Fretting fatigue in marine diesel engine bedplate bearing interfaces

Master’s Dissertation by

JENS GUNNARSSON & HENRIK RINGDAHL

Supervisors:

Solveig Melin, Div. of Mechanics

Christer Persson, Div. of Materials Engineering

Henrik Andersson, MAN B&W Diesel A/S
Preface
This Master’s Thesis has been conducted at the division of Material science and Mechanics at Lunds Institute of Technology in cooperation with MAN B&W Diesel A/S in Copenhagen during the summer and fall of 2005. The purpose of this study has been to investigate effects of fretting in contact between bearings and bearing seats in large bore two stroke diesel engines.

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Jens Gunnarsson and Henrik Ringdahl
Abstract
MAN B&W Diesel A/S develops large bore two stroke diesel engines for ships, trains and power station applications. When designing these engines it is not possible to avoid relative motion between metal surfaces in contact. As an example, the bearings in the crankshaft and in the crosshead are exposed to a type of normal and tangential forces that could cause what is called fretting damage due to the relative motion between the back of a bearing and its adjacent seat. The question is if this constitutes a threat and a potential problem as regards engine life.

In this Master’s dissertation the fretting damage between material from a bearing and its adjacent seat are investigated for different load situations and relative displacement amplitudes. Experimental setups are manufactured to get accurate material data and to conduct flat on flat fretting tests. The tests are performed in a servo hydraulic tensile test machine and output data are logged to a PC. Except for the machine output an external load cell is also used to govern the normal force. The flat on flat fretting tests are performed within the average contact pressure in the interval 5 MPa to 10 MPa and relative displacements of between 6 µm to 200 µm. One set of experiments are performed using cyclic fatigue loading of the specimens and one set with zero axial fatigue stresses, aimed at investigating the wear aspect of the fretting damage. The surface degradation achieved in the tests is examined in a scanning electron microscope. A 2D plane strain model for the FE-solver Abaqus is created using HyperMesh. The model is correlated against literature data and used to simulate the relative displacements from the experimental tests.

It has been found that the dominating surface degradation for the conditions in the contact between the bearing and the seat is due to fretting wear. To minimize damage the experiments performed suggest that the relative displacement should be kept below 10 µm. An increase in average contact pressure from 5 MPa to 10 MPa may allow an increase in the suggested relative displacement to 15 µm and still result in minimal damage. It is also found that the friction undertake a severe increase after only a couple of hundred cycles. A relative slip amplitude of 100 µm will change the coefficient of friction from 0.13 to 0.8. The friction does not, however, show any noticeable response to contact pressure.

Furthermore it is found that it is possible to use FE calculations to predict locations and size of fretting damage. To get fairly accurate results the model needs to be very thorough regarding material models and surface interactions. The friction shows a strong dependency on relative displacement amplitude, something that also needs to be accounted for.
Nomenclature

\( \alpha_{kl} \) \hspace{2cm} \text{deviatoric backstress tensor}

\( \Gamma \) \hspace{2cm} \text{modified SWT parameter}

\( \Delta \sigma_a \) \hspace{2cm} \text{axial stress range in the specimen}

\( \Delta \varepsilon \) \hspace{2cm} \text{strain range}

\( \Delta \tau, \text{SSR} \) \hspace{2cm} \text{shear stress range}

\( \Delta L \) \hspace{2cm} \text{length increment}

\( \varepsilon_a \) \hspace{2cm} \text{total strain amplitude}

\( \dot{\varepsilon}_f \) \hspace{2cm} \text{fatigue ductility coefficient}

\( \varepsilon_{max} \) \hspace{2cm} \text{maximum strain}

\( \varepsilon_{min} \) \hspace{2cm} \text{minimum strain}

\( \theta \) \hspace{2cm} \text{polar angle of the critical plane}

\( \lambda \) \hspace{2cm} \text{coefficient of friction}

\( \lambda_{\text{press/pull}} \) \hspace{2cm} \text{coefficient of friction when specimen are pressed/pulled}

\( \nu \) \hspace{2cm} \text{poisson’s ratio}

\( \rho \) \hspace{2cm} \text{resistivity}

\( \sigma_f' \) \hspace{2cm} \text{fatigue strength coefficient}

\( \sigma_{ij} \) \hspace{2cm} \text{stress tensor}

\( \sigma_n \) \hspace{2cm} \text{contact pressure}

\( \sigma_T \) \hspace{2cm} \text{axial stress just below the contact surface}

\( \sigma_t \) \hspace{2cm} \text{tensile strength}

\( \sigma_{xx} \) \hspace{2cm} \text{normal stress perpendicular to contact surface}

\( \sigma_y \) \hspace{2cm} \text{initial yield stress}

\( \sigma_{yy} \) \hspace{2cm} \text{normal stress parallel to contact surface}

\( \tau \) \hspace{2cm} \text{shear stress}
\( \tau_{\text{max}} \)  maximum shear stress
\( \tau_{\text{min}} \)  minimum shear stress
\( \tau_{xy} \)  shear stress in the contact surface
\( \psi \)  fractional chance in resistivity
\( \omega \)  angular frequency
\( A \)  area of the contact zone
\( b' \)  fatigue strength exponent
\( c' \)  fatigue ductility exponent
\( d \)  relative displacement amplitude
\( d_e \)  elastic relative displacement amplitude
\( d_i \)  relative slip in transition from stick slip to gross slip
\( E \)  modulus of elasticity
EOC, COC  edge of contact, centre of contact
\( F \)  axial force
FE  Finite Element
\( k \)  gauge factor
\( K \)  isotropic hardening parameter
\( K^\alpha \)  Includes kinematic and isotropic hardening parameters
\( L \)  initial length of specimen
MSSR  modified shear stress range
\( N_f \)  number of reversals to failure
\( N_i \)  number of cycles to crack initiation
\( P \)  normal force
PMC  PCI Mezzanine Card standard is a family of slim modular cards.
\( R_i \)  resistance, i=1-4
$R_G$ resistance in strain gauge

$R_e$ strain ratio

$R_{\tau}$ shear stress ratio

$s_{kl}$ deviatoric stress tensor

SL1, SL2 symmetry lines

SSR shear stress range

SWT Smith-Watson-Topper

t time

$T$ frictional force parallel to slip plane

TDC top dead center

$T_e$ elastic frictional force

$T_s$ frictional force in transition from stick slip to gross slip

$V_S$ supply voltage

$V_0$ output voltage

$V_0$ output voltage

$V_S$ supply voltage

x Normalized pad length, $0 < x < 1$
1 Introduction

When designing an engine it is not possible to always avoid relative motion between surfaces in contact. The question is if this constitutes a threat and a potential problem as regards engine life. Figure 1 shows the structure surrounding the main bearing in a MAN B&W two stroke diesel engine. The bearings in the crankshaft and in the crosshead are exposed to a type of normal and tangential forces that could cause what is called fretting damage due to relative motion between the back of a bearing and its adjacent seat. This relative slip between bearing and seat during one revolution is illustrated through displacement levels over the contact area in Figure 2.

Fretting damage occurs between two bodies in mechanical contact undergoing small oscillatory displacements relative to each other. It is widely accepted that fatigue loading coupled with pressure between two touching components cause unexpected failures at stress levels well below the fatigue limits, c.f. Söderberg [1]. However, the extent of damage caused through fretting is extremely difficult to predict. The geometry of the components exposed to fretting and the load conditions will affect the type and the amount of damage in a number of ways. The material being used and the composition of the surrounding atmosphere will also have a great influence on the fatigue life. Furthermore, the local surface topology is of great importance to the development of fretting damage. All of these and several other factors seem to contribute to failure by fretting fatigue in some unknown, collaborative way, making the fretting process extremely complicated. Unfortunately it is difficult to isolate and examine the effects of specific variables one by one, and due to the complexity there is no stringent definition of the phenomenon of fretting and no exact limiting values for the displacements when fretting transits to reciprocating sliding, were the effects from fretting wear are neglectable and any damage inherits from pure wear. However, Ohmae and Tzukizoe [2] and Gordelier and Chivers [3] have proposed an upper limit of relative displacement between surfaces for such a transfer to take place in the interval of 150 – 300 µm.

Fretting failure, or unintended friction welding, i.e. sticking between the surfaces of the bearings, cannot yet be predicted accurately from conventional fatigue analysis. Instead, engine designers need to overcompensate for these phenomena by the use of coatings and high-viscosity oils, or by forcing the components to stick to each other unnecessarily tight. The result might be less effective bearings which, in the end, will lead to a lower specific power of the engine. Also, because of the lack of understanding of the origin of fretting damage, the maintenance crew needs to spend extra effort and money when serving the engines, for example due to the removal of a 100 kg bearing in the crankcase which is welded stuck to the bulk bedplate. With a better understanding of the failure mechanisms, engine design engineers can produce more efficient engines. The vessels will be safer with a reduced chance of critical engine failure and maintenance costs will be reduced as prediction of failure becomes more accurate. As the steel prices are roaring upwards, also the potential weight savings are quite welcome as a cost-down.
Figure 1. Structure around main bearing in a MAN B&W two stroke diesel engine. The stresses shown arise from the pre-tensioning of the main bearing studs.

Figure 2. Relative slip between the main bearing, its support and the main bearing cap during one revolution shown as displacement levels.

2 Objective

The objective of this work is to find the limiting relative displacement amplitude for a known contact pressure to prevent fretting damage and to investigate how the friction and the contact pressure interact with respect to fretting. Furthermore, experiments correlated with finite
element calculations will be used to predict if fretting is present, and the potential consequences of a critical failure.

3 Experimental and numerical approach

The main bearing supporting the crank shaft in the crank house, c.f. Figure 1, will be investigated with respect to fretting damage in the contact between the bearing and the seat. An experimental setup with materials cut from a bearing that has been in service and seating material from an engine installed in the crude oil tanker Star Ohio, c.f. Figure 3, has been developed in cooperation with Mr. Zivorad Zivkovic, Research engineer, at the laboratory at the division of Material Science at Lund University. A first set of experiments were aimed at estimating the material properties and a second to basically simulate different load conditions in a running engine. If fretting damage was generated, the type and also the amount of fretting were determined by investigating the contact surfaces in a Scanning Electron Microscope (SEM). Theoretical predictions and experiments were preformed alongside. The initial numerical models based on assumptions of material parameters as well as of friction behaviour use a matlab script developed by Andersson and Persson [4]. More advanced models, based on data from the experiments, were developed using Hyper Mesh [5] form Altair Engineering. For Finite Element (FE) calculations the commercial FE code Abaqus [6] was used.

Figure 3. The crude oil tanker Star Ohio built in 1992. Length: 274m.

4 Theory of fretting

The bank of experiments on fretting is immense, but due to the confusion in literature concerning the distinction between different types of fretting mechanisms it is difficult to compare results between different reports. This chapter will outline some general concepts regarding different types of fretting and how they have been reported in various articles.

4.1 Fretting degradation

Due to fretting damage three different types of surface degradation can be observed, generally identified as fretting fatigue, fretting wear and fretting corrosion [7]. The life of a component
is determined by one or a combination of these effects, and the damage caused by fretting tends to look very different depending on which of the effects that dominates.

Fretting wear is the first documented type of fretting [8]. It is a surface damage phenomenon and arises due to abrasion between two surfaces in contact. Fretting wear is distinguished from sliding wear by the small relative slip amplitudes, which ensures that the main part of the contact surfaces never are exposed to the surrounding environment [1]. This means that worn of, deformation hardened and oxidized particles have difficulties leaving the contact zone, where they thus cause severe damage, c.f. Figure 4. On the other hand, if metal to metal contact is prevalent, the damage might result in local welding, roughening of the surfaces and an increase in the coefficient of friction $\lambda$ [9].

Fretting corrosion is caused by the combination of corrosion and abrasive wear effects due to corrosion product debris. Obviously the rate of the chemical attack varies with material and surrounding atmosphere, but fretting corrosion also shows other characteristics, such as fretting scars, c.f. Figure 4 [7], possibly due to changes in temperature and to diffusivity. An example of fretting corrosion is shown in Figure 5, where a PMC connector exposed to fretting corrosion is shown.

![Figure 4. Micrograph of fretting wear scar with cylinder on flat contact [10].](image-url)
Fretting fatigue denotes the initiation and propagation of cracks in the contact zone, even when the stresses are below the critical values at pure fatigue conditions; Figure 6 shows the development of surface cracks in a Ti-6Al-4V alloy due to fretting fatigue. Due to localised stress concentrations in the contact zone just before the surfaces start to slip, stress gradients are likely to be very high [8], which might lead to fretting fatigue damage.
4.2 Fretting regimes

Experimental observations as well as theoretical analyses show that the contact conditions during fretting change with displacement amplitude [1]. Depending on whether the contact surfaces stick to each other or slip relative to each other, three different types of fretting regimes can be identified. The first is known as the stick regime, the second as the mixed stick/slip regime and the third as the gross slip regime.

Many of the investigations published on fretting reports a variation in fatigue life with slip amplitude, where the life decreases with increasing slip up to a slip magnitude of approximately 50 µm, after which the life again increases [1], c.f. Figure 7. This conclusion, however, is drawn under the conditions of partial slip, i.e. the contact displays partial stick and partial slip, and variation in slip amplitude as a critical parameter may equally well be interpreted as a variation in frictional force $T$ in many of these reports [8].

The stick regime is characterized by extremely low displacement amplitudes. There is no interfacial sliding and the displacements between the contacting surfaces are mainly accommodated by elastic deformations in the regions near the surfaces of the two components. Figure 8 shows the relation between relative surface displacement amplitude $d$ and friction force $T$ acting parallel to the slip plane. The stick contact conditions will be maintained by adhesively joined asperities, which tend to be plastically sheared in the fretting direction. The contribution from this micro plasticity to the total frictional work can, however, be neglected as compared to the elastic deformation of the bulk material near the surfaces [13]. Thus both wear and corrosion mechanisms can be neglected in this regime. The nucleation and propagation of surface cracks due to the cyclic straining during fretting cannot be excluded, and although this effect is found to be small, [7, 14], fretting fatigue is the dominating mechanism in the stick regime.

The effects during the mixed stick/slip regime are visible on the parts of the surfaces where the interaction between the surfaces shows partly stick, as mentioned above, and partly includes other parts of the interaction area, where the surfaces slip relative one another. The sticking part of the interacting surfaces shrinks with increasing slip amplitude, as expected. The predominantly elastic shear deformation in the stick zone will transit to plastic shear deformation as the tangential force and the relative displacement exceed their critical values, and the conditions change to the slip regime.

In the mixed stick/slip regime plastic deformations occur in the asperities as well as in the underlying bulk material. Close to the stick/slip boundaries the material will experience stress gradients, and an extensive formation of surface cracks is nucleated in the annular slip area close to these boundaries. Possible propagation of the cracks depends on the material and the surrounding conditions; however there is no doubt that the formation of such cracks markedly reduces the fatigue life during cyclic loading in this regime. Wear and corrosion are more pronounced than during the pure stick regime but their effects on life are still neglectable as compared to the effects of crack formation.

The transition to gross slip is recognized by a drop in the frictional force as the entire contact surfaces start to slip relative to each other and the static friction conditions are replaced by kinetic, c.f. Figure 10. All asperity contacts are broken, which results in extensive plastic surface shear, eventually leading to delamination of scales. As mentioned before these...
particles have difficulties leaving the contact area, and thus further enhance the wear rate. Under corrosive conditions the delaminating and cracking of flakes is even further enhanced. Nucleated crack embryos will wear away before they can propagate [1, 7] and this will actually increase the fatigue life as compared to the mixed stick/slip regime.

Figure 7. Schematic diagram of the variation of fretting fatigue life and fretting wear rate with slip amplitude [8].

Figure 8. The elastic behaviour of frictional force parallel to slip plane $T$ vs. relative displacement amplitude $d$ in the stick regime.
4.3 Fretting Maps

The varying characteristics of the different types of fretting presented above demonstrate the importance of making clear distinctions between the three different regimes of fretting contact. One approach to classify experimental data is the use of fretting maps [7]. This is a diagram in two variables which shows the different regimes of fretting and the critical values for the regime boundaries. Each regime is characterised by its fretting mechanism and can therefore be distinguished from dynamic measurements of test parameters such as frictional
force, normal force, specimen displacement or frequency of vibration, to name a few. The
construction of a fretting map requires the independent variation of two variables, while those
remaining are considered constant parameters. It is well known that different testing
conditions affect the obtained fretting damage, and the ability to distinguish two variables and
to keep the rest constant determines the accuracy of the fretting map. Despite the some times
questionable accuracy due to lack of information about important parameters, fretting maps
can be used as tools to interpret experimental data from different reports. Figure 11 shows a
fretting map visualizing the three different fretting regimes with average contact pressure $P/A$
and displacement amplitude $d$ as the two independent variables.

![Fretting Map](image)

Figure 11. A fretting map of average contact pressure $P/A$ vs. relative displacement amplitude $d$ showing the different fretting regimes for anodised aluminium.

### 4.4 Prediction of crack initiation zones

Fretting involves large normal and shear stresses in the contact zone. High stress magnitudes
and a multi-axial stress state initiate and nucleate cracks at or near the contact zone which,
eventually, might cause failure. There have been several studies comparing fretting and pure
fatigue in order to predict the fretting fatigue life by introducing one or more test variables
[12,15, 16]. To describe and predict the crack initiation and nucleation with respect to all
different parameters which affect the outcome in numerous ways is, however, a mammoth
undertaking, and much work still needs to be done in this area. Some different approaches to
predict the location sites and behaviours of cracks are presented below.

In some early attempts a stress intensity factor (SIF) that incorporates the effects of
alternating axial stress and friction force was developed by Namjoshi et al. [12]. In this
fracture mechanics approach, an initial crack length, or an estimate of a threshold for flaw
size that can be tolerated to provide an infinite fretting fatigue life, is assumed. Currently such
approaches are utilized to determine the loading conditions which could lead to crack growth.
Volvo Aero, amongst others, uses this method as a numerical approach to fretting in turbines.

Ruiz et al. [17] proposed two parameters specific to fretting fatigue conditions, where damage
due to fretting fatigue depends on the amount of work done by the friction forces between
contacting bodies. The first parameter is a measure of the friction energy density and is
defined as the maximum value of the product of shear stress in the contact surface, $\tau$, and
relative slip amplitude, \(d\). However, crack nucleation can also depend on the normal stress just below the contact surface, \(\sigma_T\), and, therefore, the second parameter is chosen as the peak value of the product \(\sigma_T d\). This product has been used to predict the location of fretting fatigue cracks. However, studies [16] have proven the method inadequate to demonstrate any clear correlation to number of cycles to crack initiation.

Strain based parameters have been developed from the observation that the fatigue life of a component depends on the response of the material to the applied strain at critical locations. The number of cycles to failure can be estimated as a linear function of the applied strain amplitude on a log-log scale so that:

\[
\left( \frac{\Delta \varepsilon}{2} \right) = \varepsilon_f \left( 2N_f \right)^{c'}
\]  

(4.1)

This is commonly referred to as the Manson-Coffin relation, Coffin [18] and Manson [19], where \((\Delta \varepsilon/2)_p\) is half the plastic strain range, \(\varepsilon_f\) the fatigue ductility coefficient, \(N_f\) the number of reversals to failure and \(c'\) is the fatigue ductility exponent. The plastic strain range \(\Delta \varepsilon_p\) is defined as:

\[
\left( \frac{\Delta \varepsilon}{2} \right)_p = \left[ (\varepsilon_{\text{max}} - \varepsilon_{\text{min}})/2 \right]
\]  

(4.2)

With \(\varepsilon_{\text{max}}\) and \(\varepsilon_{\text{min}}\) denoting the maximum and minimum values of \(\varepsilon_p\), respectively.

A similar relation for stress-life data as (4.1) was proposed by Basquin [20] and shows a linear behaviour on a log-log scale according to:

\[
\left( \frac{\Delta \varepsilon}{2} \right)_e = \frac{\sigma_f'}{E} \left( 2N_f \right)^{b'}
\]  

(4.3)

Here \((\Delta \varepsilon/2)_e\) is the elastic strain range, \(\sigma_f'\) the fatigue strength coefficient and \(b'\) the fatigue strength exponent. The total strain amplitude \(\varepsilon_a\) can then be expressed in terms of cycles to crack initiation \(N_i\) as the sum of equations (4.1) and (4.3):

\[
\varepsilon_a = \frac{\sigma_f'}{E} \left( 2N_i \right)^{b'} + \varepsilon_f \left( 2N_f \right)^{c'}
\]  

(4.4)

The highest value of \(\varepsilon_{\text{max}}\) does not necessarily result in the highest strain range since \(\varepsilon_{\text{min}}\) also varies. This variation in strain ratio, \(R_e\), needs to be accounted for in order to compare different data. The strain amplitude in equation (4.4) should thus be interpreted as maximum strain and corrected for the strain ratio to obtain:

\[
\varepsilon_a = \varepsilon_{\text{max},R_e} = \varepsilon_{\text{max}} \left( 1 - R_e \right)^m
\]  

(4.5)

which was suggested by Walker in 1970 [21] and where \(m\) is a fitting parameter and the strain ratio is defined as:
The strain-fatigue life relation equation (4.4) was modified by Smith et al. [22] to include the effects of mean stress for pure fatigue conditions. They used Basquin’s formulation (4.3) and multiplied this by the strain-life equation (4.4) according to:

$$
\sigma_{\text{max}} \varepsilon_a = \left( \frac{\sigma^\prime_i}{2} \right)^{2/b} \sigma^\prime_i \varepsilon^{\prime\prime}_f (2N_i)^{1/n_i}
$$

(4.7)

The left hand side is commonly referred to as the Smith-Watson-Topper (SWT) parameter. Except for the difference in magnitude the variation of the SWT parameter along a chosen contact interface trajectory is similar to the load amplitude variation along the same trajectory [16].

Under fretting conditions, the state of stress in the contact zone is complex and the principal stresses vary as functions of time during the load cycle. It is therefore of interest to define multi-axial fatigue parameters when predicting crack initiation. The critical plane approach has been developed based on observations that fatigue cracks often nucleate on particular crystallographic planes. Quantities, such as stress and strain, on the critical plane are then used to define the fatigue parameters. Basically, the critical plane approach finds the critical plane as the one at which the maximum resolved shear strain amplitude appears, and then uses the maximum normal stress acting upon this plane to calculate a mean stress.

To modify the SWT parameter to a critical plane multi-axial fatigue parameter, $\Gamma$, it is assumed that crack initiation occurs on the plane where the product of the strain amplitude and the maximum stress normal to the plane $\sigma_{\text{max}}$ takes on its largest value. Thus, $\Gamma$ is defined as:

$$
\Gamma = \sigma_{\text{max}} \varepsilon_a
$$

(4.8)

By using computed stresses and strains from FE analyse, the critical plane with polar angle $\theta$, $0 \leq \theta \leq \pi$, in a Cartesian coordinate system $(x,y)$, with origin at the surface, c.f. Figure 12, can be located using the stress transformation equation:

$$
\sigma = \frac{\sigma_{xx} + \sigma_{yy}}{2} + \frac{\sigma_{xx} - \sigma_{yy}}{2} \cos 2\theta + \tau_{xy} \sin 2\theta
$$

(4.9)

It has been shown [12] that the modified SWT parameter predicts location of crack initiation satisfactorily even if it has a strong pad size dependency and the predicted orientation of crack initiation is inaccurate.

The shear stress range (SSR) critical plane parameter is based on the shear stress $\tau$ on a critical plane. The shear stress range $\Delta \tau$ is defined as:

$$
\Delta \tau = \tau_{\text{max}} - \tau_{\text{min}}
$$

(4.10)
\( \Delta \tau \) is calculated in the contact region with \(-90^\circ < \theta < 90^\circ\), where the maximum SSR provides a critical plane. \( \tau_{\text{max}} \) and \( \tau_{\text{min}} \) are maximum and minimum shear stress values as a result of maximum and minimum applied axial load due to the cyclic load conditions, and are calculated through the following transformation equation:

\[
\tau = \frac{\sigma_{xx} - \sigma_{yy}}{2} \sin 2\theta + \tau_{xy} \cos 2\theta
\]  

(4.11)

In order to include the effect of the mean axial/shear stress on the critical shear stress range \( \Delta \tau_{\text{crit}} \), equation (4.5) proposed by Walker is once again employed, resulting in:

\[
\Delta \tau_{\text{crit}} = \tau_{\text{max}} \left(1 - R_{\tau}\right)^m
\]  

(4.12)

where \( R_{\tau} \) is the shear stress ratio \( R_{\tau} = \tau_{\text{min}} / \tau_{\text{max}} \) and \( m \) is a fitting parameter. The use of SSR has proven to predict orientation and initiation satisfactorily. However, it shows a dependence on pad geometry \cite{12}.

The modified shear stress range parameter (MSSR) includes both the maximum value of the normal stress \( \sigma_{\text{max}} \) and the shear stress range \( \Delta \tau_{\text{crit}} \) in the critical plane, which eliminates the effect of pad geometry \cite{12}. The MSSR parameter is considered as one of the prime fretting fatigue predictive parameters and determined by:

\[
\text{MSSR} = A\Delta \tau_{\text{crit}}^B + C\sigma_{\text{max}}^D
\]  

(4.13)

In this approach the first term \( \Delta \tau_{\text{crit}} \) is the same as in equation (4.12), \( \sigma_{\text{max}} \) is the maximum normal stress on the critical plane defined by maximum shear stress range, and \( A, B, C \) and \( D \) are fitting parameters.

![Figure 12. Illustration of equation 4.9 and 4.11](image.png)
5 Strain gauge measurements

The most common way of measuring strain is to use a strain gauge. This is a device in which the electrical resistance varies in proportion to the strains that the gauge is exposed to. The most widely used metallic strain gauge has a metallic foil arranged in a grid pattern, but it might also consist of very fine wires. To reduce effects of shear strain and the Poisson’s effect, the cross sectional area of the grid is minimized. The metallic grid pattern, or the wire, is bonded to the carrier, which is a thin backing that is attached directly onto the specimen. The material in a carrier is paper or plastic, with a thickness of between 0.02 and 0.1 mm. This backing allows the strain gauge, shown in Figure 13, to experience the same strain as the specimen at the point of application.

At small strains the strain gauge linearly changes its electrical resistance due to the change in strain in the specimen. The attachment onto the specimen, often by glue, has to be chosen so that the gauge is capable to follow the specimen movements. The ends of the wires are either welded to thicker connecting wires or to solder tabs.

![Figure 13. Schematic strain gauge](image)
Strain, $\varepsilon$, is a measure of the amount of deformation of a body due to an applied force. As pictured in Figure 14, strain is defined as the fractional change in length:

$$\varepsilon = \frac{\Delta L}{L}$$  \hspace{1cm} (5.1)

where $L$ is the initial length of the specimen and $\Delta L$ is the length increment due to the force.

![Figure 14. Definition of strain](image)

In unloaded condition the resistance $R$ in the wire will be given by:

$$R = \frac{4\rho L}{D^2 \pi}$$  \hspace{1cm} (5.2)

where $L$ and $D$ are the linear measures of the unloaded wire, c.f. Figure 14, and $\rho$ is the resistivity of the unloaded specimen. When the wire is exposed to load it will stretch, and the new length $L_1$ will be:

$$L_1 = L + \Delta L = L(1 + \varepsilon)$$  \hspace{1cm} (5.3)

The diameter of the wire will also change, to $D = D_1$, according to:

$$D_1 = D(1 - \nu \varepsilon)$$  \hspace{1cm} (5.4)

where $\nu$ is Poisson’s ratio, which is about 0.3 for metals. The resistivity will change to $\rho = \rho_1$ according to:

$$\rho_1 = \rho + \Delta \rho = \rho + \psi \rho = \rho(1 + \psi)$$  \hspace{1cm} (5.5)

where $\psi$ is defined as the fractional change in resistivity.

The new resistance, $R_1$, can thus be written as:

$$R_1 = R + \Delta R = R(1 + \varepsilon) = \frac{4\rho L (1 + \varepsilon)(1 + \psi)}{\pi D^2 (1 - \nu \varepsilon)^2}$$  \hspace{1cm} (5.6)

where $r$ is defined as the relative change in resistance:
For small deformations the expression can be approximated by:

\[ R_1 \approx R(1 + \varepsilon)(1 + \psi)(1 + 2\nu \varepsilon) \]  

(5.8)

By performing the multiplications and neglecting terms of second or higher order, the relative change in resistance can be written as:

\[ r = R_1 - R = \varepsilon + \psi + 2\nu \varepsilon \]  

(5.9)

and the gauge factor \( k \), defined as the ratio between relative change in resistance and strain, becomes [23]:

\[ k = \frac{r}{\varepsilon} = 1 + 2\nu + \frac{\psi}{\varepsilon} \]  

(5.10)

### 5.1 Bridge configurations

In general, strain measuring involves small strains, up to a few millistrain with millistrain defined as, \( \varepsilon \times 10^{-3} \). Therefore it requires very accurate measurements to detect the very small changes in resistance. To measure such small changes, the strain gauge is mounted in a Wheatstone bridge configuration with a supply voltage source \( V_S \), c.f. Figure 16 a.

The output voltage \( V_0 \) is given by the bridge equation:

\[ V_0 = \left( \frac{R_3}{R_3 + R_4} - \frac{R_2}{R_1 + R_2} \right) V_s \]  

(5.11)

When the resistance ratios satisfies:

\[ \frac{R_1}{R_2} = \frac{R_2}{R_3} \]  

(5.12)

the output voltage in the bridge equals zero and the bridge is said to be balanced. When a change in any of the resistances appears the output voltage will be non-zero. If e.g. the resistance \( R_3 \) is replaced by an active resistance in terms of a strain gauge, as shown in Figure 16 b, it will unbalance the bridge and produce a non-zero output voltage.

Now let the nominal resistance in the strain gauge be \( R_3 = R_G \). Then the strain induced change in resistance \( \Delta R \) can be written as:

\[ \Delta R = R_G k \varepsilon \]  

(5.13)
If it is further assumed that $R_1=R_2$, the bridge equation (5.11) can be written as function of strain:

$$V_0 = -\frac{k\varepsilon}{4} \left(1 + k \frac{\varepsilon}{2}\right) V_e \quad (5.14)$$

Note that the output voltage is nonlinear due to the presence of the second term involving the strain and the gauge factor.

One problem that arises during service is that the materials in the strain gauge are sensitive to temperature changes. Manufacturers of strain gauges are, today, attempting to build strain gauges that minimize such sensitivity by using gauge materials that compensate for thermal expansion in the specimen material for which the strain gauge is intended. These compensating strain gauges are, however, just reducing the sensitivity of thermal expansion, but they do not remove it. To reduce the effect even further, a second strain gauge can be introduced. This second gauge, called a dummy gauge and replacing $R_1$, is applied at the specimen in a way so that the strain in the specimen has little influence on it, for example as shown in Figure 17, where the dummy is placed transverse to the applied load. The two gauges are mounted in a half-bridge according to Figure 16c.

The temperature change in the specimen will affect the two gauges, the active and the dummy, in the same way, whereas the ratio between their resistances will not change. In this way the output voltage $V_0$, shown in Figure 16, will not change due to temperature changes in the specimen.

To make the bridge even more sensitive, the dummy gauge can be made active and included in a half-bridge configuration, as shown in Figure 16c. In the half-bridge $R_3$ is the original, active gauge and $R_1$ the dummy.

This way the output voltage will be linear and approximately twice the output from a quarter-bridge. Often one gauge is mounted at the specimen in tension and the other in compression, c.f. Figure 15.

Figure 15. Half bridge mounted with one gauge in tension and one in compression.
The output voltage is given by:

\[ V_o = -\frac{k\varepsilon V_s}{2} \]  \hfill (5.15)

To further increase the sensitivity of the bridge, all four arms can be active strain gauges as shown in Figure 16 d. The output voltage will in this case once again double to:

\[ V_o = -k\varepsilon V_s \]  \hfill (5.16)

![Figure 16. a. Wheatstone bridge, b. Quarter bridge, c. Half bridge, d. Full bridge.](image-url)
However, in equations (5.14)-(5.16) for the three bridges it is assumed that each bridge is initially balanced and generates zero output when no strain is applied. This is, however, an approximation. The tolerance of the resistances and the strain induced through mounting of the bridge generates an initial voltage offset. The offset voltage can be dealt with in two ways. One is to use a special offset-nulling or balancing circuit to rebalance the bridge. Another is to measure the initial offset voltage and compensate for it in the software [24].

6 Experimental procedures

6.1 Tensile test

Two different steels, from the seat and from the bearing, c.f. Figure 18 and Figure 19, from which the fretting test setup is crafted, are cut from a main bearing seating and a main bearing that has been in service. This suggests that the material properties have changed as compared to virgin properties, and to get as accurate material parameters as possible for the FE analyses a tensile test is a necessary step.
Figure 18. The bedplate of a S50MC engine where the location of the bearing seat is shown.

Figure 19. Crank house bearings of a L70MC-C engine.
Three test specimens from each material with the same geometry as used for the fretting experiments were produced. The specimen geometry is shown in Figure 34. During the crafting of the specimens the temperature was carefully supervised so that it would not increase sufficiently to affect the material properties. Due to maintenance of the servo hydraulic test rig for cyclic loading (MTS) some of the specimens from the seat had to be tested in a test rig where the strain rate was maintained constant during the entire test, and thereby no cyclic loading was possible.

**Seat steel**

The seat steel, which is cut out from an engine previously installed in the crude oil tanker Star Ohio shown in Figure 3, was examined in a common tensile test machine as well as in a servo hydraulic tensile test rig (MTS), and subjected to an increasing axial tensile force until failure. The test rig was given a constant velocity of 1 mm/s between the two grips holding the specimen. The relative velocity of the grips is not a very exact way to get an understanding of what happens to the specimen in the elastic region because of the rather small displacements induced. This is due to the slip that always occurs between the specimen and the grips. To gauge the change in length of the specimen in a proper way an extensiometer was placed onto the specimen. The output variables: force, time and change of length, were imported to a PC and stored in a text-file.

Knowing the length of the extensiometer (25 mm) and the change of length at each time increment, the strain can easily be determined through:

\[
\varepsilon = \frac{\Delta L}{L}
\]  \hspace{1cm} (6.1)

The axial stress can be determined from the force and the cross section area by:

\[
\sigma = \frac{F}{A}
\]  \hspace{1cm} (6.2)

Finally, through Hook’s law, the Young’s modulus \(E\) is determined as:

\[
E = \frac{\sigma}{\varepsilon}
\]  \hspace{1cm} (6.3)

One specimen from the seat was tested in the MTS by cyclic loading. The relative soft seat steel provided stabilized loops in the strain range from 0.1<\(\Delta \varepsilon\)<1.5. The force, time and strain were imported into a PC, and the stress/strain curve is plotted in Figure 20.

**Bearing steel**

The bearing steel, which is taken from a bearing that has been in service, is examined in a servo hydraulic tensile test rig which enables cyclic loading and thereby a better estimate of material properties such as hardening parameters in the von Mises criterion according to:

\[
f(\sigma_y, K) = \left[ \frac{3}{2} (s_{ij} - \alpha_{ij}) (s_{kl} - \alpha_{kl}) \right]^{1/2} - \sigma_y - K = 0
\]  \hspace{1cm} (6.4)
where $s_{kl}$ is the deviatoric stress tensor, $\sigma_y$ denotes the initial yield stress, $\alpha_{kl}$ is the deviatoric backstress tensor which describes the kinematic hardening and $K$ is a hardening parameter which describes the isotropic part of the hardening [25].

The first specimen was loaded until rupture in order to get an approximation of the yield strength $\sigma_y$ and the tensile strength $\sigma_t$. The two other specimens were subjected to cyclic strain with constant amplitude until they showed a stable loop, c.f. Figure 21. By increasing the strain amplitude in relatively small steps after the curve had stabilized, it was possible obtain several sets of stabilized loops, all from the same test specimen, c.f. Figure 22. The strain was monitored by an extensiometer, and a sine shaped strain with frequency 1 Hz was applied to the specimen. The output variables: force, time and strain were imported to a PC and stored in a text-file.

![Figure 20. Stabilized curves at different strain amplitudes for one specimen from the seat steel.](image-url)
Figure 21. A stress/strain diagram which shows a stabilized loop at a constant strain amplitude of 0.2%.

Figure 22. Stabilized curves at different strain amplitudes for one specimen from the bearing steel.
Test results
The tensile test curves from the seat steel show a very good reproducibility, indicating that the material properties are kept during the machining of the specimens. The yield strength $\sigma_y$ and tensile strength $\sigma_t$ are estimated to $\sigma_y = 260$ MPa and $\sigma_t = 450$ MPa, respectively, c.f. Figure 23. The Young’s modulus is calculated form the curves by using Matlab to evaluate a linear curve, c.f. Figure 24. The average value of Young’s modulus is determined to $E = 184$ GPa.

The tensile test curve form the first specimen of bearing steel, c.f. Figure 25, shows approximated values of the yield strength and the tensile strength to $\sigma_y = 290$ MPa and $\sigma_t = 436$ MPa. The first cyclic tensile test, where the material not yet has suffered any work hardening, confirms the approximated yield strength, c.f. Figure 21. A cyclic loop well below the yield strength provides sufficient data for a good approximation of Young’s modulus by using the curve fitting tools in Matlab, c.f. Figure 26. The Young’s modulus is estimated as $E = 187$ GPa. The stabilized curves from the cyclic tensile tests are shown in Figure 20 and Figure 22.

![Figure 23. Stress-strain curves for the three specimens of seat steel. The reason that the two curves, ending at approximately $\varepsilon = 0.12$ and $\varepsilon = 0.14$, having such brusque endings is the waist formation and fracture outside the extensometers gauge zone.](image-url)
Figure 24. Plus marks represent data from tensile test and the full line the evaluated Young’s modulus for the seating material.

Figure 25. Engineering Stress/strain curve from bearing steel.
To evaluate the bearing performance MAN B&W has developed an in-house software [4, 26]. The input to the bearing analysis is based on the output of another in-house software, COUVIP [27]. COUVIP calculates the resulting forces between the crankshaft and its support as functions of crankshaft angle, c.f. Figure 27. The software package has been used to optimise the performance of main bearings and, furthermore, to evaluate production related deviations and to give proposals for repair modifications. When applied in evaluating the performance of a bearing, a very fine mesh is required in the areas of the fluid, were steep pressure gradients are present, and due to the support from local refinement of the fluid mesh the calculations can be made at reasonable computational time. The outcome of these Elasto-Hydro-Dynamic analyses is oil film pressure and oil film thickness distributions as function of the crank angle, c.f. Figure 28. This program is also capable to calculate the orbit of the centre of the, by the bearing supported, crank shaft or piston bolt, c.f. Figure 29. The frictional forces that arises due to the tangential movement of the bearing can also be generated, c.f. Figure 30. These frictional forces will tend to rotate the main bearing, which is the main source of relative motion.

6.2 Fretting experiment

Figure 26. Dashed lines are obtained from cyclic tensile test and the solid line is the approximated Young's modulus.
Figure 27. Resulting force in a main bearing as a function of crankshaft angle with respect to TDC in a cylinder [27].
Figure 28. Normalized oil film pressure in a bearing at different crank angles [26].
Figure 29. The orbit of the centre point of a bearing during one revolution [26].

Figure 30. Friction force as function of crank shaft angle calculated by MainBeast for a main bearing [26].
Simplifications between engine condition and experimental setup
Since the bearing is shrink fitted into the seat, and the loads arise from the rotating crank shaft via the oil film, it is necessary to make simplifications as regards the structure and the loading conditions in order to investigate the failure mechanism. The circumfencial contact due to the circular seat is replaced by a finite flat on flat contact, which represents one particular section of the entire contact. The complicated load conditions from the oil film are approximated by a constant normal force $P$, controlling the contact pressure, and a cyclically varying axial load $F$ which introduces the relative displacement amplitude, c.f. Figure 31. The normal force can be changed between different setups and depending on which part of the contact to be simulated contact pressures from 5 MPa up to 10 MPa will be investigated. The engine pretty much maintains a constant rpm most of its running time, which suggests that the cyclic load should be applied with constant amplitude.

Experimental setups

Setup A
A setup with flat on flat contact is created using four pads on two clamps mounted on a dog bone shaped rod. The clamp is manufactured from the material of the bearing with dimensions according to Figure 34. The dog bone shaped rod is made from the somewhat softer seat material. The contact surfaces on the rods as well as the pads are carefully machined and polished to achieve a very smooth finish, in order to avoid any discontinuities from one setup to another. The fretting damage should actually increase with a more highly polished contact surface because it in such a case is more difficult for the debris in escaping into hollows with smoother surface, and less relative motion can be absorbed by elastic deformation of the asperities. The load $F$ contributes to a tension in the specimen, and thereby relative motion will arise in the contacts between the pads and the specimen. The normal load $P$ is controlled by a screw and monitored by a load cell, c.f. Figure 32. The load cell, built by Mr. Zivorad Zivkovic, consists of a s-shaped steel construction with two strain gauges measuring the elastic tension at the centre.

Setup B
A second setup, shown in Figure 33, is manufactured with the purpose of eliminating fatigue of the specimens due to the cyclic load $F$ and, thereby, provokes fretting wear to be the dominating type of degradation of the contact. The load $F$ is now governed by the motion control by the MTS and $F$ depends on the friction in the contact and is logged to a PC.
Figure 31. Test setup with normal and axial load.

Figure 32. Experimental Setup A.
Figure 33. Experimental Setup B.

Figure 34. Dimensions of rod and clamp in experimental setup.
7 Numerical simulations

7.1 FE modelling

3D vs. 2D
Three-dimensional (3D) effects due to finite contact width will be present in the experimental setup. A resent study [10] has, however, shown that the fretting characteristics very well can be approximated by two-dimensional plane strain if the loading takes place under plane conditions. A 3D model with flat on flat contact which measured 6.35x13.97mm were investigated. Stresses and relative displacements along the edges were compared to the corresponding stresses and relative displacements along the centre of the specimen in order to visualize eventual differences, c.f. Figure 35. The differences in maximum peak stress between centre and edge of the specimen were 3\% and 17\% for normal ($\sigma_{xx}$) and shear ($\tau_{xy}$) stresses, respectively. Crack initiation was examined using the modified shear stress range (MSSR) parameter which showed that the effect due to the free edge boundary was less than 2\%. Due to the reasonably good approximations and the relative numerical efficiency the numerical simulations will be conducted for the two-dimensionally case, only.
Figure 35. Boundary effects from 3D analysis for flat pad: (a) contact pressure, and (b) relative displacement [10]. EOC denotes end of contact.
Geometry
A two dimensional model based on the plane strain assumption is built in HyperMesh. Because of the symmetry conditions the FE-model only involves one quarter of the specimen and one half of one of the pads, c.f. Figure 36 and Figure 37. To avoid unnecessary high stress gradients along the boundaries, the pads have a round off radius of 0.25 mm.

Mesh layout
The elements used are Abaqus CPE3 and CPE4, which are triangular and rectangular plane strain elements of first order with three and four nodes, respectively. The element density is increasing towards the contact region where the smallest elements are located, as Figure 38 shows. The element size in the contact region is 9.5x9.9 µm, and the total number of elements in the model is 59 040.

Figure 36. Simplified complete model with symmetry lines.
Figure 37. Model showing boundaries and load steps.
Constitutive models
The model involves two materials and therefore needs two material definitions, one for the bearing steel in the clamp and one for the seat steel in the specimen. The value of Young’s modulus $E$ is taken from the tensile tests described earlier in the text and the Poisson’s ratio $\nu$ is chosen to be 0.3. To define the plastic behaviour of the materials, values from the curves generated in the tensile tests are chosen, c.f Figure 20 and Figure 22. These values start with respective materials initial yield strength, with zero plastic strain, and ends up with the respective tensile strength with given plastic strain. The cyclic tests for the bearing steel provide several stabilized curves, each from which the kinematic hardening can be determined according to the stabilized cycle algorithm in Abaqus [28].

Boundary condition
The model represents, as mentioned, one quarter of the specimen, and one half of the clamp. This means that the rest of the specimen and clamp is replaced by boundary conditions in order for the model to behave as if it is a full scale model. On the specimen two boundaries occur, one along the symmetry line in the $x$-direction ($\text{SL1}$ in Figure 36) and one along the symmetry line in the $y$-direction ($\text{SL2}$ in Figure 36). Along $\text{SL1}$ movement is prevented in the $y$-direction and at $\text{SL2}$ movement is prevented in the $x$-direction. This means that all deformations in the model will be relative to the cross sections of the two symmetry lines. In the second experimental setup with motion control the boundary condition at the clamp are described by a cyclic displacement according to equation 7.1.

Figure 38. The mesh and the enlarged upper contact region.
**Interaction**

To provide interaction between the specimen and the clamp the HyperMesh tool Contact Manager is used. This establishes a relationship between a slave surface (nodes on the clamp) and a master surface (nodes on the specimen). The manager determines which segments of the master surface that interacts with which nodes on the slave surface, and provides an algorithm for the transfer of loads between the two surfaces. The Lagrange multiplier formulation of friction is used. With this the sticking constraints at the interface is enforced exactly, which means there is no relative motion between the two closed surfaces until the criterion $\tau = \tau_{\text{crit}}$ is met, with $\tau_{\text{crit}}$ denoting a critical shear stress value determined by the stress perpendicular to the contact surface $\sigma_{xx}$ and the coefficient of friction $\lambda$. The Lagrange multiplier is used in fretting calculations where the relative displacements in contact are small.

**Loading conditions**

To produce the loading conditions needed in the model, the HyperMesh tool Step Manager is used. This tool allows the user to create loads that will be applied to the model in steps. In these simulations two steps are used, where each step applies one load sequence onto the model. The first one is a ramped concentrated load $C_{\text{load}}$ in the negative $x$-direction on the right hand side of the clamp. This load provides and controls the necessary contact pressure between the clamp and the specimen. The second load is a cyclic surface load $D_{s\text{load}}$ in the $y$-direction at the top of the specimen. The cyclic load is a sinusoidal load defined by a Fourier series according to:

$$ a = A_0 + \sum_{n=1}^{N} \left[ A_n \cos \left( n\omega \left( t - t_0 \right) \right) + B_n \sin \left( n\omega \left( t - t_0 \right) \right) \right] \quad \text{for} \quad t \geq t_0 \tag{7.1} $$

$$ a = A_0 \quad \text{for} \quad t < t_0 \tag{7.2} $$

where $t$ denotes time and $A_0$, $B_n$ and $t_0$ are constants. The constant term $A_0$ is constituted somewhat higher than the total stress range in order to achieve an initial tension in the specimen and, thereby, a stress ratio of $0 < R_{\varepsilon} = \sigma_{a,\text{max}} / \sigma_{a,\text{min}} < 1$. In the second experimental setup, where cyclic fatigue is not present, the second load in the $y$-direction is replaced by the displacement boundary condition.

**Numerical validation**

To verify the potency of the model, it is compared to results from previous works and experiments. Older models are limited by the lack of numerical efficiency as compared to the performance of today’s computers and, therefore, only more recent models should be considered. However, most of the recent experimental setups and models are preformed with aluminium and titanium alloys or very high strength steels. These materials are, of course, different from common steel in several aspects. They are, however, linear elastic, and their behaviour in the elastic region is much alike the one for common steel. This suggests that they very well can be compared to steel within these frames in terms of stress distribution. In order to achieve the same relative conditions as the experimental setups to be compared with, the load conditions are chosen within the elastic range limited by the material definition for the commercial steel used in this investigation.

The contact pressure $\sigma_n$ is validated against a model which investigates the 3D effects of finite contact [10]. The flat on flat contact area $A$ is 13.97 x 6.35mm and normalized contact pressure is shown in Figure 39. The material used in the model is alloy Ti-6Al-4V, which
often is found in aircraft applications due to its high strength to weight ratio. The length of the elements in the contact zone is about 7.5 µm. The normalized contact pressure \( \sigma_n A/P \) is compared in Table 1. The somewhat asymmetric contact pressure inherits from the geometry of the clamp, which distributes more stress towards the lower edge of contact as compared to the upper.

The stress distributions in the contact zone at the peak axial tension are compared to a similar model [29] based on alloy Ti-6Al-4V, and with a minimum element size of approximately 6.5 µm. The contact area \( A \) is 19.05 x 10.00 mm and the average contact pressure \( P/A \), and axial stress range \( \Delta \sigma_a \) is 110 MPa < \( \Delta \sigma_a \) < 275 MPa. The stresses from the edge to the centre of the contact are compared in Table 2 and visualized in Figure 40. The somewhat different shapes of the curves may be explained by some bending of the clamp around the z axis due to the friction forces in the contact.

Crack initiation, based on the SSR parameter, is compared to an experimental setup, much similar to the setup created in this investigation, by Nakazawa et al. [30] on austenitic stainless steel. Two clamps with the pad size of 2 x 8 mm are mounted on a dogbone shaped specimen with a cross sectional area measuring 8 x 10 mm. A round off radius of 1 mm for the specimen results in the contact zone \( A \) of 6 x 2 mm. The average contact pressure \( P/A \) and the axial stress range \( \Delta \sigma_a \) together with the corresponding crack initiation predictions are presented in Figure 41. The actual failures of the specimens are seen in Figure 42. Extreme peaks in the stress distribution at the EOCs (end of contact zone) are inevitable due to the non continuous geometry in the round off edges of the model. Despite these peaks, the SSR parameter suggests that failure might occur at \( x = 0.48 \) in the critical plane, where \( x \) is normalized pad length, described by \( \theta = 44.3 \) degrees for the lower contact pressure. If the contact pressure is increased, the critical SSR value transits towards the lower EOC with \( x = 0.03 \) and \( \theta = -10.3 \) degrees.

| Table 1. Comparison of normalized contact pressure between calculated results and data achieved from an investigation of 3D effects of finite contact [10]. |
|-------------------------------------------------|----------------|----------------|----------------|----------------|----------------|----------------|
| \( P/A \) (MPa) | \( A \) (mm²) | \( \sigma_{normal,0} \) | \( \sigma_{normal,0.10} \) | \( \sigma_{normal,0.30} \) | \( \sigma_{normal,0.50} \) |
| Literature data (MPa) | 89.9 | 88.7 | 7.5 | 1.1 | 0.8 | 0.75 |
| Calculated values (MPa) | 90 | 28.0 | 5.5 | 1.1 | 0.9 | 0.8 |
Table 2. Comparison of stress distribution around the upper deformed edge of contact between data achieved from a recent paper [29] and calculated values.

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<th>$\tau_{xy,\text{max}}$</th>
<th>$\sigma_{xx,\text{min}}$</th>
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</table>

Figure 39. Normalized contact pressure for a. model from [10] and b. current model.

Figure 40. Stress distribution around the upper deformed edge of contact for a. model from [29] and b. current model. Positive stresses shows compression.
Figure 41. The SSR parameter and the direction of critical plane for simulations with contact pressure of a. 7.5 MPa and b. 50 MPa and an axial range of 150 MPa.
7.2 Software

To model the experimental setup Matlab [31] and Hypermesh [5] have been used. The finite element code Abaqus [6] has been used for the FE calculations, and to view the results Abacus and Hyperview [5] have been employed.

The Matlab script was first developed by Persson and Andersson [4] at LTH in 1998. This script is used to produce a simple 2D-mesh to be implemented in Abaqus. The script has been used in this thesis to produce the first simple models.

For building the more complex model the program Hypermesh from Altair Engineering is used. Hypermesh is a pre- and postprocessor for finite element solvers, such as Abaqus. The program generates an Abaqus input file, INP-file, which can be directly imported to ABAQUS/Standard.

ABAQUS/CAE is divided into two modules, one for logical aspects of modelling such as meshing and generating an input file, and one module for monitoring and visualization of the output. The first module is not used in this analysis, as all the modelling and meshing is done using Hypermesh/Matlab.

ABAQUS/Standard is used to solve the finite element model. During the analysis, the program sends information to ABAQUS/CAE allowing monitoring the progress of the run, and generates an output database. The second module, the visualization module in
ABAQUS/CAE, can now read the information and visualize the results from the output database.

Altair HyperView is a complete post-processing and visualization environment for finite element analysis, multi-body system simulations, video and engineering data. HyperView supports many common CAE solver formats and in this thesis an Abaqus database has been directly imported.

8 Analysis of results

8.1 Experimental results

Setup A
The first experiments on Setup A used a global contact pressure of $P/A = 10$ MPa and an axial stress range of $\Delta \sigma_a = 200$ MPa. These experiments showed a broad variety of contact pressures over the contact area. This was due to that the relatively low global contact pressure $P/A$ resulted in local pressure peaks because of differences in topology, and the fact that the areas not were perfectly plane. Debris from the softer seat material in the specimen gathered in the local pressure peaks and built up a new, smoother, surfaces over the irregularities in the cyclic loading direction, c.f. Figure 43 and Figure 44. As new surfaces emerge perpendicular to the contact direction more material is torn off from the seat steel and the fretting scar grows deeper. At these local peaks in topology, local pressure peaks will increase further and, eventually, lead to crack initiation as shown in Figure 45.

The relative slip between different pads and the specimen varies somewhat depending on which of the four global contact zones that is investigated. Plastic deformations shown in Figure 46 and Figure 47 points towards a relative slip of $d$ in the interval $10 \mu m < d < 30 \mu m$, depending not only on choice of global contact zone but also on number of cycles $N$ since different layers show varying lengths of their scars. The amount of debris and flaws in the area affected by fretting and the fact that cracks only initiate in the build up surfaces prove that, even though fatigue stresses are present trough the axial stress $\sigma_a$, the dominating surface degradation mechanism is fretting wear.

Further experiments on Setup A using a constant axial stress range of $\Delta \sigma_a = 250$ MPa and different normal loads $P$ show the effect of increasing contact pressure. The relative slip amplitude $d$ shows no visible change in the interval of $5$ MPa $< P/A < 10$ MPa, c.f. Figure 48. The fretting scar after $10^5$ cycles from the setup with average contact pressure 10 MPa is, however, almost twice the size compared to the scar from the setup using an average contact pressure of 5 MPa.

Setup B
Results from Setup B proves the fact that a specific contact zone shows signs of variation in slip amplitude $d$ with number of cycles $N$. Large displacement amplitudes, up to $98 \mu m$, allows the deviations to be distinguished with the naked eye and clearly shows that, instead of sharing the relative motion between the upper and lower pads about equally, the surfaces alternates in absorbing almost the entire relative motion at one side. The alternation between the sides continues until the friction has reached its maximum and stabilizes on one side after approximately $500 < N < 1000$, c.f. Figure 49.
Further experiments on Setup B were performed with the upper side of the pads fixed to the specimen using fast hardening glue. Hereby the slip in the remaining two contact zones are known at all times, and this allows a more thorough investigation of the friction behaviour. Different slip amplitudes were applied and the resulting coefficients of friction are shown in Table 3 and Figure 50. The coefficient of friction varies slightly depending on whether the specimen is pressed up or pulled down, and these values are represented as $\lambda_{\text{pull}}$ and $\lambda_{\text{press}}$. At a slip amplitude of $d = 8 \mu m$, i.e. a total relative displacement of $16 \mu m$, the fretting scar is almost non-existent after $10^5$ cycles, and at the slip amplitude $d = 3 \mu m$ the axial load stabilizes to constant pressure after the first $10^4$ cycles, c.f. Figure 51. This suggests that most of the relative displacement of about $6 \mu m$ is entirely absorbed by elastic motion within the contact zone, and therefore do not cause any substantial fretting damage.

The friction shows a strong increase with number of cycles in the beginning of the experiment for $d > 15 \mu m$ and peaks at about $500 < N < 1000$ cycles after which it slowly decreases. After $10^4$ cycles the continuous decrease in friction is impossible to distinguish from the instantaneous variations, and hereafter the coefficient of friction is referred to as the stabilized coefficient of friction. The increase in friction coefficient is shown in Figure 52 for different slip amplitudes. The variation in the stabilized coefficient of friction at different slip amplitudes is shown in Figure 53 for $10^4 < N < 10^5$.

Results from the first set of experiments, using Setup B, aimed at investigate the friction dependence on contact pressure and the results are shown in Figure 54. The plot shows the stabilized coefficients of friction against maintained contact pressure during $10^4$ cycles. The slip amplitude is kept at $d = 28 \mu m$ and the normal force governing the contact pressure is also shown in the plot on the right Y-axis. The results does not show any obvious dependency on contact pressure for the friction coefficients, and the slowly decrease in coefficients of friction with number of cycles is in accordance with previous results, c.f. Figure 53.

Next set of experiments with Setup B investigating friction dependency on contact pressure is performed by continuously changing the normal force within the range $5\text{ MPa} < P/A < 10\text{ MPa}$ instead of momentary adjustments, when the axial displacement is kept constant. In agreement with previous experiments regarding stabilized friction and momentary increase in the normal force, the results shown in Figure 55 reveal that there is no significant change in coefficient of friction with contact pressure in the range $5\text{ MPa} < P/A < 10\text{ MPa}$.

**Fretting map**
A number of experiments have been conducted in an attempt to create a fretting map, relating contact pressure and relative slip. Several slip amplitudes were tested at different contact pressures, and the stick regime were distinguished by viewing the behaviour of the load. The initial coefficient of friction of $\lambda = 0.13$ permits gross slip at very low slip amplitudes. However, prior to 500 cycles the friction has increased, and a transition to the stick regime taken place. Because the friction is not identical between press and pull of the specimen, stick in one direction does not necessarily mean stick in the opposite direction. This will often lead to a translation of the position of equilibrium, and has as consequence that the axial load no longer oscillates around 0 N, c.f. Figure 51. Figure 56 shows a transition to the stick regime after approximately 300 cycles. The results suggest that stick is introduced at different slip amplitudes for different average contact pressures. From corresponding values of $P/A$ and $d$ the fretting map in Figure 57 is constructed.
The fretting scars on the specimens were also investigated in the electron microscope. The relative slip seems to very well coincide with the gathered data, c.f. Figure 58. The difference in wear between a specimen exposed to fretting during the stick regime and a specimen exposed to fretting during the slip regime are clearly separable regarding debris formation, c.f. Figure 59. A fretting map of wear induced by debris formation after 50,000 cycles is presented in Figure 60 and it is obvious that it coincides with the transition from the stick regime to the gross slip regime.

Figure 43. Debris gathering around the asperities perpendicular to the cyclic loading direction at the edge of a local pressure peak on the pad. Load direction is vertical.
Figure 44. Fretting scar at a local pressure peak. Debris is filling up in the scratch perpendicular to the loading direction. Load direction is vertical.

Figure 45. Cracks on the surface of the pad in material build up by debris from the specimen.
Figure 46. Scars in the specimen due to relative motion in the lower contact zone. Load direction is vertical.

Figure 47. Scars in the specimen due to relative motion in the upper contact zone. Load direction is vertical.
Figure 48. Pictures of the scar after a cyclic axial load peak at 255 MPa and average contact pressures of 5 MPa, left figures, and 10 MPa, right figures.
Figure 49. Coefficient of friction vs. number of cycles from a test with \(d=28\mu m\) and \(P/A=10\) MPa

Table 3. The coefficients of friction as \(\lambda_{\text{pull}} / \lambda_{\text{press}}\) at different slip amplitudes and number of cycles.

<table>
<thead>
<tr>
<th>(d)</th>
<th>(N=1)k</th>
<th>(N=10)k</th>
<th>(N=40)k</th>
<th>(N=100)k</th>
</tr>
</thead>
<tbody>
<tr>
<td>16 (\mu m)</td>
<td>0.35/0.35</td>
<td>0.29/0.30</td>
<td>0.23/0.31</td>
<td>0.02/0.47</td>
</tr>
<tr>
<td>26 (\mu m)</td>
<td>0.67/0.75</td>
<td>0.54/0.64</td>
<td>0.53/0.63</td>
<td>0.52/0.61</td>
</tr>
<tr>
<td>46 (\mu m)</td>
<td>0.68/0.66</td>
<td>0.74/0.77</td>
<td>0.68/0.71</td>
<td>0.64/0.67</td>
</tr>
<tr>
<td>96 (\mu m)</td>
<td>0.79/0.90</td>
<td>0.73/0.82</td>
<td>0.75/0.82</td>
<td>0.67/0.75</td>
</tr>
<tr>
<td>196 (\mu m)</td>
<td>0.81/0.82</td>
<td>0.82/0.85</td>
<td>0.78/0.80</td>
<td>-</td>
</tr>
</tbody>
</table>
Figure 50. 3-D plots of the coefficient of friction versus relative displacement and number of cycles.
Figure 51. Stabilized axial load at $N = 10^4$ for $d = 3\mu m$ and $P/A = 10\text{MPa}$. 
Figure 52. The coefficient of friction vs. number of cycles for different slip amplitudes, upper left $d = 16 \, \mu m$, upper right $d = 26 \, \mu m$, lower left $d = 46 \, \mu m$, lower right $d = 96 \, \mu m$, at a contact pressure of 10 MPa.
Figure 53. The coefficient of friction for $16 \, \mu m < d < 96 \, \mu m$ and $10 \, N < 10^5$.

Figure 54. Coefficients of friction and normal force vs. number of cycles. The contact pressure increases from 5 MPa up to 10 MPa and decreases again to 5 MPa. Slip amplitude 60 µm.
Figure 55. Variations of friction coefficient with average contact pressure $P/A$ altered continuously.
Figure 56. Axial load and axial load / normal load vs. number of cycles visualizes a transition from the slip regime to the stick regime.
Figure 57. Fretting map of stick regimes relating contact pressure and relative slip.

Figure 58. Lucky shot of a scratch made by a particle partially sliding sideways through the contact zone. The average contact pressure is 10 MPa and the relative slip is estimated to 16 µm. Slip direction nearly vertical.
Figure 59. The upper picture shows a scar from the stick regime after 50,000 cycles at $P/A = 10 \text{ MPa}$ and $d = 15 \mu\text{m}$. The lower picture shows a scar from the gross slip regime after 50,000 cycles at $P/A = 10\text{MPa}$ and $d = 20 \mu\text{m}$. 
8.2 Numerical simulations

The first calculations simulating the conditions of Setup A with the coefficient of friction kept constant at $\lambda = 0.13$ are performed to estimate the relative motion of the contact zones and to get a first estimate of the stress distribution, c.f. Figure 61 and Figure 62, because there is a factor of 10 between the stress in the clamp and in the specimen when the axial load is applied, the von Mises equivalent stresses is visualized on two different scales in Figure 62. There are stress peaks at the EOC:s, and at the lower EOC are somewhat higher because the normal load elastically bends the clamp around the Z-axis. The results shown in Table 4 suggest that the contact pressures are too low to affect the slip amplitude more than 0.5 $\mu$m, approximately. The variation in slip within the contact zones is shown in Figure 63.

FE simulations of Setup B show a significant change in stress distribution depending on if the specimen is pressed down or pulled up, c.f. Figure 64. Based on results from the experiments the coefficient of friction is set to $\lambda = 0.7$. The displacement of the pad is shown in Figure 65 as a function of the displacement of the specimen. The result suggests that approximately 4 $\mu$m are absorbed as elastic tension and bending of the clamp and any further movement of the specimen will result in relative slip in the contact zone.
Figure 61. The stress distribution with the normal load resulting in an average contact pressure of 8MPa.
Figure 62. Stress distribution when axial load and normal load provides tension and pressure of 300 MPa and 8 MPa, respectively.
Table 4. Estimates of relative slip for $\lambda = 0.13$.

<table>
<thead>
<tr>
<th>$P/A$</th>
<th>$\sigma_a = 150$ MPa</th>
<th>$\sigma_a = 200$ MPa</th>
<th>$\sigma_a = 220$ MPa</th>
<th>$\sigma_a = 250$ MPa</th>
<th>$\sigma_a = 280$ MPa</th>
<th>$\sigma_a = 300$ MPa</th>
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<td>$COC$</td>
<td>$EOC_{lower}$</td>
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<td>5 MPa</td>
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<td>11.3 µm</td>
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<tr>
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<td>9.9 µm</td>
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Figure 63. Axial load vs. slip during one load cycle at different locations within the contact zone. The average contact pressure is 8 MPa.
Figure 64. The difference in stress distribution when the specimen is pressed down, left figure, and pulled up, right figure.
Figure 65. The pad displacement vs. specimen displacement in mm along the vertical axis during one load cycle. Start and end of the load cycle at * and ◊, respectively.

8.3 FE model as compared to experimental conditions

To compare the model of Setup A, the relative slip calculated numerically in Abaqus using axial loads, normal loads and an estimated coefficient of friction similar to the specific conditions of each experiment are displayed in Figure 66 together with estimates of the actual experimental slip achieved from investigations in the electron microscope. The estimates seem fairly agreeable to the actual slip achieved from the experiments, although the exact slip amplitude is difficult to determine due to the variations in topology over the specimen surfaces.

The model shows a distinct variation in stress distribution whether the specimen is pulled up or pressed down. The change in stress distribution suggests that different parts of the contact zone contributes with local stress peaks depending on in which direction the specimen is moving. Different parts of the contact have different friction and different changes in friction and, thereby, the different coefficients of friction achieved in the experiments can be explained from the FE model.
9 Discussion

In Setup A, the fretting scar after $10^5$ cycles is almost twice the size at an average contact pressure of 10 MPa as compared to a contact pressure of 5 MPa. This is a phenomenon that can be explained by that the contact surfaces not are entirely flat. Small peaks in the surface topology remain from the machining of the specimens. These peaks can, at low contact pressures around 5 MPa, counteract a full contact, but when the contact pressure has increased to about 10 MPa, some of these peaks are plastically flattened out and a larger contact zone is possible. However, it is important not to confuse this with the fretting map in Figure 60, where wear induced by debris formation increases at lower contact pressures because of the transition between fretting regimes.

In Setup B the pads do not share the relative motion equally in the first load cycles, up to about $500 < N < 1000$. Instead they alternate in accommodating the displacement between the upper and the lower pad until the coefficient of friction stabilizes. This means that it is hard to predict the slip amplitude, and the location of slip in this interval. But it is not impossible that, for some cases, the slip amplitude divides between the pads. This slip can take place in different directions between the pads and allows the entire clamp to slide along the specimen. The sliding often occurs in the very beginning of the experiments, when the fretting scar is starting to establish and different fretting regimes exists in different directions.
The surface finish of the specimen and the clamp in the contact zone has been produced using a polishing machine, where the parts have been hand held. This can have produced parts with surface finishes that is slightly different from experiment to experiment due to, for example, wear of the polishing paper, or uneven pressure between the paper and the pads. However, it seems that this have had little effect on the results but should anyway be mentioned since fretting is dependent on surface finish.

Due to limited time and limitations in the experimental setups, all experiments are conducted at a constant frequency of 10 Hz at room temperature. Perhaps a lower frequency and a higher temperature better simulates and correlates with the real conditions in a large bore diesel engine, and might produce slightly different results. The limited time for this work has made it impossible to investigate the dependence of frequency and temperature, and these topics are suggested to be further investigated in future works.

It is, however, very important to keep in mind that the relative slip in the experimental setups presented in this paper are based on qualified guesses based on investigations of the fretting scars. There is no exact telling of the slip amplitude since any elastic slip is undetectable, hence impossible to assign to one specific contact zone. The number of cycles is also an unknown factor when post-investigating the scar in the electron microscope, and the scars cannot be pinned down to a particular number of cycles.

10 Conclusions

It has been found that the dominating surface degradation for the conditions in the contact between the bearing and the seat is fretting wear. To prevent fretting wear the relative displacement in the contact zone should be elastically absorbed and the characteristics should be within the stick regime. Experiments performed under conditions simulating the actual contact suggests that the relative slip should be kept below 10 µm to maintain the stick regime and, thereby, low fretting wear.

Furthermore it is found that the friction undertakes a severe increase after only a couple of hundred cycles. Slip amplitudes concerning a relative displacement of 200 µm will increase the coefficient of friction from 0.13 to the stunning value of 0.8. A relative displacement of only a couple of µm will change the friction radically. The friction does not, however, show any noticeable response to the contact pressure within the interval 5 MPa < \( \frac{P}{A} \) < 10 MPa.

The contact pressure can affect the contact in various ways. At relative displacements around 15 µm an increase in contact pressure can force down the damage caused by fretting wear through changing of fretting regime to stick. At larger displacements amplitudes, where gross slip is the only option, an increase in contact pressure can, in fact, increase the fretted area by plastically flatten out peaks in the topography and thereby force the contact zone to grow.

It is absolutely possible to use FE calculations to predict location and size of fretting damage. The material models need to be very accurate though, as the cyclic loads often lead to non linear hardening of the material. Even more important, perhaps, is the description of interaction in the contact zone. The friction shows a very strong dependency on slip amplitude, which needs to be accounted for.
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27. In-house software, MAN B&W., COUVIP
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B Matlab script Input file

clear all
global ELEMENT NODE MPC object TOLER
TOLER=1e-4;

a=5;
b=20;
c=4;
d=5;
e=40;
f=1;

h=e/2+10;
c3=3;
c2=e/2+c3;
c1=e/2-d-c3;
b3=2;
b1=a/2+c;
b2=a/2+b;

p=[0 0            %1
   0 c1            %2
   a/2 c1          %3
   a/2 0           %4
   0 c2            %5
   a/2 c2          %6
   0 h             %7
   a/2 h           %8
   a/2 c1+c3       %9
   a/2 c2-c3       %10
   a/2+b3 c1+c3    %11
   a/2+b3 c2-c3%12
   b1 c1+c3        %13
   b1 c2-c3        %14
   b2 c2-c3        %15
   b2 c1+c3        %16
   b2 0            %17
   b1 0            %18
];

% p=p.*10^-3;
% yta 1 - ytan nederst
k=0;
if 1
  k=k+1;
  initobject
  object(k).side(1).num=15;
  object(k).side(1).type='line';
  object(k).side(1).point=[p(1,:); p(4,:)];
object(k).side(2).type='line';
object(k).side(2).point=[p(4,:); p(3,:)];
object(k).side(2).num=9;
object(k).side(3).type='line';
object(k).side(3).point=[p(3,:); p(2,:)];
object(k).side(4).type='line';
object(k).side(4).point=[p(2,:); p(1,:)];

% yta 2 - mitten
k=k+1;
initobject
object(k).side(1).type='line';
object(k).side(1).point=[p(2,:); p(3,:)];
object(k).side(1).num=object(1).side(1).num;
object(k).side(2).type='line';
object(k).side(2).point=[p(3,:); p(6,:)];
object(k).side(2).num=30;
object(k).side(3).type='line';
object(k).side(3).point=[p(6,:); p(5,:)];
object(k).side(4).type='line';
object(k).side(4).point=[p(5,:); p(2,:)];

% yta 3 - överst
k=k+1;
initobject
object(k).side(1).type='line';
object(k).side(1).point=[p(5,:); p(6,:)];
object(k).side(1).num=object(1).side(1).num;
object(k).side(2).type='line';
object(k).side(2).point=[p(6,:); p(8,:)];
object(k).side(2).num=4;
object(k).side(3).type='line';
object(k).side(3).point=[p(8,:); p(7,:)];
object(k).side(4).type='line';
object(k).side(4).point=[p(7,:); p(5,:)];
end

% Now, model the clamp
if 1
% yta 4 - kontakt
k=k+1;
initobject
object(k).side(1).type='line';
object(k).side(1).point=[p(9,:); p(11,:)];
object(k).side(1).num=10;
%object(k).side(1).scale=0.5;
object(k).side(2).type='line';
object(k).side(2).point=[p(11,:); p(12,:)];
object(k).side(2).num=10; %object(1).side(2).num;
for k=1:length(object)

    object(k).side(3).type='line';
    object(k).side(3).point=[p(12,:); p(10,:)];
    object(k).side(4).type='line';
    object(k).side(4).point=[p(10,:); p(9,:)];

    % yta 5 - benlångd
    k=k+1;
    initobject
    object(k).side(1).type='line';
    object(k).side(1).point=[p(11,:); p(13,:)];
    object(k).side(1).num=4;
    object(k).side(2).type='line';
    object(k).side(2).point=[p(13,:); p(14,:)];
    object(k).side(2).num=object(4).side(2).num;
    object(k).side(3).type='line';
    object(k).side(3).point=[p(14,:); p(12,:)];
    object(k).side(4).type='line';
    object(k).side(4).point=[p(12,:); p(11,:)];

    % yta 6 - hörn
    k=k+1;
    initobject
    object(k).side(1).type='line';
    object(k).side(1).point=[p(13,:); p(16,:)];
    object(k).side(1).num=4;
    object(k).side(2).type='line';
    object(k).side(2).point=[p(16,:); p(15,:)];
    object(k).side(2).num=object(4).side(2).num;
    object(k).side(3).type='line';
    object(k).side(3).point=[p(15,:); p(14,:)];
    object(k).side(4).type='line';
    object(k).side(4).point=[p(14,:); p(13,:)];

    % yta 7 - benhöjd
    k=k+1;
    initobject
    object(k).side(1).type='line';
    object(k).side(1).point=[p(18,:); p(17,:)];
    object(k).side(1).num=4;
    object(k).side(2).type='line';
    object(k).side(2).point=[p(17,:); p(16,:)];
    object(k).side(2).num=object(1).side(2).num;
    object(k).side(3).type='line';
    object(k).side(3).point=[p(16,:); p(13,:)];
    object(k).side(4).type='line';
    object(k).side(4).point=[p(13,:); p(18,:)];
end
object(k).elset='[surf_,num2str(k)];
object(k).material='steel';
object(k).eltype='CPS4';
object(k).thickness=f;
object(k).creation='defined by points'; % Is not really used except for copies.
end

MeshEngine3D_ver_1

% Extract nodes in specimen and clamp as they can not be equivalenced together
spec_nodes=[object(1:3).node];
clamp_nodes=[object(4:7).node];
equivalence(clamp_nodes,spec_nodes)
equivalence(spec_nodes,clamp_nodes)
figure(1)
plotmesh('off','off',1e-8)

% Create node and element sets for later use, e.g. contact surfs
% sets....
sets(1).type='element';
sets(1).list=[1:length(ELEMENT)];
sets(1).name='AllE';

sets(2).type='element';
sets(2).list=[object(4:7).element];
sets(2).name='ClampE';

sets(3).type='element';
sets(3).list=[object(1:3).element];
sets(3).name='SpecimenE';

sets(4).type='element';
sets(4).list=findelement((p(3,1)-(p(3,1)-p(2,1))/object(2).side(1).num/2)*0.999, (p(3,1)-(p(3,1)-p(2,1))/object(2).side(1).num/2)*1.001,p(3,2),p(6,2));% Find the element at the contact surf
sets(4).name='spec_contactE';

sets(5).type='element';
sets(5).list=findelement(p(7,1),p(8,1),(p(7,2)-(p(7,2)-p(5,2))/object(3).side(2).num/2)*0.999, (p(7,2)-(p(7,2)-p(5,2))/object(3).side(2).num/2)*1.001);% Find the element at the contact surf
sets(5).name='TopE';

sets(6).type='element';
sets(6).list=findelement((p(9,1)+(p(11,1)-p(9,1))/object(4).side(1).num/2)*0.999, (p(9,1)+(p(11,1)-p(9,1))/object(4).side(1).num/2)*1.001,p(9,2),p(10,2));% Find the element at the contact surf
sets(6).name='clamp_contactE';
sets(7).type='node';
sets(7).list=findnode(0,0,0,h)';
sets(7).name='XzeroN';

sets(8).type='node';
sets(8).list=findnode(0,100,0,0)';
sets(8).name='YzeroN';

sets(9).type='node';
tmp=findnode(a/2*0.9999,a/2*1.0001,0,h)'; % Find the nodes on the plane of the contact surf
sets(9).list=intersect(tmp,object(2).node);
sets(9).name='spec_contactN';

sets(10).type='node';
sets(10).list=intersect(tmp,object(4).node);
sets(10).name='clamp_contactN';

sets(11).type='node';
sets(11).list=findnode(p(7,1),p(8,1),p(7,2)*0.99,p(8,2)*1.01)';
sets(11).name='topN';

sets(12).type='node';
sets(12).list=findnode(p(17,1)*0.99,p(17,1)*1.01,p(16,2),p(15,2)*0.999)';
sets(12).name='clamp_pressN';

sets(13).type='element';
sets(13).list=findelement((p(17,1)-(p(17,1)-p(18,1))/object(7).side(1).num/2)*0.999,
(p(17,1)+(p(17,1)-p(18,1))/object(7).side(1).num/2)*1.001,(p(17,2),p(16,2)));% Find the
element at the clamp force surf
sets(13).name='clamp_pressE';

% Create abaqus .INP file
mesha3D(sets)
C Abaqus input file setup A

*Heading
** Job name: plant2D_5_abq Model name: plant2D_5
*Preprint, echo=NO, model=NO, history=NO, contact=NO
**
** PARTS
**
*Part, name=PART-1
*Node
...
*Element, type=CPE4
...
*Element, type=CPE3
...
** Region: (Section-1-CLAMP:CLAMP)
** Section: Section-1-CLAMP
*Solid Section, elset=CLAMP, material=BEARING_STEEL
10.,
** Region: (Section-2-SPEC:SPEC)
** Section: Section-2-SPEC
*Solid Section, elset=SPEC, material=SEAT_STEEL
8.,
*End Part
**
**
** ASSEMBLY
**
*Assembly, name=Assembly
**
*Instance, name=PART-1-1, part=PART-1
*End Instance
**
*Nset, nset=_M5, internal, instance=PART-1-1
  10975, ..., 43820
*Nset, nset=_M6, internal, instance=PART-1-1
  20144, ..., 54096
*Nset, nset=_M7, internal, instance=PART-1-1
  11044,
*Nset, nset=_M8, internal, instance=PART-1-1
  11045, 11046, 11047, 11048
*Nset, nset=_M9, internal, instance=PART-1-1, generate
  11130, 11133, 1
*Nset, nset=contactstress, internal, instance=PART-1-1, generate
  20229, 20679, 1
*Nset, nset=contactslip, internal, instance=PART-1-1, generate
  042, 410, 1
*Elset, elset=_PAD_S3, internal, instance=PART-1-1
  152031, ..., 156276
*Elset, elset=_PAD_S1, internal, instance=PART-1-1
  151935, ..., 156126
*Elset, elset=_PAD_S4, internal, instance=PART-1-1
  147949, ..., 151911
*Elset, elset=_PAD_S2, internal, instance=PART-1-1
  149778, 149798, 149817, 150905, 150927
*Surface, type=ELEMENT, name=PAD
  _PAD_S3, S3
  _PAD_S1, S1
  _PAD_S4, S4
  _PAD_S2, S2
*Elset, elset=_SPEC_S2, internal, instance=PART-1-1

112882, ..., 130775
*Surface, type=ELEMENT, name=SPEC
   SPEC_S2, S2
*Elset, elset=__M10_S3, internal, instance=PART-1-1
   147491, 147492, 147493, 147730, 147731, 147732
*Surface, type=ELEMENT, name=__M10, internal
   __M10_S3, S3
*Elset, elset=__M11_S1, internal, instance=PART-1-1
   147438, 147439, 147442, 147454, 147455, 147459, 147505,
   147506, 147509, 147510, 147660, 147661, 147664, 147665
   147676, 147677, 147680, 147681, 147716, 147717, 147720, 147721
*Surface, type=ELEMENT, name=__M11, internal
   __M11_S1, S1
*End Assembly
**
*Amplitude, name=CYCLIC, definition=PERIODIC
   1, 6.2832, 0., 0.55
-0.5, 0.
**
** MATERIALS
**
*Material, name=SEAT_STEEL
*Elastic
   184000., 0.3
*Plastic, Hardening=combined, Data type=stabilized
   60,0,0.006
   120,0.001,0.006
   180,0.004,0.006
**
   50,0,0.009
   160,0.0025,0.009
   210,0.007,0.009
**
   50,0,0.014
   150,0.0025,0.014
   220,0.0115,0.014
**
*cyclic hardening
   260,0
   280,0.0125
**
*Material, name=bearing_STEEL
*Elastic
   187000., 0.3
*Plastic, Hardening=Combined, DATA TYPE=STABILIZED
   110,0
   130,0.0002648,0.004
   155,0.0004561,0.004
   195,0.0008172,0.004
   230,0.0013701,0.004
   235,0.0014533,0.004
**
   100,0
   130,0.0004048,0.006
   140,0.0005013,0.006
   180,0.0010574,0.006
   212,0.0017663,0.006
   220,0.0020235,0.006
   244,0.0032951,0.006
**
   100,0,0.008
**Cyclic Hardening**
290, 0
250, 0.0005
350, 0.006

** INTERACTION PROPERTIES
**
*Surface Interaction, name=FRETTINGINTER
8.,
*Friction, lagrange
0.7,

** INTERACTIONS
**
** Interaction: FRETTINGINTER
*Contact Pair, interaction=FRETTINGINTER
PAD, SPEC

** STEP: Step-1
**
*Step, name=Step-1, nlgeom=YES
*Static
1., 1., 1e-08, 1.

** BOUNDARY CONDITIONS
**
** Name: Disp-BC-1 Type: Displacement/Rotation
*Boundary
_M5, 2, 2
** Name: Disp-BC-2 Type: Displacement/Rotation
*Boundary
_M6, 1, 1

** LOADS
**
** Name: CFORCE-1 Type: Concentrated force
*Cload
_M7, 1, -31.1
** Name: CFORCE-2 Type: Concentrated force
*Cload
_M8, 1, -62.2
** Name: CFORCE-3 Type: Concentrated force
**Cload
_M9, 1, -70
**
[5 MPa = 31,1N/node]
[10 MPa = 62,2N/node]

** OUTPUT REQUESTS
**
*Restart, write, frequency=0

** FIELD OUTPUT: F-Output-1
**
*Output, field, variable=PRESELECT
**
** HISTORY OUTPUT: H-Output-1
**
*Output, history, variable=PRESELECT
*End Step
**  -------------------------------------------------------------------------------------------------
**
** STEP: Step-2
**
*Step, name=Step-2, inc=1000, n1geom=YES
*Static
  .01, 1., 1e-09, 0.02
**
** LOADS
**
** Name: DISTFORCE-1 Type: Pressure
*Dsload, amplitude=cyclic
_M10, P, -200.
** Name: DISTFORCE-2 Type: Pressure
*Dsload, amplitude=cyclic
_M11, P, -200.
**
** OUTPUT REQUESTS
**
*Restart, write, frequency=0
**
** FIELD OUTPUT: F-Output-1
**
*Output, field, variable=PRESELECT
**
** HISTORY OUTPUT: H-Output-1
**
*Output, history, variable=PRESELECT
*End Step
**D Abaqus input file setup B**

*Heading
** Job name: plant2D_6_abq Model name: plant2D_6
*Preprint, echo=NO, model=NO, history=NO, contact=NO
**
** PARTS
**
*Part, name=PART-1
*Node
...
*Element, type=CPE4
...
*Element, type=CPE3
...
** Region: (Section-1-CLAMP:CLAMP)
** Section: Section-1-CLAMP
*Solid Section, elset=CLAMP, material=BEARING_STEEL
10.,
** Region: (Section-2-SPEC:SPEC)
** Section: Section-2-SPEC
*Solid Section, elset=SPEC, material=SEAT_STEEL
8.,
*End Part
**
**
** ASSEMBLY
**
*Assembly, name=Assembly
**
*Instance, name=PART-1-1, part=PART-1
*End Instance
**
*Nset, nset=displacement, internal, instance=PART-1-1
43790, ..., 43820
*Nset, nset=displacement2, internal, instance=PART-1-1, generate
54012, 54042, 1
*Nset, nset=_M5, internal, instance=PART-1-1
10975, ..., 11044
*Nset, nset=_M6, internal, instance=PART-1-1
20144, ..., 54096
*Nset, nset=_M7, internal, instance=PART-1-1
11044,
*Nset, nset=_M8, internal, instance=PART-1-1
11045, 11046, 11047, 11048
*Nset, nset=_M9, internal, instance=PART-1-1, generate
11130, 11133, 1
*Nset, nset=contactstress, internal, instance=PART-1-1, generate
20229, 20679, 1
*Nset, nset=contactslip, internal, instance=PART-1-1, generate
042, 410, 1
*Elset, elset=_PAD_S3, internal, instance=PART-1-1
152031, ..., 156276
*Elset, elset=_PAD_S1, internal, instance=PART-1-1
151935, ..., 156126
*Elset, elset=_PAD_S4, internal, instance=PART-1-1
147949, ..., 151911
*Elset, elset=_PAD_S2, internal, instance=PART-1-1
149778, 149798, 149817, 150905, 150927
*Surface, type=ELEMENT, name=PAD
_PPAD_S3, S3
*Elset, elset=_SPEC_S2, internal, instance=PART-1-1
112882, ... , 130775
*Surface, type=ELEMENT, name=SPEC
_SPEC_S2, S2
*Elset, elset=_M10_S3, internal, instance=PART-1-1
147491, 147492, 147493, 147730, 147731, 147732
*Surface, type=ELEMENT, name=_M10, internal
__M10_S3, S3
*Elset, elset=_M11_S1, internal, instance=PART-1-1
147438, ... , 147721
*Surface, type=ELEMENT, name=_M11, internal
__M11_S1, S1
*End Assembly
**
*Amplitude, name=CYCLIC, definition=PERIODIC
1, 6.2832, 0., 0.0
-0.0, 1.
**
** MATERIALS
**
*Material, name=SEAT_STEEL
*Elastic
184000., 0.3
*Plastic, Hardening=combined, Data type=stabilized
60, 0.006
120, 0.001, 0.006
180, 0.004, 0.006
**
50, 0.009
160, 0.0025, 0.009
210, 0.007, 0.009
**
50, 0.014
150, 0.0025, 0.014
220, 0.0115, 0.014
**
cyclic hardening
260, 0
280, 0.0125
**
*Material, name=bearing_STEEL
*Elastic
187000., 0.3
*Plastic, Hardening=Combined, DATA TYPE=STABILIZED
110, 0.004
130, 0.0002648, 0.004
155, 0.0004561, 0.004
195, 0.0008172, 0.004
230, 0.0013701, 0.004
235, 0.0014533, 0.004
**
100, 0.006
130, 0.0004048, 0.006
140, 0.0005013, 0.006
180, 0.0010574, 0.006
212, 0.0017663, 0.006
220, 0.0020235, 0.006
244, 0.0032951, 0.006
**
100, 0, 0.008
158, 0.000755, 0.008
226, 0.00239, 0.008
258, 0.00423, 0.008
260, 0.00522, 0.008
**
*Cyclic Hardening
290, 0
250, 0.0005
350, 0.006
**
** INTERACTION PROPERTIES
**
*Surface Interaction, name=FRETTINGINTER
8.,
*Friction, lagrange
0.7,
**
** INTERACTIONS
**
** Interaction: FRETTINGINTER
*Contact Pair, interaction=FRETTINGINTER
PAD, SPEC
**
** STEP: Step-1
**
*Step, name=Step-1, nlgeom=YES
*Static
1., 1., 1e-08, 1.
**
** BOUNDARY CONDITIONS
**
** Name: Disp-BC-1 Type: Displacement/Rotation
*Boundary
_M5, 2, 2
** Name: Disp-BC-2 Type: Displacement/Rotation
*Boundary
_M6, 1, 1
**
** LOADS
**
** Name: CFORCE-1 Type: Concentrated force
*Cload
_M7, 1, -31.1
** Name: CFORCE-2 Type: Concentrated force
*Cload
_M8, 1, -62.2
** Name: CFORCE-3 Type: Concentrated force
**Cload
_M9, 1, -70
**
** [5 MPa = 31.1N/node]
** [10 MPa = 62.2N/node]
**
** OUTPUT REQUESTS
**
*Restart, write, frequency=0
**
** FIELD OUTPUT: F-Output-1
**
*Output, field, variable=PRESELECT
**
** HISTORY OUTPUT: H-Output-1
**
*Output, history, variable=PRESELECT
*End Step
**
-----------------------------------------------
**
** STEP: Step-2
**
*Step, name=Step-2, inc=1000, nlgeom=YES
 *Static
  .01, 1., 1e-09, 0.02
**
** LOADS
**
** Name: DISTFORCE-1  Type: Pressure
 *Boundary, amplitude=cyclic
  displacement2, 2, 2, 0.001
**
** OUTPUT REQUESTS
**
*Restart, write, frequency=0
**
** FIELD OUTPUT: F-Output-1
**
*Output, field, variable=PRESELECT
**
** HISTORY OUTPUT: H-Output-1
**
*Output, history, variable=PRESELECT
*End Step
E Matlab file SSR calculation

clear all
close all

% plottar SSR mot noder samt kritiska planet för varje nod
% corresponds only to results from plant2D_5_abq.inp including [Sxx, Sxy, Syy]

[filn,pt]=uigetfile('*.*','Load');
sigma=load(strcat(pt,filn));
time=sigma(:,1)-sigma(1,1);
sigma=sigma(:,2:length(sigma(1,:)));

[M N] = size(sigma);
Nnod=N/3;               %antal noder i kontakten som skall undersökas
m=M;                    %tidssteg
teta=90;                %halva vinkeln som skall undersökas
step=0.1;               %steglängden med vilken vinkeln undersöks

xaxis=1:(N)/3;
xaxis=xaxis-50;
xaxis=xaxis./350;

tau_temp=[];
deltatau=zeros(1,Nnod);
fi_crit=zeros(1,Nnod);
tau=[];

for j=1:Nnod
    for i=-teta:step:teta
        tau_temp= -( sigma(:,j)-sigma(:,j+2*Nnod)/2*sin(2*pi*i/180)+sigma(:,j+Nnod).*cos(2*pi*i/180);  %-(sigmaxx-sigmayy)/2*sin(2*i*pi/180)+sigmaxy*cos(2*i*pi/180);
        deltatau_temp=max(tau_temp)-min(tau_temp);
        if deltatau_temp>deltatau(j)
            deltatau(j)=deltatau_temp;
            fi_crit(j)=i;
            tau(:,j)=tau_temp;
        end
    end
end

plot (xaxis,deltatau,'+-')
figure
plot(xaxis,fi_crit,'*')