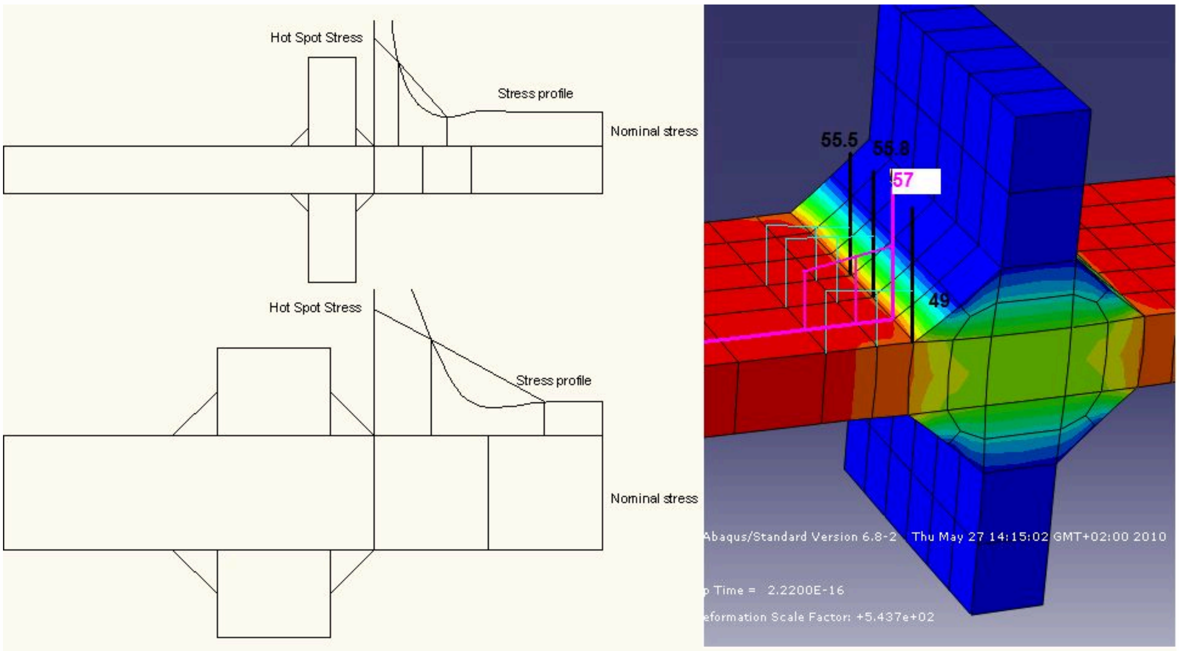

A STUDY OF THE THICKNESS EFFECT IN FATIGUE DESIGN USING THE HOT SPOT STRESS METHOD

Master of Science Thesis in the Master's Programme Structural Engineering and Building Performance Design

GUANYING LI
YIDONG WU

Department of Civil and Environmental Engineering
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Steel Structures
CHALMERS UNIVERSITY OF TECHNOLOGY
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Master's Thesis 2010:103

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Department of Civil and Environmental Engineering
Division of Structural Engineering
Steel Structures
Chalmers University of Technology
SE-412 96 Göteborg
Sweden
Telephone: + 46 (0)31-772 1000

Cover:
Extrapolation method to get the hot spot stress

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ABSTRACT

The thickness effect is the phenomenon that the fatigue strength of a welded connection decreases when the thickness of load carrying plate increases. The thickness effect is observed in details where the fatigue crack initiation takes place at the weld toe.

The hot spot stress approach is a fatigue design approach based on the structural stress obtained from Finite element Analysis (FEA) results. This method is more suitable when calculation of load effects is made using finite element method. Even though the method has been used for more than 20 years, it is yet not fully completed. One problem in this respect is how to account for the thickness effect, a problem which is not included in design codes and recommendations. Instead, the thickness effect conditions for the nominal stress method have been used with the hot spot stress. The thickness effect using the hot spot stress approach for stress determination for welded joints is studied in this thesis.

A database containing fatigue test results with a number of selected welded joints is chosen for the analysis. Different types of modelling technique (according to the International Institute of Welding) have been applied to analyze the selected joints and the models were analysed using the FE program ABAQUS. The structural hot spot stresses are determined according to the IIW and other stress determination methods. Finally the result data obtained from the FEA are compared with the result from fatigue test data.

The results show that the thickness effect correction factors in the hot spot stress approach are the same as the values obtained for the nominal stress approach for two of studied details. For the third detail, the thickness effect correction factor in the hot spot stress approach is larger than that for the nominal stress approach. There is no thickness effect when the so called "1 mm stress approach" is used for the cruciform joint.

Key words: Thickness effect, hot spot stress approach, nominal stress approach, "1 mm stress method", T-S curves, S-N curves, thickness effect correction factor.

Contents

Preface

In this study, the effect of plate thickness on fatigue strength using hot spot stress has been studied. Current code recommendations for the thickness effect by hot spot stress assessment approach are not clear and might be non-conservative. This has made the department of steel structures at Chalmers University of technology interested in an investigation on this subject.

The master thesis has been carried out from February 2010 to June 2010. The project is carried out at the Department of Structural Engineering, Steel Structures, and Chalmers University of Technology, Sweden.

First of all we would like to thank our supervisors Associate Professor Mohammad Al-Emrani and Dr Mustafa Aygül at Chalmers University of Technology. We also send our appreciation to Professor Bo Edlund, who gave us many suggestions during final seminar. Finally, we want to thank our master program coordinators Professor Engström, Björn and Senior Lecturer Plos, Mario for all support during these two years.

Göteborg June 2010

Guanying Li & Yidong Wu

Notations

$\nabla\sigma$	Applied stress range
γ_{Ff}	Partial factor for fatigue loading in EC3
γ_{f}	Partial factor for fatigue loading in IIW
σ_{a}	Stress amplitude
σ_{b}	Shell bending stress
σ_{hs}	Hot spot stress (HSS)
σ_{ln}	Notch stress
σ_{m}	Membrane stress
σ_{mean}	Mean stress
σ_{max}	Maximum stress in stress history
σ_{min}	Minimum stress in stress history
σ_{nl}	Nonlinear stress peak
σ_{nom}	Nominal stress (NS)
σ_{s}	Structural stress

A	Cross section
a	Leg length
b	Width of specimen
F	Axial force
K_t	Stress concentration factor
l	Length of specimen
m	Slop of S-N curve
M	Bending moment
n	Fitting parameter
N	Fatigue life, cycles of failure
N_B	Fatigue life for reference plate thickness
R	Stress ratio
S	Fatigue strength of the joint under consideration
S_B	Fatigue strength of the joint using the basic S-N
ΔS	Stress range, Fatigue strength
ΔS_o	Fatigue strength for reference thickness
t	Main plate thickness
t’	Attachment plate thickness
t_B, t_o	Thickness corresponding to the basic S-N curve.
W	Bending resistance

1 Background and Introduction

1.1 Background

The hot spot stress approach is a fatigue design approach based on the structural stress obtained from Finite element Analysis (FEA) results. This method is more suitable with application of FEA than the nominal stress approach. Even though the method has been used for more than 20 years, it is yet not fully completed. One problem in this respect is how to account for the thickness effect, a problem which is not included in design codes and recommendations. Instead, the thickness effect conditions for the nominal stress method have been used for the same purpose. The thickness effect using the hot spot stress approach for stress determination for welded joints needs to be studied. Although in some codes, it has been mentioned how to consider the thickness effect by hot spot stress approach, it is not clear and seems non-conservative.

1.2 Aim

The aim of the thesis is to propose a formula by mean of which a more accurate estimation of the thickness effect can be made when the hot-spot method is used. The study also aims at providing an understanding for this parameter affect the results derived from FE-analysis.

Also, improve an understanding of how to get a right hot spot stress. How stress distributes along surface from weld toe outwards, and the stress distribution through thickness.

1.3 Method

The following steps describe the methods and procedures used in this thesis:

- Literature survey about fatigue and thickness effect and how this effect is discovered and considered.

- A database of fatigue test with a number of selected welded joints is chosen for the analysis.

- Calculate the thickness correction factors using the nominal stress method.

- FE-models with various complexities and plate thicknesses will be constructed and analyze the fatigue strength data. Different types of modelling technique (according to the international institute of welding), the 3-D solid element model and the 2-D plain strain shell element model will be used to analyze the chosen joints and the models will be built up with ABAQUS.

- The structural hot spot stress will be determined according to the IIW and other stress determination methods.

- Finally the result data obtained from the FEA will be compared with the result from fatigue test data and presented in form of S-N curve.

1.4 Limitations

There are some limitations of this thesis:

- Limited number of joint types are studied in this thesis, detail type b) have not been mentioned.
- FE-models are not close to the reality. The analyzed specimens are assumed to have to material defects and the stress analysis does not consider residual stresses. And same material parameters are assumed for all different specimens. In addition, only linear FE analysis is used, without considering the non-linear effect.
- The hot spot stresses are derived by the IIW recommendation, which may have uncertainty itself. As can be seen from Chapter 6, the extrapolation rule is not so reliable.
- The slopes for the S-N curves are not forced to 3.
- Only two types of elements are used for FE modelling.

2 Literature Survey

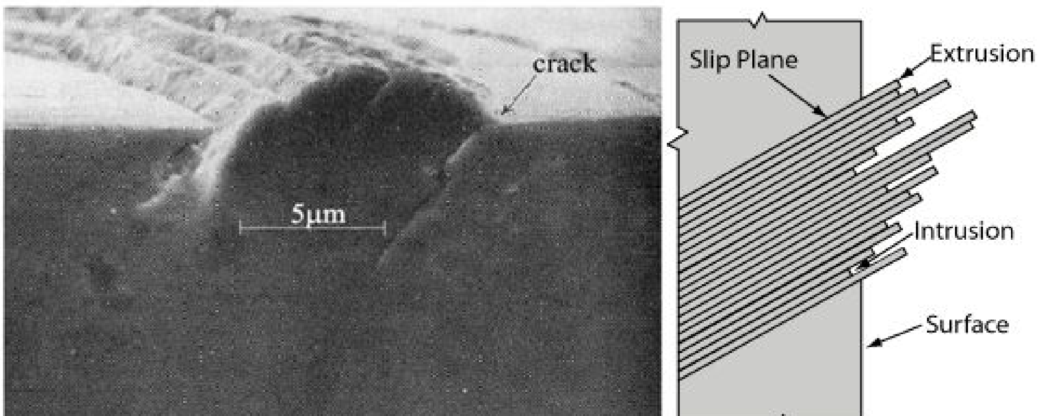
2.1 Fatigue introduction

Fatigue: “the process of progressive localized permanent structural change occurring in a material subjected to conditions that produce fluctuating stresses and strains at some point or points and that may culminate in cracks or complete fracture after a sufficient number of fluctuations”, by ASM (1985) [1].

W. A. J. Albert (Schutz 1996) who published the first article on fatigue found that fatigue is not depending on the overload but the number of repeating loaded cycles which refer to fluctuating loading. There is commonly recognized that a material failure may happen even the maximum stresses are well below the ultimate tensile stress limit. It has long been confirmed that fatigue is one of the primary reasons for the failure of structural components. In fact, 80% to 95% of all structural failures occur through a fatigue mechanism, Adarsh Pun (2001) [2].

The fatigue is defined as the deterioration of a component caused by the crack initiation or by the growth of a crack, new IIW recommendation (2008) [3]. Accordingly, three stages of the progress relevant to fatigue are indicated as follows:

- *Crack initiation:* After several micro-cracks caused by initiation process, one crack becomes dominant and other micro-cracks start to interact as suggested by Miller, Miller K.J. (1987) [4]. The stage that occupy commonly more time in the fatigue life since damage develops slowly in this phase.



Suresh 1991

Figure 2.1 Crack initiation

- *Crack propagation:* the stage that the dominant micro-crack in previous starts to accelerate the local stress field near the crack front when the cross-section decreases. [5]

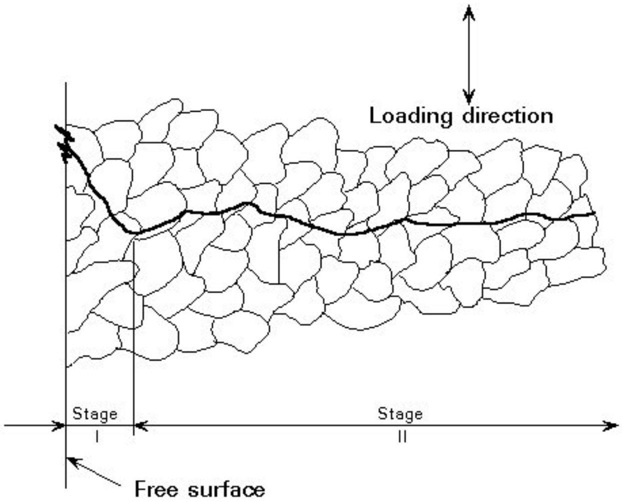


Figure 2.2 Crack initiation and crack propagation (ESDEP Lecture Note)

- *Final sudden fracture:* It is the stage that the rapid propagation rate results in a not enough remaining area to support load.

However, when it comes to welding details, the crack initiation phase is regard as a completed stage. This is mainly because of the defects has already happened in welding heat-affected zone. Therefore, crack propagation and final failure are the only two phases for welded structures.

2.2 Fatigue Actions (Loading)

In fatigue assessment, the stress range influences the fatigue life most. Other influencing factors include load-acting direction, geometry of critical point, weld type as well as residual stress. The fatigue load is expressed as applied stress range, $\Delta\sigma$. In the EC3 and the IIW, the characteristic values of the fatigue actions are assumed to be a value with an appropriate partial safety factor. [5]

The fatigue action is:

$$\gamma_{Ff} * \nabla\sigma \tag{2.1}$$

γ_{Ff} : Partial factor for fatigue loading (γ_{Ff} in EC3 and γ_f in IIW)

$\Delta\sigma$: Applied stress range

As shown above, the two factors influencing the fatigue actions are partial factor and applied stress range. The partial factor for fatigue loading covers three uncertainties in estimating: the applied load level, the conversion of loads into stresses and stress ranges, and the equivalent stresses from variable amplitude fatigue loading.

2.2.1 Fatigue load

What types of loads result in fatigue load effects and cause fatigue damages? All types of fluctuating loads have to be considered. The repeated actions can be imposed loads, dead weights, snow load, fluid pressure, wind load, temperature variation and hydrodynamic load etc. The resulting stresses from above actions, no matter the action is moment, axial force or combine, lead to fatigue load effects at the weakened position.

Exactly confirming fatigue actions is one of the greatest problems and includes many uncertainties. In application, usually, only estimations of the stress history can be made.

2.2.2 Definitions – fatigue loading analysis

Generally, there are three loading related factors:

- Type of loading: bending, shear, fretting etc, see also Section 2.2.1
- The loading levels; the stress cycles, see Figure 2.3
- The number of loading cycles as well as the frequency of loading cycles, see Figure 2.4

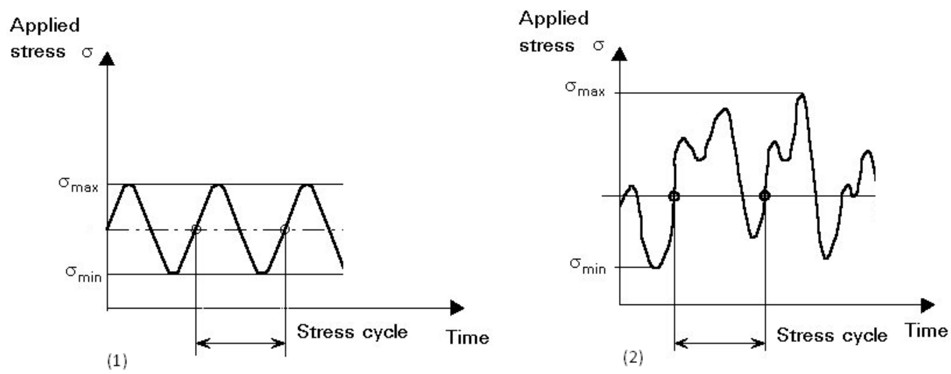


Figure 2.3 loading levels for (1) Constant amplitude stress history and (2) Variable amplitude stress history. [5]

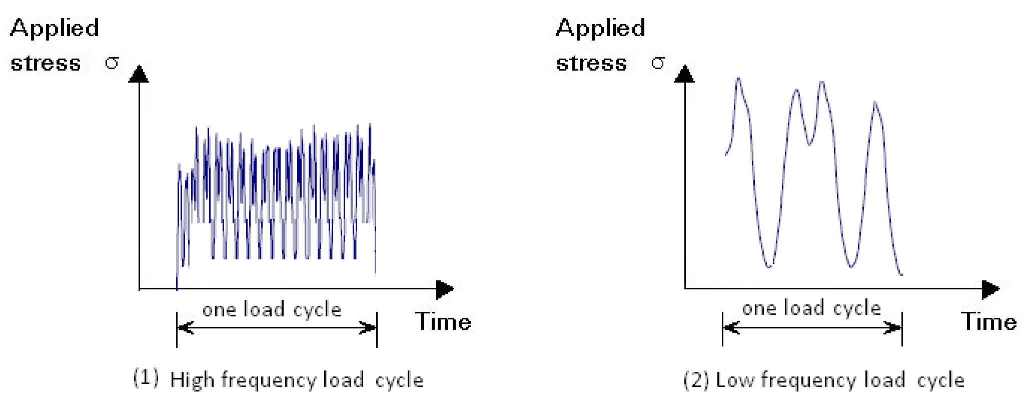


Figure 2.4 Frequency of loading cycles. (1)High frequency of loading cycle and (2) Low frequency of loading cycle [5]

For one cycle of fatigue loading, Figure 2.5 illustrates the definition of fatigue stress cycles.

Stress range, $\Delta\sigma$, is the peak to peak stress:

$$\Delta\sigma=\sigma_{max}-\sigma_{min} \quad (2.2)$$

Stress amplitude, σ_a, is the amount the stress deviates from the mean:

$$\sigma_a=\Delta\sigma/2 \quad (2.3)$$

Mean stress, σ_{mean}, in the cycle is the average value of the stresses:

$$\sigma_{mean}= (\sigma_{max}+\sigma_{min})/ 2 \quad (2.4)$$

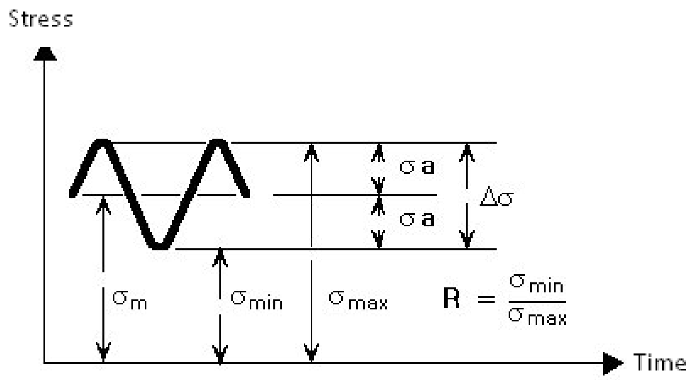


Figure 2.5 Definition of fatigue stress cycles [5].

The stress range is a main parameter that influencing fatigue strength. But the stress ratio will also influence the fatigue strength. Figure 2.6 indicates the stress radio for different types with respect of constant load history.

Stress radio: $R= \sigma_{min}/ \sigma_{max}$

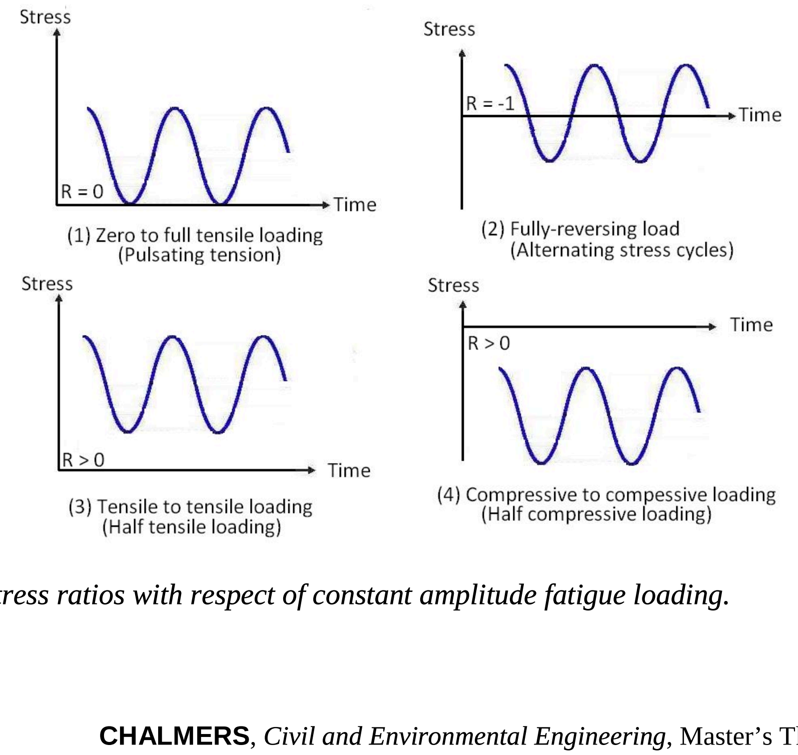


Figure 2.6 Stress ratios with respect of constant amplitude fatigue loading.

With the same stress range, the fatigue damaging degree is: (3) > (1) > (2) > (4). The modification factors in different codes are recommended for this phenomenon.

The more complex loading history is the variable amplitude fatigue loading; see Figure 2.3(2), which can be calculated by the cycle counting method and the cumulative frequency diagram (stress spectrum), IIW (2008) [3].

The deformation type with low-cycle fatigue distinguishes with high-cycle fatigue. Low-cycle fatigue means where stress is high enough to occur plastic deformation. (by L. F. Coffin in 1954 [6]) High-cycle fatigue is defined that fatigue life require more than 104 cycles where stress is low and deformation mainly elastic. The IIW recommendation mainly focuses on the later one.

2.2.3 Stress categories

According to various stress analysis methods, there are mainly three stress categories used in fatigue analysis of weld structures.

- Nominal Stress
- Geometric Stress (Structural Stress)
- Notch Stress

Those stress terms are defined in detail below.

2.2.3.1 Nominal Stress

The nominal stress is a stress calculated by simply elasticity theory, ignoring stress raisers of the weld joints and plastic flow, but including the stress raising effects of the macro-geometric shape of the component near the joint, such as cutouts, frame edges and unequal stress distribution [8].

For simply geometric structures, the nominal stresses are readily to calculate directly. (Equation 2.5) Care must be taken to ensure that all stress raising effects of the structural details of the welded joints are excluded when calculating the modified (local) nominal stresses. As a result, a certain distance away from welded joint section is considered. However, up to now, no common codes illustrate the nominal stress determination from FEA results, A.F. Hobbacher (2008) [7].

$$\sigma_{nom} = \frac{N}{A} + \frac{M}{W} \tag{2.5}$$

Where, N is the axial force, A is the cross section, M is the bending moment and W is the bending resistance, divided by the distance from gravity centre to the calculate point.

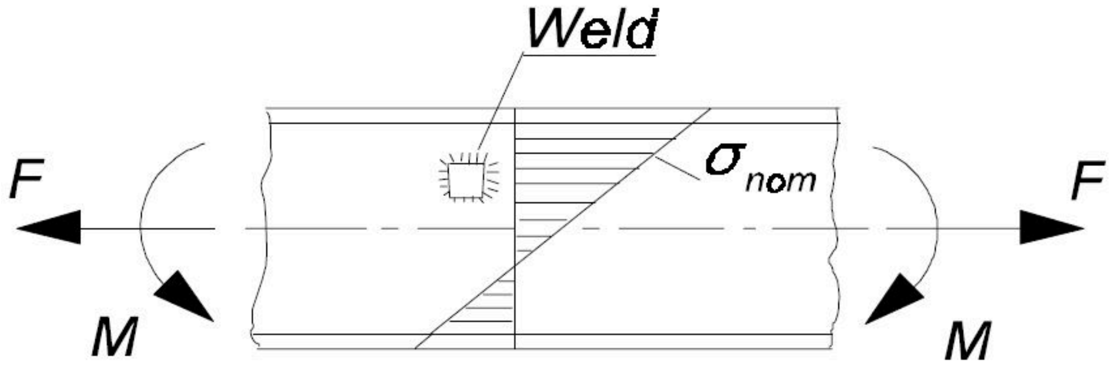


Figure 2.7 The nominal stress in a beam-like component [3].

When analysing modified nominal stress (Figure 2.8), the geometrical concentration factor K_t calculated from FEA results is used.

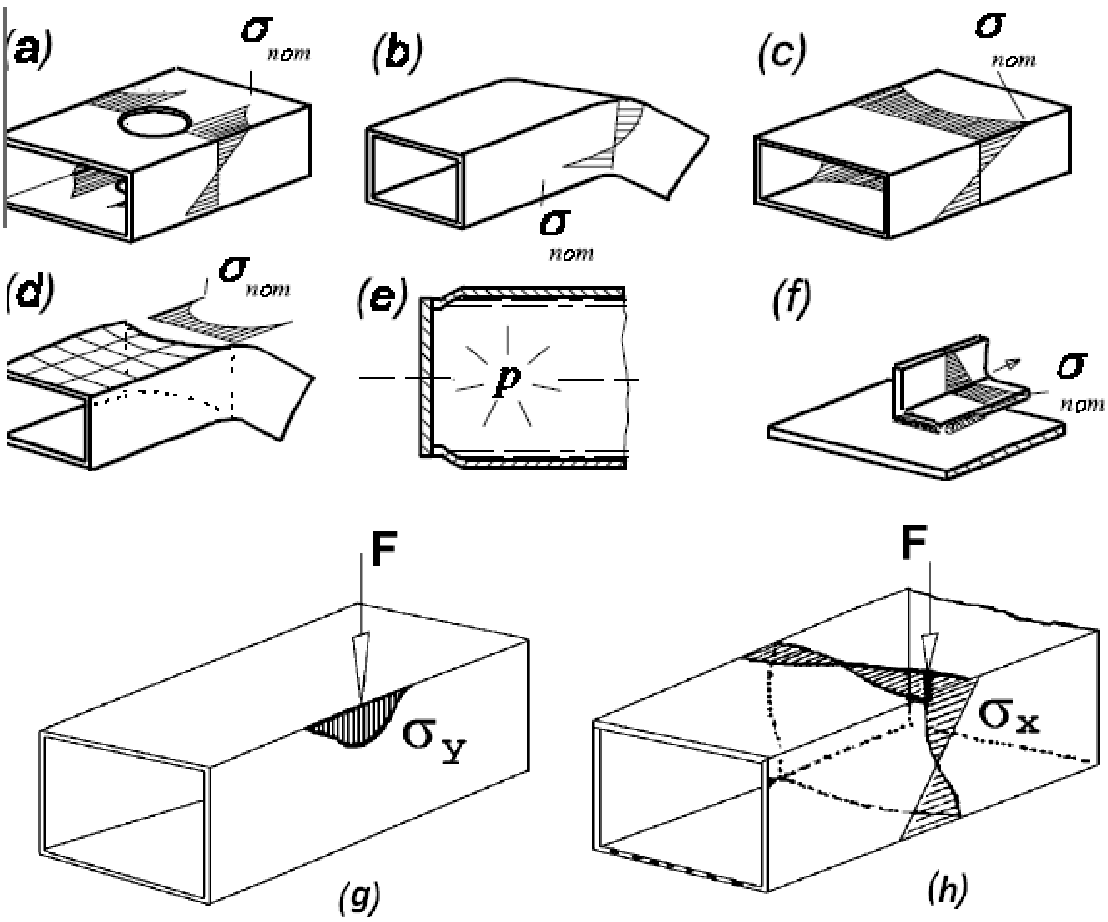


Figure 2.8 (a) to (e) Modified nominal stress due to macro geometric effect (g) and (h) Modified nominal stress due to concentrate force [3]

For even more complicated details, other stress evaluation approach, the hot spot method should be used instead of nominal stress.

2.2.3.2 Geometric Stress (Structural Stress)

The geometric stress, also called structural stress (σ_s), includes two stress components, membrane stress (σ_m) and shell bending stress (σ_b). All stress concentration factors are taken into account except the notch effect due to the weld itself.

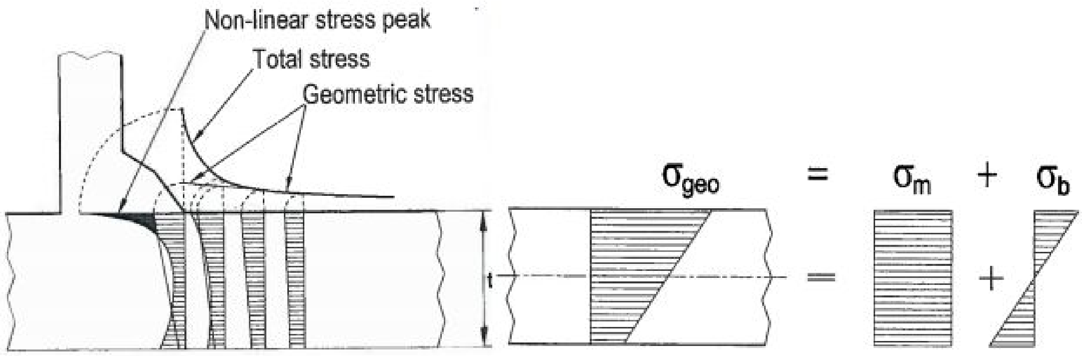


Figure 2.9 Geometric stress includes two stress components. [8]

The geometric stress can be determined by finite element method or multiply by a geometric stress concentration factor K_s. However, it is difficult to find such a suitable stress concentration factor.

And what is hot spot stress? The hot spot stress (σ_{hs}) is the geometric stress on the surface at the critical point (hot spot), such as geometry discontinuity and weld point where a fatigue crack is expect to grow. (Figure 2.10)

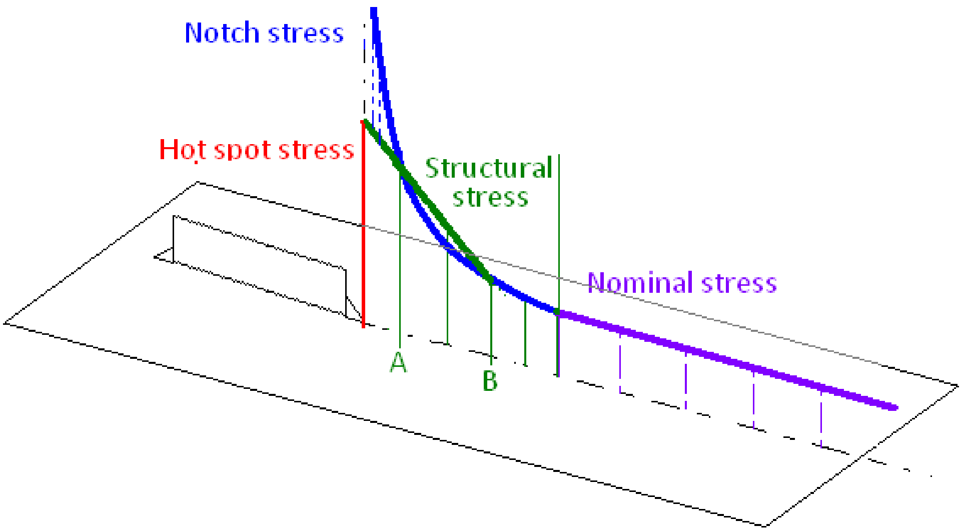


Figure 2.10 Notch stress, structural stress and hot spot stress on surface. A and B are the extrapolation measuring point in hot spot approach

There are mainly six approaches to measure hot spot stress using finite element method.

- Linear stress extrapolation (IIW 2008) [3]
- Quadratic stress extrapolation (IIW 2008) [3]
- Batelle (Dong) structural stress approach (Dong 2002) [9]

- One millimetre structural stress approach (Xiao and Yamada 2004) [10]
- One point stress determination (Fricke 2002) [11]
- Through thickness structural stress approach (Radaj 1995) [12]

Methods one, two and four will be introduced later in Section 5.2.2.

2.2.3.3 Notch stress

There are three stress components expressed the total stress in the notch through the thickness. They are membrane stress (σ_m), shell bending stress (σ_b) and non-linear stress peak (σ_{nl}), see Figure 2.11

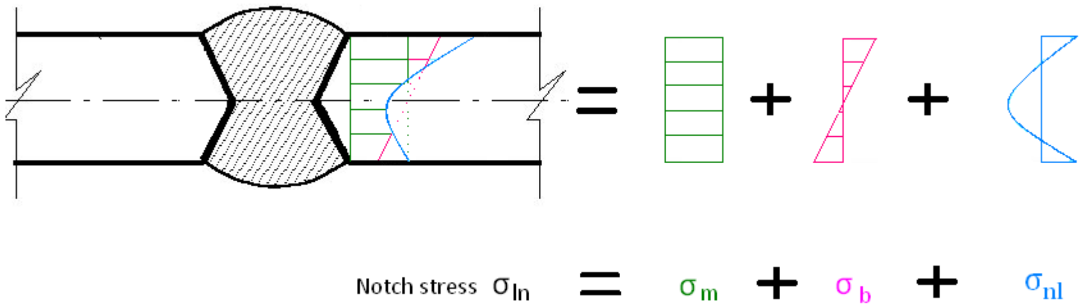


Figure 2.11 Notch stress at weld toe, divided into three stress components [8]

The membrane stress (σ_m) is the mean stress that is constant through the thickness.

The shell bending stress (σ_b) is the linearly distributed stress determined by drawing the stress curve through the midpoint, where the membrane stress intersects the mid-plane of the plate. The non-linear stress is the value remaining at weld toe due to weld radius and welding angle, also called non-linear peak stress. [3]

One method to determine the effective notch stress is FEM. Rare effective notch stresses are obtained from an FE model for a complete structure directly. This is mainly because a very fine mesh is required in order to accurate enough.

The other approach is using the stress concentration factor, K_t, see Equation 2.6. K_t is defined as the maximum stress at the effective radius, divided by the structural stress in the plate, IIW (2008) [3]. It depends on the weld throat thickness, plate thickness, effective radius and the type of load.

$$K_t = \frac{\sigma_{nl}}{\sigma_{nom}} \tag{2.6}$$

The IIW gives the instructions for evaluation using the effective notch stress method.

Finally, different stress categories are together shown in figure 2.12.

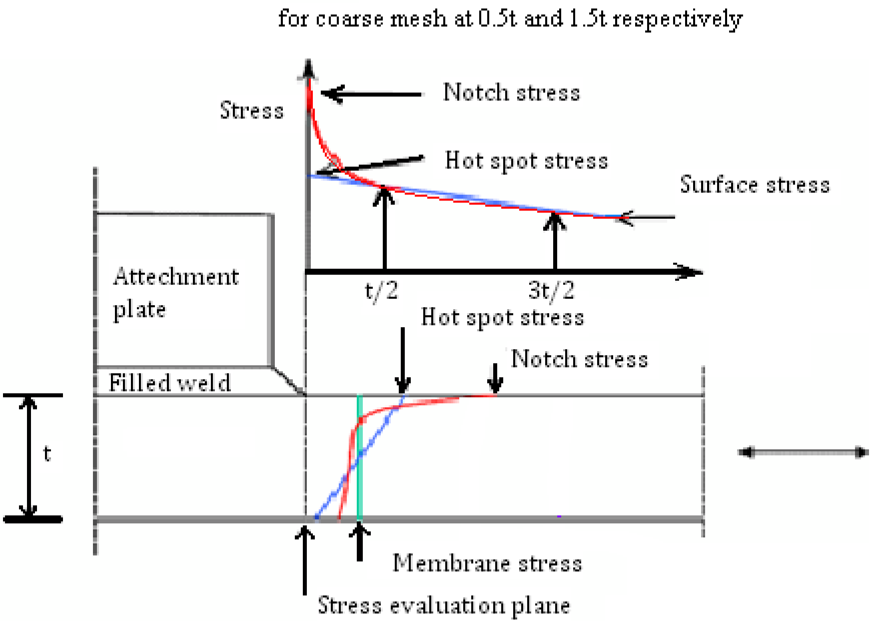


Figure 2.12 the fatigue stress categories and the expression of through thickness stress on the surface)

2.3 Fatigue capacity (Resistance)

The fatigue resistance is usually derived from experimental fatigue test data. There are two ways to present fatigue capacity: the first way is S-N curves given in section 2.3.1 and the second one is using fracture mechanics analyses. The fatigue resistance data are in the form of relationships between range of stress intensity factor (ΔK) and the rate of fatigue crack propagation (da/dN), which will be omitted in the paper.

2.3.1 S-N curves and fatigue classes

The fatigue design curve or the S-N curve is a logarithmic graph between the stress range (S) and the number of stress cycles to failure (N), which is also called fatigue life, based on a large number of fatigue test for a certain detail. All the test data generally produce a wide spread of results, the S-N curve is determined for a certain failure probability, see figure 2.13, e.g. 50% for mean fatigue strength, 2.3% for characteristic fatigue strength. In different codes, they recommend different slopes for S-N curves.

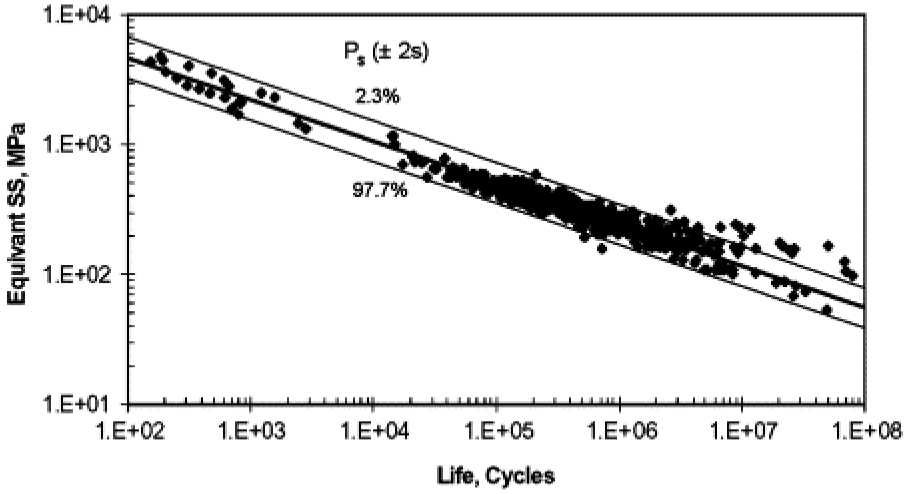


Figure 2.13 S–N curve according to Dong (2005) with scatter band for two probabilities of survival Ps.

Fatigue classes are defined as the characteristic fatigue strength, also called fatigue category, in MPa, at 2 million load cycles, see Figure 2.14. These values are the fatigue classes (FAT in the IIW 2008 and Detail category $\Delta\sigma_c$ in the EC3)

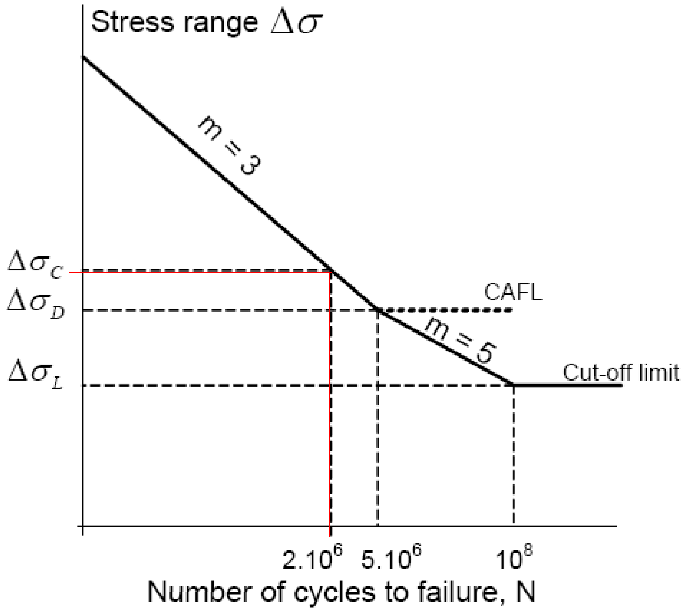


Figure 2.14 S–N curve definition in EC3

For the nominal stress method, IIW Recommendations (2008) [3] provides 13 S-N curves for consideration of normal or shear stress ranges for more than 100 different welded joints, while 14 S-N curves for normal stress and 12 S-N curves for shear stress are illustrated in the EC3.

Whereas, for assessing the fatigue resistance of a detail on the basis of the structural hot spot stress, there are only two S-N curves in the IIW 2008, FAT90, FAT100, and three S-N curves in the EC3, Detail category 112, 100 and 90. Nevertheless, the hot spot approach is not able to use for the fatigue from weld root and the shear stress fatigue.

2.3.2 Factors influence the fatigue resistance

The fatigue problem of steel structural components, especially welded steel detail, is a particularly complex problem, and many factors may exert an influence on the fatigue strength. Since the S-N curves in the IIW (2008) [3] are based on the experimental data, in which small effects of misalignment, high residual stress, and welded imperfections with normal fabrication standards, for instance Undercut, Porosity and inclusions, Crack-like imperfections, have already been included. However, the following factors which are not caused by weld imperfections should be taken into account to modify the fatigue strength.

- Stress ratio
- Wall thickness
- Improvement techniques
- Effective of elevated temperatures
- Effective of corrosion

All the modification factors are given in the IIW (2008) [3], and the wall thickness effect will be discussed further in chapter 3.

2.4 Fatigue evaluation methods

The fatigue action and the fatigue resistance are introduced above, in section 2.2 and 2.3. Now an appropriate assessment procedure is needed to relate these two elements together. Three procedures are presented in the IIW [3], respectively, the S-N curve approach, the crack propagation approach, and the direct experimental approach. However, the focus of the description will be on the S-N curve approach.

For constant amplitude loading, the characteristic resistance stress ranges $\Delta\sigma_R$ should be determined at required number of cycles. Then $\Delta\sigma_R$ should be divided by the partial safety factor γ_{MF} for the final resistance. And in section 2.2, the fatigue stress have already introduced. Finally, the following fatigue criterion (Equation 2.7) should be checked, which is the only method will be used in this paper:

The design criterion is:

The design stress range for the specific detail for the design life, N_{life}

$$\gamma_{Ff} \cdot \Delta\sigma \leq \frac{\Delta\sigma_R}{\gamma_{MF}} \qquad (2.7)$$

Safety factor for load

Applied stress range

Safety factor for fatigue strength

If the load produces both normal and shear stresses, the following criterion must be met, and here CV is a recommendation value which is given in a table in the IIW [3].

For the variable amplitude loading, a cumulative damage calculation procedure is applied to calculate the total damage, IIW (2008) [3]. See Equation 2.8.

$$\left(\frac{\Delta\sigma_{S,d}}{\Delta\sigma_{R,d}}\right)^2 + \left(\frac{\Delta\tau_{S,d}}{\Delta\tau_{R,d}}\right)^2 \leq \mathrm{CV} \tag{2.8}$$

In the following sections the nominal stress method and the hot spot stress method will be compared, since only these two methods will be used in this thesis.

2.4.1 Nominal stress approach

The nominal stress approach is the earliest known and commonly used method, and there are a lot of S-N curves classified by different types of joints are available in codes that are based on the nominal stress approach. Section 2.3.2 indicates that in fatigue class determination and S-N curves, a lot of effects have already been considered. However, the macro-geometric effects are not generally taken into account. Therefore, these should be considered when calculating the nominal stress. In all cases the fatigue strength is given as a nominal stress range. [8] This approach is therefore, if possible, especially for hand calculation preferred among the others for its simplicity.

On the other hand, nowadays, fabricated structures are often so geometrically complex that the determination of the nominal stress is difficult. Moreover, in finite element analyses only the local stress can be obtained at some points and it is hard to transfer it into nominal stress. To overcome these difficulties, the structural hot spot approach is developed and applied to welded structures.

2.4.2 Structural hot spot method

The structural hot spot method is based on the structural stress at the hot spot, which needs to be obtained from appropriate stress analyses, more detail see section 2.2.3.2. This method was originally developed for the offshore industry, and has been used for the fatigue design of pressure vessels and welded tubular connections since 1960s, Marshall and Wardenier (2005) [13]. In the early 1990, the classification companies introduced fatigue assessment procedures based on the hot spot stress concept also for plated structures, Lotsberg (2006) [14]. Now, in the car industry and the bridge industry, engineers are recommended to use this method for fatigue analyses as well.

In welded structures, Niemi (1993) has listed several cases where the hot spot approach is more suitable than the nominal stress approach, Gary Marquis & Asko Kähönen (1995) [15]:

- There is no clearly defined nominal stress due to complicated geometric effects,
- The structural discontinuity is not comparable with any classified details included in the design rules,

- For the above-mentioned reasons, the finite element method is used,
- Field testing of a prototype structure is performed using hot spot strain gauge measurement, and
- Offset or angular misalignments exceed the fabrication tolerances, thus invaliding some of the basic condition for using the nominal stress approach.

Furthermore, compared with the nominal stress approach, this method needs only two S-N curves, but gives comparable more accurate results.

But this method still has disadvantages. Firstly, the hot spot stress are depended on modelling technique, mesh size and arrangement, while mesh-size insensitive structural stress definition was developed in order to save computational resources and convenient for FEM analyst, Dong (2006) [16]. Secondly, this method is not suitable for the analysis of fatigue cracks initiated from embedded weld defects or weld roots.

Although, the structural hot spot stress approach has been used for the fatigue assessment for over 40 years in the offshore industry, the method is still not completed for the application of welded plate structures. In this thesis, the thickness effect using the hot spot stress approach will be discussed.

3 Thickness effect

The thickness effect is the phenomenon that the fatigue strength of a welded connection decreases when the thickness of load carrying plate increases. The thickness effect is observed in details where the fatigue crack initiation and propagation take place at the weld toe.

As mentioned in Section 2.3.2, the thickness effect factor is one of the modification factors which will influence the fatigue resistance. In 1979 Gurney [17] pointed out on the basis of fracture mechanics analysis and experimental evidence that the effect of plate thickness on fatigue strength could be significant. On the basis of nominal stress method with S-N curve data for tubular joints, covering a range of plate thickness up to 50mm, Gurney (1981) [18], proposed an empirical thickness correction for fatigue strength, see Equation 3.1.

$$S = S_B (t_B / t)^{0.25} \tag{3.1}$$

where S_B refers to fatigue strength for a reference plate thickness t_B.

With an S-N relation given by

$$N * S^3 = const. \tag{3.2}$$

the corresponding thickness correction for fatigue life is

$$N = N_B (t_B / t)^{0.75} \tag{3.3}$$

where N_B refers to fatigue life for a reference plate thickness t_B.

In 1984 a thickness correction factor for fatigue strength was been implemented in offshore design codes like the Department of Energy, (1990) [19], see Figure 3.1

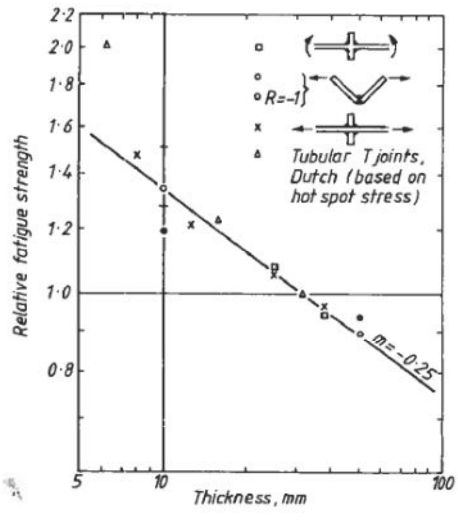


Figure 3.1 Influence of plate thickness on fatigue strength (normalised to a thickness of 32mm). All test at R=0 except where stated, Gurney (1989) [20]

Later, due to the researching works done by Berge, Webster, Haagensen and other Norwegian researchers, this factor was also included in the Norwegian Standards Steel Design Code, and many other Codes. The main body of results was presented and summarized at the Delft conference on steel in Marine Structures (1987) [21], Berge (1990) [22]. However, all of these recommended factors were derived from experimental results and were based on the nominal stress method. There are still rather limited documents which discuss about this factor and which give guidance on how to use this factor in the design when the hot spot stress approach is used.

The IIW (2008) [3] recommends a modification factor proportional to 0.1 to 0.3 power of the thickness for a wall thickness up to 25mm according to different details. This factor is applied to both nominal stress approach and hot spot stress approach. In the EC3 (2003) [23], it is recommended to use 0.2 for some specific details for nominal stress approach. As the FE-analysis is frequently used today, the hot spot stress approach becomes more practical. A new recommended thickness effect correction factor for this method should be studied further.

Moreover, the hot spot method is more and more penetrating into bridge and construction industry. And thick plates are often used in civil engineering application, so it is important to clarify this problem. And more detailed discussion will be presented in the following chapters.

3.1 Causes of thickness effect

There are several reasons to why an effect of plate thickness may appear in fatigue of welded joints. Three main explanations for the thickness effect can be distinguished as follows:

- Statistical, Örjasäter (1987) [24]

  For thicker plate, a larger volume of material is stressed. The probability of the initial defects refers to the dimension of the joint, so the larger volume structural component involves more imperfection and initial defects than thinner plate, which gives a weaker fatigue capacity for thicker plate. The length of weld toe from which the cracks initiate is therefore an influencing likelihood of initiation and failure of the welded joint, Overbeeke and Wildschut (1987) [25].

- Production-related (technological)

  A technological size effect is due to different material properties, different fabrication processes and different surface finish method experienced by large and small components. Because of more restrain in a thicker plate, the residual stress in large joint is higher than thinner one, which influences the fatigue life of welded structures.

- Geometric factors

  The radius of the weld toe does not depend on the wall thickness, but results in a relatively smaller radius for thicker joint components, Berge (1990) [22]. See Figure 3.2. (a_i=a_i, but $a_i/T_1<a_i/T_2$). Therefore, for the thicker plate, the stress at weld toe is close to the surface stress, but for the thinner one, since its large

stress gradient, the stress at weld toe drops dramatically from surface, which result a relatively small stress at the tip of the weld toe. Gurney T.R. (1979) [17] demonstrated based on experimental evidence and fracture mechanics analyses that the effect of the plate thickness could be significant with the increased plate thickness of same geometry subjected to the same magnitude of stress.

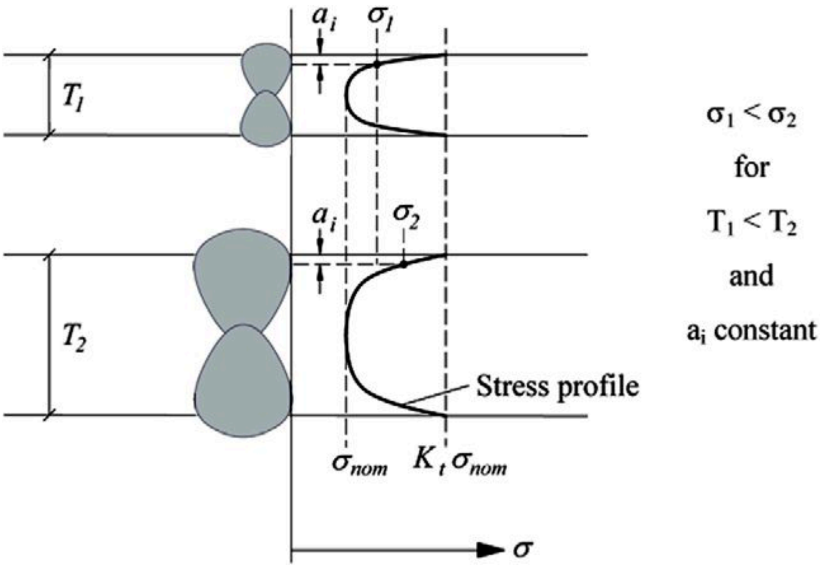


Figure 3.2 A simplified model for describing the geometric effect of thickness in weld toe fatigue failures, Berge (1990) [22].

Schumacher (2003) [26] defined three possible scenarios of geometrical size effect. Proportional scaling, where two welded details are identical except in scale was considered to be the true "thickness effect". Most of the tests on plated joints that were used to establish thickness correction factors have proportional scaling. Non-proportional scaling with constant attachment plate or tube thickness was referred to as the "scale effect". Non-proportional scaling with varying main plate or tube thickness was considered to be a combination of the thickness and scale effect.

According to Örjasäter (1988) [24], the two major factors influencing the thickness effect are the technological size effect and the statistical size effect. But, Niemi [27] states that the geometrical effect, i.e. technological size effect and stress gradient effect, alone often dominates the contribution to the thickness effect.

In some reports, it was clarified that thickness effect depends not only on the main plate thickness, but also on the welded attachment size. And this is also varied depending on loading modes such as tension and bending, Yagi (1993) [28], but these effects are directly accounted for in the hot spot method.

3.2 Literature survey

3.2.1 The methodology of deriving the thickness effect correction

Gurney (1981) [18] indicated that the thickness effect could be demonstrated using both fracture mechanics theory and experimental work (S-N curve based on nominal stress approach).

- Fracture mechanics

  The two assumptions for a simplified model to describe thickness effect in weld toe fatigue failures are, Berge (1985) [29]:

  a) Welded joints of the same type in various plate thicknesses are geometrically similar (typical for load-carrying welds), which leads to a steeper stress gradient in the thinner joints.
  b) Initial conditions of fatigue crack growth are independent of plate thickness (a_i=constant), which experience a smaller stress than the initial crack of the same length in the thicker plate, causing a smaller initial crack growth rate in the former case.

  For notched machined components, Haagensen (1988) [30] pointed out in geometrically similar parts with similar microstructures, the stress gradient is steeper in the smaller part and a surface grain will experience a lower average strain than a grain at the surface of a thicker part, at the same surface stress. On this basis, a prediction model was formulated. Berge (1989) [31]

  $$\Delta S/\Delta S_0=(t_0/t)^n \tag{3.4}$$

  $$n=0.1+0.14\log K_t \tag{3.5}$$

  K_t is stress intensity factor

- Experimental work

  The well known thickness correction factor was proposed by Gurney (1981) [18], see Chapter 3, Equation 3.1&3.3. Berge (1985) [29] also perform the same work, and obtained the same factor which is 0.25. Later work by Gurney (1989) [20] showed that n=0.21 for t>20mm and n=0.18 for t<20mm. Maddox (1987) [32] did not show any slope transition in the fatigue strength and thickness relation but found the thickness exponent over the whole range of thickness considered could be adequately represented as 0.2. These values are less than the exponent value of 0.25. But the research reported by Xue (1990) [33] suggested the exponent to be 1/3, which means that the empirical thickness correction proposed by Gurney may be unconservative. So it can be found the suggested thickness correction factor exponents are in a range of 0.1 to 1/3.

  Gurney T.R. (1989) [20] proposed that the reference plate thickness t_B as 32mm, see Figure 3.1. In the work by Smith and Hurworth (1984) [34], it was shown that the thickness effect did indeed continue to smaller thicknesses than 22mm with a transition in slope at t=22mm. On the other hand, Gurney (1995) [35] studied "relatively thin" weld joints, and concluded that the thickness effect extends back to at least a thickness of 10mm and probably further.

According to the discussion above, there is still a lot of disputation about the values of the exponent and reference thickness. Berge (1989) [31] showed the thickness effect was essentially unaffected by and superimposed on any effects of post weld heat treatment, variable amplitude load history, and sea water environment. However, Yagi et al. (1993) [28] claimed that weld improvement makes fatigue strength increase and thickness effect decrease. They gave three exponents, 1/10, 1/5 and 1/3, for different detail and different weld treatment. Furthermore, some researchers have found that there are other factors that will influence the thickness effect. Cole (1993) [36] showed that thickness effect depended on the stress range, and gives an exponent of $3*[K-(\Delta\sigma*C)]$, where K and C are constants depending on the environment (air or cathodic protection) and the type of joint.

3.2.2 Consideration in various Design Standards

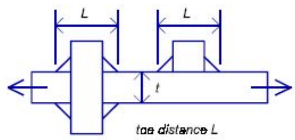
Many codes give different recommended exponent values and reference plate thicknesses for different details. In the EC3 [23], the thickness effect are only considered for the nominal stress approach. While, for the hot spot stress approach, only few standards recommend some values. Some of the standards are shown in Table 3.1 as following.

From the Table 3.1, it is shown that the formulas and the methodologies to consider the thickness effect are almost same. The thickness effect is considered by the similar formula, $S=S_B(t_{ref}/t_{eff})^n$. But the parameters of n and t_{ref} vary in different codes. The value of n roughly ranges from 0.1 to 1/3 depending on the type of details and weld treatment. t_{ref} value is between 16 and 32. The most common values are n=0.25 and t_{ref} =25.

Here, all the standards above are based on the nominal stress method, and only IIW (2008) [3] gives the recommendation value for the hot spot method which is the same as the nominal stress method.

Table 3.1 Standards and recommendations for thickness effect (nominal stress approach) [37]

Standard	Rule	Remarks
Department of Energy (1990) guidelines [19]	$S=S_B(t_B/t)^{0.25}$	a) For nodal joints the basic S-N curve is the Class T-curve and relates to a thickness of 32mm,i.e. $t_B=32$ b) For a non-nodal joints the basic curves are the Class B-G curves and relate to thicknesses up to 22mm, i.e. $t_B=22$, A value of t=22 should be used for calculating endurance N when the actual thickness is less than 22mm
BS 7608 (BSI 1993) [38]	$S=S_B(16/t)^{0.25}$	Basic S-N curves relate to a reference thickness of 16mm for welded joints. For welded joints of other thicknesses greater than 16mm, the correction factors on stress range should be applied to produce a relevant S-N curve.
Hobbacher (1996) [39]	$S=S_B(25/t)^{0.25}$	The influence of plate thickness on fatigue should be taken into account in cases where cracks start from the weld toe on plates thicker than 25mm. The thickness correction exponent, n, is dependent on the effective thickness and joint category as follows: 1) n=0.3, for as-weld cruciform joints, transverse T-joints, plates with transverse attachments 2) n=0.2, for toe ground cruciform joints, transverse T-joints, plates with transverse attachments 3) n=0.2, for as-welded transverse butt welds 4) n=0.1, any butt weld ground flush, base material, longitudinal welds or attachment
AS4100-1998 (SAA 1998) [40]	$S=S_B(25/t)^{0.25}$	No thickness correction is required except for transverse fillet or a butt welded connection involving plate thickness (t) greater than 25mm
EC3 (2003) [23]	$S=S_B(25/t)^{0.2}$	The size effect due to thickness or other dimensional effects is applied mainly to: I) Transverse spices in plates and flats with uniform or taped widths. II) Flange and web splices in plate girders and III) Full cross section butt welds of rolled sections with or without cope holes
	$S=S_B(30/\phi)^{0.25}$	For bolts and rods with rolled or cut threads in tension. Size effect for bolt or rod diameters,

		ϕ >30mm
IIW Recommendations (2008) [4]	$S=S_B(t_{ref}/t_{eff})^n$	a) The thickness correction exponent n is dependent on the effective thickness t_{eff} and the joint category, 1) n=0.3, for as-weld cruciform joints, transverse T-joints, plates with transverse attachments 2) n=0.2, for toe ground cruciform joints, transverse T-joints, plates with transverse attachments 3) n=0.2, for as-welded transverse butt welds 4) n=0.1, any butt weld ground flush, base material, longitudinal welds or attachment b) If L/t >2, then t_{eff}=t; if L/t<2 then t_{eff}=0.5L or $t_{eff}=t_{ref}$ which is the larger. Where L is the toe distance including attachment thickness. 

3.2.3 Summary of research on thickness effect

Some of the tests carried out by previous researchers are illustrated below, see Table 3.2. All the tests listed here are on plated specimens, and in the majority, the main plate and transverse plates are usually of equal thickness. However, the main plates and attachments are of different thicknesses also tested, which show that there is a thickness effect associated with non-proportional scaling, Mashiri and Zhao (2005) [37]. The following summary shows the type of joints tested, the load type and the thicknesses of the plates. Some of the data will be used in this thesis, and it will be discussed in Chapter 4.

Table 3.2 Summary of research on thickness effect

References	Type of joint	Loading	Thickness range [mm]	Remarks
Mohaupt et al (1987) [41] Vosikovsky et al(1989) [42]	Plate T joints	3- point bending constant and variable amplitude in air	16, 26, 52, 78, 103	Joints tested in the as welded condition, Stress ration R=0.05, joints with proportional and non-proportional scaling test, improved weld profile tested
Booth (1987) [43]	Plate T-joints	4-pointbending, constant amplitude in air	25,38,50,75,100	Joints tested in as welded condition
Overheeke and wildschut(1987) [44]	Plate T-joints	Pure bending, constant amplitude in air	20, 100, 150	Joints tested after PWHT (post weld heat treatment)
Orjasaeter et al(1987) [24]	Plate T-joints	Cantilever bending; 3-point bending; constant amplitude in air	30, 70, 100, 130, 160	Joints tested in as welded condition and agter PWHT; R=0.1
Noordhoek et al(1987) [45]	Plate T-joints	4-point bending; constant amplitude in air	Main plate t=70 and 160mm with transverse/longitudinal plates of t-20 and 45mm	Joints tested in the as welded conditions; thickness effect due to non-proportional scaling of main and attachment plate thickness; R=0
Gurney (1989) [20]	Plates with longitudinal edge attachments	Tensile cyclic loading	Width between longitudinal attachments, W=40,80,125,200	Joints tested in a stress relieved condition; R=0

Eide and Berge(1987) [46]	Plate girders	4-point bending	20,40,60	Joints tested in the as welded condition
J.Yagi, S. Machida (1993) [47]	Cruciform joint	Pulsating tension	10, 22, 40, 80	Two types of improving treatment methods: profiling and toe-grinding
		3-point bending		
	Plate T-joint	Pulsating tension	22, 40, 80	
		3-point bending		
Gurney (1995) [35]	K butt welds	Axially loaded	10, 13, 20, 25	Joints tested in the as welded condition R=0
	Transverse fillet welds	Axially loaded	10, 12, 13, 14, 15, 16, 19, 20, 25	Joints tested in the as welded condition R=0
			13, 20, 25	Joints tested in a stress relieved condition R=0
		Bending	6.4, 8, 11, 13, 20, 25, 26	

3.3 Effects of decreasing thicknesses

The fact that even if the thickness effect is not considered in standards when the thickness below the reference thickness, thinner plates still give an increase in the fatigue strength. Some research papers discuss the subject.

For the t_0 factor, Fricke (2001) [48] indicates that there is no fatigue strength improvement for thicknesses under t_0, since the thickness effect does not consider the effect of misalignments. When the thickness becomes larger, the effect of the misalignment will be reduced. Thus there exists a reversed thickness effect. Fricke speculates that this reversed thickness effect might be responsible for the existence of the reference thickness t_0. Below t_0 the thickness effect and the reversed thickness effect approximately is cancelling each other.

Gurney (1995) [35] investigated for relatively thin weld joints and found out that there is clear evidence that the “thickness effect” extends back to at least a thickness of 10mm and probably further.

Mashiri and Zhao (2005) [37] showed the weld defects such as undercut become a major negative influencing factor for the weld with thickness less than 4mm. And for this very thin detail, fatigue strength actually decreases as the member failing under fatigue becomes thinner. This observation in thin walled joints can be attributed to the greater negative impact that weld toe defects such as undercuts have on fatigue crack propagation life.

As the development of the welding technology, the misalignment effect will be less. And according to Gurney and Mashiri, the t_0 factor could be between 4mm and 10mm, which indicates that the t_0 factor from the above standards, 16mm to 32mm are too large.

4 Data Collection & Database

All the test data which are used in this thesis come from previous work which is published in various journal papers and reports. In these tests, the strain gages were only placed on the surface of the plate and far away from the weld toe which means only the nominal stresses are recorded. Without strain readings near the weld toe, the hot spot stresses are unknown directly from test.

4.1 Rules to select data

Since there are a lot of articles which are related to the studied subject, and some of them listed the test data in the form of table, while the others are only show the results in plots. For most of thickness and fatigue strength plots are log-log scale, it is hard to calibrate and find an accurate value. In this case, only the data with accurate number will be selected in the report. Further on, the following three rules are adopted in selecting a more usable data.

1) Choose the results for each detail from the same researcher at the same test, which will reduce the error and uncertainty by manufacturing and material difference.

2) The environment condition is only restricted to the condition in the air. The test in sea water will not be used in this work.

3) Geometric proportional details are considered, which means the thickness of the plates will be always the same as the thickness of the attachments.

Even if those restrict rules are followed, there are still many influencing factors which might result in some uncertainties. For example, stress ratios are not the same in all case, and the numbers of test at each stress level are not the same at the same detail, etc. All the effects by these will be omitted, since these effects seem too small to consider.

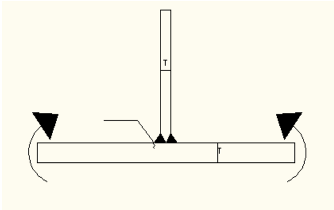
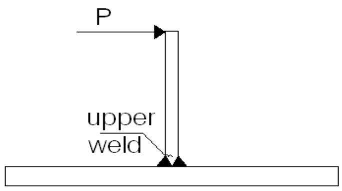
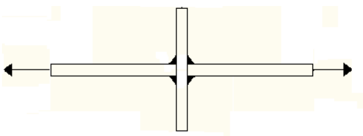
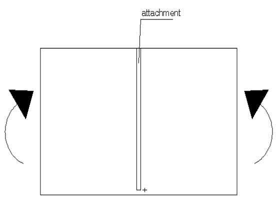
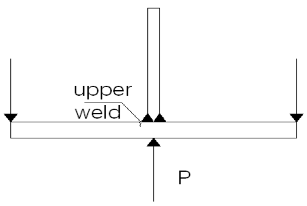
However, the cruciform joint is worth to be studied, because it is a very common detail. Even if the tests are from different researchers (rule 1) and non-geometric proportional (rule 3), they are also applied in a further studying.

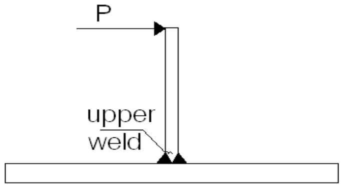
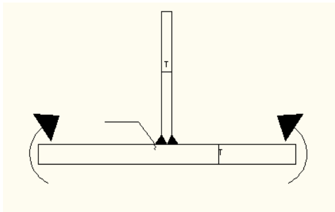
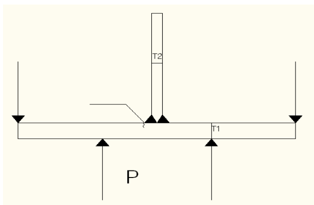
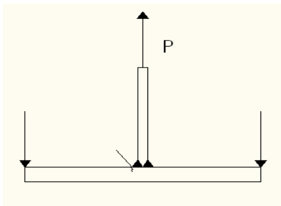
4.2 Database for this report

4.2.1 Data list

In this report, nine details are selected for studying. See Table 4.1. The first three details show better results when using the nominal stress method. So these three details are chosen for further study by using the hot spot method, see Chapter 5. For these three joints' detailed information, numbers of cycles at different stress levels are shown in Table 4.2 to Table 4.4.

Table 4.1 Nine different details and analysis result using nominal stress approach

Detail name	Reference	Drawings	Thickness and test times		Result of n factor	S-N curve slope m
Detail No.1	J.L.OVERBEEKE (1987)[25]		16mm	2*	0.163	3.998
			25mm	2*		
			40mm	2*		
			70mm	2*		
Detail No.2	O.ÖRJASAETER (1987)[24] (PWHT)		30mm	15	0.200	3.203
			70mm	4		
			100mm	4		
			130mm	4		
			160mm	4		
Detail No.3	R. OLIVIER & W.RITTER (1981)[49]		7.9 mm	5	0.33	5.104
			12 mm	3		
			14 mm	5		
			16 mm	3		
			25 mm	2+2		
			32 mm	3		
			38 mm	2+3		
Detail No.4	O.I.EIDE& S.BERGE (1987) [46]		20mm	6	0.23	1.978
			40mm	4		
			60mm	2		
Detail No.5	O.ÖRJASAETER (1987)[24]		30mm	4	0.12	2.346
			100mm	4		
			160mm	4		

Detail No.6	O.ÖRJASAETER (1987)[24] (As welded)		30mm	11	0.14	3.220
			100mm	5		
Detail No.7	G.S.BOOTH (1987) [43]		25mm	2*	0.309	3.242
			38mm	2*		
			50mm	2*		
			75mm	2*		
			100mm	2*		
Detail No.8	C.NOORDHOEK (1987) [45]		70mm	2*	0.197	3.651
			160mm	4		
Detail No.9	U.H.MOHAUPT (1987) [41]		16mm	2	0.326	3.517
			26mm	3		
			52mm	3		
			78mm	4		
			108mm	2		

*means there are more than 2 experiments had been done, but the authors only provided two stress-cycle points in order to fix the experimental S-N curve.

In chapter 5, detailed discussion about how the values of n (thickness effect exponent factor) and m (average slope of S-N curves for each detail) factors come from will be presented.

From the above table, it can be easily noticed that all the n values are between 0.1 and 1/3, which are quite fulfil the standards from Table 3.1. And other phenomenon is the n values of 7 details in the total 9 are smaller than 0.25, which means 0.25 is a quite conservative value for n. On the other hand, the m values are various, and diverse from the standards' recommended 3.

Due to enough different plate thicknesses and a large amount of test data for the first three details, in the next chapter, only these three details will be further discussed and analysed. Furthermore, these three details have reasonable n and m values.

In addition, the test of detail 8, C.NOORDHOEK (1987) [45], is special, non-proportional detail is discussed. The thicknesses of main plate are 70mm and 160mm, and the thicknesses of attachment are 20mm and 45mm for the 160mm thick plate. The result shows that thickness effect is existent and in the same manner, when increasing the attachment thickness, fatigue life and fatigue strength are decreasing. If the relation between attachment thicknesses and fatigue strengths are plotted, the exponent value will be 0.264. It has to be noted that this is typical case in which H.S. is good because it takes into account the effect of attachment thickness!

Table 4.2 Test results of detail No.1, source from J.L.OVERBEEKE (1987) [25]

DETAIL No.1							
T=16mm		T=25mm		T=40mm		T=70mm	
S(MPa)	logN	S(MPa)	logN	S(MPa)	logN	S(MPa)	logN
200	5.451	200	5.325	200	5.23	200	5.198
120	6.392	120	6.227	120	6.152	120	5.981

Table 4.3 Test results of detail No.2, source from O.ÖRJASAETER (1987) [24]

DETAIL No.2					
T=30mm		T=70mm		T=100mm	
S(MPa)	N	S(MPa)	N	S(MPa)	N
379	91120	276	156440	320	101120
360	117660	190	548630	180	373670
345	135130	183	492520	171	480000
301	130290	177	3865000	150	2313000
299	200000				
278	110290	T=130mm		T=160mm	
186	891300	S(MPa)	N	S(MPa)	N
186	742310	310	74200	300	44060
173	1295130	150	622320	151	445840
172	2221060	135	774500	115	860000
169	1555200	120	2096630	105	2990000
168	2850330				
152	2765420				
152	2221060				
145	2765420				

Table 4.4 Test results of detail No.3, source from R. OLIVIER & W.RITTER (1981) [49].

DETAIL No.3							
T=7.9mm		T=12mm		T=14mm		T=16mm	
Ref. [49] p.p. 80		Ref. [49] p.p. 23		Ref. [49] p.p. 294		Ref. [49] p.p. 58	
S(MPa)	N	S(MPa)	N	S(MPa)	N	S(MPa)	N
124	180000	55	1175000	80	750000	74	420000
108	360000	54	1060000	79	841800	59	950000
93	610000	52	1732000	76	1181600	58	1150000
85	1590000			75	2008100		
77	2490000			70	1920000		
T=25mm		T=32mm		T=38mm			
Ref. [49] p.p. 326&328		Ref. [49] p.p. 64		Ref. [49] p.p. 330&332			
S(MPa)	N	S(MPa)	N	S(MPa)	N		
45	1340100	68	320000	45	1011600		
38	2993300	49	1000000	40	2372200		
78	222800	44	2000000	133	47800		
45	2591200			80	250100		
				45	2216400		

Furthermore, these S-N curves of three details are shown in Chapter 5.1, Figure 5.2, Figure 5.4 and Figure 5.5 respectively.

4.2.2 Test specimens and testing description for first three details

In this section information about the test specimens and fatigue testing methods will be presented, e.g. metal quality, geometry of detail, weld condition, and experimental condition, etc.

- Detail No.1 (J.L.Overbeeke (1987) [25])

  - Material

    According to the literature from J.L.Overbeeke (1987) [25], the steel of this detail was a hot rolled carbon-manganese steel that complied with grade FeE 355KT, Euronorm 113-72.

  - Specimens geometry

    The main plate of the specimens were 220mm wide and 800 to 1600 mm long (depending upon the plate thickness). A gusset plate of the same thickness was welded to the main plate by a full penetration K-weld, see Figure 4.1. The length of all specimens was in the rolling direction. The height of this gusset plate was 2.25 times its thickness, which was judged to be high enough to regard it as free from stress concentration point of view.

- Welding condition

  The weld geometry, see Figure 4.1, and the welding procedure was Manual Metal Arc welding (MMAW) at the weld root, submerged arc welding (SAW) is used to fill the gap and finally finishing runs MMAW welding. The weld throat angle α was 60°. Post weld heat treatment, stress relieved condition were used in this detail.

- Experimental

  Tests were carried out with servo-hydraulic machinery, and loaded in 4 point bending, see Figure 4.2. The environment was laboratory air with a temperature of about 20°C and a relative humidity of 40 to 80%. Each series contained 10 specimens of one thickness and the thicknesses used were 16, 25, 40 and 70 mm. The weld throat angle was 60°. From these 10 specimens, 4 of them were tested at $\Delta S=200$ N/mm^2 and 6 at $\Delta S=120$ N/mm^2.All testing was done by sinusoidal loading at a stress ratio R=0.1. The frequency was 5 cps.

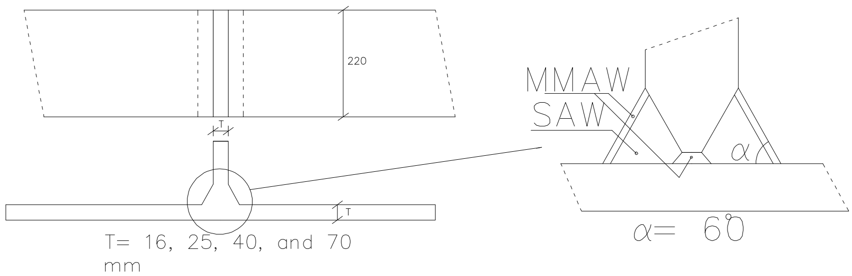


Figure 4.1 Geometry and welding detail of the specimen

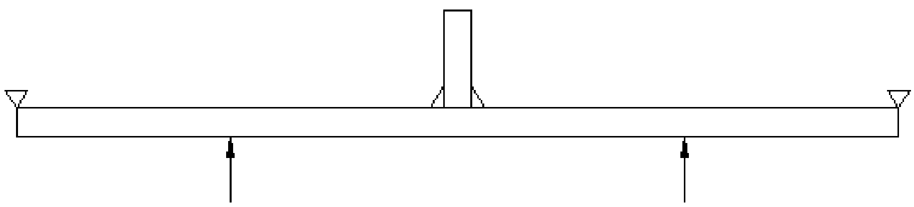


Figure 4.2 Loading condition of detail No.1

- Detail No.2 (O.Örjasatter (1987)[24])

  - Material

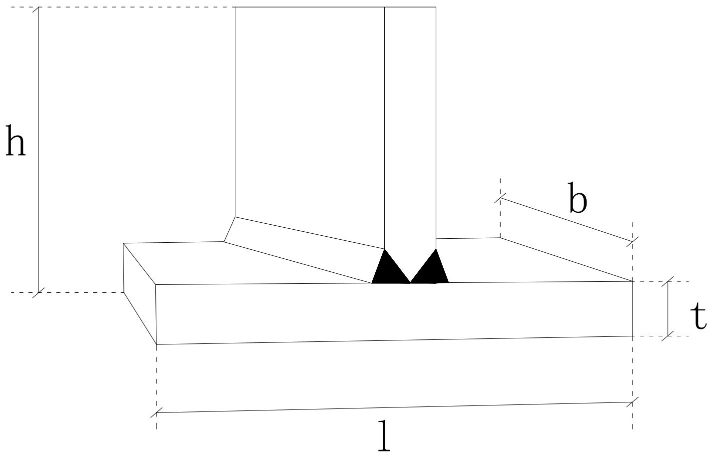
    According to the literature from O.Örjasatter (1987) [24], the material was a low carbon micro-alloyed steel (Statoil Grade 1) with a chemical composition which was nearly identical for all thicknesses. The mechanical properties show a slight decrease in static strength and charpy impact values for increased thickness.

  - Specimens geometry

    The specimens were produced in large lengths with a vertical up stand welded on to a horizontal plate. The weld was a full penetration fillet weld with a 45^o K-groove. The dimensions of the cantilever bending test specimens are shown in Figure 4.3.

  - Welding condition

    In the main part of the program manual metal arc welding was used. The electrode diameter was 3.25-5 mm for the final passes (AWS E7018). The welds were treated with post weld stress relieved (PWHT). The crack plane was parallel to the rolling direction.



t (mm)	b (mm)	l (mm)	h (mm)
30	90	281	218
70	211	656	508
100	301	938	725
130	392	1219	943
160	482	1500	1160

Figure 4.3 Geometry and welding detail of the detail No.2

- Experimental

All tests were performed in laboratory air with a temperature of about 20 to 25°C and a relative humidity of 30 to 50%. The loading frequency was 1 to 25 Hz depending on specimen deflection and type of test. The testing equipment comprised several servo hydraulic machines with maximum load capacities from 10 to 1000 kN. Load condition is cantilever bending with upper weld toe cracking, see Figure 4.4.

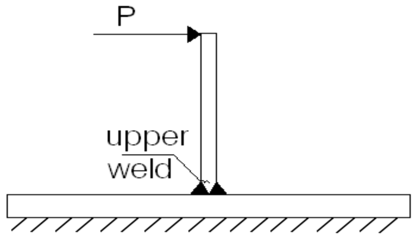


Figure 4.4 Loading condition of detail No.2

- Detail No.3 (R. OLIVIER & W.RITTER (1981) [49])

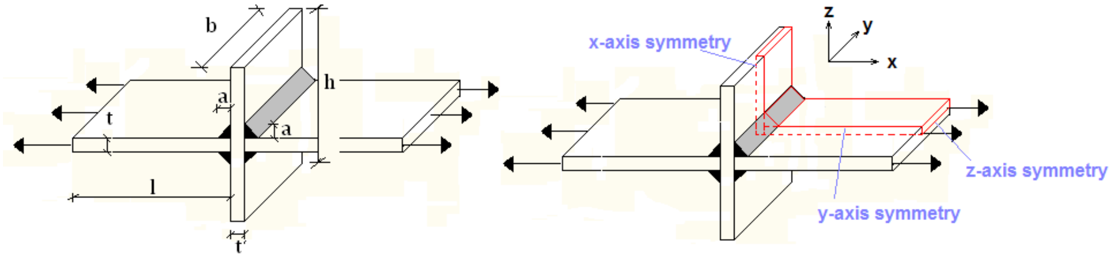
  - Material

According to the Catalogue of S-N-curves of welded joints in structural steels part 3: Cruciform joint from R.Olivier and W.Ritter (1981) [49], the test of cruciform joint in toe failure was chosen as target. There are four different materials in 9 tests; BS 15, ST 37, RST 37-2 and BS 4360 respectively. The original data are shown in Table 4.5.

Table 4.5 Material and experimental information of detail No.3

Thickness [mm]	Welding	Electrode	Leg length [mm]	Width [mm]	R	Plate Material	Ref. [49] Page
7.9	MANNAL ARC	Rutile	12.7	101.6	0	BS 15	80
12	MANNAL ARC	Low hydrogen	11.3	50	0.26	ST 37	23
14	MANNAL ARC	Rutile	16	70	-1	RST 37-2	294
16	MANNAL ARC	Ilmenite	18	60	0.11	SM 41	58
25	MANNAL ARC	Preheat to 428 K	19	100	0	BS 4360	326
25	MANNAL ARC	Preheat to 428 K	22	100	0	BS 4360	328
32	MANNAL ARC	Ilmenite	30	60	0.13	SM 41	64
38	MANNAL ARC	Preheat to 428K	24	100	0	BS 4360	330
38	MANNAL ARC	preheat to 428K	33	100	0	BS 4360	332

- <u>Specimens geometry</u>



t (mm)	t' (mm)	l (mm)	b (mm)	h (mm)	a (mm)
7.9	38	91.7	101.6	64.9	12,7
12	12	131.3	50	82.6	11.3
14	14	156	70	102	16
16	16	178	60	116	18
25	25	272	100	169	22
25	25	272	100	169	19
32	32	350	60	220	30
38	38	404	100	238	24
38	38	412	100	254	32

Figure 4.5 Geometry, welding detail and loading condition of the detail No.3

- <u>Welding condition</u>

  The welding technique is manual arc welding. However, Rutile electrode was applied in 7.9mm and 14mm specimens; low hydrogen electrode was used in 12mm specimens, ilmenite electrode was applied in 16mm and 32mm specimens and preheat to 428K electrode was used in 25mm and 38mm specimens. See Table 4.5 above. Varying leg lengths (a) were used in the 25mm and 38mm specimens.

- <u>Experimental</u>

  Experimental conditions are not mentioned in the literature for this detail. Specially, for the specimen of t=7.9mm, the attachment plate thickness is 38mm, but others are the same as main plate.

The common condition of these three details is testing environment (in air). The detail in seawater will not be discussed in this report. And a clearer table is established to summarise the description of different details, see Table 4.6.

Table 4.6 Summarization of details' information

	Weld treatment	Loading condition	Material	Throat angle	Type of weld
Detail No.1	Manual Metal Arc welding	4 point pure bending	Mg-steel	60	Full penetration fillet
Detail No.2	Manual Metal Arc welding	Cantilever bending	Low-carbon micro-alloyed steel	45	Full penetration fillet
Detail No.3	Manual arc welding	Axial load	Varies steel	45	Fillet weld

5 Modelling and Analysis

In this chapter, thickness effect will be discussed and the thickness correction factor will be calculated by using two approaches. In section 5.1, the classical method, nominal stress approach will be used. The way of using this method to calculate n (thickness effect exponent factor) and m (average slope of S-N curves for each detail) factors will be introduced, and the results will be discussed in view of previous works. The results from the three details' studied will also be compared with each other, see Table 4.1. In section 5.2, a new method is used to study the thickness effect by using hot spot stress, but actually, the methodology is totally the same. The only difference is how to get the stress, and here hot spot stress is used. After presenting the FEM modelling, two methods calculating hot spot stresses are introduced, these are the extrapolation method and the one-millimetre method. The latter one was supposed to have already considered the thickness effect, which means that it shouldn't need a thickness correction factor at all. Finally, a small conclusion will be given for this chapter.

5.1 Nominal stress approach analysis

As noted previously, the thickness effect correction factor was first obtained by Gurney (1981) [18]. The details with thickness between 16mm and 32mm plate were tested, and the tests indicated a small influence of thickness, but then when 6mm thick joints were tested. It showed that the thicker plates, between 16 mm and 32 mm, had a remarkable reduction in strength. There are indications that the results for the thicker joints will show a further reduction in strength. Later, Gurney assumed not only plate joints but also tubular joints and even for other type of welded joints will behave in the same way with respect of the thickness effect. Then he tried to derive a suitable correction factor for this. Figure 5.1 shows the test data from Gurney. And an empirical relationship between thickness and fatigue strength at 2 million cycles of the form, see Equation 3.1 were established.

Gurney also noted that Figure 5.1 does not include some relevant data obtained by Wildschut et al [25] which indicated a lack of thickness effect in T butt specimens of 40 and 70mm thickness. However, the data were limited and in any case they fall within the scatter of data in Figure 5.1. In his article, the reference thickness was chosen as 32mm

Gurney (1981) [18] claimed that the actual correction factors for tubular and non-tubular joints would have to be different, and the factors would vary with $t^{0.25}$.

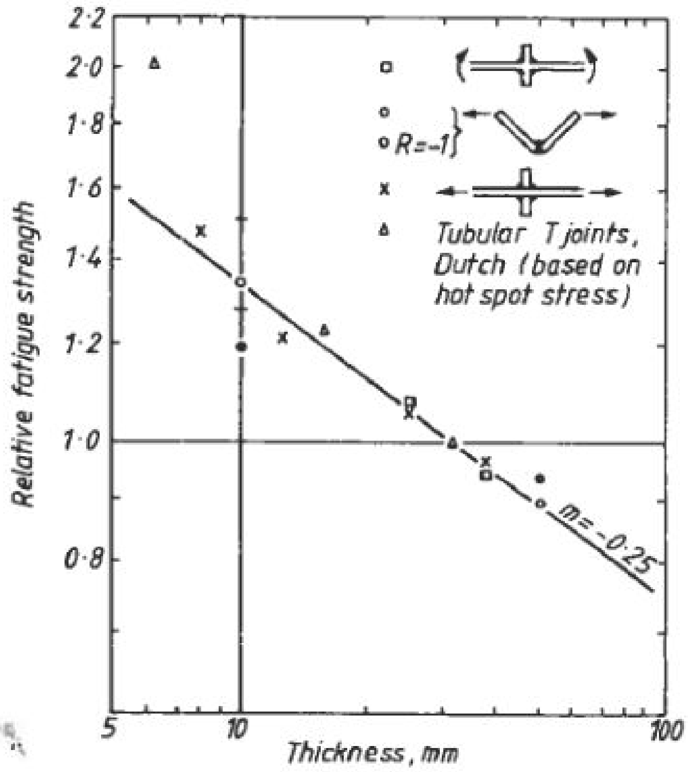


Figure 5.1 Influence of plate thickness on fatigue strength (normalised to a thickness of 32mm). All test at R=0 except where stated [18].

Gurney's method of obtaining the thickness correction factor will be used to study the results from the details in Table 4.1. Here, the detail No.1 was chosen to be an example to show how the thickness correction factor can be derived. And other details will be treated in the same manner.

Firstly, based on the test data from Table 4.2, four different S-N curves can be plotted for four different thicknesses with log-log scale, see Figure 5.2.

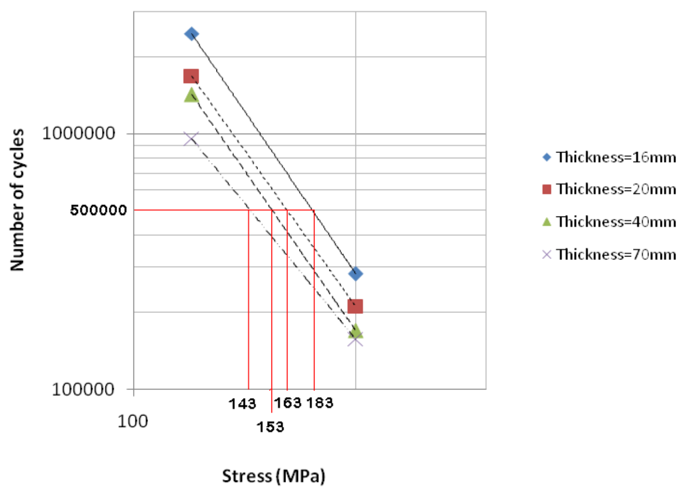


Figure 5.2 S-N curves for four different thicknesses of detail No.1

Then, using any mathematical program, e.g. excel, the power relationship of this curve can be formulated, shown in Table 5.1. The exponent value is so called m factor for S-N curve. And the average m value for this detail is m=3.99825.

Table 5.1 Formulation derived from S-N curves of detail No.1

Thickness	16mm	25mm	40mm	70mm
Formula	$y = 2E+15x^{-4,242}$	$y = 5E+14x^{-4,066}$	$y = 6E+14x^{-4.156}$	$y = 2E+13x^{-3,529}$

According to Gurney’s method, he fixed the number of cycles at 2 million cycles, but in this case, for the lowest curve, the highest test data is below 2 million. It means if the number of cycles fixed at 2 million, the stress will be chosen at the extended line out of the test data, which might give an inaccurate value. So in this case, the point of 0.5 million cycles was chosen, and the stresses at this cycle level were written down in Table 5.2.

Table 5.2 Stresses for different thicknesses at 0.5 million cycles for detail No.1

Thickness	16mm	25mm	40mm	70mm
Nominal Stress (MPa)	183	163	153	143

Finally, plot a curve in log-log scale with x coordinate of thickness and y coordinate of Stress at 0.5 million cycles, see Figure 5.3. Using mathematical program to formulate this relationship and obtain the thickness correction formula. The exponent value is the correction factor n for this certain detail, which supposes to be -0.25 by Gurney.

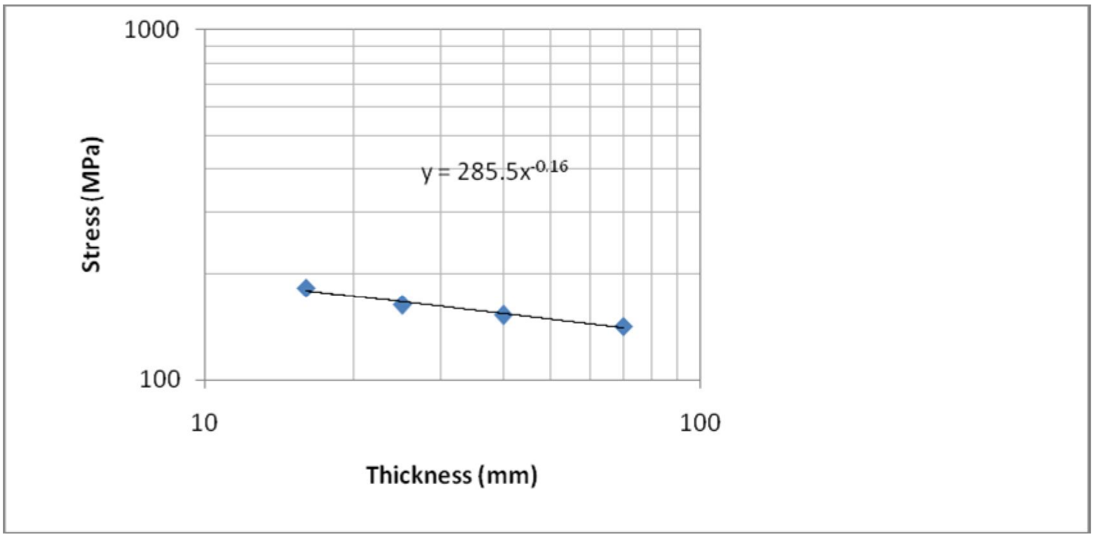


Figure 5.3 The thickness and stress relationship at 0.5 million cycles of detail No.1

From the above figure, is shows the test points fit the power relation line very well.

According to this methodology, nine details in Table 4.1 are calculated, and all the result data are available in that table. Here, for Detail No.2 and Detail No.3 more detailed information will be shown as following tables and figures.

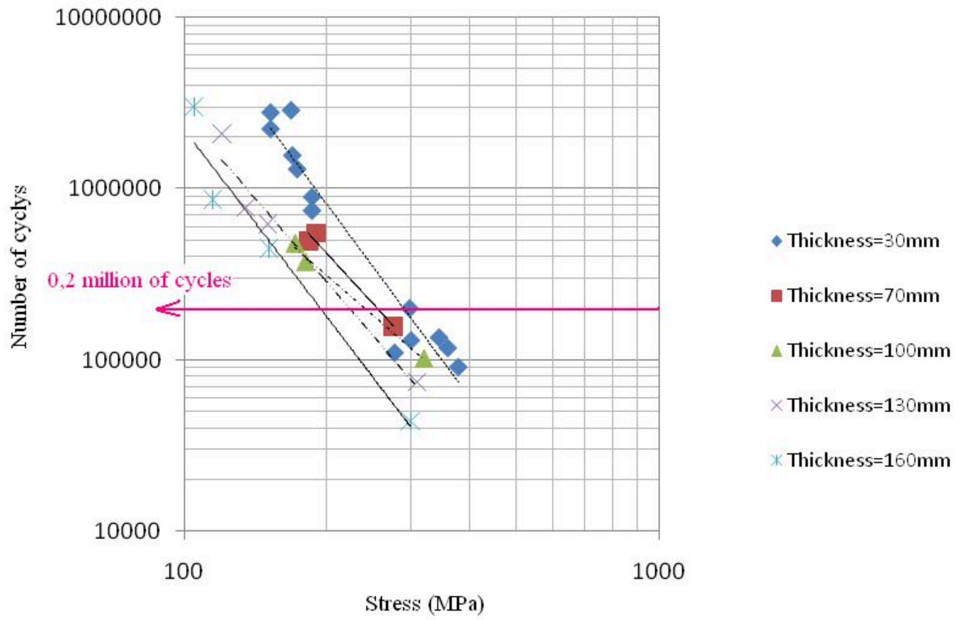


Figure 5.4 S-N curves for four different thicknesses of detail No.2

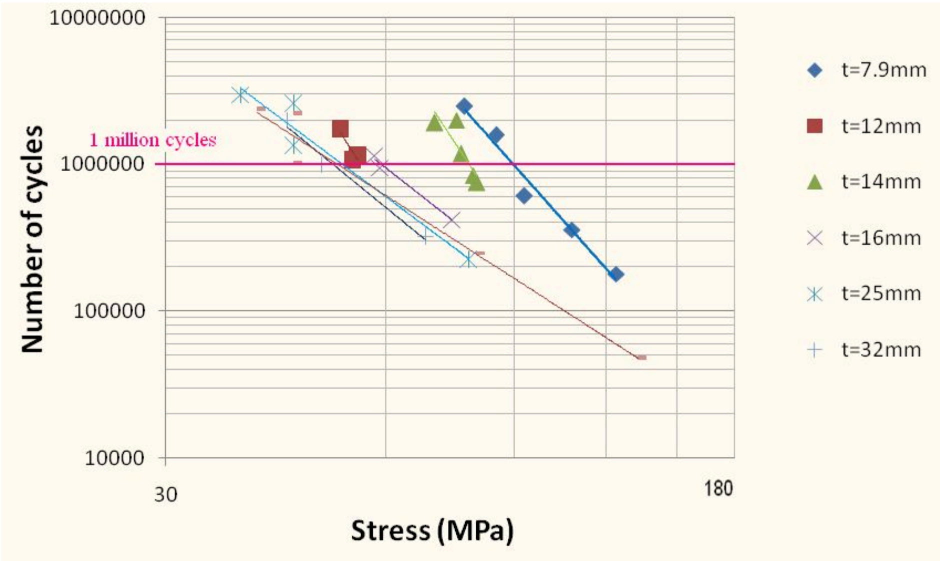


Figure 5.5 S-N curves for four different thicknesses of detail No.3

Then, for detail No.2, the number of cycles was fixed at 0.2 million cycles. Moreover, for detail No.3, this value was 1 million cycles. S-N curves formulation and stresses at certain number of cycles for these two details are shown in Table 5.3 and Table 5.4.

Table 5.3 S-N curves formulation and stresses at 0.2 million cycles for detail No.2

Thickness	30mm	70mm	100mm	130mm	160mm
Formula	$3E+14x^{-3.74}$	$4E+12x^{-3.02}$	$1E+11x^{-2.39}$	$8E+12x^{-3.22}$	$4E+13x^{-3.63}$
N.S.(MPa)	284	262	242	230	194

Table 5.4 S-N curves formulation and stresses at 1 million cycles for detail No.3

Thickness	7.9mm	12mm	14mm	16mm	25mm	32mm	38mm
Formula	$8E+16x^{-5.57}$	$4E+19x^{-7.82}$	$2E+20x^{-7.51}$	$8E+12x^{-3.90}$	$2E+12x^{-3.70}$	$8E+12x^{-4.03}$	$3E+11x^{-3.20}$
N.S.(MPa)	91	55	80	59	50	52	51

For detail No.3 of 12mm and 14mm thick plate, the m factors are quite large compared with 3, which is S-N relation recommended value. This means the data might be not so reliable for these two S-N curves. However, fortunately, the tests data at these two specimens are close to 1 million cycles, and the stress value at this number of cycles can be reliable.

After obtaining the relations between nominal stress at certain number of cycles and thicknesses, the thickness correction factors can be easily derived, see Figure 5.6 and Figure 5.7.

Further discussion of exponent of nominal stress is shown in Chapter 6, with the comparison of that of the hot spot stress.

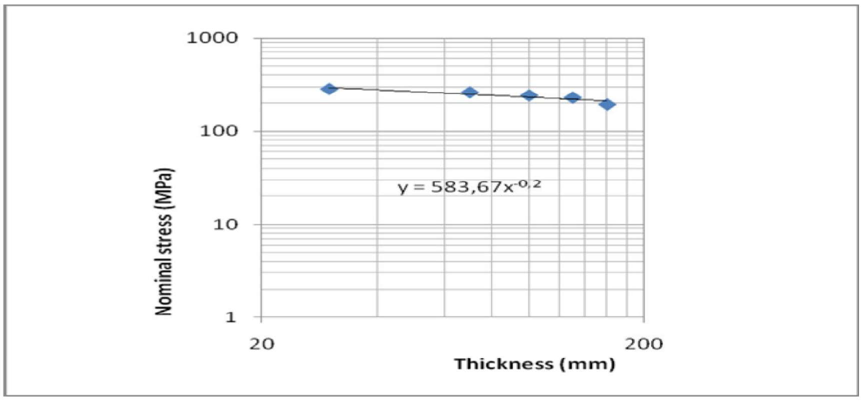


Figure 5.6 The thickness and stress relationship at 0.2 million cycles of detail No.2

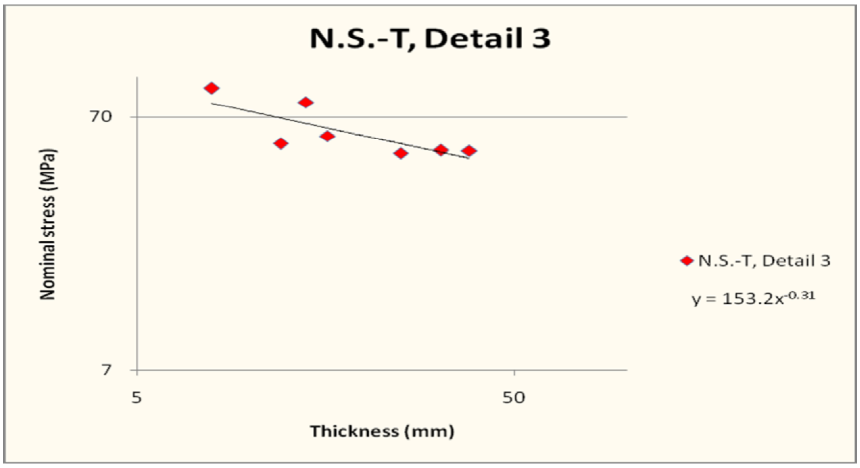


Figure 5.7 The thickness and stress relationship at 1 million cycles of detail No.3

All of these results will be concluded and compared in Section 5.3.

5.2 Hot spot stress approach analysis

The methodology of this approach is the same as the one used for the nominal stress approach. Here, the emphasis will be on how to obtain a reasonable hot spot stress. According to section 2.2.3.2, there are more and 6 methods available to obtain the hot spot stress. In section 5.2.2, two of them will be picked out and used for further study.

5.2.1 FEM-modelling method

To calculate the hot spot stresses for the specimens, every specimen is modelled using the finite element analysis program, ABAQUS 6.8-2. The program includes solving linear and nonlinear static and dynamic structural problems. For each detail, two types of elements are used for modelling two different models. Firstly, 3-D solid element is used for modelling 3-D model. In addition, 2-D model is made using 2-D plain strain shell element. The following sections introduce all the important input data to establish these two models. Here taking detail No.1 as an example and other two details follow the same procedures.

5.2.1.1 3-D model

In order to model the specimens as close to the reality as possible, the models are firstly built up with 20-nodes 3-D solid elements. For saving computer resources, the models are established symmetric with x-z plane and y-z plane, see Figure 5.6.

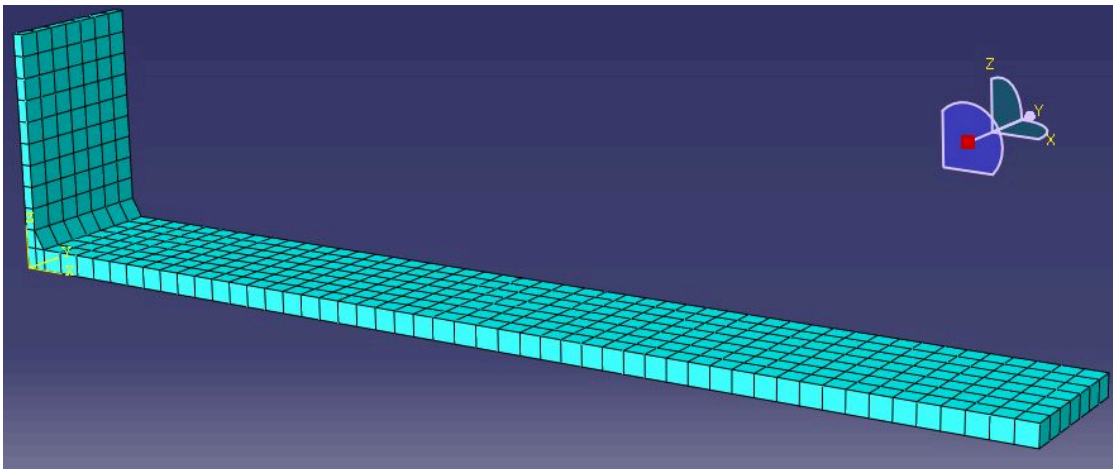


Figure 5.6 3-D solid model of Detail No.1 for 16mm thick plate.

Three parts are created, which are attachment, main-plate and weld. The type of these parts is defined as 3D deformable solid extrusion element.

Gurney has claimed that the fatigue strengths of similar welded details in mild steel and high tensile steel can be represented by the same S-N curve. Therefore, the material properties used for all models are chosen to be the same. The material properties as isotropically elastic with:

Elastic modulus, E=210000 MPa and Poisson's ratio, ν=0.3

The aspect ratio between largest and smallest dimensions for an element is limited up to 3. The welds are easy to create and incorporate in the models.

The boundary condition at the symmetry plane x-z is set to be U2=UR1=UR3=0, and at y-z plane it is defined as U1=UR2=UR3=0. At the edge of the specimen, the movement of the top edge is fixed at z direction. The load is calculated to insure the resulting nominal stress, at the point far away from the weld, the same as experimental nominal stress. It is transformed to be a pressure loaded at the bottom and middle of the main-plate. See Figure 5.7

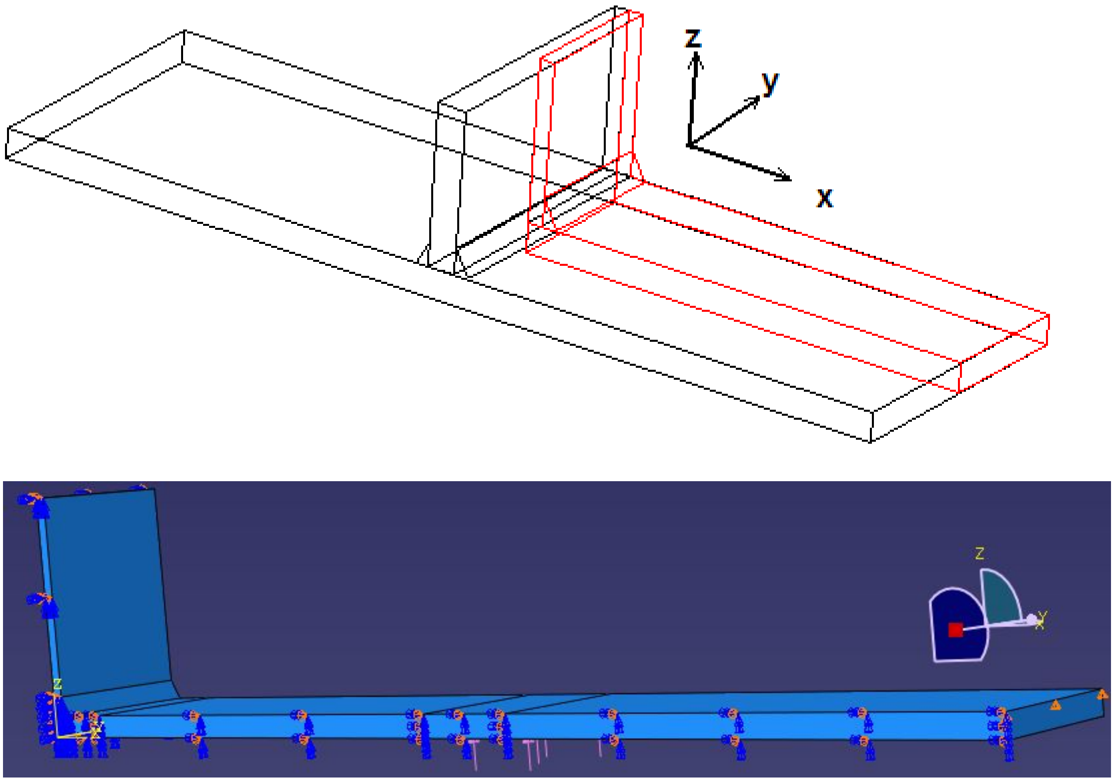


Figure 5.7 Load and boundary condition of detail No.1 for 3-D model

The model is meshed in three different ways in order to fulfil the later analysis, which are called, coarse mesh, fine 2-point mesh and fine 3-point mesh. It will discussed further in Section 5.2.2

After creating the jobs, and submitting the files, the results are available. See Figure 5.8. The nominal stress of this model is checked at the top surface far away from weld until reaching the loading area, and it is the same as experimental nominal stress which means the model input data is correct. Data analysis will be in Section 5.2.3.

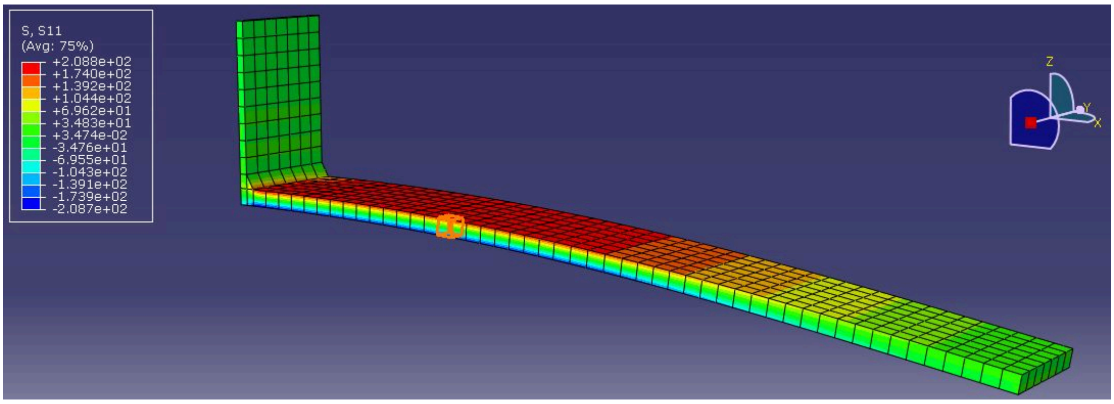


Figure 5.7 Result of detail No.1 for 3-D model

It is generally accepted that a finite element model with 20-node 3-D solid elements including the weld is the most realistic model, Wei and Maddox (2007) [51]. So in the following discussion, this model will be accepted as relatively "true" model compared with 2-D model.

5.2.1.2 2-D model

A simplified 2-D plain strain model is also established in order to compare the results with 3-D solid model. Furthermore, in order to obtain "1 mm stress", if 3-D model is created it will be very time consuming. But using 2-D model, it will not only save time but also give an accurate value compared with the result from 3-D model. So 2-D plain strain model will be used for the "1 mm method".

For a long "extruded body", the load and cross-section are constant along the weld, using a special membrane elements with a plane strain formulation is enough to present this detail. The strain in the longitudinal axis of the body is set to zero and all shear components (strains and stresses) out of the plane are equal to zero. This type of elements allows the model to be made considerably smaller than if the entire joint were modelled using solid elements. [8] In order to save computer resources, the models are established symmetric with y-axis, see Figure 5.8.

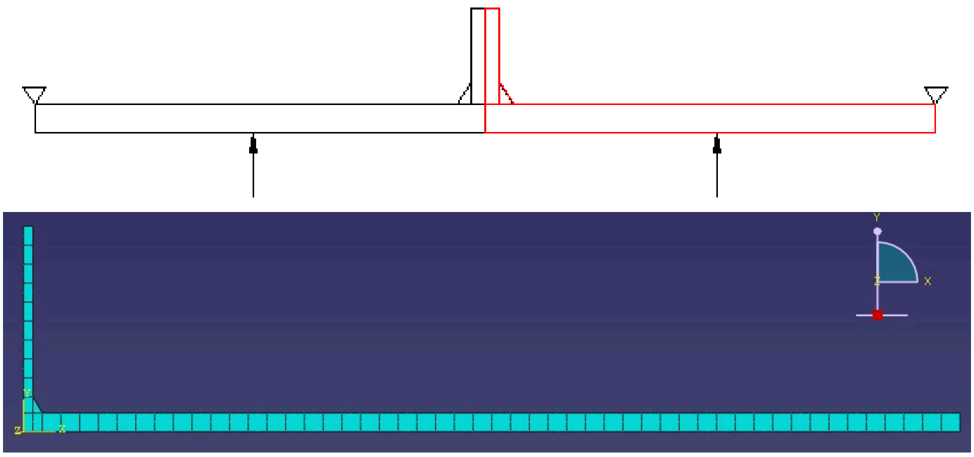


Figure 5.8 2-D model of detail No.1 for 16mm thick plate.

Three parts are created, which are attachment, main-plate and weld. The type of these parts is defined as 2D planar deformable shell element.

The material properties as isotropically elastic with: Elastic modulus, E=210000 MPa, Poisson's ratio, ν=0.3, which are the same as previous 3-D model.

The aspect ratio between largest and smallest dimension for an element is limited up to three. The welds are easy to create and incorporated in the models.

The boundary condition at symmetric line is set to be U1=UR2=UR3=0. At the edge of the specimen, the movement of the top edge is fixed at z direction. The load is calculated to insure the resulting nominal stress, at the point far away from the weld, the same as experimental nominal stress. A concentrated force is applied at the bottom and middle of the main-plate. See Figure 5.9

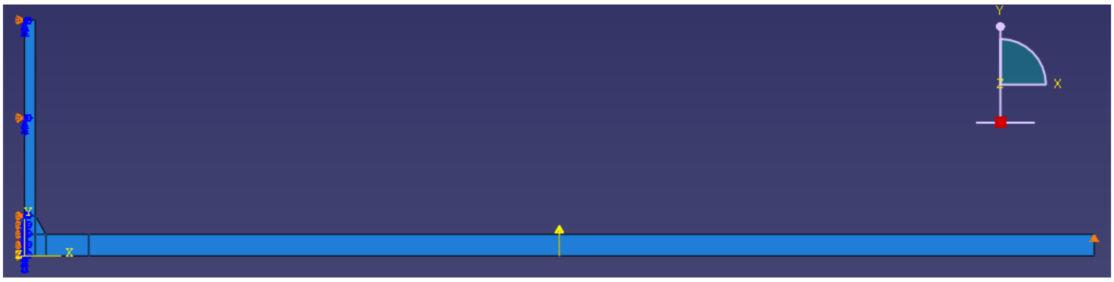


Figure 5.9 Load and boundary condition of Detail No.1 for 2-D model

The "1 mm method's" element type is also used in 2-D model, besides the extrapolation method mesh. After creating the jobs, and submitting the files, the results are available. See Figure 5.10. The nominal stress of this model is checked, and it is the same as experimental nominal stress, which means the model input data is correct. Data analysis will be in Section 5.2.3.

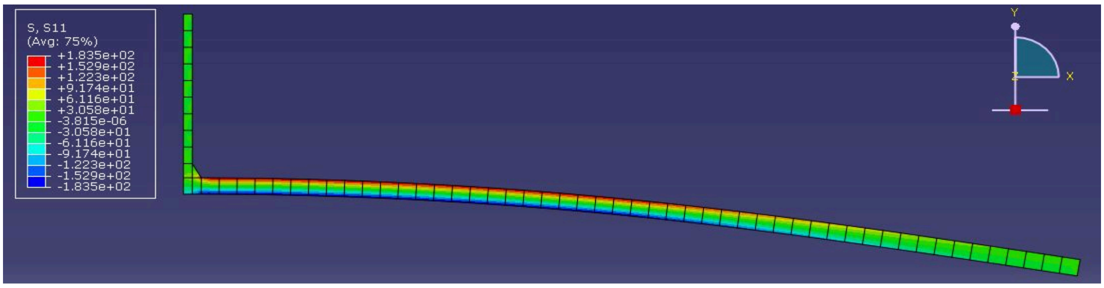


Figure 5.10 Result of Detail No.1 for 2-D model

5.2.2 The approaches to obtain hot spot stress

The hot spot stress of five typical details can be calculated using FE analyses are shown in Figure 5.11.

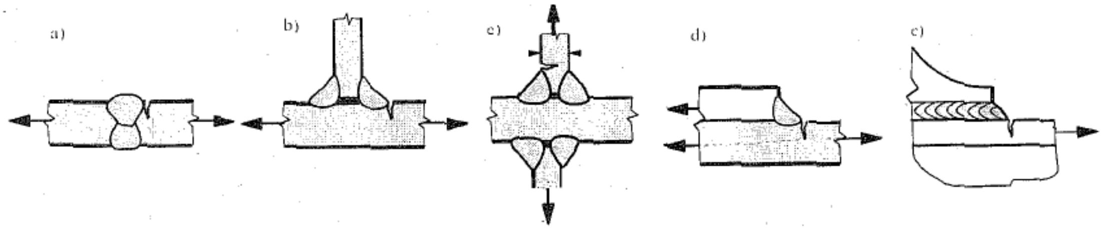


Figure 5.11 Locations of crack propagation in weld joints

For the determination of structural hot spot stress, two types of these details are distinguished, see Figure 5.12:

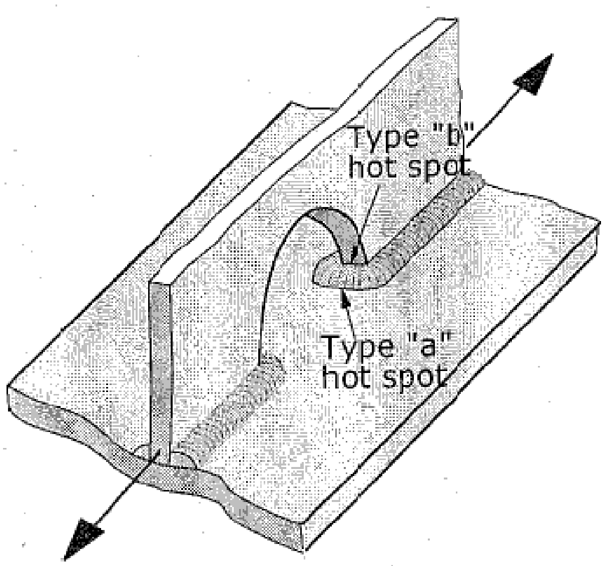


Figure 5.12 Examples of the two hot spot types [16]

5.2.2.1 Extrapolation method

In section 2.2.3.2, six approaches to obtain hot spot stress has been mentioned. And the linear stress extrapolation method and the quadratic stress extrapolation method are the most commonly used methods. It is recommended to using quadratic type elements, which are 8 nodes shell element for 2-D model and 20 nodes solid element for 3-D model. The recommendation from IIW for how to calculate the hot spot stress of these two kinds of weld details is given as follow, see Table 5.5 and 5.6.

Type (a): stress transverse to weld toe on plate surface

Table5.5 Linear extrapolation cases a1, a2 and a3

Case	Mesh size	Extrapolation point	Recommendation
a1	Fine mesh (2-point)	0.4T and 1.0T Nodal points	0.4T at the hot spot σ_{hs}=1.67 $\sigma_{0.4t}$ -0.67 $\sigma_{1.0t}$
a2	Fine mesh (3-point)	0.4T, 0.9T, and 1.4T Nodal points	where the hot spot tends towards a singularity σ_{hs}=2.52$\sigma_{0.4t}$ -2.24 $\sigma_{0.9t}$+0.72$\sigma_{1.4t}$
a3	Coarse mesh	0.5T and 1.5T Mid-side points	higher order 3-D elements sized σ_{hs}=1.50 $\sigma_{0.5t}$ -0.50 $\sigma_{1.5t}$

Type (b): stress transverse to weld toe on plate edge. For type (b) hot spots, two different extrapolations of FEM stress to the weld toe (or mid-plane intersection in the case of thin-shell elements) are allowed:

Table5.6 Linear extrapolation cases b1 and b2

Case	Mesh size	Extrapolation point	Recommendation
b1	Fine mesh (3-point)	4mm, 8mm, 12mm Nodal points	Quadratic extrapolation σ_{hs}=3σ_{4mm} -3 σ_{8mm}+σ_{12mm}
b2	Coarse mesh	5mm and 15 mm Mid-side points	Midside stresses with higher order elements σ_{hs}=1.50 σ_{5mm} -0.50 σ_{15mm}

The element size for coarse mesh of detail type (a) is defined as t*t, and for detail type (b) is 10mm*10mm. And for relatively fine mesh of type (a) detail is 0.4t*t, for detail type (b) is 4mm*4mm.

Figure 5.13 shows reference points at different types of meshing.

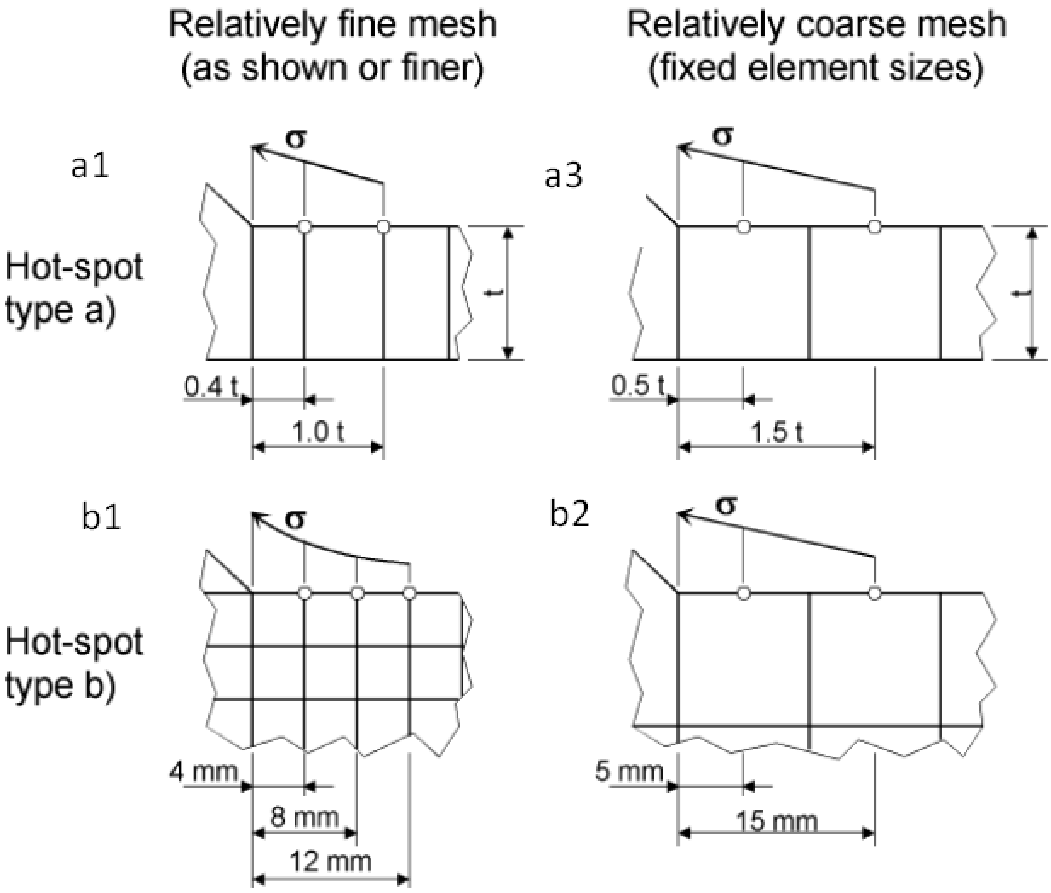


Figure 5.13 Reference points at different types of meshing

It should be mentioned that in this thesis all these three details are type (a) detail.

5.2.2.2 "1 mm method"

Another approach that considers the thickness effect quite well is the one millimeter approach, Xiao and Yamada (2004) [10], by computing stress that 1 mm below the surface in the direction corresponding to the expected crack path and the data range of cruciform joints. This method is based on a generalized crack propagation analysis for weld toe cracks. The drawback of this method is 1 mm fine mesh is very time consuming. Furthermore, the one-millimeter method needs similar geometry as reference details to compare with. Xiao and Yamada (2004) [10] concluded when compared to the surface extrapolation technique for structural hot spot stress evaluation, the "1 mm method" has the additional advantage in that it is able to account for the size and thickness effect observed in welded joints.

Xiao and Yamada (2004) [10] also found that for non load carrying cruciform joints models with different sizes of element, in surface direction, convergence occurs at about 2mm away from weld toe, while in thickness direction, the converging point is less than 1 mm away from weld toe, see Figure 5.14. This indicates that the affecting range of weld toe geometries is rather localized. And the stress values at these convergence points are more stable and neglect the non-linear stress part.

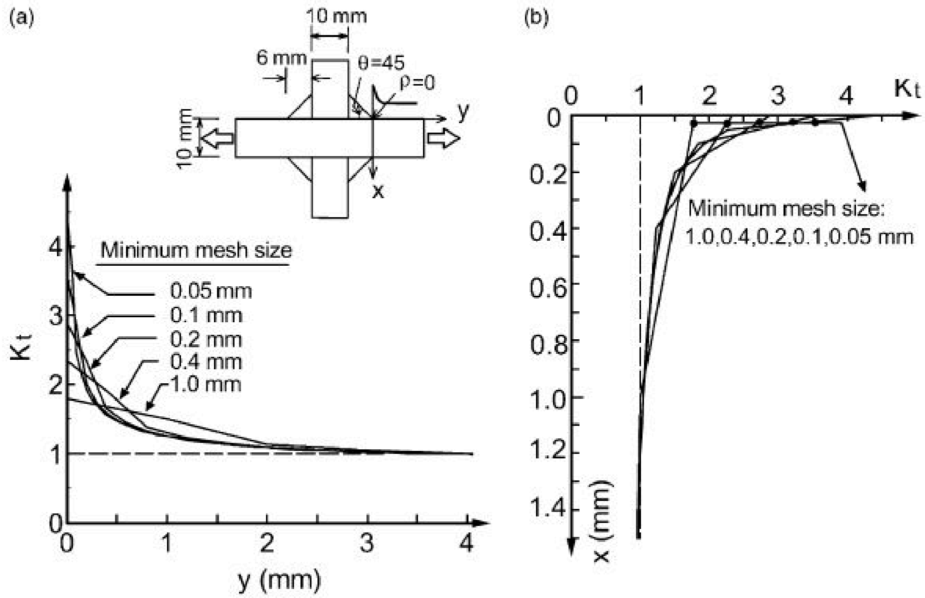


Figure 5.14 Stresses calculated with different mesh size: (a) along surface; (b) through thickness.

However, it should be pointed out that the "1 mm method" cannot be used for the cases that bending stresses are dominant, especially when the thickness of main plate is small, Xiao and Yamada (2004) [10], which means only detail No.3 can be checked using this method, since other two details are bending stress dominated. Xiao and Yamada recommended using only first order elements but not quadratic elements. The reason for this is that the notch singularity at the weld toe which influences the stress gradients drastically.

The other four methods outlined in Section 2.2.3.2 will not be applied in this thesis, and there are even other more methods are recommended, e.g. Fricke published one point stress determination to further simplification. Fricke recommends that the hot spot stress at a position 0.5t from the weld toe and assessed with a design S-N curve reduced by one fatigue class (or increasing the $\sigma_{0.5}$ by multiply 1.12, DNV 2008).

5.2.3 Result from HSS approach analysis

In section 5.1, for nominal approach analysis, the different nominal stress (or fatigue strength at certain number of cycles) for plates with different thicknesses have already calculated for each detail. Therefore, in HSS approach analysis, the first step is to calculate the right hot spot stress that should ensure that original membrane stress (or fatigue strength at certain number of cycles) at weld toe in FE model is the same as the experimental nominal stress. In addition, the next step is to plot log-log scale curve in the relation between hot spot stresses and thicknesses. Finally, the thickness effect correction factor can be easily obtained.

5.2.3.1 Calculating hot spot stress

According to IIW (2008) [3], the hot spot stress should be obtained from the FE analysis. Maximum stress perpendicular to the weld toe was picked out to be fatigue stress, e.g. S11 for detail No.1, see Figure 5.7. Extrapolation points followed IIW, see Table 5.5. For 3-D model, coarse mesh stress, fine mesh two point stress and fine mesh three point stress were calculated. For 2-D plain strain model, except previous three stresses, “1 mm stresses” were picked out as well.

5.2.3.1.1 Detail No.1

Table 5.7 to Table 5.10 show the result of the hot spot stresses for each detail as obtained from 3-D models. The top of the tables show the input data and detail information. Pressure width means the length of loading area. The nominal stress is chosen from experimental S-N data, which represents the fatigue strength at a given number of cycles. The nominal stress in detail No.1 is fixed at 0.5 million cycles. Pressure is the applied load at the loading area, according to the fixed N.S. and the geometry.

Table 5.7 Hot spot stress by using 3-D model for 16mm thick plate of detail No.1

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
16		70	183	0,279576	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	24, 0, 16	208,793	214,412	1,168638
	1,5t	40, 0, 16	197,555		
Fine mesh 2 point S11 (0,4t*t)	0,4t	22,4, 0, 16	209,474	214,5673	1,169485
	1,0t	32, 0, 16	201,872		
Fine mesh 3 point S11 (0,4t*t & 0,5t*t)	0,4t	22,4, 0, 16	213,777	231,0068	1,259087
	0,9t	30,4, 0, 16	201,013		
	1,4t	38,4, 0, 16	197,997		

Table 5.8 Hot spot stress by using 3-D model for 25mm thick plate of detail No.1

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
25		70	163	0,608203	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	37,5, 0, 25	179,82	185,037	1,131829
	1,5t	62,5, 0, 25	169,386		
Fine mesh 2 point S11 (0,4t*t)	0,4t	35, 0, 25	180,421	185,5304	1,134848
	1,0t	50, 0, 25	172,795		
Fine mesh 3 point S11 (0,4t*t & 0,5t*t)	0,4t	35, 0, 25	180,44	186,9425	1,143485
	0,9t	47,5, 0, 25	174,003		
	1,4t	60, 0, 25	169,445		

Table 5.9 Hot spot stress by using 3-D model for 40mm thick plate of detail No.1

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
40		70	153	1,456823	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	60, 0, 40	162,048	165,7455	1,083542
	1,5t	100, 0, 40	154,653		
Fine mesh 2 point S11 (0,4t*t)	0,4t	56, 0, 40	162,105	165,9803	1,085077
	1,0t	80, 0, 40	156,321		
Fine mesh 3 point S11 (0,4t*t & 0,5t*t)	0,4t	56, 0, 40	162,032	167,5899	1,0956
	0,9t	76, 0, 40	157,133		
	1,4t	96, 0, 40	154,51		

Table 5.10 Hot spot stress by using 3-D model for 70mm thick plate of detail No.1

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
70		70	143	4,159634	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	105, 0, 70	147,076	149,0385	1,045033
	1,5t	175, 0, 70	143,151		
Fine mesh 2 point S11 (0,4t*t)	0,4t	98, 0, 70	147,064	149,133	1,045696
	1,0t	140, 0, 70	143,976		
Fine mesh 3 point S11 (0,4t*t & 0,5t*t)	0,4t	98, 0, 70	147,106	150,2642	1,053627
	0,9t	133, 0, 70	144,403		
	1,4t	168, 0, 70	143,083		

Table 5.11 to Table 5.14 show the result of hot spot stresses for each certain thick detail respectively by using 2-D models. Input data and detail information are shown at the top of the tables. The nominal stresses are chosen from experimental S-N data at 0.2 million cycles. Pressure point is in the quarter point of main plate.

Table 5.11 Hot spot stress by using 2-D model for 16mm thick plate of detail No.1

Thickness (mm)		Nominal stress (MPa)		Force (N)	
16		183		4305,469069	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0.5t	24, 16, 0	183,472	183,472	1,000002
	1.5t	40, 16, 0	183,472		
Fine mesh 2 point S11 (0,1t*0,1t)	0.4t	23.2, 16, 0	182,269	181,8456	0,991137
	1.0t	32, 16, 0	182,901		
Fine mesh 3 point S11 (0,1t*0,1t)	0.4t	22.4, 16, 0	182,547	182,9366	0,997084
	0.9t	30.4, 16, 0	182,645		
	1.4t	38.4, 16, 0	183,393		

Table 5.12 Hot spot stress by using 2-D model for 25mm thick plate of detail No.1

Thickness (mm)		Nominal stress (MPa)		Force (N)	
25		163		9366,32182	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0.5t	37.5, 25, 0	163,485	163,485	1,000001
	1.5t	62.5, 25, 0	163,485		
Fine mesh 2 point S11 (0,1t*0,1t)	0.4t	35, 25, 0	162,661	162,45	0,99367
	1.0t	50, 25, 0	162,976		
Fine mesh 3 point S11 (0,1t*0,1t)	0.4t	35, 25, 0	162,661	163,009	0,997089
	0.9t	47.5, 25, 0	162,748		
	1.4t	60, 25, 0	163,415		

Table 5.13 Hot spot stress by using 2-D model for 40mm thick plate of detail No.1

Thickness (mm)		Nominal stress (MPa)		Force (N)	
40		153		22435,07299	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0.5t	60, 40, 0	152,967	152,967	1,000004
	1.5t	100, 40, 0	152,967		
Fine mesh 2 point S11 (0,1t*0,1t)	0.4t	56, 40, 0	152,196	151,999	0,993676
	1.0t	80, 40, 0	152,49		
Fine mesh 3 point S11 (0,1t*0,1t)	0.4t	56, 40, 0	152,196	152,5222	0,997096
	0.9t	76, 40, 0	152,277		
	1.4t	96, 40, 0	152,901		

Table 5.14 Hot spot stress by using 2-D model for 70mm thick plate of detail No.1

Thickness (mm)		Nominal stress (MPa)		Force (N)	
70		143		64058,36302	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0.5t	105, 70, 0	142,616	142,616	1
	1.5t	175, 70, 0	142,616		
Fine mesh 2 point S11 (0,1t*0,1t)	0.4t	98, 70, 0	141,897	141,7128	0,993666
	1.0t	140, 70, 0	142,172		
Fine mesh 3 point S11 (0,1t*0,1t)	0.4t	98, 70, 0	141,897	142,1983	0,997071
	0.9t	133, 70, 0	141,974		
	1.4t	168, 70, 0	142,555		

An interesting finding here is the hot spot stresses calculated by 2-D are all smaller than nominal stresses which cannot be used. This phenomenon will be discussed later in Chapter 6.

"1 mm method" hot spot stresses are shown in Table 5.15. And the convergence study of how fine mesh can obtain relatively accurate hot spot stress will be in Chapter 6. It shows that 0.5 mm*0.5 mm mesh can generate an accurate stress which has only 5% difference from 0.1 mm*0.1 mm mesh result.

Table 5.15 “1 mm method” hot spot stress by using 2-D model of detail No.1

Thickness (mm)	“1 mm stress” (MPa)	Nominal stress (MPa)	3-D coarse mesh hot spot stress (MPa)	Ratio (“1 mm stress”/Nominal stress)
16	169	183	214	0,91897
25	179	163	185	1,0949
40	198	153	166	1,2963
70	226	143	149	1,5841

Table 5.15 shows the “1 mm stress” is quite different from other hot spot stresses. “1 mm stress” increasing with thickness, but other hot spot stresses and nominal stresses are on the contrary. Since this “1 mm” stress can not represent the fatigue stress very well, it will not be used for fatigue assessment.

The locations of obtaining the hot spot stresses and the distributions of the stresses along the surface and through the thickness will be discussed in Chapter 6.

After getting all the results of the hot spot stress for three details, the calculations of the thickness effect correction factors are shown in Section 5.2.3.2.

5.2.3.1.2 Detail No.2

Since for 2-D plain strain model, the hot spot stresses are smaller than nominal stress, these results will omit here. Only the hot spot stresses from 3-D model are shown in Table 5.16 to Table 5.20. The nominal stresses in detail No.2 are fixed at 0.2 million cycles.

Table 5.16 Hot spot stress by using 3-D model for 30 mm thick plate of detail No.2

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
30		23	284	11,47	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	-15, -30, 30	264,492	294,352	1,036012952
	1,5t	-15, -30, 60	204,772		
Fine mesh 2 point S11 (0,4t*t)	0,4t	-15, -45, 27	273,874	302,27664	1,063904829
	1,0t	-15, -45, 45	231,482		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	-15, -45, 27	273,481	308,31956	1,085173729
	0,9t	-15, -45, 42	237,185		
	1,4t	-15, -45, 57	208,947		

Table 5.17 Hot spot stress by using 3-D model for 70 mm thick plate of detail No.2

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
70		53	262	10,7	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	363, 15,67, 140	242,744	269,9415	1,032083732
	1,5t	363, 15,67, 210	188,349		
Fine mesh 2 point S11 (0,4t*t)	0,4t	363, -19,5, 133	251,479	277,50515	1,061002294
	1,0t	363, -19,5, 175	212,634		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	363, -19,5, 133	251,165	283,44908	1,083728083
	0,9t	363, -19,5, 168	217,811		
	1,4t	363, -19,5, 203	192,236		

Table 5.18 Hot spot stress by using 3-D model for 100 mm thick plate of detail No.2

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
100		75	242	10,02	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	0, 150,5, 100	224,446	249,773	1,030459177
	1,5t	0, 150,5, 200	173,792		
Fine mesh 2 point S11 (0,4t*t)	0,4t	0, 150,5, 90	233,101	257,23306	1,061236272
	1,0t	0, 150,5, 150	197,083		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	0, 150,5, 90	232,809	262,73732	1,083944552
	0,9t	0, 150,5, 140	201,884		
	1,4t	0, 150,5, 190	178,165		

Table 5.19 Hot spot stress by using 3-D model for 130 mm thick plate of detail No.2

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
130		98	230	9,44	
Stress name	Location	Original coordinate	Stress at point (MPa)	HSS (MPa)	Ratio (HHS/NS)
Coarse mesh S11 (t*t)	0,5t	0, 196, 130	212,632	236,624	1,030771911
	1,5t	0, 196, 260	164,648		
Fine mesh 2 point S11 (0,4t*t)	0,4t	0, 196, 117	220,843	243,75834	1,061850235
	1,0t	0, 196, 195	186,641		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	0, 196, 117	220,79	252,78304	1,101163269
	0,9t	0, 196, 192	187,681		
	1,4t	0, 196, 267	162,219		

Table 5.20 Hot spot stress by using 3-D model for 160 mm thick plate of detail No.2

Thickness (mm)		Pressure width (mm)	Nominal stress (MPa)	Pressure (MPa)	
160		120	194	8	
stress name	location	original coordinate	stress at point (MPa)	HSS (MPa)	ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	0, 321,333, 160	179,646	199,7865	1,032221648
	1,5t	0, 321,333, 320	139,365		
Fine mesh 2 point S11 (0,4t*t)	0,4t	0, 241, 144	186,112	205,38187	1,061130819
	1,0t	0, 241, 240	157,351		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	0, 241, 144	185,879	209,77932	1,083850788
	0,9t	0, 241, 224	161,184		
	1,4t	0, 241, 304	142,245		

It is also found that the hot spot stresses in 2-D model are equal to the nominal stress in coarse mesh and smaller than the nominal stress in fine mesh. Further discussion will be presented in Chapter 6.

5.2.3.1.3 Detail No.3

The same reason here, only the hot spot stresses from 3-D model are shown in Table 5.21 to Table 5.29. The nominal stress in detail No.3 is fixed at 1 million cycles.

Table 5.21 Hot spot stress by using 3-D model for 7.9 mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
7,9		91	91	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	99,8097	104,468	1,15218
	1,5t	90,4927		
Fine mesh 2 point S11 (0,4t*t)	0,4t	93,9037	96,8646	1,06832
	1,0t	89,4844		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	94,1284	100,925	1,1131
	0,9t	89,4058		
	1,4t	88,8749		

Table 5.22 Hot spot stress by using 3-D model for 12 mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
12		55	55	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	59,1009	60,9481	1,11057
	1,5t	55,4066		
Fine mesh 2 point S11 (0,4t*t)	0,4t	56,1052	57,2569	1,04331
	1,0t	54,3862		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	55,6086	60,5714	1,10371
	0,9t	52,9801		
	1,4t	54,3238		

Table 5.23 Hot spot stress by using 3-D model for 14 mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
14		80	80	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	86,9867	90,3342	1,12608
	1,5t	80,2917		
Fine mesh 2 point S11 (0,4t*t)	0,4t	82,0389	84,0142	1,0473
	1,0t	79,0907		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	81,0589	88,8226	1,10724
	0,9t	76,9055		
	1,4t	78,9201		

Table 5.24 Hot spot stress by using 3-D model for 16 mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
16		59	59	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	63,6291	65,762	1,11669
	1,5t	59,3634		
Fine mesh 2 point S11 (0,4t*t)	0,4t	60,3299	61,7592	1,04872
	1,0t	58,1966		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	60,4388	64,7294	1,09916
	0,9t	58,1298		
	1,4t	59,2144		

Table 5.25 Hot spot stress by using 3-D model for 25 mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
25		50	50	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	54,285	55,9678	1,10915
	1,5t	50,9194		
Fine mesh 2 point S11 (0,4t*t)	0,4t	51,561	52,5878	1,04217
	1,0t	50,0284		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	51,066	55,5342	1,10056
	0,9t	48,7189		
	1,4t	49,9698		

Table 5.26 Hot spot stress by using 3-D model for 25(b) mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
25		50	50	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	54,1644	55,7851	1,10553
	1,5t	50,923		
Fine mesh 2 point S11 (0,4t*t)	0,4t	51,5153	52,4906	1,04024
	1,0t	50,0596		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	51,1255	55,5238	1,10035
	0,9t	48,7955		
	1,4t	49,9853		

Table 5.27 Hot spot stress by using 3-D model for 32 mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
32		52	52	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	55,9535	57,9888	1,12316
	1,5t	51,883		
Fine mesh 2 point S11 (0,4t*t)	0,4t	53,2771	54,8204	1,06179
	1,0t	50,9736		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	53,3691	57,5777	1,1152
	0,9t	50,9706		
	1,4t	51,7524		

Table 5.28 Hot spot stress by using 3-D model for 38 mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
38		51	51	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	53,7743	54,992	1,06822
	1,5t	51,339		
Fine mesh 2 point S11 (0,4t*t)	0,4t	52,7633	53,857	1,04617
	1,0t	51,1309		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	52,8532	56,0895	1,08954
	0,9t	51,066		
	1,4t	51,7879		

Table 5.29 Hot spot stress by using 3-D model for 38(b) mm thick plate of detail No.3

Thickness (mm)		Nominal stress (MPa)	Pressure (MPa)	
38		51	51	
Stress name	Location	Stress at point (MPa)	H.S.S. (MPa)	Ratio (HSS/NS)
Coarse mesh S11 (t*t)	0,5t	55,6247	57,3656	1,11433
	1,5t	52,143		
Fine mesh 2 point S11 (0,4t*t)	0,4t	52,8521	54,0714	1,05034
	1,0t	51,0322		
Fine mesh 3 point S11 (0,4t*t & 0,5*t)	0,4t	52,9394	56,4682	1,0969
	0,9t	50,9939		
	1,4t	51,7878		

Table 5.30 "1 mm method" hot spot stress by using 2-D model of Detail No.3

Thickness (mm)	"1 mm stress" (MPa)	Nominal stress (MPa)	3-D coarse mesh hot spot stress (MPa)	Ratio ("1 mm stress"/Nominal stress)
7,9	91	91	103	0,99946
12	60	55	61	1,09416
14	91	80	90	1,13844
16	69	59	66	1,17724
25	67	50	56	1,32065
25	66	50	56	1,31594
32	73	52	58	1,41657
38	76	51	55	1,47004
38	76	51	57	1,48365

“1 mm method” hot spot stresses are shown in Table 5.30. And the convergence study of how fine mesh can obtain a relatively accurate hot spot stress will be in Chapter 6. 0.1 mm*0.1 mm mesh and 0.25 mm*0.25mm mesh are used in this detail.

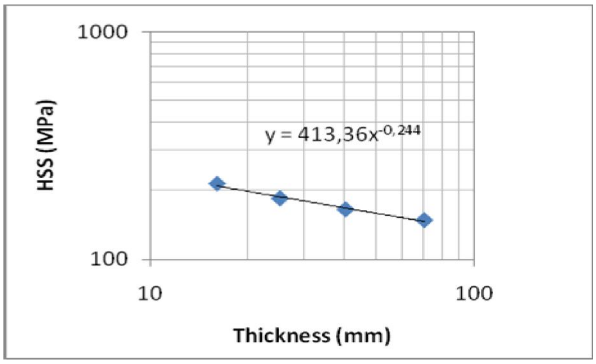
Table 5.30 shows the “1 mm stress” is quite different from other hot spot stresses. The “1 mm stress” tends to the same value whatever the nominal stress changes. It means when the fatigue lives are the same, the “1 mm stresses” for the plates with different thicknesses are similar. There might be no thickness effect correction factor when using the hot spot stress approach, see Chapter 6.

5.2.3.2 Thickness effect factor based on hot spot stress

Since the hot spot stress for each thick plate has already been obtained, the next step is to plot the log-log relation between stress and thickness, and add a trend line. Hot spot stresses from 2-D model are not usable, so the following sections only calculate the thickness correction factor based on hot spot stress of 3-D model. For Detail No.3, “1 mm hot spot stress method” is compared as a comparison.

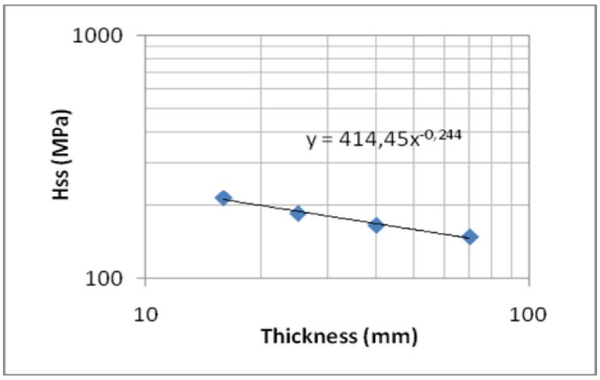
5.2.3.2.1 Detail No.1

Figure 5.15 to Figure 5.17 show the relation between the hot spot stress and the thickness. The correction factors are 0.244, 0.244 and 0.283, larger than 0.166, which is the value from the nominal stress analysis approach. The result will be further discussed in Chapter 6.



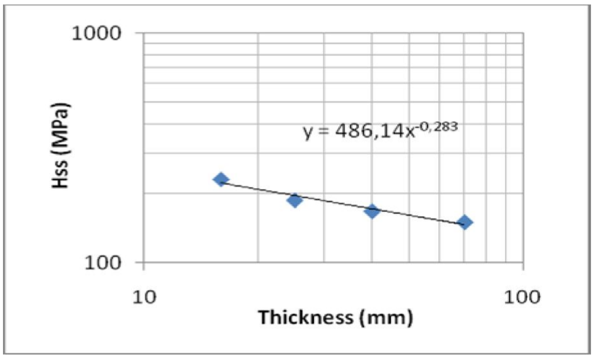
DETAIL No.1 (coarse mesh hotspot)	
N=500000	
Thickness (mm)	Hot spot stress (MPa)
16	214
25	185
40	166
70	149

Figure 5.15 The thickness and stress relationship at 0.5 million cycles of detail No.1 according to coarse mesh hot spot stress



DETAIL No.1 (fine mesh 2 point)	
N=500000	
Thickness (mm)	Hot spot stress (MPa)
16	216
25	186
40	166
70	149

Figure 5.16 The thickness and stress relationship at 0.5 million cycles of detail No.1 according to fine mesh 2 point hot spot stress

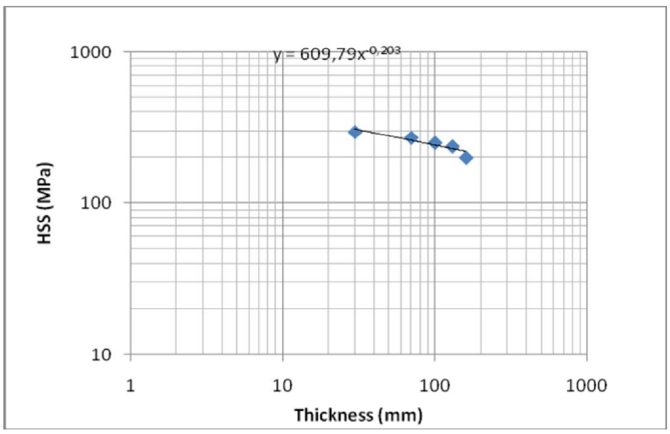


DETAIL No.1 (fine mesh 3 point)	
N=500000	
Thickness (mm)	Hot spot stress (MPa)
16	231
25	187
40	168
70	150

Figure 5.17 The thickness and stress relationship at 0.5 million cycles of detail No.1 according to fine mesh 3 point hot spot stress

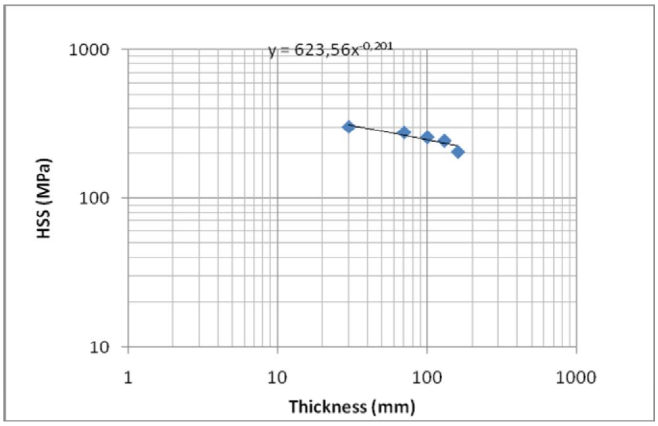
5.2.3.2.2 Detail No.2

Figure 5.18 to Figure 5.20 show the relation between the hot spot stress and the thickness. And the correction factors are 0.203, 0.201 and 0.197, almost equal to 0.2, which is the value from the nominal stress analysis approach.



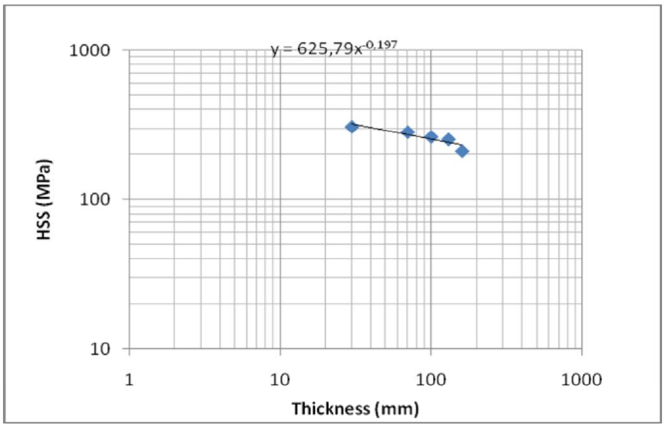
DETAIL No.2 (coarse mesh)	
N=200000	
Thickness (mm)	Stress (MPa)
30	294
70	270
100	250
130	237
160	200

Figure 5.18 The thickness and stress relationship at 0.2 million cycles of detail No.2 according to coarse mesh hot spot stress



DETAIL No.2 (fine mesh 2 point)	
N=200000	
Thickness (mm)	Stress (MPa)
30	302,27
70	278
100	257
130	244
160	205

Figure 5.19 The thickness and stress relationship at 0.2 million cycles of detail No.2 according to fine mesh 2 point hot spot stress

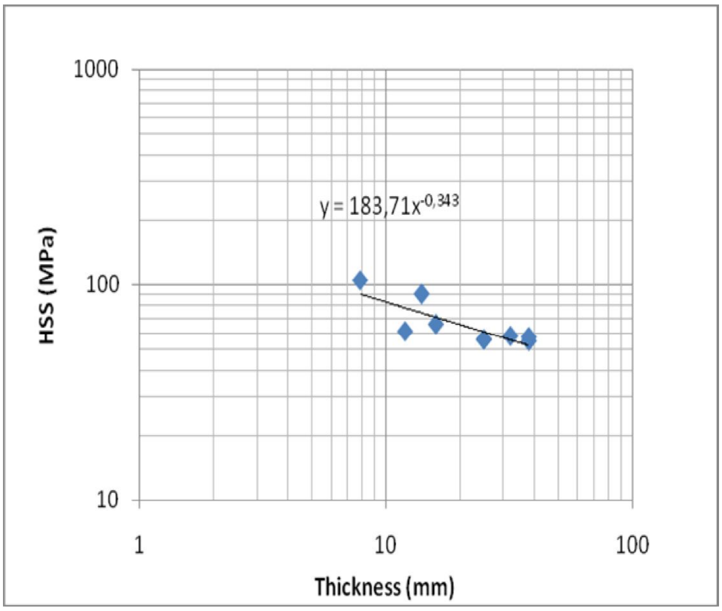


DETAIL No.2 (fine mesh 3 point)	
N=200000	
Thickness (mm)	Stress (MPa)
30	308
70	283
100	263
130	253
160	210

Figure 5.20 The thickness and stress relationship at 0.2 million cycles of detail No.2 according to fine mesh 3 point hot spot stress

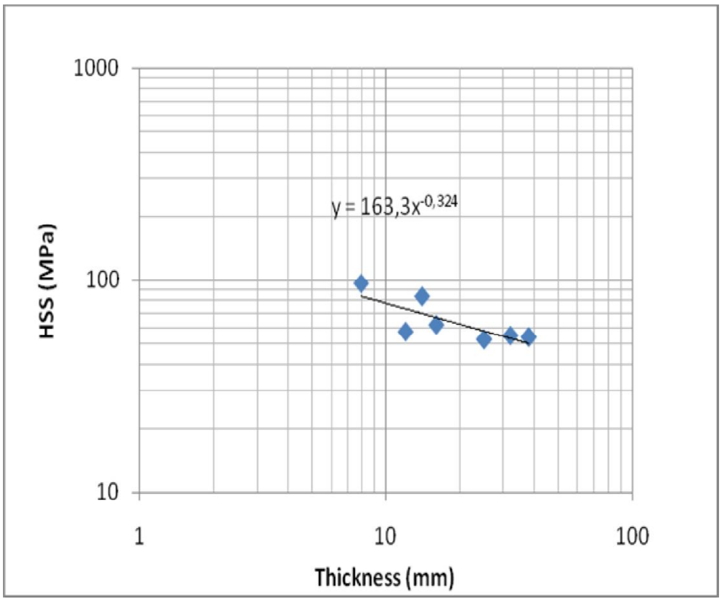
5.2.3.2.3 Detail No.3

Figure 5.21 to Figure 5.23 show the relation between the hot spot stress and the thickness. And the correction factors are 0.343, 0.323 and 0.325, almost equal to 0.31, which is the value from the nominal stress analysis approach.



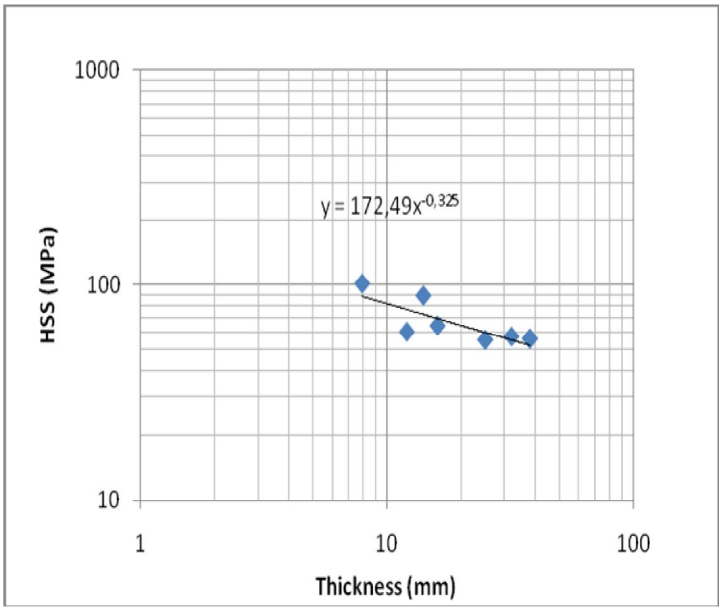
DETAIL No.3 (coarse mesh)	
N=1000000	
Thickness (mm)	Stress (MPa)
7,9	104
12	61
14	90
16	66
25	56
25	56
32	58
38	55
38	57

Figure 5.21 The thickness and stress relationship at 1 million cycles of detail No.3 according to coarse mesh hot spot stress



DETAIL No.3 (fine mesh 2 point)	
N=1000000	
Thickness (mm)	Stress (MPa)
7,9	97
12	57
14	84
16	62
25	53
25	52
32	55
38	54
38	54

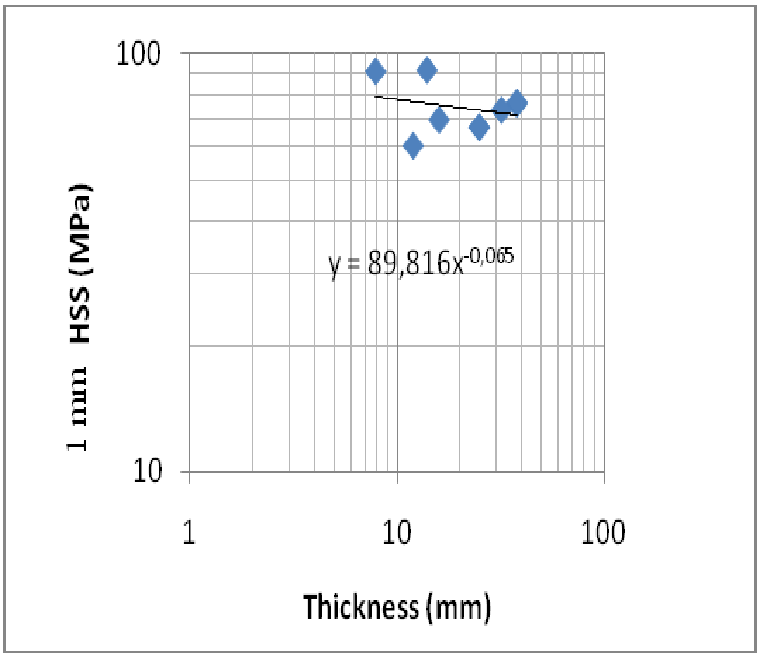
Figure 5.22 The thickness and stress relationship at 1 million cycles of detail No.3 according to fine mesh 2 point hot spot stress



DETAIL No.3 (fine mesh 3 point)	
N=1000000	
Thickness (mm)	Stress (MPa)
7,9	101
12	61
14	89
16	65
25	56
25	56
32	58
38	56
38	56

Figure 5.23 The thickness and stress relationship at 1 million cycles of detail No.3 according to fine mesh 3 point hot spot stress

The next figure shows the relationship between “1 mm stress” and thickness.



DETAIL No.3 (“1 mm stress”)	
N=1000000	
Thickness (mm)	Stress (MPa)
7,9	91
12	60
14	91
16	69
25	67
25	66
32	73
38	76
38	76

Figure 5.24 The thickness and stress relationship at 1 million cycles of detail No.3 according to “1 mm hot spot stress”

Figure 5.24 shows the thickness correction factor is 0,065 almost 0, which means there is no thickness effect when using the hot spot stress approach. It means the “1 mm method” has already considered the thickness effect.

5.3 Conclusion of thickness correction factor

The thickness correction factor by using hot spot method is very dependent on details and stress calculation method.

Table 5.31 The final results of thickness correction factor

Detail Number	Detail No.1	Detail No.2	Detail No.3
Thickness correction factor by N.S. method	0,166	0,2	0,31
Thickness correction factor by H.S.S method (coarse mesh)	0,244	0,203	0,343
Thickness correction factor by H.S.S method (fine mesh 2 point)	0,244	0,201	0,323
Thickness correction factor by H.S.S method (fine mesh 3 point)	0,283	0,197	0,325
Thickness correction factor by "1 mm stress method"	-	-	0,0065

Table 5.31 shows the final results of thickness correction factor. For detail No.1, the hot spot stress method correction factor n is larger than the nominal stress method correction factor. So if still following 0.25 it will be on the unsafe side. But for detail No.2 and detail No.3 the n factors obtained by the hot spot method are quite similar to n factors by the nominal stress method, and the thickness correction factor can follow the nominal stress method recommendation. Using the "1 mm method" for detail No.3 seems already including the thickness effect so that the n factor is almost equal to 0.

According to the research, different methods or even different mesh size will give dramatically different results. Some of them are even smaller than nominal stresses, which are of course incorrect. Therefore, how to decide which method gives a reasonable hot spot stress becomes a new problem, see Section 6.1.

The comparison of thickness correction exponents between achieved results and IIW recommendations [3] is shown in Table 5.32.

Table 5.32 The FEM results of thickness correction factor compared with the IIW recommendation (2008) [3]

Joint category	Condition	Loading condition	n (IIW[3])	n (FEA)
Transverse T-joints (detail 1)	As-welled	4-Point bending	0.3	0.24/0.24/0.28
Transverse T-joints (detail 2)	As-welled	Cantilever bending	0.3	0.2/0.2/0.2
Cruciform joints (detail 3)	As-welled	Axial force	0.3	0.34/0.32/0.33

Table 5.32 shows the thickness correction exponents from FEA in detail 1 and 2 are smaller than that in the IIW recommendations [3]. The reason might be that the component in bending condition is favourable to the fatigue compared with that in axial force when the nominal stress is the same. Nevertheless, the thickness correction factor from FEA of detail 3 is larger than that in IIW [3], which means it should be carefully considered in fatigue design. Therefore, further discussion need to be made for detail 3. The models with a small distance between main plate and attachment plate for detail No.3 will be shown in Chapter 6.

6 Discussion and conclusion

When using the hot spot stress approach to fatigue stress calculation, there were some interesting phenomenon, which will be discussed in this chapter.

6.1 ‘True’ and ‘false’ hot spot stress

- 2-D plain strain model with coarse mesh: hot spot stress equal to nominal stress.

  From the results of the hot spot stress analysis of the 2-D plain strain models (see Section 5.2), it is obvious that the stresses at the extrapolation points are equal to the nominal stress. This generates a hot spot stress equal to the nominal stress.

- 2-D plain strain model with fine mesh: hot spot stress smaller than the nominal stress.

  This appears in all three details. The reason is that the stress at 0.4*t extrapolation point is smaller than that at the extrapolation point at 1*t. These stresses are “false” hot spot stress, since they are smaller than nominal stress, and they cannot be used for fatigue assessment. Furthermore, it is because the 2-D plain strain models just represent the mean stresses along the width, and in 3-D models the hot spot stress extrapolation point should be at the maximum stress spot which are not in the middle of detail No.3. Figure 6.1 shows the location where the extrapolation point is taken in the 3D model.

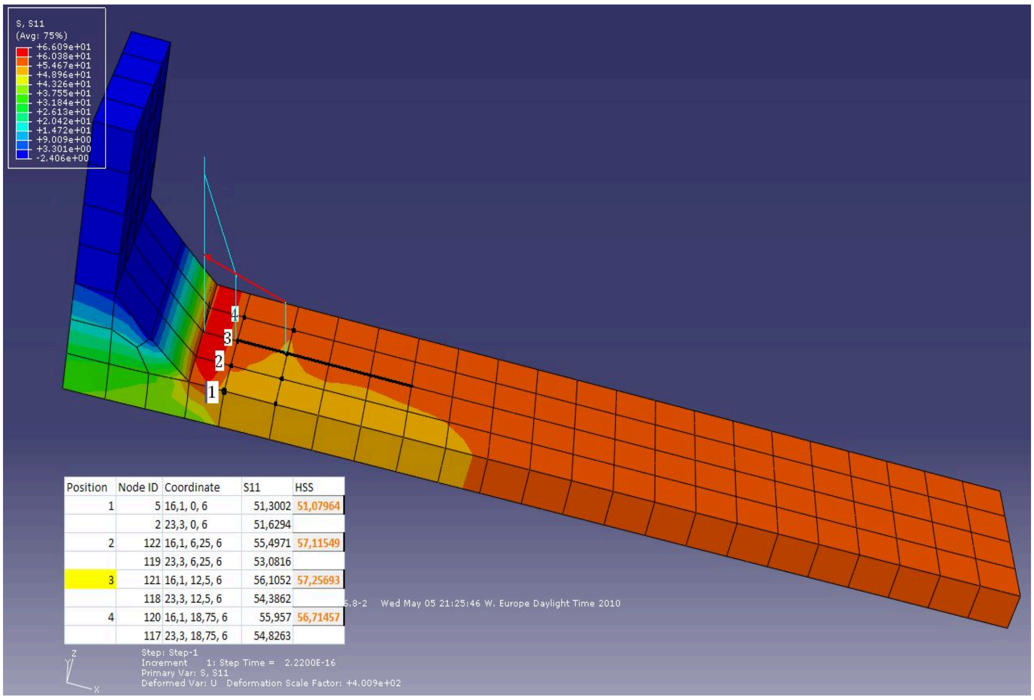


Figure 6.1 The hot spot location of detail No.3 with12mm thick plate

6.2 Stress along the surface

Taking detail No.1 with 25mm thick joint as an example, Figure 6.2 shows the stress of 3-D solid element model (0.2t*0.2t mesh and fine mesh 2 point method, 0.4t, 1.0t mesh) the stress along the surface away from the weld toe does decrease constantly, but not for 2-D model. Actually, for 2-D plain strain model (0.25mm*0.25mm mesh and fine mesh 2 point method, 0.1t*0.1t mesh), it can be seen that from the weld toe outwards, the stress decrease dramatically, and it reaches the minimum value at about 0.55*t away from the weld toe. After that the stress returns back to the nominal stress slowly, at about 2*t, the stress will equal to the nominal stress. This phenomenon was observed in other details of different thicknesses as well.

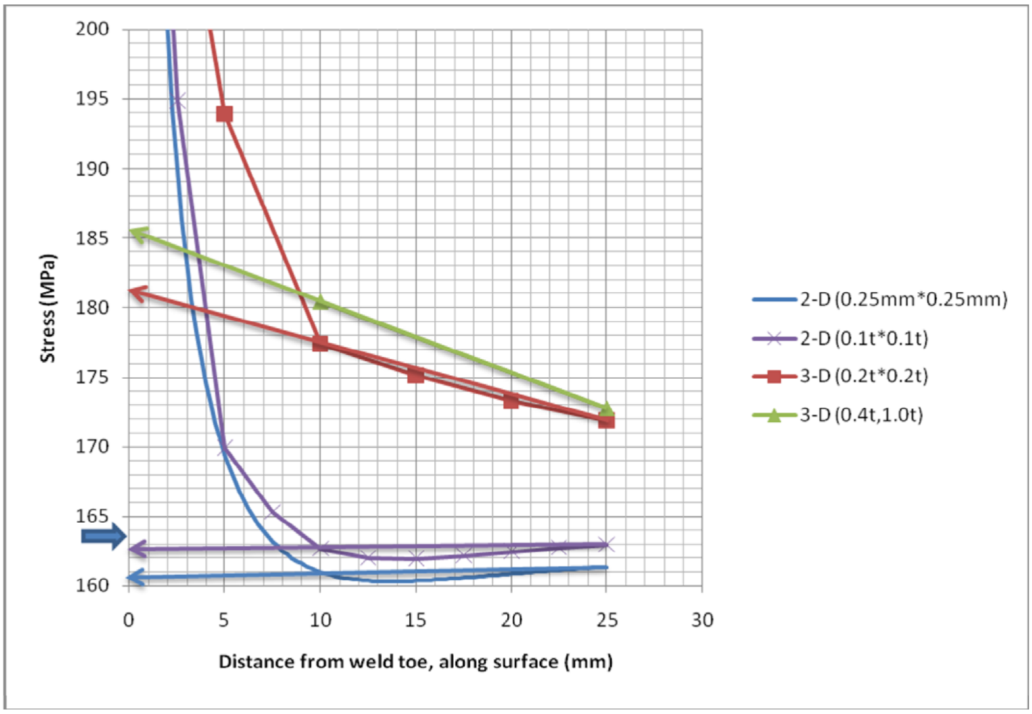


Figure 6.2 Surface stresses perpendicular to the weld along a path from the weld toe outwards of detail No.1 with 25mm thick joint.

Table 6.1 shows the location of minimum stress for detail No.1. The hot spot stresses are very mesh dependent, even for the same method, fine mesh 2 point extrapolation method for 3-D model, the result of 0.2t*0.2t mesh and the result of 0.4t*1.0t differ substantially.

*Table 6.1 Minimum stress locations for detail No.1 for 2-D plain strain (0.25mm*0.25mm mesh)*

Thickness (mm)	Distance from weld toe (mm)	Location (distance/thickness)
16	8,75	0,546875
25	13,75	0,55
40	22	0,55
70	38	0,542857

For detail No.3 12mm thick plate, the trend of stress for both 3-D and 2-D model are the same as trend of detail No.1 2-D model, which has a minimum stress near 0.6*t away from weld toe. Surface stresses of detail No.3 12mm thick plate are shown on Figure 6.3. From the Figure, all the 3-D model hot spot stresses are larger than the nominal stress, and all the 2-D model hot spot stresses are smaller than the nominal stress.

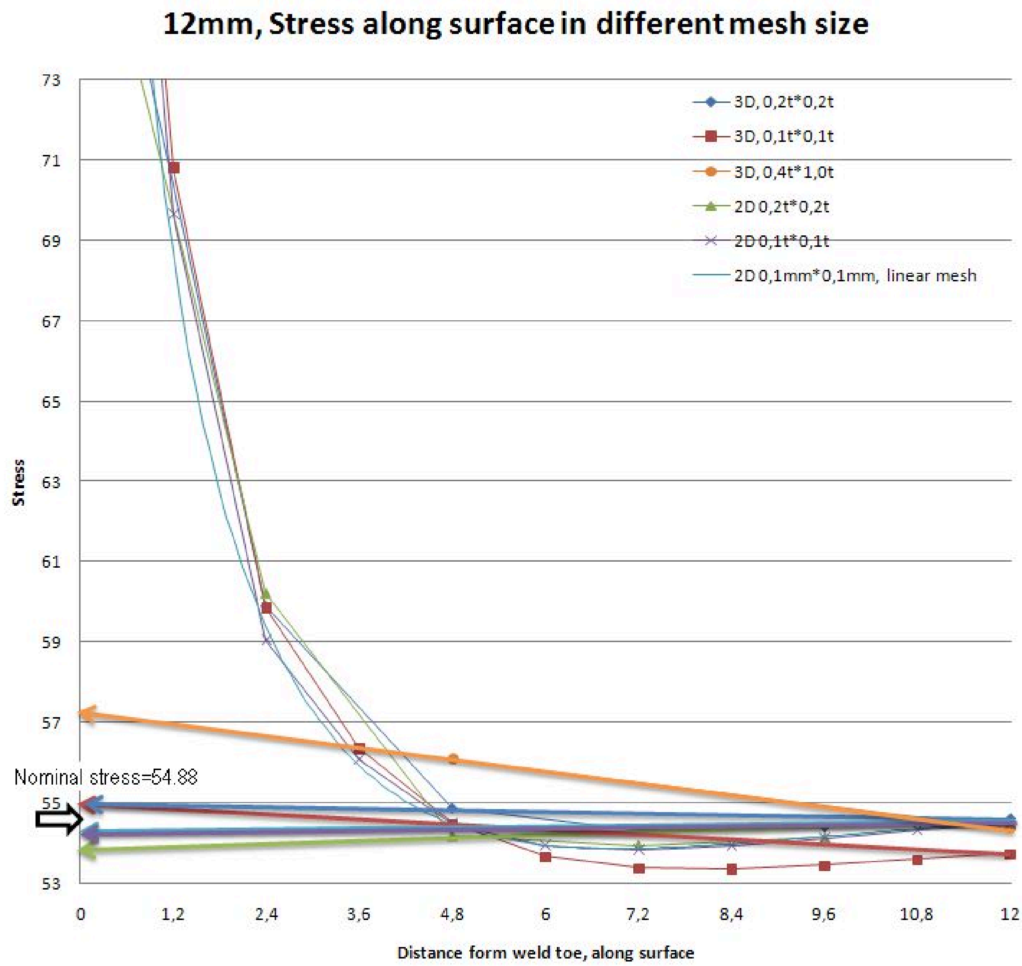


Figure 6.3 Surface stresses perpendicular to the weld along a path from the weld toe outwards of detail No.3 12mm thick joint.

Table 6.2 shows the minimum stress locations for detail No.3 for the plates with different thicknesses. Even if these specimens are not so proportional, the location points are almost the same where are 0.5*t to 0.6*t away from the weld toe. Whatever 2-D and 3-D model, the locations are the same.

*Table 6.2 minimum stress location for detail No.3 for 3-D model (0.1t*0.1t mesh)*

Thickness (mm)	Distance from weld toe (mm)	Location (distance/thickness)
7,9	3,95	0,5
12	7,2	0,6
14	8,4	0,6
16	9,6	0,6
25	15	0,6
25	15	0,6
32	19,2	0,6
38	22,8	0,6
38	22,8	0,6

6.3 Stress through the thickness

Here detail No.3 12mm thick plate is studied. Figure 6.4 shows the stresses through half of the thicknesses at different sections using different mesh models. Generally speaking, at 0.4t section the stress through thickness is almost linear, and at t section the stress is definitely linear.

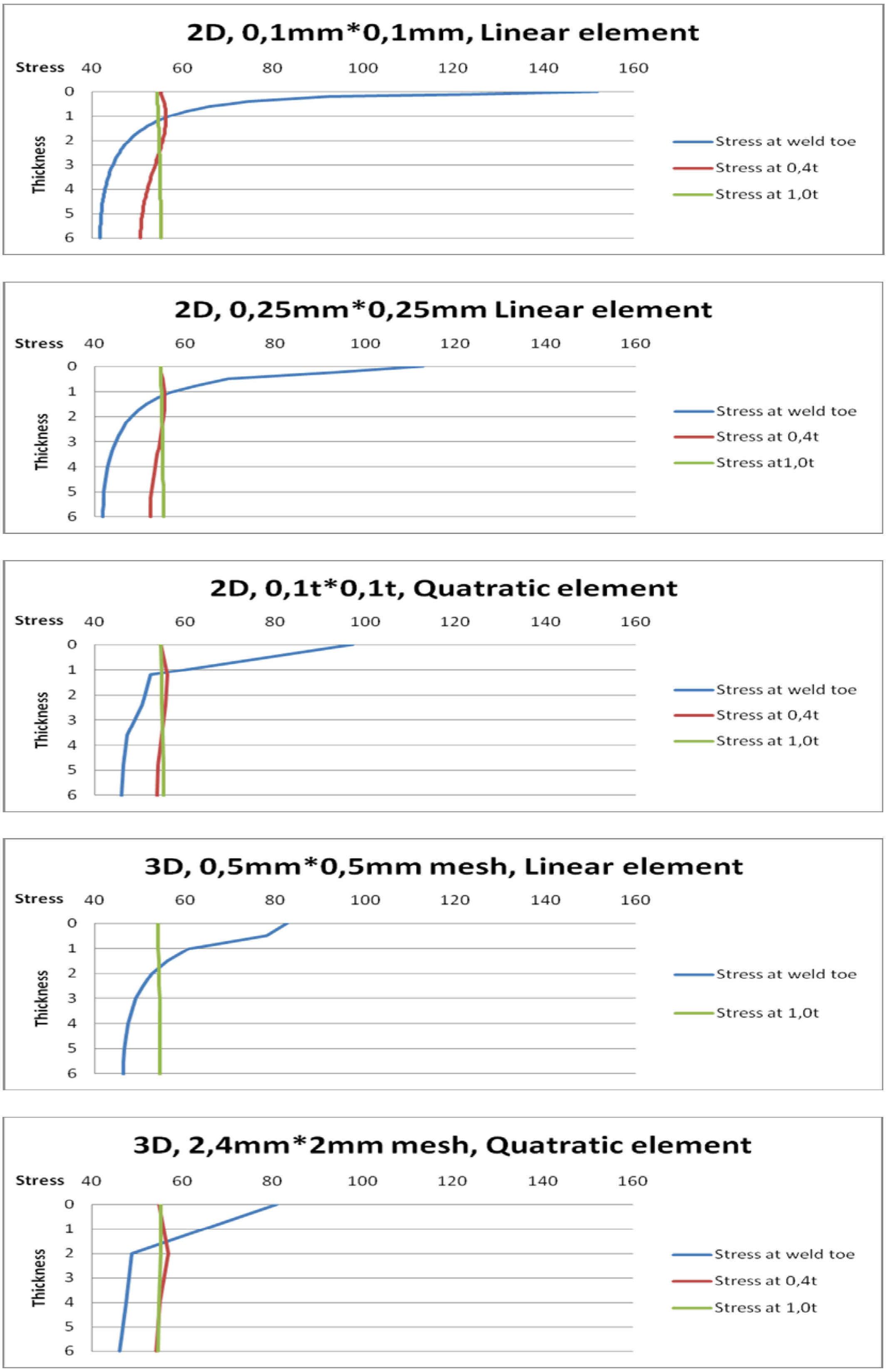


Figure 6.4 The stress through the thickness at different sections for detail No.3 with 12 mm thick plate

6.4 Convergence study of “1 mm stress method”

The convergence characteristics of the “1 mm stress method” have been studied to check how fine the mesh should be in order to obtain a relatively accurate value. Table 6.3 and Table 6.4 are convergence studies about detail No.1 25 mm thick plate and detail No.2 12 mm thick plate, respectively.

Table 6.3 Convergence study of “1 mm stress” for detail No.1 25mm thick plate

Detail No.1, Thickness=25 mm, Nominal stress=163MPa			
Mesh size	Element type	“1 mm stress”	Difference from 0.1 mm*0.1 mm mesh result
0,1 mm*0.1 mm	2-D plain strain shell, linear	179	0
0,25mm *0,25 mm	2-D plain strain shell, linear	179	0,13%
0,5 mm*0,5mm	2-D plain strain shell, linear	178	-0,28%
1 mm*1 mm	2-D plain strain shell, linear	196	9,8%
0,5 mm*0,5mm	2-D plain strain shell, quadratic	180	0,73%
1 mm*1 mm	plain strain shell, quadratic	155	-13,5%
1 mm*1 mm	3-D solid, linear	228	27,4%

For the 2D plain strain model, 4-nodes elements with 0.5 mm*0.5 mm were enough to give a ‘right’ value. It has only 0.73% error compared with very fine mesh result.

Table 6.4 Convergence study of “1 mm” stress for detail No.3 12mm thick plate

Detail No.3, Thickness=12 mm, Nominal stress=55MPa			
Mesh size	Element type	“1 mm stress”	Difference from 0.1 mm*0.1 mm mesh result
0,1 mm*0,1 mm	2-D plain strain shell, linear	60,05	0
0,25mm *0,25 mm	2-D plain strain shell, linear	60,09	0, 067%
0,5 mm*0,5mm	2-D plain strain shell, linear	59,88	-0, 28%
1 mm*1 mm	2-D plain strain shell, linear	63,90	6,407%
0,5 mm*0,5mm	3-D solid, linear	60,75	1,167%

The convergence study of detail No.3 shows the 2-D (0.5mm*0,5mm mesh) result fit well with the 3-D (0.5mm*0,5mm mesh) result and fine enough to represent the “1 mm stress”.

Since detail No.1 3-D FE model with a 0.5 mm*0.5 mm mesh is very time consuming, 1 mm *1 mm mesh was examined. This model was also time consuming with bad accuracy. Detail No.3 also shows the 2-D (0.5mm*0,5mm mesh) result fit well with the 3-D (0.5mm*0,5mm mesh) result which were accurate enough. This model has only 0.28% error. So the 2-D 0.5mm*0,5mm mesh model is recommended for “1 mm method” FE modelling.

6.5 The models with a small gap between the main plate and the attachment plate for detail No.3

Since the thickness correction factor did not comply with the IIW recommendations [3] for detail No.3 with the previous model (Section 5.3), another modal is created. The only difference between this modal and previous modal is that there is a small gap, 0.01mm, between the main plate and the attachment plate, see figure 6.5. The result of HSS is shown in Table 6.5. Figure 6.6 illustrates the relationship of T-S exponent in different models of varies mesh sizes.

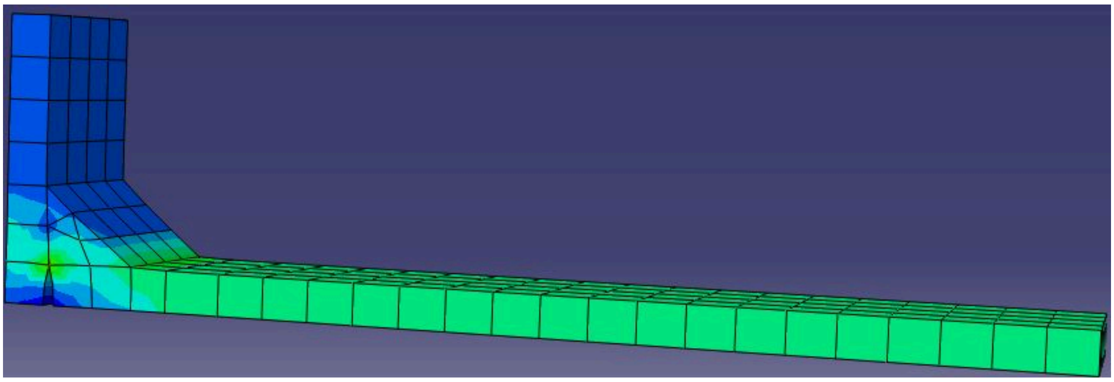


Figure 6.5 3D model for detail No.3 with a small gap between the main plate and the attachment plate

Table 6.5 Comparison of HSS in 3D modal with and without distance between main plate and attachment plate of detail No.3

	3D with distance			3D without distance			
T(mm)	HSS in Coarse mesh	HSS in Fine mesh (2 points)	HSS in Fine mesh (3 points)	HSS in Coarse mesh	HSS in Fine mesh (2 points)	HSS in Fine mesh (3 points)	NS
7.9	103	95	101	104	97	102	91
12	64	56	59	61	57	61	55
14	92	83	87	90	84	89	80
16	68	61	65	66	62	65	59
25	58	51	54	56	53	56	50
25	59	52	55	56	52	56	50
32	60	55	57	58	55	58	52
38	62	54	58	55	54	56	51
38	61	53	56	57	54	56	51
n	0.29	0.31	0.31	0.34	0.32	0.33	0.31

Table 6.5 shows that the thickness correction exponent is 0.29 and 0.31 in coarse mesh and fine mesh. The result is close to the n factor given by the IIW recommendation [3]. However, a special case t=7.9mm is going to be studied in next section.

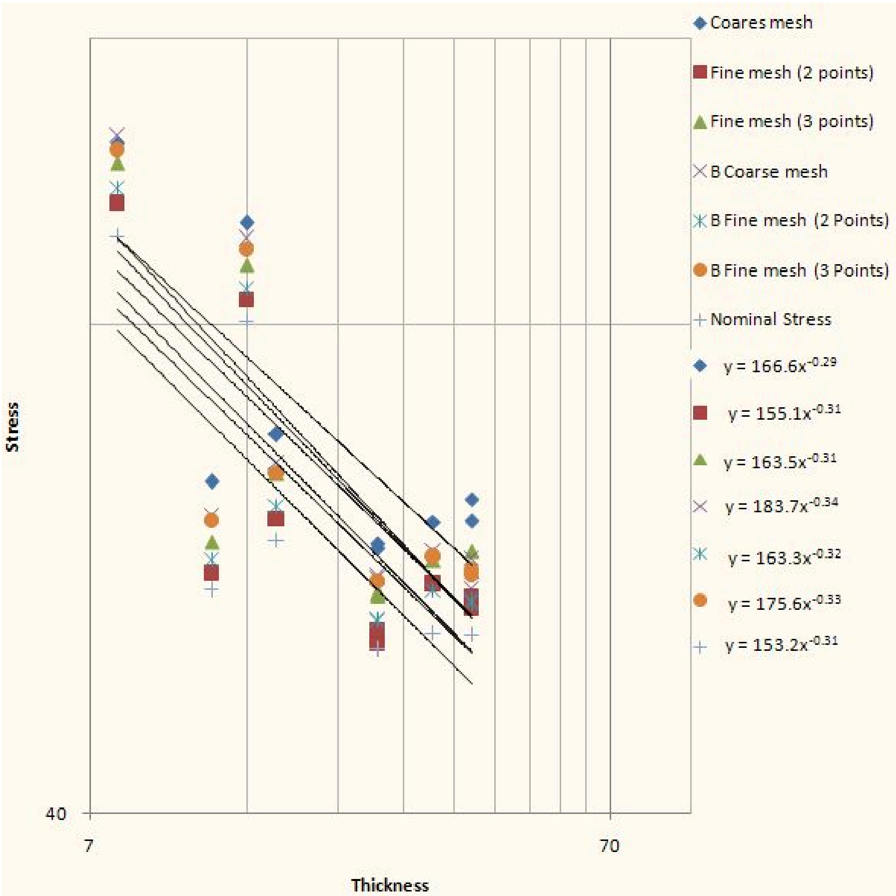


Figure 6.6 The thickness and stress relationship at 1 million cycles of detail No.3 according to 3D (B is the model without a small gap)

6.6 Thickness correction factor exclude non-proportional case t=7.9mm

The specimen, t=7.9mm, is a special case in which the attachment plate is 38mm. See Figure 6.7 and table 4.5. Therefore, the thickness correction factors excluding the data of t=7.9mm is worth to be discussed.

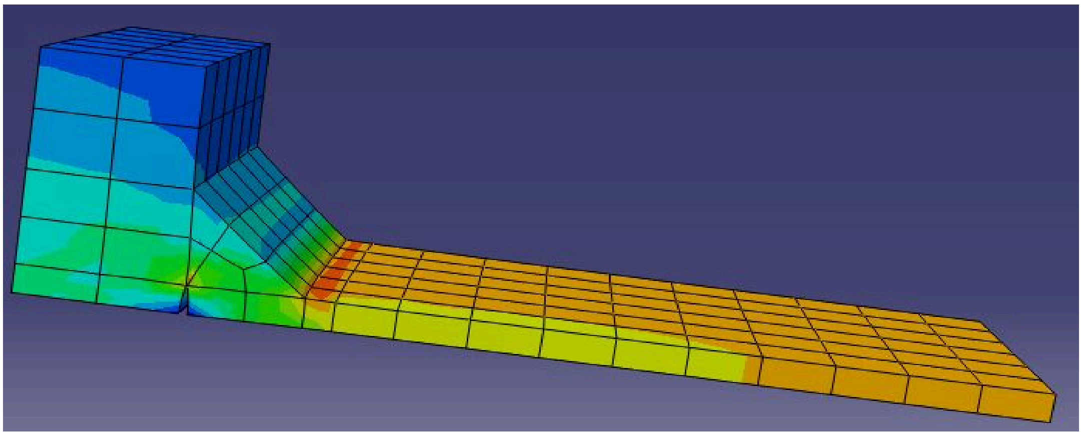


Figure 6.7 The specimen of main plate is 7.9 mm and the attachment plate is 38mm.

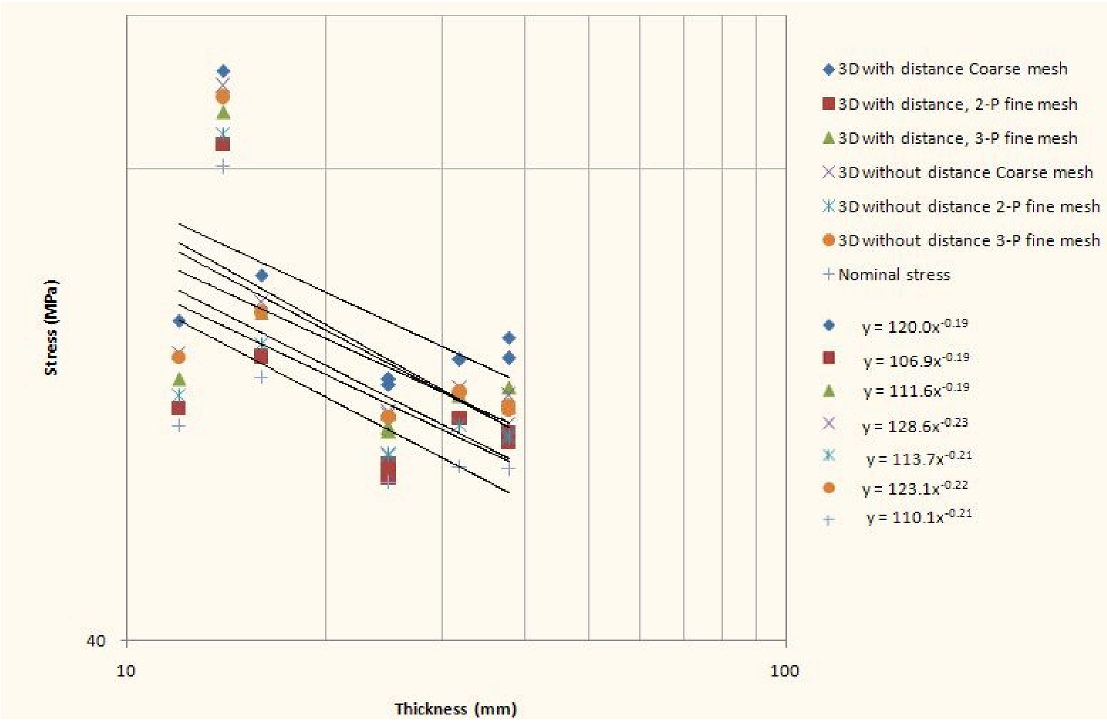


Figure 6.8 The thickness and stress relationship at 1 million cycles of detail No.3 according to 3D without specimen t=7.9mm

From Figure 6.8, it could be observed that the thickness correction factor is smaller than before. Thus, a comparison of thickness correction exponent is shown in Table 6.7.

Table 6.7 Comparison of thickness correction factor between the case with and without specimen t=7.9mm.

	3D with distance			3D without distance			NS
	HSS in Coarse mesh	HSS in Fine mesh (2 points)	HSS in Fine mesh (3 points)	HSS in Coarse mesh	HSS in Fine mesh (2 points)	HSS in Fine mesh (3 points)	
T-S exponent	0.29	0.31	0.31	0.34	0.32	0.33	0.31
T-S exponent Without t=7.9mm	0.19	0.19	0.19	0.23	0.21	0.22	0.21

In addition, for detail No.3, there is a large scatter. The result will be effected no matter which case is not taken into account, especially for specimen t=14mm.

6.7 Further study

1) Limited numbers of joint type are studied in this thesis.

F.S: More joints should be studied.

2) FE-models not so close to the reality.

F.S: The material defects, the gap between main-plate and attachment and the residual stress should be considered.

3) Only linear FE analysis is used.

F.S: Non linear FE analysis could be studied.

4) The hot spot stresses are derived only by the IIW extrapolation method and the "1 mm method".

F.S: Other H.S.S. calculation method can be studied further.

5) Only two types of model are used for FE modelling.

F.S: Other types of modelling techniques can be further studied.

Furthermore, the reference thickness is not studied in this thesis, but it should be studied as well. And will weld treatment influence the factor?

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