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Crack Propagation in Cruciform Welded Joints

Study of Modern Analysis

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Abstract

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This thesis is investigating how the effective notch method can be used for fatigue assessment of welded joints. The effective notch method is based on a finite element analysis where the joint is modeled with all notches fictitiously rounded with a radius of 1 mm. Analyses are performed on a cruciform fillet welded joint where parameters such as, load case, steel plate thickness and weld size, are varied. The achieved lifetime estimations are then compared to calculations with other fatigue assessment methods, linear elastic fracture mechanics and the nominal method. The goal is to draw conclusions about pros and cons of the effective notch method. The results are also compared to experimental fatigue tests performed on the same geometry.

The results indicates that the effective notch method tends overestimating the lifetime, especially when the steel plate thickness is small. This leads to a non conservative method that is dangerous to use as guidance when designing. The estimations are though better when considering a toe crack then when considering a root crack.

Due to a large scatter in experimental test results, it is hard to validate a fatigue assessment method in an absolute sense. More results from experimental fatigue tests are needed before the effective notch method can be fully used. For relative analysis, when variations of the same design needs to be compared, the effective notch can be a very powerful tool. This is because of the flexibility for different geometries that this method grants.

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Sammanfattning

Detta examensarbete undersöker hur ”effective notch” metoden kan användas för utmattningsdimensionering av svetsade förband. Effective notch metoden baserar sig på en finita element modell där alla skarpa anvisningar på förbandet är avrundade med en fiktiv radie på 1 mm. Analyser är utförda på ett korsformat förband med kälfogar där parametrar som t ex lastfall, plåttjocklek och svetstjocklek är varierade. Analyserna resulterar i en livstidsuppskattning som sedan är jämförd med beräkningar med andra metoder för utmattningsdimensionering, brottmekanik och nominella metoden. Målet är att dra slutsatser om för- och nackdelar med effective notch metoden. Resultaten är också jämförda med experimentella utmattningsprover genomförda på samma geometri.

Resultaten indikerar att effective notch metoden tenderar att överskatta livslängden för förbanden, speciellt när plåttjockleken är tunn. Detta leder till en icke konservativ metod som är farlig att använda som vägledning vid design. Livstidsuppskattnignarna är dock bättre om sprickan antas starta i svetstån än om den skulle starta i svetsroten.

På grund av stor spridning i experimentella resultat så är det mycket svårt att validera en utmattningsdimensioneringsmetod i en absolut mening. Det är också fallet för effective notch metoden, och därför behövs mycket mer resultat från experimentella utmattningstester innan effetive notch metoden kan användas fullt ut. För relativa analyser, när variationer av en och samma design ska utvärderas mot varandra, kan dock effective notch metoden vara ett mycket kraftfullt verktyg. Detta på grund av den flexibilitet för olika geometrier som denna metod medger.

Contents

1 Introduction

What do you do if you want to cut off a steel wire, but you do not have any pliers? You could bend the wire back and forth. A small crack will form, and after a certain numbers of bending the wire will break. You have accomplished a low-cycle fatigue failure. A fatigue failure is a failure that can arise in a structure, loaded way below the static rupture limit, if the load is cyclic and are applied for enough long time. Microcracks are formed due to high stress concentrations at small material or geometrical defects in a structure. These microcracks grow little by little when the load is varied in time, eventually they will reach a critical size. At this critical size, the remaining material is not enough, even for a load as small as this, and a failure will occur. [3]

At Atlas Copco, vehicles for construction and mining are built, see Figure 1. The vehicles are exposed to fluctuating and cyclic loads when they are operating. The vehicles are joined together with welded sheets, and the weld becomes a limitation in quality and lifetime for the vehicle due to fatigue failures. At the same time, an over dimensioning of the weld joints are costly since they are a big part of the expense in the construction. Today, a method of nominal stresses is used for dimensioning against fatigue failure at the division of Applied Mechanics at Atlas Copco.

Figure 1 *Examples of Atlas Copco products, a three-boom drilling rig and an underground loader.*

The method of nominal stresses is a well-tried and quite old method. Different kinds of joints are categorized into classes and norms. Standards are recommending how to dimension a certain joint. As the technical resources are increasing more advanced computer calculations are available and finite element analyses has become an important tool for fatigue assessment. With the finite element analyses the calculations can be made more accurate and perhaps the nominal method has unnecessary large safety margins. Hopefully these analyses can give guidance to a dimensioning much closer to the limit, but at the same time still conserve the safety of the construction.

The scope of this thesis is to investigate some of these finite element analysis methods. The main topic will be the *Effective Notch Method* and to compare results from this method with the *Linear Elastic Fracture Mechanics* (LEFM) method and with the nominal method. In the effective notch method all sharp edges that gives singularities, and therefore problems in the finite element analysis, are replaced by a fictitious radius. In LEFM a small crack is initiated and then propagated through the material.

1.1 Problem description

In this thesis, cruciform welded joints are modelled with the effective notch method, see Figure 2. For different load cases it is studied how the effective notch method estimates the lifetime when parameters are varied, for example the thickness of the material or the weld size. The same joint is also modelled with LEFM and the results are compared for the different methods, and with the nominal method.

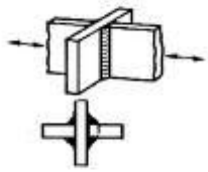


Figure 2 *Cruciform joint*

Some real fatigue tests are also performed on cruciform joints, both when the weld is load-carrying and when the weld is not load-carrying. The results from these tests are used to validate the calculated results. The distribution of the results are often very large for fatigue tests, therefore also test results from the literature are used to get more data. Results are then evaluated from a statistical point of view.

1.2 Goals and purpose

The goal is to compare how different calculation methods estimates the lifetime in a fatigue loaded structure. The estimation of the lifetime is compared for the effective notch method LEFM, the nominal method and real performed fatigue tests. Finally, conclusions about pros and cons of the effective notch method are drawn. What limitations are in the method? And are there certain geometries or cases where the method works better then for others?

1.3 Method

The thesis is done at the division of Applied Mechanics at Atlas Copco Rock Drills AB in Örebro. The cruciform joint is modelled in the finite element program Ansys 12.0 for different dimensions and load cases. Lifetime estimations are performed with the different methods, effective notch, LEFM and nominal method. Since the methods based on FE analysis, are methods still under development, a massive literature study is performed to gather information and recommendations on how the modelling are to be done.

An experimental fatigue test is also performed at the division of Applied Mechanics. For different dimensions on the weld throat, and for different load cases, a cruciform joint is applied to a cyclic load until failure occurs. The results from the calculations are compared with the test results. The result has a rather large statistical deviation and this is also taken under consideration.

For the nominal method two different standards are used. One is the BSK 99 standard which is a Swedish standard and also the standard that Atlas Copco uses. The other one is the standard of the International Instituite of Welding (IIW).

1.4 Reader's Guide

In chapter 1 and 2 background theory about the phenomenon of fatigue and the different approaches for fatigue assessment can be found. In chapter 4 it is described how the modelling is made, assumptions and boundary conditions for the different methods. An example of a non load-carrying cruciform joint is used through the whole chapter, and it is investigated how the steel plate thickness affects the lifetime estimations. In chapter 5 the effect of the weld throat thickness is analysed. For what weld sizes will the crack originate in the weld root and when will it start at the weld toe? In chapter 6 experimental fatigue tests are performed and in chapter 7 and 8 the results from these tests are analysed.

All concepts, notations and recommendations used, are in consistence with IIW [5] unless nothing else is mentioned.

1.4.1 Nomenclature

σ_r - stress range ($\sigma_{max} - \sigma_{min}$)
σ_{mean} - mean value of the stress
σ_a - stress amplitude
R - stress ratio ($\sigma_{min}/\sigma_{max}$)
N - number of load cycles
φ_Q - factor defining the failure probability
σ_{nom} - nominal stress
m - constant defining the slope of the S-N curve
K_t - stress concentration factor ($\sigma_{max}/\sigma_{nom}$)
ρ - radius of the notch
K_I - stress intensity factor (I = mode I)
ΔK_I - stress intensity factor range ($K_{I,max}$-$K_{I,min}$)
ΔK_{th} - threshold value. Below this value the crack won't grow.
ΔK_{IC} - critical value. Above this value immediate failure occurs.
a - crack length
C - material constant in Paris' law
n - material constant in Paris' law, coupled to m
K_{eff} - effective stress intensity factor
t - steel plate thickness
θ - the weld angle
l - weld leg length
s - weld throat thickness
E - coefficient of elasticity
v - Poisson's ratio
φ_{dim} - thickness factor
$s_{critical}$ - critical value of the weld throat thickness where the crack moves from the root to the toe
L - width of the test specimen

2 The phenomenon of fatigue

Materials that are exposed to a load, varying in time, can fracture even though the magnitude of the load is below the static yield limit of the material. This kind of fracture is called fatigue failure and is one of the most common causes for breakdown in steel structures. It is known that 80-90% of all breakdowns in machines and constructions are due to fatigue failure. [2]

The fatigue process usually divides into three phases:

- *The crack initiation*. The number of cycles it takes to form a microscopic crack from a material or geometrical defect.
- *The crack propagation*. The crack is growing with increasing number of load cycles.
- *Final breakage*. The crack has grown so much that the remaining material can not support the load and an immediate failure occurs.

In this thesis only welded joints are considered, and then the crack initiation phase is assumed to already be passed. A material that is affected by a weld always contains small cracks or slag inclusions that acts as notches for crack propagation. Therefore only the crack propagation phase and the final break remains for analysis. [1]

2.1 General concepts and loads

For determination of fatigue data, for a certain structure or joint, the specimen is usually loaded with a cyclic constant amplitude load. The number of load cycles required to cause fatigue failure for this particular stress range is then used as a measure of the fatigue life for the structure. The stress range, σ_r, is defined as the difference between the maximal stress, σ_{max}, and the minimal stress, σ_{min}, occurring in the material when the load is fluctuating.

$$\sigma_r = \sigma_{\max} - \sigma_{\min} \qquad (2:1)$$

The stress range affects the fatigue life of a structure, but also the mean stress, σ_{mean}, is of great importance.

$$\sigma_{mean} = \frac{1}{2}\left(\sigma_{\max} + \sigma_{\min}\right) \qquad (2:2)$$

To be able to describe the constant amplitude load, two more characteristic quantities are introduced:

stress amplitude: $$\sigma_a = \frac{1}{2}\left(\sigma_{\max} - \sigma_{\min}\right) \qquad (2:3)$$

stress ratio: $$R = \frac{\sigma_{\min}}{\sigma_{\max}} \qquad (2:4)$$

Figure 3 shows a fluctuating stress with constant amplitude and constant mean stress.

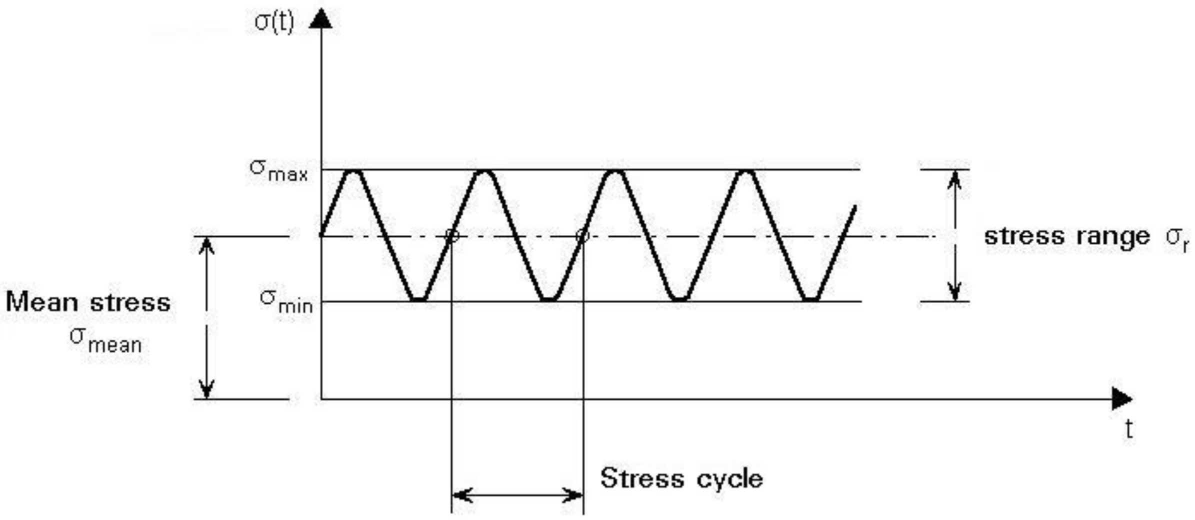


Figure 3 *Fluctuating stress with constant stress amplitude and constant mean stress.*

There are two special cases of the constant amplitude load, alternating load when $R = -1$, and pulsating load when $R = 0$. The two cases are schematically shown in **Figure 4**. [3]

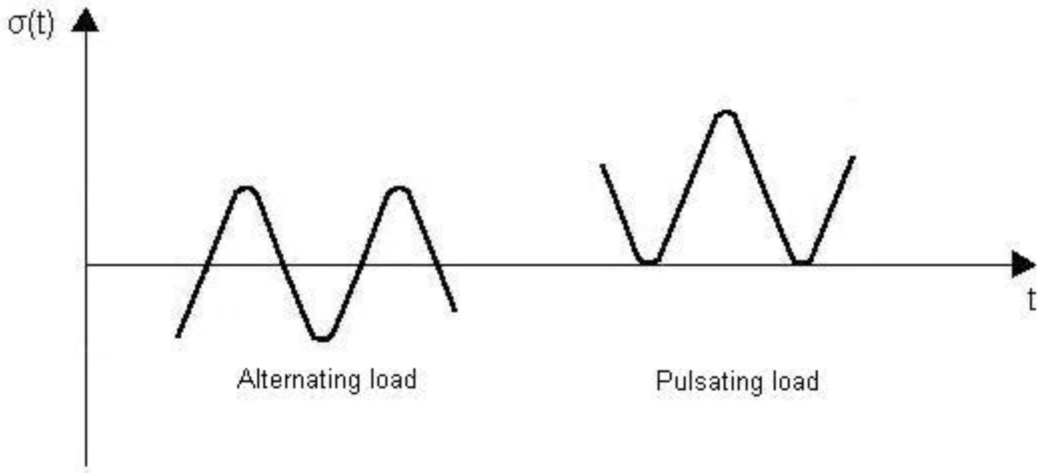


Figure 4 *Alternating and pulsating loads.*

An important note is that when tests are performed on specimens, a constant amplitude load is often applied since this requires less advanced equipment. On the other hand, when operating, the structures are often applied to loads with varying amplitude. In many cases the actual load function is unknown when the structure is designed. This is often the most uncertain parameter when doing a fatigue analysis for designing.

2.2 S-N curves and FAT- classes

The result from a big number of tests performed with different stress ranges can be concluded in an S-N curve (Wöhler curve), see **Figure 5**. The S-N curve is relating the stress range to the number of cycle to failure.

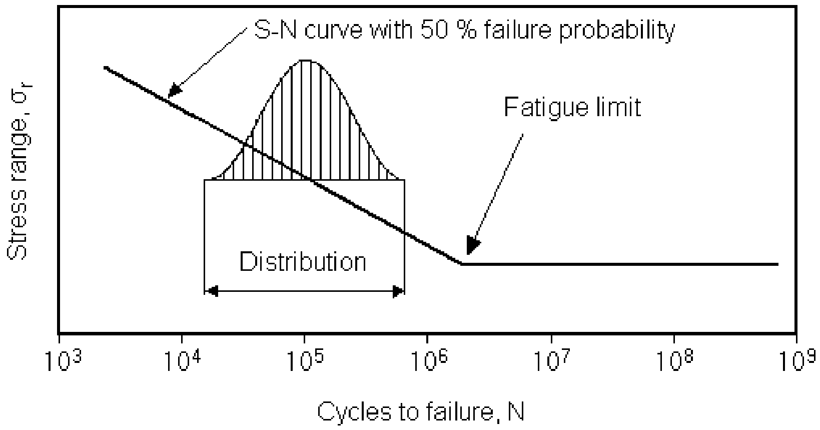


Figure 5 *Example of an S-N curve relating the stress range to the number of cycles to failure.*

The results from tests are usually obtained with a great dispersion. When many tests have been performed the S-N curve can be established for a certain failure probability. [1]

The S-N curve can be assumed to be a straight line in a log-log-plot, described by the equation below.

$$\sigma_r = C \cdot \left(\frac{N_0}{N} \right)^{1/m} \qquad (2:5)$$

where C and N_0 are constants representing stress range and cycles to failure at a certain point on the curve. m is for welded joints often approximately 3.

During the years, a lot of welded joints have been tested for fatigue failure. The results from these tests have been put together for different kinds of joints and many different classification systems and standards have been set up. International Institute of welding (IIW) uses different FAT-classes (fatigue classes) and the BSK 99 standard uses a C-value to classify the joints [1][5]. Both of these standards define the classes from the stress range in MPa at $N = 2 \cdot 10^6$ cycles, for 2.3 % probability of failure. This means that if a certain joint has a 97.7% chance to live for $2 \cdot 10^6$ cycles or more, applied to the constant amplitude load where the stress range is x MPa. Then this joint is in the class FAT = x for IIW, or C = x for BSK 99. Equation (2:5) can then for a certain FAT-class be written as

$$\sigma_r = FAT \cdot \left(\frac{2 \cdot 10^6}{N} \right)^{1/3} \qquad \textbf{(2:6)}$$

2.2.1 Statistical deviation

Under the assumption of normal distribution the stress range for a failure probability of 50 % has to be subtracted by two standard deviations to get the stress range for a failure probability of 2.3 %. Or the other way around, the stress range has to be added up by to standard deviations to go from 50 % failure to 2.3 % failure. A 50 % failure probability corresponds to the mean value and is the value that a real fatigue test is supposed to be compared to. One standard deviation in log N is ~0.17 [1]. If a S-N curve with 2.3 % failure probability are to be compared to test results the S-N curve should be moved $2 \cdot 0.17 = 0.34$ to the right. If the current S-N curve has a slope *m* = *3*, the permitted stress range must be lowered by a factor, $\varphi_Q = 10^{0.34/m} = 10^{0.34/3} = 1.3$.

In Table 1 the factor, φ_Q, is calculated for different probabilities of failure. Note that these values only are valid if the slope of the S-N curve is 1/3.

Table 1 The factor, φ_Q, for different failure probabilities for *m = 3. [1]*

Probability of failure	50 %	15.9 %	2.3 %	1 %	1.35 ‰	0.001	10^{-4}	10^{-5}
Number of standard deviations	0	1	2	2.33	3	3.10	3.72	4.27
φ_Q	1.30	1.14	1.00	0.95	0.88	0.86	0.80	0.74

3 *Different methods for fatigue assessment*

There are essentially four different approaches for calculating the fatigue life of a structure, the nominal stress method, the hot spot method, the effective notch method and linear elastic fracture mechanics (LEFM). The nominal stress method is the oldest and most well-tried method, the others are newer and usually performed by FE-calculations. The hot spot method will only be mentioned briefly, since it has shown not to be so useful. It has shown that the hot spot method is full of special cases, exceptions and limitations that make the method very problematic. Research is now instead focusing on the effective notch method. Hopefully the effective notch method is as simple as it is at first sight, but more work and investigations needs to be done. The focus in this thesis will be on the effective notch method.

Martinsson [8] has made a picture for a comparison of the different methods, see **Figure 6**. The nominal method is the simplest method of the four, but it is also the least accurate one, at least for complex structures. LEFM is much more accurate, but at the expense of a larger working effort. According to this picture the effective notch seems to be a promising method for fatigue assessment.

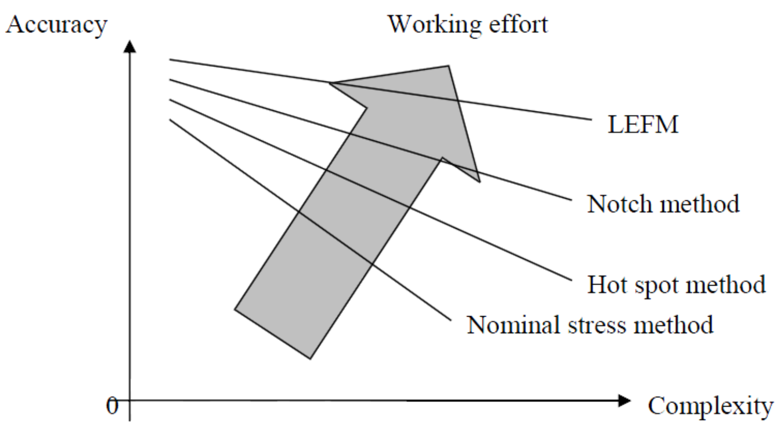


Figure 6 *There are essentially four methods for dimensioning against fatigue failure. The nominal stress method is the least complex and least time consuming one, but also the least accurate one. Linear elastic fracture mechanics is considered to be the most accurate one but at the expense of complexity and more working effort. [4]*

3.1 The Nominal Stress Method

Nominal stress, σ_{nom}, is defined as the stress in the sectional area under consideration, here it is the sectional area of the metal plate. The local stress raising effect of the welded joint is disregarded, but the stress raising effect of the macrogeometric shape of the component in the vicinity of the joint is included. A macrogeometric shape is a geometric detail that affects the stress distribution all over the sectional area, for example a large cut out, or a change of the sectional area.

In the simplest cases the nominal stress can be calculated by linear elastic theory according to equation (3:1).

$$\sigma_{nom} = \frac{F}{A} + \frac{M}{I} \cdot z \qquad \textbf{(3:1)}$$

where

F = normal force
A = cross- sectional area
M = bending moment
I = moment of inertia
z = distance from the centre of gravity of the section to the considered location.

Sometimes it can be hard to find the nominal stress using this equation. For example if the geometry is very complicated, or if a statically indetermination of the problem complicates it. Then a finite element analysis (FEA) of the structure can be used to determine the nominal stresses.

The classified structural details (FAT- classes) are defined by the nominal stress for which the specimen will hold for $2 \cdot 10^6$ cycles with a probability of 97.7 %. When a structure is designed by the nominal stress method the calculated nominal stress is compared to the fatigue resistant curve (S-N curve) for the right class. [1]

3.2 The Hot Spot Method

The hot spot is the critical point in a structure from where a crack can be assumed to propagate. In a welded structure this point is usually located by the weld toe or by the weld root. The hot spot stress is the value of the geometrical stress in the hot spot. The geometrical stress includes all stress raising effects except the non-linear stress peak caused by the local notch.

When evaluating by the hot spot method, just as for the nominal stress method, the maximal stress range is determined by different structural classes. Unlike for the nominal stress approach, the fatigue strength is here stated as a geometrical stress range in each class. Stress raising effects due to the local geometry of the joint is included in the structural class.

Therefore a unique structural class for each joint type is not needed and the number of different classes can be decreased.

The hot spot method will not be considered any deeper in this thesis. For further information see [1], [5] and [17].

3.3 The Effective Notch Method

The notch stress is the total stress at the root of a notch taking into account the stress concentration caused by the local notch, consisting of the sum of geometrical stress and non-linear stress peak. When using the effective notch method a linear-elastic material behaviour is assumed. Then sharp edges in the weld toe and weld root will cause discontinuities in the equations. Therefore the actual radius is replaced by an effective notch root radius. For structural steel, Hobbacher [5] recommends to use a radius of **ρ = 1 mm** which has been verified to give consistent results, see Figure 7. For fatigue assessment, the effective notch stress is compared with a single fatigue resistance curve. FAT = 225 is recommended by IIW [5] but this has later been questioned and suggested to be lowered for special cases of joints and geometries [6].

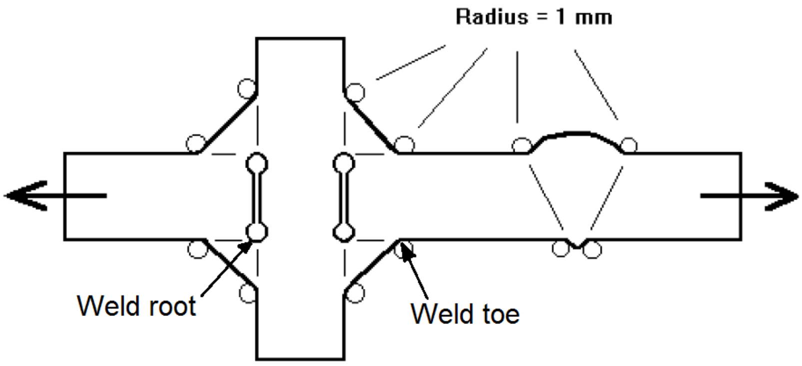


Figure 7 *In the effective notch method the notches at weld toe and weld root are fictitiously rounded. [5]*

When using the effective notch method one often refers to the stress concentration factor K_t. This is defined as the ratio of the maximal notch stress and the nominal stress.

$$K_t = \frac{\sigma_{\max}}{\sigma_{nom}} \qquad \textbf{(3:2)}$$

The theory about a fictitious radius has its origin from the microstructural support effect. [7] The actual radius is being increased by the material dependent microstructural length, ρ^*, multiplied by the support factor, s. The effective radius, ρ_f, to be used in the model is given by:

$$\rho_f = \rho + s\rho^* \qquad \textbf{(3:3)}$$

where ρ is the actual radius of the notch, see Figure 8. When doing a calculation of the stress concentration close to a notch you often assume that the material is elastic. In fact you will get a plasticizing occurs in the microstructural length. Due to this the actual stress is smaller then the one obtained with a complete elastic model. By increasing the radius of the notch this effect is simulated and the stress concentration is lowered.

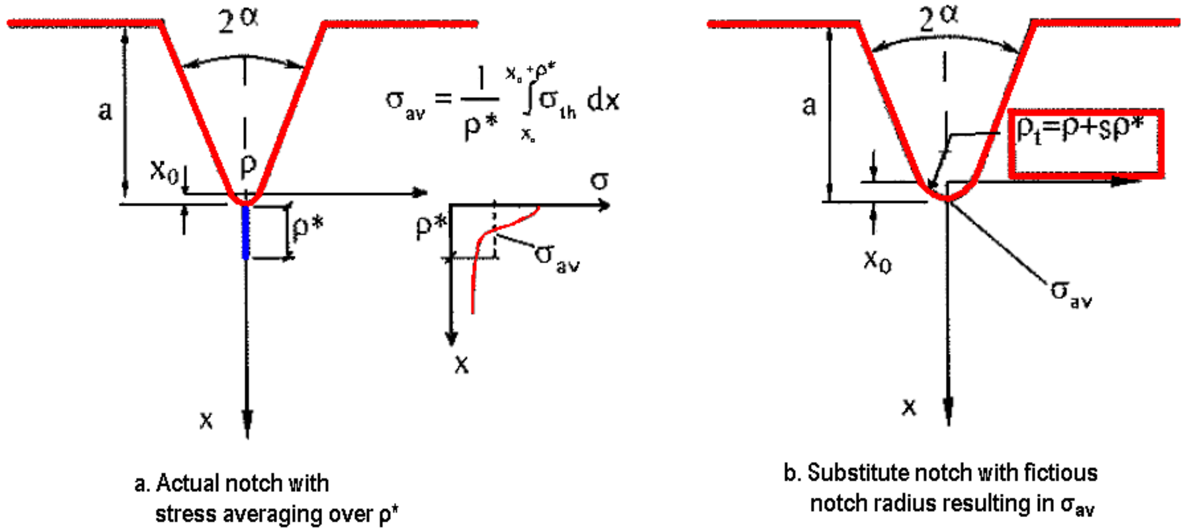


Figure 8 *The actual notch is substituted by a fictitious notch radius to obtain a stress concentration that includes the microstructural support effect. [9]*

Radaj [7] states that the conservative estimates, $\rho^* = 0.4\ mm$, $s = 2.5$ and $\rho = 0\ mm$ have proved to be realistic for welded joints in structural steel. This means, a fictitious effective radius of *1 mm*.

3.4 Linear Elastic Fracture Mechanics

Linear elastic fracture mechanics, LEFM, is used to predict the behaviour of a crack in structures subjected to loads. The method is investigating the conditions in the vicinity of a fictitious initiated crack. LEFM calculates the number of cycles to fracture, assuming linear elastic material. This imposes that the stresses at the crack tip tend to infinity. [1]

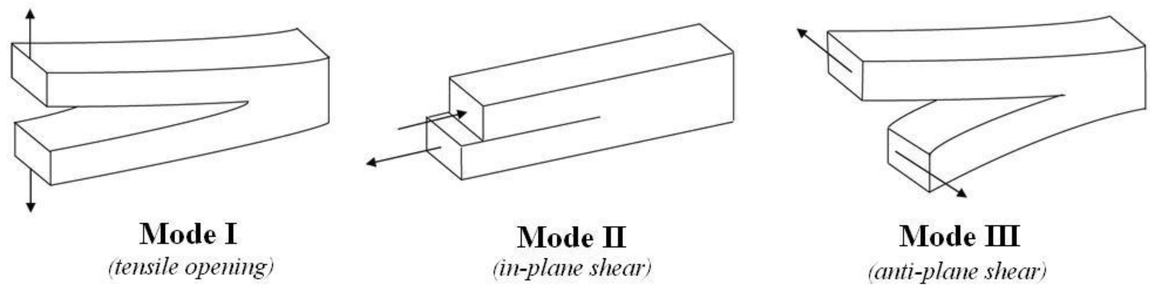


Figure 9 *A crack can be loaded in three different modes. Mode I is the most dangerous way to load a crack since this will cause the greatest stress intensity.*

There are three different load cases, three different modes, that can occur in the surrounding of a crack, see Figure 9. In Mode I the crack is open and this is therefore the mode where risk of failure is largest. This mode gives rise to the greatest stress intensity factor. The stress intensity factor, K_I (I = mode I), is defined as:

$$K_I = \sigma_{nom}\sqrt{\pi a} \cdot f \qquad \textbf{(3:4)}$$

where σ_{nom} is the nominal stress, a is the crack length and f is a dimensionless function depending on the geometry and load. The function f can be found in tables and handbooks for different load cases and geometry, but the stress intensity factor can also be calculated by FE.

When using the LEFM method for fatigue analysis, the range of the stress intensity factor is of interest. The range controls the crack propagation.

$$\Delta K_I = K_{I,\max} - K_{I,\min} \qquad \textbf{(3:5)}$$

If the load is negative, compressing the crack, the load does not have any influence of the number of cycles to failure. Therefore the stress intensity range is defined as an always positive quantity.

$$\Delta K_I = \begin{cases} K_{I,\max} - K_{I,\min} & \textit{for } K_{I,\min} \geq 0 \\ K_{I,\max} & \textit{for } K_{I,\min} \leq 0 \\ 0 & \textit{for } K_{I,\max} \leq 0 \end{cases} \qquad \textbf{(3:6)}$$

In a cyclic load case the time to failure is measured in the number of load cycles, N. From experimental studies the result can be represented in a log-log-diagram with the crack increment per load cycle, $\frac{da}{dN}$, as a function of stress intensity range, ΔK_I. In such a plot three different areas are distinguished, a threshold area, a middle area and an instable area, see Figure 10.

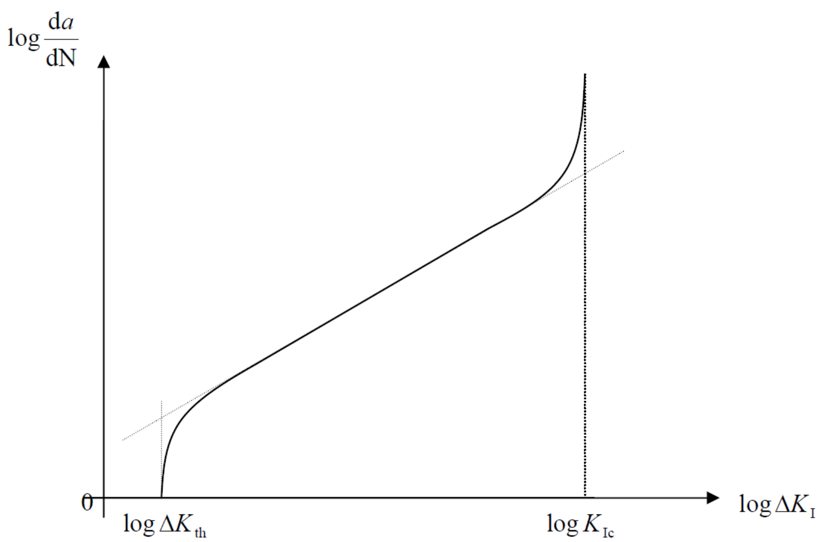


Figure 10 *The crack increment per load cycles is plotted logarithmically as a function of the stress intensity rate. Below the threshold value of the stress intensity rate,* ΔK_{th}*, the crack will not propagate. Above the critical value,* ΔK_{Ic}*, a fracture will occur momentarily.*

The threshold area is the area to the left of the asymptote $\Delta K_I = \Delta K_{th}$, where the stresses are to small for the crack to propagate. The instable area is the area to the right of the asymptote $\Delta K_I = \Delta K_{Ic}$. Here the crack growth is approaching infinity and a failure will occur momentarily. In the middle area the curve is linear in the log-log-diagram. An analytical expression for the relation can be written as

$$\frac{da}{dN} = C(\Delta K_I)^n \qquad \textbf{(3:7)}$$

This expression is called Paris' law. *C* and *n* are material constants. [3]

When calculating the lifetime of a structure, Paris' law has to be integrated. It is a separable differential equation, but since ΔK_I depends on the function *f*, that is hard to integrate analytically, a numerical integration method is normally used. [4]

If a crack is loaded in several modes at the same time, the stress intensity factor in Paris' law is replaced by an effective stress intensity factor, K_{eff}. There are numerous ways to calculate the effective stress intensity factor. The most common ones are presented below. [8]

$$\Delta K_{eff} = \left(\Delta K_I^4 + 8\Delta K_{II}^4\right)^{\frac{1}{4}} \quad \text{“Crack tip displacement”} \qquad \textbf{(3:8)}$$

$$\Delta K_{eff} = \left(\Delta K_I^2 + \Delta K_{II}^2\right)^{\frac{1}{2}} \quad \text{“Strain energy release”} \qquad \textbf{(3:9)}$$

$$\Delta K_{eff} = \left(\Delta K_I^2 + \Delta K_I \Delta K_{II} + \Delta K_{II}^2\right)^{\frac{1}{2}} \quad \text{”Cross product”} \qquad \textbf{(3:10)}$$

In this thesis, equation (3:9) is used.

4 Modelling in ANSYS 12.0

For the simulations and calculations done in this thesis, the FE program ANSYS 12.0 is used. Three different variations of the cruciform joint are analyzed, one where the weld is not load-carrying and two where the weld is load-carrying. For the non load-carrying joint, the crack assumes to start at the weld toe, but for the load-carrying joint the crack can originate either from the root or from the toe depending on the weld size and shape of the structure. In this section it is explained how the FE modelling, for the different approaches of fatigue strength determination, is made. The non load-carrying cruciform welded joint is used as example.

4.1 The Geometry

All finite element analyses are made in the program ANSYS 12.0. The models are made in 2D to minimize the computational time. Figure 11 shows a sketch of the geometry. On a steel plate two other steel plates, with the same thickness, are transversely attached by fillet welds. $\Delta\sigma_{nom,1}$ gives the non load-carrying joint, and $\Delta\sigma_{nom,2}$ gives the load-carrying joints.

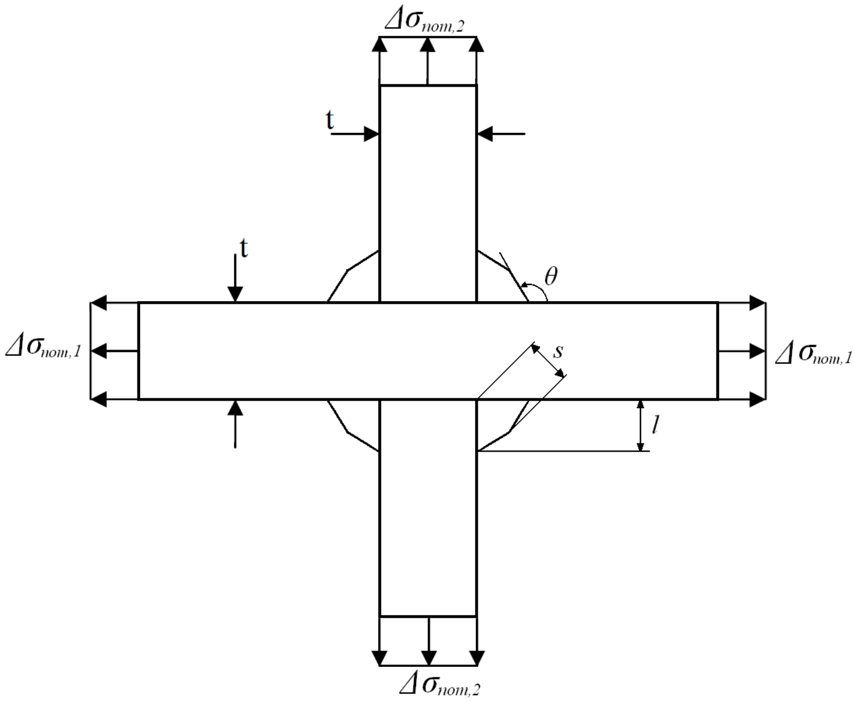


Figure 11 *The cruciform joint used for analysis. $\Delta\sigma_{nom,1}$ gives a non load-carrying joint and $\Delta\sigma_{nom,2}$ gives a load-carrying joint.*

The dimensions in Figure 11 are varied in the analyses, but in this section a non load-carrying joint with the dimensions given in Table 2 are used as an example. The material is plate steel and two material constants are needed for the calculations, the coefficient of elasticity, *E*, and Poisson's ratio, *v*. For steel, the values are approximately $2.1 \cdot 10^5$ *MPa* and *0.3* respectively. See Table 2, for a summary of the values of the variables.

Table 2 A summary of the variables used for the calculations of the non-load carrying cruciform joint used as example in this section.

Variable	Value
θ	*45 º*
σ_{nom}	*0- 120 MPa* (R=0)
s	*t/2*
t	*5 - 35 mm*
E	$2.1 \cdot 10^5$ *MPa*
v	*0.3*

To be able to build this model in Ansys one has to know how far away from the weld the material has to be modelled. How far away from the notch can the stress be assumed to be equal to the nominal stress? In Figure 12 the stress at the surface of the material is plotted against the distance from the weld toe. The steel plate thickness is, *t = 15 mm*.

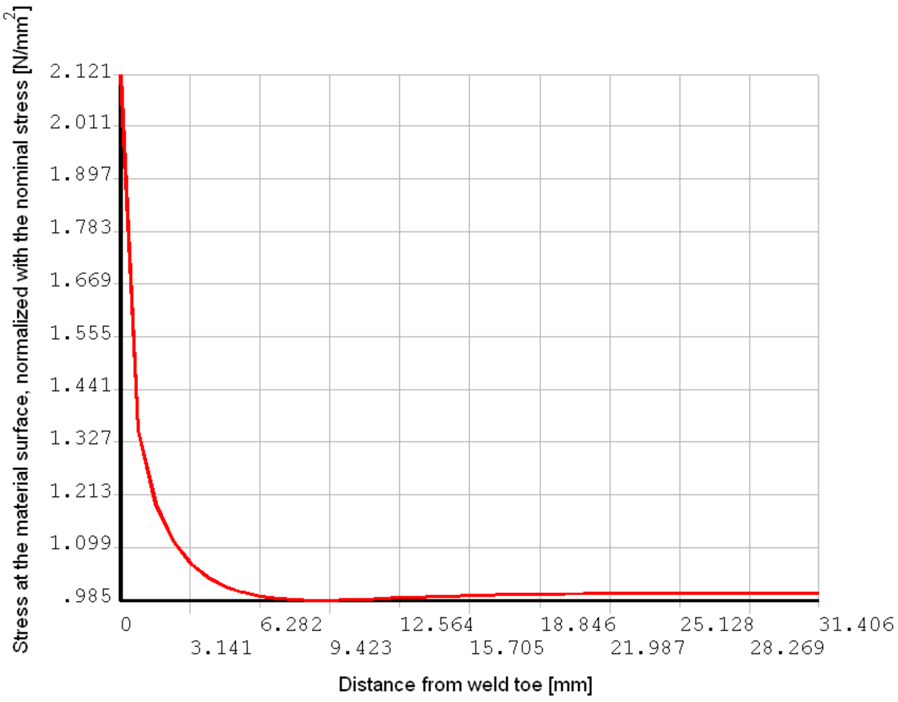


Figure 12 *The stress at the material surface decreases as the distance from the weld increases. Eventually the stress equals the nominal stress. The plate thickness is 15 mm.*

From the graph it can be seen that at a certain distance the stress becomes approximately equal to the nominal stress. This distance will depend on the thickness, *t*. In Figure 13 it can be seen that this dependence is linear. It should though be noted that here *t=s/2*, so the dependence is also linear in *s*.

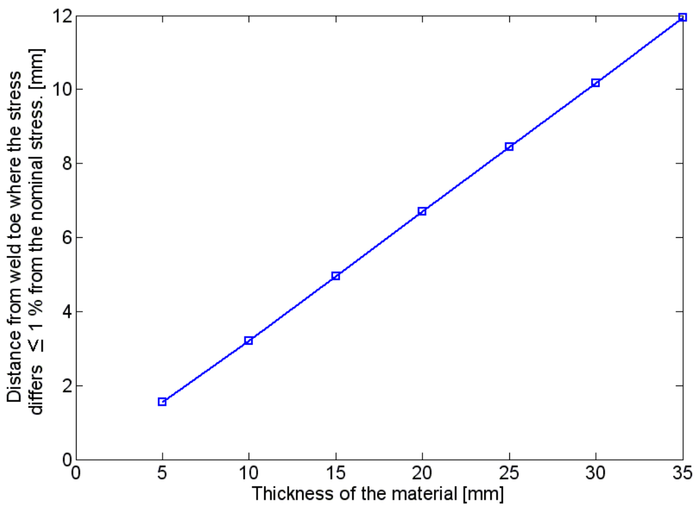


Figure 13 *The distance from the weld toe where the stress is approximately equal to the nominal stress depends linear in the thickness of the material. (Note that here, s=t/2, so the dependence is also linear in s.)*

From Figure 13 it can also be seen that the distance where the stress is equal to the nominal stress is quite small compared to the thickness of the material. Therefore it is not necessary to model the steel plates very long at all. Although, for the studies in this thesis, the plates are modelled with a length of $4.25 \cdot t\ mm$ away from the weld toe, and this is hereby proven to be more than enough.

All calculations in this thesis are made assuming plane stress, since this gives a more conservative result. Plain stress or plain strain is determined by the load case and joint type. The structure has approximately a plain stress condition on all non loaded surfaces. Plain strain is when a geometrical constrain prevents deformation in one direction. Plain stress gives a larger affected zone around the crack, i.e. plane stress is the worst case. Therefore this is used here. [4]

4.2 Effective Notch

IIW [5] has some recommendations for how to model when using the effective notch method. The effective notch method is not valid for thicknesses smaller than *5 mm* and the angle is supposed to be modelled as 45º when nothing else is known. This is in agreement with the geometry used for the non load-carrying example. These recommendations are also followed through the whole work.

When using the effective notch method the symmetry of the geometry can be used to minimize the computational work, see Figure 14. The edges in the weld toes are replaced by a radius of *1 mm* and also in the weld root a radius is introduced. As boundary conditions the symmetry conditions are enough and the nominal stress is applied as a line pressure on the horizontal line.

The mesh has to be rather fine in the radii to resolve the problem. At least 6 elements per 90º are required. Also, subdivision should be fine normal to the surface in order to model the steep stress gradient in thickness direction [11]. In Ansys *plane183* is used as element type.

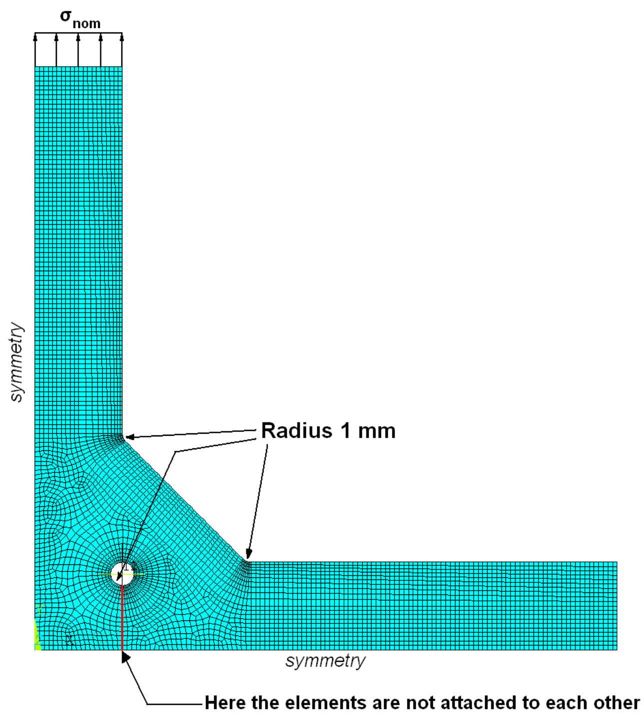


Figure 14 *The elements have to be relatively small in the radii. At least 6 elements per 90º are needed. Also in the normal direction the elements has to be rather small due to the large stress gradient in that direction. [10] [11]*

4.2.1 The Root Notch

As seen in Figure 14 the radius in the root can modelled as a keyhole shaped cut-out. This is not the only way to introduce a radius in the weld root. Fricke [11] gives another opportunity of an oval shaped root notch, compare Figure 15 (a) and Figure 15 (b).

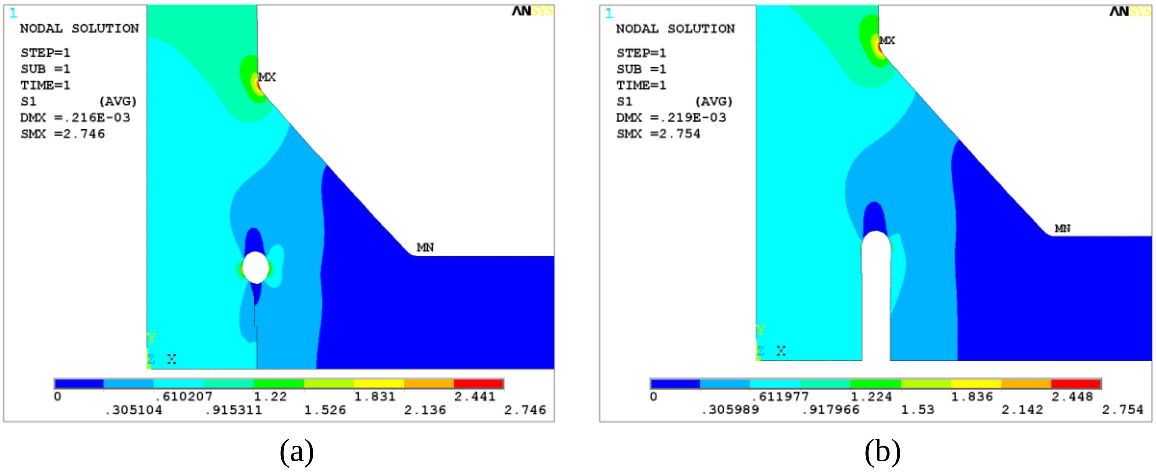


Figure 15 *The root notch is modelled with a keyhole or with an oval shape. The contour is the first principal stress.*

For a non load-carrying joint the keyhole notch may overestimate the real notch effect as the upper and lower parts of the notch do not exist in the real structure. For this load case, the oval shape seems more realistic. In Figure 16 the stress concentration factors, K_t, are given for the different shapes of the notch. The stress concentration factor is the same in the toe for oval and keyhole shaped root notch, while the oval shape gives lower values in the root. The stress concentration factors are constantly higher in the toe then in the root so the failure will come from the toe, where the shape of the notch does not affect the solution. So, for this particular geometry the shape does not matter.

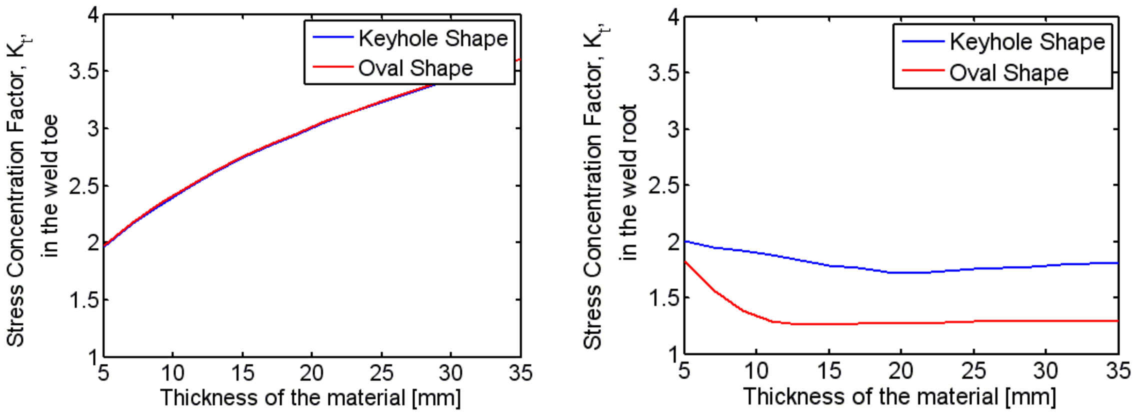


Figure 16 *The difference in the stress concentration factor when modelling with a keyhole shaped root and an oval shaped root. For the result in the weld toe there is no significant difference but for the weld root the differences are greater.*

For the load-carrying joint the stresses in the toe are affected by how the root is modelled. In Table 3 stress intensity factors, K_t, are calculated for two load-carrying joints with different weld sizes, modelled with the different root shapes.

Table 3 *The stress intensity factors are calculated in the root and in the toe for two different load-carrying joints. The root is modelled with either a keyhole shape or with an oval shape.*

Oval/Keyhole	*s* [mm]	*t* [mm]	$K_{t,root}$	$K_{t,toe}$
Oval	6	15	6.231	(5.127)
Keyhole	6	15	6.051	(4.908)
Oval	12	15	(3.821)	3.252
Keyhole	12	15	(3.798)	3.196

The stresses are higher when the root is modelled with an oval shape. Fricke [11] has made a comparison similar to this one and he concludes: “The oval shape is connected to a weaker structure, causing increased stresses at the weld toe.” Especially for small weld throat thicknesses the structure is weakened by the oval shaped root. Even though the differences are not that large, the root is modelled with the oval shape for all load-carrying joints to cover the worst case.

4.2.2 Estimation of the Lifetime

When estimating the lifetime with the effective notch method the largest principal stress is used for calculating the stress concentration factor [1][5]. Radaj et al [17] is suggests that the von Mises stress sometimes is a better alternative, but in this thesis the recommendations of IIW [5] is followed so the 1st principal stress is used. A nominal stress of *1 MPa* is applied on the geometry in the model, the stress concentration factor, K_t, can then be determined by looking at the maximal principal stress in the notch. This is of course if R=0 is assumed. The nominal stress used, (for the non load-carrying example $\sigma_{nom} = 120\ MPa$) are then multiplied by K_t to get the maximal stress range in the notch. The strength with the effective notch method is that only one FAT-value has to be considered, independently of structure and load case. IIW [5] recommends $FAT = 225$, and since the estimated lifetime is to be compared to real fatigue tests, the FAT-value has to be multiplied by, $\varphi_Q=1.3$ to get the right failure probability, see section 2.2.1. This corresponds to multiplying the result by two standard deviations and therefore a 50 % risk of failure instead of a 2.3 % risk is obtained [1]. The maximal stress range can then be inserted in a modification of equation (2:6).

$$N = 2 \cdot 10^6 \cdot \left(\frac{\varphi_Q \cdot FAT}{K_t \cdot \sigma_r} \right)^3 \quad \textbf{(4:1)}$$

This is the same whether the joint is load-carrying or non load-carrying, or whether the crack originates in the root or in the toe. When the crack propagates from the root, K_t is of course calculated in the root notch.

For the non load-carrying example with a material thickness, $t = 15\ mm$, the stress concentration factor in this case is 2.74 in the toe. The estimated lifetime is 1 408 000 cycles.

$$N = 2 \cdot 10^6 \cdot \left(\frac{\varphi_Q \cdot FAT}{K_t \cdot \sigma_r} \right)^3 = 2 \cdot 10^6 \cdot \left(\frac{1.3 \cdot 225}{2.74 \cdot 120} \right)^3 = 1\,408\,000 \text{ cycles}$$

4.2.3 Results

For the non load-carrying example the stress concentration factor for different plate thicknesses can be seen in Figure 17. The stress concentration factor increases as the steel plate thickness increases. In this figure the weld size, $s = t/2$, other ratios are treated in section 5.

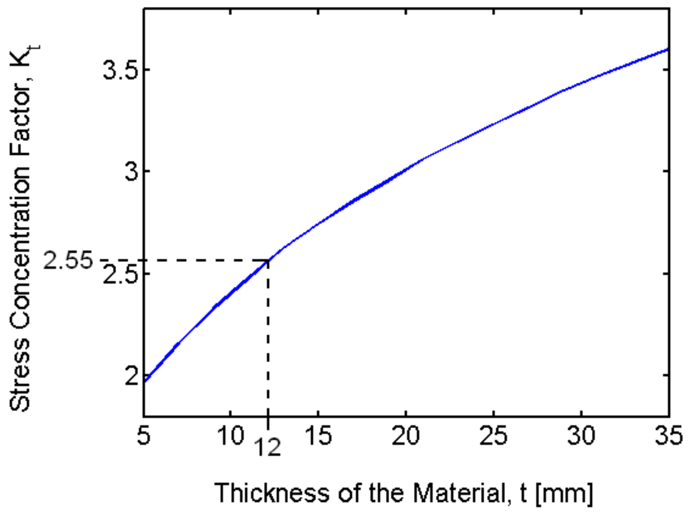


Figure 17 *The stress concentration factor increases when the material thickness increases.*

To validate the model and the calculations, the results for the non load-carrying joint, are compared to results obtained by Martinsson [8]. He has performed calculations of the stress concentration factor on a similar geometry with the steel plate thickness of *12 mm*. The only difference is that he has an angle of $\theta = 60°$, compared to $\theta = 45°$ that is used here. He got a stress concentration factor, $K_t = 2.6$. This is a little higher than 2.55 that are obtained here, see Figure 18. But a larger angle would give a sharper notch and therefore a higher stress concentration factor. So, the results obtained here are realistic.

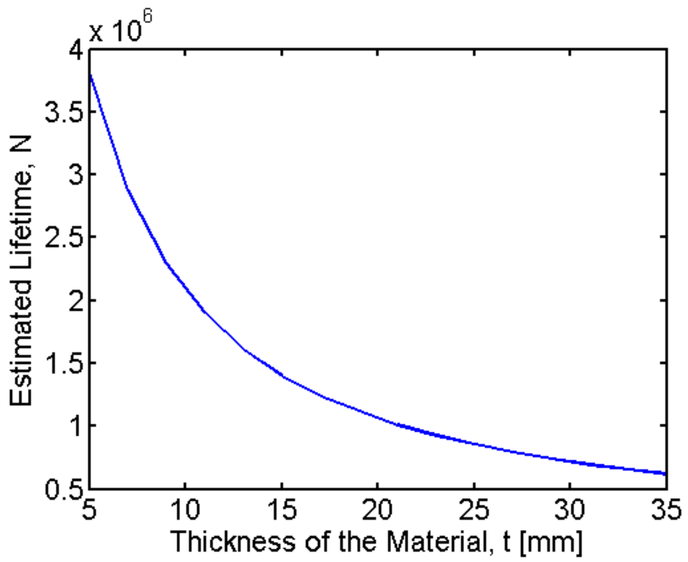
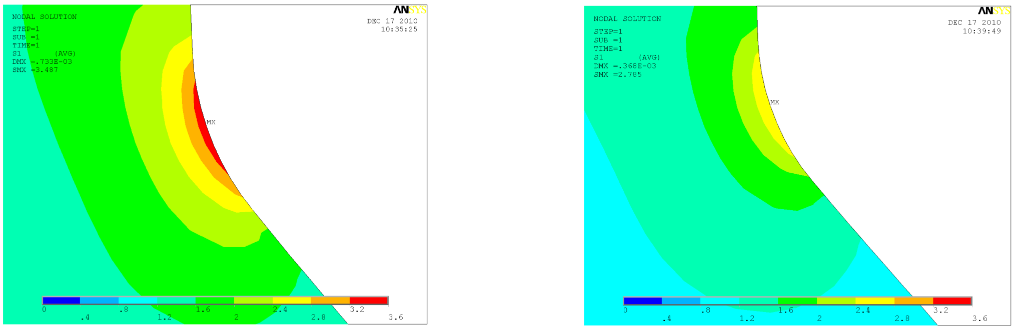


Figure 18 *Estimations of the lifetime due to the effective notch method is highly dependent on the steel plate thickness.*

If the estimated lifetime is calculated as in the previous section the result can be seen in Figure 18. The estimated lifetime decreases when the thickness increases. This is not surprising since it is a fact that the fatigue lifetime decreases with increased material thickness. This is due to a combination of statistical, technological and geometrical factors. The statistical effect implies that in a larger amount of material the probability of a defect that can affect the lifetime is greater. The technological effect originates from the fact that thick materials has lower fatigue resistance than the same material that has been, mechanically processed to a thinner dimension. The geometric effect is dependent on the stress gradient in the direction of the crack propagation, the effect of the stress gradient increases when the material thickness increases, since the radius in the weld toe is constant. For a propagating crack, the stress decreases more slowly for a thick sheet compared to a thin one. [1] In Figure 19 this effect can be seen. The thicker steel plate, Figure 19 (a), has a larger stress gradient than the thinner steel plate, Figure 19 (b).



(a) *(b)*

Figure 19 *The stress gradient is larger for a thicker steel plate. This decreases the fatigue resistance in the structure.*

4.3 LEFM

In this section, the same structures as was analysed by the effective notch method in the previous section, are analysed by the LEFM method.

4.3.1 Geometry Modelling

When calculating the lifetime using the LEFM method, a small crack is already assumed to be present. The propagation of the crack through the material is simulated. The initiation phase has already past. For the non load-carrying example, the crack is assumed to propagate from the weld toe and therefore a small crack is initialized there. The transition between the weld and the material are modelled sharp, without a rounding radius as before with the effective notch method.

The element *plane82* is used in Ansys and a simple symmetry is used to reduce computation time, see Figure 20. The boundary conditions used are the symmetry condition and one point is also fixed in the y-direction to prevent rigid body motion. The crack is initiated perpendicular to the maximal principal stress.

For the load-carrying joint the crack can also propagate from the weld root. Then, a fictitious crack initiation is not needed since the root gap acts like the initiation. The model is for those simulations made with double symmetry, as the models for effective notch are. This is to get away with the problem that there would be two cracks propagating at the same time. Theory for this is complicated and uncertain, and it is not in the scope of this thesis.

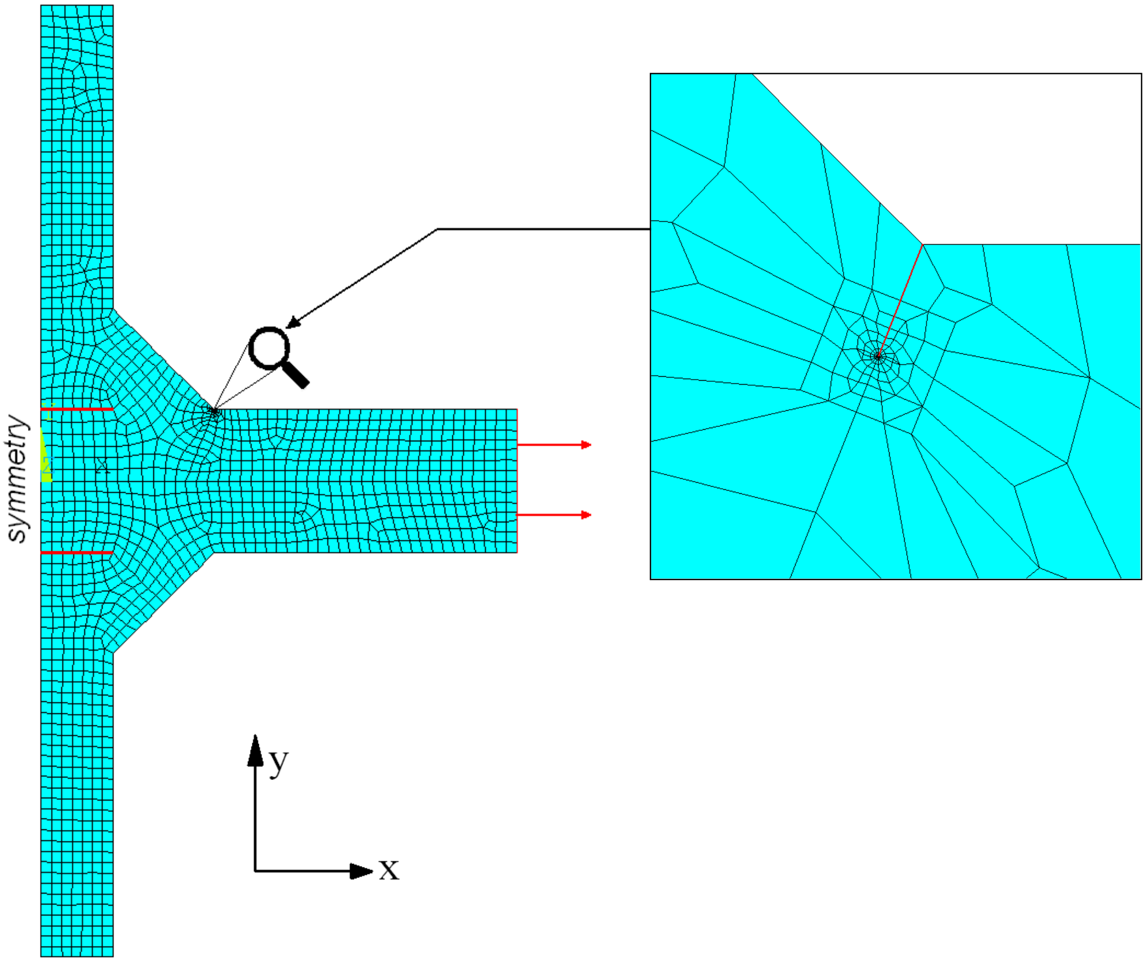


Figure 20 *With the method LEFM a single symmetry model is used. The crack is initiated perpendicular to the maximal principal stress and the elements have to be relatively small around the crack tip.*

Stress and deformation field around the crack tip generally have high gradients therefore the mesh has to be much finer in the vicinity of the crack tip than in the remaining of the model. To be able to retain this as the crack propagates into the material a square encloses the crack tip. Inside this square the elements are set to be smaller than outside the square. The square moves along with the crack tip as the crack propagated through the material. Ansys has a special way to handle the elements in direct contact to the tip of the crack. The stresses and strains are singular at the crack tip, varying as $1/\sqrt{r}$, see Figure 21 a). This requires a, what Ansys calls, *singular element* near the crack tip. If a fracture mechanics analysis is to be made in Ansys, quadratic elements have to be used. The midside nodes are then skewed to a quarter of the distance from the crack tip, see Figure 21 b), to produce the singularity effect. [13]

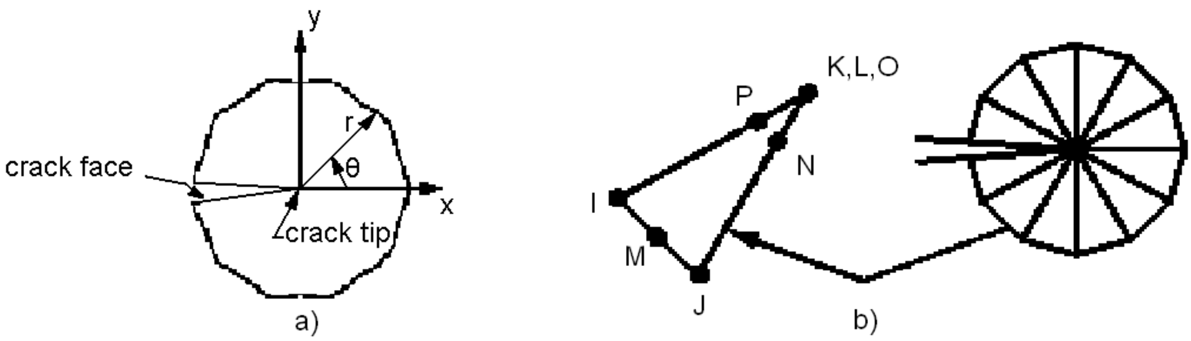


Figure 21 *In Ansys the singularity in the crack tip is modelled by skewing the midside nodes to a quarter of the distance from the tip. [13]*

Due to the symmetry in the model, the crack is mirrored falsely. To verify that the symmetry does not effect the estimation of the number of cycles to failure, calculations are made with a full model, without symmetries. The result in Figure 22 shows that there are no significant different in the lifetime when using the symmetry. Therefore the symmetry is used to reduce computational time.

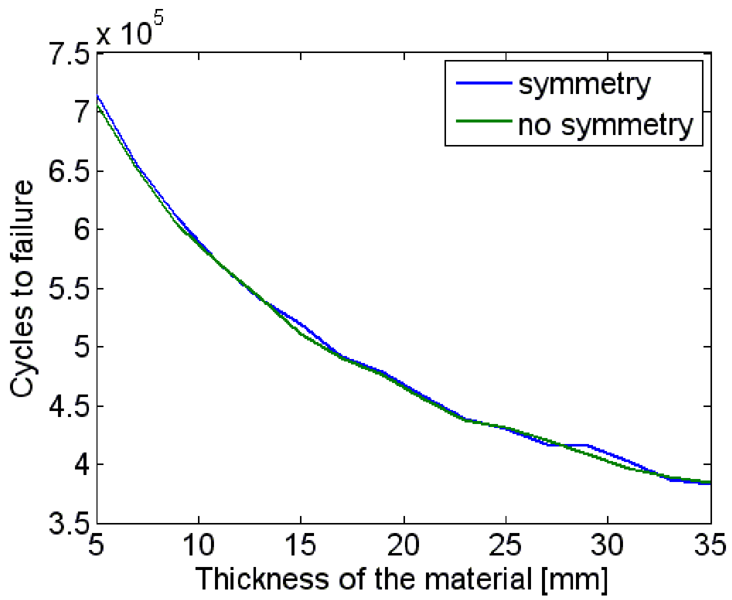


Figure 22 *There is no significant difference for the estimated lifetime when the model is made with or without symmetry.*

4.3.2 Crack initiation and propagation

For the cruciform joint, the crack is loaded in a combined mode I and II. The interaction theory for mode I/II is that the crack will move in the direction of maximum circumferential stress around the crack tip [12]. The crack is initiated as a straight crack perpendicular to the largest principal stress in the weld toe. A mayor question is now, in what direction will the crack grow? According to Byggnevi [14] the mixed mode I/II almost always results in a kinking of the crack and the kink grows in the mode I direction, i.e. in the direction perpendicular to the largest principal stress. In the simulations the crack is propagating with a fixed crack growth increment, Δa, until the crack reaches half the thickness of the material. Figure 23 shows how the crack growth increment and the deflection angle, φ, decide where the next crack tip will be placed. The deflection angle is determined by the direction of the largest principal stress.

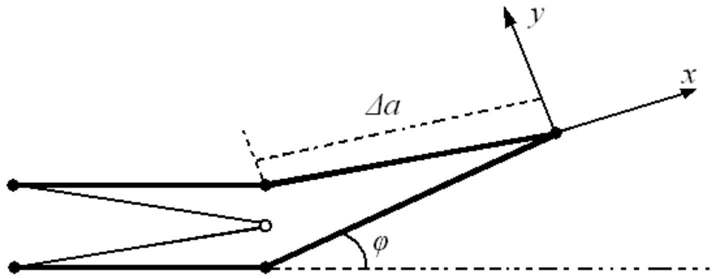


Figure 23 *The crack growth increment and the deflection angle between the crack front and the crack extension decide where the next crack tip will be placed.*

4.3.3 The Constants in Paris' Law

The fatigue life, *N*, is calculated using Paris' law, equation (3:7), with the effective stress intensity factor calculated with the strain energy release theory, equation (3:9). For this case, though, ΔK_{II} is much smaller than ΔK_I, so all the equations for ΔK_{eff} will give approximately the same result. The choice of equation will be negligible compared to other uncertainty factors such as, the crack initiation length and the choice of the *C*-value in Paris' law.

The variables *C* and *n* in Paris' law are an issue when simulating the crack propagation. The *n*-value is coupled to the *m*-value giving the slope of the S-N curve. Almost every studies done on welded steel joints with LEFM uses *n=3* [8] [12] [14], that is also what the International Institute of Welding (IIW) are recommending [5]. This is despite the fact that many fatigue tests performed suggest this is only valid for certain geometries and load cases [6] [16]. In this thesis the value of *n=3* is used since no other value are proposed for this particularly structure.

The value of *C* depends on the geometry too, but is also depends on the probability of failure that is desired. To be able to later on compare the results obtained here with some real fatigue tests, the probability of 50 % is used. IIW [5] recommends $C = 5.21 \cdot 10^{-13} MPa\sqrt{mm}$ for a failure probability of 2.3 %. Barsoum [12] and Martinsson [8] are using a value of $C = 1.502 \cdot 10^{-13} MPa\sqrt{mm}$ for a failure probability of 50 %, based on the recommendations of IIW. This value is used here as well.

4.3.4 Simulations

For each step in the simulation of the crack propagation the deflection angle, the stress intensity factors, K_I and K_{II}, and also the effective stress intensity factor, K_{eff}, are calculated. The crack is propagated to half the thickness of the material, see Figure 24. For a root crack the crack is propagated through the whole weld.

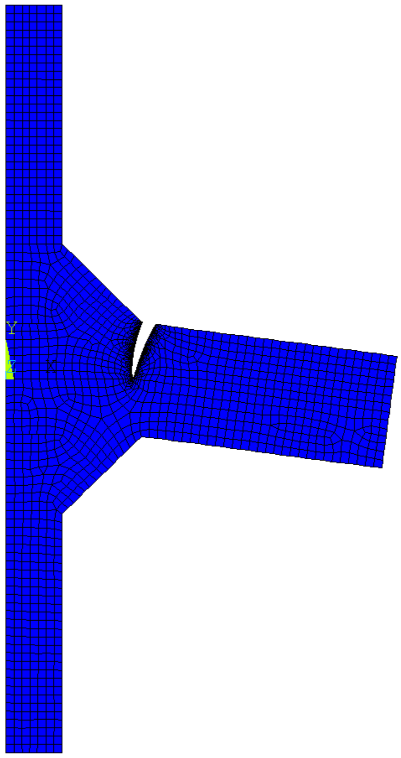


Figure 24 *For a toe crack, the crack is propagated to half the thickness of the material. In this plot the scaled deformation is showed.*

In Figure 25 the effective stress intensity range is plotted versus the crack length for the non load-carrying joint with a plate thickness of *15 mm*. The stress intensity increases as the crack length increases. Eventually it will reach the critical value, ΔK_{IC}, where the crack will grow to failure instantly. The smallest value of ΔK_{eff} is about $200\, MPa\sqrt{mm}$. IIW [5] gives a value of the threshold, $\Delta K_{th} = 63\, MPa\sqrt{mm}$, so the value of ΔK_{eff} is clearly over the threshold value here, and the crack will grow. For all simulations done with the LEFM method, this threshold value has been checked.

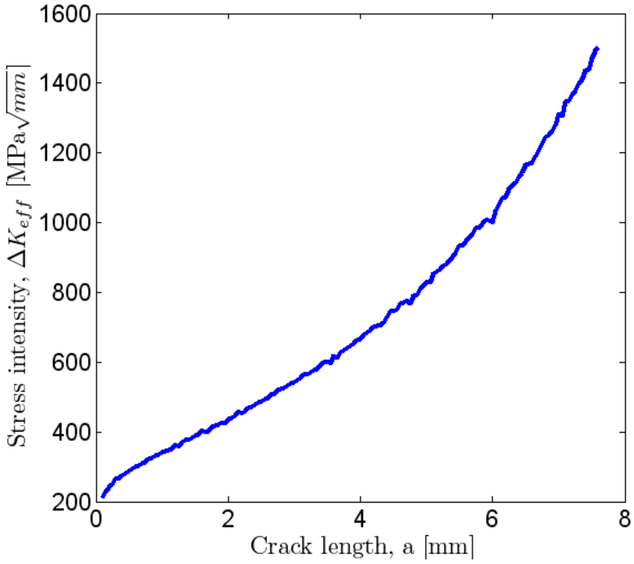


Figure 25 *The stress intensity factor increases as the crack length increases. Eventually it will reach the critical value,* ΔK_{IC}. *The thickness of the material is here 15 mm.*

In Figure 26 the crack length is plotted as a function of the number of cycles so far applied on the specimen. The thickness of the material for this example is *15 mm*. When the crack length is approximately *7.5 mm* (half the thickness) the curve is almost vertical. This means that now, a few numbers of cycles causes the crack to grow much. There are not so many cycles left before a complete failure will occur. Most of the lifetime in the specimen is when the crack length is short, and the curve in Figure 26 has a small slope. As the crack length grows the lifetime left in the material decreases and when the crack has reached half the thickness the lifetime left is negligible.

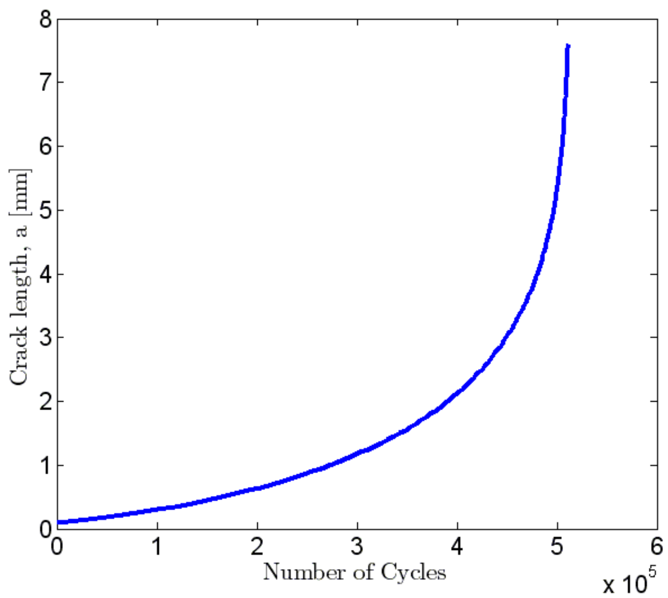


Figure 26 *The thickness of the material is 15 mm. When the crack length approaches half the thickness (7.5 mm) the number of cycles required, to make the crack grow, are fewer. The lifetime left before failure is negligible.*

4.3.5 Calculation of the lifetime

As mentioned in section 3.4, Paris' law has to be integrated to get the lifetime. From the Ansys simulation the effective stress intensity factor are given for each extension of the crack. For further details on how this is done, see the ANSYS help [13]. The integration of Paris' law can then be approximated by

$$N_{i+1} \approx N_i + \frac{\Delta a}{C \cdot \left(\frac{\Delta K_{eff,i+1} + \Delta K_{eff,i}}{2} \right)^n} \qquad \text{where } N_0 = 0 \qquad \textbf{(4:2)}$$

The mean value of the stress intensity factor is taken from the present step and the previous step, to prevent a constant over- or underestimation of the lifetime. If Paris' law is integrated straight forward it would result in this:

$$\int_0^N dN = \int_{a_i}^{a_f} \frac{1}{C(\Delta K_{eff}(a))^n} da \qquad \textbf{(4:3)}$$

where a_i is the initial crack length and a_f is the final crack length. The curve $g(a) = \frac{1}{C(\Delta K_{eff}(a))^n}$ has to be integrated numerically since there is very hard to find an analytical expression of *g(a)*. This can be done with either an over- or an under sum, see Figure 27. The upper sum will result in an over estimation of the lifetime and the lower sum will result in an underestimation. Equation (4:2) will give the same result as the mean of the upper and lower sum, and this will give the most accurate result.

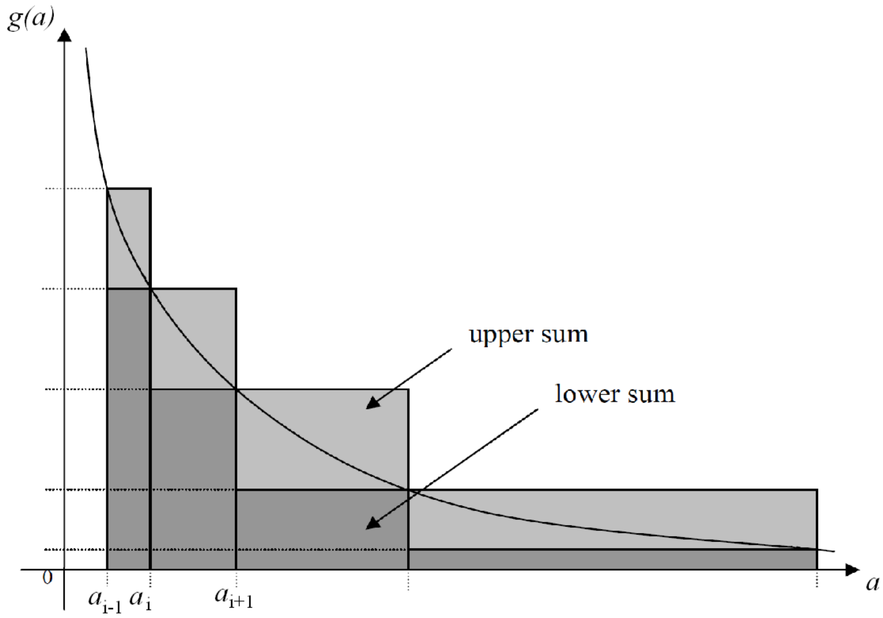
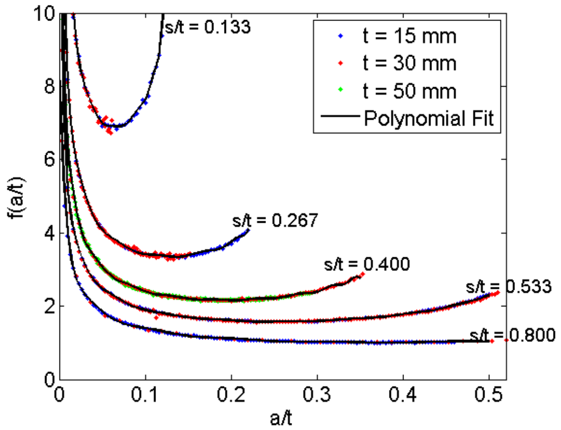


Figure 27 *A mean value of the upper and lower sum will give the most accurate result when integrating numerically.*

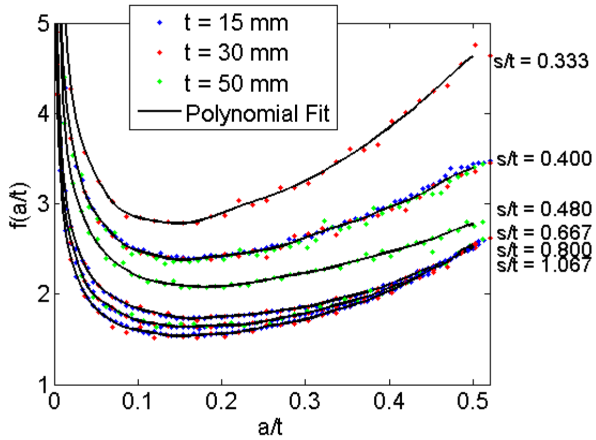
For a steel plate thickness of *15 mm* the estimated lifetime for LEFM method is, N = 518 716 cycles, taken from the data to Figure 22. This is to be compared to the estimation with the effective notch method, N = 1 408 000 cycles.

4.3.6 Function *f(a/t)* in the LEFM analysis

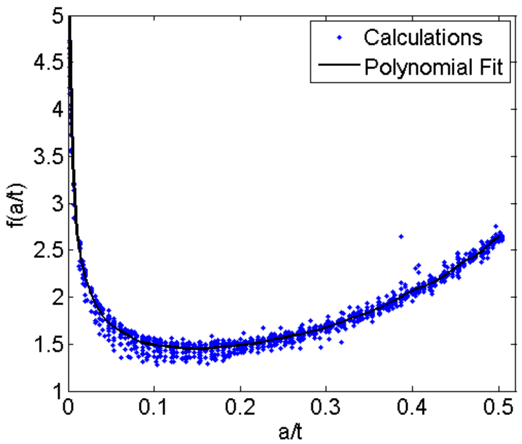
In equation (3:4) it is stated that $K_I = \sigma_{nom}\sqrt{\pi a} \cdot f$. The function *f* is tabulated for different crack types in handbooks, for example [24]. By doing a lot of simulations for different values of *a/t* this function can be found by a polynomial fit of the simulation results. In Figure 28 this can be seen for the different joints handled here. The non load-carrying joint is not very dependent of the weld throat thickness, *s*, so there all data points are fitted into one curve.



(a) Load-carrying joint with the crack assumed to start at the weld root.



(b) Load-carrying joint with the crack assumed to start at the weld toe.



(c) Non load-carrying joint.

Figure 28 *The function f from equation (3:4) is plotted versus the quotient a/t.*

4.3.7 Crack increment length, *Δa*

There are mainly four critical variables that you have to choose when doing a LEFM simulation of this kind, Δa, a_i, and the material constants C and n. The first one is the crack increment length, Δa, and the second one is the crack initiation length, a_i. Here it is investigated how Δa affects the estimated lifetime and in the next section the crack initiation length is analysed.

It would be optimal to use a so small crack increment as possible, because that would be more accurate. The limitation is the computational time that increases as the increment decreases. In Figure 29 the estimated lifetime for different crack increments are presented as a function of the crack length. The final result is stabilizing as the crack increment is decreased.

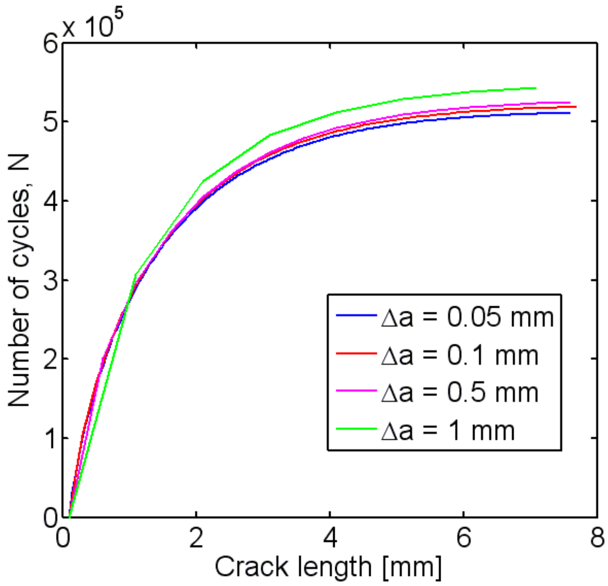


Figure 29 *For a material thickness of 15 mm the estimated lifetime decreases when the crack increment decreases. The result for the smallest crack increment should be the most accurate one.*

Simulations for the non load-carrying joint are performed for different material thicknesses, *t*, and different crack increment sizes. A table of the result can be seen in appendix A. The conclusion from the results is that these sizes of the crack increment are suitable and gives a sufficiently accurate estimation,

$$\Delta a = \begin{cases} 0.05mm, & t < 15mm \\ 0.1mm, & 15mm \le t < 25mm \\ 0.5mm, & 25mm \le t < 50mm \\ 0.75mm, & 50mm \le 5 \end{cases}$$

These values are used for all the simulations with LEFM in this thesis. Although when the simulations are done for a crack propagating from the root, the weld throat thickness is considered instead of the thickness of the material.

$$\Delta a = \begin{cases} 0.05mm, & s < 7.5mm \\ 0.1mm, & 7.5mm \le s < 12.5mm \\ 0.5mm, & 12.5mm \le s < 25mm \\ 0.75mm, & 25mm \le s \end{cases} \quad \textbf{(4:4)}$$

4.3.8 Crack initiation length, a_i

The crack initiation length is the most difficult variable to define when doing a LEFM analysis of this kind. Fracture mechanics was originally intended to use when a flaw was detected and you want to know whether or not the design will hold anyway. But when LEFM is used to get design conditions, the initial crack is only a small defect of the magnitude of the microstructure of the material. Many investigations, similar to this one, have used an initiation length of *0.1 mm* [4] [8]. Radaj et al [17] recommends, $a_i \ge 0.1mm$, for the crack to be larger than a “short crack”. For a “short crack” the theory of LEFM presented above are invalid. The small crack will give so small values of ΔK_{eff} that even if they are above the threshold value perhaps they are not in the linear part of the curve in Figure 10. Then Paris’ law is not valid, at least not for the given values of *C* and *n*. In all the simulations the initial crack size is *0.1 mm*. The estimated lifetime is highly dependent on the initial crack length, since much of the lifetime is consumed while the crack is small. This parameter is therefore analysed further.

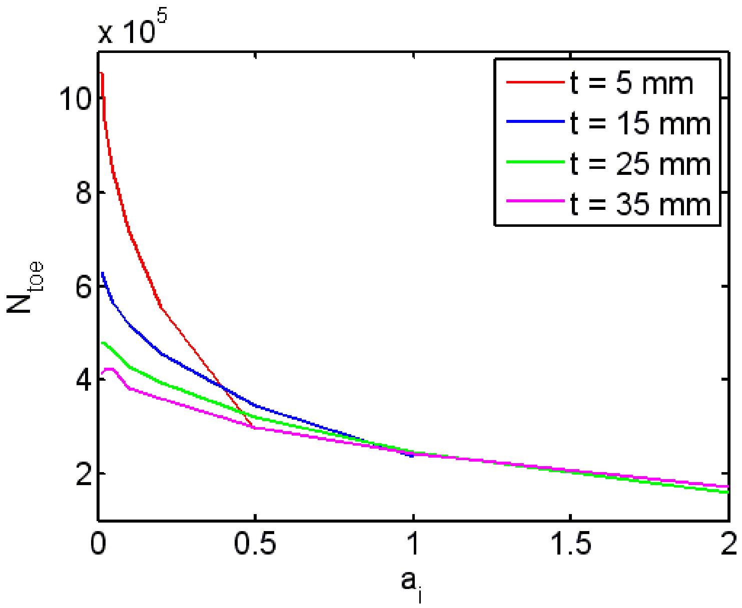


Figure 30 *The estimated lifetime vs. the length of the initiated crack in mm. The curves are steep when the initiated crack is small, indicating that much of the lifetime is consumed while the crack is small. There is a huge different in lifetime for a small and a large initial crack, especially when the material thickness is small.*

Figure 30 shows how the estimated lifetime varies when the initiated crack length increases. The curves are steep when the initiated crack is small, indicating that much of the lifetime is consumed while the crack is small. There are a huge different in lifetime for a small and a large initial crack, especially when the material thickness is small. Therefore the choice of initial crack size is critical to the result. This is one of the largest weaknesses with the LEFM method. It should though be noted that this is not a problem when a root crack is considered. Then the root gap is used as the initial crack so it is expected for the LEFM method to be more accurate for root cracks than for toe cracks.

4.4 Nominal Method

The nominal method uses, as mentioned in section 3.1, standards and norms to determine the lifetime. Here two different standards will be used, the Swedish standard BSK 99 and the standard of IIW. For the lifetime estimations with the nominal method no FE analyses are used.

4.4.1 BSK 99

When performing a fatigue assessment with the nominal method you have to define basically two things, the nominal stress and what C-value (FAT-value for IIW standard) to use for this specific joint. The nominal stress is very easy to determine for toe failures, the nominal stress is simply the applied stress, for the non load-carrying example, *120 MPa*. It is harder for root failures when the stress through the weld has to be considered. BSK 99 has specified how this should be done and it can be seen in Appendix B. When determine the C-value you look in the standard for a similar geometry and load case. For the non load-carrying cruciform joint, there is no obvious choice in the BSK 99 standard. Here a C-value of 56 is chosen (joint number 39) [18]. This means that for $2 \cdot 10^6$ cycles the probability of failure is 2.3 % if the nominal stress range is *56 MPa*. To get the lifetime estimation this is then put into equation 2:6.

BSK99 includes a thickness effect that increases the lifetime when the thickness is less than *25 mm*.

$$\varphi_{\dim} = \left(\frac{25[mm]}{t} \right)^{0.0763} \geq 1 \qquad \textbf{(4:5)}$$

σ_r from equation (2:6) are multiplied by the thickness factor, φ_{dim}, to get the right result. For a nominal stress of *120 MPa* this results in an estimated lifetime according to Figure 31. The estimated lifetime is greater for smaller thicknesses and when the thickness is bigger than *25 mm* the thickness has no effect at all and the curve is horizontal.

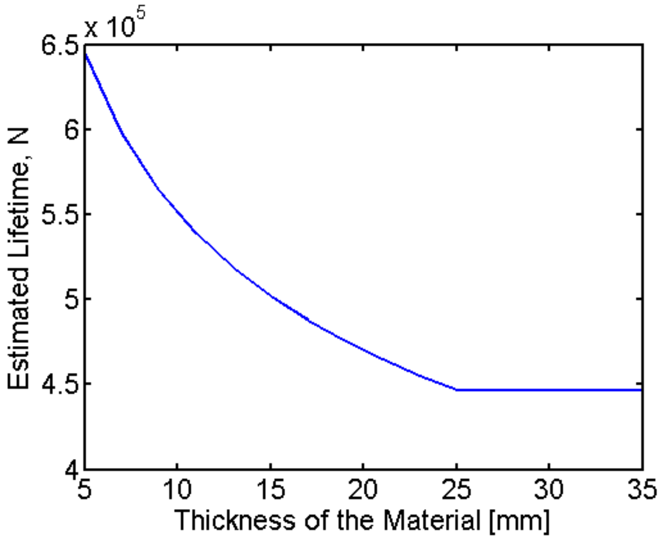


Figure 31 *For the nominal method, due to BSK 99, the thickness of the material only have effect on the lifetime when the thickness is thinner than 25 mm.*

For a non load-carrying joint with a steel plate thickness of *15 mm*, the estimated lifetime due to BSK 99 standard is, N = 501 951 cycles.

4.4.2 IIW

The IIW standard is very similar to the BSK 99 standard. The only differences for the non load-carrying example are that IIW uses a different FAT-value and that the material thickness is handled in a different way. For a root failure IIW also has a different way of calculating the stress through the weld, this can be seen in Appendix C.

For IIW there is a standard geometry that matches the non load-carrying cruciform joint almost exact. The FAT-value is 80, quite much higher than the value given by BSK 99 standard. Instead of adding extra lifetime to thicknesses smaller than *25 mm*, that BSK 99 suggests, IIW are subtracting lifetime for material thicknesses over *25 mm*. IIW has for this joint a thickness factor,

$$\varphi_{\dim} = \left(\frac{25[mm]}{t_{eff}} \right)^3 < 1 \qquad \textbf{(4:6)}$$

where t_{eff} also depends on the weld throat thickness, see [5]. The results are presented in Figure 32.

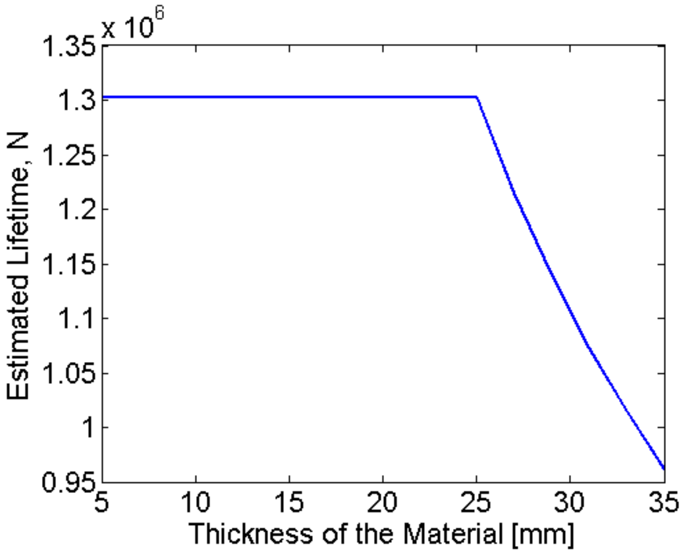


Figure 32 *For the nominal method with the standard of IIW the thickness only affects the estimated lifetime if the thickness is above 25 mm.*

For a non load-carrying joint with the material thickness of *15 mm* the IIW standard predicts a lifetime of, N = 1 301 926 cycles. This is more then twice as much as BSK 99.

4.5 Comparison of the methods

Curves for all of the methods for the non load-carrying example, are put together for comparison in Figure 33. The estimated lifetime is plotted against the thickness of the material.

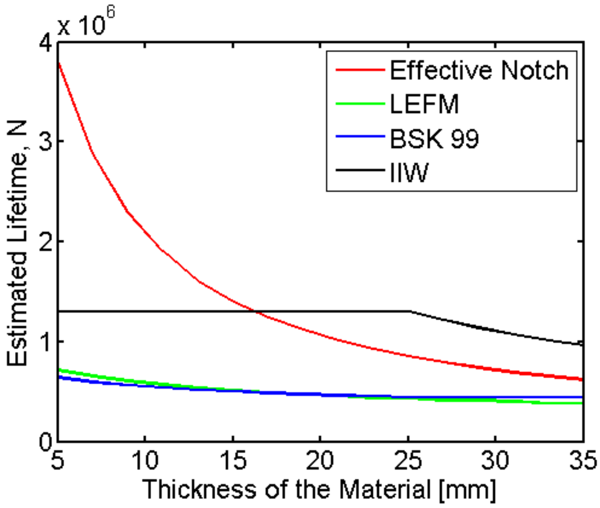


Figure 33 *The effective notch method, LEFM, BSK 99 and IIW all estimates the lifetime different. The BSK 99 and LEFM are quite similar. IIW are slightly higher and the effective notch method predicts a much longer lifetime, especially for small thicknesses.*

For all of the methods the lifetime decreases when the thickness increases. This is due to the geometrical, statistical and technological factors affecting this. BSK 99 and LEFM gives approximately the same estimations while the effective notch method gives a much higher estimated lifetime, especially for the rather small thicknesses. IIW are twice as high than BSK 99 and LEFM but not as high as effective notch for the small thicknesses. In section 6 fatigue tests are performed to get real test results to compare with.

5 The Weld Throat Thickness

In this section the effect of the weld throat thickness is analysed. How are the different methods treating a change in the weld throat thickness for different plate thicknesses? Three different plate thicknesses, $t = 15\ mm$, $t = 30\ mm$ and $t = 50\ mm$, are used. For the load-carrying joint two cases are possible, either the crack initiates from the weld toe or it originated from the weld root. In a non load-carrying joint the crack generally starts in the weld toe. For the load-carrying joint it is investigated for what weld throat thickness the crack initiation point will move from the root to the toe.

5.1 Crack initiation point

For the load-carrying cruciform fillet weld joint, the crack initiation point can be either in the weld root or in the weld toe. The location of the initiation point depends on the weld size, plate thickness and weld penetration. Previously experimental results have indicated that the fatigue crack initiation point also depends on the magnitude of the stress range, but the reason for this is not yet clear and more can be read about it in Kainuma et al [21]. The weld size at which the crack origination point changes from the root to the toe is the critical size, $s_{critical}$. The critical weld size is generally normalized by the plate thickness. Many studies of this has been made and fatigue test has been gathered and compiled by Gurney [19], giving the ratio range from 0.85 to 1.0 for the weld leg length and the plate thickness. Assuming a weld angle of, $\theta = 45^{o}$ as used here, this can be converted to a ratio between the weld throat thickness, s, and the plate thickness.

$$s_{critical} / t = [0.601, 0.707] \qquad \textbf{(5:1)}$$

This is a value based on experimental test results. In section 6 fatigue tests are performed on structures with a plate thickness of *15 mm* and a weld throat thickness of *12 mm*. This means a ratio *s/t = 0.8*, which is larger than the critical value. In this structure the crack should originate from the weld root. In another test performed the weld throat thickness is *6 mm*, and the ratio, s/*t = 0.4*. This is below the critical value and the crack should initiate in the weld root.

In reality the critical weld size is given by equation (5:1), but how well are the different assessment methods predicting this critical value? In Figure 34 the estimated lifetime is plotted against the weld throat thickness, *s*, for different plate thicknesses and for different calculation methods. Both the case where the crack originates in the root and the case where it is initiated in the toe are plotted in the same picture. Where the two lines are crossing each other the critical value is found.

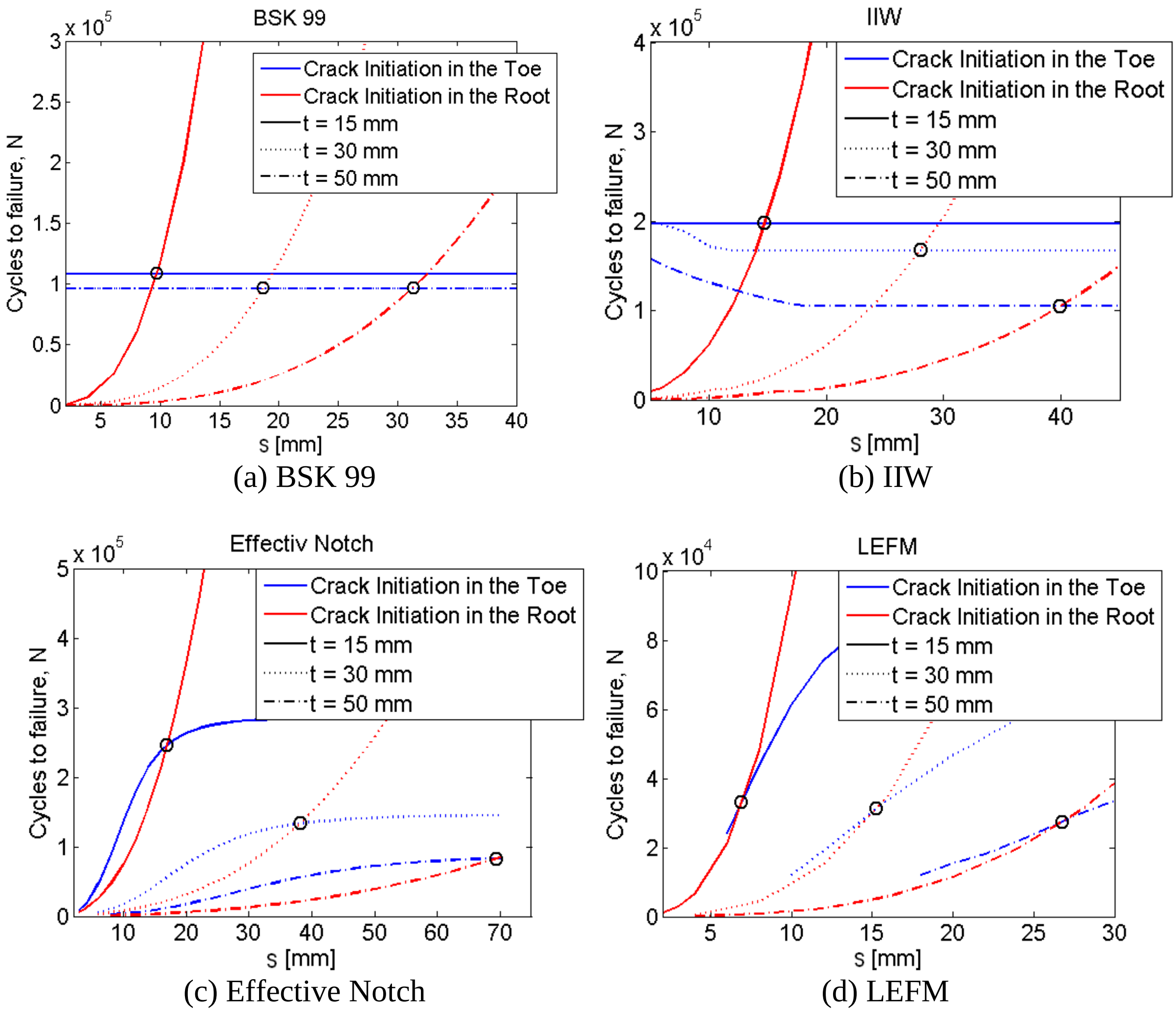


Figure 34 *For the four evaluation methods the critical value of the weld throat thickness,* $s_{critical}$*, is determined for when the crack moves from a root crack to a toe crack. Blue lines are for analyses where the crack is assumed to start at the weld toe and red lines are for analyses where the crack is assumed to start at the weld root. The different dotted lines are for different plate thicknesses.*

In the plots in Figure 34 the critical values are marked with a black circle. The values where the lines are crossing each other are presented with a better accuracy in Table 4. The critical value is approximately constant for a fixed ratio of *s/t* and the mean value from the three thicknesses used in the analyses is stated in the last row of the table.

Table 4 *The limit for the weld throat thickness for when the crack initiation moves from the root to the crack. The values are from the plots in* **Figure 34** *for the different calculation methods and plate thicknesses.*

	BSK 99			IIW			Effective Notch			LEFM		
t [mm]	**15**	**30**	**50**	**15**	**30**	**50**	**15**	**30**	**50**	**15**	**30**	**50**
$s_{critical}$ [mm]	**9.69**	**18.69**	**31.21**	**14.73**	**28.01**	**40.05**	**16.93**	**38.10**	**69.40**	**6.82**	**15.15**	**26.61**
$s_{critical}/t$	**0.646**	**0.623**	**0.624**	**0.982**	**0.934**	**0.801**	**1.129**	**1.270**	**1.388**	**0.455**	**0.505**	**0.532**
mean of $s_{critical}/t$	**0.63**			**0.91**			**1.26**			**0.50**		

BSK 99 is the only evaluation method that predicts this critical value correct. IIW and effective notch gives to large values, meaning that the weld size according to them should be larger than actually needed. LEFM underestimates the weld size needed for the crack to change initiation point from the root to the toe. The critical value is too low with the LEFM method.

It should be noted that this kind of reasoning is not the whole truth for the LEFM method. With this method you can follow the crack as it propagates through the material and not just look at the assumed number of cycles to failure. Even for larger value of *s/t* than 0.5 the range of the stress intensity factor, ΔK_{eff}, is larger in the root than in the toe in the beginning, when the crack is small. This indicates that a crack should first be initiated in the root. It should though be noted that this does not prevent an additional crack initiation in the toe. In this analysis it is assumed that the crack grows either from the root or from the toe. In reality it is not impossible for two cracks to propagate at the same time, in fact this is often the case. So even though a structure has a lower estimated lifetime if a single crack is assumed to grow from the toe than if a single crack is assumed to grow from the root, it is not certain that the failure will occur in the toe. When a small crack is initiated in one location this could redistribute the forces and the crack could stop grow and another one initiates. If this phenomenon is desired to be under consideration within a LEFM analysis, theory for this is needed. This kind of analysis is complex and is beyond the scope of this thesis.

5.1.1 Misalignment

The nominal methods are built up by experimental tests, and therefore a tolerance for misalignments in the structures is included in the analysis. For the effective notch method on the other hand, no misalignment is covered. To investigate how misalignment is affecting how the effective notch method estimates the critical weld size, a model of a load-carrying joint without symmetry is made, and a small axial misalignment is included in the model, see Figure 35. The plate thickness is *15 mm* and the weld size is, *s = 12 mm*.

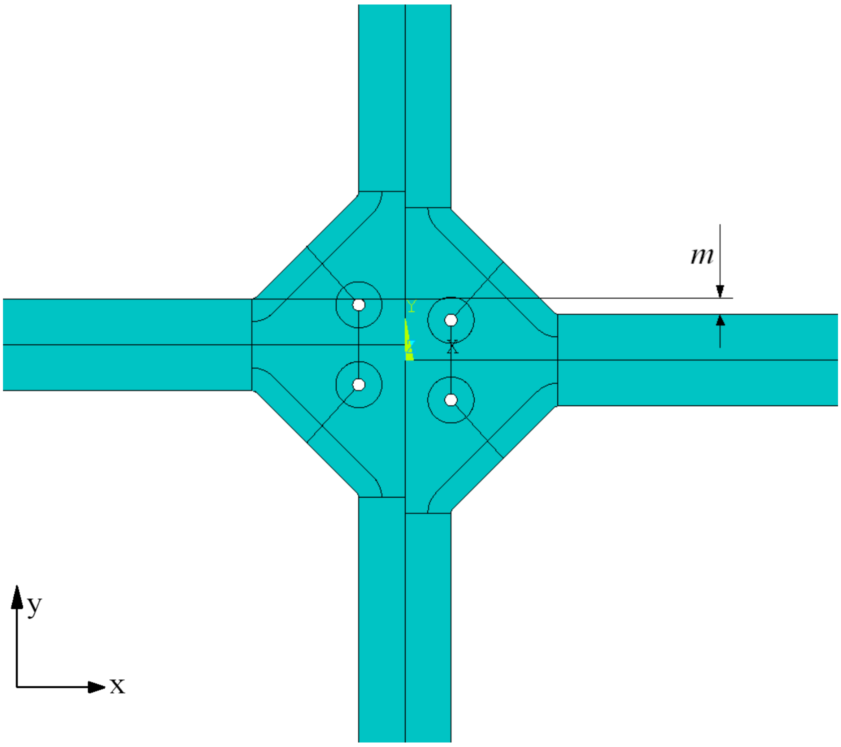


Figure 35 *An axial misalignment is added to the model to investigate how this will affect the result. This is a load-carrying cruciform joint. The steel plate thickness is here 15 mm and s = 12 mm.*

As boundary conditions, the nodes on the left side are fixed in the x-direction, and also one point on the left side is fixed in the y-direction to prevent rigid body motion. On the right side a stress is applied in form of a line pressure. The nodes on the right side are also coupled so that they always will be in a straight vertical line. This is because a bending force is induced in this problem, and the edges of the plates will otherwise twist in the model.

Due to the induced bending forces, this analysis is dependent of the length of the steel plates. In the model, the steel plates are modelled so long that the stress concentration factor does not change in 3 significant digits when the model is made even longer.

In Figure 36 you can see how the stress concentration factors are affected when axial misalignment is included in the model. When the misalignment is *1.36 mm* or larger the effective notch method predicts a crack from the toe instead of a crack from the root. This is 9.1 % of the plate thickness (*15 mm*), and seems to be quite much. It is unlikely that the misalignment is this high in a structure. For this structure the ratio of the weld size and the plate thickness is, $s/t = 0.8$, and the crack should start at the toe, see equation (5:1). So, for the effective notch method to give a statistically correct result, at least this much misalignment is needed to reach the correct critical value.

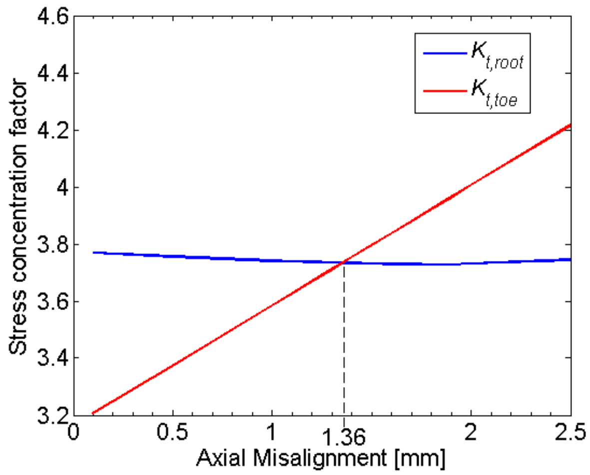
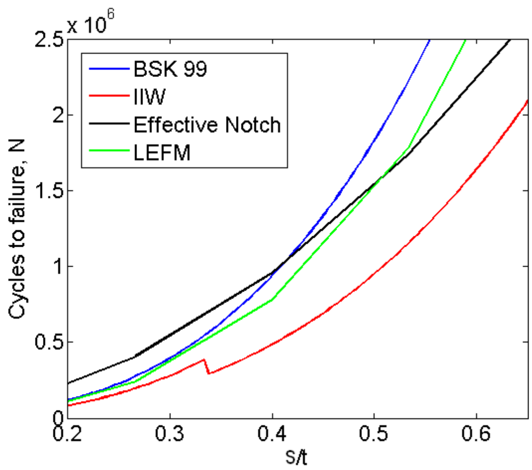


Figure 36 *A load-carrying joint, with t = 15 mm and s = 12 mm, is modelled with misalignment to see how this affects the relation between the stress concentration factor in the root and the factor in the toe. When the misalignment is larger than 1.36 mm, the stress concentration factor in the toe becomes larger then the one in the root.*

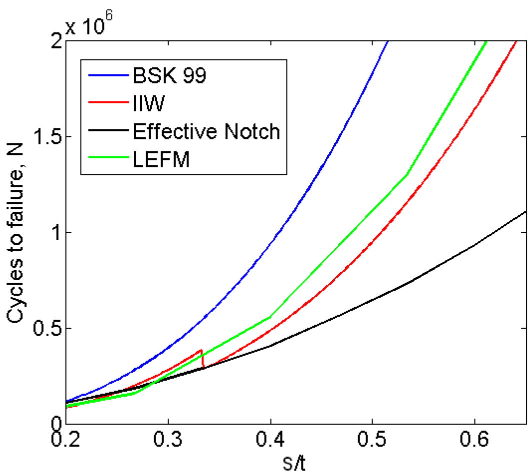
Axial misalignment is not the only misalignment possible for this type of cruciform joint. Also an angular misalignment could be present. This is not included in the model, but this will also make the stress concentration factor higher in the toe because of an increasing bending effect. If an angular misalignment would be included, the limit in Figure 36, where the crack initiation moves from the root to the toe, would move to a lower value of the axial misalignment.

5.2 Load-carrying root crack

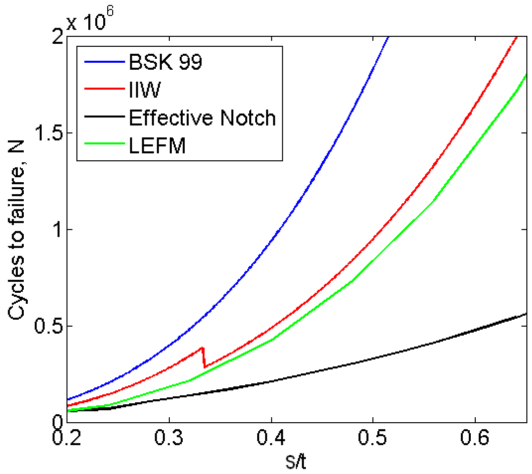
In this section it is assumed that the crack initiates in the weld root. Figure 37 shows how the estimated lifetime is affected by the weld throat thickness for the three different plate thicknesses.



(a) *t = 15 mm*



(b) *t = 30 mm*



(c) *t = 50 mm*

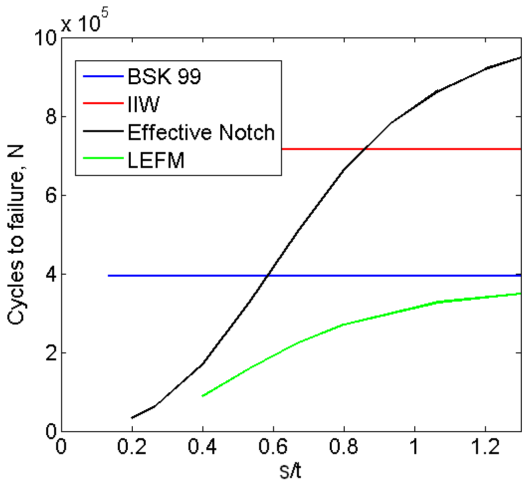
Figure 37 *Number of cycles to failure is plotted, for the four different calculation methods, against the quotient of weld throat thickness, s, and plate thickness, t. The nominal stress applied is 60 MPa. BSK 99 and IIW gives the same estimation of the lifetime for all plate thicknesses, while the lifetime decreases when the thickness is increased for effective notch and LEFM. Especially for the effective notch method, this effect is significant.*

For *t = 15 mm* the four methods are quite well in agreement. When the material thickness is increased the lifetime for the effective notch and for LEFM is decreased while it is held constant for the nominal methods. The nominal methods do not at all take the plate thickness under consideration in this case. The sharp turn in the IIW curve is because of a change in the FAT-value when *s/t* > 1/3. Larger values of *s/t* than present in the figure, is not of interest here since, in that case, the crack will originate at the weld toe. The nominal stress range used to create these plots is *60 MPa.*

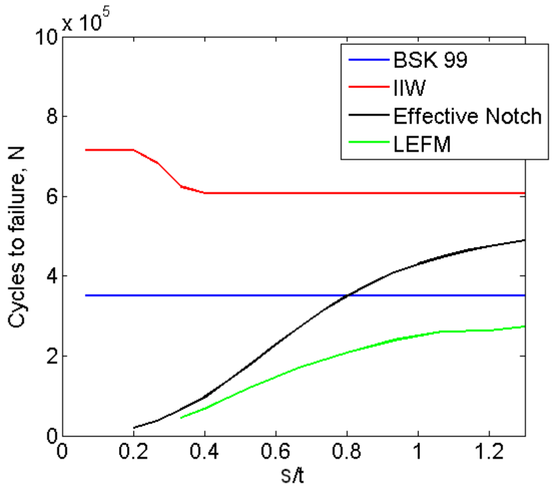
Effective notch is the method that is most dependent on the plate thickness. LEFM has a small dependency of the thickness while the nominal methods are not affected at all of this. The curves for the different methods have very similar shapes, they are all varying in the same manner as the weld size is varied. LEFM should be best for this joint since here one of the obstacles with this method is not present, the crack initiation length. Therefore the results for the LEFM method should be reliable for this structure.

5.3 Load-carrying toe crack

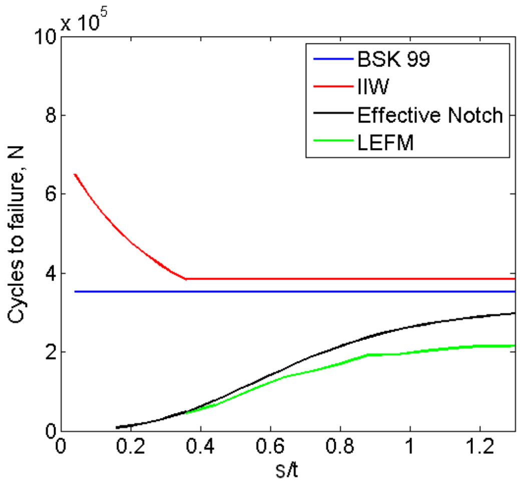
Here is it assumed that the crack is originated in the weld toe. Results from calculations are presented in Figure 38. Results for *s/t* < 0.6 is not really of interest here since then the crack would be initiated in the weld root, and results for *s/t* > 1.2 is not a realistic size of the weld. It is too costly to make a weld of that size, especially for large plate thicknesses.



(a) *t = 15 mm*



(b) *t = 30 mm*



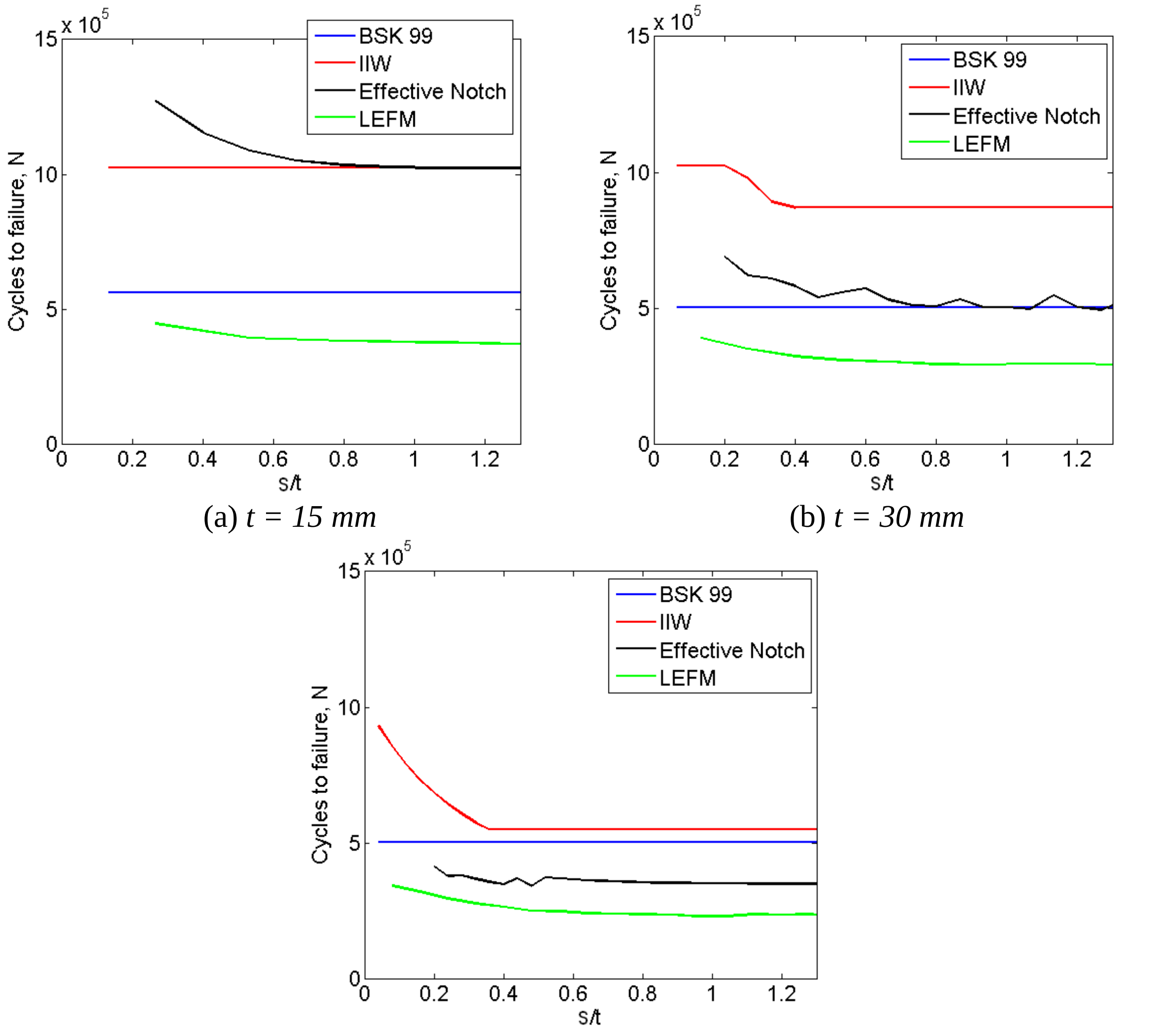
(c) *t = 50 mm*

Figure 38 *The number of cycles to failure is plotted against the quotient a/t for the load-carrying joint where the crack is initiated in the weld toe. The nominal stress applied is 130 MPa. Results for s/t<0.6 is not really of interest here since then the crack would be initiated in the root.*

In the interval of interest for *s/t* the nominal methods are not dependent of *s*. The curves are horizontal in the plots. The estimation of lifetime is though dependent of the thickness, the curves are not on the same level in all three plots (a), (b) and (c). Effective notch and LEFM gives the same shape of the curve but effective notch has a much steeper curve, it is more independent of the weld throat thickness. Effective notch is also the method that gives the largest difference when the plate thickness is increased. LEFM is the most conservative method here, for all plate thicknesses.

5.4 Non load-carrying

For a non load-carrying cruciform joint, the crack initiation point is in general located at the weld toe. Results from calculations are shown in Figure 39. None of the methods gives a large difference in the estimated lifetime when the weld throat thickness is changed. This is not surprising since the weld is not load-carrying. So the size of the weld does not have so much impact on the force distribution in the structure.



(a) *t = 15 mm*

(b) *t = 30 mm*

(c) *t = 50 mm*

Figure 39 *For the non load-carrying joint the number of cycles to failure is plotted versus the quotient s/t. The nominal stress range applied is 130 MPa.*

The nominal stress range applied for all the plate thicknesses are *130 MPa*. The lifetime is, just as expected, decreasing when the thickness is increasing, and as for the other joint types, effective notch is the method that is most dependent on the plate thickness.

LEFM and effective notch are more conservative than the nominal methods for *t = 50 mm*. For toe cracks LEFM is more conservative than the other methods. Those estimations are also very uncertain since a fictitious crack has to be initiated in the toe, and the length of this crack has a great impact on the result. Much of the lifetime is consumed when the crack length is below this initial crack and this lifetime is completely disregarded by the LEFM method in those cases.

6 Fatigue tests

6.1 The Specimen

Fatigue tests are performed for verification of the calculated results. The geometry used can be seen in Figure 40. The fatigue test is performed with two different load cases on the same joint, a non load-carrying (F_1) and a load-carrying joint (F_2), respectively. Two different weld sizes are used for the load-carrying case, to obtain one case where the crack is supposed to start at the weld root, and one where it is supposed to propagate from the weld toe. These values are *s = 12 mm* and *s = 6 mm*, respectively. For the non load-carrying joint the value is, *s = 12 mm*. For each joint type, 10 specimens are manufactured and tested.

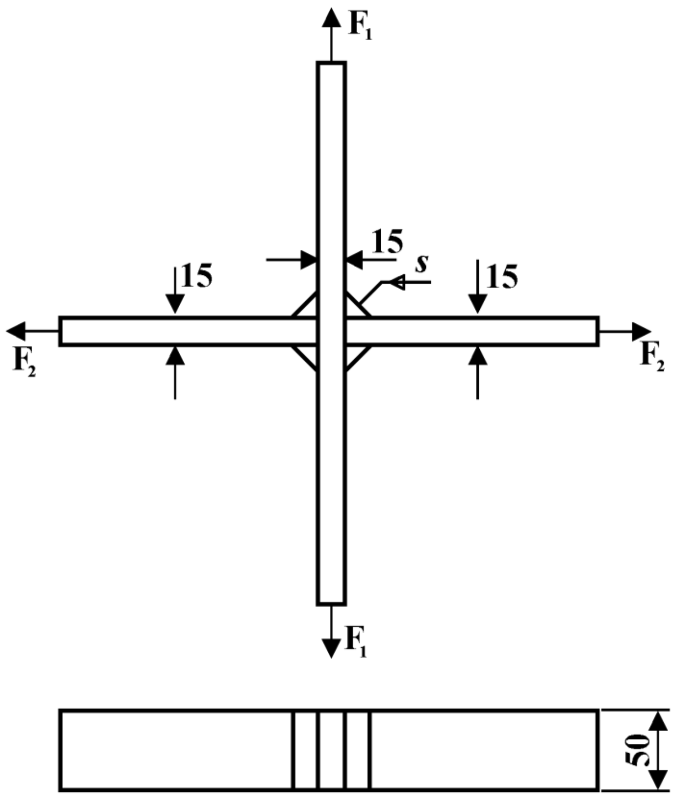


Figure 40 *The cruciform joint used for fatigue tests.*

The welding process is manually MAG, and the material used for the specimens are non-alloyed, as rolled steel bars, 1312-00 in Swedish standard. Mechanical properties at 20ºC can be seen in Table 5.

Table 5 *Mechanical properties for the steel bars used in the test specimens.*

R_{eL} [N/mm^2]	R_{eH} [N/mm^2]	R_m [N/mm^2]	A_5 [%]
220	240	360-460	25

To be certain of always getting a complete positive tensile stress in the joint, a base load of *10 kN* is applied to the specimen. The minimum value in the load cycles are *10 kN* and then the ΔF are applied above this value. This is to reduce the effect of potential residual stresses induced during the welding process.

Figure 41 *The specimen is attached to the test machine. This is a non load-carrying joint.*

6.2 Results

The specimens are applied to a sinusoidal constant amplitude load with 10 Hz, and the number of cycles is counted before the structure breaks into two parts. All the results can be seen in Table 11 in Appendix D.

6.2.1 Non Load-Carrying Joint

For the non load-carrying joint, all the failures started, as expected, at the weld toe. In Figure 42 the experimental results for this joint is plotted. The mean curve and the characteristic curve, 2.3 % failure probability, are obtained according to the recommendations of IIW [5]. The *m-value*, see equation 2:5, determining the slope of the curve is here $m = 3.46$. This is a bit higher than the standard value $m = 3$. The characteristic stress range for $2 \cdot 10^6$ cycles is *77.3 MPa*, which is very close to IIW's FAT-value of 80 for this joint.

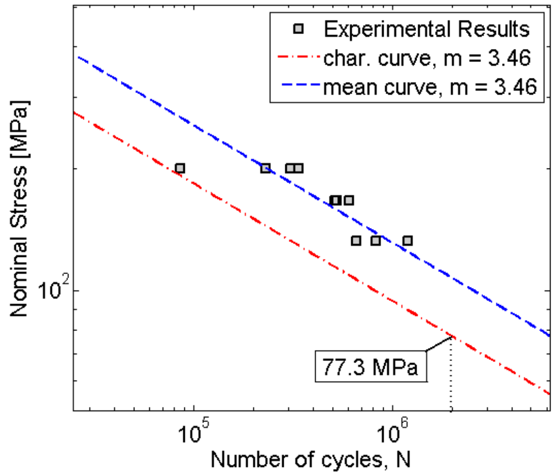


Figure 42 *Experimental results for the non load-carrying joint. The slope of the curve, m, is determined by linear regression and log C, in equation 2:5, is determined by statistics according to the recommendations from IIW. [5]*

Barsoum et al [22], Martinsson [8] and Barsoum et al (LOST) [23] has performed fatigue tests similar to the one performed here. To verify that the experimental results obtained here are reasonable, comparisons to their results are made. In Figure 43 their results are plotted in the same figure as the experimental results obtained here. In Figure 43 (a) the nominal stress range is plotted versus the number of cycles to failure. Also the curves for 50 % failure probability is plotted according to IIW standard and BSK 99 standard, respectively. The curve for IIW seems reasonable since approximately half the number of the dots are plotted below the curve and the other half above the curve. The curve for BSK 99 is too low for this particular joint. It should be noted that in the BSK 99 standard there are no obvious choice of *C*-value for this joint and perhaps this is an explanation to the poor prediction in this case.

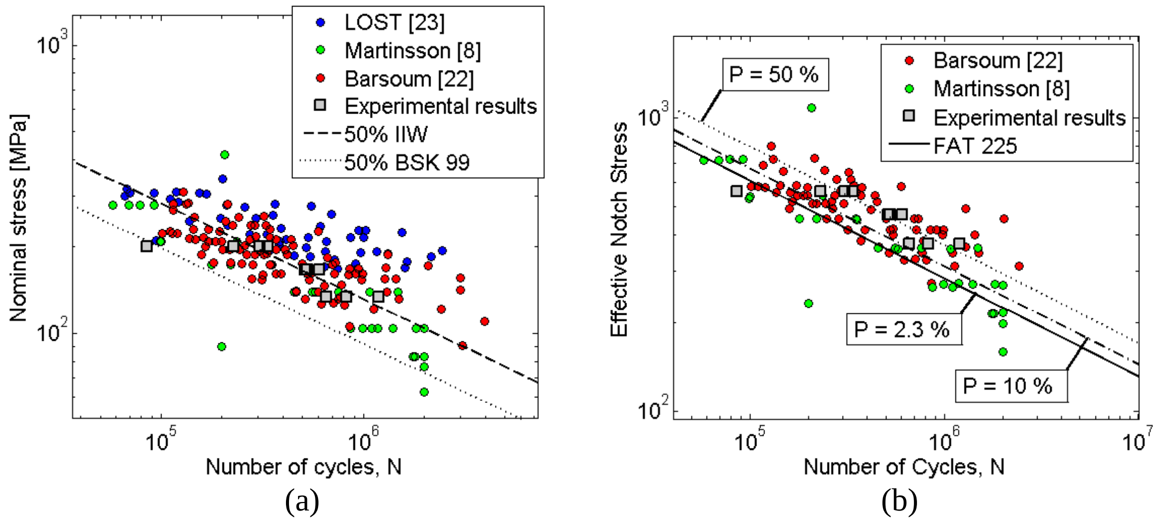


Figure 43 *The experimental results are compared to previously done fatigue tests. In (a) the nominal stress range is on the y-axis, and the curves for 50 % failure probability are plotted for IIW standard and BSK 99 standard, respectively. In (b) the effective notch stress range is on the y-axis, and the curves for different failure probabilities are included in the figure.*

In Figure 43 (b) the effective notch stress range is on the y-axis. Two of the previously done experiments also had an effective notch analysis of their geometry. The experimental results obtained in this thesis are in well agreement with the previous results obtained by others. It seems though as the FAT 225 curve is a bit too high here. A lot more dots are located under the 50 % failure probability curve than above it.

One of the test specimens falls out from the others since it failed already after 85636 cycles. The others, applied to the same stress range, had a longer endurance limit. When this specimen was investigated it showed that the specimen was not attached centralized in the test machine. To see how this could have affected the result, a 3D model of the structure is made. The model is made according to the conventions of the effective notch method. Notches are rounded by a radius of *1 mm* and the weld is modelled with a 45º angle. The specimen is placed in the test machine and fasted by circular plates. In the 3D model this is modelled by applying a displacement on the nodes located at the surface of the model inside the circular pattern. The circular pattern is moved *3 mm* from the centre on both sides. In Figure 45 the largest principal stress is plotted along the path defined in Figure 44. The difference in stress on the two sides of the structure is no more than 1.3 %. This is not enough to explain the big experimental difference in lifetime. Perhaps the boundary conditions applied are not modelling the reality or perhaps this particular specimen had some other defect decreasing the lifetime.

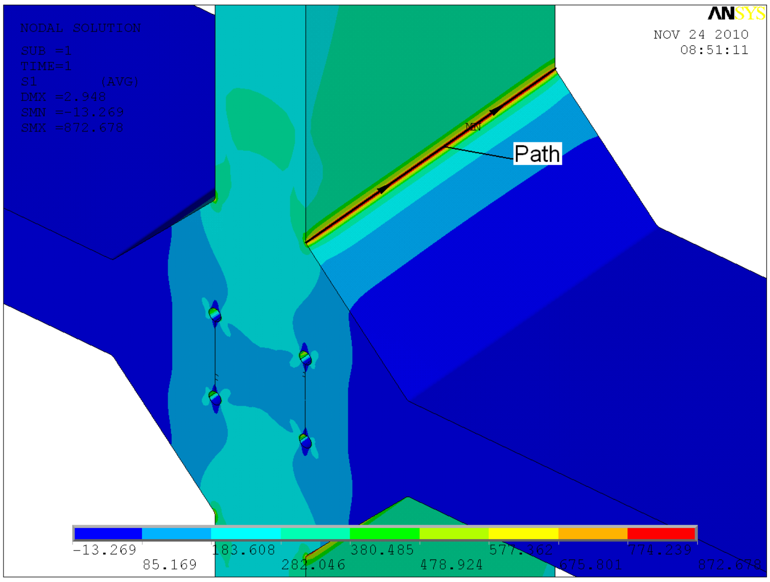


Figure 44 *The 3D modelled is made to see how a decentralized attachment in the test machine affects the result. The stress is measured along the path marked in the picture.*

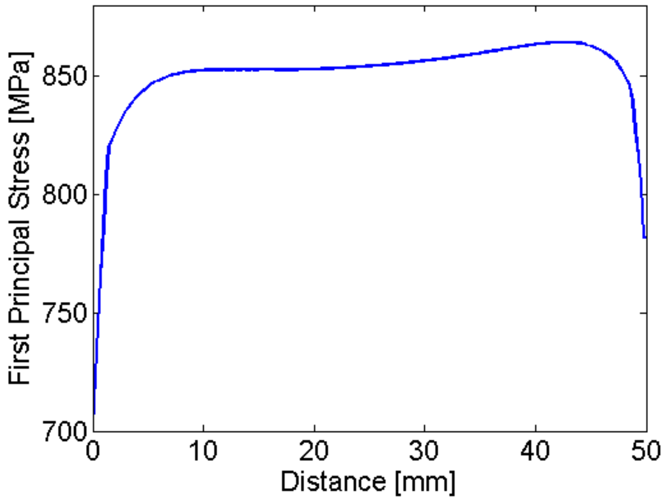


Figure 45 *The largest principal stress varies along the path in* **Figure 44**. *The non symmetrical difference in the stress is according to the non symmetrical fastening in the test machine. The difference is approximately 1.3 %.*

6.2.2 Load-Carrying Joint, Large Weld Size

The load-carrying joints with the large weld size were expected to fail from the toe, see section 5.1. For all the tests performed here, the final failure occurred from the root. In some of the tests a crack is visible also in the toe. So, a crack is initiated both in the toe and in the root, but the final failures have come from the root. In Figure 46 the results can be seen in a plot with the nominal stress on the y-axis and the experimental lifetime on the x-axis.

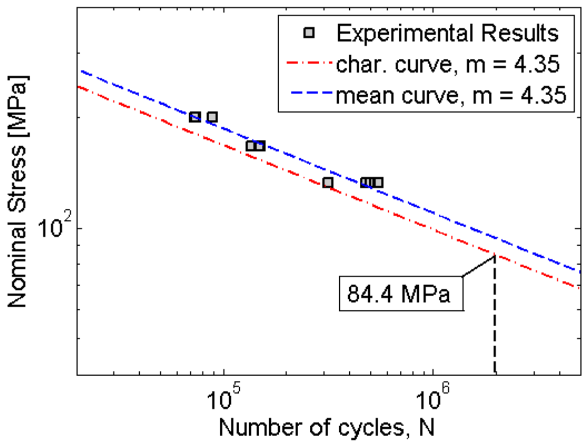


Figure 46 *Experimental results for the load-carrying cruciform joint with the big weld size. The slope, m, is determined by linear regression and log C, in equation 2:5, is determined by statistics according to the recommendations from IIW [5].*

The characteristic stress range for $2 \cdot 10^6$ cycles is here *84.4 MPa*. This is higher than the FAT-values given by both BSK 99 (*C = 56 MPa*) and IIW (*FAT = 71 MPa*). The high value is a result from the flat curve, $m = 4.35$ is quite much higher than the standard value $m = 3$.

A crack is initiated in the toe on the specimens even though the final failure occurred from the root. This could strengthen the discussion in section 5.1, about the crack initiation point for the LEFM analysis when two cracks are propagating simultaneously. It is not self-evident where the final failure will occur for this joint. The effective notch method is predicting a failure from the root and in Figure 47 the effective notch stress is plotted versus the experimental lifetime. Curves for FAT 225 are plotted for different failure probabilities. For the results to correspond to FAT 225 one of the ten dots should lie under the 10 % -curve. None of the dots are below this curve, but on the other hand only 3 of the 10 dots are located above the 50 %-curve.

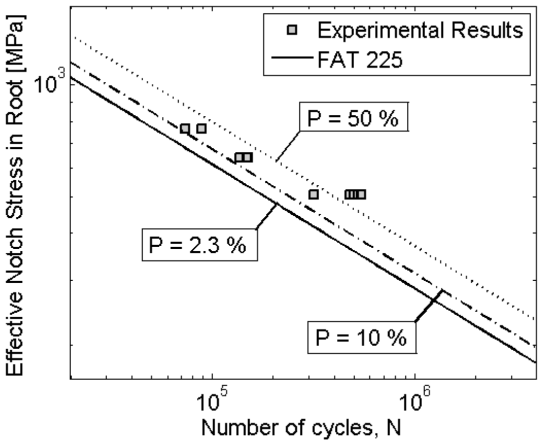


Figure 47 *The effective notch stress is plotted versus the experimental lifetime. Curves for FAT 225 are plotted for different failure probabilities. For the results to correspond to FAT 225 one of the ten dots should lie under the 10 % -curve.*

6.2.3 Load-Carrying Joint, Small Weld Size

For the load-carrying joint with the small weld size, all the failures started, as expected, at the weld root. In Figure 428 the experimental results for this joint is plotted. The mean curve and the characteristic curve are obtained according to the recommendations of IIW [5]. The *m-value*, see equation 2:5, determining the slope of the curve is here $m = 4.10$. This is a bit higher than the standard value $m = 3$. The characteristic stress range for $2 \cdot 10^6$ cycles is *41.4 MPa*, compared to the IIW value of *36 MPa* and the BSK 99 value of *56 MPa*.

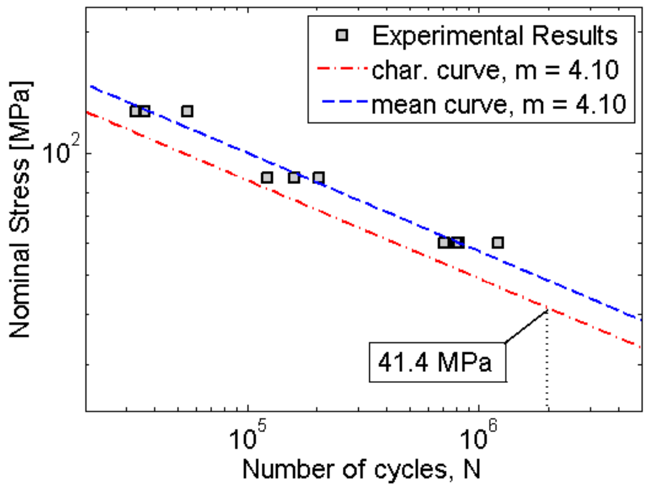


Figure 48 *Experimental results for the load-carrying cruciform joint with the small weld size. The slope, m, is determined by linear regression and log C, in equation 2:5, is determined by statistics according to the recommendations from IIW [5].*

The nominal stresses used in the tests can, by the stress concentration factor, be converted to the effective notch stress. In Figure 49 the effective notch stress is plotted versus the lifetime. The FAT 225-curve for different failure probabilities is included in the plot. For the FAT-value to agree with the results, one of the ten specimens should lie below the 10 % -curve and the other ones should lie above it. Here five of the specimens are located below and five are located above the curve. So, FAT 225 is too high according to the test results.

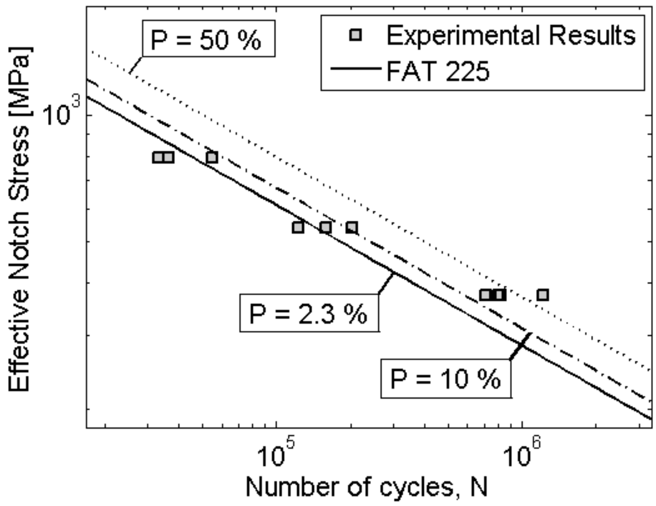


Figure 49 *The effective notch stress is plotted versus the lifetime. Curves for FAT 225 are plotted for different failure probabilities. For the FAT-value to agree with the results, one of the ten specimens should lie below the 10 % -curve and the other ones should lie above it.*

6.3 Strain Gauge

To one of the load carrying test specimen with the large weld size three strain gauges are attached to measure the actual stress in the structure. The gauges are attached ca *5 mm* from the edge of the steel bar, see Figure 50.

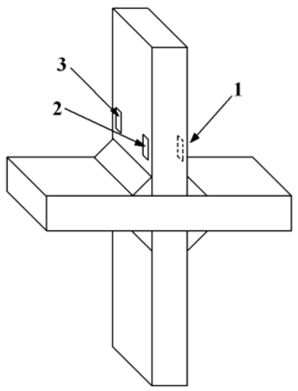


Figure 50 *Three strain gauges are attached to a test specimen in this way. They are located ca 5 mm from the edge of the steel bar.*

The load case for this particular joint is, *F = 10 – 110 kN*. The nominal strain should therefore be:

$$\varepsilon = \frac{\sigma}{E} = \frac{F}{E \cdot A} = \frac{F}{2.1 \cdot 10^{11} \cdot (0.015 \cdot 0.050)} \qquad \textbf{(6:1)}$$

where A is the cross section area. Calculated for the force applied this gives, $\varepsilon = 63.5 \cdot 10^{-6} - 698.4 \cdot 10^{-6}$. The range is, $\Delta\varepsilon = 635 \cdot 10^{-6}$ and the mean strain is, $\varepsilon_{mean} = 381 \cdot 10^{-6}$.

Those calculated values should be compared to the measures values, see Table 6 obtained from Figure 51.

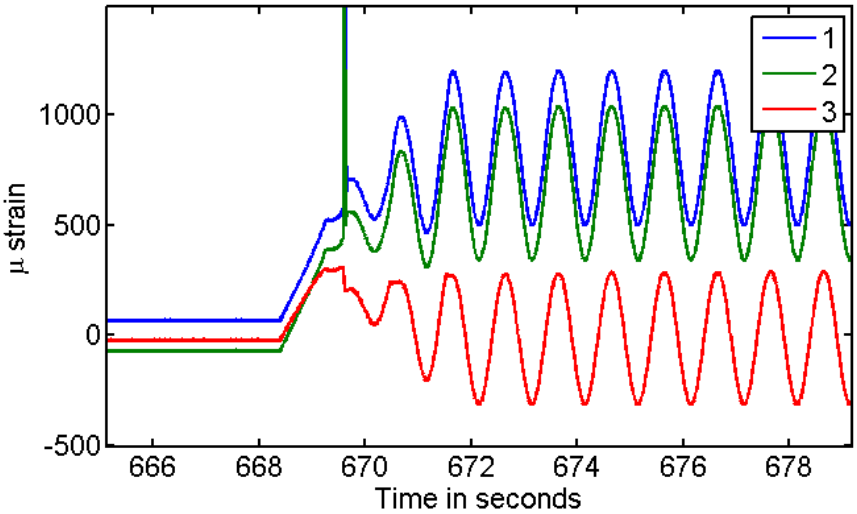


Figure 51 *For the three strain gauges the strain is measured in time. The unit on the y-axis is µ strain (strain ·10*$^{-6}$*).*

Table 6 *These values are received from* **Figure 51**. *The ranges are in good agreement with the theory. The mean values have a big deviation.*

Time	Gauge 1	Gauge 2	Gauge 3
start	68	-70	-18
672.7 s	1200	1046	285
673.1 s	506	346	-306
range	694	688	581
ε_{mean}	853	702	-6

The strain ranges are in good agreement with the nominal values, but the mean values have big deviations. This could be caused by many different things and it is definitely worth more investigations, but these investigations are beyond the scope of this thesis. Here it is just stated what the causes could be. It could be misalignments or non centralization in the specimen. The test machine could apply the loads biased or the specimen could slip when it is already fastened in the test machine.

The deviations in the mean values question the accuracy in the test results. According to this, the stresses are close to the yield strength of the material. Therefore it would be very interesting to know what are causing this deviation.

7 Comparison of Experimental Results and Calculated Results

7.1 Calculated Results

For all the three cases that were fatigue tested in the previous section, calculations and analyses are made with the nominal method, effective notch method and LEFM method. For the load-carrying joint with the large weld size the crack originated from the weld root even though it was expected to start from the weld toe. Therefore calculations for this joint is performed both when the crack is assumed to propagate from the root and when it is assumed to propagate from the toe. The results from analyses with the four different methods for the three different joint types can be seen in the tables below. In Table 7, Table 8 and Table 9 results for the non load-carrying joint, the load-carrying joint with the large weld size and the load-carrying joint with the small weld size can be found, respectively.

Table 7 *Calculated results for the non load-carrying joint.*

ΔF [kN]	$\Delta\sigma_{nom}$ [MPa]	*s [mm]*	*C*	*FAT*	$K_{t,toe}$	N_{toe}			
						BSK 99	IIW	effective notch	LEFM
150	200	12	56	80	2.80	108 421	281 216	284 999	104 683
125	167	12	56	80	2.80	186 233	483 037	489 535	179 810
100	133	12	56	80	2.80	368 680	956 258	969 122	355 970

Table 8 *Calculated results for the load-carrying joint with the large weld size, s= 12 mm. The failure occurred in the root even though it was supposed to occur in the toe. Therefore results for both a root and a toe failure are calculated. Results for a toe failure can be found in the parentheses.*

ΔF [kN]	*C*	*FAT*	$K_{t,root}$ ($K_{t,toe}$)	N_{root} (N_{toe})			
				BSK 99	IIW	effective notch	LEFM
150	56	71	3.821 (3.252)	202 564 (108 421)	104 963 (196 583)	112 147 (181 872)	143 037 (74 270)
125	56	71	3.821 (3.252)	347 939 (186 233)	180 292 (337 665)	192 631 (312 397)	245 691 (127 572)
100	56	71	3.821 (3.252)	688 807 (368 680)	356 921 (668 467)	381 348 (618 445)	486 389 (252 551)

Table 9 *Calculated results for the load-carrying joint with the small weld size, s = 6 mm.*

ΔF [kN]	$\Delta\sigma_{nom}$ [MPa]	*s [mm]*	*C*	*FAT*	$K_{t,root}$	N_{root}			
						BSK 99	IIW	effective notch	LEFM
95	127	6	56	36	6.23	98 890	51 242	101 022	82 285
65	87	6	56	36	6.23	307 613	159 397	314 244	255 962
45	60	6	56	36	6.23	937 797	485 941	958 013	780 333

For toe failures BSK 99 gives a much more conservative result then IIW. This is the usual case when these to standards are compared [1]. For root cracks, IIW gives a more conservative result then BSK 99. It should though be noted that for this particular specimen the choice of the *C-value* in the BSK 99 standard is very uncertain since no geometry matches this one perfectly. In the IIW the geometry is directly presented.

For the load-carrying joint with the large weld size, BSK 99 and LEFM predict a toe failure, while IIW and effective notch predict a root failure.

7.2 Comparison

In this section the calculated results from the previous section is compared to the experimental results obtained from the fatigue tests. In Figure 52 the calculated lifetime is plotted against the actual experimental result for the non load-carrying joint. The different methods are presented by dots with different shape and colour. Optimal would be if all the dots lie on the straight line, y = x.

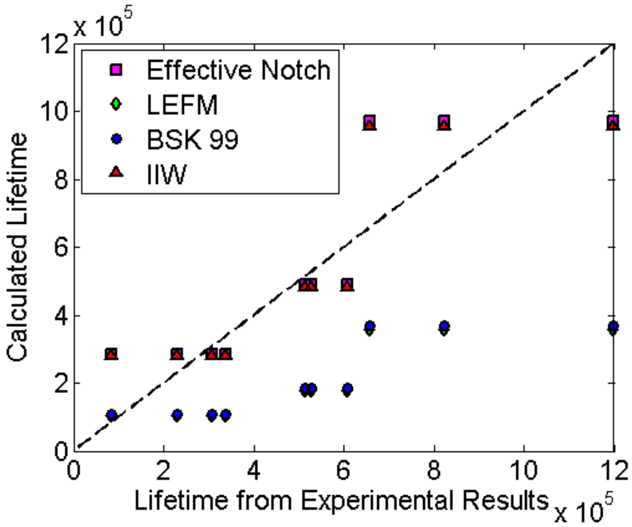
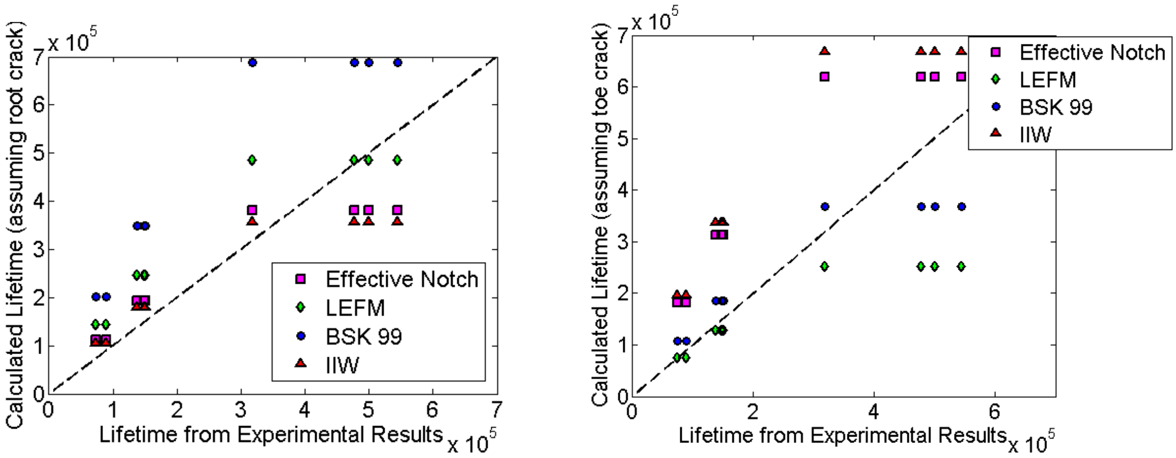


Figure 52 *Non load-carrying joint. BSK 99 and LEFM give estimations similar to each others and IIW and effective notch give approximately the same results.*

For the non load-carrying joint the results from the effective notch method and IIW are similar to each other and the result obtained by LEFM and BSK 99 are approximately equal. IIW and effective notch gives a better estimation for this joint.

It is not surprising that IIW gives a better result than BSK 99 since IIW has a geometry that completely represents this case. BSK 99 does not have any obvious choice of the *C*-value. LEFM underestimates the lifetime for this joint. Perhaps this is because of the fictitiously initiated crack in the toe. For the LEFM method the initiation phase is completely ignored and it is a fact that much of the lifetime is consumed while the crack still is very small.

For the load-carrying joint with the large weld size, all failures occurred at the weld root even though it was expected for this joint to fail from the toe. Therefore the endurance limit is estimated both for a root crack and for a toe crack for all the four methods. In Figure 53 (a) the failure is assumed to start at the toe and in Figure 53 (b) the failure is assumed to start at the root.



(a) *(b)*

Figure 53 *Load-carrying joint with the large weld size. In (a) the calculations are based on the assumption that the crack grows from the root and in (b) that the crack grows from the toe. Effective notch and IIW are most conservative for a root crack, while LEFM and BSK 99 are predicting a toe crack.*

BSK 99 and LEFM predict a toe failure while IIW and effective notch predicts a root failure. IIW and effective notch got right in this case but according to section 5.1 the failure should actually have started in the toe. This could be because of a small weld penetration for the test specimen. In Gurney [19] it is not stated for what weld penetration it is valid that the crack should start in the toe, and if the penetration is small the lifetime will decrease. It could also be because of some boundary effect. In many experimental test, similar to this one performed here, the specimens are first welded in large steel plates and then cut into the right thickness [8] [12]. Here the thickness was already *50 mm* and each specimen was welded separately. When welding, the starts and the stops are critical places and when each specimen is welded separately each weld gets very short and many starts and stops are required. The weld could have a not so good quality at the edges and this could cause a crack to start there and grow faster than expected. It should also be noted that many of the specimens has a small crack initiated in the weld toe even though the final failure occurred in the root.

In the load-carrying joint with the small weld size, all the failures occurred in the weld root, as expected. In Figure 54 the comparison of the calculated result and the experimental result can be seen. Perhaps LEFM can be considered to be the best method for this joint. This is not surprising since when the crack originates from the root no fictitious crack has to be initiated with this method. So the LEFM method should be best for root cracks. The effective notch method overestimates the lifetime for this joint. This is not good, an underestimation is better because in that case the method is still conservative.

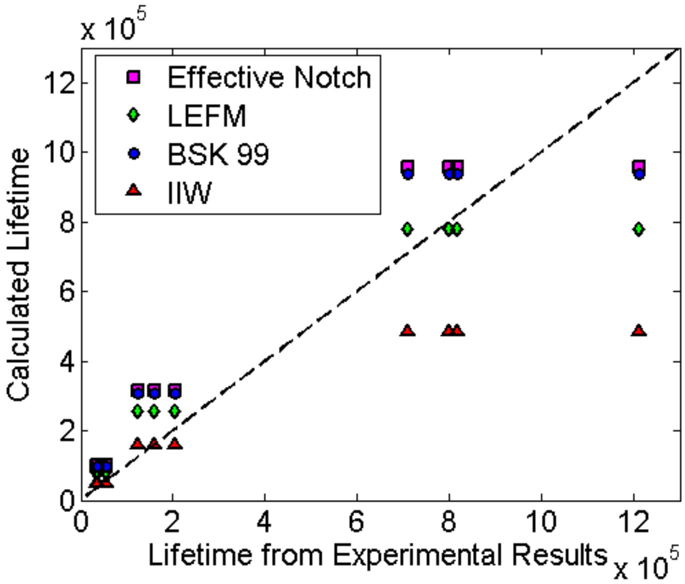


Figure 54 *Load-carrying joint with small weld size. IIW underestimates the lifetime but the other methods are quite good. Perhaps LEFM is the best method for this particular joint.*

8 Discussion and Conclusions

Generally, there can be large differences in the lifetime estimations for the different methods. Small changes in the input parameters can give large changes in the results. The scatter, in both the experimental and calculated results, makes it very hard to validate and rank the different methods.

In Figure 6 the methods for fatigue assessment are compared to each other. The results obtained from the experiment here can not totally verify this ranking of the methods. First of all, the LEFM method is not always the most accurate one. LEFM seems, not surprisingly, to be more accurate for root cracks, where the fictitious crack initiation is not needed, than it does for toe cracks. For the non load-carrying joint, both the effective notch method and IIW gives better results than LEFM, this is not in agreement with the figure. It should also be noted that the time consumption for the calculations with the LEFM method is huge compared to the time needed for the effective notch method, and this is for a very simple structure. One can only imagine the time needed for a more complex model. Figure 6 is not generally true, it is not certain that the effective notch method and the LEFM method always are more accurate than the nominal method. Radaj et al [17] think that a nominal method should be used as a first guideline for dimensioning from which further design improvements, based on local approaches, may originate. By “local approaches” they are referring to methods and variations of effective notch and LEFM method.

The effective notch method insinuates that the critical weld throat thickness, $s_{critical}$, should be $1.2 \cdot t$. This is an unreasonable high value and is not in agreement with any experimental test performed. One contribution to this erroneous value could be that only 2D models are made. Perhaps some boundary effect not included in a 2D model could influence this value. The fact that the misalignment is not included in an evaluation with the effective notch method could be another contribution to this strange value. But if the effective notch method is applied to a more complex structure it would be very complicated to include all the possible misalignments into the model. The time consumption for an analysis would increase drastically.

The effective notch method gives quite good lifetime estimations for the joints used in this thesis, but the method tends overestimate the lifetime rather then underestimate it. This makes the method not conservative, and dangerous to use as guidance. Perhaps some safety factor should be included to cover things such as misalignment. Or the FAT value should be lowered, at least for some geometries. But if this is done the big advantage with the effective notch method is lost. The simplicity of only needing one FAT value and not have to worry about what factors to use is what is so appealing with the effective notch method.

From the results one can see that the effective notch method gives more accurate results for toe cracks than for root cracks. It can be concluded that the shape of the root, oval or keyhole, does not have a great impact on the result. Either of the shapes introduces a hole in the structure that really is not there. This gives unnatural stresses in that region, especially for thin structures, and the results for root crack are less reliable that for toe cracks.

Results from the experiment for the non load-carrying joint are compared to experiments made by others. No large deviation from there results can be seen and therefore it can be assumed that the experimental test performed in this thesis is having an adequate good quality. Therefore it is strange that the load-carrying joint with the large weld size failed from the root instead from the toe, as expected. The weld penetration was perhaps unusually small here causing the joint to be weaker in the root. Or some boundary effect from the fact that the specimens are not cut out from wide steel plate to the right width, but rather manufactured directly from material of the right width, are affecting the tests. Other similar experimental fatigue tests performed are using larger plates and then cutting them to the correct width. This makes the effect of the welding start and stop point less significant. But on the other hand are real structures, actually used for service, never cut to the right width afterwards, so why should the test specimen be manufactured in that way?

Even though the effective notch method has some disadvantages it also has a big benefit compared to the nominal method. In the nominal method it is necessary that the test specimen correspond in respect of all essential influencing parameters to an already determined FAT class. This is not needed for the effective notch method. This method is much more flexible for complex and unconventional designs. Therefore the effective notch method could be a useful tool for optimization and innovation. Effective notch also has a large advantage compared to the LEFM method. Even though the LEFM method sometimes, (but not always, as stated above) are more accurate it has a downside with all the parameters. Many variables as the constants in Paris' law, crack initiation length, crack path and crack shapes, has to be set. The effective notch method does not have all these parameters. Also, the microscopic crack phase is completely neglected in the LEFM method, even though it is a fact that much of the lifetime is consumed when the crack is small.

A definite answer for when and where the effective notch method is accurate enough can not be made from this study. Due to the scatter it is hard to validate a method for fatigue assessment, and more experimental results are needed to draw a definite conclusion. But one can conclude that the effective notch method has a tendency to overestimate the fatigue strength of the structure. At least that is the case for all the geometries used in this thesis. Perhaps the FAT 225 is too high and should be lowered. This action has already been proposed for certain geometries [6]. Perhaps it is even wrong to compare all structures and geometries to one FAT class. But, as mentioned above, more tests are needed, and it is definitely a method worth more research and development. Even though the difficulties with the validation makes it problematic to use this method for absolute statements, is can be a useful tool for design optimization. What fatigue strength can be expected with a new design variant in relation to a well-proven former design? Methods as effective notch and LEFM can also be useful for backtracking. Why did a structure fracture?

8.1 Limitations and Further Work

In this thesis many assumptions and limitations are used to obtain the results. Many of them are mention in the text where it is relevant. Some of these limitations could though be worth looking closer into. For example only constant amplitude cyclic loads are considered. Since machines in service almost always are applied to variable cyclic loads, it would be very interesting to extend this study to variable amplitude load.

Another thing than would be interesting to dig deeper into is the complexity of the geometry. It would be desired to do this analysis on more complex structures to see if the same tendencies can be seen even then. It could also be interesting to look at different stress ratios, R. In the effective notch method no effect of the stress ratio is included, but one can imagine that this can have some effect of the lifetime. It should also be noted that the effective notch method is not applicable if there is a significant stress component parallel to the weld. This is a weakness of the method.

From the figures in section 5 it can be seen that the effective notch method is the method that is most affected by the steel thickness. For larger thicknesses effective notch gets more conservative, and since this is one of the problems with the method it would be interesting to investigate thicker steel plates. Perhaps the effective notch method is more conservative and gives a more accurate result for thicker materials.

8.2 Conclusions

The analysis and discussion can be concluded into some separate statements.

- The effective notch method is better for toe failures than for root failures where the fictitious radius gives unnatural stresses. Effective notch also tends overestimate the lifetime. This is dangerous since this gives non conservative results.

- LEFM has a big problem with all the parameters that has to be set. Despite this, LEFM is the most accurate method for root cracks. One downside is the huge working effort needed.

- Effective notch is not verified enough to use for absolute statements, but it is definitely a useful tool for design optimization when different design variations are compared to each other.

- It is hard to validate and verify the effective notch method due to the scatter in test results. More research and experimental tests needs to be performed and it is definitely a method interesting enough to be worth the effort. Especially more research for more complex structures are needed since not so much is done here.

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Appendix A

Table 10 *Different crack increments affects the estimated lifetime for different thicknesses of the material. For large thicknesses the smallest increments results in too long calculation time. Using a too long increment in relation to the material thickness result in very little iteration before the crack reaches half the thickness of the material, the stop criterion. The stop criterion can also be fulfilled way too early if the next increment does not fit in the remaining of the material until the crack reaches half the thickness.*

Δa [mm]	Estimated Lifetime, *N*			
	t = 5 mm	*t = 15 mm*	*t = 25 mm*	*t = 35 mm*
0.02	706278	-	-	-
0.03	709451	-	-	-
0.05	714741	516241	-	-
0.1	728189	518716	425932	364059
0.5	676864	524995	430055	383212
1	561700	543421	438431	391383
2	269745	485954	444685	398426
3	-	-	408565	391705
5	-	-	-	319571

Appendix B

For BSK 99 [18], where the crack is assumed to grow from the weld root, the stress through the weld is used for evaluation. The *C-value* for a load-carrying joint with the crack assumed to start in the root is according to BSK 99 *C = 56*.

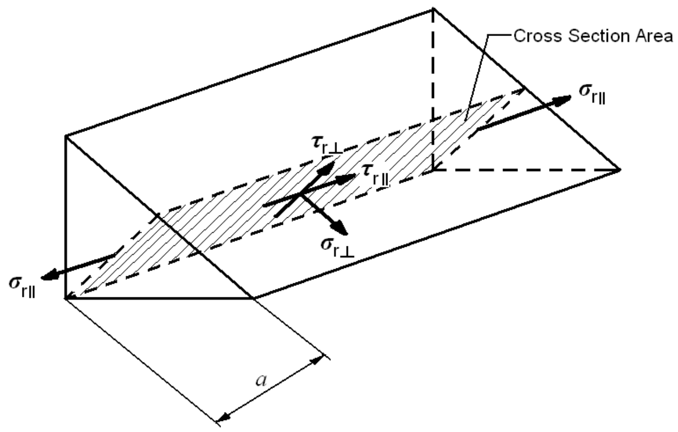

Figure 55 *The stress components in the cross section area through the fillet weld.*

BSK99 states how the stress components are to be calculated through the weld, see Figure 55. For the relationship between all the components this condition has to be fulfilled,

$$\sqrt{\frac{\sigma_{rd\parallel}^2}{f_{rd\parallel}^2}+\frac{\sigma_{rd\perp}^2}{f_{rd\perp}^2}+\frac{\tau_{rd\parallel}^2}{f_{rvd}^2}+\frac{\tau_{rd\perp}^2}{f_{rvd}^2}} \leq 1.10 \qquad \textbf{(B:0:1)}$$

where the *d* in the indexes stands for the dimensioning condition and

$$\sigma_{rd} \leq f_{rd} \qquad \textbf{(B:0:2)}$$

for dimensioning against fatigue failure with pure normal stress, and

$$\tau_{rd} \leq f_{rvd} \qquad \textbf{(B:0:3)}$$

if the stress range consists of only shear stresses and $f_{rvd} = 0.6 f_{rd}$. [18]

For the specimen considered here, a load-carrying cruciform joint, the force situation can be seen in Figure 56.

Figure 56 *The force situation in a load-carrying cruciform welded joint.*

Due to symmetry, half the figure can be considered and the force balance will look like this:

$$\left.\begin{array}{l} \uparrow: \left(\dfrac{\sigma_{r\perp}}{\sqrt{2}} - \dfrac{\tau_{r\perp}}{\sqrt{2}}\right)\cdot s \cdot L = 0 \\ \rightarrow: \left(\dfrac{\sigma_{r\perp}}{\sqrt{2}} + \dfrac{\tau_{r\perp}}{\sqrt{2}}\right)\cdot s \cdot L = \dfrac{F}{2} \end{array}\right\} \Rightarrow \quad \sigma_{r\perp} = \tau_{r\perp} = \frac{F}{2\sqrt{2}\cdot s \cdot L} \qquad \textbf{(B:0:4)}$$

where a is the weld throat thickness and L is the width of the material, here $L = 50\ mm$.

Put this into relation (B:1).

$$\sqrt{\frac{\sigma_{rd\perp}^2}{f_{rd\perp}^2} + \frac{\sigma_{rd\perp}^2}{0.36 f_{rd\perp}^2}} \le 1.10 \;\Rightarrow\; \frac{1.36\sigma_{rd\perp}^2}{0.36 f_{rd\perp}^2} \le 1.21 \qquad \textbf{(B:0:5)}$$

Combine this with equation (7:4) and you get:

$$\frac{1.36}{0.36}\cdot\frac{F^2}{8\cdot s^2 \cdot L^2}\cdot\frac{1}{1.21} \le f_{rd\perp}^2 \quad \Rightarrow \quad 0.624713\frac{F}{s \cdot L} \le f_{rd\perp} \qquad \textbf{(B:0:6)}$$

This value of f_{rd} is then put into equation (2:6) to get a relation between the stresses and the estimated lifetimes. The thickness factor, φ_{dim}, does not have to be taken under consideration here since it only is valid if the crack grows through the material thickness. The fact that we want a probability of failure of 50 % has to be taken under considered, though. To get the correct probability of failure, the FAT-value has to be multiplied by two standard deviations, according to [1] this corresponds to multiply by 1.3. Put this into equation (2:6) and you get:

$$N = 2\cdot 10^6\left(\frac{\varphi_Q \cdot C}{f_{rd}}\right)^3 = 2\cdot 10^6\left(\frac{\varphi_Q \cdot C \cdot s \cdot L}{0.624713\cdot F}\right)^3 \qquad \textbf{(B:0:7)}$$

Appendix C

The load-carrying joint, with the crack propagating from the weld root, has according to IIW standard a FAT-value of 36. The stress used for lifetime estimations are due to IIW not the same as BSK 99 suggests. Instead this equation should be used for the stress:

$$\sigma_w = \frac{F}{\sum (s_w \cdot L)} \qquad \textbf{(C:1)}$$

In this case the stress is shared by two welds, one on each side of the steel plate, with the same value of *s*. So here the equation will look like this,

$$\sigma_w = \frac{F}{\sum (s_w \cdot L)} = \frac{F}{s \cdot L + s \cdot L} = \frac{F}{2sL} \qquad \textbf{(C:2)}$$

Put this into equation (2:6) and you get:

$$N = 2 \cdot 10^6 \left(\frac{\varphi_Q \cdot FAT}{f_{rd}} \right)^3 = 2 \cdot 10^6 \left(\frac{\varphi_Q \cdot FAT \cdot 2 \cdot s \cdot L}{F} \right)^3 \qquad \text{(C:3)}$$

Appendix D

Table 11 *Experimental Results*

Test #	non load/load	root/toe	*s [mm]*	ΔF [kN]	N
1	non load	toe	12	150	85363
2	non load	toe	12	150	336945
3	non load	toe	12	100	1197673
4	non load	toe	12	100	658921
5	non load	toe	12	150	229744
6	non load	toe	12	125	527727
7	non load	toe	12	125	515742
8	non load	toe	12	125	607955
9	non load	toe	12	150	307850
10	non load	toe	12	100	823467
11	load	root	12	150	89170
12	load	root	12	150	73755
13	load	root	12	150	73413
14	load	root	12	125	148072
15	load	root	12	125	137065
16	load	root	12	125	149966
17	load	root	12	100	317796
18	load	root	12	100	477095
19	load	root	12	100	544930
20	load	root	6	95	55064
21	load	root	6	65	159597
22	load	root	6	45	709538
23	load	root	6	45	799496
24	load	root	6	45	817770
25	load	root	6	65	204111
26	load	root	6	65	122922
27	load	root	6	95	33210
28	load	root	6	95	36272
29	load	root	6	45	1212529
30	load	root	12	100	499872