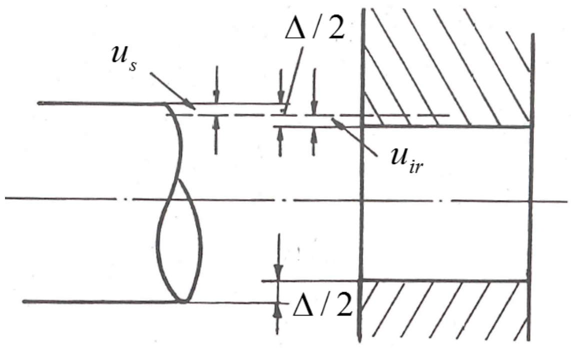

Figure 3.5: Shows how the interference is defined based on the displacements of the inner ring and the shaft.

The displacement part of Equation 3.7 and Equation 3.8, is then inserted into Equation 3.10 at a radius of $d_{ir,is}/2$ to establish a final relationship between the parameters. This is resulting in Equation 3.11.

$$\Delta = pd_{ir,is}\left[\frac{1}{E_{ir}}\left(\frac{1+k^2}{1-k^2}+\nu_{ir}\right)+\frac{1}{E_s}(1-\nu_s)\right] \tag{3.11}$$

When assuming the shaft to be solid and the inner ring to have the same material properties as the shaft, one can do some simplifications to the equation and obtain Equation 3.12.

$$\Delta = \frac{pd_{ir,is}}{E}\cdot\frac{2}{1-k^2} \tag{3.12}$$

From this equation, one can identify the relationship between the pressure and the interference. Equation 3.12 is then inserted into the circumferential stress component of Equation 3.9 to determine the relationship between the hoop stress and the interference. This final relationship can be seen in Equation 3.13.

$$\sigma_{\varphi,ir}\left(\frac{d_{ir,os}}{2}\right) = \frac{\Delta}{d_{ir,is}}k^2E \tag{3.13}$$

From Equation 3.13, one can draw the conclusion that the hoop stress will linearly increase with increasing interference, Δ, and decrease with inverse effect when the shaft diameter, d, is increasing. The factor k has a second degree influence which is nonlinear. The equation is also affected by Young's modulus which is a material constant and has a proportional effect. Due to this analyse, the only parameter which can be varied to obtain different values of the hoop stress for a specific bearing is the interference, since the other parameters are material or bearing specific.

3.4.4 Interference fit practice

Generally when designing an application, and a bearing is needed, the load and the shaft diameter are vital for the functionality of the bearing. Basically the shaft diameter governs the size of the bearing where multiple series of bearings are available, which further on are suitable for different ranges of loads. After that have been issued, the required interference to handle the load needs to be determined. Since it can be difficult to determine and control an exact value of the required interference, a fit is used instead which is a suitable range of diameters the bearing would fit onto the shaft with. To make it simple for the application engineer, or the customer, the loads have been assigned into different load cases described previously. The different load cases are connected to a recommended fit which is supposed to handle that particular load. Different levels of fittings are presented in Appendix, Figure A.2, with origin from the SKF Catalogue [3]. The underlying theories are created with the intention to avoid the appearance of ring creep and fretting and are based on mathematical equations. From these equations, a value of 0.08% Δ/d is obtained as an interference to not fracture the ring [8].

3.5 Fatigue of the inner ring

The fatigue of the inner ring have been divided into two parts depending on what phenomena are affecting the inner ring. The phenomena affecting the surface are included in surface fatigue and the phenomena affecting the sub-surface, are included in sub-surface fatigue. Fatigue on the surface of the inner ring considers factors such as lubrication, surface finish and amount of dirt but also loads and friction. The sub-surface fatigue considers rather what happens inside the material. This includes remaining stresses from the mounting procedure and the continuously alternating strains in the contact between the inner ring and the rollers. These are the two main factors influencing the fatigue of the inner ring's sub-surface and will be described in more detail in the following sections.

3.5.1 Contact pressure

When an application is running, the bearing within will be subjected to loads of different characteristic and proportion. Let's say as an example that a radial load is applied on a bearing while the shaft is rotating. The rollers in the bearing will then engage and disengage with a force on the inner ring's surface. This will ensue time varying shear stresses in the sub-surface of the inner ring. These stresses may have the potential effect of lowering the performance of the bearing. The shear stress effect will be amplified due to the stresses caused by the interference.

The cross section of a SRB is not equally thick over the whole inner ring, which results in a variation of the outer diameter of the inner ring, which can be seen in Figure 1.1 and Figure 1.2. Due to this, an approximation of where the maximum stress will occur was done. However, it is assumed to occur at a certain angle, or more specific, where the roller will be centred. The ideal outer ring diameter at this point is then calculated by trigonometry.

The contact pressure between the rollers and the inner ring is dependent on the number of rollers in the bearing as well. In a design perspective, one can have many rollers of a smaller diameter or few rollers with a larger diameter. Many rollers give a lower load on each roller and hence a lower stress level in the material at a given bearing load which is beneficial from a life time point of view. On the other hand, larger rollers also give a lower stress level in the material due to larger contacts between the inner ring and the roller for a given roller load. These two options are contradicting each other from an optimisation point of view. More rollers result in more contact points and smaller rollers mean more sliding. Both of these factors lead to higher friction which in its turn results in a higher temperature in the bearing and thus a shorter service life time. The optimal design of a bearing in this regard is the one with as thin inner ring as possible in theory. This is not possible in practice due to the occurrence of fretting and ring creep which have been presented in previous chapters. As can be concluded from this section, the contact pressure has a large impact on the bearing performance. The contact pressure is in itself affected by underlying factors which need to be taken into account [8].

3.5.2 Hoop stress

Hoop stress is the second stress having a major impact on the sub-surface fatigue. It is as mentioned earlier created by the mounting process due to interference fit. The hoop stress can be seen as a constant structural stress unlike the contact pressure and corresponding shear stresses which rather can be seen as pulsating stresses. Conclusions drawn from Section 3.4.4 are that the hoop stress will, apart from larger interferences, increase with a thinner inner ring which equals to a larger k factor. Also the type of material is of importance since Young's modulus is included.

3.6 Curve fitting theory

There are some different ways of creating a curve fit from a set of data points. The methods that have been taken into consideration in this project are described in brief next.

Interpolation

A method to create a curve fit is by interpolate between the data points to achieve an acceptable curve fit. Functions that are most often used for this purpose are splines or polynomials with a degree corresponding to the number of data points. When evaluating the curve fit based on a set of data points, it will fit the data points exactly which is a benefit. In between or outside the boundaries though, the behaviour may not be as desired and an error is to be expected.

Smoothing

The smoothing method is attempting to capture important patterns in the data and leave noise out. This gives a more natural behaviour of the analysed data since the different curve fits are based on mathematical functions such as hyperbolic functions or trigonometric functions. Related to this, there is the regression analysis which is a statistical method estimating the relationship between variables. By regression analysis one can understand relations between the studied parameters and their effect on the result. By using the least square method one can minimise the total error which the generated curve is producing. The least square formula can be seen in Equation 3.14 [10]. This formula allows for a comparison between the errors the different curve fits produce. The best curve fit can then easily be selected based on the smallest error obtained. The error is expressed in Equation 3.15.

$$\chi^2 = \sum_{i=1}^{n} \frac{1}{\sigma_i^2} \left(y_i - y(x_i) \right)^2 \tag{3.14}$$

$$\varepsilon^2 = \frac{\sum_{i=1}^{n} \frac{1}{\sigma_i^2} \left(y_i - y(x_i) \right)^2}{n} \tag{3.15}$$

4

Procedure

Relevant theories and equations had at this point been established which allowed for the project to be specified in more detail. The stages in the procedure were based on the method and were further on related to the essentials for each and every of the comprised phases accordingly.

4.1 Define

A proper and thoroughly pre-study was performed to frontload the project with knowledge. As a part of this, the stakeholders were identified to capture and establish their requirements and needs. A deeper understanding for the task was gained which allowed for properly stated system boundaries that came to limit the project. These segments were further on developed through multiple interviews and a large portion of information gathering. Also, the software had to be investigated and understood by the project group to identify the parameters affecting the result. Much time was devoted on these investigations early in the project with the intention to reduce the amount of errors and mistakes at later stages in the project.

4.1.1 Stakeholders

For this project, the stakeholders were identified as individuals or departments having a connection to the project. The major stakeholder, which was seen to be benefiting the most from the project, was SKF as a company. Within SKF, other stakeholders were identified. The primary stakeholder among those was the product development department of spherical roller bearings, which had ordered the study to evaluate and report the essential findings of the subject. The other secondary stakeholders within SKF were the research department, product engineering department, application engineering department and computational department. All of these stakeholders had direct or indirect requirements on the project which they wanted to be fulfilled and therefore had to be considered. On the other hand, it existed stakeholders outside of SKF as well. For example Chalmers University of Technology which also had requirements on the project. An illustration of the stakeholders within SKF can be seen in Appendix, Figure A.3 to create an understanding of how the stakeholders interacted with the project.

4.1.2 Customer needs

To gather and collect the knowledge needed to perform the project, interviews with stakeholders but also with experienced personnel at both SKF and Chalmers were held. For example interviews with the product development department at SKF were held to understand the problem better but also to capture and understand their requirements on the project better. Other meetings and interviews were held with the computation department at SKF, which was responsible for the BEAST model. Based on these interviews and meetings, a foundation for the project was settled which allowed for a proper understanding of the stakeholders, the customer needs and the problem itself. The customer needs were then divided in three levels based on the stakeholders. The first level covered the researchers' expectations as a scientific model, where the main need was to evaluate the sub-surface fatigue exclusively. This means more specifically a model which includes hoop stress and contact pressure between the inner ring and the rollers to describe the life time. The researchers did not need the underlying factors affecting the hoop stress and such parameters were thus not needed in their model. On the next level, the product development department of SRBs was identified. They were interested in the closest parameters affecting the hoop stress in order to evaluate the life time. These parameters were the interference and the load more specifically and came to be called the engineering level. The third level included the application engineers and the other departments, an application level. They wanted to have an application specific model with input for their needs, which was mounting force for example. However, since the product development department had ordered the study and due to the time limit, only needs up to the second level were considered. This resulted in an investigation of the first and second level of parameters meanwhile the third level of parameters were excluded. Anyhow, the meta model was aimed to enable for an adaptation of the third level of parameters in the future if needed and wanted. The levels of stakeholders in relation to the customer needs are visualised in Appendix, Figure A.3. The identified customer needs are presented next accordingly to the information given in this section which is based on discussions with the different stakeholders.

- Level 1: σ_φ and p_{max} - Hoop stress and maximum contact pressure.
- Level 2: Δ and P - Interference and load.
- Level 3: Mounting force and application.

4.1.3 System boundaries

Relevant boundaries were identified based on the information stated in the previous sections. The system boundaries were used to narrow the project down to the essentials of the subject to make the project more efficient. The study was thus limited to an investigation of spherical roller bearings with a cylindrical bore that are mounted onto the shaft by interference fit. Bearings that have a tapered inner ring and are further on forced onto the shaft by an axial displacement were excluded due to the asymmetric load distribution which is then inflicted. In combination to this, only solid shafts were considered, thus excluding hollow shafts. The shaft and the inner ring were assumed to be of the same material, resulting in the same Young's modulus and other material parameters. It was also assumed that the inner ring and the shaft have negligible temperature differences. Only the sub-surface fatigue of the inner ring was evaluated in the life time model, which is dependent on the contact stress field between the rollers and the inner ring in addition to the hoop stress. The surface fatigue theory, which instead consider lubricant and viscosity etc. was on the other hand excluded from the study since it is independent of the hoop stress. The sub-surface fatigue of a bearing is dependent of a lot of other factors. However, only the effect of the hoop stress and contact load were examined during this project. The hoop stress is occurring in the inner ring due to interference fit. Hence the study was limited to investigations of only the inner ring.

The parametric study was aimed to result in a mapping of how the interference fit in combination with the bearing load affect spherical roller bearings' life. The study consisted of two separate models to describe this. The first model was a theoretical model of interference fit and described how the hoop stress varies regarding its included parameters. The hoop stress was based on the assumption of a plane stress case with an exact analytical solution. The second model was the life time model in BEAST, where the calculated variations of the hoop stress were used as inputs.

The first model generated a collection of different values on the hoop stress, when dimensions and interference had been varied. The result was then brought into the life time model presented in BEAST. This stage was predicted to be very time-consuming due to all analyses and the overall data handling of the life time calculations from BEAST. In an attempt to reduce the estimated time, a selection of bearings was investigated in more detail at a first stage to set a baseline. This selection of bearings, levels of interference and bearing loads were based on discussions to iterate reasonable samples. Generally it should preferably be chosen to consider a span as large as possible, for example the largest and the smallest bearings for different proportions or series. The system boundaries and the proposed procedure that were decided upon are visualised together with the information flows in Appendix, Figure A.4.

4.2 Measure

The following sections will describe how all data was planned to be gathered and created, both to the interference model but also to the life time model based on the procedure described in the previous section. In more detail, how all parameters were planned to be varied, established and collected for the whole project will be accounted for. Due to the many parameters and their sometimes vaguely defined relationships, the problem could be seen as a multi-disciplinary problem. A multi-disciplinary problem imposes requirements on how the data creation and data analysis are made but also how the different parameters are varied. Hence when varying parameters, it must be done by following a structure, an own-defined structure or a methodology. A deeper investigation of suitable methods to varying parameters and perform the parametric study was done to see which of all methods was the most suitable for this type of project.

It was decided to perform the project in three smaller but significant steps. The first was said to be a pilot study and was only supposed to include a few bearings to understand and possibly identify improvements of the process for the larger upcoming studies. Also identification of trends and correlations between parameters which could be used in the larger studies were going to be searched for. This pilot study was followed by two sequential studies, one minor study and one major study, hereby their names. During the minor study it was intended to establish a foundation for the meta model to build upon. The minor study was going to identify the area of interest for the final major study as well. The major study covered the majority of the assortment with a focus on the findings from the minor study.

A hypothesis was going to be stated at first, prior to the pilot study, based on the underlying theory regarding which parameters were believed to have a large impact on the hoop stress and life time respectively. From this hypothesis, the crucial parameters were going to be focused and investigated more in detail but still with consideration to other factors. When the most impacting parameters had been determined, these would be the foundation of the meta model. Next though, the data collection will be explained in a more principle level to obtain a deeper understanding of the data creation process.

4.2.1 Input

To run a simulation in BEAST, a couple of different types of parameters are needed. A short description of each is presented next. All of the inputs are needed to be given as numeric values respectively. For the sake of the parametric study, a new simulation had to be performed for each variation. Luckily enough for larger studies, they can be sent to SKF's cluster where simulations in BEAST can be performed in parallel and more importantly faster.

Dimensions

To define the bearing in BEAST and allow for the software to produce a result at all, a lot of bearing specific dimensions and attributes are required for the model. This originates from a simulation point of view but also due to visualisation. All required product data are available in SKF's product database and were downloaded for all bearings as a preparation for the study.

Load

In the BEAST model, the radial load is an input. As for this project, the contact pressure would be advantageous to have as an input parameter instead. However, this contact pressure may be calculated by programs already developed by SKF. With this mathematical transformation, one could obtain the contact pressure from the corresponding force.

Hoop stress

Another input parameter, which was the essential of the topic, is the hoop stress. As mentioned in previous chapters as well, the hoop stress is based on underlying factors such as interference and dimensions of the inner ring. The parametric levels to use were determined by evaluating the assortment with the help of Equation 3.13. The identified maximum and minimum hoop stress for reasonable user cases were representing the boundaries of the parametric levels.

But in order to be able to obtain the different values of the hoop stress for a specific bearing, the interference had to be varied. For this purpose, the interference described as 0.08% Δ/d in Section 3.4.4 was used. The lowest possible value for Δ/d is 0, which corresponds to no interference. A consequence from the choice of Δ/d as a parameter is that the fraction will be dimensionless, since both Δ and d are lengths, and would therefore not be dependent on the load, which was desired.

Others

The rotational speed and the bearing clearance are also input values but according to previous chapters, they were both set to constants during the study in order to reduce the number of varying parameters.

4.2.2 Output

From BEAST, a lot of different kinds of outputs can be extracted due to the wide functionality BEAST offers. For the case of this project, only the life time of the bearing and the contact pressure between the inner ring and the rollers were of importance and as a consequence, the only outputs that were collected.

The format of the output have been programmed by the developers in such way that for each run, a separate output file is received. With other words, for a large set of runs in BEAST, an equally large set of output files will be received. In this project, a lot of data was predicted to be handled at later stages and thus must the files be put together to ease the analysis. For small sets, this merging process was expected to be done manually with some effort. However, a script that merged the separate files into one single master file would be advantageous and was further on investigated in order to focus the limited time on analysing the data instead of being obstructed by the data handling.

Life time

The most relevant output for this project was the calculated life time accordingly to the newly implemented theory, which is considering the hoop stress. The life time was collected for all runs and all variations to be analysed at a later stage.

Contact pressure

The maximal contact pressure between the rollers and the inner ring is also an output, which was consequently controlled to not vary more than a pre-determined error rate. If another value of the pressure would be simulated than the expected pressure, an error in the life time would also be expected and had to be remade. With other words, the contact pressure was used as a control mechanism to verify the validity of the simulation to be correct.

4.2.3 Parametric study

Since a lot of combinations of loads, interferences and dimensions were going to be varied, both the variation and evaluation process were needed to be performed in a structured manner. A short investigation of available methods to vary parameters was made but it was soon realised that the methodology behind Design of Experiments, DoE, was beneficial. DoE was advantageous since it gives clear indications of how parameters are affecting the result, which was desired to evaluate in this project. It is also a more efficient method than other similar methods since it effectively reduces the amount of runs needed to perform a full factorial analyse [11]. Another advantage was that in BEAST, there is an existing tool which builds on the methodology of DoE, allowing for an implementation. Thus a factorial experiment was used as a help to investigate the effect of the produced parameters and the interacting factors. The full factorial design was built with two factors, contact pressure and hoop stress as presented earlier. This was possible since the produced parameters had been made independent from each other, which this methodology requires for a reliable result.

4.3 Analyse

Generally, when data have been collected, it needs to be analysed in order to be able to draw any conclusions. The result needs to be understood and documented as well to achieve a deeper understanding of the phenomenon. Relevant questions need to be posed based on the result to see if there are any obvious correlations or synergies but also unexpected correlations between parameters. Further on different parts of the theoretical models can be plotted against relevant variables to identify any signs of correlations.

Once the data had been collected and merged into one file, it had to be analysed to draw relevant conclusions. In an attempt to do this, the life time when a hoop stress had been inflicted would be compared against the case when there was no hoop stress at all, which had been entitled to the reference point, for the same loading condition. This produces a factor of the life time reduction between the two cases and can then be visualised in a plot to tell how large impact an inflicted hoop stress has on the life time. The largest number possible to achieve through this calculation is 1, for the case when there is no hoop stress for the both cases. The other extreme case will result in a value converging towards 0 when the relative life time is reduced considerably. Hence, the formulation will be between 0 and 1. A mathematical description of the formulation is shown in Equation 4.1.

$$0 < \frac{\textit{Life time at } (\sigma_\varphi)}{\textit{Life time at } (\sigma_\varphi = 0)} \leq 1 \tag{4.1}$$

Since so many parameters were assumed to affect the result, a plot of all of them together would not to be useful for a visualisation purpose since it would be a function in hyperspace due to its many dimensions. A solution to this was to project the function in two dimensions at a time to see how only one parameter affected the life time. This results in a multitude of scatter plots where trends and correlations could be seen from the two dimensional plots and further conclusions could be drawn.

4.4 Improve

The plots that were produced in the analyse phase were desired to be translated into a mathematical function which would be the foundation of the meta model later on. Proposals of such mathematical function would have to be done based on the shapes of the plots. As a help, plots with more data points than the intended set were used to identify the behaviour of the phenomenon more easily. Once a suitable shape of the function had been identified, it had to be optimised to fit the whole assortment of SRBs as good as possible. When the mathematical function was able to describe the occurring phenomenon well enough, it could be seen as the final meta model, describing the scenario. In order to achieve a meta model describing the whole product range, the process would have to be iterated in order to tune and adjust the meta model to make it more accurate. If the case would occur that it would be impossible to create only one meta model solely for the whole product range, an additional meta model would have to be created to handle that specific area of the assortment.

The produced mathematical functions were evaluated with the least square method, which is a method that generates a comparable value that shows how the function is performing compared to the real values [10]. The least square method can be seen in Equation 3.14. Since the measurements were executed at the same position the whole time for a certain set, the standard deviation was not needed to consider. The produced error for a generated function was compared to the error for a competing function. In this manner the function with the least error was selected since it would perform the best.

The interpolation method was disregarded in this project since the functions' behaviour between and outside the data points may be unpredictable and may not represent the reality, which was desired even though this method would create a fit that exactly matches the data points.

4.4.1 Meta model - Scientific level

First of all, a meta model was constructed within the scientific level, described in the system boundaries, Section 4.1.3. This level describes how the assortment was behaving relative to the contact pressure and the hoop stress.

4.4.2 Meta model - Engineering level

Secondly, the meta model for the scientific level had to be transformed into the second level presented in Section 4.1.3. This level had to be adjusted from the researchers' point of view towards the product development engineers' requirements, which more in detail are considering interference, load and dimensional parameters. This was advantageously done by a mathematical transformation.

4.5 Control

During the control stage, the result needs to be verified as functioning and improved. For this case, the constructed models' performance and accuracy had to be evaluated. This was done by collecting data between the data points that had built the models to verify that the error in between was below the accepted rate and the accuracy of the models was good enough. If the models were not performing as desired, additional data points need to be added to the models to improve them until a desired result have been generated.

When the models were stable enough, guidelines for researchers and product developers were elaborated to support them when analysing and using the models to investigate the behaviour of existing bearings but also as a support during development of new bearings.

4.6 Project process

The whole process from data collection to the analyse phase, and finally a creation of the meta model is visualized in Figure 4.1. It summaries the project procedure that has been described previously in this chapter.

Figure 4.1: Shows the process of the project.

5 Results

The detailed process of the project had now been defined and could be executed. The execution involved the stages of measuring, analysing and improving in an iterative manner to achieve a result as accurate as possible and will be explained in very detail. First of all, the pilot study was performed to identify the overall behaviour of the bearings when applying contact pressure and hoop stress. A general boundary was here set to achieve a stable model. Secondly the minor study would present the behaviour over the assortment of bearings. Next the major study would span over the whole assortment to be able to draw conclusions of the overall behaviour. A final meta model for the scientific level would at last be created to visualise the result of how the hoop stress and the contact pressure are affecting the life time. This meta model would then come to do a mathematical transformation to visualise the result for the engineering level instead, which rather considered interference and load. All of the plots presented in this chapter have been created through the interpolation method to obstruct an interpretation of detailed values, when in reality they were built from lines and data points. The data points, grid lines and tick marks have been removed in the plots for the same reason. With this said, the result is still in principle correct.

5.1 Pilot study

To increase the knowledge and understanding of the result, the pilot study was performed first in order to make a more efficient and effective analysis for the larger upcoming studies. First the parameters were investigated to determine suitable levels of them to be varied. These levels were used in the upcoming studies. The settings of the software were tuned and confirmed by a test run of the most extreme bearings, the largest bearing and the smallest bearing. A third bearing was chosen in addition as a complement to represent a medium sized bearing.

5.1.1 Investigation of parameter levels

In the theory study it was found that the most interesting and relevant parameters to vary were the load, described as the maximum constant contact pressure between the rollers and the inner ring, and the hoop stress based on the interference. One could then argue to randomise the ranges and intervals completely but the target was to find a result for states that are seen as reasonable and are used by the customer in their applications. This was one of the main considerations when determining the intervals. Also to ease the visualisation and understanding of the evaluation, linear intervals were desired. Another major decision which had to be made was the number of varying levels. More levels would give a longer simulation time but would also give a more accurate meta model. Depending on the number of levels, it would result in the same number of data points the meta model could be created from. With other words, only one level would give one point to build the meta model from, two levels give a straight line, three or more levels give a curve segment with increasingly accuracy. Due to this, four levels were decided to use for each case, plus an additional level for the reference value of the hoop stress. Presented next is how the parametric levels were identified and decided based on the given information from this section for both of the parameters.

Loads

To determine the parametric levels of the load, it was desired to match the levels with corresponding maximum contact pressure between the rollers and the inner ring. This was due to the fact that the stress based life time equation is dependent on this pressure. By keeping the contact pressure constant, the dependency of it would be eliminated.

To find relevant levels of the contact pressure, an investigation was performed over the whole assortment of bearings with varied loads within the recommended intervals stated in Section 3.3. The result showed that the maximum contact pressure between the most loaded roller and the inner ring varied from 1000 MPa to 2500 MPa. Due to this outcome, the range was selected accordingly. Since linear intervals were desired, four levels of the inner ring contact pressure were determined based on the range. The chosen levels can be seen next.

- Pressure level 1 = 1000 MPa
- Pressure level 2 = 1500 MPa
- Pressure level 3 = 2000 MPa
- Pressure level 4 = 2500 MPa

Hoop stress

As have been presented in earlier chapters, the hoop stress was successfully made independent by using Δ/d as a parameter to be varied. A reasonable range for Δ/d to be varied within was from 0%, which corresponds to no interference, to 0.08% based on the information from Section 3.4.4. The calculated maximum hoop stress at the inner ring contact was observed to vary up to 140 MPa. The levels of the hoop stress were selected up to this value, including the reference point which had been defined as 0 MPa.

5.1.2 BEAST evaluation

Before the main study could be started definitely, the BEAST model had to be tested to work as planned and to give results which could be analysed. Two separate test runs with different purposes were performed to ensure this. As a help to do this, three bearings were examined. These bearings were chosen in regard to their sizes. The smallest and largest bearing possible were chosen together with a bearing in medium size. The three bearings are listed next from the largest to the smallest. The bearings have also been marked in a principle way with a blue colour in a matrix of the assortment, seen in Appendix, Figure A.5.

- 248/1800
- 24072
- 22205

The aim of the first test run was to evaluate the accuracy and reliability of the model. What was done in more detail was that, for all of the three bearings, they were evaluated with 64 levels each of increasing loads accordingly to the range of reasonable contact pressures described in the previous section. What could be expected was increasing contact pressure and decreasing life time as the load was increasing. With this expectation in mind, the output did clearly give some confusing results at first sight. The results can be seen in Appendix, Figure A.6 and Figure A.7. What was confusing was that, for some increasing loads, the contact pressure increased but at the same time the life time was longer, which can be seen at the first curve in blue colour, in Figure A.6. For other cases, with increasing loads, the contact pressure was varying, which can be seen at the first curve in blue colour, in Figure A.7 instead. For different levels of hoop stress, the same behaviour was occurring. This problem had to be issued before the pilot study could be continued. As the first test run showed, some tuning inside BEAST was needed in order to optimise the simulation and be able to evaluate the result in a reasonable fashion.

Rotational speed

Originally a static model, with zero rotational speed, was preferred since the effect of rotational speed was not included in the study. However, since BEAST is a dynamic software, a small consistent speed was chosen to counteract the dynamic properties of the model. A problem by defining the rotational speed as 0 may occur since the rotational speed defines the direction of the coordinate system in the model. BEAST is not preventing you from doing this and undesirable result can then be obtained. Hence, the rotational speed was defined as 1 rpm for all bearings and would also be for all runs performed at later stages. The effect of the rotational speed parameter in the BEAST model was needed to be investigated to prove that it did not have a significant effect on the simulation result. When the effect of this parameter was investigated in BEAST, an extremely low rotational speed and an extremely high rotational speed were used for comparison. The difference between the two cases showed that a maximum difference of 0.03% of the life time was obtained. For the contact pressure instead, only as much as 8.3 Pa in difference was seen. From this evaluation it could be assumed that the rotational speed had a negligible effect on the life time and contact pressure calculation for the sub-surface fatigue model in BEAST. In order to see the true effect of the rotational speed, one has to adapt Equation 3.13 to also include the factor of rotational speed, ω, from Equation 3.4.

Accuracy

Once assured that the rotational speed had no major effect on the result, other factors that were believed to disturb the result were examined. As a repetition, the problems were that, at some levels of increasing contact pressure, the life time increases seen in Appendix, Figure A.6. In other scenarios, the contact pressure decreased with a higher load, seen in Figure A.7. Both of the cases are intuitively wrong, therefore an extensive investigation was initiated to identify the cause. Some different factors were investigated to track down the root cause of this behaviour. After several investigative runs and meetings, the problem seemed to appear from the generally low accuracy of the model for especially some load cases. As an attempt to solve this problem, the solver precision was increased from a value of 5 (rough) to 2 (fine) at first. This change resulted in stabilised values of the contact pressures. The contact pressures were no longer alternating but instead followed a smooth shape, seen as the second curve in Figure A.7. Another effect was that the output file from BEAST shows a performance measure over the whole simulation of the contact pressure and not the maximum contact pressure at the end of the simulation. When looking at the performance measure of the contact pressure, one should be aware that this is not the true maximal contact pressure but can be seen as a good approximation. However, most importantly did the contact pressure always increase for increasing loads which it should intuitively do. The issue with alternating life times was still not resolved and the previous change had no visible impact on the life time.

Due to the previously successful solution, other parameters affecting the overall accuracy of the model were investigated. One of these factors which was thought to have a large impact on the accuracy was the resolution of certain surfaces. From a standard resolution value of 1, the resolution was doubled to see the effect at the affected surface, which only was the surface of the inner ring. The result showed a significant improvement, the life times were not alternating as much as in previous studies, but still the problem existed. The resolution was then doubled once more to see if an additional improvement could be achieved, and it could. The alternations were gone but still some irregularities remained. By increasing the resolution to infinity, the problem was expected to disappear. This was tested by increasing the resolution 10 times from the original state. The result showed an even better curve with less irregularities and no alternations. However, by increasing the resolution 10 times, the calculation time was increasing tremendously. An accepted rate of resolution thus had to be decided. When doing this, the accuracy and time were the two factors competing against each other. A resolution of 0.25 was then decided to give an acceptable result at an acceptable time. The acceptable resolution are shown in the second curve in Figure A.6. The final rate of the settings can be seen next where the standard value is stated within the parentheses.

- Resolution: 0.25 (1) for the inner ring's surface, s6
- Solver precision: 2 (5)

Range

Another issue which needed to be addressed was the infinite life time at lower loads. As seen in Appendix, Figure A.6, a lower value of the contact pressure was giving a life time which was reaching infinity. BEAST handles such situation by setting a maximal life time of 10^{30} of all contacts which makes it impossible to analyse low load contacts. This shows that contact pressures at 1400 MPa and below were not enough to obtain sub-surface fatigue of the inner ring. Hence, the range for the contact pressure parameter was instead decided to vary between 1500 MPa to 2500 MPa as a result. However, the previous tunings with a resolution of 0.25 had shown that the life time model was rather unreliable at lower contact pressures and higher contact pressures within the range. According to Figure A.6, the somewhat more stable section seemed to be within the range of 1700 MPa to 2300 MPa for all of the three bearings. Due to this, the previously stated range of contact pressures were restricted a bit more to a range where it was more stable.

- Pressure level 1 = 1700 MPa
- Pressure level 2 = 1900 MPa
- Pressure level 3 = 2100 MPa
- Pressure level 4 = 2300 MPa

Iteration of contact pressure

As mentioned in the procedure, the contact pressures were estimated by a program produced by SKF. Since the contact pressures given by this program differed from the contact pressure simulated in BEAST, each of the desired pressure levels of the selected bearings were needed to be iterated. During this study, the error differed between 6% to 20%. The most problematic bearings seemed to be the largest bearings. A reason for the difference is that BEAST is using the modified Hertz-theory. The iterations were performed until the error was between 0% to 1%, which had been set as an acceptable error.

5.1.3 Final configuration

As a final test before the larger studies could be initiated, the life time calculation was performed for the three bearings. All of the contact pressures were collected together with the life times. The life times were then divided by the life times without any hoop stress to create a factor to analyse further. The framework of the data handling can be seen in Figure 5.1.

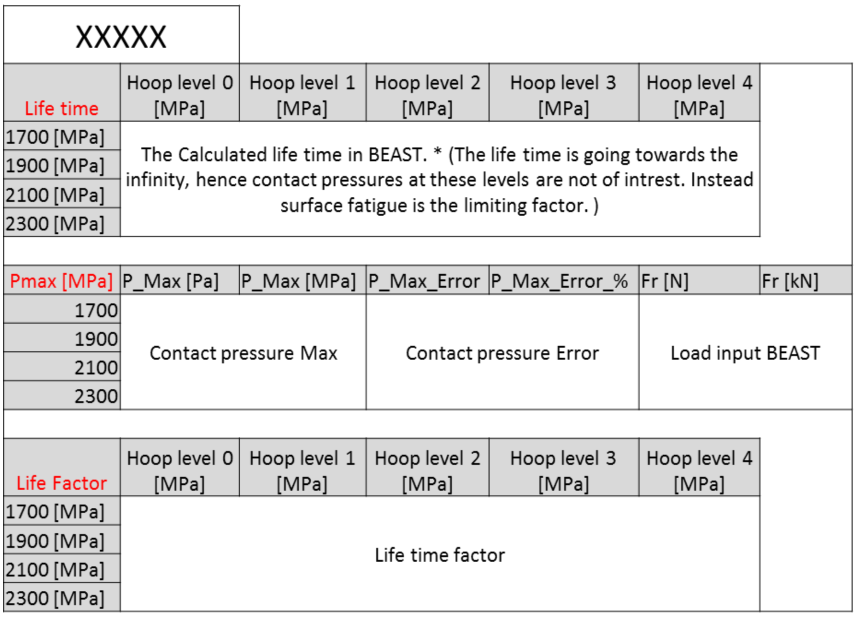

Figure 5.1: Shows the standard configuration which was used to handle the collected data.

For all levels of contact pressure, the values of the life time factor were then plotted against the hoop stress in order to visaulise the behaviour. As mentioned in the beginning of this chapter, all plots presented in this chapter have been adjusted to make it more difficult to interpret the exact values that were used for this study. In combination to this, all data points, grid lines and tick marks have also been removed in the plots for the same reason. The principle result for bearing 24072 can be seen in Figure 5.2. What could be seen from the graph was that all of the curves were going from 1 to 0 with similar shapes, as expected accordingly to Equation 4.1. When comparing the curves, it seemed as the hoop stress had a larger relative impact on the life time factor when the bearing is applied to a lower load. This is due to the fact that when the bearing has a very long life, as for the case of 1700 MPa contact pressure, the added hoop stress is dominating and is giving a larger reduction of the life time then for the level of 2300 MPa contact pressure.

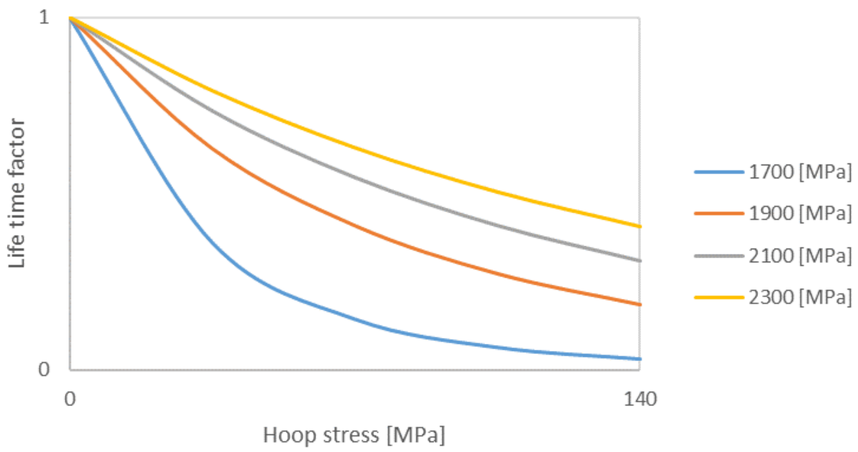

Figure 5.2: Shows the life time factor plotted against the hoop stress, with fixed levels of contact pressure for bearing 24072.

The same behaviour was also identified for the other two bearings that were investigated during the pilot study. Even though the same behaviour was occurring, some differences could be seen between the bearings when they were subjected to the same contact pressure, which is shown in Figure 5.3.

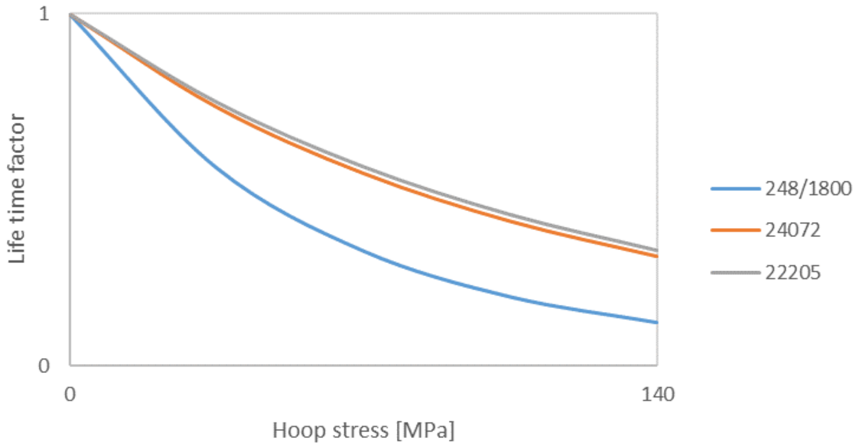

Figure 5.3: Shows the life time factor plotted against the hoop stress, with 2100 MPa of contact pressure for all of the three bearings.[1]

It seemed as for larger bearings, the life time factor reduced faster with increasing hoop stress. The reduction seemed to be more significant for larger differences in dimensions meanwhile for somewhat smaller differences, it did not have the same effect. From this result, it was understood that more bearings of both smaller and larger sizes had to be investigated in more detail in the minor study to see the true effect of the hoop stress.

Shape identification

In order to find a mathematical function which suited for the collected data sample, a smaller study was performed to find available mathematical functions which seemed suitable to investigate further. The target was to find a function with high accuracy in relation to the identified shape of the data points. What was done more in detail was a search of mathematical functions on the Internet and in mathematical handbooks that seemed to look the same as the data sample presented in Figure 5.2 [12]. A couple of promising alternatives were identified and are listed next. In combination, the functions have been plotted to visualise their shapes in Figure 5.4.

- Hyperbolic tangent ($tanh$)
- Inverse tangent ($atan$)
- Exponential function (e^{-x})
- Hyperbolic secant ($sech$)

[1]This result is updated further on in the study.

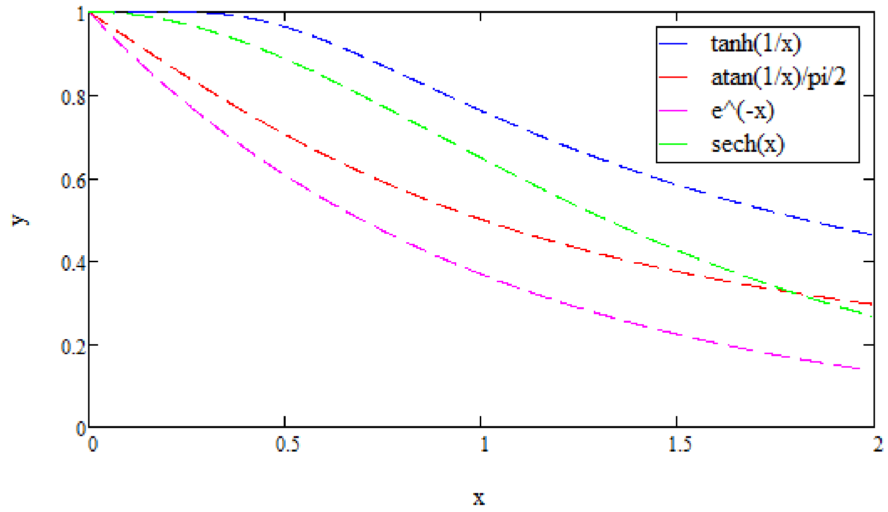

Figure 5.4: Shows all of the considered mathematical functions with a relevant shape.

To get more accurate shapes of the mathematical functions, they were all optimised in Mathcad with the function "genfit". This function generates a curve fit of a mathematical function in relation to a set of data points according to the least square method implemented in the software. The set of data points for this case was the collected data from the test runs with good accuracy. When all functions had been optimised accordingly, they were plotted against each other together with the data points to see which of the mathematical functions that matched the data points the best. The plot is shown in Figure 5.5.

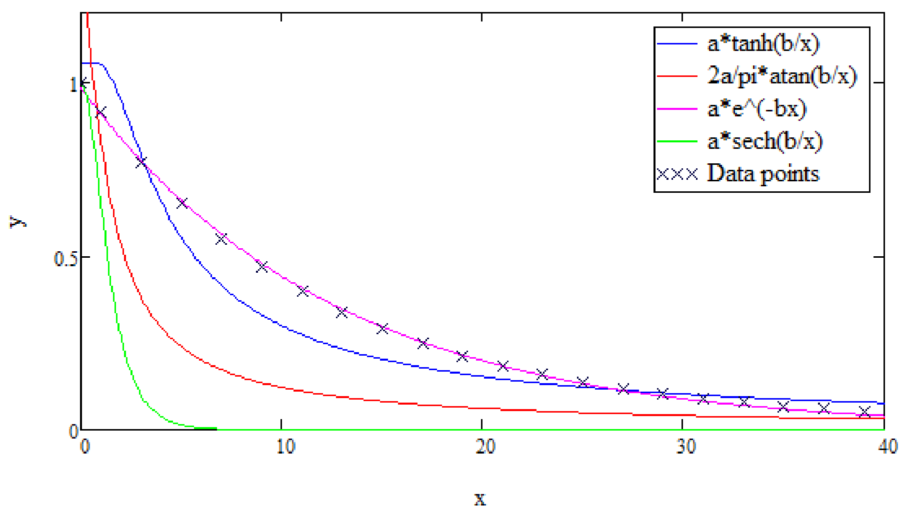

Figure 5.5: Shows all of the considered mathematical functions, optimised to fit the chosen data set.

Based on the Figure 5.5 and a least square error comparison, a final decision could be made. The exponential function e^{-x} was chosen since it seemed to be the best according to accuracy and other mathematical properties. The function is continuously derivative, which was of good use since the derivative was needed for the curve fitting function. Worth mentioning is that all of the hyperbolic functions are based on the exponential function in foundation, which produces their characteristics. Hence, all produced curves are based on, and have a shape close to e^{-x} in some parts.

5.1.4 Curve fitting 1

As previously identified and described, the exponential function seemed to be a good option to describe the behaviour of the life time factor as a function of the hoop stress. All of the curves exemplified in the Figure 5.2 seemed to behave in the same manner. Due to this, a curve fit was generated for each of the curves with fixed contact pressure. The general function can be seen as the first function in Equation 5.1. In Equation 5.1 there are two constants, a and b. These constants needs to be determined to achieve an accurate curve fit. This was done with the "genfit" function in Mathcad, which is based on the least square method described in Section 3.6. The first constant, a, was fixed to 1 since it was a boundary condition. This was because the function starts at 1 when the life time with no hoop stress is divided by itself, accordingly to Equation 4.1. The first function could then be simplified as the second function in Equation 5.1.

Figure 5.6 shows the result of the curve fit for 24072. This was further on done for all of the bearings included in the pilot study. The behaviour of the hoop stress had now been described at 4 levels of different contact pressure.

$$LT(\sigma_\varphi) = ae^{-b\sigma_\varphi} \rightarrow LT(\sigma_\varphi) = e^{-b\sigma_\varphi} \tag{5.1}$$

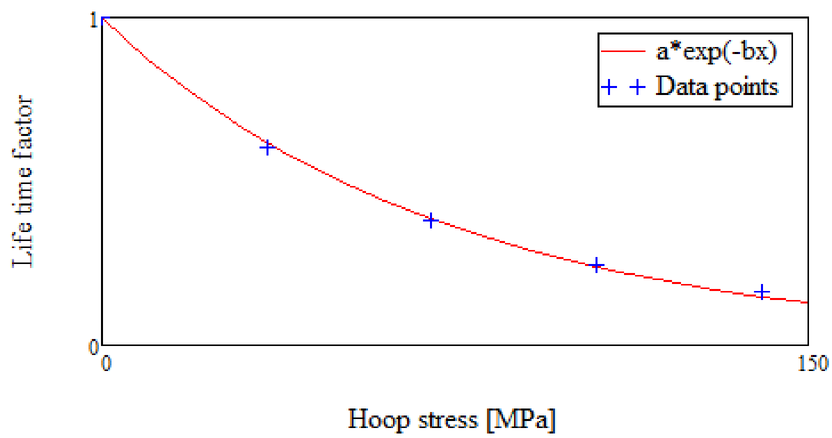

Figure 5.6: Shows the curve fit of the hoop stress effect on the life time factor for bearing 24072.

5.1.5 Curve fitting 2

To obtain a function that handles both hoop stress and contact pressure, investigations were performed on how the previously generated functions could be combined. When plotting the values of the constant b, that had previously been generated, they could all be described with an exponential function. The curve fit of the contact pressure could then be described by Equation 5.2 and is illustrated in Figure 5.7 where the data points are the values of b.

$$b(p_{\max}) = ce^{-(dp_{\max})} \tag{5.2}$$

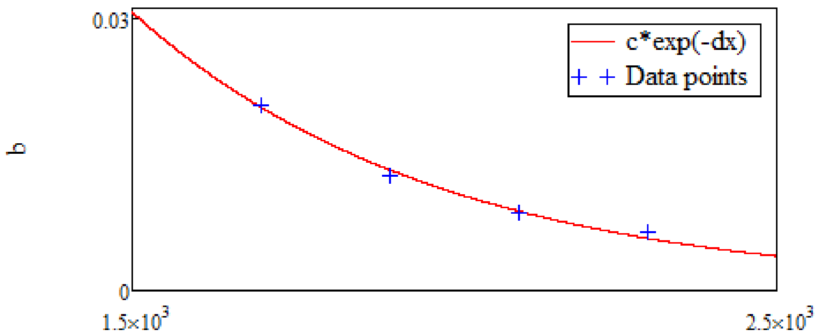

Figure 5.7: Shows the curve fit of the contact pressure effect in regard to the b values.

Now the model of the life time factor could be described by one single equation as a function of the hoop stress and the contact pressure, seen in Equation 5.3. The model is also illustrated in a 3D plot in Figure 5.8. The red colour represents a high value of the life time factor, which means a relatively small effect of the hoop stress. The blue colour represents a low value of the life time factor, which means a relatively large effect of the hoop stress.

$$LF(\sigma_{\varphi}, p_{\max}) = e^{-(ce^{-(dp_{\max})})\sigma_{\varphi}} \tag{5.3}$$

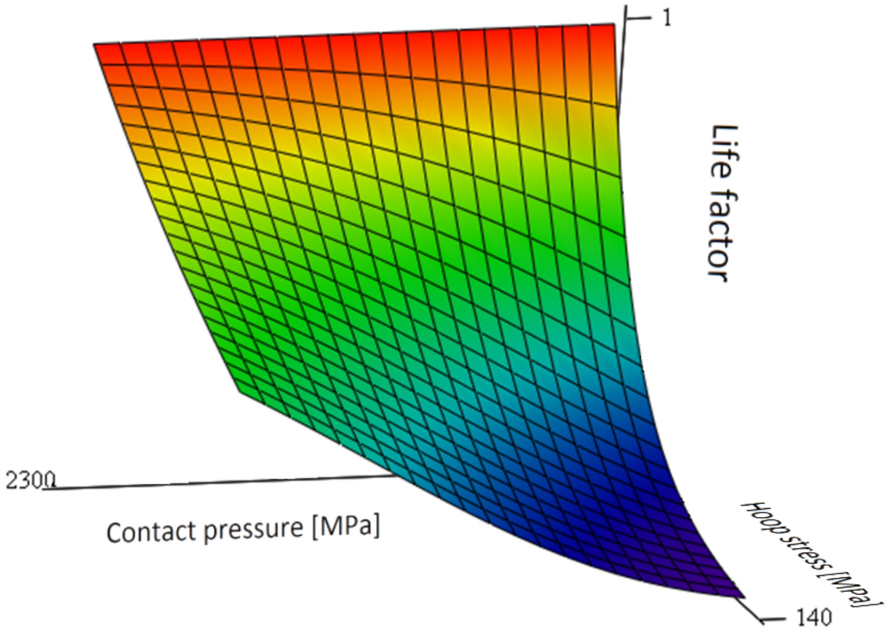

Figure 5.8: Shows the surface which describes the life time factor as a function of the hoop stress and the contact pressure for bearing 24072.

5.2 Minor study

Based on the thoroughly performed pilot study, the procedure could be followed as planned. In the minor study, a larger number of bearings were going to be evaluated based on the result from the pilot study. What was more interesting to evaluate during the minor study, was the relationship between different series, bore diameters and widths.

5.2.1 Measure

As BEAST now had good enough settings to perform a proper analysis, based on the result from the pilot study, no further adjustments of the settings were made. The levels of the input parameters had also been proven to be suitable to obtain reasonable results. Hence no adjustments or corrections of these levels were made either.

In order to collect data to be able to analyse the series, bore diameter and width effect, bearings of different proportions had to be evaluated. This was done by selecting two areas of bearings, one area of smaller bearings and one area of larger bearings. Within both of the areas, bearings with different series, bore diameter and width were chosen. The chosen bearings will not be listed but the principle can be seen in Appendix, Figure A.8 how they were chosen on a principle level. The two areas the bearings were taken from are both shaped as crosses. This was because of how the original matrix was built up, to make it easy to visualise the differences in series, bore diameter and width. As the original matrix cannot be shown, the selection will only be shown on a principle level from here on. When collecting data for measuring the effect within the series, the bearings that lay vertically in the matrix were simulated. To measure the effect of having the same bore diameter, the bearings that lay horizontally in the matrix were simulated instead. For the effect over the width on the other hand, two in a row in the horizontal direction were picked for evaluation. By picking the bearings like this, according to the matrix, it was ideal for the analyse phase since all combinations were gathered and could be analysed separately. The number of simulations that had to be performed were also minimised by choosing the bearings in this way.

5.2.2 Analyse - Effects within series

To analyse the effect over the same series, the bearings laying in the same column accordingly to Appendix, Figure A.8 were evaluated. The result from this analysis for a thick series and a contact pressure of 1900 MPa can be seen in Figure 5.9. Interesting enough was that all of the bearings within that series gave the same result. They all had the same life time factor with a given hoop stress which can be seen as all of the lines are located on top of each other and hence appear to only show one line. From this figure, it seemed as there was no effect between bearings within the same series.

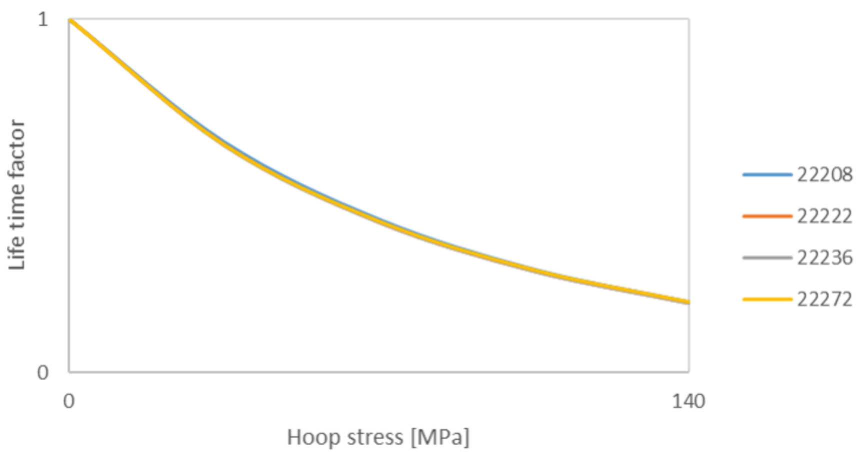

Figure 5.9: Shows the hoop stress effect for smaller bearings belonging to series 222 with a contact pressure of 1900 MPa.

In order to be able to compare the results with each other, the same contact pressure was needed. The data had been collected for four different levels of contact pressure but due the large amount of data and to ease the evaluation, comparisons at a fixed contact pressure was decided to be presented in this report as it was representative for all of the other contact pressures in the minor study. This could be done since the behaviour between the contact pressures did not differentiate. The behaviour was rather identical as the lines in the both plots were located on top of each other which can be seen by comparing the behaviour in Figure 5.9 with a contact pressure of 1900 MPa and the behaviour in Figure 5.10 with a contact pressure of 1700 MPa instead.

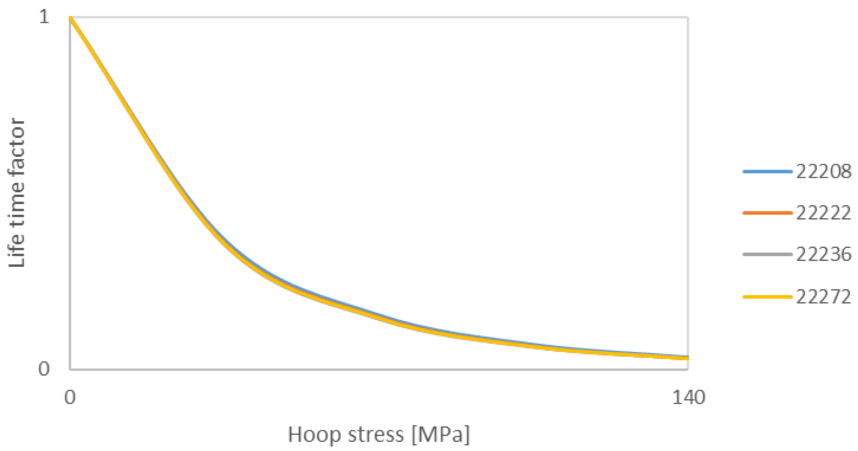

Figure 5.10: Shows the hoop stress effect for smaller bearings belonging to series 222 with a contact pressure of 1700 MPa.

The same procedure followed by creating plots for all of the other cases and from this, it could with certainty be said that all of the bearings were following the same behaviour for different contact pressures. Hence only plots corresponding to a contact pressure of 1900 MPa will be presented further on in the minor study even though plots have been produced for the other contact pressures as well. From the recently shown result, it was expected for the larger bearings to follow the same behaviour as for the smaller bearings, to be located on top of each other in the plots. Even more interestingly, based on the previous result, was that they did not as can be seen in Figure 5.11. The smaller bearings in the series were a bit more packed together meanwhile the largest bearing was distinctly divergent. However, even the smaller bearings within the series were not as packed as for the bearings in Figure 5.9. Worth mentioning is that the smallest bearing in the series had a larger life time factor reduction than the closest bearing in size. A potential effect between sizes had been found but this had to be investigated further in order to be able to draw any final conclusions since it was not clear due to the previously mentioned phenomenon.

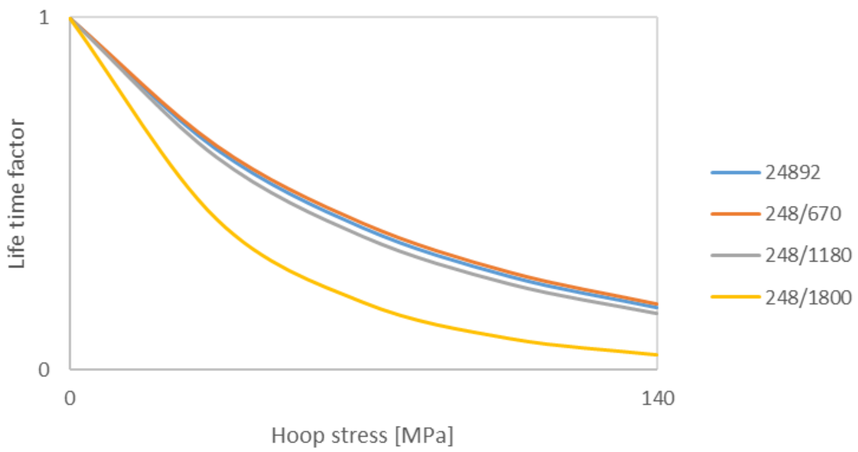

Figure 5.11: Shows the hoop stress effect for larger bearings belonging to series 248 with a contact pressure of 1900 MPa.[2]

5.2.3 Analyse - Effects over bore diameter

The same procedure followed for the analyse regarding bearings with the same bore diameter. As for the effect within the series, the result between different contact pressures did not differ and thus only results with a contact pressure of 1900 MPa are presented. For the sample of bearings with a bore diameter of 110 mm, the result turned out as seen in Figure 5.12. Likewise did this result match the behaviour when measuring within series for smaller bearings, the curves were located on top of each other.

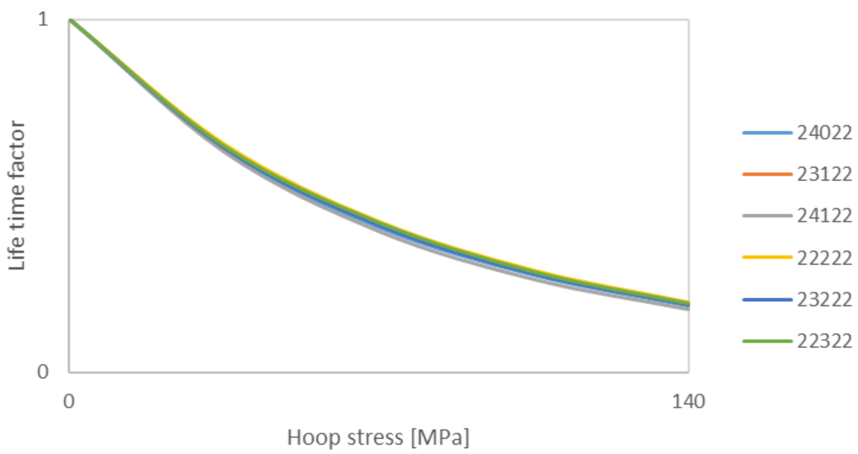

Figure 5.12: Shows the hoop stress effect for bearings with a bore diameter of 110 mm and a contact pressure of 1900 MPa.

[2]This result is updated further on in the study.

The question then was whether a similar result would be obtained for larger bearings as well or if they would diverge like they had done when analysing the effect within series. The result can be seen in Figure 5.13 and it turns out that they were following the secondary option. The larger bearings did diverge from each other for this case also.

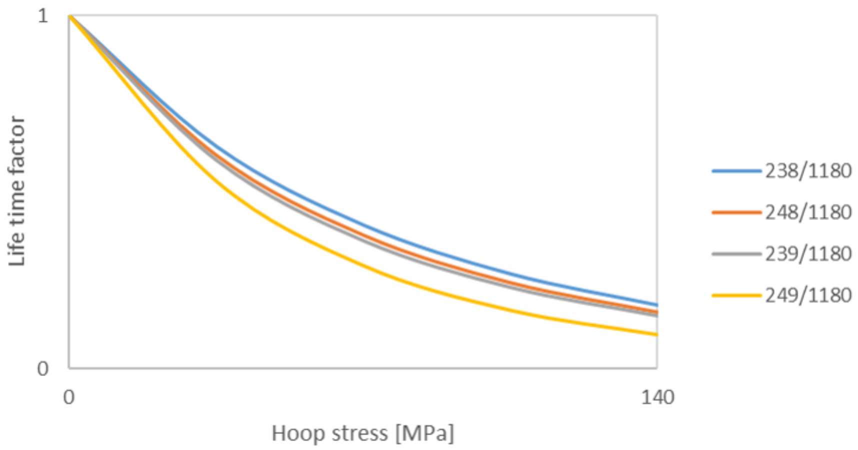

Figure 5.13: Shows the hoop stress effect for bearings with a bore diameter of 1180 mm and a contact pressure of 1900 MPa.[3]

5.2.4 Analyse - Effects of width

As had been established from both of the previous analyses, it seemed as there was no effect influencing the result for smaller bearings. For larger bearings on the other hand, some sort of effect seemed to influence the result. To evaluate this further but also to see the effect of having the same outer diameter of bearings, the effects of having different widths were evaluated next. In order to see the width effect, two bearings with the same outer diameter but with different widths were compared against each other. This resulted in five comparisons in total, three for the smaller bearings and two for the larger bearings. Based on the previous results, a new hypothesis was stated. It was assumed that for the smaller bearings, no influence would be seen meanwhile for the larger bearings, they would diverge from each other. A comparison between two smaller bearings and a comparison between two larger bearings can be seen in Figure 5.14 and Figure 5.15 respectively. The same behaviour was obtained for the other comparisons respectively as well and are thus not shown in this report.

[3]This result is updated further on in the study.

What could be seen was as expected, the life time factor for smaller bearings was the same meanwhile for the larger bearings, a smaller deviation was seen. For the two other comparisons in regard to smaller bearings, their curves were located on top of each other and for the other comparison in regard to the larger bearings, another deviation was seen which confirmed the hypothesis.

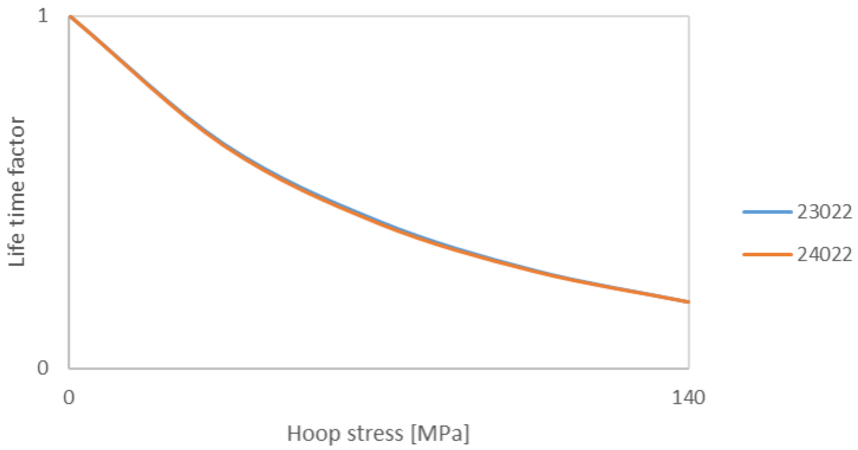

Figure 5.14: Shows the hoop stress effect for smaller bearings with the same bore diameter of 110 mm and a contact pressure of 1900 MPa.

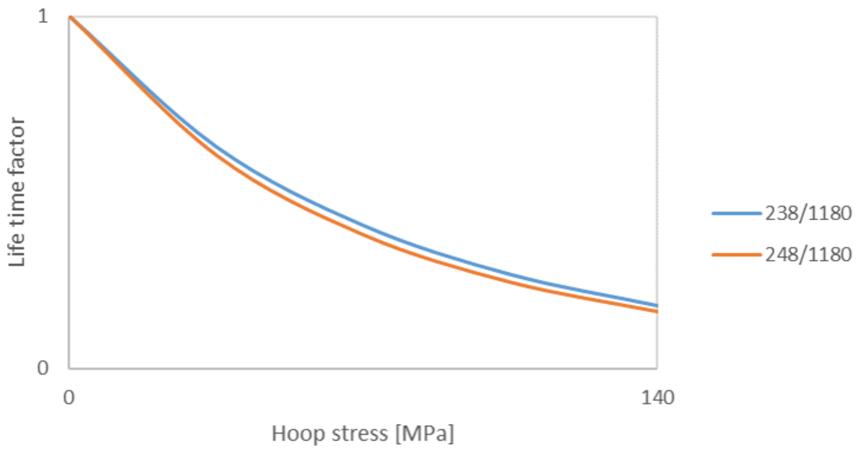

Figure 5.15: Shows the hoop stress effect for larger bearings with the same bore diameter of 1180 mm and a contact pressure of 1900 MPa.[4]

[4]This result is updated further on in the study.

5.2.5 Conclusion of minor study

For the smaller bearings, the conclusion could be drawn that the lines in the plots were located on top of each other for all cases. This meant there was no difference in life time factor for a certain hoop stress between smaller bearings. On the other hand, the opposite seemed to be true for the larger bearings. As the curves for the larger bearings were diverging from each other, the phenomenon behind had to be investigated further to identify the root cause. To understand and find answers for this behaviour, more and deeper investigations had to be done in the major study.

5.3 Major study

As stated in the minor study, it seemed to be a different behaviour between larger bearings and smaller bearings. Based on this understanding, the major study was going to include investigations of which point the effect starts to affect the result. In addition to this, it was also said that the cause of this behaviour had to be investigated to obtain a deeper understanding of the subject. In order to answer these questions and draw reasonable conclusions, the major study would include bearings over the whole assortment.

5.3.1 Measure

In the major study the same settings for the BEAST simulations were used as before, since it showed stable results. The same levels of the input parameters were used here as well for the same reasons.

In order to cover the whole assortment as effectively as possible, it was decided to choose bearings in each series with reasonable distance between them. The selected bearings of each series were matched with the other series so that conclusions about series, bore diameter and width could be drawn. The principle of how the bearings were chosen can be seen in Appendix, Figure A.9. By choosing this configuration of bearings, the number of simulation were minimised in relation to the information generated. A meta model would be created for every bearing in the same manner as in Section 5.1.4 and Section 5.1.5.

5.3.2 Analyse

When comparing all of the bearings in the major study at the same time, there seemed to be an infliction point depending both on the bore diameter and the volume of the bearings, visualized in Appendix, Figure A.10. All bearings above this infliction point seemed to behave similar, but under it, they behaved differently. The hoop stress seemed to have larger effect on the bearings below this infliction point. The most critical bearings seemed to be the bearings with large inner diameter and a thick profile as for example the bearing 241/1250, which was the most divergent bearing in this study. All of the diverging bearings were then identified and marked in the matrix of the assortment. The principle result of this can be seen in Appendix, Figure A.11. Worth mentioning in this context is that the area seemed to represent a border of bearings limited to the southeast corner. The southeast corner of the matrix corresponds to bearings with large bore diameters and relatively thick inner rings, a large volume with other words.

A common thing between the bearings in Figure A.11, except from the divergent behaviour, was that for all of the bearings, it had been difficult to iterate a desired contact pressure. Even though an error close to zero had been obtained, the next iteration could give a larger error than the previous. This was remarkable already when this was discovered but since the error was close to zero, it was assumed to be good enough and could not be improved. During the iteration of the smaller bearings, they did converge rapidly to a negligible error and this problem was not seen. Since it was the same bearings that showed confusing results for the both cases, a deeper investigation of the problem was initiated. After several meetings with experienced personnel at SKF, it seemed as the problem was regarding unstable calculations of the contact pressure. The value had not yet stabilised as it was measured. This can be seen in Figure 5.16. For the larger bearings, it took longer time to stabilise and hence a value was measured early in the simulation where the contact pressure had not yet stabilised. This would also explain why it was difficult to iterated the exact contact pressure. By measure the contact pressure at a later time during the simulation, a more exact contact pressure was expected to be obtained.

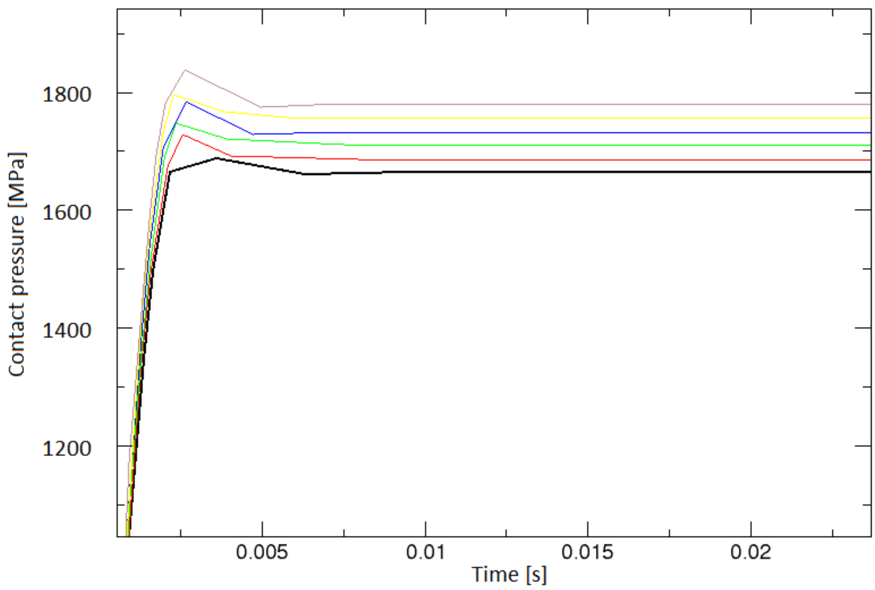

Figure 5.16: Shows how the contact pressure is fluctuating before the simulation stabilises and an exact value can be measured. The lines represents different loading conditions.

The variable "StartWriteTime" was the one which had to be adjusted. By setting this variable to 0.2*SimTime, a good result was expected. This means the measurement would be done at 20% of the total simulation time, which would be within good margins to find a stabile value. A test run with bearing 232/670 was done to see how the new settings would perform. This bearing was chosen since it was one of the problematic bearings, according to A.11, and was a bit faster to simulate than the others. The result can be seen in Figure 5.17.

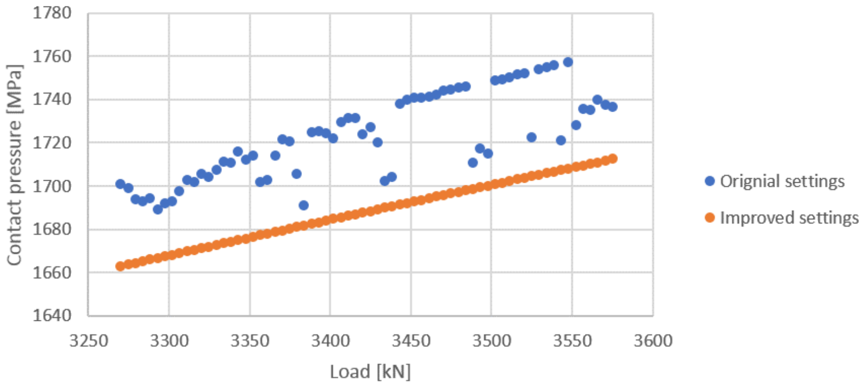

Figure 5.17: Shows how the contact pressure is differing between the two cases of settings for bearing 232/670.

As one can see, the improved settings gave a much more reliable result. Hence it was decided upon to restart the simulations of the diverging bearings to see if a better result could be obtained. When the data had been collected, the same procedure followed as for creating Figure A.10, seen in Appendix. The new result of this compilation can be seen in Appendix as Figure A.12. All of the bearings had now converged to a single curve, meaning they all would be able to be described by one meta model.

5.3.3 Conclusion of major study

Based on the previous result, it was understood that the data for all bearings converged, after some additional tuning of the model. What was seen in the minor study regarding diverging bearings was hence a cause of unstable calculations. Even if this change would have been discovered earlier, all of the tests would have to be performed anyway to determine that there were no differences between the bearings in regard to series, diameter and width. Since the new result showed that all bearings converged, this meant that a single meta model was possible to create. The question was still whether the meta model would be bearing specific or valid for all of the bearings in a common meta model. A bearing specific model was expected to have higher accuracy but would also be a bit slower to use since a lot of bearing specific parameters would have to be inserted into the model by the user. A common model for all of the bearings would instead be more manageable since only one model would be needed with already produced constants to match the behaviour as good as possible. The disadvantage though would probably be a bit lower accuracy for some of the bearings.

5.4 Final meta model - Scientific level

As elaborated in the previous section, due to the similar behaviour, a common meta model was possible to create. The question was still though if this was better than the other option of having a bearing specific model. A more investigative analysis was needed in order to evaluate this. To do this, the data points from the major study would have to build the different models. Hence, curve fittings were created for all bearings included in the major study and the constants were compiled. The methodology which was used to produce this data is presented in Section 5.1.4 and Section 5.1.5.

In contradiction to the first level, where the a constant was set to 1 due to the boundary condition given by the life time factor function, the similar constant for the second level, c, was instead investigated if it could be optimised to obtain a better result. This was done by not locking it to 1 when creating the curve fits. The same boundary condition applies in practice for the second level as well as for the first level which meant that the case when c was locked to 1 had to be calculated as well. With other words, there would be four different variations to evaluate. First and foremost if a common or a specific model would be better and secondly whether an optimised value of c would improve the meta model. The four cases are listed next.

- Alternative 1: c=1 and d is optimised for each bearing and then made an average for a common model
- Alternative 2: c and d are optimised for each bearing and then made an average for a common model
- Alternative 3: c=1 and d is optimised for each bearing and remains bearing specific
- Alternative 4: c and d are optimised for each bearing and remains bearing specific

For Alternative 1 and 3, the curve fits were created with c locked to 1 and d as a free variable. The factor d was then made an average over all of the obtained d values from the process for Alternative 1 meanwhile for Alternative 3, the d factor was a parameter, depending on the bearing. The same procedure followed for Alternative 2 and 4 but instead of locking c to 1, it was free. The framework which was used for this task can be seen in Figure 5.18.

XXXXX

Life time	Hoop level 0 [MPa]	Hoop level 1 [MPa]	Hoop level 2 [MPa]	Hoop level 3 [MPa]	Hoop level 4 [MPa]	a	b	c1	d1	ε1
1700 [MPa]	The Calculated life time in BEAST. * (The life time is going towards the infinity, hence contact pressures at these levels are not of intrest. Instead surface fatigue is the limiting factor.)					Curve fit	Curve fit hoop stress	Curve fit contact pressure		
1900 [MPa]										
2100 [MPa]								c2	d2	ε2
2300 [MPa]								Curve fit contact pressure		

Pmax [MPa]	P_Max [Pa]	P_Max [MPa]	P_Max_Error	P_Max_Error_%	Fr [N]	Fr [kN]	C	P/C
1700	Contact pressure Max		Contact pressure Error		Load input BEAST		C value	P/C factor
1900								
2100								
2300								

Life Factor	Hoop level 0 [MPa]	Hoop level 1 [MPa]	Hoop level 2 [MPa]	Hoop level 3 [MPa]	Hoop level 4 [MPa]
1700 [MPa]	Life time factor				
1900 [MPa]					
2100 [MPa]					
2300 [MPa]					

Figure 5.18: Shows the final configuration which was used to handle the collected data.

A meta model for each of the four cases was created and compared against the data from BEAST. To evaluate the best option of the four, the error was calculated according to Equation 3.15. The total least square error for each of the alternative models is listed further on. The smallest error was obtained for Alternative 1 and hence chosen to continue with. A bit surprising was that Alternative 1 was the best since only one parameter was optimised and then made an average, while other alternatives were optimised for two parameters and additionally were bearing specific. This means it was not better to optimise the model by letting c be a free variable. By instead keeping it locked to the boundary condition, as it should have been, was the best alternative.

- Alternative 1: 4,105 %
- Alternative 2: 15,739 %
- Alternative 3: 6,383 %
- Alternative 4: 4,427 %

Since the difference in error between Alternative 1 and 4 was so small, one could argue that more investigations were needed to do a definite choice. But with the knowledge that a bearing specific model would be much more demanding on the user, everything points at a common model, as Alternative 1, was the best.

The result of the final meta model for the scientific level can be seen in Equation 5.4, where d was obtained as a constant but cannot be shown exactly due to confidentiality and c as fixated to 1. A very important note to mention in this context is that the model and its data points are only valid for the given levels, with other words from 0 MPa to 140 MPa of hoop stress and 1700 MPa to 2300 MPa of contact pressure. This is due to the fact that the model was built upon these levels and values outside of these levels cannot be guaranteed to be valid. Since all of the bearings were showing the same result with minor differences, a general example is shown in Example 5.1 which will apply for all bearings.

$$LF(\sigma_\varphi, p_{\max}) = e^{-(e^{-(d \cdot p_{\max})})\sigma_\varphi} \quad (5.4)$$

Example 5.1

<u>Question:</u> A project team is investigating how the life time of a specific bearing is affected by the hoop stress. They want to know how much the life time is reduced at the levels of 2000 MPa contact pressure and 100 MPa hoop stress.

<u>Answer:</u> The common model is used as described in previous section, see Equation 5.4. The d value is assumed to be 0,003 in this case.

$$\begin{cases} c = 1 \\ d = 0{,}003 \end{cases}$$

Insertion into the life time model, one get the following equation.

$$LF(\sigma_\varphi, p_{\max}) = e^{-(e^{-(0,003\, p_{\max})})\sigma_\varphi}$$

The life time is acquired at.

$$\begin{cases} p_{\max} = 2000\ MPa \\ \sigma_\varphi = 100\ MPa \end{cases}$$

The life time factor then becomes.

$$LF(100, 2000) = 0{,}78$$

The life time is reduced by 22%, in relation to not having any hoop stress.

Conclusions about the life time reduction can be drawn when comparing against the absolute life time. For this case, the L_{10} formula is used, see Equation 3.1. The load at 2000 MPa for the bearing is around 94 kN, the c value is used but not mentioned in this official report. Inserting this in the L_{10} formula gives the life time of 1,6*10^8 revolutions, decreasing this life time with 22% gives 1,25*10^8 revolutions. This show at what magnitude the life time is reduced.

If the bearing is running at 20% of the reference speed 3400 r/min, the bearing will last about 3921 operating hours without hoop stress effect and 3058 operating hours with the hoop stress effect. Worth mentioning is that a contact pressure of 2000 MPa contact pressure is very high because it represents around 0,2 P/C.

5.4.1 Verification of the scientific model

To verify the accuracy of the constructed scientific model, a few bearings were chosen to act as verification samples, see Appendix Figure A.13. As can be seen in the principle matrix, they were chosen randomly but they were also aimed to not be too close to each other. The general procedure was performed for these bearings but instead of iterating the previous levels of contact pressure, the iteration was aimed for values between the ordinary levels that the meta model was built upon. This was done to verify both other bearings but also other levels of contact pressure. A full factorial design was executed for all of the samples. The aimed contact pressure are listed below.

- Pressure level 1 = 1800 MPa
- Pressure level 2 = 2000 MPa
- Pressure level 3 = 2200 MPa
- Pressure level 4 = 2400 MPa

Same goes for the levels of the hoop stress, these were also selected between the parameter levels. The reference point was still needed in order to produce the life time factor according to Equation 4.1. The life time factors calculated in BEAST were then compared with the scientific meta model. The result of the verification process for a contact pressure of 2000 MPa can be seen in Figure 5.19. From the figure, one can see that the model was performing good for the level of 2000 MPa contact pressure. The data points did not differ much from the meta model even though a small variation existed. A similar result was obtained for the contact pressures of 1800 MPa and 2200 MPa, which can be seen in Appendix, Figure A.14 and Figure A.15 respectively.

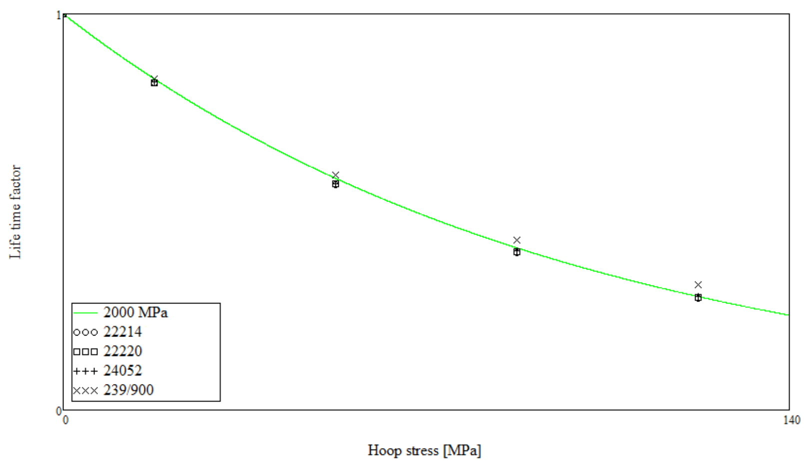

Figure 5.19: Shows how the meta model is performing for bearings the model was not built upon with a contact pressure of 2000 MPa.

When evaluating the meta model for the contact pressure of 2400 MPa, a larger error was obtained which can be seen in Figure 5.20. This is due to the boundaries of the model which it was built upon. 2400 MPa is located outside of the original boundaries of 1700 MPa to 2300 MPa which may explain the larger error obtained. The least square error method was used to evaluate the result in more detail and was done for all of the sample points. The error was calculated for two cases, first only including the contact pressures within the original boundaries. The second calculation included all four contact pressures. The final error for the first case was obtained as 3,1%. Still some variations were existing but as a whole, this was considered as an acceptable error, in addition to the benefits of having a common model for the whole assortment. For the second case, the error was 16,1%. By verifying against values outside of the model, the error rate increased fast. This means one should be cautious when using the model outside of the stated boundaries due to the expected large error.

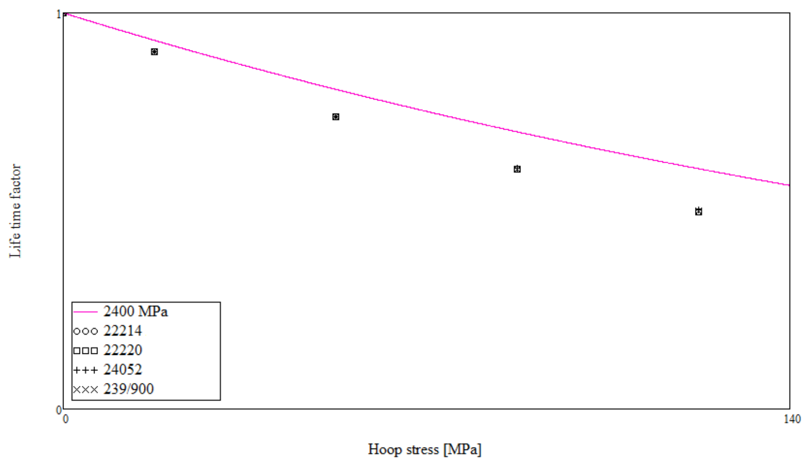

Figure 5.20: Shows how the meta model is performing for bearings the model was not built upon with a contact pressure of 2400 MPa.

5.5 Transformation - Hoop stress to interference

Once the final meta model for the scientific level had been settled, it had to be transformed to suit for the engineering level. As have been elaborated in Section 3.4.3, the hoop stress can be described by Equation 3.13. This equation was used to transform the meta model from the scientific level to the meta model of the engineering level. In addition to this it was investigated how the bearings over the whole assortment vary in relation to the interference instead of the hoop stress.

5.5.1 Measure

As mentioned, the transformation was performed with the help of Equation 3.13. According to the equation, the hoop stress at the surface is dependent on the relation between Δ/d, k and Young's modulus. Young's modulus was in this study assumed to be constant and was set to be 207 GPa [9]. The k factor was varying over the assortment and Δ/d was set as the pre-determined parametric levels. These levels were chosen between 0 and 0,08% Δ/d, as described in Section 3.4.4. For this evaluation, the whole assortment of all available bearings were considered instead to find the extreme values and be able to draw reasonable conclusion regarding this matter as well.

5.5.2 Analyse

The k factor was the part of the equation which varied without prediction due to the many factors affecting k. The result showed that it varied between 0,64 and 0,94. It turned out to be a factor of approximately 2 when comparing the least affected bearing with the most affected bearing. The least affected bearing was 21304, this was due to the thickness of the inner ring and the small contact angle to the point where the stress concentration was located. The most effected bearing instead was 248/1180 due to its large bore diameter and the large angle. From the matrix, it was understood that the largest and smallest bearings did not necessarily have to be the extreme cases in this matter, since for example 248/1800 is larger than 248/1180. The tendency though was such that with increasing bore diameter and thinner inner ring, the higher was the hoop stress but this was not distinct due to the variation in angles and k.

When analysing the hoop stress levels that occur when varying Δ/d levels from 0 to 0,08%, the least affected bearing 21304 varies from approximately 0 to 75 MPa, meanwhile the most affected bearing, 248/1180, varies approximately between 0 to 150 MPa. The factor could from here be verified by dividing 150/75 and receive a value of 2. In Figure 5.21, one can see the two extreme cases. In between the two lines, all of the other bearings would be located.

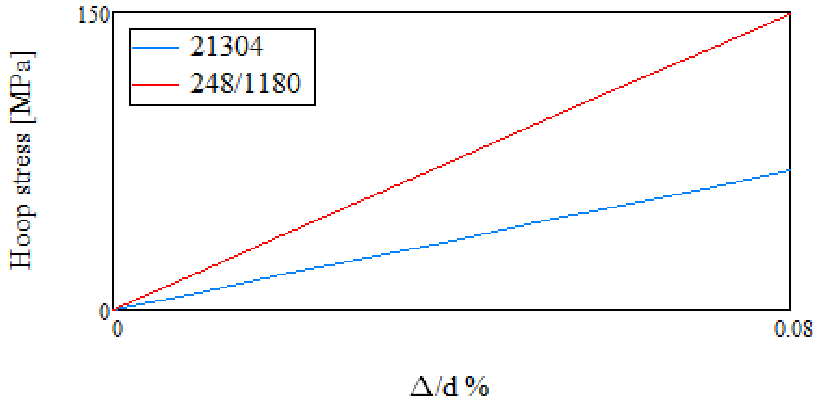

Figure 5.21: Shows the hoop stress levels increase with increasing Δ/d for the bearings 21304 and 248/1180.

5.6 Transformation - Contact pressure to load

Once the transformation from the hoop stress to interference was established, the relation between loads and the contact pressure was investigated. To do this, a model developed within SKF was used. The model is based on the Hertz theory considering contacts between surfaces. This theory is elaborated in more detail in Section 3.2.2.

5.6.1 Measure

The model which was used to transform the loads to contact pressure is based in a SKF program. Similar principles that are used for all the BEAST simulations are also implemented in this software. The Mathcad program needed geometrical properties of the bearing which was studied, these properties were imported from the SKF product database. The generated output was the maximum contact pressure between the most loaded roller and the inner ring, the same parameter that was used to develop the meta model in the scientific level. The levels for the loads that were investigated were chosen between 0,1 and 0,5 P/C to collect cases over the whole assortment and are listed next.

- Load level 1: 0,1 P/C
- Load level 2: 0,2 P/C
- Load level 3: 0,3 P/C
- Load level 3: 0,4 P/C
- Load level 4: 0,5 P/C

The generated contact pressures were then inserted into the scientific level model, together with the interference levels listed in section 5.5.1. The investigated bearings were the same bearings that were investigated in the major study.

At first, the analysis was taking all levels of contact pressures and loads into consideration. This was to investigate the general behaviour of the assortment. Later on, more realistic assumptions were taken into consideration, to elaborate more practical cases. At the later stages, all contact pressures below 1700 MPa were excluded since these low levels of contact pressure gave unrealistically long life times, which makes this area of less importance. A bearing will most certainly fail due to other causes before it reaches these levels, as for example due to surface fatigue. The levels of the loads were also restricted during the later analysis to only consider maximum 0,3 P/C. This level was chosen after discussions with the different stakeholders.

5.6.2 Analyse

The general behaviour that was identified for the assortment was that for bearings with low levels of contact pressure, the hoop stress had a larger relative impact than for the bearings with high levels of contact pressure. Generally bearings with a small bore diameter had larger levels of contact pressure at the set P/C levels which made the interference of less importance. An example of this can be seen in Figure 5.22 for bearing 22308. By performing the same analysis for another bearing with larger diameter, as for example 22372, the contact pressure decreased and the effect of the interference had higher impact, as seen in Figure 5.23.

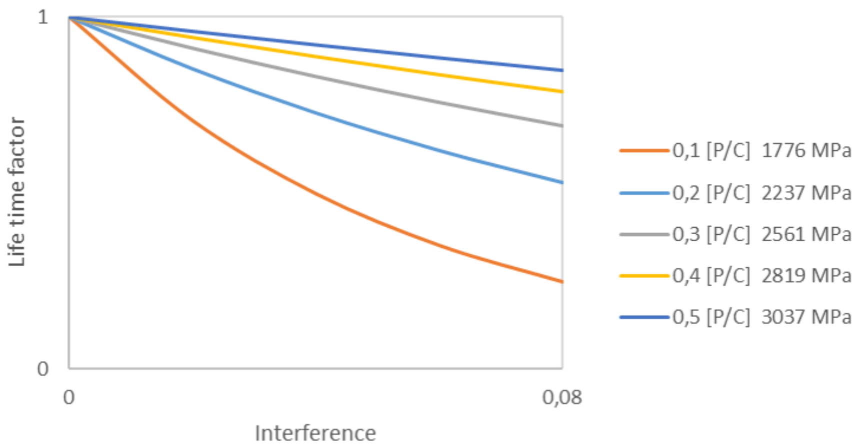

Figure 5.22: Shows the evaluation of bearing 22308 at the set engineering levels of load and interference.

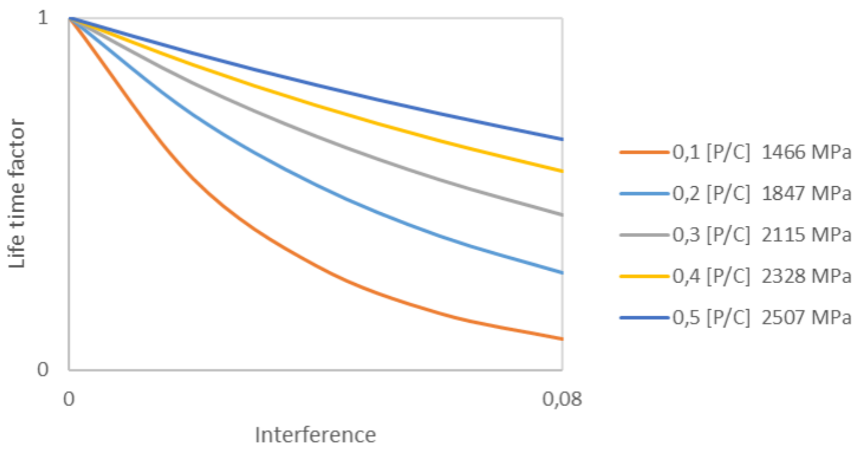

Figure 5.23: Shows the evaluation of bearing 22372 at the set engineering levels of load and interference.

The same behavior appears when comparing over the series, thin profiled series as for example series 239 had lower contact pressures than for example the 223 series. This made the thinner series more affected by the interference, as show for 23972 in Figure 5.24 when comparing against Figure 5.23.

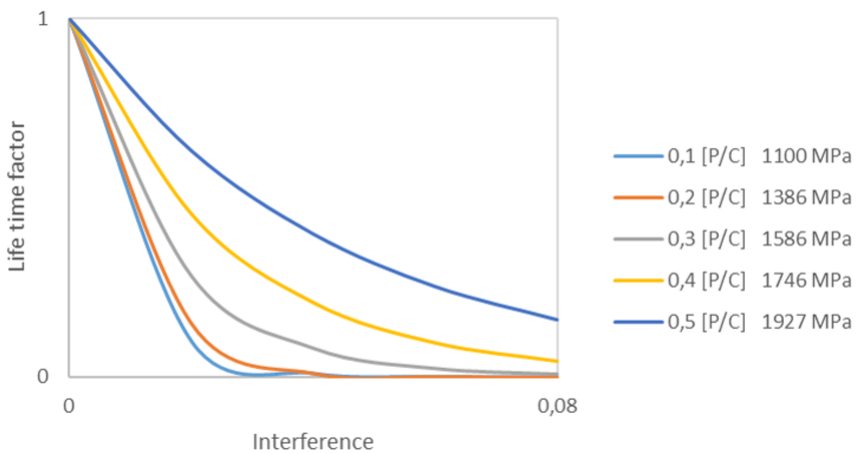

Figure 5.24: Shows the evaluation of bearing 23972 at the set engineering levels of load and interference.

The bearings with a thin profile and a large bore diameter then ultimately gets the highest effect of the interference as a consequence. As seen for example in Figure 5.25 when evaluating 238/1180. What was noticeable here was that these bearings had very low levels of contact pressure, which makes them in practice not of interest because of the long life time they will sustain. With other words one can say that the stress levels are below sub-surface fatigue limit.

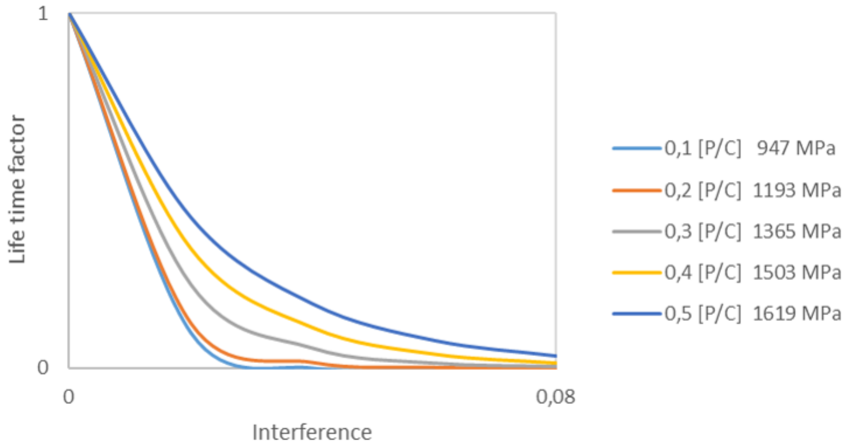

Figure 5.25: Shows the evaluation of bearing 238/1180 at the set engineering levels of load and interference.

From here on, more realistic levels of the contact pressure and loads were evaluated since many of the bearings were not reasonable to evaluate for these occurring levels. It was decided that bearings with a lower value than 1700 MPa of contact pressure between 0,1 and 0,3 P/C would be excluded since they could be considered to have infinite life time in regard to sub-surface fatigue. The principle schematic of these bearings can be seen in Appendix, Figure A.16, marked with a light blue colour. The bearings of most interest were hence the bearings closest to the blue area and the bearings furthest away from the blue area but for two different reasons. The bearings closest to the blue area are more affected by the interference, but the contact pressure is still relatively low for the relatively high levels of loads. For example was 0,3 P/C the only eligible level of load for bearing 241/670. This is shown in Figure 5.26. On the other hand, bearings further away from the blue area were of interest for all the levels of loads, as seen for bearing 22308 in Figure 5.27.

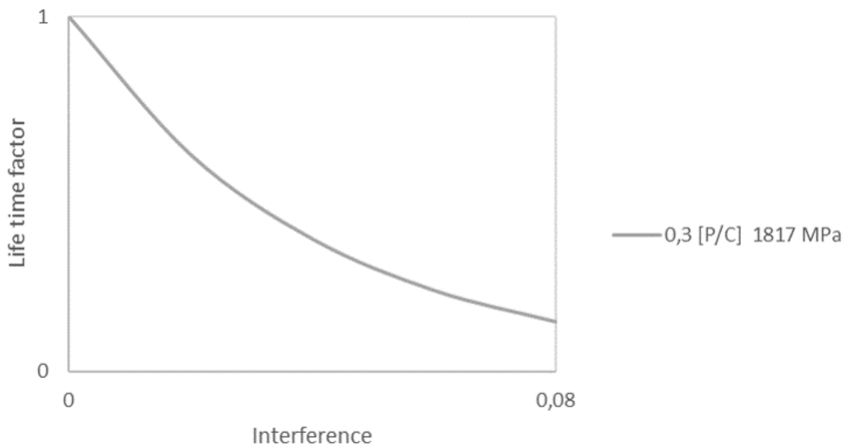

Figure 5.26: Shows the evaluation of bearing 241/670 at the set engineering levels of load and interference.

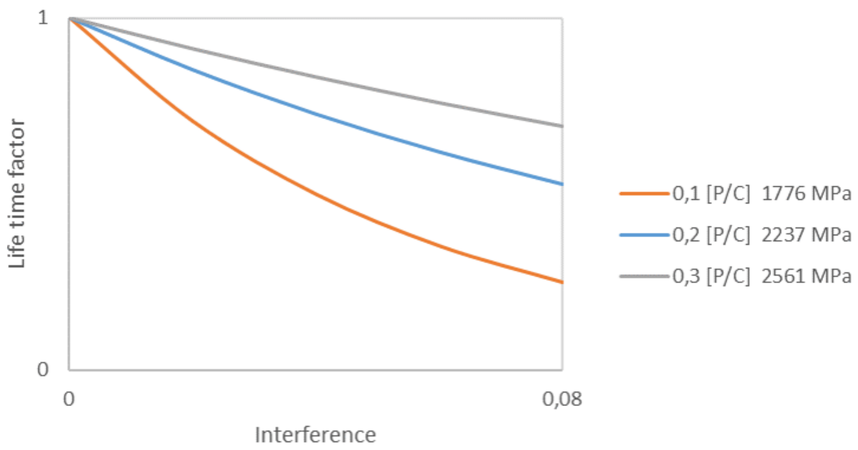

Figure 5.27: Shows the evaluation of bearing 22308 at the set engineering levels of load and interference.

One conclusion that could be drawn from this analysis was that small and thick bearings had a higher contact pressure at reasonable loads, which makes the interference interesting both for low and high loads. For large and thin bearings, the obtained contact pressure was relatively low which makes the interference important only for high loads. If one not consider the reasonable assumptions which have been made in regard to load and contact pressure, the general behaviour of the hoop stress can be seen in Appendix, Figure A.17. What can be seen is that for a lower contact pressure, the hoop stress has a larger impact on the performance.

Relations to the seen behaviour may be drawn to the factor C_{ISO}, described in Section 3.2.1. The number of rollers generally increases for a bearing with a larger bore diameter and a thinner profile, which decreases the C_{ISO} value according to the equation. The same behaviour applies for the roller length. But the roller diameter on the other hand, is increasing C_{ISO}. This behaviour could be seen by transforming the loads, from P/C to contact pressure. Generally bearings with many rollers, with small roller diameter and long roller length got the lowest values of contact pressure in relation to the set P/C levels. The contrary applies for bearings with few rollers, large roller diameter and short roller length. This outcome indicates that it is the variation of C_{ISO} which is seen throughout the study.

When looking at absolute values, a typical life time for bearings at 1700 MPa from 0 to 0,08% interference was between 10^9 to 10^7 revolutions which is a very long life time even though the reduction was relativity high. Instead the life time at 2300 MPa from 0 to 0,08% interference was from 10^4 to 10^3 revolutions which is instead a short life time. So the life time factor may be deceiving regarding where the actual point of interest may be, and must be taken into consideration when drawing conclusions.

5.7 Final meta model - Engineering level

The final meta model for the engineering level consists of three parts, visualised in Figure 5.28. First, there is the scientific model with the life time factor as an output. The contact pressure and the hoop stress are inputs to that model. The contact pressures were generated based on the desired loads through the Hertz theory with help from the Mathcad program. The hoop stress was transformed through the analytic equation, described in Section 3.4.4. In Example 5.2 it is exemplified how the engineering model can be used.

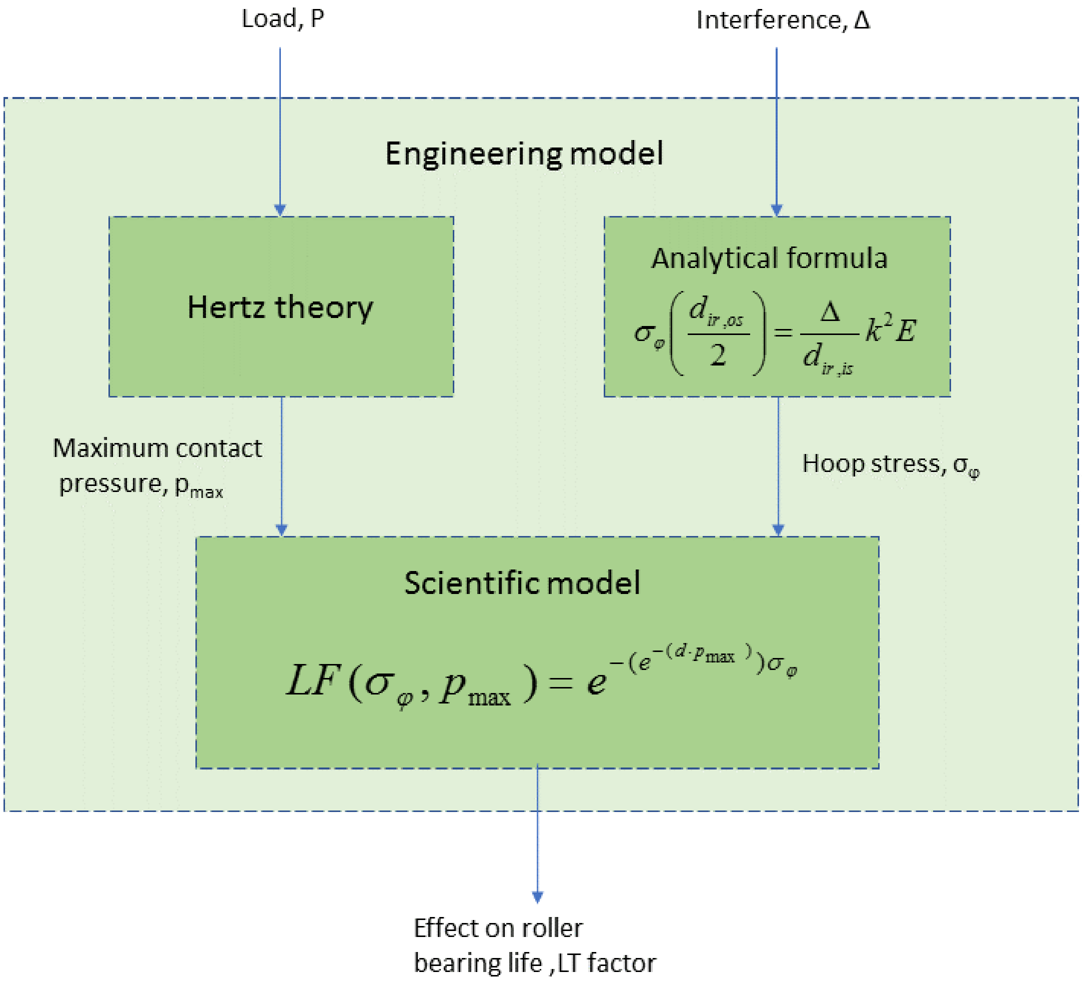

Figure 5.28: Shows the principles of the final meta model for the engineering level.

Example 5.2

Question: A project team are investigating how the life time of a specific bearing is affected by the Interference fit. They want to know how much the life time is reduced at the levels of a heavy load at 0,2 P/C and interference fit class, n5 (theoretical interference).

Answer: The common model is used as described in previous section, see Equation 5.4. Insertion in the life time model, one get the following equation.

$$LF(\sigma_\varphi, p_{\max}) = e^{-(e^{-(0{,}003\, p_{\max})})\sigma_\varphi}$$

A transformation from load to contact pressure is needed and for this the Hertz theory is used based in a Mathcad program developed within SKF. The calculated contact pressure then becomes.

$$Load = 0{,}2\ P/C \rightarrow p_{\max} = 2150\,MPa$$

The hoop stress is transformed by the analytical function presented in section 3.4.3. The inserted parameters are selected accordingly. Assumed diameter $d_{ir,os}$ for critical contact point is calculated using trigonometry. The interference Δ is obtained in the SKF Catalogue.

$$\begin{cases} E = 207000\,MPa \\ d_{ir,is} = 40\ mm \\ d_{ir,os} = 51\,mm \\ k_{ir} = 0{,}784 \\ \Delta = 40\ \mu m \end{cases}$$

$$\sigma_\varphi\left(\frac{d_{ir,os}}{2}\right) = \frac{\Delta}{d_{ir,is}} k^2 E \rightarrow \sigma_\varphi\left(\frac{51}{2}\right) = \frac{0{,}04}{40} \cdot 0{,}784^2 \cdot 207000 = 127\ MPa$$

Insertion in the life time model gives.

$$LF(127{,}2150) = 0{,}82$$

The life time is reduced by 18%, at 0,2 P/C and interference fit class, n5.

Conclusions about the life time reduction can be drawn when comparing against the absolute life time. For this case, the L_{10} formula is used, see Equation 3.1. The load at 2150 MPa for the specific bearing is around 20 kN, the C value is not mentioned due to confidentiality. Inserting this into the L_{10} formula gives the life time of 2,1*10^8 revolutions, decreasing this life time with 18% gives 1,72*10^8 revolutions. This shows at what magnitude the life time is reduced.

If the bearing is running at 20% of the reference speed 8000 r/min, the bearing will last about 445 operating hours without hoop stress effect and 365 operating hours with the hoop stress effect.

5.8 Summary of result

A common meta model for the whole assortment of spherical roller bearings could be created and be valid for the scientific level. Despite what was believed in the beginning of the project, that a common meta model would not be able to be created for the scientific model and in addition, be better than a bearing specific model, it was in the end. For the engineering level on the other hand, it was not possible to conclude a common meta model but it rather had to be bearing specific due to the unique design of all bearings. When transforming the hoop stress to the interference, an analytic formulation was used. For the transformation from contact pressure to load, a significant relationship could not be found. Instead of then creating a new model, trying to transform the contact pressure to the load as good as possible, the already existing Mathcad tool which SKF already used was chosen for this purpose. Even though a common model could not be created for the engineering level, the result is considered successful since the meta model can reproduce the result from the BEAST model with only a small error in a fraction of time of what the BEAST model can do. The final common meta model for the scientific level can be seen in Figure 5.29. Worth mentioning is the similarities with Figure 5.8 which further on shows the negligible effect between a bearing specific model and a common model which Figure 5.29 is.

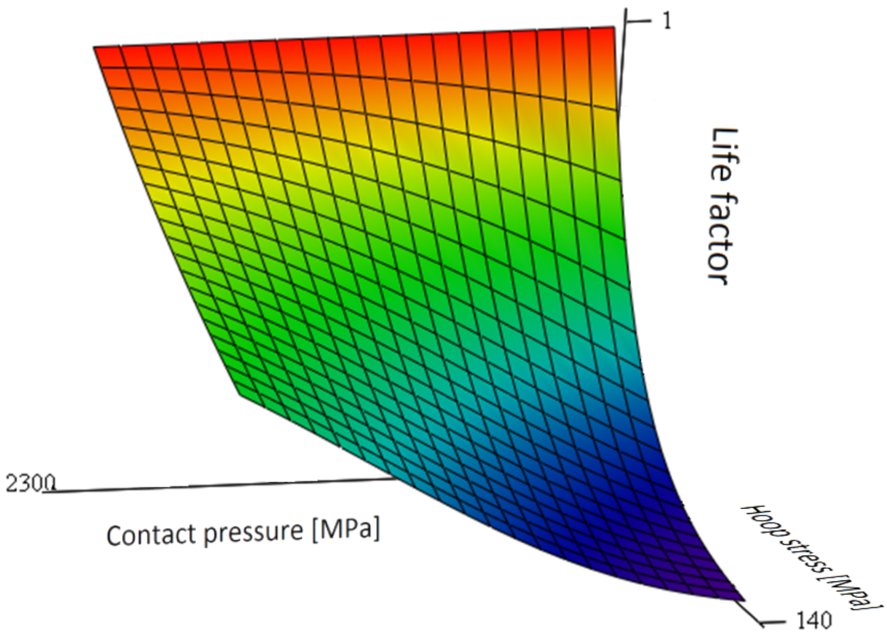

Figure 5.29: Shows the final common meta model for the scientific level visualised in a 3D plot.

6

Recommendations

As the result has been concluded, further recommendations of what to continue to work with in relation to this project follows. The recommendations includes possible improvements of the working procedure but also direct tasks which were not managed in the project due to either the time limit or the scope of the project. Some of the detailed recommendations are not included in this report but instead handed over to SKF directly.

If a new project within the same area of interest would be performed, the methodology and the procedure of this project is recommended. This will minimise the work load and the identified problems which have been solved within this project will not repeat themselves. Since this project have been performed mainly manually, a great opportunity would be if the same procedure could be performed automatically. This is expected, with a small effort, to be adjusted to relatively easy due to the thoroughly method which have been developed by this project. A tool such as modeFRONTIER or similar would be a beneficial option to choose to perform such an automation. The opportunity to analyse more parameters at once would be a consequence from this choice, which would be very advantageous. Hence a both faster process and maybe even a better result may be obtained if such route is chosen.

By studying Equation 3.4 in more detail, one may see that there are other factors affecting the hoop stress as well. These are more specifically the rotational speed, the material and the temperature. The three parameters can be investigated in a similar manner as have been done through this project, in order to evaluate the effect of the three.

7 Discussion

7.1 Methodology

For the project, the methodology of DMAIC was used as a help to guide the project forward. It was of great help to follow a structure like this cause once a problem occurred, one could go back to the foundation of the methodology and look for tools that could solve the identified problem. The focus on data collection and analyse DMAIC provides was the major reason the methodology was chosen, and with hindsight, that was the correct choice. In addition to the methodology to rely on, the Gantt-chart created early in the project to plan the process was a good help to keep the time limits throughout the whole project. The Gantt-chart did push the project forward when it was realised that a certain task had to be done in order to keep to the schedule. Since the Gantt-chart was an approximation of the process, with hindsight it turned out to be accurate enough to be able to wrap the project together without needing to restrict the scope of the project.

A more specific method which was used to produce and analyse the result was the least square method. This was used to evaluate and calculate the smallest error among the available options that were able to choose between. This method was considered to be good as an evaluation tool since the error it produces can easily be compared between the alternatives. The advantage with the least square method is that it gives an error over the whole set of points, a measure of how large the total error over the whole set is.

When deciding the type of curve fits, there were mainly two distinctive options to choose between. The first was interpolation, which would fit the data points in detail but in between the data points building up the curve, the result could be diverging. This is because of how the curve fit is built, which most often are from higher degrees of polynomial functions, which can oscillate between the data points. Already a degree of 2 or 3 of the polynomial function can cause problems. This method was hence not optimal since the produced curve fittings were going to be able to show an accurate result in between the data points as well. The other option was the method of a smooth curve fit, meaning that with an accepted error for the accuracy of the data points, the area between the data points will follow the pre-defined function. Hence this was needed in order to obtain an accurate result and became the chosen method.

When creating the different parameter levels, a lot of methods of how to combine these in the most efficient way exist. The choice of DoE was made since it is a common method which is known to effectively optimise the number of simulation in order to obtain a reliable result. Within DoE, one can choose between full factorial and fractional factorial design. Fractional factorial is better to use when handling many factors to decrease the number of samples, but in this study only two factors were analysed. Also the problem was known as highly nonlinear. This means a full factorial analysis would not be that large to execute, hence this method was chosen.

7.2 Parameters

The effect of the interference fit on the performance of the bearings in relation to reasonable loads were to be investigated in this study. But since the whole assortment was going to be studied, a reasonable interference and load for one bearing is not necessarily reasonable for another bearing. The problem in itself was thus a multi-disciplinary problem due to all parameters influencing each other and possibly affecting the result. Because of this, the parameters were needed to be chosen in a clever way to be able to make a comparison over the whole assortment. For the scientific level, the parameters were chosen to be contact pressure and hoop stress, since these components are directly affecting the life time calculations within the BEAST model.

For the engineering model, the load was chosen to be factors of P/C because this makes the model comparable over the assortment with reasonable loads. To use only a load as a parameter would not be feasible since the recommended loads for the bearings are significantly different, dependent on the size and dimensions of the bearings. Unfortunately when adding the C_{ISO} value as a parameter, the effects behind the C_{ISO} value were added into the problem.

The same argument for P/C goes for the selection of Δ/d as a parameter. The option to only use the interference, Δ, as a parameter would not be feasible since the bearings have very different interferences in practice. This is why the quotient between Δ and d is used since this relation is more dimensional feasible. Another positive thing by choosing this parameter was that the effect of the k factor was isolated in Equation 3.13, which then only varies between series for different thickness of the inner ring.

7.3 Modelling

In general, the life time of the inner ring is depending directly on the shear stresses in the material. As shown with the scientific model, the contact pressure and corresponding shear stress give the same behaviour for all bearings. Hence a common scientific model was created for all bearings. The life time calculation in BEAST includes a material fatigue limit which has been implemented as a single constant for all of the bearings in the assortment in the BEAST model. So when knowing this, it is not that strange that the scientific model is behaving the same over the whole assortment. It may be argued that this fatigue limit should be dependent on the bearing size instead. According to the ISO standard, the fatigue limit should be decreased for larger bearings [5]. An implementation of this recommendation in the BEAST model will probably give a variation between bearings instead and hence, an update of the scientific model will be needed.

When creating the engineering model, the scientific model was chosen as a base. From this point, the scientific model was expanded to the engineering level with help from Hertz contact theory and an analytic equation of the interference. Another option to do this would be by directly develop the engineering model from data points collected in BEAST. This option was disregarded since it would be very time consuming because the need of running a lot of simulations and in addition, no common behaviour was seen which a common model would require. Further on, probably all of the bearings in the assortment would have to be evaluated to draw reasonable conclusions. Since the route to use Hertz contact theory to transform the contact pressure to load was chosen, the already created data building up the scientific model was used once again to save time instead of creating new data for the engineering model. Another benefit with this solution was the ease of usage for both of the developed models for the product development team.

7.4 BEAST model

As have been elaborated throughout the whole report, the current BEAST model may not give a result as reliable as was thought. Besides that, the software was very manageable and overall a very good software to use. If the identified problems are issued and resolved, the model has a good opportunity to be very successful. The project was performed with version 13.2 of BEAST and is hence the only version the result from this study can be said to be valid for. Even though other values of the simulations would be obtained after potential updates, the principle result from this study, that the hoop stress has a larger relative effect on larger bearings with thinner inner rings, is believed to remain the same.

7.5 Further investigation of parameters

This section will briefly elaborate on the topic regarding other parameters that may be of interest to investigate further. It will bring up parameters that have been excluded in the main study due to the scope, to not exceed the stated time limit of the project.

Manufacturing techniques

Different manufacturing techniques of the inner ring and especially the heat treatment have an effect on the structural stresses and will be residual in the material. These stresses may be tensile or compressive depending on the type of manufacturing technique and thus affect the hoop stress in different ways.

Surface fatigue

The surface fatigue is heavily affected by the bearing load, lubrication, rotational speed and temperature [3]. The lubrication has in itself additional parameters affecting the performance. One affecting parameter is the thickness of the film created by the lubrication. This film cannot be too thin in order to acquire sufficient lubrication nor it must be too thick, which will increase the friction. Moreover, there is the humidity of the lubricant which is of importance for the surface fatigue. This area has thoroughly been investigated by SKF, and the effect of this is included in correction factors for the general life time equation seen in Equation 3.1.

Rotational speed

The analytic theories described in Section 3.4.3, give that a rotational speed will decrease the hoop stress as the rotational speed reduces the interference. This is due to the negative sign of the circumferential part in the two first equations in Equation 3.4. This parameter may be of interest to investigate even further since it will have an effect on the hoop stress.

Temperature gradients

Temperature gradients are also a factor in the analytic theories that has not been investigated in this study. Naturally higher temperatures in the shaft will increase the hoop stress in the ring, since the shaft will increase in volume. For the other way around, a higher temperature in the ring than in the shaft will instead decrease the hoop stress in the ring. These temperature gradients may be of interest for further investigative studies within the subject, to investigate the amplitude and the effect of the hoop stress during normal operating conditions in regard to temperature.

7.6 Social, ethical and ecological aspects

The bearing as a product itself does not have any larger effect on the society. However, bearings are indirectly related to the product which they are mounted in, which can be both related to problematic businesses or not. Since bearings are such a common component in rotating products, they can be assumed to be a part in all of them even though they are not visible. An effective and efficient bearing supports the product to be more energy efficient and minimises the energy losses. This is why SKF have to produce as good and effective bearings as possible. This will in itself contribute to more productive processes and further on benefit the society in a positive way.

Regarding the ethical perspective, one can argue that the bearing producer has an ethical responsibility regarding what end products the bearings are used for. For example, if it is implemented in warfare or oil industry products that will increase the productivity in that manner, it will have a negative effect in these aspects. This is a problem the project group had no influence over and had therefore to be accepted, but could at the same time be considered to be outside of the scope of the project. The same yields for the ecological aspect, it may affect the environment in a positive or a negative manner with respect to what product the bearing is a part of. The project as it have been embodied does not affect any of the social, ethical or ecological aspects directly but hopefully, the project will have a positive effect in regard to the development phase of new products. From this point of view, the society may benefit from this project through resource reduction in many different levels long term.

8

Conclusion

The project started with an aim to create a meta model which could describe the whole assortment of SKF's spherical roller bearings. The result would come to be close to the stated aim since a common model for the engineering level could not be created easily due to diverging behaviour between bearings. The methodology of DMAIC was a help to structure the project and divide it in significant parts. It was also of good use to define a supposed path of the project, as a procedure to follow. The thoroughly described procedure guided the project into the right direction continuously.

One of the main questions during this project was what parameters are affecting the hoop stress and by which degree they influence. Already early in the project during the theory study it could be seen how the interference, Δ, the ratio between the inner and outer diameter of the inner ring, k, and the material through Young's modulus affect the hoop stress, as described with the help of Equation 3.13. These parameters are building up the analytic expression of how the hoop stress is calculated and thus also the factors directly affect the hoop stress but there may be second degree effects of the same. Also the analytic expression is approximate due to the assumption of an infinitely thin inner ring. Another question in relation to this was, what could be considered as reasonable limits of the hoop stress at the customer. By following the recommendations given in the SKF Catalogue, an interval up to 140 MPa could be seen as reasonable for the hoop stress. In combination to this, it was also interesting to investigate what designs were most affected by an increase of hoop stress.

When keeping the contact pressure fixed in the scientific model, it was seen that there was no effect of the hoop stress between the bearings in the assortment since all curves aligned. When comparing between different levels of contact pressures, a larger effect was seen for a lower contact pressure.

On the other hand, when analysing the engineering level including interference and load instead, it was realised that the most affected bearings are the ones with larger diameter and thinner inner ring which can be seen in Appendix A.17. But as the hoop stress affects more, the contact pressure is also decreasing, making them unimportant in the context since the surface fatigue will be dominating a long time before an effect of the hoop stress takes place. With other words does a limiting line appear which bounds the area of reasonable contact pressures. Depending on this line, the closest bearing to this border will be the bearings most affected by the hoop stress for reasonable loading conditions, which can be seen in Appendix A.16.

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A

Appendix

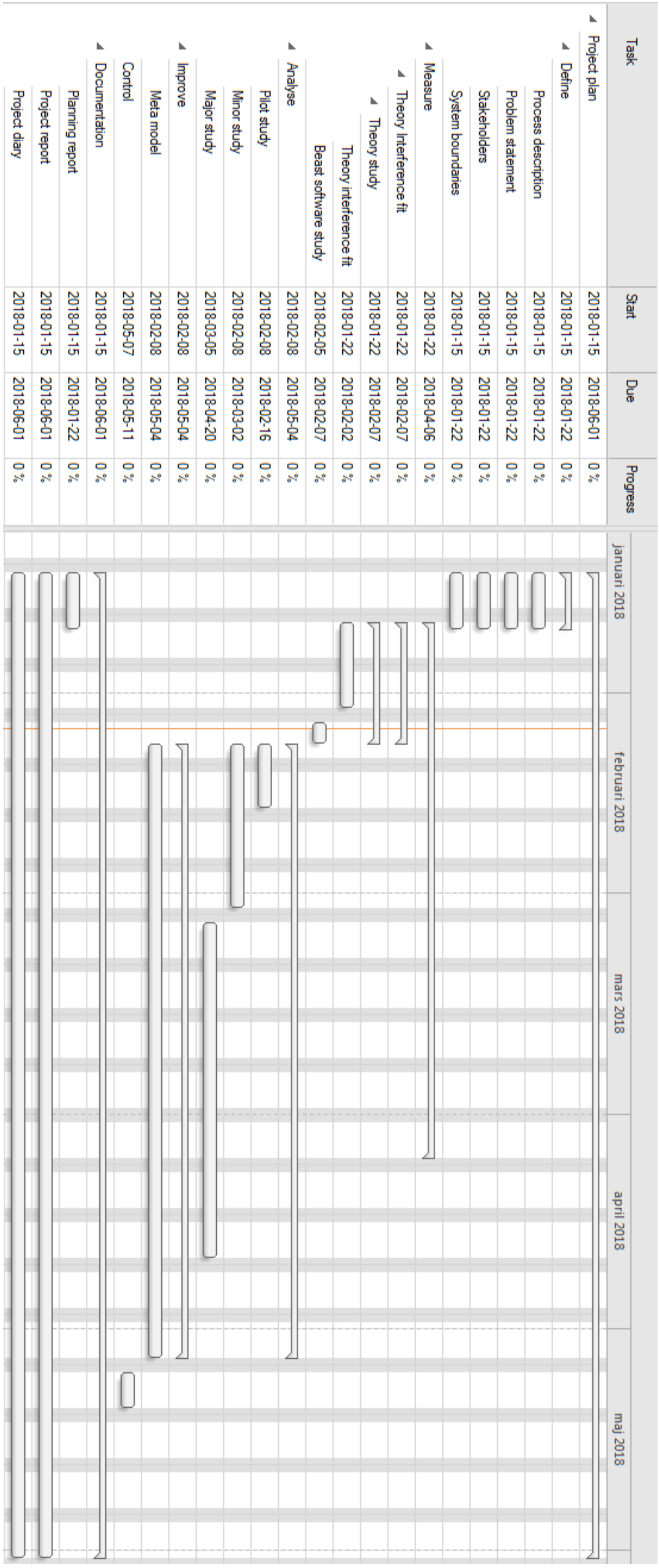

Figure A.1: Shows the Gantt-chart of the project process.

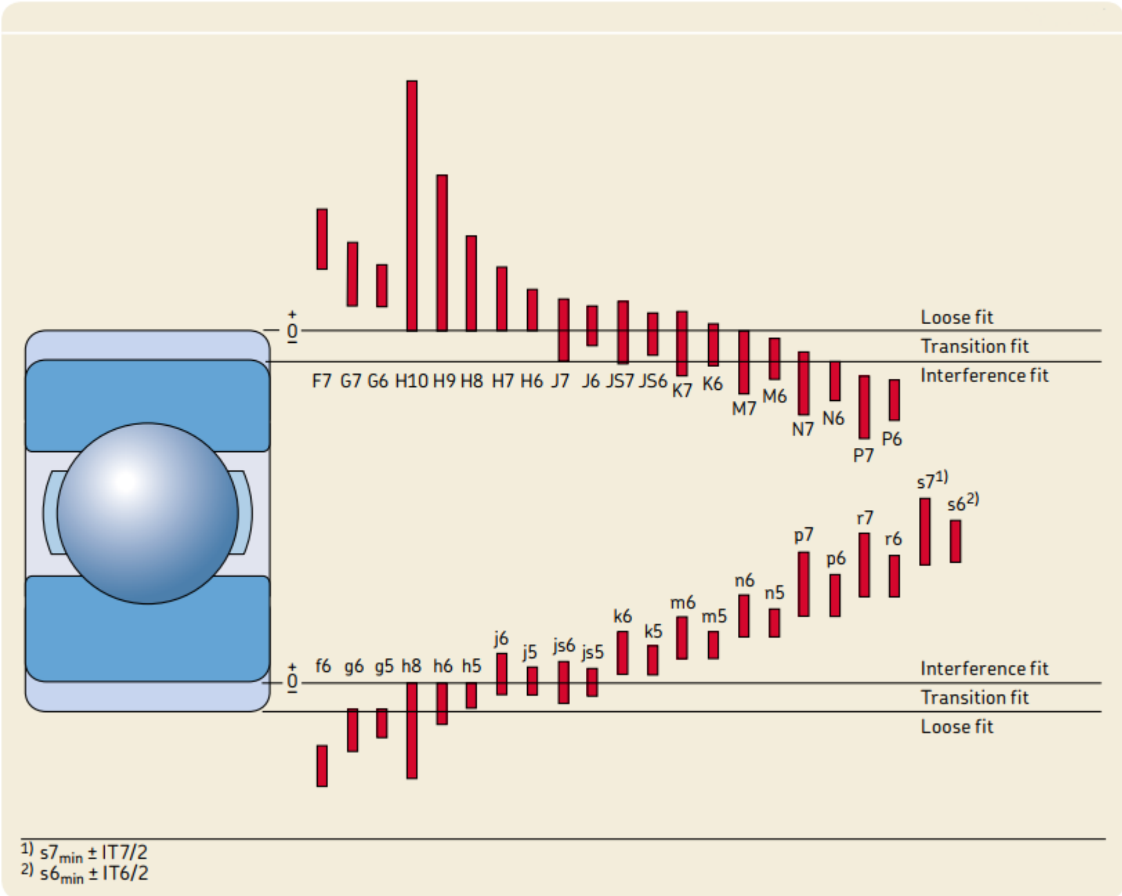

Figure A.2: Shows the range of possible fits.

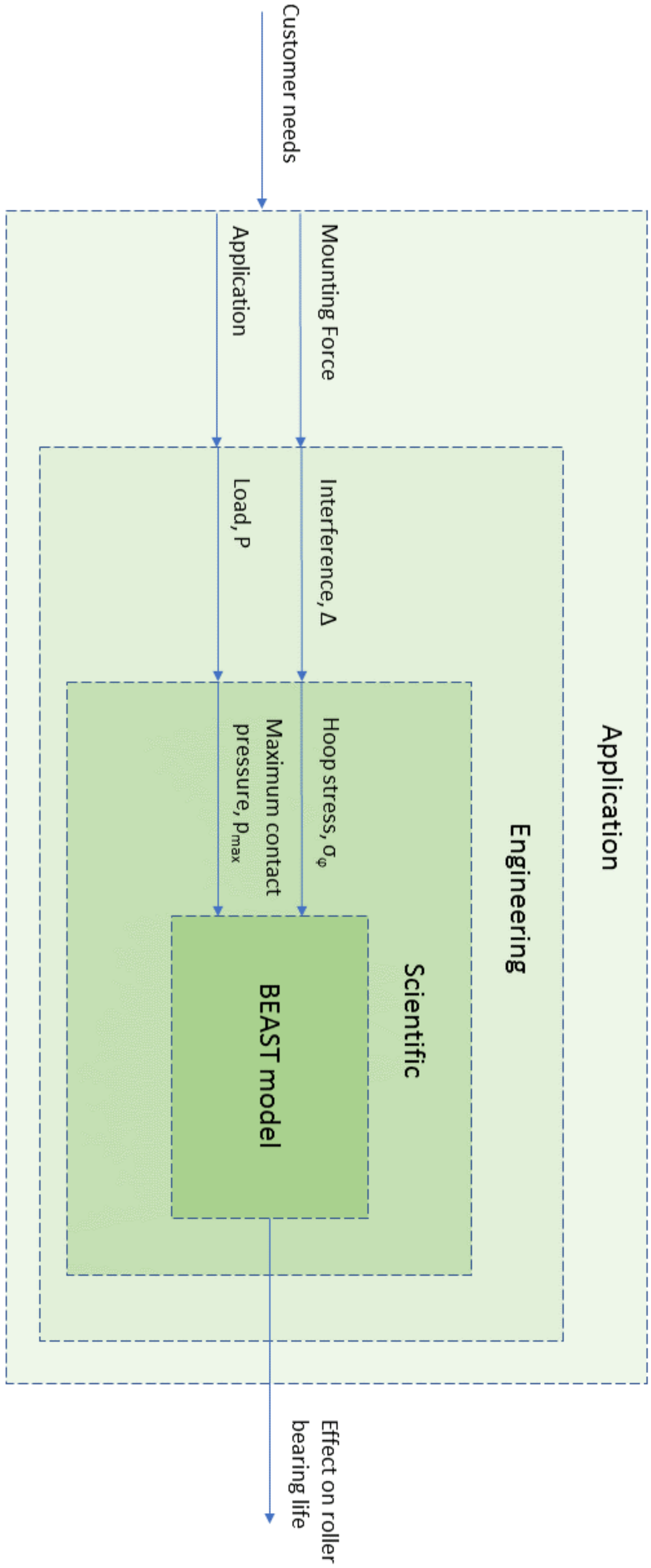

Figure A.3: Shows the levels of stakeholders.

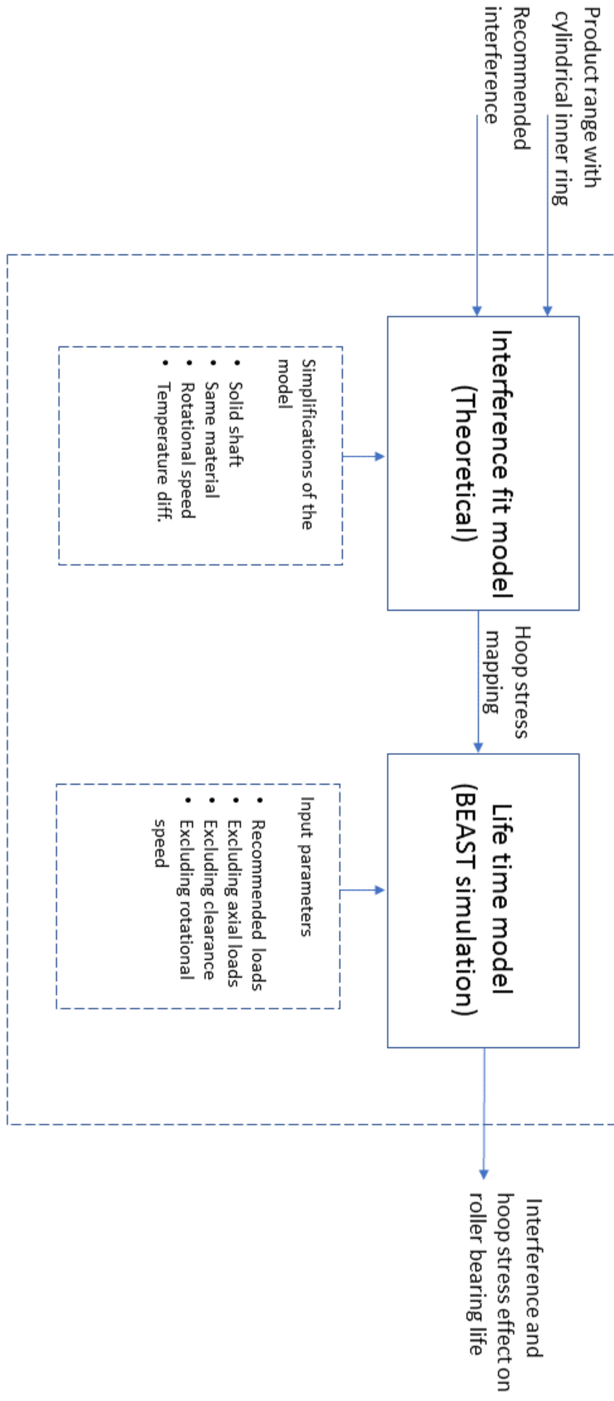

Figure A.4: Shows the system boundaries of the project.

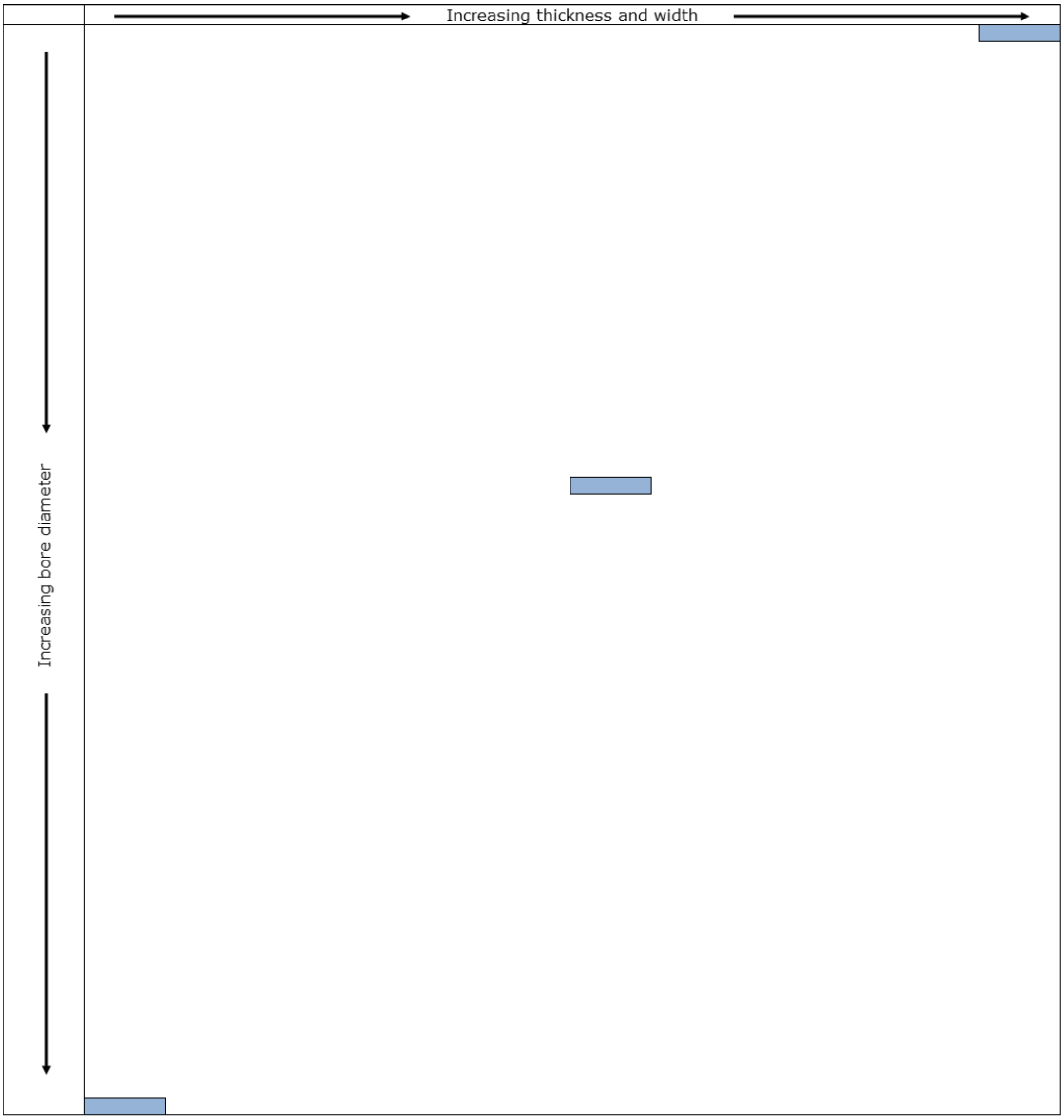

Figure A.5: Shows the bearings that were evaluated in the pilot study. The bearings are marked with blue colour. The assortment matrix is modified due to confidentiality

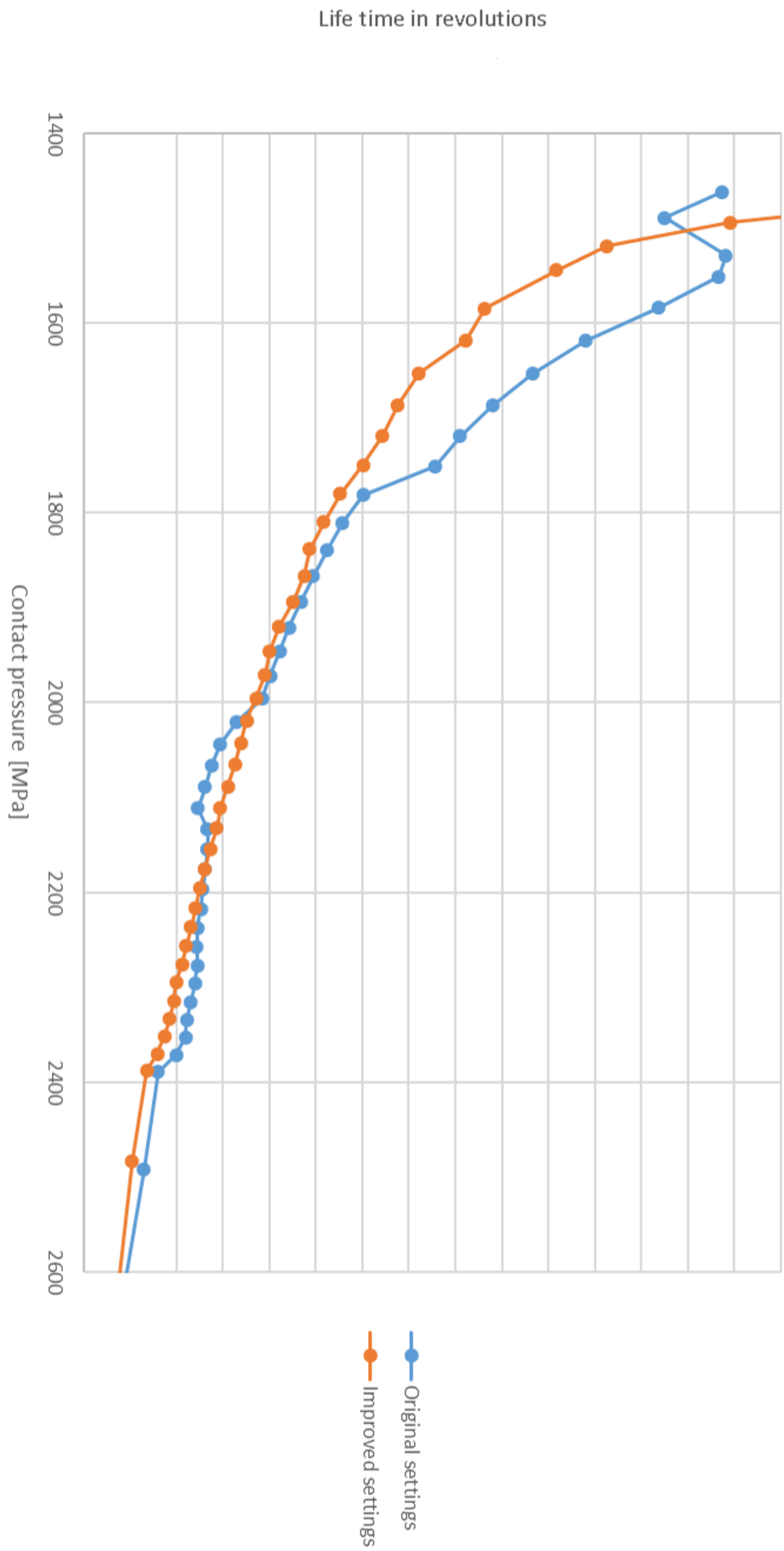

Figure A.6: Shows that with increasing contact pressure, the life time increase at some locations. The absolute life time values are removed due to confidentiality. The original settings had a resolution of 1 and a solver precision of 5. The improved settings had a resolution of 0.25 on the inner ring surface and a solver precision of 2.

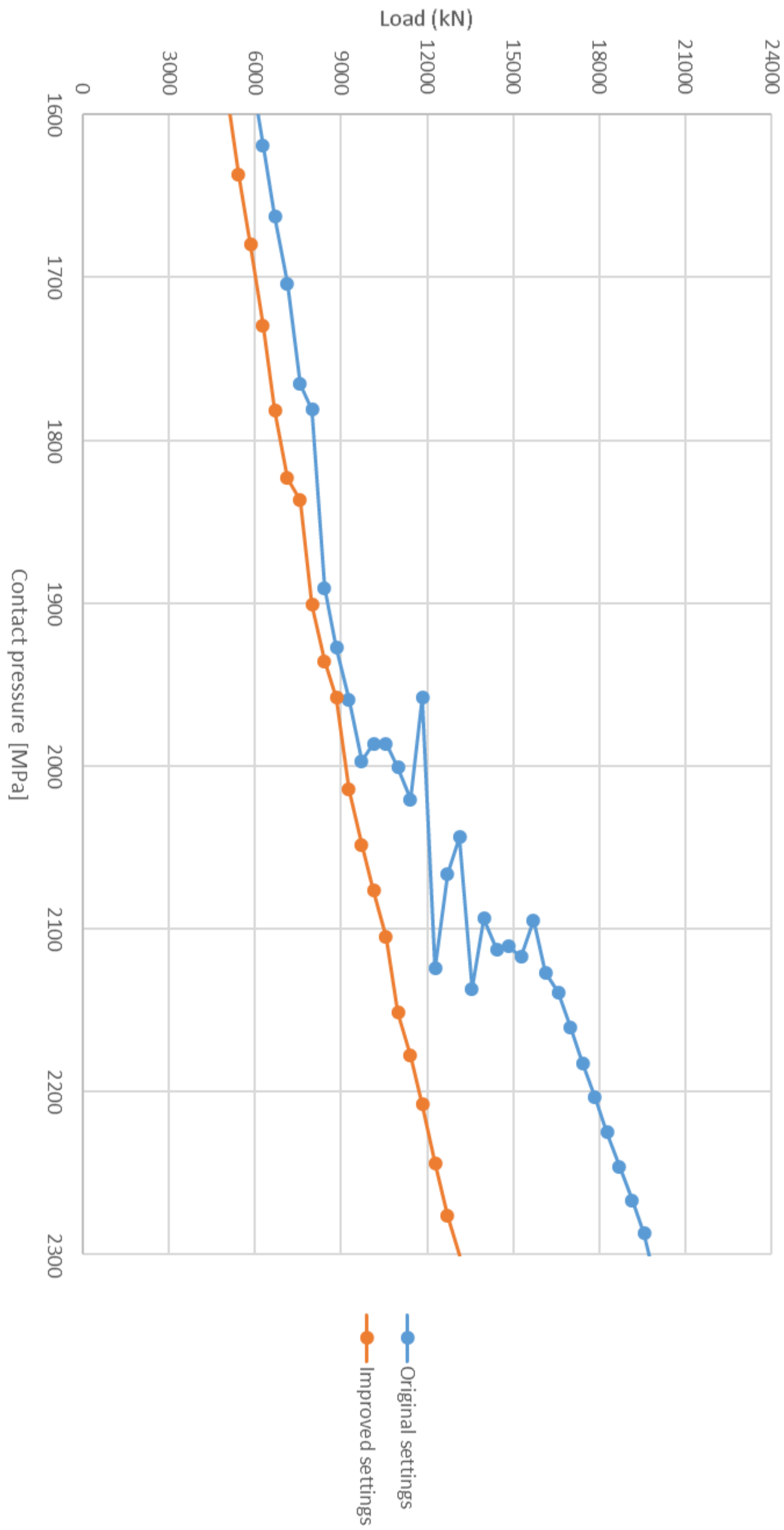

Figure A.7: Shows that with increasing load, the contact pressure fluctuates. The original settings had a resolution of 1 and a solver precision of 5. The improved settings had a resolution of 0.5 and solver precision of 2.

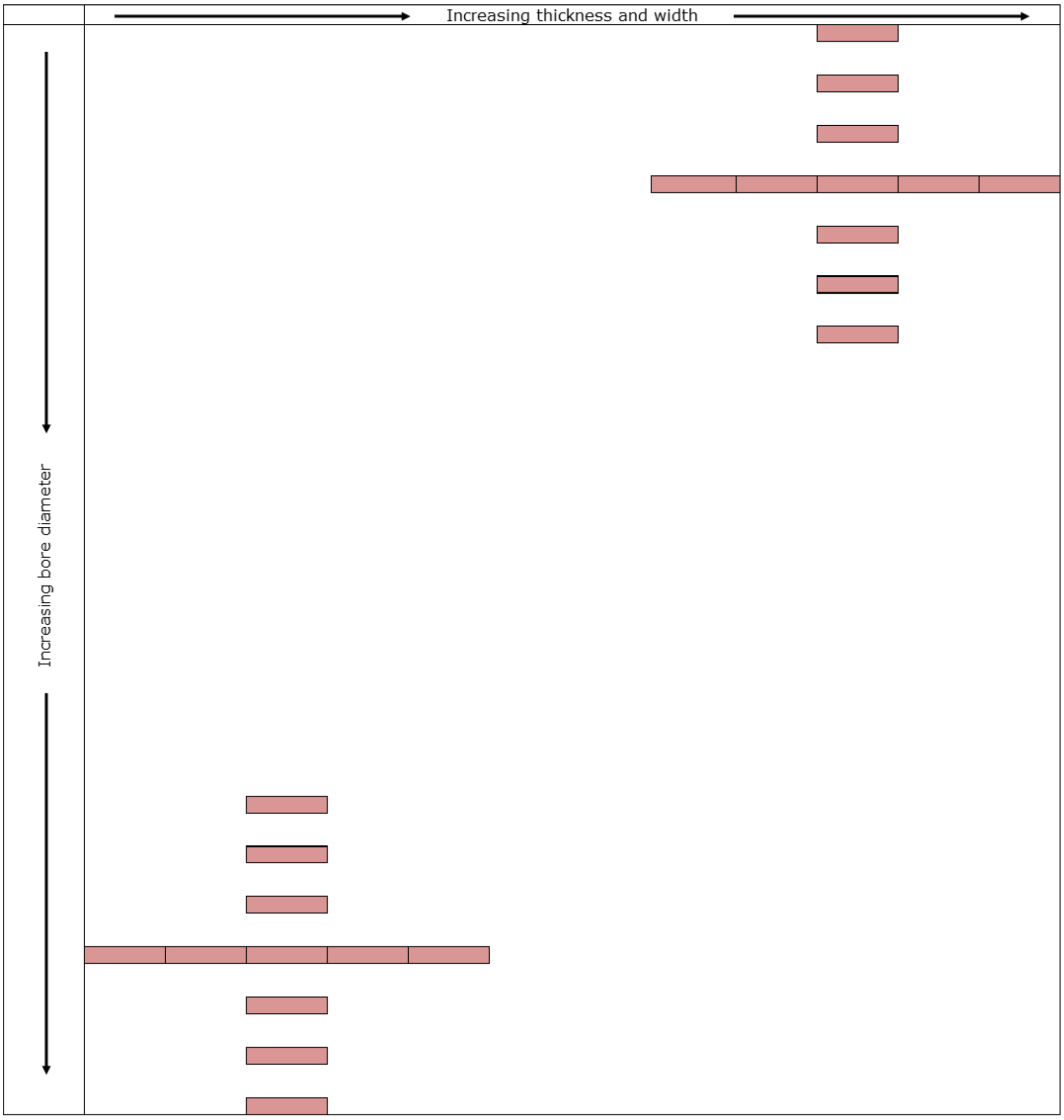

Figure A.8: Shows the bearings that were evaluated in the minor study. The bearings are marked with red color. The assortment matrix is modified due to confidentiality.

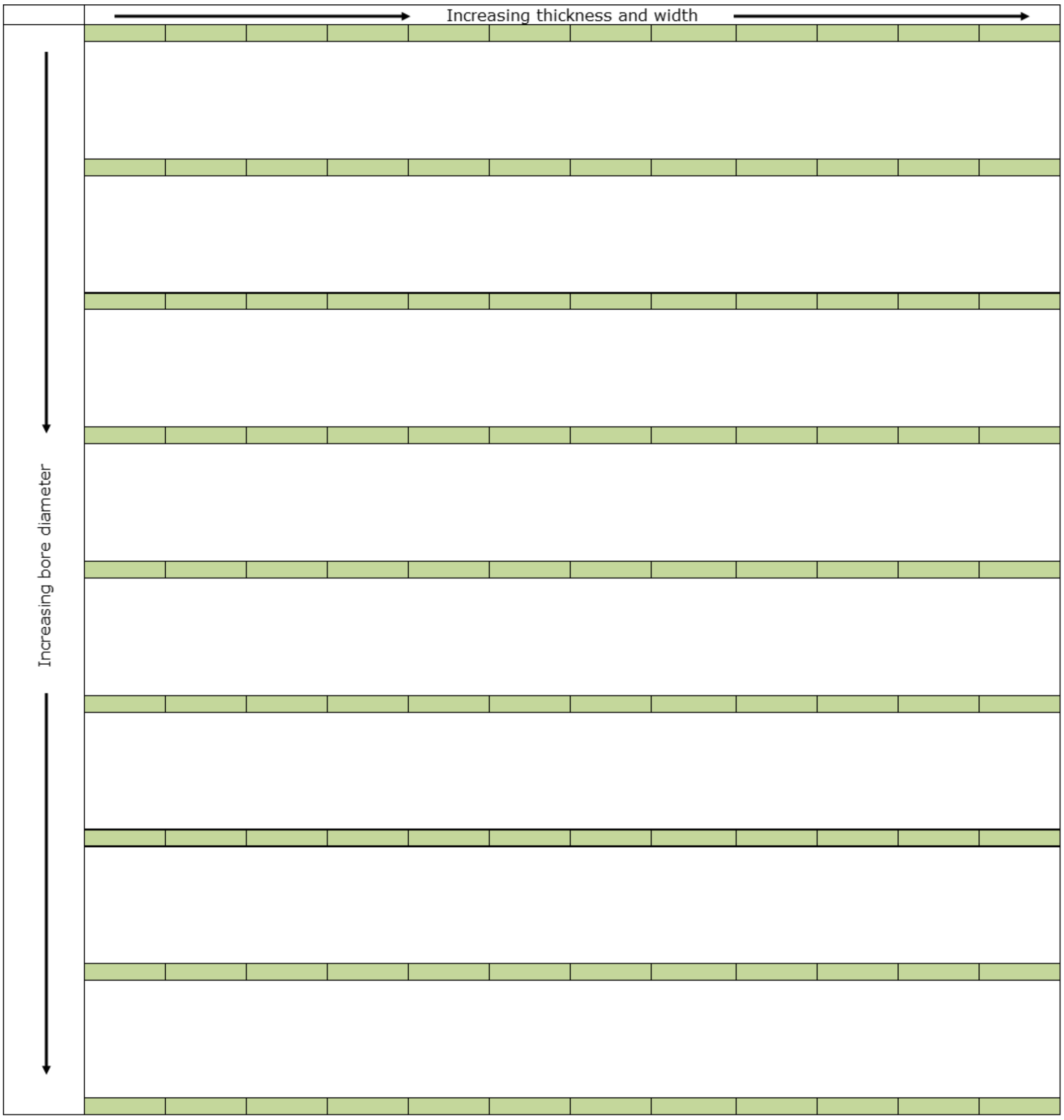

Figure A.9: Shows the bearings that were evaluated in the major study. The bearings are marked with green colour. The assortment matrix is modified due to confidentiality.

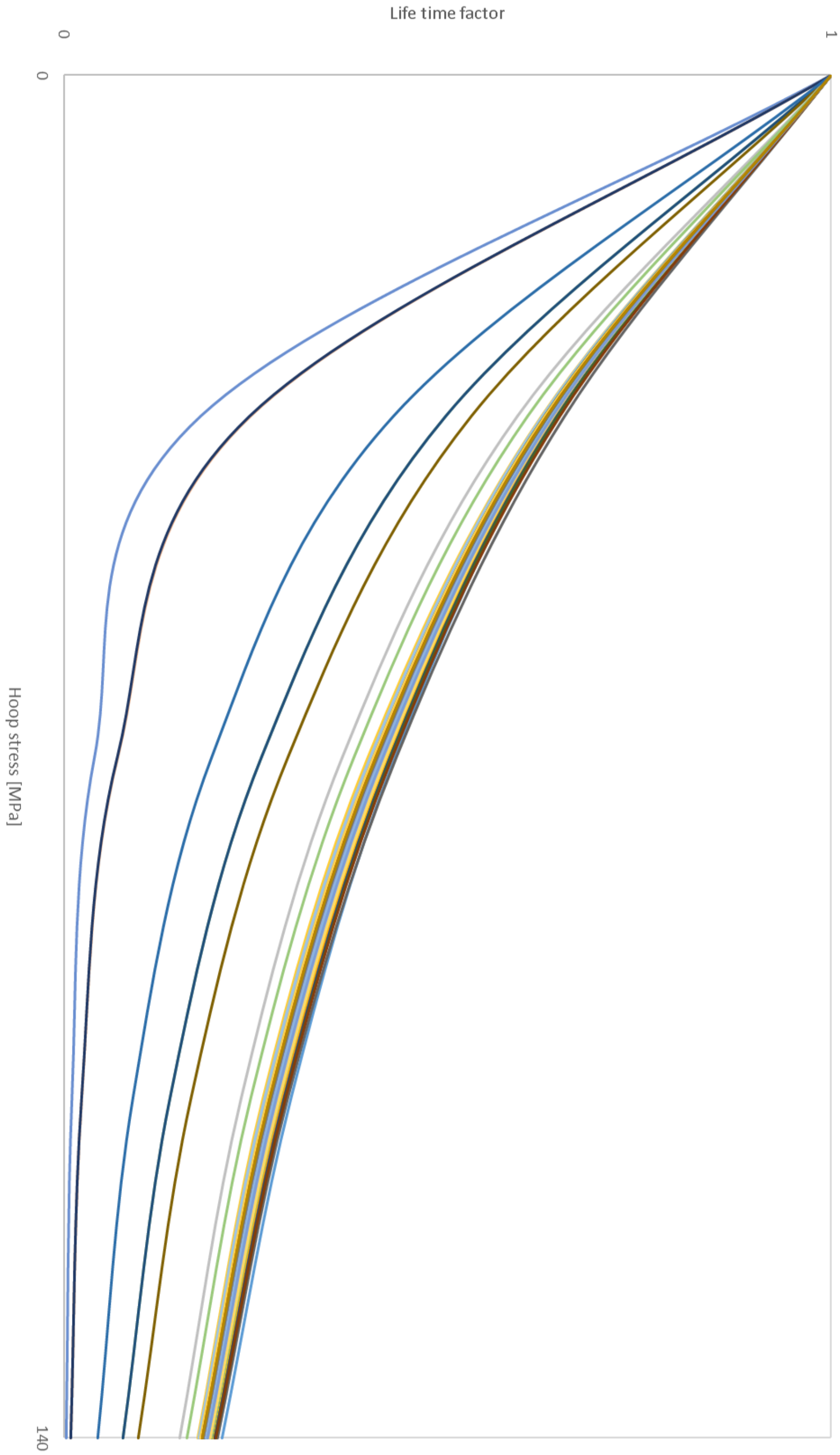

Figure A.10: Shows the effect of hoop stress for all bearings included in the major study at a fixed contact pressure of 1900 MPa.

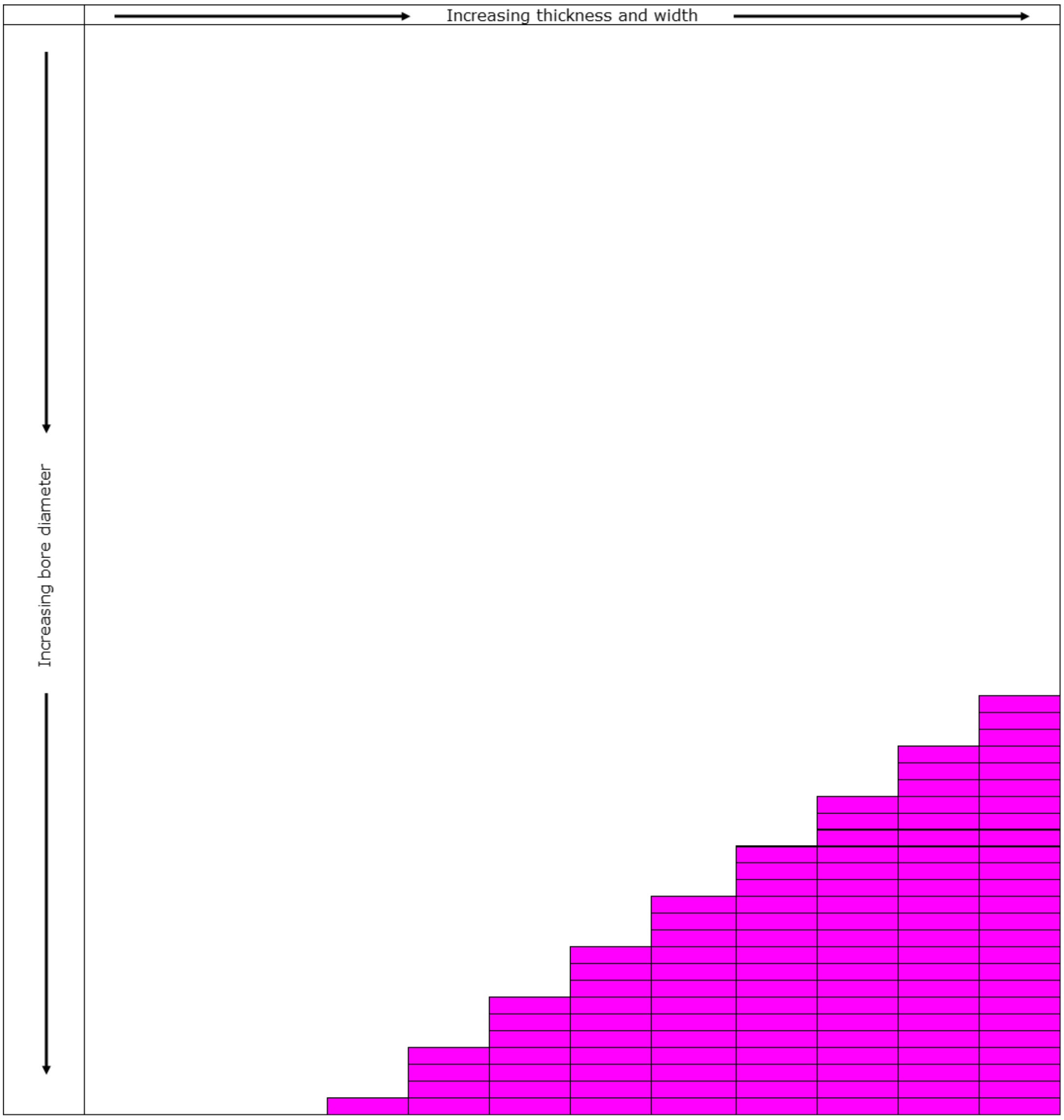

Figure A.11: Shows the area of bearings which were expected to diverge from the others based on the major study. The bearings are marked with purple colour. The assortment matrix is modified due to confidentiality.

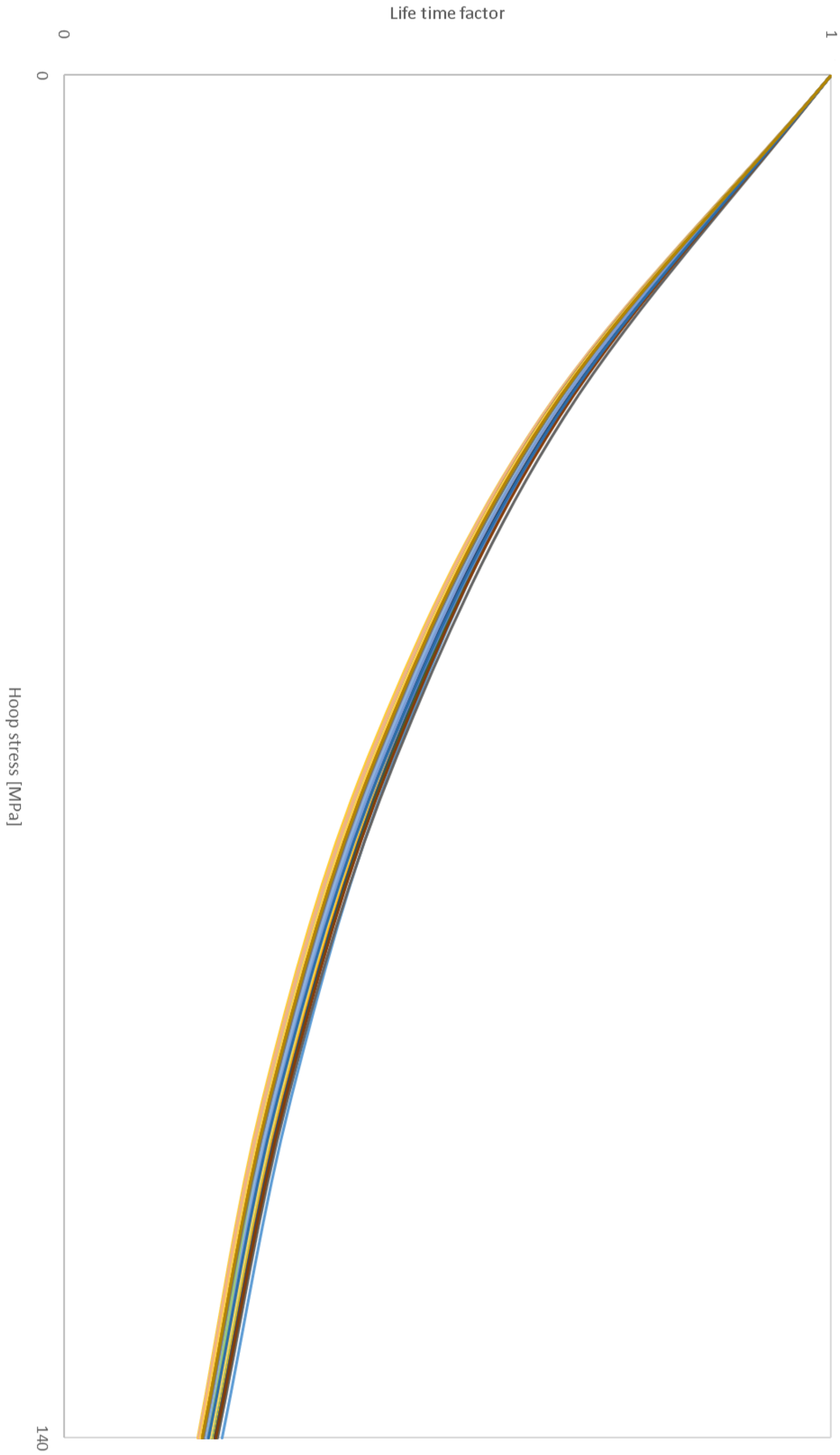

Figure A.12: Shows the effect of hoop stress for all bearings included in the major study with the updated settings at a fixed contact pressure of 1900 MPa.

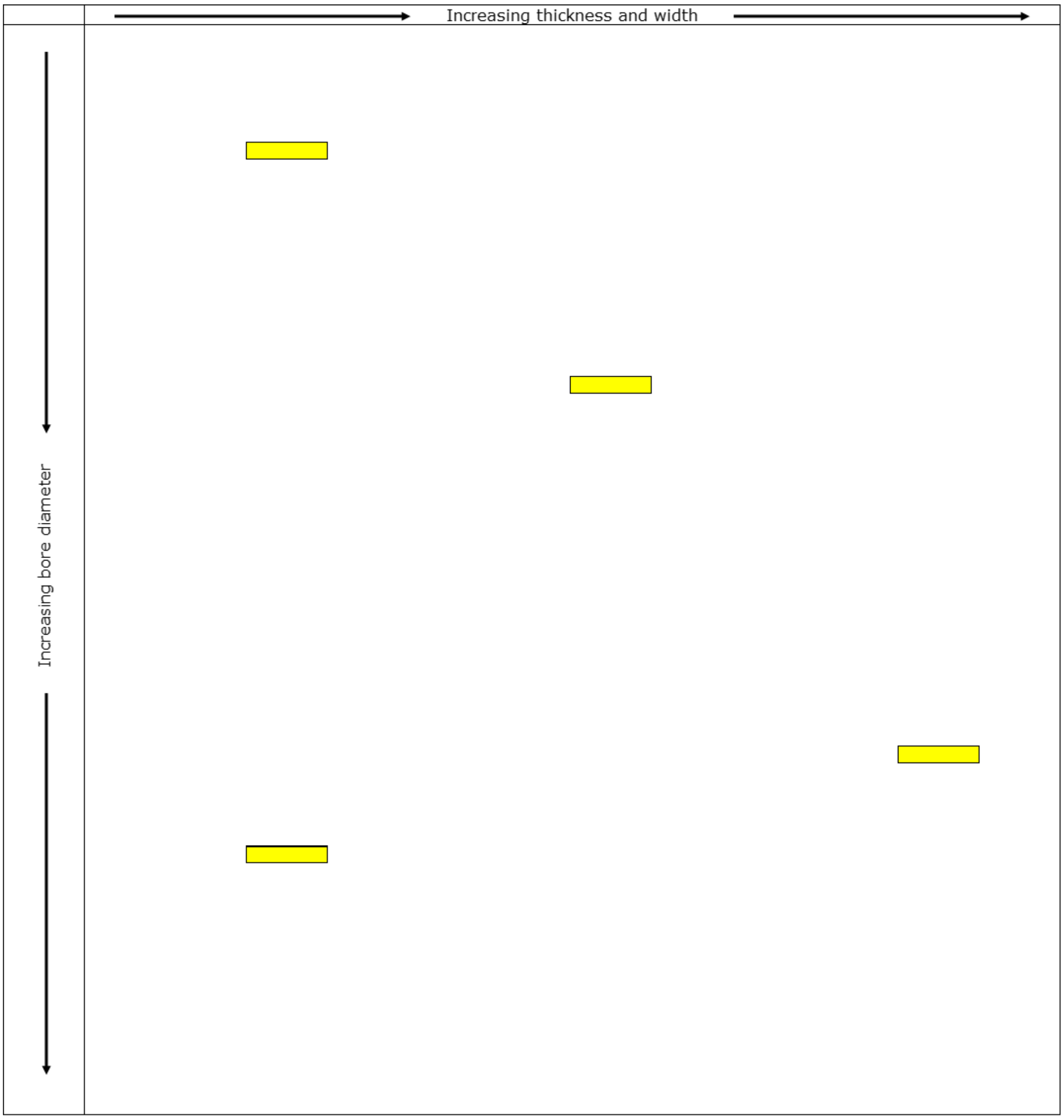

Figure A.13: Shows the bearings which were used to verify the validity of the scientific model. The bearings are marked with yellow colour. The assortment matrix is modified due to confidentiality.

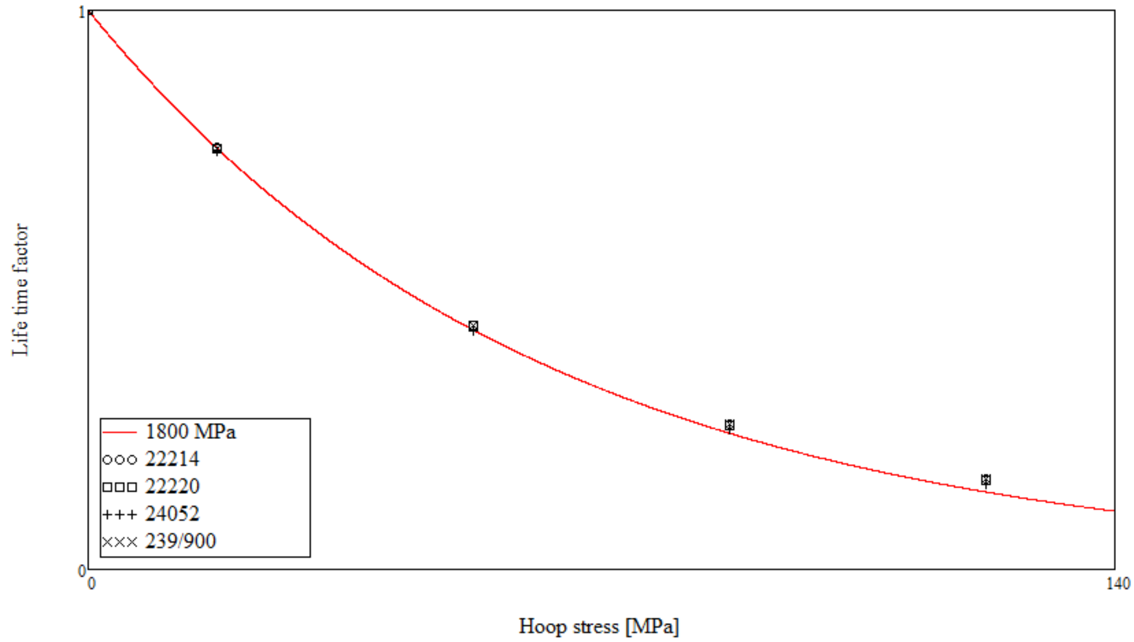

Figure A.14: Shows how the meta model is performing for bearings the model was not built upon with a contact pressure of 1800 MPa.

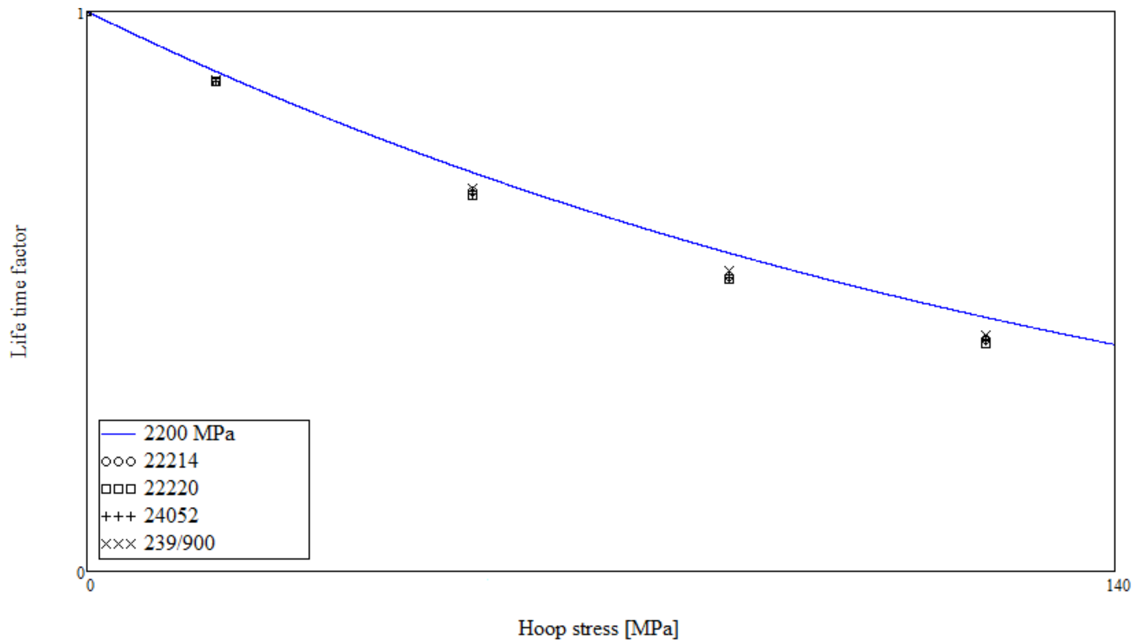

Figure A.15: Shows how the meta model is performing for bearings the model was not built upon with a contact pressure of 2200 MPa.

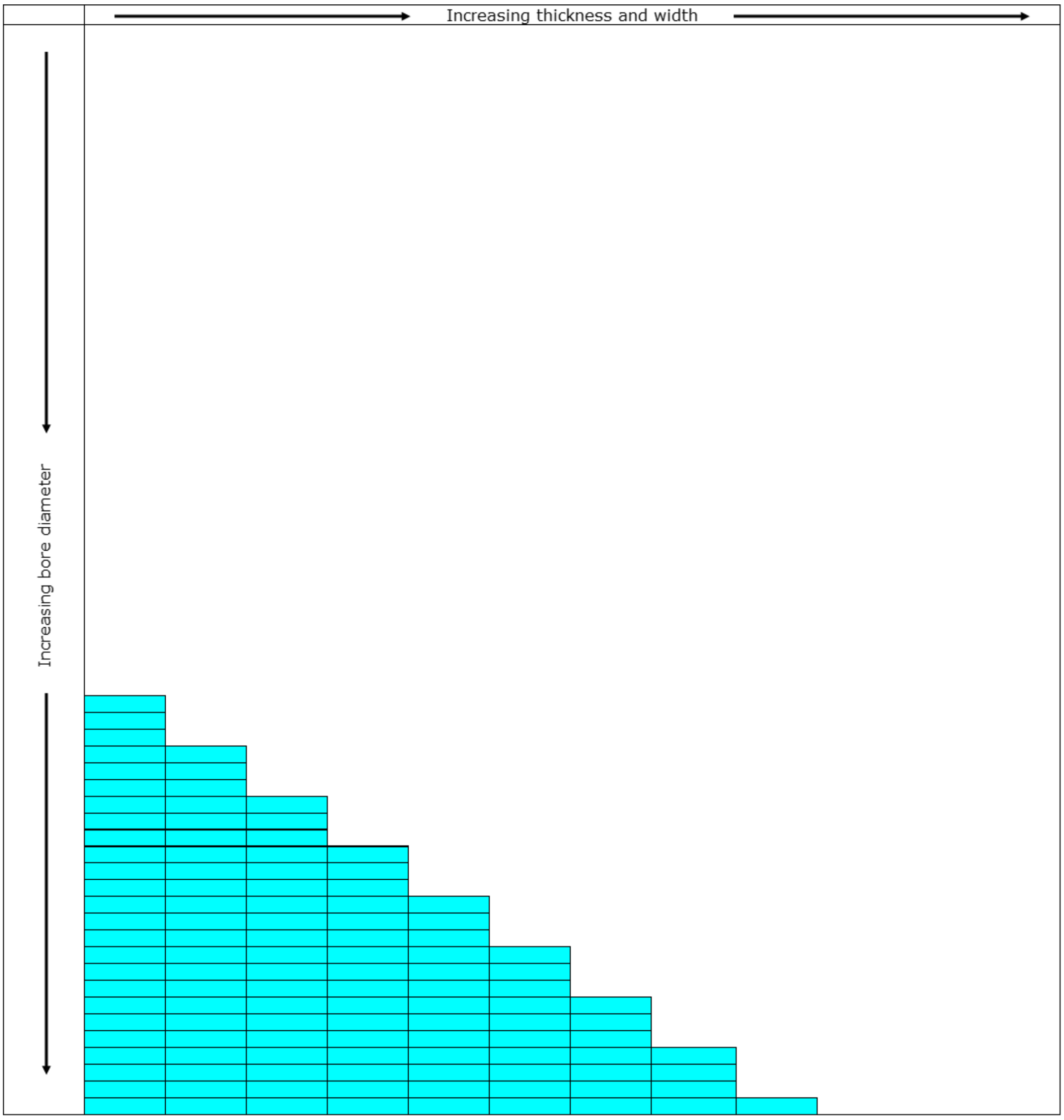

Figure A.16: Shows the area of bearings which were expected to have unreasonable conditions based on the engineering model. The bearings are marked with light blue colour. The assortment matrix is modified due to confidentiality.

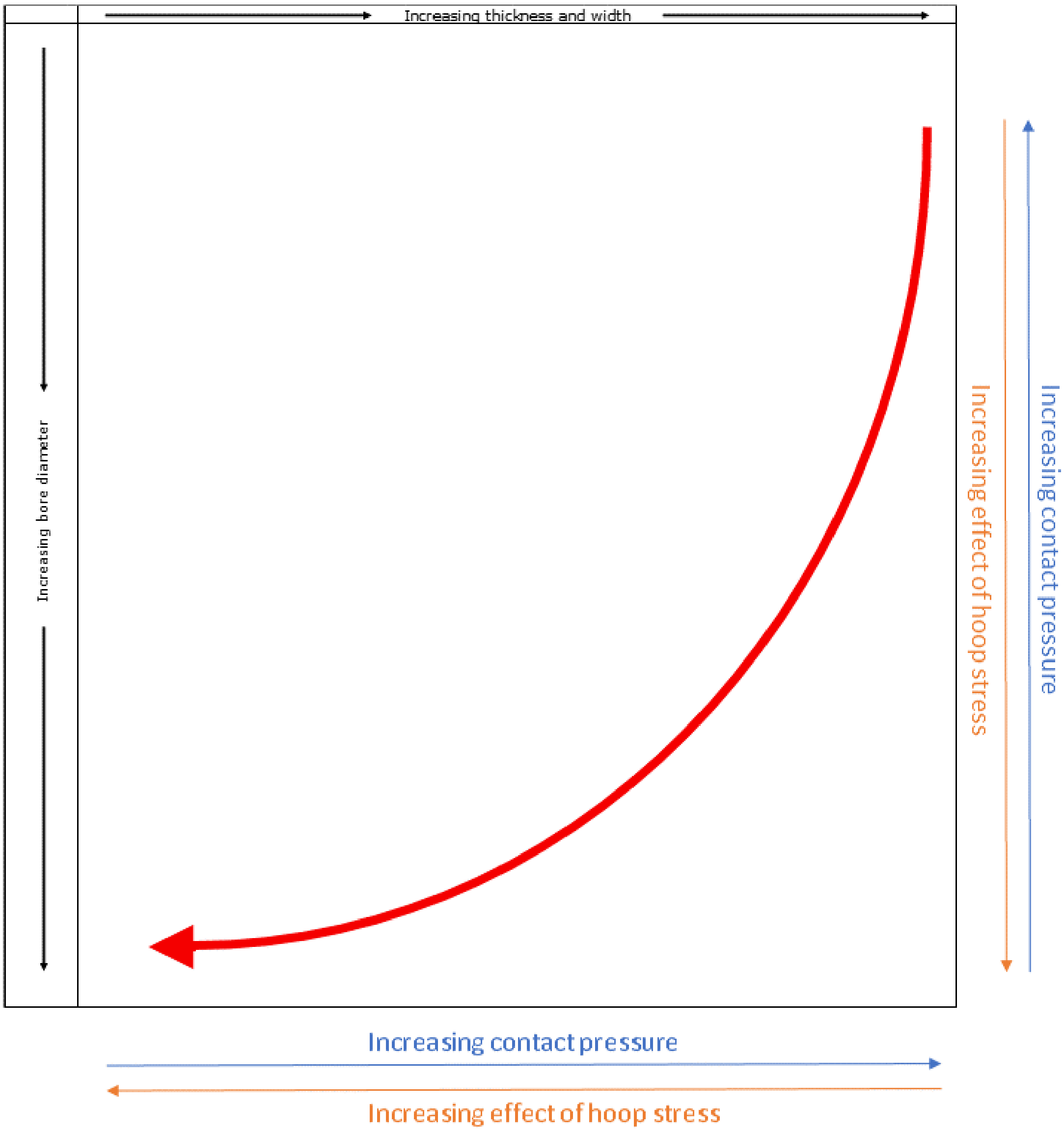

Figure A.17: Shows the behaviour of the contact pressure and the hoop stress over the assortment. The red arrow shows the direction of increasing effect of hoop stress. The assortment matrix is modified due to confidentiality.