

ISTANBUL TECHNICAL UNIVERSITY★GRADUATE SCHOOL OF
SCIENCE ENGINEERING AND TECHNOLOGY

**NUMERICAL FATIGUE STRENGTH ASSESSMENT OF WELDED STEEL
STRUCTURES**

M.Sc. THESIS

Amr ÖZDEŞ

Department of Mechanical Engineering

Solid Mechanics Master Programme

DECEMBER 2015

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**KAYNAKLI ÇELİK YAPILARIN SAYISAL OLARAK YORULMA
DAYANIMLARININ BELİRLENMESİ**

YÜKSEK LİSANS TEZİ

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To my family,

FOREWORD

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FOREWORD

FEA	: Finite Element Analysis
EC3	: EuroCode 3
MKJ	: Moore Kommers Jasper
FEM	: Finite Element Method
HAZ	: Heat Affected Zone

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NUMERICAL FATIGUE STRENGTH ASSESSMENT OF WELDED STEEL STRUCTURES

SUMMARY

Fatigue of metals has been an area of mechanical sciences that yet has not been completely well understood. There has been much research and many studies about mechanical behavior of fatigue evaluation of metals. Besides this, fatigue of welded components is even more complex. Apart from the thermal cycles during welding, which enormously affect the base material and introduce residual stresses and distortion to the weld zone, the fusion process of filler leads to very heterogeneous micro structures, varying the materials' mechanical properties and chemical composition throughout the neighbouring area of the joint. Moreover, weld defects like porosity, slag inclusions, undercuts and overlap, among others, can significantly reduce the fatigue strength. All these reasons create a proper environment for especially weld toe cracking, which is the most critical area of welding (strongly depends on the loading conditions). Based on identified source of problems, several improvement techniques have been developed and they have proven to increase the fatigue strength of welded area by inducing compressive stresses in weld toe, modifying the mechanical design into smoother geometries, modifying the welding techniques and processes, and similar.

Fatigue mechanisms around the welded area of structures has been a major area of investigation for engineers especially for those working in the area of rail vehicle, automotive, naval and construction equipment industries. Mainly being initiated from these industries, many studies and codes have been developed for evaluating the fatigue conditions welded areas by both academic and industrial authorities.

In this study, numerical fatigue strength assessment in welded areas were considered on a preferential basis according to internationally accepted codes, including FKM (Forschungkuratorium Maschinenbau), DVS 1612, and EuroCode 3 (EC3, EN-1993-1-9). Each of these international codes have their own assessment theory and technique for the fatigue strength assessment of welded structures.

To be able to understand the fatigue behaviour in the welded areas, structural fatigue theories should have been well understood. In this study, stress-based fatigue analysis theory have been processed and implemented via numerical methods.

Fatigue strength assessment of welded structures and evaluation of stresses in notched areas can be processed numerically either with manual calculations or commercial software packages. In this study, for fatigue stress assessment in welded areas a software package *LIMIT 2015* was used in accordance with *ABAQUS/Standard*.

It has been seen that; numerical methods can be used for strength assessment by comparing theoretical solutions to numerical outputs for different codes. Based on the specifically chosen examples, it can be concluded that, analysing the fatigue behaviour in welded areas has a strong relationship with the structural design process. To be able

to increase the fatigue life of welded areas of the structure, it may be needed to alter the structural design in accordance with the fatigue behaviour in welded areas. Even statically safe and approved designs have to be fully analysed according to fatigue behaviour in their welded areas or joints.

KAYNAKLI ÇELİK YAPILARIN YORULMA DAYANIMLARININ SAYISAL OLARAK BELİRLENMESİ

ÖZET

Tekrarlı yüklemelere maruz bırakılan makina parçalarında ve/veya mekanik yapılarda oluşan hasarların başlıca nedeni yorulmadır. Hasar analizi sonuçlarına dayanılarak, imal edilen çelik yapılarda yorulma hasarının çoğu kez gerilme yığılmasının çok yüksek olduğu, çentik etkisine sahip kaynak geometrilerinden başladığı görülmüştür.

Gerilme yığılmasının olduğu bölgelerin haricinde, bağlantının yorulma dayanımına etki eden çok sayıda faktör bulunmasından dolayı, kaynaklı bağlantı bölgeleri, ana taşıyıcı elemanlara göre çok daha düşük yorulma dayanımına sahiptir. Bahsi geçen faktörler arasında en önemli etkiye sahip olanlar, ısı tesiri altındaki bölgenin mikro yapısal etkileri ve kaynak işleminden dolayı meydana gelen artık gerilmelerdir.

Artık gerilmeler ve ısı tesiri altındaki bölgenin mikro yapısal etkilerinin haricinde, kaynak işlemi sırasında kaynak dolgu malzemesinin bölgede meydana getireceği homojen olmayan yapı, kaynak işlemi sırasında genellikle işçilik kusurları sebebiyle meydana gelen kaynak bölgesindeki gözenekli yapı, işlem sırasında oluşan dikiş bindirmeleri gibi problemler bölgenin mekanik ve kimyasal kompozisyonunu değiştirmekte ve yapısal olarak yorulma dayanımının önemli ölçüde azalmasına neden olmaktadır.

Bahsi geçen bu etmenler sebebiyle kaynaklı bölgenin özellikle kaynak dibi veya kaynak kökünden (yükleme şartına ve kaynak konfigürasyonuna bağlıdır) hasara uğraması kaçınılmazdır. Hasarı engellemek adına, akademisyenler ve mühendisler, ilgili bölgelerin yorulma mukavemetini arttırabilmek adına çeşitli yöntemler geliştirmişlerdir. Kaynaklı bölgeye artık gerilmeyi giderebilmek adına ısıl işlem uygulanması, kaynak dibindeki gerilme karakteristiğinin değiştirilmesi (basma gerilmesi uygulanması) ve yapısal tasarımı, kaynaklı bölgeye gelebilecek gerilmeleri azaltmaya yönelik yeniden geliştirme bunlardan bazılarıdır.

Çoğunlukla raylı taşımacılık, otomotiv endüstrisi, denizcilik endüstrisi alanında çalışan mühendisler için, yapısal tasarımda kaynaklı bağlantı kullanılması kaçınılmaz bir imalat yöntemidir. Bu nedenle, özellikle kaynaklı bağlantıların yorulma mukavemetinin bulunması ve kaynaklı bölgenin yorulma karakteristiğinin belirlenmesi, bu alanda çalışan mühendisler için çok önem arz etmektedir. Bu bahsedilen alanlardan başlamak şartıyla numerik veya analitik yöntemlerle kaynaklı bölgelerin yorulma mukavemetinin belirlenmesi adına birden fazla uluslararası standart ve kod geliştirilmiştir, geliştirilmektedir. Bu tez kapsamında bunlardan en önemlileri olan FKM, DVS1612 ve EuroCode 3 incelenmiştir. Her üç kod da kendine ait prensiplere, kaynak kalite parametrelerine ve yorulma mukavemetini tahakkuk etme teorilerine sahiptir.

Bu kodların özet olarak nasıl işlediklerinden bahsetmeden önce bu tez kapsamında gerilme temelli yorulma mukavemeti değerlendirme metodu işlenmiştir. Dolayısıyla

her üç kod da yapıdaki gerilmelerden yola çıkarak yorulma mukavemeti tayini yapmaktadır.

İlk olarak FKM, Alman mühendis ve akademisyenler tarafından geliştirilen ve hala üzerinde çalışılan, analitik olarak yapıdaki ana taşıyıcıların ve kaynaklı bölgelerin yorulma mukavemeti tayini amacıyla, Avrupa başta olmak üzere dünyanın birçok yerinde kullanılan uluslararası bir teknik kılavuzdur. FKM kendi içinde, deneysel verilerden alınan parametreleri yine kendine ait teorileri kullanarak yapılarda yorulma mukavemeti tayini yapabilme kabiliyetine sahip prensipler içermektedir. Bu prensipleri oluşturan parametreler arasında: yorulma dayanımına en çok etki eden ortalama gerilmenin etkisi, geometrik çentik faktörleri, gerilmenin hesap edileceği bölgenin konumu gibi önemli kıstaslar yer almaktadır. FKM kılavuzu (kodu) mekanik yükler altında çalışan tüm yapısal tasarımlara uygulanabilmektedir.

DVS 1612, yine Alman mühendis ve akademisyenler tarafından temelde raylı taşımacılık endsütrisinde kullanılmak üzere, kaynaklı yapılarda yorulma mukavemeti tayin etmek amacıyla geliştirilmiş / geliştirilmekte olan bir kılavuzdur. Kılavuz FKM 'de olduğu gibi kendine ait teoriler ve kaynak kalite parametreleri içermektedir. Bu parametreler vasıtasıyla yapıdaki kaynaklı alanların, verilen yükleme şartları altında yorulma dayanımını belirlemektedir.

EN 1993-1-9 (EuroCode 3), İngiliz mühendisler ve akademisyenler tarafından geliştirilen ve kaynaklı bölgelerinin mukavemetini belirlemek amacıyla genellikle taşıyıcı yapılarda kullanılan uluslararası bir kılavuzdur. FKM ve DVS1612'de olduğu gibi, EuroCode 3 de kendine has kaynak kalite parametreleri ve yorulma dayanımını tayin edebilmek amacıyla teoriler barındırmaktadır. Diğer iki teoriden en önemli farkı, ortalama gerilme etkisinin bu kılavuz kapsamında, yapıdaki kaynaklı bölgelerinin yorulma mukavemetini tayin etmek üzere kullanılmaması / hesaba katılmamasıdır.

Bahsedilen her üç teori de dünyanın birçok yerinde mekanik yapısal tasarım aşamasında kullanılmaktadır.

Tez kapsamında, bahsi geçen üç kod, bir demiryolu aracının yorulma mukavemetini tayin etmek amacıyla kullanılmıştır. İstanbul Teknik Üniversitesi Mukavemet ve Biyomekanik Laboratuvarı akademisyenleri tarafından sayısal ve deneysel gerilme analizleri başarıyla yapılmış olan bu yük taşıma vagonunun yorulma dayanımının belirlenmesi amacıyla LIMIT yazılımı kullanılmıştır. LIMIT yazılımı, kendi içinde yukarıda bahsi geçen her üç kılavuzu da kullanabilmektedir. Tüm sayısal gerilme analizi çalışmaları İstanbul Teknik Üniversitesi Mukavemet ve Biyomekanik Laboratuvarı'nda bulunan süper bilgisayarlar aracılığıyla yapılmıştır. Sayısal vagon modelinin oluşturulması ve yükleme şartlarının vagona sonlu elemanlar ortamında uygulanması işlemi Hypermesh 13.0 yazılımı ile yapılmıştır. Oluşturulan sayısal vagon modeli, ABAQUS 6.13 programında çözdürülmüş ve sonuçlar, yorulma mukavemetinin tayin edilmesi amacıyla LIMIT yazılımına aktarılmıştır. Birbirine bağlı şekilde yapılan tüm nümerik çalışmalar neticesinde, daha önce bahsi geçen her üç kılavuz doğrultusunda demiryolu yük vagonunun yorulma mukavemeti tayini yapılmıştır.

Yapılan çalışmalar ve sonuçlar doğrultusunda, değişken mekanik yükler altında çalışan makina parçaları ve mekanik yapıların kaynaklı bölgelerinin yorulma tayini yapılması çok fazla önem arz etmektedir. Yapıların hem statik, hem de dinamik yükler altında çalışması sebebiyle, statik ve yorulma dayanımlarının, yapının hem ana taşıyıcılarında hem de birleşim noktalarında, kaynak yapılması gerekmektedir. Kaynaklı bölgelerde yapılan yorulma mukavemeti tayini, yapının hem ana tasarımını değiştirebileceği gibi,

yapının kritik bölgelerinde tahribatsız periyodik muayene yapılması gerekliliğini ve kaynak işlemi sonrası kaynaklı bölgede iyileştirme çalışması yapılması gerekliliğini de ortaya çıkarabilmektedir.

Çalışmanın sonucu olarak, kaynaklı bölgelerin yorulma karakteristiğinin incelenmesi ile yapısal tasarım arasında güçlü bir bağ bulunmaktadır. Kaynaklı bölgelerin yorulma dayanımının arttırılması, tasarımın değiştirilmesi ile mümkün olacağı gibi, kaynak işlemi sonrası kaynaklı bölgede iyileştirme çalışmalarının yapılması ve yapının ek parçalar ile güçlendirilmesi ile de mümkündür. Yapının ek parçalar ile güçlendirilmesi yapıya ek bir kaynak getirmesi sebebiyle, temel olarak yapının yorulma mukavemetinin belirlenmesi ve güvenli tasarımın ortaya çıkarılabilmesi, yapının gerilme durumunun çok iyi okunabilmesi ile mümkündür.

1. INTRODUCTION

Fatigue of metals has been an area of mechanical sciences that yet has not been completely well understood. There has been much research and many studies about mechanical behavior of fatigue evaluation of metals. Static or quasi static loading is rarely observed in modern engineering components or structures. For this reason, engineers must address themselves to the implications of repeated load, fluctuating loads, and rapidly applied loads. Such loading induces fluctuating or cyclic stresses that often result in failure by fatigue.

Besides this, fatigue of welded components is an even more complex area. Either welding together a few relatively simple parts or fabricating large, complex structures, weld fatigue is likely to be the most-common failure mode if the part or structure is subjected to alternating stresses.

Fatigue failures in steel structures often occur at welded joints where stress concentrations due to joint geometry are relatively high and fatigue strength of the weld is reduced in comparison to that of the base material. Fatigue of welded details can be a limiting factor for the design of more efficient steel structures.

In many cases, the fatigue performance of severely loaded details can be improved by employing good detail design practices, for example by upgrading the welded detail class to one having a higher fatigue strength. As an alternative to strengthening the structure at a considerable increase in costs, procedures which reduce the severity of the stress concentration at the weld, remove imperfections, and introduce local compressive stresses at the weld can be used for improvement of the fatigue strength. Similarly, these fatigue improvement techniques can be applied as remedial measures to extend the fatigue strength of critical welds that have failed prematurely.

Within the scope of this thesis, detailed fatigue strength assessment procedures of welded components have been introduced based on international fatigue codes like FKM, EC3 and DVS1612. A comprehensive study of weld fatigue assessment has been numerically applied to a sample freight wagon by using different codes, and the results are compared to each other.

2. HISTORY

The fatigue failure of metals is the well-known type of failure that occurs after the cyclic stresses are applied to the component or to the specimen. The phenomenon that causes such a failure is quite complicated and not yet completely understood in completely a physical basis, despite one and half century long of studies and a large investment in research.

Fatigue failure develops in two steps: crack initiation and crack propagation, which may not be noticed macroscopically and the final failure is caused then by sudden catastrophic failure of the remaining section in the component or part.

The fatigue cracks are caused by repeated application of loads which individually are too small to cause failure and usually initiate from the surface of a component. This phenomenon is called crack initiation. The crack may then propagate and this phenomenon is called crack propagation. Finally the component may fracture.

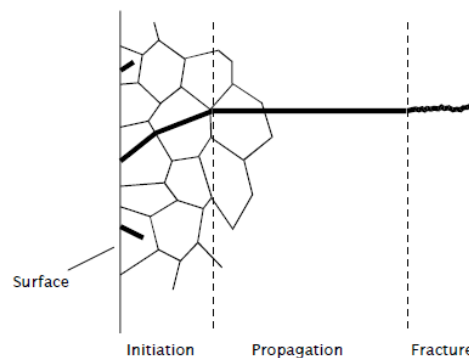


Figure 2.1: Three stages of fatigue failure

Modern fatigue theories provide separate analyses for each phase of fatigue phenomenon. Theories that try to estimate crack initiations are based on the assumption that fatigue cracks are initiated by the local stresses and/or strains on the surface of a component. Crack propagation theories relate the crack growth to the stress in the component. Earlier theories treated the whole of the fatigue life as a single entity and related fatigue life to the calculated engineering stresses in the structure.

Figure 2.2 shows the failure surface of a shaft due to fatigue which was tested in bending. Crack initiation has occurred at the surface. The progression of the crack is shown by the bands, called striations, or bench marks.

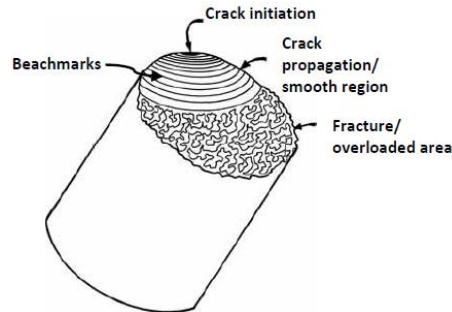


Figure 2.2: Schematic fatigue fracture surface in bending

An early example of testing to determine the fatigue strength of a component was reported by W.A.J. Albert in 1813. He investigated the failure of chains used to haul trucks in mine shafts due to repeated loads. [1] Examples of the chains were tested using a test rig built as to purpose shown in Figure 2.3

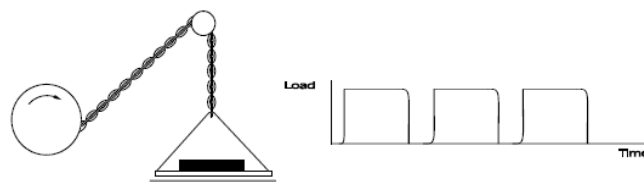


Figure 2.3: Time history of applied loading on the chain

The weight was raised and lowered, applying repeated loads to the links of the chain. If a new chain is used for each test, and each chain is tested using a different load, the relationship between the magnitude of the applied load and the number of applications to cause fracture of the chain, had been plotted in Figure 2.4.

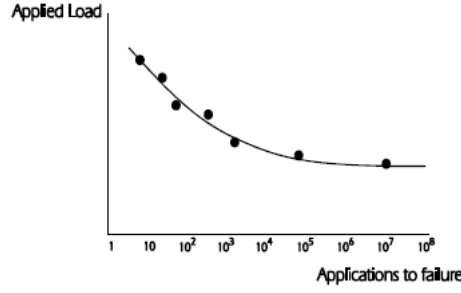


Figure 2.4: Fatigue test results

The horizontal axis shows the number of load applications required to cause failure. This is the life curve of the chain, and is referred to as *endurance curve*. The vertical axis shows the magnitude of the applied load.

With the introduction of railways of train industry, a number of failures started to be observed in components subjected to cyclic loads. These failures occurred under loads with values fairly lower than those would produce rupture, it was accepted the cyclic nature of the problem [2]. Wöhler proposed in 1860 a method to avoid these collapses, and created the S-N curve concept. By that time, designers applied high safety factor to avoid this kind of failures, but they did not go deeper into the reasons of the problem. The existence of cracks in the components was not also considered until then.

2.1 S-N Curves

The basic method of compiling engineering data is by means of Wöhler Curve, where the stress σ is plotted against the number of cycles to failure, N. Data for the latter are usually demonstrated in a logarithmic scale. The value of the nominal stress employed can be:

$$\sigma_a = \frac{\sigma_{\max} - \sigma_{\min}}{2} \quad (2.1)$$

where $\sigma_{\max}$ the maximum stress that occurs during cyclic loading and $\sigma_{\min}$ is the minimum stress that occurs during cyclic loading

Examples of Wöhler curves are shown schematically in Figure 2.5. It is important to note that each curve is determined for a specified value of:

$$\sigma_m = \frac{\sigma_{\max} + \sigma_{\min}}{2} \quad (2.2)$$

$$R = \frac{\sigma_{\min}}{\sigma_{\max}} \quad (2.3)$$

where σ_m stands for mean stress value and quantity R; “stress ratio” is introduced as a useful parameter.

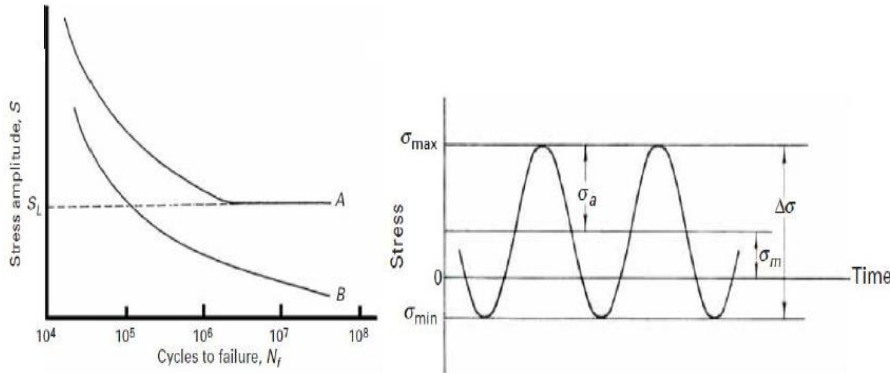


Figure 2.5: S-N curve for ferrous (A) and non-ferrous (B) materials, and related characteristic parameters.

Investigating the S-N curves in Figure 2.5 ferrous materials have been to have a clear endurance limit whereas another (B) exhibits failure even at very low values of σ . Nevertheless, the behavior is similar in both cases. With increasing stress amplitudes the fatigue life is decreased. Thus special care has to be taken when dealing with high amplitude of loads.

Most non-ferrous materials, like aluminium, magnesium and copper alloys have an S-N curve similar to B in Figure 2.5, with no evident limit. For these cases, it is usual to set the fatigue strength at an arbitrary high number of cycles, such as 10^8 . Ferrous materials and titanium exhibits a fatigue limit, below which the material is seen to withstand infinite number of cycles without failure theoretically. [3]

2.2 High Cycle Fatigue (HCF) and Low Cycle Fatigue (LCF)

The Wöhler diagram is shown in Figure 2.6 in an arrangement that shows its typical fatigue characteristics, for regions defined as:

- Zone K where failures occur at very low number of cycles and very high stress,
- Zone Z where stress is lower, number of cycles is higher, but final failure still occur,

- Zone D of stress equal or lower than the fatigue limit σ_D , if exists, or where the S-N line has a very small slope.

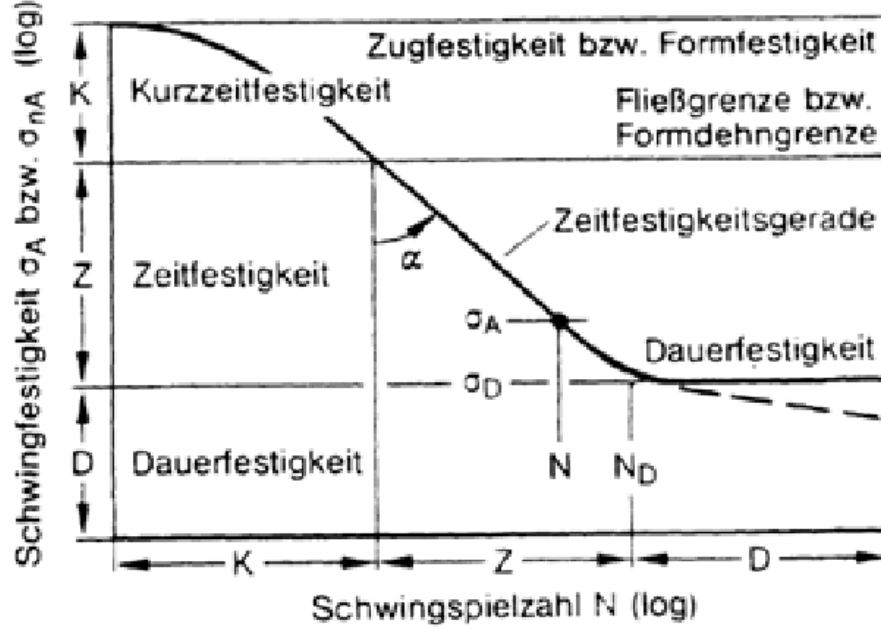


Figure 2.6: Schematic of the Wöhler Diagram in the log-log representation with the fields K, Z and D

The so-called *knee* of the Wöhler Curve is observed at the number of cycles named N_D that depends on the type of the material. Using the log-log representation, for zone Z the Wöhler curve can be easily linearized, obtaining the well-known equation:

$$\sigma_A = \left(\frac{N_D}{N_A} \right)^k \sigma_D \quad (2.4)$$

That links to the value of σ_A , the stress on the specimen or component, with the number of cycles N_A of final breakage. The exponent k is defined as the slope of the line. The location of the elastic limit σ_E on the Wöhler diagram depends on the materials characteristics, but as a rule of thumb, for a large selection of hard and also of mild steels, we can assume that $\sigma_E > \sigma_D$ and consequently that no plasticity occurs at stresses not far different from the fatigue limit.

There is no fixed rule to separate the concept of LCF from the HCF approach but, as a rule of thumb, the LCF refers to life from 10 to 10^6 cycles to failure, for higher number of cycles or lower stresses, the phenomenon is called High Cycle Fatigue (HCF). In HCF region stresses are in pure elastic region. Life is usually infinite or finite with very high number of cycles.

2.3 Mean Stress Effect

Much of the fatigue data in the literature has been determined for fully reversed stress cycles with zero mean stress ($\sigma_m = 0$) that is $R = -1$. However, the effects of mean stress is important, and an increase in mean stress will always cause a reduction in the fatigue life [4]

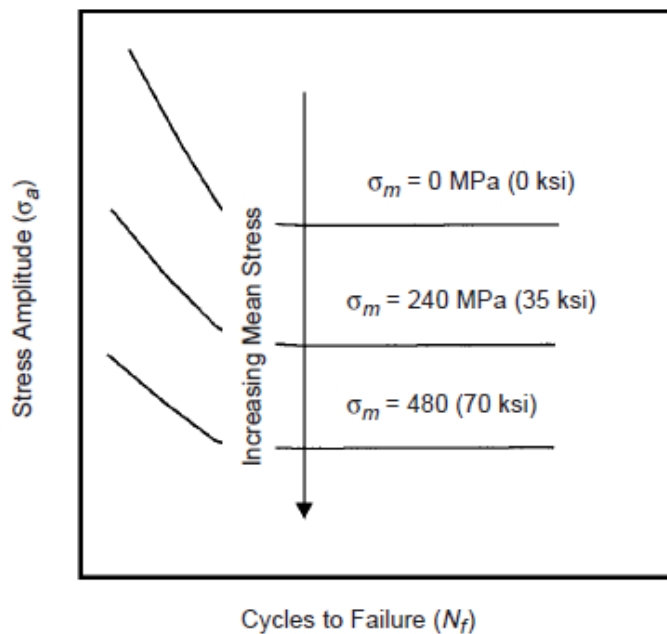


Figure 2.7: Effect of mean stress on fatigue life

2.4 Factors Which Influence Fatigue Strength

Fatigue cracks usually initiate from the surface of a component, where fatigue damage initiates as shear cracks on crystallographic slip planes.

The surface shows the slip planes as intrusions and extrusions. This is Stage I crack growth. Stage II crack growth takes place in a direction normal to the applied stress. Finally the crack becomes unstable and fracture occurs.

For many components Stage I growth occupies most of the useful life before fatigue failure is observed. The life to fatigue crack initiation has been found to be a function of surface stress or strain combined with a critical feature such a weak grain boundary.

Because crack initiation is a mostly surface phenomenon assuming that the material is perfectly homogeneous, any process which effects the surface properties may have a significant influence on fatigue strength. Important factors that affect the surface properties related with fatigue phenomenon are:

- The quality of the surface finish, for example machined, rolled, as-cast, as-forged.
- Surface treatments such as plating or cladding.
- Residual stresses introduced during manufacture either by the fabrication process or by special treatments such as shot peeling or rolling.
- The operative environment, for example existence of corrosive gases, salt sprays and operating temperature.

The effect of environmental factors on fatigue strength heavily depends on the material and on the precise nature of the surface treatment or the operating environment. For practical fatigue calculations where environmental effects are to be included, reference is generally redirected to specialized research papers. To illustrate their importance, examples of the effect of the environment and surface treatment on fatigue strength levels are shown in Figure 2.8 and Figure 2.9.

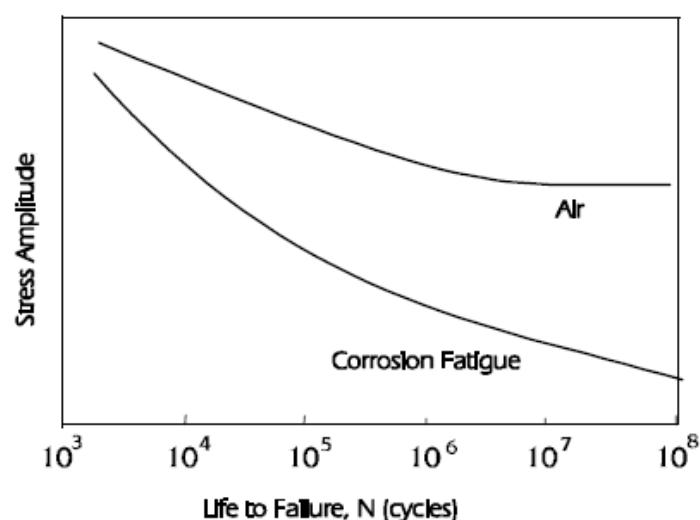


Figure 2.8: Example of the effect of the environment on fatigue strength

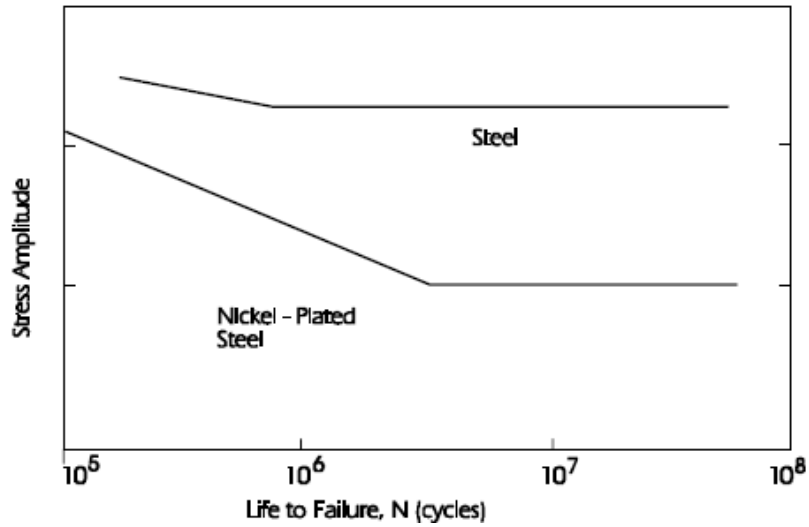


Figure 2.9: Effect of a nickel plating on the fatigue strength of a steel

The effects of residual stresses are usually limited to low endurance because the local yielding associated with low cycle fatigue eliminate the residual stresses introduced by shot peeling or rolling.

The cycle frequency of the applied loading, and the wave form, can be ignored in many fatigue calculations, and tests conducted at frequency range of 20Hz-20 kHz have shown little difference in the total cycles to failure. Exceptions are corrosive environments, high temperature fatigue, and situations where the initiating crack is subjected to internal pressure.

3. STRESS BASED UNIAXIAL FATIGUE ANALYSIS

The uniaxial fatigue analysis is used to estimate the fatigue life of a component under cyclic loading when damage is initiated under uniaxial state of stress. The fatigue life of a component refers to the fatigue initiation life defined as the number of stress cycles (N) or reversals (2N). By definition one cycle consists of two reversals.

The stress in a cycle can be described either by stress amplitude σ_a , and mean stress σ_m or by maximum stress σ_{max} and minimum stress σ_{min} as shown in Figure 3.1.

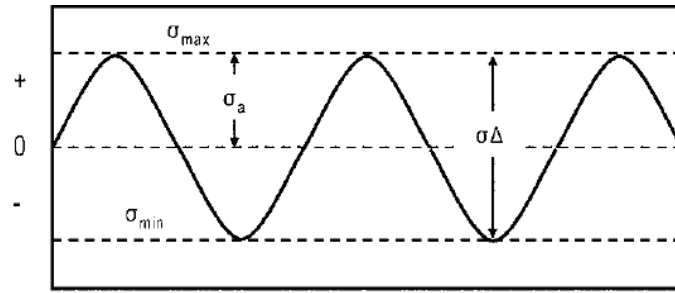


Figure 3.1: Symbols used with cyclic stresses and cycles

Since stress amplitude σ_a is a primary factor affecting cycles to failure N, it is often chosen as the controlled or independent parameter in fatigue testing, and consequently, N is the dependent variable on σ_a .

The choice of the dependent and independent variables places an important role in performing a linear regression analysis to define the S-N relations. As the stress amplitude becomes larger, fatigue life is expected to be shorter. The stress and life estimation (namely, the constant amplitude S-N curve) can be generated by fatigue testing material specimens or real components at various stress levels. For this type of fatigue testing the mean stress is generally held constant, commonly is equal to zero.

The constant amplitude S-N curve is often plotted by a straight line on log-log coordinates, representing fatigue data in the high cycle fatigue regime. In German,

constant amplitude S-N curve is named the Wöhler curve, to the honour Mr. Wöhler for his contribution to the first fatigue study in modern world.

Generally, an S-N curve can be constructed as a piecewise-continuous curve consisting of two distinct linear regimes when plotted on log-log coordinates. For the typical S-N curve of a component made of steels is schematically illustrated in Figure 3.2.

The parameters used to define the inclined linear segment of an S-N curve are the fatigue properties. The slope of an S-N curve in the HCF regime can be denoted as b (the height-to-base ratio) or as k (negative base-to-height ratio).

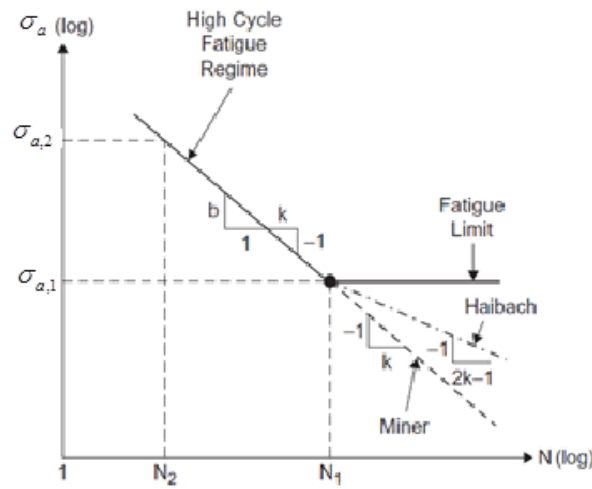


Figure 3.2: Schematic constant amplitude S-N curve of a components made of steels

The parameter k is the slope factor. The two slopes are related in the following expression:

$$k = -\frac{1}{b} \quad (3.1)$$

Any two S-N data points in the HCF regime can be related by the slope b or the slope factor k in the following equation:

$$\frac{N_2}{N_1} = \left(\frac{\sigma_{a,1}}{\sigma_{a,2}} \right)^k = \left(\frac{\sigma_{a,1}}{\sigma_{a,2}} \right)^{-1/b} \quad (3.2)$$

The S-N curve is commonly expressed as ;

$$N\sigma_a^k = A \quad (3.3)$$

or

$$\sigma_a = \sigma_f' (2N)^b \quad (3.4)$$

where A stands for as the fatigue parameter and σ_f' stands for the fatigue strength coefficient defined as the fatigue strength at one reversal.

The fatigue limit of a component made of steels can be defined as the fully reversed stress amplitude at which the fatigue initiation life becomes infinite or when fatigue initiation failure does not occur. The fatigue limit can be interpreted from the physical perspective of the fatigue damage phenomenon under constant amplitude loading.

The fatigue limit might be negatively influenced by other factors such as periodic overloads, elevated temperatures or corrosion. When Miner's rule is applied in variable amplitude loading, the stress cycles with amplitudes below the fatigue limit could become damaging if some of the subsequent stress amplitudes exceed the original fatigue limit.

Two methods such as the Miner rule and the Miner-Haibach model as shown Figure 3.2 were proposed to include the effect of periodic overloads on the stress cycle behavior below the original fatigue limit. The Miner rule extends the S-N curve with the same slope factor k to approach zero stress amplitude, while the Miner –Haibach model extends the original S-N curve below the fatigue limit to the zero stress amplitude with a flatter slope factor 2k-1. Stanzl et al. (1986) concluded that a good agreement is found for measured and calculated results according to the Miner-Haibach model [5].

Through many years of experience and testing, empirical relationships that relate the data among ultimate tensile strengths and endurance limit for steels at 10^6 cycles have been developed.

3.1 Ultimate Tensile Strength of a Component

The mean ultimate tensile strength of a standard specimen can be determined by averaging the static test results of several smooth, polished round test specimens. If the test data are not available the ultimate tensile strength of a component can be estimated by a hardness value.

Lee and Song (2006) reviewed this relationship between hardness and mean ultimate tensile strength of a component [6]. They propose an equation as:

$$\sigma_{t,u,std} (MPa) = 3.45HB \quad (3.5)$$

Where $\sigma_{t,u,std}$ is the mean ultimate tensile strength of a standard material test specimen and HB is the Brinell hardness.

It has been also strongly proven that surface treatment and local notch geometry have a strong effect on ultimate tensile strength of a notched component. Taking into consideration the size effect of a specimen, the ultimate tensile strength of a real component ($\sigma_{t,u}$) with a survival rate (reliability) of R_r is estimated as follows:

$$\sigma_{t,u} = C_D C_R \sigma_{t,u,std} \quad (3.6)$$

where C_R is introduced as the reliability correction factor and C_D is the size correction factor.

3.2 Reliability Correction Factor

If test data are not available, a statistical analysis cannot be performed to account for variability of the ultimate tensile strength. In the absence of the statistical analysis, the suggested reliability values for various reliability levels are given in Figure 3.3.

These values are derived on the assumption of a normally distributed ultimate tensile strength

Table 3.1: Reliability correction factors, C_R (Adapted from FKM-Guideline)

Reliability	C_R
0.5	1.000
0.90	0.897
0.95	0.868
0.975	0.843
0.99	0.814
0.999	0.753
0.9999	0.702
0.99999	0.659

It has been specified that the ultimate strength of a component for a design should be based on the probability of a 97.5% survival rate, meaning corresponding C_R value of 0.843 [7].

3.3 Stress Correction Factors

The stress correction factor is used to correlate the different material strengths in compression or shear with respect to that in tension, and can be found in Figure 3.4. Also note that $C_\sigma = 1.0$ for tension, for general steels.

Table 3.2: Stress correction factors in compression and in shear

Materials	C_σ	C_τ
Case hardening steel	1	$1/\sqrt{3} = 0.577^*$
Stainless steel	1	0.577
Forging steel	1	0.577
Steel other than above types	1	0.577
Steel castings	1	0.577
Ductile irons	1.3	0.65
Malleable cast iron	1.5	0.75
Grey cast iron	2.5	0.85
Aluminum alloys	1	0.577
Cast aluminum alloys	1.5	0.75

*Note that $1/\sqrt{3} = 0.577$ is based on the von Mises yield criterion.

3.4 Load Correction Factors

The stress gradient of a component in bending or torsion can be taken into account by the load correction factor, also called the section factor. The load correction factor is defined as the ratio of the nominal stress at global yielding to the nominal stress at the initial notch yielding.

The load correction factors also depend on the type of material according to FKM Guideline [7]:

For steels, steel castings, ductile irons,

$$C_{b,L} = \sqrt{\frac{\sigma_{t,y \max}}{\sigma_{t,y}}} \quad (3.7)$$

where $C_{b,L}$ is introduced as the load correction factor, $\sigma_{t,y\max}$ for the ultimate tensile strength and $\sigma_{t,y}$ for tensile yield strength of related material.

3.5 Component Endurance Limit Under Fully Reversed Loading

The endurance limit is defined as the stress amplitude for a fully reversed loading at an endurance cycle limit ($N_E=10^6$ cycles). Since R is defined as the ratio of minimum stress to maximum stress, a fully reversed loading is also called and $R=-1$ loading. Even though the endurance limit is occasionally expressed in terms of range in some books, it is worth noting that the endurance limit in here is clearly defined in amplitude.

With the probability of an $R_r\%$ survival rate, the endurance limit for a smooth, polished component, under fully reversed tensile or shear stress can be found from $\sigma_{t,u}$, which has already taken into account the factors for size and reliability:

$$\sigma_{s,\sigma,E} = C_{\sigma,E} C_{E,T} \sigma_{t,u} \quad (3.8)$$

$$\sigma_{s,\tau,E} = C_\tau \sigma_{s,E,\sigma} \quad (3.9)$$

where $C_{E,T}$ is the temperature correction factor for the endurance limit and $C_{\sigma,E}$ is the endurance limit factor for normal stress. C_τ stands for shear stress correction factor.

It has been found that the endurance limit of a notched component is affected by the residual stress due to high stress concentration or stress gradient. These effects have been empirically quantified by FKM Guideline [7].

For example, the endurance limit values of a notched, rod shaped component under fully reversed loading in axial, $\sigma_{s,ax,E}$ bending $\sigma_{s,b,E}$ and shear $\sigma_{s,s,E}$ and torsion $\sigma_{s,\tau,E}$ can be obtained as follows:

$$\sigma_{s,ax,E} = \frac{C_s \sigma_{s,\sigma,E}}{K_{ax,f} + \frac{1}{C_{\sigma,R}} - 1} \quad (3.10)$$

$$\sigma_{s,b,E} = \frac{C_s \sigma_{s,\sigma,E}}{K_{b,f} + \frac{1}{C_{\sigma,R}} - 1} \quad (3.11)$$

$$\sigma_{s,s,E} = \frac{C_s \sigma_{s,\tau,E}}{K_{s,f} + \frac{1}{C_{\tau,R}} - 1} \quad (3.12)$$

$$\sigma_{s,t,E} = \frac{C_s \sigma_{s,\tau,E}}{K_{t,f} + \frac{1}{C_{\tau,R}} - 1} \quad (3.13)$$

where C_s is the surface treatment factor, $C_{\sigma,R}$ and $C_{\tau,R}$ are the roughness correction factors for normal and shear stresses respectively and K stands for related fatigue notch factor.

3.5.1 Fatigue Notch Factors

It was believed that at the same crack initiation life near the endurance cycle limit of 10^6 cycles, the local surface stress (σ_E^e) at the stress concentration location of a notched component would be identical to the surface stress of a smooth components. Since this belief provides $\sigma_E^e = \sigma_{s,E}$ and $\sigma_E^e = K_t \sigma_{s,E,notch}$ where K_t and $\sigma_{s,E,notch}$ are elastic stress concentration factor and the nominal stress of notched component. To conclude that $\sigma_{s,E,notch}$ is smaller than $\sigma_{s,E}$ by a factor K_t .

Tryon and Dey (2003), presented a study revealing the effect of fatigue strength reduction for steels [8]. The study has indicated at the same endurance cycle, the presence of a notch on a component under cyclic stressing reduces the nominal stress of a smooth component by a factor K_f instead of K_t . The K_f is termed the fatigue notch factor or fatigue strength reduction factor defined as follows:

$$K_f = \frac{\sigma_{s,E,smooth}}{\sigma_{s,E,notched}} \leq K_t \quad (3.14)$$

The stress field intensity theory is valid in the endurance cycle limit regime where the peak local surface stress is approximately equal to the true surface stress. According to stress field intensity theory, the average stress is responsible for the fatigue life. The average stress is defined as $K_f \sigma_c$ as opposed to the local surface stress $K_t \sigma_c$ where σ_c is the nominal stress of a notched component.

Figure 3.3 schematically shows two notched components with the same local surface stress and steel material. Note that the subscripts numbers 1 and 2 denote the notched components 1 and 2.

Based on the stress field intensity theory the K_f factor is closely related to a notch root radius (or a stress gradient) and the strength of materials. Therefore, several empirical methods have been developed to determine the $K_t - K_f$ relationship based on any combination of the earlier two parameters.

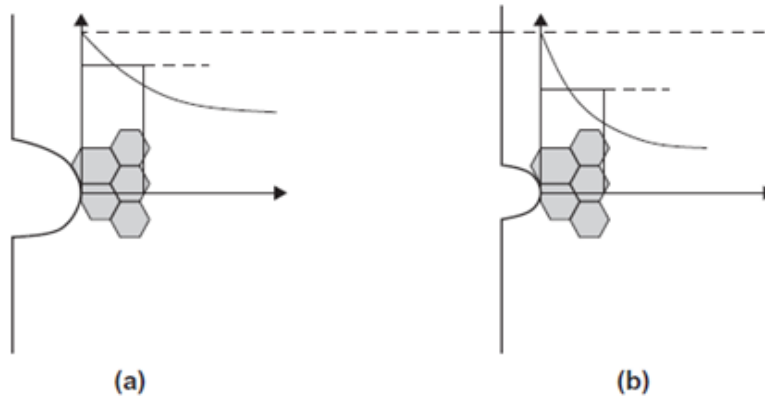


Figure 3.3: Effect of notch size and stress gradient on K_f (a) large notch radius and mild stress gradient and (b) small notch radius and steep stress gradient.

For example, a notch sensitivity factor q was introduced by Peterson (1959) as follows:

$$q = \frac{K_f - 1}{K_t - 1} \quad (3.15)$$

where q is a function of a notch root radius and the ultimate tensile strength of a material. Also the K_t / K_f ratio or the supporting factor (n_k) was developed:

$$n_k = \frac{K_t}{K_f} \quad (3.16)$$

where n_k depends on a relative stress gradient and tensile yield strength or on a notch root radius and ultimate tensile strength [7].

Constant amplitude stress-life curve for a notched component under fully reversed loading.

3.5.1.1 Constant amplitude local stress-life curve

The constant amplitude local stress-life curve for a notched component is preferable if the local stress is determined by a linear elastic analysis. The following equations are valid:

$$\sigma_E^e = K_t \sigma_{s,E,notch} \quad (3.17)$$

$$K_f = \frac{\sigma_{s,E,smooth}}{\sigma_{s,E,notched}} \quad (3.18)$$

$$n_k = \frac{K_t}{K_f} \quad (3.19)$$

where σ_E^e stands for local endurance limit, n_k is called the supporting factor. Also $\sigma_{s,E,smooth}$ and $\sigma_{s,E,notched}$ are introduced endurance limit of a smooth component and endurance limit of a notched component, respectively.

These equations lead to a new reference stress point at 10^6 cycles as

$$\sigma_E^e = K_t \sigma_{s,E,smooth} \quad (3.20)$$

With the assumption of the identical slope factor specified by Haibach (2003) the constant amplitude local stress-life curve can be determined [7]. Figure 3.4 shows the concept of defining the local endurance limit with respect to the endurance limit of a smooth component as well as the comparison between the constant amplitude S-N and σ_E^e -N curves for a notched component made of steels.

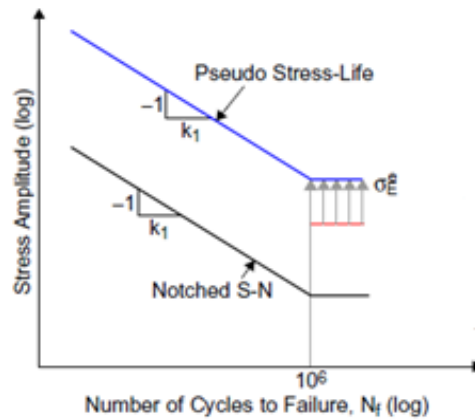


Figure 3.4: Constant amplitude s-n and σ_E^e -n curves for a notched component made of steel.

Local stresses are the elastic stresses calculated from a linear elastic analysis. The local stresses can be directly employed for stress-based fatigue strength assessment in the high cycle fatigue calculations with usage of notch factors [9].

3.5.1.2 Stress-life curve for a component under variable amplitude loading

For a component subjected to variable amplitude loading over time, a rainflow cycle counting technique is typically used to convert a complicated time-varying stress history to a series of discrete simple constant amplitude stress events that consist of a mean level and a number of stress cycles.

The fatigue life $N_{f,i}$ corresponding to the number of cycles to failure at the specific stress can be estimated from the component constant amplitude S-N curve. In this case, the accumulated fatigue damage is defined as the cycle ratio ($\frac{n_i}{N_{f,i}}$).

The Palmgren-Miner linear damage rule is adopted to calculate the accumulated damage, which assumes fatigue damage occurs when the sum of the cycle ratios at each constant amplitude stress event reaches a critical damage value D_{PM} . According to their model the fatigue failure is predicted when

$$\sum \frac{n_i}{N_{f,i}} \geq D_{PM} \quad (3.21)$$

Palmgren and Miner found the critical damage value as 1.0 in their studies. For mechanical design, however, FKM Guideline (2003) recommends $D_{PM}=0.3$ for steels, steel castings and aluminum alloys.

The component constant amplitude S-N curve is supposed to be used for estimating the fatigue life of a component at a given constant amplitude stress event. But when Palmgren-Miner linear damage rule is applied to a component in variable amplitude loading, the stress cycles with amplitudes below the fatigue limit could become damaging if some of the subsequent stress amplitudes exceed the original fatigue limit.

Therefore, there is a need to modify the fatigue limit for a component subjected to variable amplitude loading history because the fatigue limit obtained from constant amplitude loading might be negatively influenced by periodic overloads. Two methods such as the Miner rule and the Miner–Haibach model were proposed to include the

effect of periodic overloads on the stress cycle behaviour below the original fatigue limit.

The Miner Rule extends the S-N curve with the same slope factor k to approach zero stress amplitude, while the Miner-Haibach model extends the original S-N curve below the fatigue limit to the zero stress amplitude with a flatter slope factor $2k-1$. Stanzl et al (1986) concluded that a good agreement is found for measured and calculated results according to the Miner-Haibach model.

3.5.2 Mean stress effect

From the perspective of applied cyclic stress, the fatigue damage of a component strongly correlates with the applied stress amplitude or applied stress range, and is secondarily influenced by the mean stress. The mean stress effect should be seriously considered in fatigue analyses. In the HCF regime, normal mean stresses have a significant effect on fatigue behaviour of components.

Mean normal stresses are responsible for the opening and closing state of micro cracks. Since the opening of micro cracks accelerates the rate of crack propagation and the closing of micro cracks retards the growth of cracks. Tensile mean stresses are detrimental and compressive mean stresses are beneficial in terms of fatigue strength. There is very little or no effect of mean stress on fatigue strength in the low cycle fatigue (LCF) regime where the large amount of plastic deformation significantly reduces according to effect of mean stress.

The mean stress effect can be represented by the mean stress (σ_m) or stress ratio (R). Both are defined as follows:

$$\sigma_m = \frac{\sigma_{\max} + \sigma_{\min}}{2} \quad (3.22)$$

$$R = \frac{\sigma_{\min}}{\sigma_{\max}} \quad (3.23)$$

where $\sigma_{\sigma, \max}$, $\sigma_{\sigma, \min}$, the maximum and the minimum normal stresses in a stress cycle.

A number of mathematical models have been developed that allow the effects of mean stress on stress amplitude to be predicted from fully reversed-bending data. Most known theories for mean stress effect on fatigue phenomenon belongs to Goodman

and Gerber. Goodman developed a linear model, while Gerber used a parabolic model (Figure 3.5). The test data for ductile material usually fall closer to the Gerber parabolic curve; however because of the scatter in fatigue data and the fact that notched data fall closer to the Goodman line, the more conservative Goodman relationship is often used in practice. If the component design is based on the yield rather than ultimate strength, as most are, then the even more conservative Soderberg relationship can be used. Mathematically, the three relationship can be expressed by:

$$\sigma_a = \sigma_e \left[1 - \left(\frac{\sigma_m}{\sigma_u} \right)^x \right] \quad (3.26)$$

The parameter x can be taken as 1 for the Goodman line, and as 2 for the Gerber curve and $\sigma_u = \sigma_y$ for Soderberg curve.

where σ_u stands for ultimate tensile strength, σ_a for fatigue strength for n cycles under the mean stress of σ_m and σ_y for yield strength of related material.

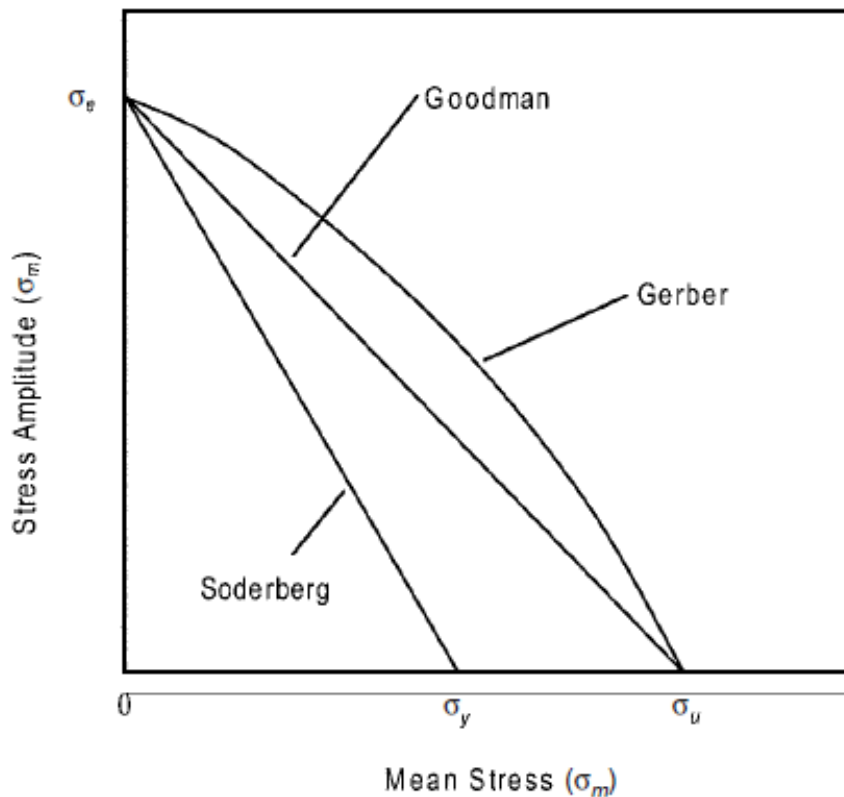


Figure 3.5: Comparison of Goodman, Gerber and Soderberg models for relating mean stress to stress amplitude

If there are no experimental data available, FKM Guideline (2003) recommends to use Haigh's diagram based on a normal stress can be classified as four regimes, as shown in Figure 3.6 and summarized as follows:

- Regime I is applied for the stress ratio $R > 1$ where the maximum and minimum stresses are under compression
- Regime II is applicable to the case of $-\infty < R < 0$ where $R = -\infty$ is the zero compression stress; $R = -1$, the fully reversed stress, $R = 0$, the alternating tension stress with a zero minimum stress.
- Regime III is for $0 < R < 0.5$ where the maximum and minimum stresses are under tension.
- Regime IV is for $R > 0.5$, the regime of high alternating tension stress.

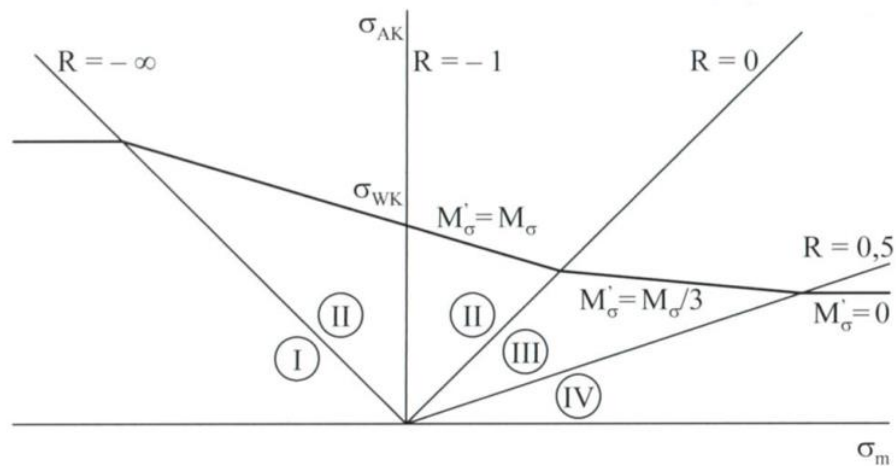


Figure 3.6:Haigh diagram for mean stress effect on normal stress

4. STRESS BASED MULTIAXIAL FATIGUE ANALYSIS

A linear elastic analysis in conjunction with a multiaxial notch stress-strain analysis is employed to estimate the true notch stresses and strains. A uniaxial fatigue analysis approach has been commonly coded in some commercial fatigue analysis tools. Multiaxial fatigue analysis is the most effective fatigue analysis for use in HCF regime.

Multiaxial cyclic stresses at high stress concentration areas can be commonly found in structures under multiaxial cyclic loads. Due to geometric constraints at notches, multiaxial loads can result in either a uniaxial stress state as applied to a notched thin metal sheet or in a multiaxial stress state as applied to a notched shaft or rod.

4.1 Fatigue Damage Models

The existing stress based multiaxial fatigue damage models can be categorized into three different classifications. Equivalent stress approach, critical plane approach, and Dang-Van multiscale approach.

Although these approaches are used to assess the damage at fatigue limit, they can be extended for the fatigue life prediction in the HCF regime by simply replacing the fatigue limit with an S-N material equation.

4.2 Equivalent Stress Approach

The existing equivalent stress models are reviewed in here.

4.2.1 Maximum Principal Stress Hypothesis

According to maximum principal stress hypothesis fatigue initiation occurs when

$$\sigma_{PS,a} = \sigma_{1,a} \geq \sigma_{E,R=-1} \quad (4.1)$$

Where $\sigma_{1,a}$ is the maximum principal stress amplitude and the subscripts PS and a represent the principal stress and amplitude, respectively. $\sigma_{E,R=-1}$ stands for fully reversed fatigue limit for normal stress

4.2.2 Maximum shear stress theory (Tresca Theory)

According to maximum shear stress hypothesis fatigue initiation occurs if

$$\tau_{MS,a} = \sigma_{1,a} - \sigma_{3,a} \geq \tau_E \quad (4.2)$$

Where $\sigma_{3,a}$ minimum principal stress amplitude and the subscript MS is represent the maximum shear τ_E stands for fully reversed fatigue limit for shear stress.

4.2.3 Von Mises hypothesis

According to Von-Mises hypothesis fatigue initiation occurs when

$$\sigma_{VM,a} + \alpha_{VM} \sigma_{VM,m} \geq \sigma_{E-R=-1} \quad (4.3)$$

where α_{VM} is the mean stress sensitivity factor and $\sigma_{VM,a}, \sigma_{VM,m}$ stands for the mean stress and Von Mises stress amplitudes, respectively.

$$\sigma_{VM,m} = \sigma_{1,m} + \sigma_{2,m} + \sigma_{3,m} = \sigma_{x,m} + \sigma_{y,m} + \sigma_{z,m} \quad (4.4)$$

$$\sigma_{VM,a} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_{1,a} - \sigma_{2,a})^2 + (\sigma_{2,a} - \sigma_{3,a})^2 + (\sigma_{1,a} - \sigma_{3,a})^2} \quad (4.5)$$

$$\sigma_{VM,a} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_{x,a} - \sigma_{y,a})^2 + (\sigma_{y,a} - \sigma_{z,a})^2 + (\sigma_{x,a} - \sigma_{z,a})^2 + 6(\tau_{xy,a}^2 + \tau_{yz,a}^2 + \tau_{xz,a}^2)} \quad (4.6)$$

here, x,y,z are the local coordinates axes and 1,2,3 are the principal axes.

4.3 Critical Plane Approach

The critical plane approach is commonly used for fatigue analyses of components under variable amplitude, multiaxial loading. It is known that non proportional loading results in rotating the maximum principal and the maximum shear stress planes at a fatigue damage initiation location where the potential damage initiation location orientation is changing as well.

A final damage location will eventually initiate on a certain orientation where the fatigue damage parameters are maximized. Therefore, these approaches essentially involve the critical plane searching technique to identify the orientations of the highest damage planes.

The fatigue damage parameter can be a function of shear stress and normal stress on the plane where the degree of damage is to be evaluated. The dominant stress used in a fatigue damage parameter depends on the fatigue damage mechanism of a specified material. For example, fatigue initiation life of ductile materials is typically dominated by crack initiation and growth on the maximum shear stress planes; therefore shear stress and normal stress are primary and the secondary damage parameters to be used respectively.

Based on the nucleation and growth properties of fatigue cracks, two types of cracks named as Case A and Case B cracks by Brown and Miller (1973) are considered. Case A crack is the surface crack, which tends to be very shallow and has a small aspect ratio. Case B crack is the in-depth crack, propagating into the surface [10]. As illustrated in Figure 4.1 and Figure 4.2, case a crack grow on the planes perpendicular to the x -axis. The critical plane searching techniques based on Case A and Case B cracks are also addressed in these figures.

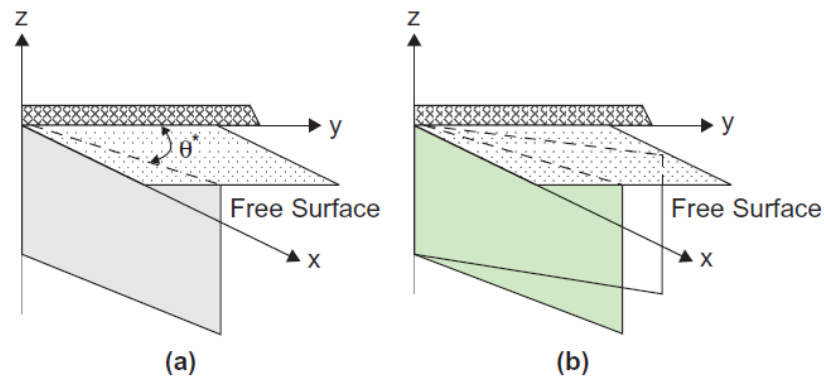


Figure 4.1: Case A cracks. (a) critical plane orientation and (b) potential critical planes at a crack initiation site)

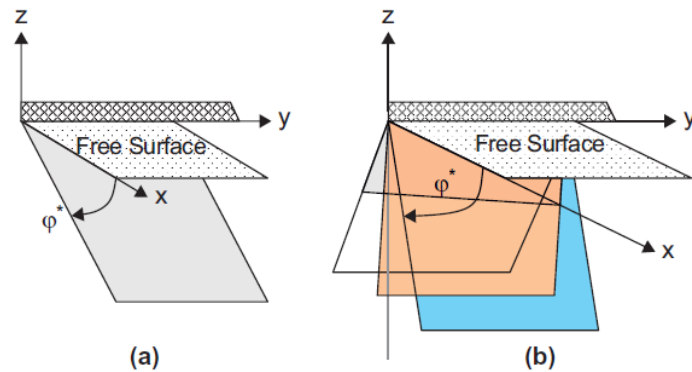


Figure 4.2: Case B cracks. (a) critical plane orientation and (b) potential critical planes at a crack initiation site)

Usually fatigue damage cracks type Case A occur in fatigue calculations. Therefore only Case A crack is to be analysed.

In Figure 4.2 the two possible critical planes that are perpendicular to the z axis can be seen, each of which locates at an interference angle θ^* with respect to y-z surface. And the typical normal and shear stresses acting on an interference plane with the angle of θ^* are depicted in Figure 4.3.

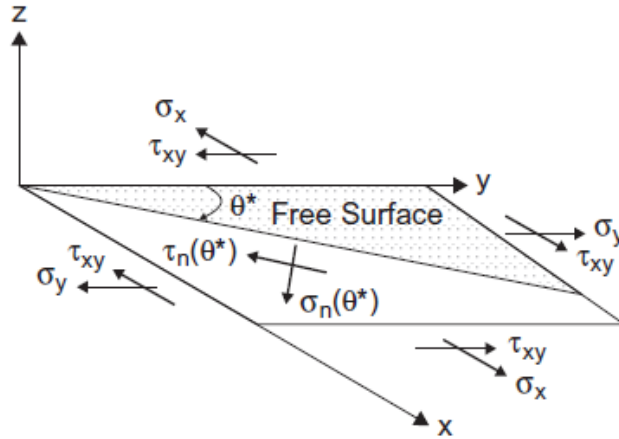


Figure 4.3: Normal and shear stresses on an interference plane θ^* from the y-z plane for case a cracks.

For the plane of maximum damage, it is necessary to assess the fatigue damage parameter on each critical plane by varying θ^* from 0° to 180° . However, for computational reasons, the continuous problem is solved at 5° intervals, and it is a common practice, but it could lead to discretization errors in some cases.

In summary, critical plane searching approach, at least in numerical sense would require an iterative stress transformation technique and the proper definition of fatigue damage parameter.

Stress Transformation

The state of stress at a point to a global xyz coordinate is given by the stress matrix

$$[\sigma]_{xyz} = \begin{bmatrix} \sigma_{xx} & \tau_{xy} & \tau_{zx} \\ \tau_{xy} & \sigma_{yy} & \tau_{yz} \\ \tau_{zx} & \tau_{yz} & \sigma_{zz} \end{bmatrix} \quad (4.7)$$

And the state of a stress at the same point relative to a local $x'y'z'$ coordinate system is noted as

$$[\sigma]_{x'y'z'} = \begin{bmatrix} \sigma'_{xx} & \tau'_{xy} & \tau'_{zx} \\ \tau'_{xy} & \sigma'_{yy} & \tau'_{yz} \\ \tau'_{zx} & \tau'_{yz} & \sigma'_{zz} \end{bmatrix} \quad (4.8)$$

It is shown that the stress transformation is

$$[\sigma]_{x'y'z'} = [\mathbf{T}] \cdot [\sigma]_{xyz} \cdot [\mathbf{T}]^T \quad (4.9)$$

$[\mathbf{T}]$ is the coordinate transformation matrix that transforms a vector in the xyz system to a $x'y'z'$ system and $[\mathbf{T}]^T$ stands for the transpose of $[\mathbf{T}]$.

4.4 Fatigue Damage Parameters

There are several fatigue damage parameters used while calculating the strength endurance limit in multiaxial stress based fatigue analysis. However, here one of the most widely used one, Findley damage parameter will be reviewed.

Findley (1959) first proposed the concept of the critical plane approach and assumed a fatigue crack will form on the plane where the damage parameter is maximized and exceeds the fatigue limit in shear τ_E . It can be expressed as follows:

$$(\tau_{findley,a})_{\max} = \tau_E \quad (4.10)$$

$$\tau_{findley,a} = \tau_a + k_F \sigma_{n,\max} \quad (4.11)$$

where k_F is the normal stress sensitivity factor, representing the influence of the maximum normal stress on the maximum shear stress amplitude [11].

k_F can be determined by fatigue limits from two loading conditions. If $\sigma_{n,\max}$ exceeds the tensile yield strength of the material, then $\sigma_{n,\max} = \sigma_{t,y}$.

As a conclusion, based on the assumption that there is very little plastic deformation in the HCF regime, the local stress components calculated from a linear elastic FEA can be used to estimate the fatigue damage parameter and predict life from the local stress-life curve.

The most widely used and best fitted to experimental data were mentioned in this section.

5. FATIGUE ANALYSIS OF WELDED STRUCTURES

5.1 Introduction

Compared to homogeneous base materials, fatigue in welds is an even more complex phenomenon. Apart from thermal cycles during welding, which strongly affect the base material, the fusion process and the existence of a filler lead to very inhomogeneous structure, varying in mechanical properties.

As mentioned earlier, fatigue cracks usually occur at the surface of components, although when dealing with welded structures some of them can nucleate away from it, in welding defects. Porosity, slugs, inclusions or lack of melting are potential sites for crack initiation and reduce the fatigue strength of the component. Moreover, residual stresses and distortions also have a major influence in fatigue behaviour. Due to these facts, endurance limits reported for metals in engineering books or in articles, have usually been obtained with smooth polished specimens, have little relevance in design of weldments.

Nonetheless, this does not mean that defects in welds subjected to cyclic loading cannot be tolerated, but the task demands a proper analysis of each situation in order to determine the severity of discontinuities.

5.2 Elastic Stress Concentration Factor

According to stress concentration effects, the peak stress, σ_p occurring in bending, close to a discontinuity can be related to the average stress on the net section, σ_{net} . The same can be defined in torsion using shear stresses instead. Both cases can be expressed as:

$$\sigma_p = K_t \sigma_{net} \quad (5.1)$$

$$\tau_p = K_{ts} \tau_{net} \quad (5.2)$$

In which K_t and K_{ts} are the theoretical stress concentration factor for tension and torsion respectively. Figure 5.1 shows the two ways of presenting stress around a hole in a sheet, subjected to uniaxial load.

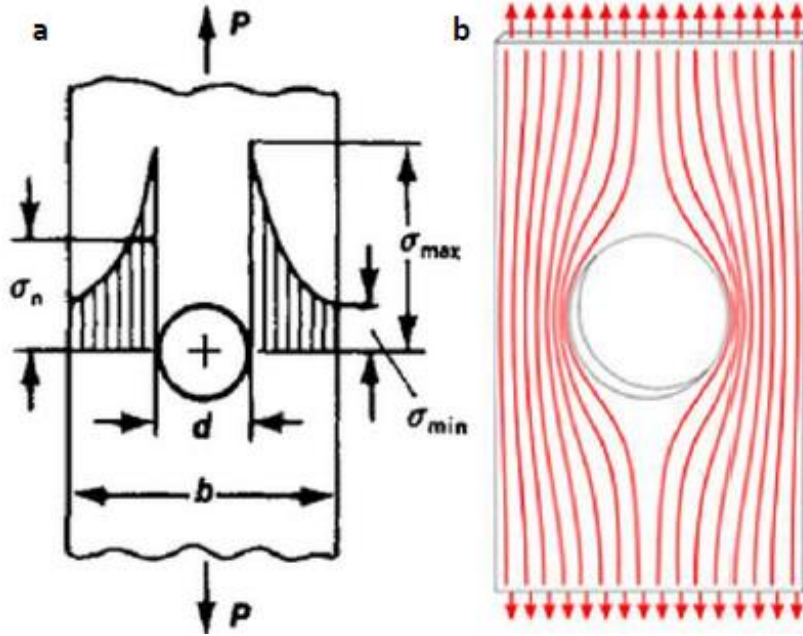


Figure 5.1: Stress profiles around a hole in a sheet

Theoretical stress concentration factors apply mainly to ideal elastic material and vary with geometry and loading. In some cases, plastic deformation can occur in front of the notch, modifying the stress distribution. Other concepts were therefore introduced, such as the effective stress concentration factor K_e , which is obtained experimentally as the ratio between the rupture load in a bars of diameter d , and the rupture load another set of round bar with a circumferential groove, with d as the diameter at the root of the groove [12].

It can be noted that $1 \leq K_e \leq K_t$, and it is dependent on geometry of the body, the nature of the load and material properties. The relationship between K_e and K_t is defined by the concept of notch sensitivity, q , presented as:

$$q = \frac{K_e - 1}{K_t - 1} \quad (5.3)$$

In fatigue design, both q and K_e concepts are used, but the latter is replaced by K_f or K_{fs} defined as:

$$K_f = \frac{\text{Fatigue_Limit_of_Unnotched_Specimen_}(Axial_or_Bending)}{\text{Fatigue_Limit_of_Notched_Specimen_}(Axial_or_Bending)} = \frac{\sigma_f}{\sigma_{nf}} \quad (5.4)$$

$$K_{fs} = \frac{\text{Fatigue_Limit_of_Unnotched_Specimen_}(Shear)}{\text{Fatigue_Limit_of_Notched_Specimen_}(Shear)} = \frac{\tau_f}{\tau_{nf}} \quad (5.5)$$

where K_f and K_{fs} are the fatigue notch factor for normal stress and shear stress, respectively for $R=-1$. Although it was originally proposed for high cycle regime, the factor be transferred to medium and low cycle fatigue. It is worth mentioning that if notch sensitivity is not taken into consideration in design and only stress concentration factor is employed, predictions may be too conservative, and therefore on the safer side.

Welds in a welded structures are not far from being a stress concentrator, thanks to differences in section, shaped of the bevels, non-welded root gaps all of which create high stress concentrations with widely varying geometric parameters.

However, the above definition cannot be directly redirected to welded joints, because the material conditions at the weld notch differ from the parental material characteristics. Moreover, residual stresses are not the same in each case. These inconveniences can be overcome by proper selection of constant K_f .

Radaj (1990) deeply describes another convenient parameter. *Fatigue strength reduction factor* compares the endurance limit of typically uniaxially loaded weld joints with the endurance limit of unwelded parent material [13]. It is particularly useful for design purposes and for predicting tendencies. In fact, fatigue strength reduction factor results from the reciprocal value of the fatigue notch factor. However, it must be considered that the latter is defined in relation to a plain polished specimen in the mill-finished condition [14].

Due to fact that local stress distribution in the notch tip is more dependent on the notch radius r than other geometrical parameters, it was found that plotting q vs. r decrease the scatter resulting from specimen geometry influence. It can be clearly seen from the Figure 5.2 that q is larger for stronger materials and for larger specimens.

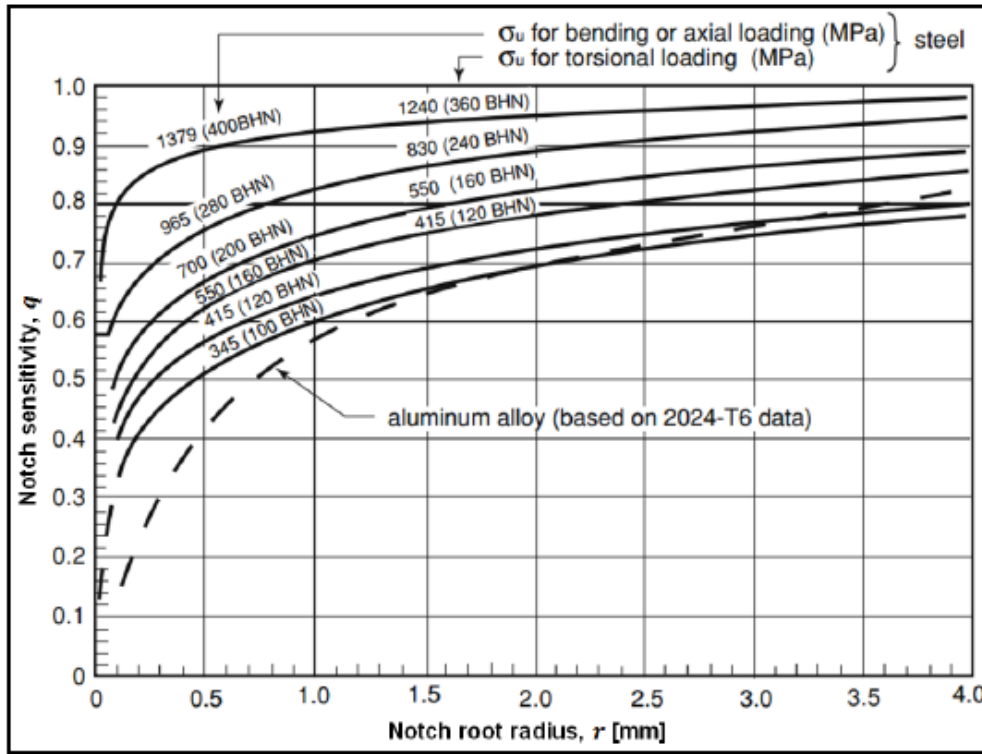


Figure 5.2: Notch sensitivity factor for steels and aluminum alloys for $sm=0$

In order to determine the relationship between K_r and K_f , Peterson (1959) made the hypothesis that fatigue failure occurs when the stress, over a certain distance below the surface where the stress gradient is steep, is equal to or greater than the fatigue limit of a smooth specimen[15]. Assuming a linear decrease of the notch tip stress field, an empirical relation is as:

$$q = \frac{1}{1 + \alpha / r} \quad (5.6)$$

Where α is a material constant that should be determined by test data. The relevant local material constant at the potential site for crack initiation should be derived from hardness measurements when dealing with the welded components.

The relationship between K_r and K_f can be seen in Figure 5.3. It is noted that the latter tends to infinity when notch radii tends to zero, whereas the fatigue notch factor is usually restricted to a narrow range, depending on the type of joint. Other equations relating both parameters can be found elsewhere in literature both for unwelded notched specimens and for welded joints [14].

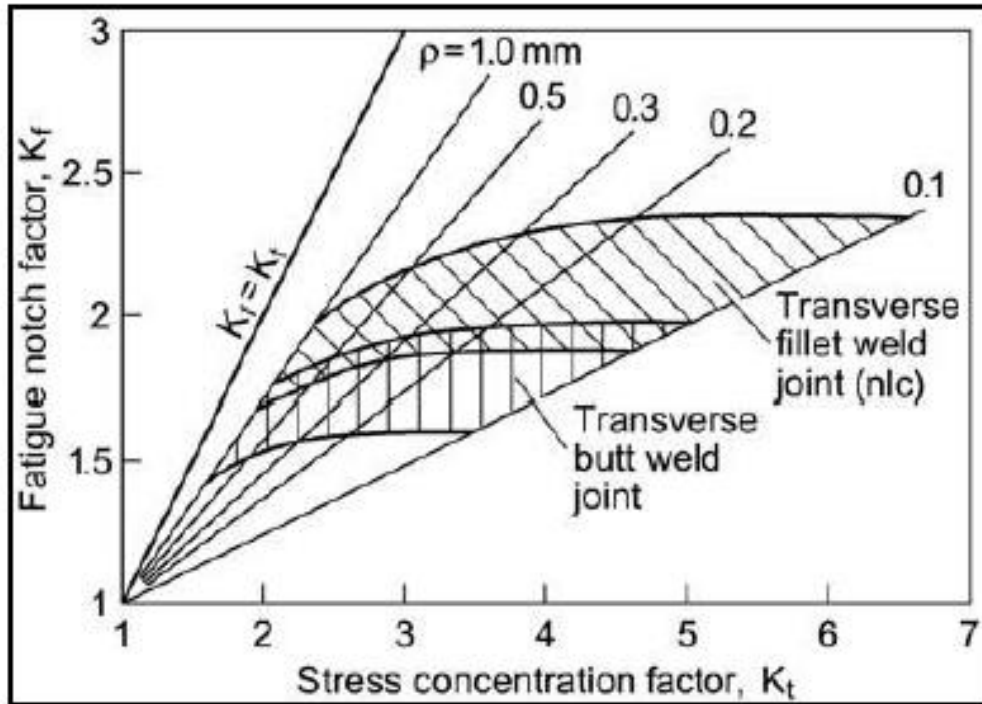


Figure 5.3: K_f and K_t for transverse non-carrying load fillet welds and transverse butt welds

It was mentioned earlier that K_e , and therefore K_f values, can vary with the loading mode. Although it is generally found for simple notch configurations that they are reduced in bending than in tension and even lower in torsion, this statement cannot be generalized for some welded joints. This can be seen in Figure 5.4, where fatigue notch factor in the toe (first number) and in the toe (second number) are shown for some typical weld designs.

Important results of what have been concluded here are first, that not only the peak stress at the notch root influences the fatigue strength, but also the stress gradient ahead of the tip; and secondly the notch radius affects fatigue strength and not just K_t , what leads to the corollaries that two similar discontinuities with the same theoretical stress concentration factor can have different effect on fatigue. Furthermore, Frost et. al. (2001) analysed a large amount of experimental data and came up with the following results:

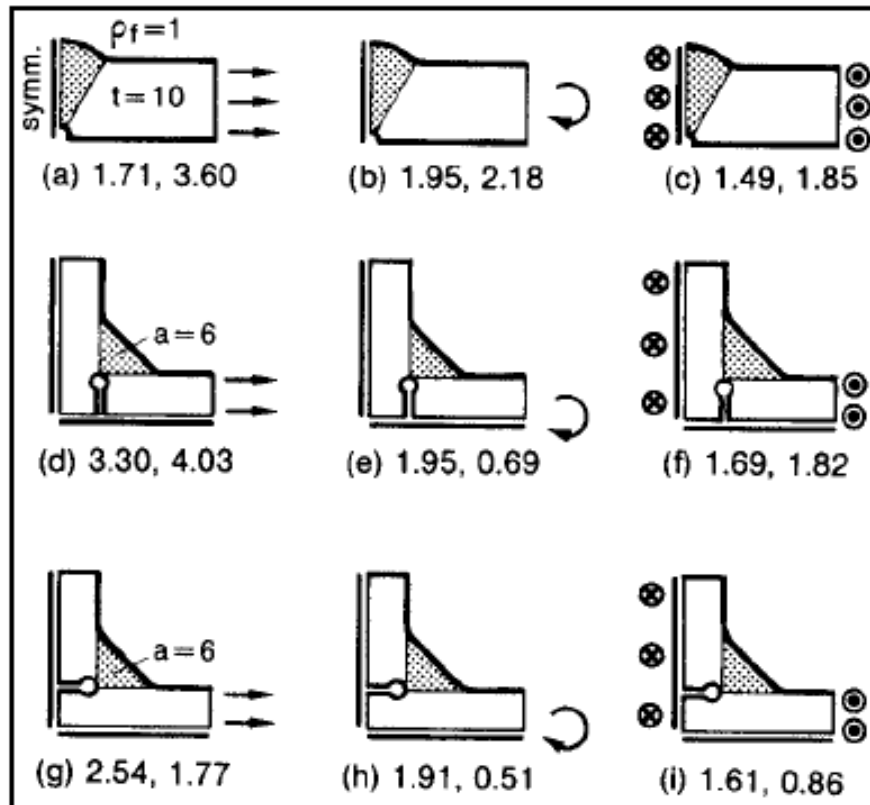


Figure 5.4: Butt joints with root (a,b,c) cruciform joint (d,e,f) and t joint (g,h,i) and their fatigue notch factors at the weld toe and root, under bending, tension and torsion

- For low K_t values, K_f is generally lower,
- For high K_t values, K_f is often quite lower than K_t ,
- For a given material and certain geometry, there is apparently a value of K_t at which K_f reaches maximum.

It has been demonstrated that the complex nature of stress raisers, enhanced when dealing with welded components [16].

5.3 Stresses in Welded Components

Stresses may arise from a variety of sources such as vibrations, temperature changes, pressure fluctuations and dead weights, and residual stresses from manufacturing processes.

5.3.1 Fatigue assessment according to local nominal stress approach

Nominal stress is defined as the stress in a component resolved usually using elementary theories of structural mechanics, such as beam theory. It is generally calculated adding the effect of stresses produced by external forces and moments in the region containing the weld detail, but sufficiently far away from the attachment to disregard any effect of it in the stress distribution. Macro geometrical effects generated by holes, edges, misalignments or other types of discontinuities, are not included in the aforementioned definition. Structural discontinuities can be seen in Figure 5.5.

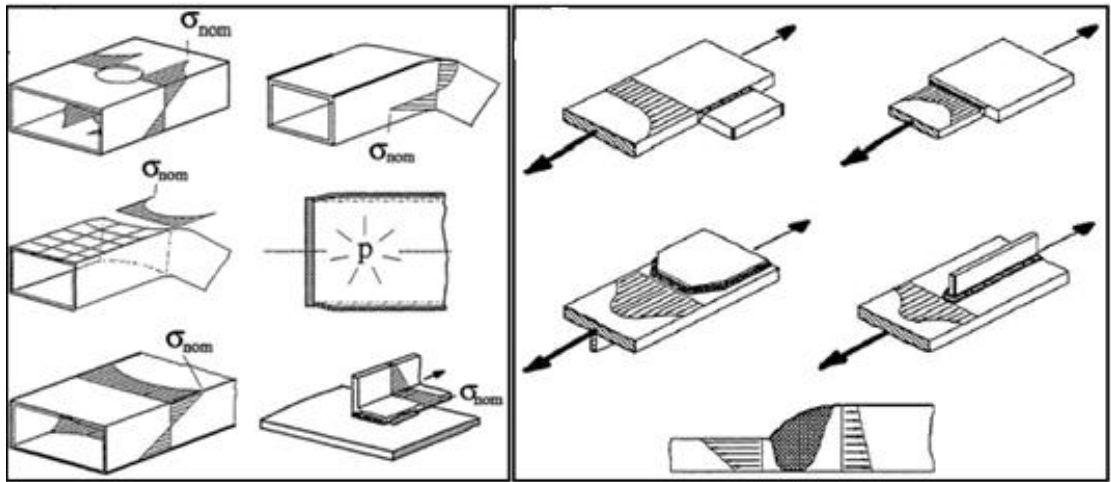


Figure 5.5: Selected examples of macro geometrical discontinuities and structural discontinuities

Nominal stress amplitude in a base material critical cross section of small specimens or near full scale beams is plotted against to failure (S-N curves). Service life result from comparison between these S-N curves and the nominal stress spectrum calculated by theory of damage accumulation.

Welding defects and typical flaws resulting from fusion welding processes are considered by the concept of quality classes for particular weld details. The IIW recommendations identify the S-N curves by the characteristic fatigue strength of the detail in MPa at 2 million cycles, at a high stress ratio, $R \geq 0.5$, with a survival probability $p_s = 97.5\%$ [9]. Its value is called the fatigue design class (FAT) and this is assumed to be valid for any R ratio [17].

Uniform design S-N curves for each design class can be found in Figure 5.6 in the case of steel structures, showing usually a slope with $k=3$ for normal stresses and the constant amplitude knee point at 10 million cycles.

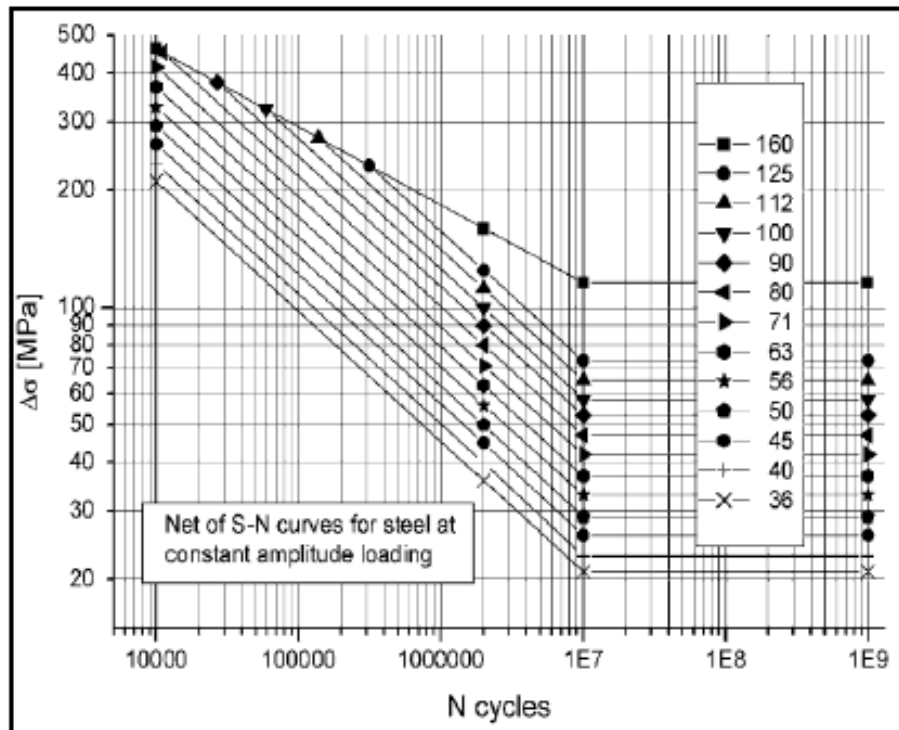


Figure 5.6: Wöhler curves for steel and for particular weld quality values

In cases where high tensile residual stresses exist and they have the same direction of the tensile applied load, S-N curves present a step in the range of medium cycle fatigue strength, due to local yielding caused by strong relief of residual stresses. This leads to reduced mean stress, which means an increase in fatigue strength. Low stress high cycle fatigue may not cause notable stress relief because yielding is not reached by the governing state of stress [13].

To conclude, nominal stress approach is a statically robust method wide spread in industry and still preferred in many areas that require only a simplified stress determination. Furthermore latest investigations have supported the revisions by weld imperfection acceptance criteria and according to IIW recommendations [18]. Many other construction and design codes make us of nominal stress approach as the basis for their guidelines.

Nominal stress concept for fatigue strength assessment of welded joints provides a good prediction on fatigue lives and strength only in the case of components with well-

defined loading conditions and simple geometries of the detail, contemplated by the standards.

5.4 Improvement Techniques of Fatigue Characteristics of Welds

Post weld improvement techniques may increase the fatigue resistance of welded joints, generally as a result of an improvement in the weld profile, the residual stress conditions or the environmental conditions of the welded area. They may be used to increase the fatigue strength of new structures, notably if a weld is found to be critical, or as a part of repair or upgrading of an existing structure.

The main improvement techniques are:

a) Methods for improvement of weld geometry

- Machining or grinding of but weld cap flush to the surface
- Machining or grinding of the weld transition at the toe
- Remelting of the weld toe by TIG-, plasma or laser dressing

b) Methods for improvement of residual stress conditions

- Peening(Hammer, needle, shot, brush-peening or ultrasonic treatment)
- Overstressing
- Stress relief

c) Methods for improvement of environmental conditions

- Painting
- Resin Coating

The effects of all important improvement techniques are sensitive to the method of application and the applied loading, being most effective in the low stress high cycle regime. They may also depend on the material, the structural detail, the applied stress ratio and the dimensions of the welded joint. Consequently, fatigue tests for the verification of the procedure in the endurance range of interest are recommended.

5.5 Fatigue Strength Assessment of Welded Components According to FKM Guideline FKM (Forschungskuratorium Maschinenbau)

FKM guideline is mostly valid for mechanical engineering and related fields. For mechanically stressed components, the guideline allows an analytical fatigue strength assessment with variable amplitude and also static strength assessment of mechanical components. Generally it can be applied to:

- Components with geometrical notches
- Components with welded joints,
- Forged steel, stainless steels, aluminum components

FKM guideline is capable of calculating fatigue strength of components with nominal stress approaches.

Basically nominal stresses can be determined for rod and shell shaped components, in most cases by applying elementary equations of technical mechanics. The main characteristics of nominal stresses are as follows:

- Reference to a defined cross-section of components
- Neglect local peak stresses occurring as a result of geometrical notches.

For welded components, as a result, the calculation of the nominal stress in the weld is performed based on the section sizes, but using the characteristic values for the cross-section of the weld.

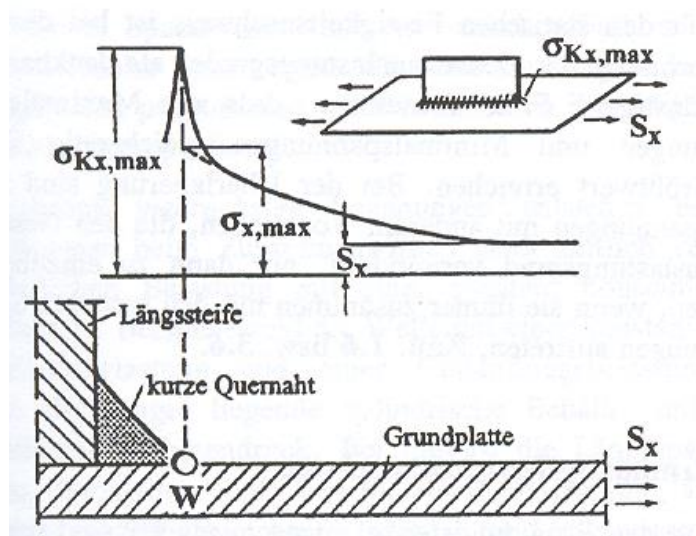


Figure 5.7: Shell-shaped welded component

5.5.1 Assessment of the fatigue strength of welded components

Fatigue strength calculation according to FKM guideline, the required parameters of stress spectrum shall be specified, the notations and relevant stress components are as follows:

- $\sigma_{a,i}$ = Amplitude in step i,
- $\sigma_{m,i}$ = Mean value with regard to $\sigma_{a,i}$,
- $\overline{N}$ = total number of cycles corresponding to the required fatigue life,
- n_i = Related number of cycles in step i,
- R = Stress ratio.

Stress components on the welds are as follows:

- $\sigma_{||}$ = Normal stress parallel to the weld.
- $\sigma_{\perp}$ = Normal stress perpendicular to the weld.
- $\tau_{||}$ = Shear stress parallel to the weld.
- $\sigma_{\perp,zd}$ = Normal stress perpendicular to the weld, resulting from an axial load.
- $\sigma_{\perp,by}$ = Normal stress perpendicular to the weld, resulting from bending around y axis.
- $\sigma_{\perp,bz}$ = Normal stress perpendicular to the weld, resulting from bending around z axis.
- $\tau_{||}$ = Shear stress parallel to the weld, resulting from the combination of the transverse forces.

5.5.2 Constant amplitude S-N curve

The constant amplitude S-N curve describes the relationship between stress amplitude and critical number of load cycles and shall be determined for each site of the assessment and each stress components. Based on the material constant amplitude S-N curve and taking into consideration the design factor, and mean stress factor, the components constant amplitude S-N curve results:

- Model I constant amplitude S-N curve exhibit one knee point and a horizontal profile (Steels and cast iron)
- Model II constant amplitude S-N curve exhibit two knee points and a horizontal profile (Aluminium and austenitic steel).

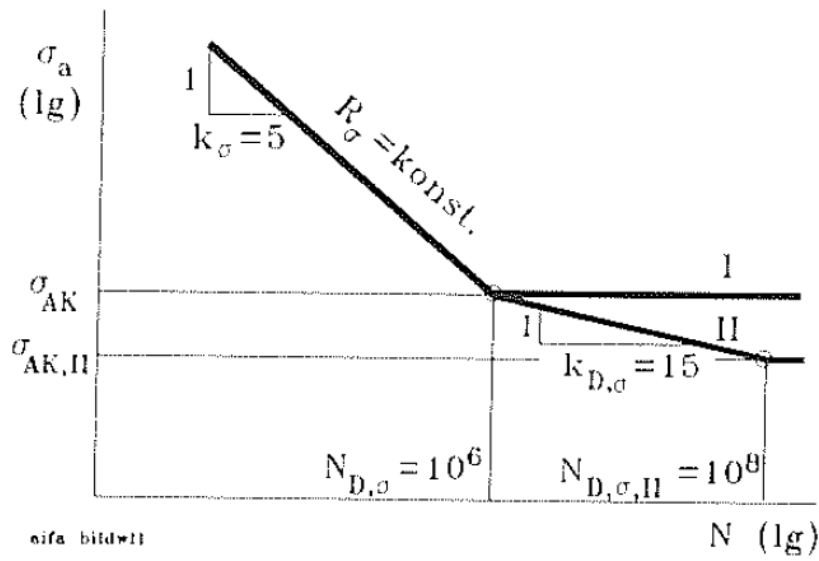


Figure 5.8: Constant amplitude S-N curve

Where σ_a stands for stress amplitude, σ_{AK} stands for component's fatigue limit, $\sigma_{AK,II}$ is component's endurance limit, N_D is the number of cycles at knee point, $N_{D,II}$ is the number of cycles at second knee point of the model and f_{II} is the factor by which the endurance limit is lower than nominal component fatigue limit.

5.5.3 Damage equivalent stress amplitude

The damage equivalent stress amplitude $\sigma_{a,qu}$ is a constant stress amplitude which is damage equivalent to the stress spectrum in question and has an assigned number of cycles equal to the number of cycles at the knee point of the component constant amplitude S-N curve, N_D .

The damage equivalent stress amplitude refers to N_D , and thus it allows the external form of the assessment.

For a given stress spectrum, the relevant damage equivalent stress amplitude can be calculated using 5.7. By using the damage equivalent stress amplitude, the assessment of the variable amplitude fatigue strength can be calculated.

$$\sigma_{a,qu} = \sigma_{a,1} / K_{BK} \quad (5.7)$$

where $\sigma_{a,1}$ is the maximum value of stress amplitude in stress spectrum, K_{BK} is the variable amplitude fatigue strength factor, σ_{equ} is the damage equivalent stress amplitude and $\sigma_m(\sigma_{\text{equ}})$ is the associated mean stress.

5.5.4 Design parameters

The following design parameters are relevant for welded components:

- $FAT_{\perp,\dots}$ = Fatigue class
- $f_{FAT,\sigma,\dots}$ = Factor for conversion from fatigue class to component fatigue limit for completely reversed stress
- f_t = Thickness factor
- K_V = Surface treatment factor

5.5.5 Fatigue class and conversion factor

The fatigue class (FAT) applies to normal stresses perpendicular and parallel to the weld and to shear stresses in materials.

The fatigue classes according to the welding detail can be found in FKM guideline.

Fatigue classes corresponds with the double amplitude existing at the reference number of cycles N_C . The stress ratio between reference number of cycles of the fatigue class and the number of cycles at knee point of the component constant amplitude S-N curve is as follows:

$$f_{FAT,\sigma} = 0.5 \left(\frac{N_C}{N_{D,\sigma}} \right)^{1/k_\sigma} \quad (5.8)$$

$$f_{FAT,\tau} = 0.5 \left(\frac{N_C}{N_{D,\tau}} \right)^{1/k_\tau} \quad (5.9)$$

where $N_{D,\sigma}$ and $N_{D,\tau}$ are the number of cycles at knee point of the components constant amplitude S-N curve. Table 5.1, N_C is the Reference number of cycles, for steels $N_C = 2 \cdot 10^6$ and k_σ and k_τ are the exponents of the constant amplitude S-N curve

Table 5.1: Steel and cast iron material model

Component	$N_{D,\sigma}$	$N_{D,II,\sigma}$	k_σ	$k_{II,\sigma}$	$f_{II,\sigma}$
Welded	$5 \cdot 10^6$		3	5	

5.5.6 The thickness factor

The thickness factor f_t accounts for the influence of the sheet metal thickness on the fatigue strength of the welds. The following equations are applied to sheet metal thickness:

$$t \leq 10mm, f_t = 1.1$$

$$10 < t \leq 25mm, f_t = \left(\frac{25mm}{t} \right)^{0.1} \quad (5.10)$$

$$t > 25mm, f_t = \left(\frac{25mm}{t} \right)^n \quad (5.11)$$

For longitudinally loaded welds, n can be taken as 0.1

For transversely loaded T joints, n can be taken as 0.3

For transversely loaded butt weld, n can be taken as 0.2

5.5.7 Component Strength Calculation

The fatigue strength shall be calculated according to component strength as follows:

- First the component fatigue limit for completely reversed stress should be identified by taking into consideration the design factors and design parameters,
- Then the amplitude of the component fatigue limit depending on the mean stress factor should be identified.
- the amplitude of the component variable amplitude fatigue strength depending on the variable amplitude fatigue strength

5.5.8 Component Fatigue Limit for Completely Reversed Stress

The critical nominal values of the component fatigue limits for completely reversed normal stress perpendicular and parallel to the weld and for completely reversed shear stresses are as follows:

$$\sigma_{WK,\perp} = FAT_{\perp} \cdot f_{FAT,\sigma} \cdot f_t \cdot K_V \cdot K_{NL,E} \quad (5.12)$$

$$\sigma_{WK,II} = FAT_{II} \cdot f_{FAT,\sigma} \cdot f_t \cdot K_V \cdot K_{NL,E} \quad (5.13)$$

$$\tau_{WK} = FAT_{\tau} \cdot f_{FAT,\tau} \cdot f_t \cdot K_V \quad (5.14)$$

where $FAT_{\perp...}$ is the fatigue class, $f_{FAT,\sigma...}$ is the factor for conversion from fatigue class to component fatigue limit for completely reversed stress, f_t is the thickness factor, K_V is the Surface treatment factor, if no surface treatment exists K_V can be taken as “1” and

$K_{NL,E}$ is the partial safety factor, for ductile materials, $K_{NL,E}=1$. The normal stress values $\sigma_{WK,\perp}$, $\sigma_{WK,II}$ apply to the number of cycles $N_{D,\sigma}=5.10^6$ and the shear stress value $\tau_{WK}=1.10^8$. The characteristic values for completely reversed stress of welded components correspond with the fatigue limit at N_D and the endurance limit is infinite.

5.5.9 Component fatigue limit as a function of mean stress

Mean stress has an important effect on the fatigue life of welded or non-welded components. Mean stress factor can be taken as 1 in the situations without the mean stress.

5.5.9.1 Amplitude of the component fatigue limit

The critical nominal values of the amplitude of the component fatigue limit for normal stress perpendicular and parallel to the weld and for shear stress are as follows:

$$\sigma AK_{\perp} = KAK_{\perp} \cdot KE_{\sigma} \cdot SWK_{\perp} \quad (5.15)$$

$$\sigma AK_{II} = KAK_{II} \cdot KE_{\sigma} \cdot SWK_{II} \quad (5.16)$$

$$\tau AK = KAK_{\tau} \cdot KE_{\tau} \cdot TWK \quad (5.17)$$

where $K_{AK,\perp}$ is introduced as mean stress factor, KE,σ is residual stress factor and $\sigma_{WK,\perp}$ is the component fatigue limit for completely reversed stress.

5.5.9.2 Mean stress sensitivity

The mean stress sensitivity effect M_σ or M_τ , in combination with the mean stress factor, describes what extent the mean stress affects the amplitude of the component fatigue limit.

In welded component mean stress sensitivity factor shall be taken account as a function of the residual stress factor.

5.5.9.3 Residual stress factor

The residual stress factor accounts for the influence of residual stresses on the fatigue strength of welded components. For welded components made of steels, the residual stress factor is given in Table 5.2 for normal and shear stress.

High residual stresses shall be assumed in the case of complex thick-walled components and in the presence of design related global residual stresses.

Moderate residual stresses shall be assumed in the case of narrow girders with longitudinal welds and with transverse welds of less than 300 mm in length, transversely stressed welds on components with a sheet thickness up to 10 mm.

Table 5.2: Residual stress and mean sensitivity for welded components

Residual Stresses	KE,σ	M_σ	KE,τ	M_τ
High	1,00	0	1,00	0
Moderate	1,26	0,15	1,15	0,09
Low	1,54	0,30	1,30	0,17

Low residual stresses shall be assumed in the case of welding with subsequent relief heat treatment.

5.5.10 Fatigue limit diagram

The mean stress factor described as K_{AK} in FKM guideline depends on the mean stress sensitivity, the field of mean stress and the type of overloading. The field of mean stress is determined by stress ratio R . Here, four fields of mean stress shall be distinguished.

Normal Stresses

For normal stresses, a Haigh Diagram with increasing amplitudes applies the tension field.

- Field I : $R > 1$, field of fluctuating compression stress. $R = +\infty$ is the zero compression stress.
- Field II: $-\infty \leq R \leq 0, R < 1$ is the field of alternating compressing stress. $R = -1$ is the completely reversed stress, $R > -1$ is the field for alternating tensile stress.
- Field III: $0 < R < 0,5$, the field of low fluctuating tensile stress. $R = 0$ is the zero tension stress.
- Field IV: $R \geq 0,5$, field of high fluctuating tensile stress.

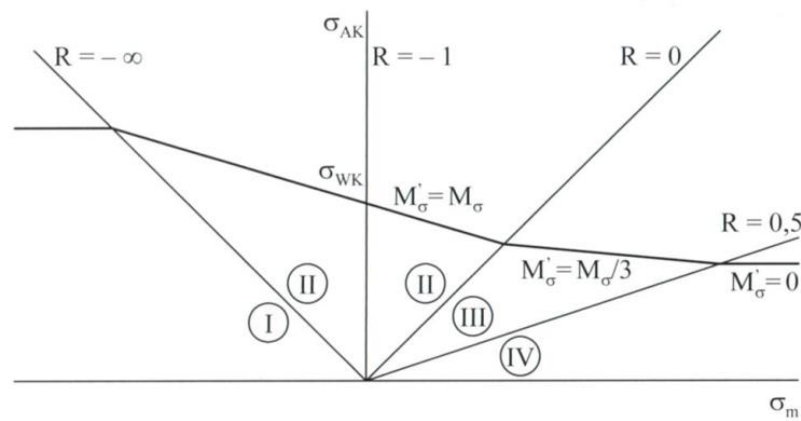


Figure 5.9: Diagram for fatigue limit for normal stresses

Critical amplitude σ_{AK} , mean stress σ_m , stress ratio R , component fatigue limit for completely reversed stress σ_{WK} .

Shear stresses

For shear stresses, a Haigh diagram which is symmetrical to $\tau_m = 0$ and exhibits no increasing amplitudes for $R < -1$ is valid.

- Field I: not existing for calculation
- Field II: $-1 \leq R \leq 0$ (lower boundary changed)
- Field III: $0 < R < 0,5$ (unchanged)
- Field IV: $R \geq 0,5$ (unchanged)

is utilized.

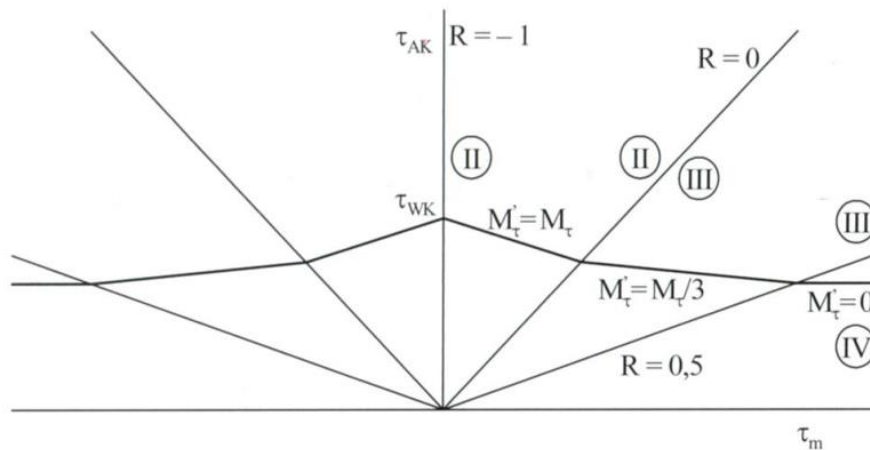


Figure 5.10: Diagram for fatigue limit for shear stresses

Critical amplitude τ_{AK} , mean stress τ_m , stress ratio R , component fatigue limit for fully reversed stress τ_{WK} .

5.5.11 Mean stress factor

The mean stress factor is calculated using mean stress sensitivity and residual stress parameters.

The mean stress factor K_{AK} depends on the type of overloading, F1 to F4. This shall be specified as a function of stress behaviour in the case of a possible load increase in service that is in the sense of safety in service. The types of overloading are as follows:
Depending on the type of overloading, the critical amplitude of the component fatigue limit changes.

Normal stresses

From Haigh diagrams,

Field I: $R > 1$, field of fluctuating compression stress. $R = +\infty$ or $-\infty$ is the zero compression stress.

$$K_{AK} = 1/(1 - M_\sigma) \quad (5.18)$$

Field II: $-\infty \leq R \leq 0$, $R < 1$ is the field of alternating compressing stress. $R = -1$ is the completely reversed stress, $R > -1$ is the field for alternating tensile stress.

$$K_{AK} = \frac{1}{1 + M_\sigma \cdot \sigma_m / \sigma_a} \quad (5.19)$$

$$\sigma_m / \sigma_a = (1 + R)/(1 - R) \quad (5.20)$$

Field III: $0 < R < 0.5$, the field of low fluctuating tension stress. $R = 0$ is the zero tension stress.

$$K_{AK} = \frac{3 + M_\sigma}{(1 + M_\sigma)(3 + M_\sigma \cdot \sigma_m / \sigma_a)} \quad (5.21)$$

Field IV: $R \geq 0.5$, field of high fluctuating tension stress.

$$K_{AK} = \frac{3 + M_\sigma}{3(1 + M_\sigma)^2} \quad (5.22)$$

Shear Stresses

The mean stress factor for shear stress shall be calculated based on the absolute value of the mean shear stress. This always results in $R \geq -1$.

Otherwise the mean stress factor for shear stress is calculated in the same manner for normal stress, provided M_σ is replaced with M_τ . Field I is omitted.

5.5.12 Individual or equivalent mean stress

In each case, R , $\sigma_{\min}$ and $\sigma_{\max}$ are determined based on the mean stress and amplitude. The mean stress may be taken either as the individual mean stress according to the type of stress or as an equivalent mean stress based on the individual mean stresses of all types of stress.

For welded components, normal stress perpendicular and parallel to the weld as well as shear stresses, the quantities $\sigma_{\min,\perp}, \dots, R_{\perp}$, $\sigma_{\min,\parallel}, \dots, R_{\parallel}$ or $\tau_{\min}, \dots, R_{\tau}$ apply.

Individual Mean Stress

As a rule, $\sigma_{\min}$, $\sigma_{\max}$ and R are determined based on the individual mean stress σ_m .

In that case, the following applies:

$$\sigma_{\min} = \sigma_m - \sigma_a \quad (5.23)$$

$$\sigma_{\max} = \sigma_m + \sigma_a \quad (5.24)$$

$$R = \frac{\sigma_{\min}}{\sigma_{\max}} \quad (5.25)$$

Equivalent Mean Stress

In the case of bending and torsion, which is typical of numerous applications in machine design, and in similar cases of normal and shear stresses, the equivalent variables $\sigma_{\min,v}$, $\sigma_{\max,v}$ and R_v shall be used with an equivalent mean stress $\sigma_{m,v}$ derived from individual mean stress of all types. In that case following applies:

$$\sigma_{\min,v} = \sigma_m - \sigma_a \quad (5.25)$$

$$\sigma_{\max,v} = \sigma_m + \sigma_a \quad (5.26)$$

$$R_v = \sigma_{\min,v} / \sigma_{\max,v} \quad (5.27)$$

where σ_a stands for individual stress amplitude, R_v is the equivalent stress ratio, $\sigma_{\min,v}$ and $\sigma_{\max,v}$ are the equivalent minimum and maximum stresses, respectively.

The equivalent mean stress for normal stress is:

$$\sigma_{m,v} = q \cdot \sigma_{m,v,NH} + (1 - q) \cdot \sigma_{m,v,GH} \quad (5.28)$$

$$q = \frac{\sqrt{3} - (1 / f_{w,\tau})}{\sqrt{3} - 1} \quad (5.29)$$

$$\sigma_{m,v,NH} = \frac{1}{2} \left(|\sigma_m| + \sqrt{\sigma_m^2 + 4 \cdot \tau_m^2} \right) \quad (5.30)$$

$$\sigma_{m,v,GH} = \left(\sqrt{\sigma_m^2 + 3 \cdot \tau_m^2} \right) \quad (5.31)$$

where q is the ductility factor, σ_m, τ_m are the individual mean stresses, $\sigma_{m,v,NH}$ and $\sigma_{m,v,GH}$ are the equivalent mean stresses according to normal stress hypothesis and Von Mises hypothesis, respectively.

The following applies to shear stress:

$$\tau_{m,v} = f_{w,\tau} \cdot \sigma_{m,v} \quad (5.32)$$

where $f_{w,\tau}$ introduced as the shear strength factor.

5.5.13 Safety factors

Basically the safety factors are valid on the condition that the characteristic strength values exhibit average probability of survival of %97.5.

Provided that reliable load designs are used , the following load safety factor applies:

$$J_s = 1,0 \quad (5.33)$$

Material safety factor for the fatigue strength depends on the possibilities of inspection and the consequences of failure.

Total safety factor for the assessment of the fatigue strength, J_D is;

$$J_D = J_s \cdot \frac{J_F}{K_{T,D}} \quad (5.34)$$

where J_s is the load safety factor, J_F is the material safety factor and $K_{T,D}$ is the temperature factor that affects the other safety factors.

5.5.14 Component variable amplitude fatigue strength

In this section, the amplitude of the component variable amplitude fatigue strength shall be computed as a function of stress spectrum and the component constant S-N curve.

The variable amplitude fatigue strength factor K_{BK} to be calculated depends on the stress spectrum, that is on the required total number of cycles (equals the required component fatigue life) and on the shape of the spectrum, as well as on the component constant amplitude S-N curve.

The critical nominal values of the amplitude of the component variable amplitude fatigue strength (the highest critical amplitude in stress spectrum) for normal stresses perpendicular and parallel to the weld as well as for shear stress are:

$$\sigma_{BK, \perp} = KBK, \perp \cdot \sigma_{AK, \perp} \quad (5.35)$$

$$\sigma_{BK, II} = KBK, II \cdot \sigma_{AK, II} \quad (5.36)$$

$$\tau_{BK} = KBK, \tau \cdot \tau_{AK} \quad (5.37)$$

where $KBK, \perp$ stands for variable amplitude fatigue strength factor and $\sigma_{AK, \perp}$ for component fatigue limit.

The critical amplitude of the variable amplitude fatigue strength σ_{BK} is restricted by the static strength σ_{SK} as well as by the component yield strength.

Thus the maximum value of the amplitude of the component variable amplitude fatigue strength, determined separately for each stress component at the reference point.

$$\sigma_{BK, \max, \perp} = 0,75 \cdot R_p \cdot n_p \quad (5.38)$$

$$\sigma_{BK, \max, II} = 0,75 \cdot R_p \cdot n_{pl} \cdot \alpha_{wl} \quad (5.39)$$

$$\tau_{BK, \max} = 0,75 \cdot R_p \cdot n_{pl} \cdot \alpha_{wl} \quad (5.40)$$

where R_p is the yield strength of the related material, n_{pl} and α_{wl} are section and weld factors, respectively.

The following applies to each stress components;

$$\text{If } \sigma_{BK} \geq \sigma_{BK_{\max}} \text{ then} \quad (5.41)$$

$$\sigma_{BK} = \sigma_{BK_{\max}} \text{ and} \quad (5.42)$$

$$KBK = \sigma_{BK_{\max}} / \sigma_{AK} \quad (5.43)$$

where KBK is the variable amplitude fatigue strength factor, σ_{BK} is the component variable amplitude strength and finally σ_{AK} is the component fatigue limit.

5.5.15 Assessment procedure

The assessment for fatigue assessment shall be carried with using each individual stress component and combines stresses.

5.5.15.1 Degree of utilization

The assessment shall be carried out based on the cyclic degree of utilization. The cyclic degree of utilization is a quotient of;

- The existing stress amplitude $\sigma_{a,1}$ and
- The allowable amplitude of the component variable amplitude fatigue strength at the reference point.
- The required total safety factor

Existing and critical amplitudes as well as the cyclic degree of utilization are always positive.

If different stress components act in combination at the reference point, the superposition shall be performed by computing a degree of utilization for the combined stress.

Individual Stress Types

The cyclic degree of utilization of welded components with normal stresses perpendicular and parallel to the weld as well as shear stresses, the following applies:

$$a_{BK,\perp} = \frac{\sigma_{a,\perp,1}}{\sigma_{BK,\perp / JD}} \leq 1 \quad (5.44)$$

$$a_{BK,\parallel} = \frac{\sigma_{a,\parallel,1}}{\sigma_{BK,\parallel / JD}} \leq 1 \quad (5.45)$$

$$a_{BK,\tau} = \frac{\tau_{a,1}}{\tau_{BK,\tau / JD}} \leq 1 \quad (5.46)$$

Combined Types of Stress

The applied strength hypothesis for combined stress types consists of two proportions in accordance with the normal stress hypothesis and Von Mises criterion.

Both proportions are controlled by constant q of by $f_{w,\tau}$ depending the ductility of the material. For steels and wrought aluminium alloys $q=0$ applies, with the result that only Von Mises criterion is valid.

The following rule is applied to the welded components:

$$a_{BK,\sigma v} = \frac{1}{2} \left(|s_{a,\perp} + s_{a,II}| + \sqrt{(s_{a,\perp} - s_{a,II})^2 + 4 \cdot t_a^2} \right) \quad (5.47)$$

with

$$s_{a,\perp} = a_{BK,\perp}$$

$$s_{a,II} = a_{BK,II}$$

$$t_a = a_{BK,\tau}$$

where $a_{BK,\perp}$ is introduced as cyclic degree of utilization.

5.6 DVS1612 Design and Endurance Strength Assessment of Welded Steel Joints in Rail Vehicle Construction

5.6.1 General

DVS1612 Technical code includes notes for the configuration and requirements for the designing of welded structures as well as a compilation of welded structure details (structural shapes) which are made of steel, are essential for rail vehicle construction advices existing in DVS1612 have proven their worth regard safety and maintenance. The joints details are mainly derived similar to DIN EN 15085-3 considerations. [19]

DVS1612 applies to the configuration and designing with regard to the endurance strength of arc-welded joints between steels utilised in rail vehicle construction for plate thickness $\geq 2\text{mm}$.

5.6.2 Welding technology and quality assurance

The requirements in DIN EN 15085-3 must be satisfied when configuring the welded joints in rail vehicle construction [19].

According to welding requirements in DIN EN 15085-3, the weldability must be guaranteed for a welded structure in rail vehicle construction [19]. The following is applicable:

- The welding quality in the fabrication is guaranteed if the structure must be manufactured while paying attention to the component class and to the welding processes possible in the plant.
- The weldability for service of the structure is guaranteed if the structure withstands the stresses taking account of the material behaviour.
- If weldability is not guaranteed in all points, it is strongly recommended to increase the weld inspection class, with a constant weld performance class, in order to assure the fabrication.

In addition to the weldability of the structure, it must be ensured that the structure is appropriate for inspection, required non-destructive tests and inspections, and appropriate for maintenance.

5.6.3 Design and fatigue assessment of welded joints

5.6.3.1 Basic principles of the design

With regard to the design-related formation of welded, following details has to be taken into consideration and executed carefully. These are listed below:

- in a way to appropriate for stress
- in a way to appropriate for fabrication
- -n a way appropriate for inspection
- with low distortion and warpage.

Designing in a way for stresses means that keeping any change in stiffness low and require any cross-sectional interfaces depending on the type of stresses, difference between tensile/ compressive, bending and torsional stresses.

A suitable design for fabrication is that provision is made for sufficient accessibility for the welding operation. Designing appropriate for the materials means that attention is paid to any possible material changes such as segregation.

5.6.3.2 Principle of the fatigue assessment

For the design of welded points in rail vehicle construction, it is necessary to comply with various standards with different requirements. The strength of car bodies is governed by EN 12663 [20]. These standards also include requirements on the designing welded joints but do not provide any concrete strength values. According to standards, it is necessary to design welded joints for the dynamic loads arising in operation. The material data should satisfy the following requirements:

- Survival probability of $p_{\bar{u}} = 97.5\%$
- Minimum number of 2×10^6 cycles for steel for a constant load amplitude.
- Classification of the structural shapes according to notch cases.

The above conditions are met if DVS1612 code is applied during dimensioning.

5.6.4 Assessment of the endurance strength

The endurance strength for welded joints is proven with nominal stress approach in this technical code. The increase in stresses caused by the structural and weld shapes is taken into account with endurance strength values depending on notch cases. For DVS1612, the assigned endurance strength values are represented on the MKJ diagrams.

The decisive normal stress result from the sectional sizes, in relation to the joint cross-section which exists at the position subjected to the highest stress or at the possible incipient crack location. If the stresses are determined with numerical calculation procedures such as finite element methods or means of measurement with strain gages, the decisive nominal stress are generally situated at a certain distance away from the weld which, in case of FE calculations, also depends on the undertaken modelling and on the mesh generation. In the case of welds subjected to transverse stresses, a guide value for the distance away from the weld toe is 1-1.5 times of plate thickness. In the case of the assessment of the endurance strength, attention must be paid to normal stresses longitudinal and transverse to the weld direction ($\sigma_{\parallel}$ and $\sigma_{\perp}$) as well as the shear stresses longitudinal to the weld direction(τ).

$\sigma_{\parallel}$, $\sigma_{\perp}$ and τ are illustrated below in Figure 5.11.

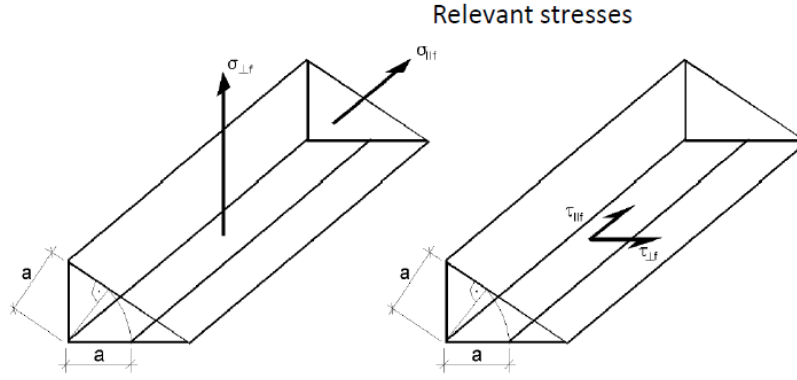


Figure 5.11: Stresses in weld region

The endurance strength must be carried out individually for each stress component:

$$\frac{\sigma_{||}}{\sigma_{||,zul}} \leq 1, \quad \frac{\sigma_{\perp}}{\sigma_{\perp,zul}} \leq 1, \quad \frac{\tau}{\tau_{zul}} \leq 1 \quad (5.48)$$

where $\sigma_{||,zul}$, $\sigma_{\perp,zul}$ and τ_{zul} are the permissible endurance strength values (according to MKJ diagrams) and apply to the plate thickness between 2 mm to 10 mm.

For the plate thickness above 10mm, strength values on the MKJ diagrams must be decreased according to following equation:

$$\sigma_{||,zul,t} = \sigma_{||,zul,MKJ} (10\text{mm}/t)^{0.1} \quad (5.49)$$

This applies analogously to $\sigma_{\perp,zul}$ and τ_{zul} .

In the case of several components, endurance strength must also be proven for multiaxial stress state via:

$$\left(\frac{\sigma_{||}}{\sigma_{||,zul}} \right)^2 + \left(\frac{\sigma_{\perp}}{\sigma_{\perp,zul}} \right)^2 - \frac{\sigma_{||}}{|\sigma_{||,zul}|} \cdot \frac{\sigma_{\perp}}{|\sigma_{\perp,zul}|} + \left(\frac{\tau}{\tau_{zul}} \right)^2 \leq 1.1 \quad (5.50)$$

The assessment of the endurance strength may also be provided according to another recognized procedure with the endurance strength values included in this technical code. In this respect, attention must then be paid to their dependence on the wall thickness as well.

If the highest stress arises at the weld toe, the nominal stress in plate cross-section is decisive. In the case of incompletely connected joints or in the case of weld joints with partial penetration, the assessment must be provided with the nominal stresses in

relation to the weld cross-section and for the weld toe. In this respect, attention must be paid to the computational throat thickness.

5.6.4.1 Permissible values for the assessment of the endurance strength

The permissible endurance strength values are represented as maximum or upper stresses on the MKJ diagrams for welded joints and for the base material made of S355, S235, S275 steels depending on the stress ratio and the notch case line.

The permissible endurance strength values on the MKJ diagrams include a safety factor 1.5 (for the mean values) and thus approximately correspond to the survival probability of %99.5.

Permissible endurance strength values must be required separately for:

- Normal stresses in the ranges of pulsating and alternating tensile stresses with $1 > R_\sigma \geq -1$

-Normal stresses in the ranges of pulsating and alternating compressive stresses with $1 > \kappa = \sigma_u / \sigma_o > -1$

- Shear stresses in the ranges of pulsating and alternating stresses with $1 > R_\tau \geq -1$

Normal stresses in the Ranges of Pulsating and Alternating Tensile Stresses (Mean stress ≥ 0)

The stress ratio R_σ results from the ratio of the minimum stress to maximum stress. In this respect, the minimum and maximum stresses must be required taking into account of the algebraic sign. The MKJ diagram for S355 is represented on Figure 5.12. It includes:

For the base material:

- Line A+ : blast cleaning and without any thermal influences
- Line A : without any thermal influences
- Line AB : with thermal influences.

For welded joints:

- Lines B to F2 whose assignment to the joint detail is specified in this technical code.

On these MKJ diagrams, the distance between the notch case lines corresponds to the factor 1.12. The notch case lines which are marked with “+” or “-“are located higher or lower by the factor of 1.04.

The permissible endurance strength value can be calculated according to equation 5.51 depending on stress ratio and the notch cases factor.

$$\sigma_{zul}(R_{\sigma}) = 150 \text{ MPa} \cdot (1.04)^{-x} \cdot \frac{2 \cdot (1 - 0.3 \cdot R_{\sigma})}{1.3 \cdot (1 - R_{\sigma})} \quad (5.51)$$

Normal stresses in the Ranges of Pulsating and Alternating Tensile Stresses (Mean stress < 0)

Instead of using R_{σ} , the stress ratio κ resulting from the ratio of existing lower stress σ_u to the existing upper stress σ_o must be applied in the compressive range.

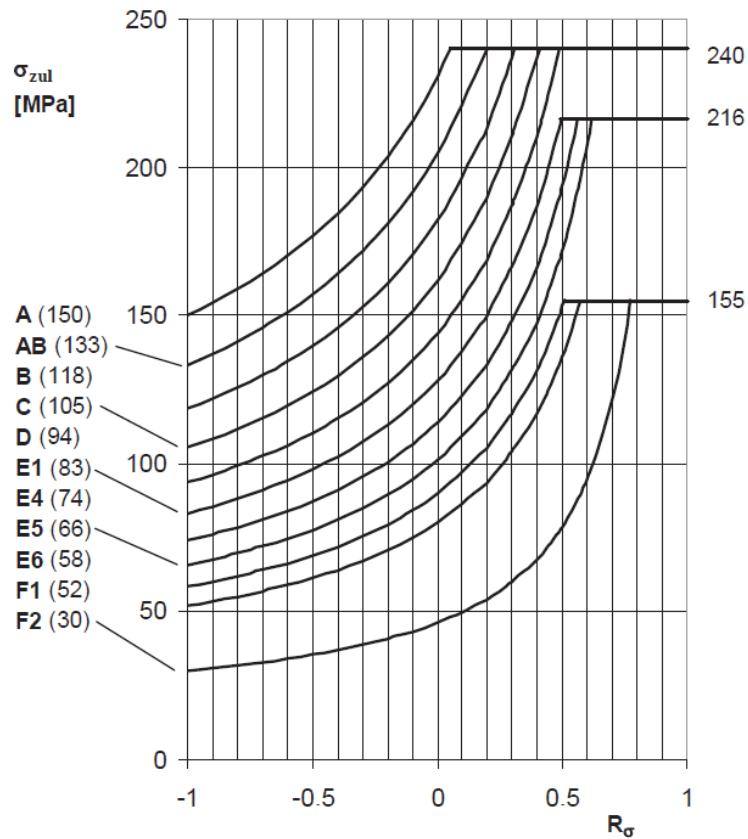


Figure 5.12: MKJ diagram for permissible normal tensile strength for S355 material.

The lower stress σ_u is the stress with the lower absolute value, without considering of the algebraic sign, and the upper stress σ_o is the highest stress absolute value, without considering the algebraic sign. In other words:

$$\kappa = 1/R_\sigma \quad (5.52)$$

$$\sigma_{o,zul}(\kappa) = 150 \text{ MPa} \cdot (1.04)^{-x} \cdot \frac{2}{1-\kappa} \quad (5.53)$$

In the compressive range, the permissible upper stress (stress with the highest absolute value) is limited by the same values on the tensile range.

Shear Stresses in the Ranges of Pulsating and alternating stresses with $1 > R_\tau \geq -1$

The permissible endurance strength values in the case of shear stress are represented as an MKJ diagram on Figure 5.13.

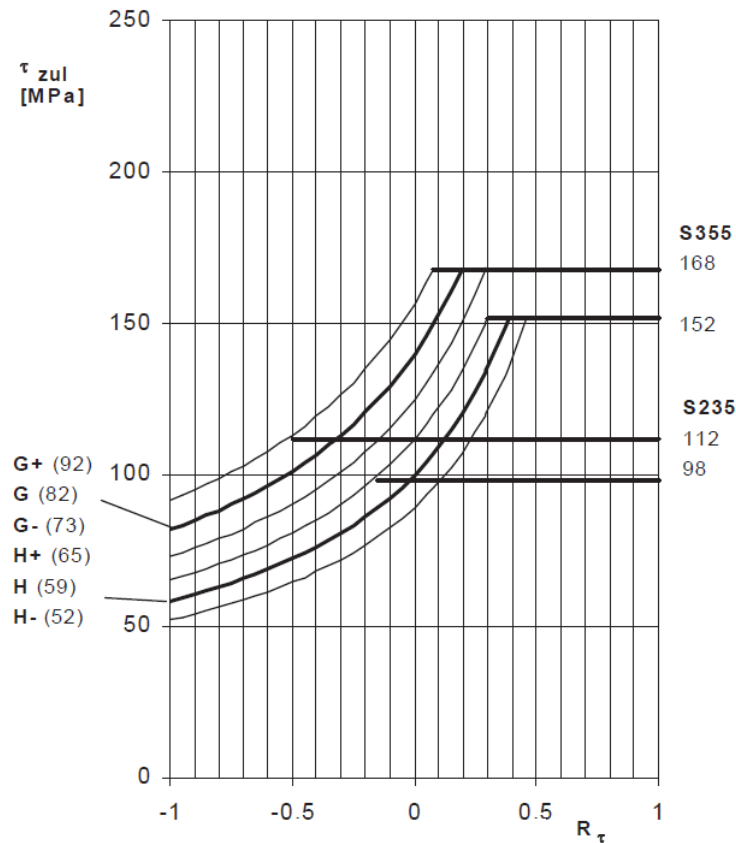


Figure 5.13: MKJ diagram for permissible shear stress for S355 material

In stress ratio R_τ , $\tau_{\min}$ is the existing shear stress with the lower absolute value (without consideration of algebraic value) and $\tau_{\max}$ is the existing maximum

absolute shear strength value (without consideration of algebraic value), the MKJ diagram exists only for $R_{\tau} \geq -1$

According to MKJ diagram:

$$\tau_{zul}(R_{\tau}) = \frac{2 \cdot (1 - 0.17 R_{\tau})}{1.17 \cdot (1 - R_{\tau})} \cdot \tau_{zul}, R = -1 \quad (5.54)$$

5.7 EC3 : Design of Steel Structures

5.7.1 Background of Eurocode programme

In 1975, the Commission of the European Community decided on an action programme in the field of construction. The objective of the programme was the elimination of technical obstacles to trade and the harmonization of technical specifications.

Within this action programme, the Commission took the initiative to establish a set of harmonized technical rules for the design of construction works which, in a first stage, would serve as an alternative to the national rules in force in the Member states and, ultimately, would replace them.

5.7.2 Eurocode 3: part 1-9: fatigue

Eurocode 3: Part 1-9: Fatigue (EN 1993-1-9) gives methods for the assessment of fatigue resistance of members, connections and joints subjected to fatigue loading.

These methods are derived from fatigue tests with large scale of specimens that include the effects of geometrical and structural imperfections from production and execution, effects of tolerances and residual stresses from welding.

The assessment methods in EuroCode 3 are applicable to all grades of structural steels, stainless steels and unprotected weathering steels except where noted otherwise in detail category tables.

5.7.3 Assessment methods

Fatigue assessment should be undertaken using either:

- Damage Tolerant Method
- Safe Life Method

The damage tolerant method should provide an acceptable reliability that a structure will perform satisfactorily for its design life, provided that a prescribed inspection and maintenance regime for detecting and correcting fatigue damage is implemented throughout the life of the structure.

The safe Life method should provide an acceptable level of reliability its design life without the need for regular in-service inspection for fatigue damage. The safe life method should be applied in cases where local formation of cracks in one component could rapidly lead to failure of the structural element or structure.

For the purpose of fatigue assessment, an acceptable reliability level may be achieved by using a partial factor of safety factor γ_{Mf} taking into account the consequences of failure.

The damage tolerant method with nominal stress evaluation is well-fitted to mechanical engineering problems. The damage tolerant method will be used according to Eurocode 3.

5.7.4 Calculations of stresses

Nominal stresses should be calculated at the site of potential fatigue initiation. Effects producing stress concentrations at details other than those included in detailed welding schemes of EuroCode 3 (EC3), should be taken into account as “modified nominal stress” multiplied by elastic stress concentration factors.

The relevant stress for details in the parent material are:

- nominal direct stress σ
- nominal shear stress τ

The relevant stresses in the welds are :

$$\sigma_{wf} = \sqrt{\sigma_{\perp f}^2 + \tau_{\perp f}^2} \quad (5.55)$$

$$\tau_{wf} = \tau_{\parallel f} \quad (5.56)$$

5.7.5 Calculation of stress ranges

The fatigue assessment should be carried out using modified nominal stress ranges where abrupt changes of section occur close to the initiation site.

The design value of stress range to be used for the fatigue assessment should be the stress ranges $\gamma_{Ff} \Delta\sigma_{E,2}$ corresponding to $N_C = 2 \times 10^6$ cycles.

5.7.5.1 Design value of nominal stress range

The design value of nominal stress ranges $\gamma_{Ff} \Delta\sigma_{E,2}$ and $\gamma_{Ff} \Delta\tau_{E,2}$ should be determined as follows:

$$\gamma_{Ff} \Delta\sigma_{E,2} = \lambda_1 \times \lambda_2 \times \lambda_3 \times \lambda_4 \dots \times \lambda_n \times \Delta\sigma(\gamma_{Ff} Q_k) \quad (5.57)$$

$$\gamma_{Ff} \Delta\tau_{E,2} = \lambda_1 \times \lambda_2 \times \lambda_3 \times \lambda_4 \dots \times \lambda_n \times \Delta\tau(\gamma_{Ff} Q_k) \quad (5.58)$$

where $\Delta\sigma(\gamma_{Ff} Q_k)$ and $\Delta\tau(\gamma_{Ff} Q_k)$ is the stress range caused by the fatigue loads.

λ_i is the damage equivalent factors depending on the spectra.

5.7.5.2 Design Value of Modified Nominal Stress Range

The design value of modified nominal stress ranges $\gamma_{Ff} \Delta\sigma_{E,2}$ and $\gamma_{Ff} \Delta\tau_{E,2}$ should be determined as follows:

$$\gamma_{Ff} \Delta\sigma_{E,2} = k_f \lambda_1 \times \lambda_2 \times \lambda_3 \times \lambda_4 \dots \times \lambda_n \times \Delta\sigma(\gamma_{Ff} Q_k) \quad (5.59)$$

$$\gamma_{Ff} \Delta\tau_{E,2} = k_f \lambda_1 \times \lambda_2 \times \lambda_3 \times \lambda_4 \dots \times \lambda_n \times \Delta\tau(\gamma_{Ff} Q_k) \quad (5.60)$$

Where k_f is the elastic stress concentration factor to take account of the local stress magnification in relation to detail geometry not included in the reference $\Delta\sigma_R - N$ curve.

5.7.5.3 Design value of stress range for welded joints

Unless more accurate calculations are carried out the design value of modified nominal stress range $\gamma_{Ff} \Delta\sigma_{E,2}$ should be determined as follows:

$$\gamma_{Ff} \Delta\sigma_{E,2} = k_1 (\gamma_{Ff} \Delta\sigma_{E,2}^*) \quad (5.61)$$

where $\gamma_{Ff} \Delta\sigma_{E,2}^*$ is the design value of stress range calculated with a simplified truss model with pinned joints and k_1 is the magnification factor.

5.7.6 Fatigue strength

The fatigue strength for nominal stress is represented by a series of $\log(\Delta\sigma_R - \Delta\tau_C)$ curves with corresponding to typical detail categories. Each detail category is designated by a number which represents in MPa.

For constant amplitude nominal stress ranges the fatigue strength can be obtained as follows:

$$\Delta\sigma_R^m N_R = \Delta\sigma_C^m 2 \times 10^6 \text{ with } k=3 \text{ for } N \leq 5 \times 10^6 \quad (5.62)$$

$$\Delta\tau_R^m N_R = \Delta\tau_C^m 2 \times 10^6 \text{ with } k=5 \text{ for } N \leq 10^8 \quad (5.63)$$

$$\Delta\sigma_D = \left(\frac{2}{5}\right)^{1/3} \Delta\sigma_C = 0,737 \Delta\sigma_C \quad (5.64)$$

Equation 5.64 is the constant amplitude fatigue limit.

$$\Delta\tau_L = \left(\frac{2}{100}\right)^{1/5} \Delta\tau_C = 0,457 \Delta\tau_C \quad (5.65)$$

Equation 5.65 is the cut-off limit.

For nominal stress spectra with stress ranges above and below the constant amplitude fatigue limit $\Delta\sigma_D$ the fatigue strength should be based on the extended fatigue strength curves as follows:

$$\Delta\sigma_R^m N_R = \Delta\sigma_C^m 2 \times 10^6 \text{ with } m=3 \text{ for } N \leq 5 \times 10^6 \quad (5.66)$$

$$\Delta\sigma_R^m N_R = \Delta\sigma_D^m 5 \times 10^6 \text{ with } m=5 \text{ for } 5 \times 10^6 \leq N \leq 10^8 \quad (5.67)$$

$$\Delta\sigma_L = \left(\frac{5}{100}\right)^{1/5} \Delta\sigma_D = 0,549 \Delta\sigma_D \text{ is the cut-off limit.} \quad (5.68)$$

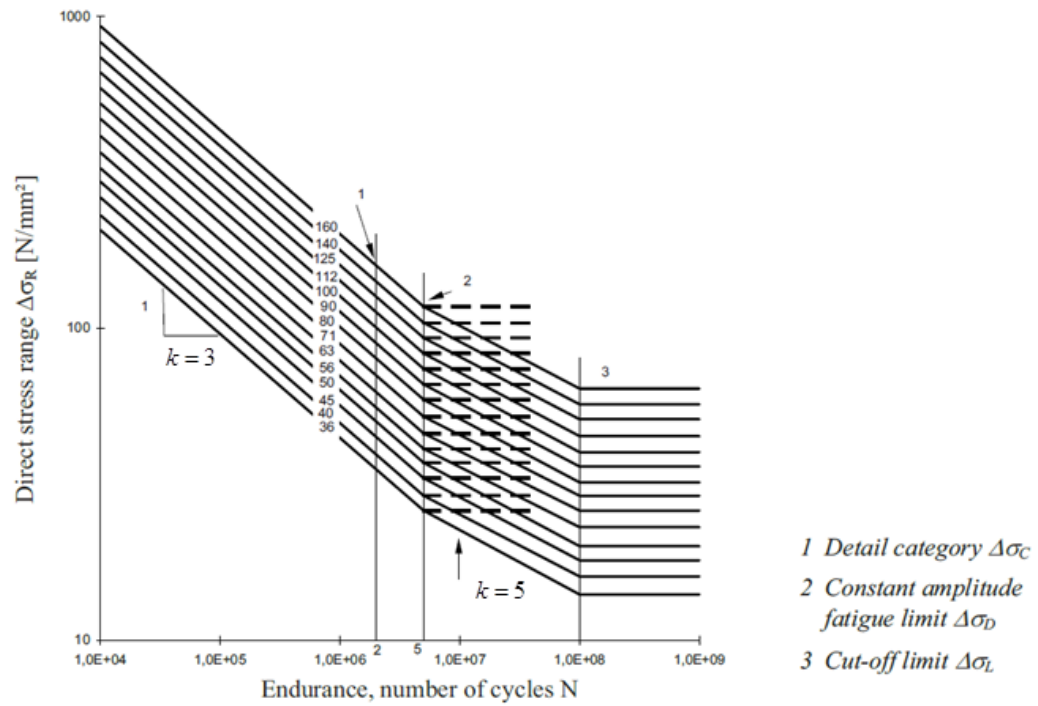


Figure 5.14: Fatigue strength of curves for direct (normal) stress ranges

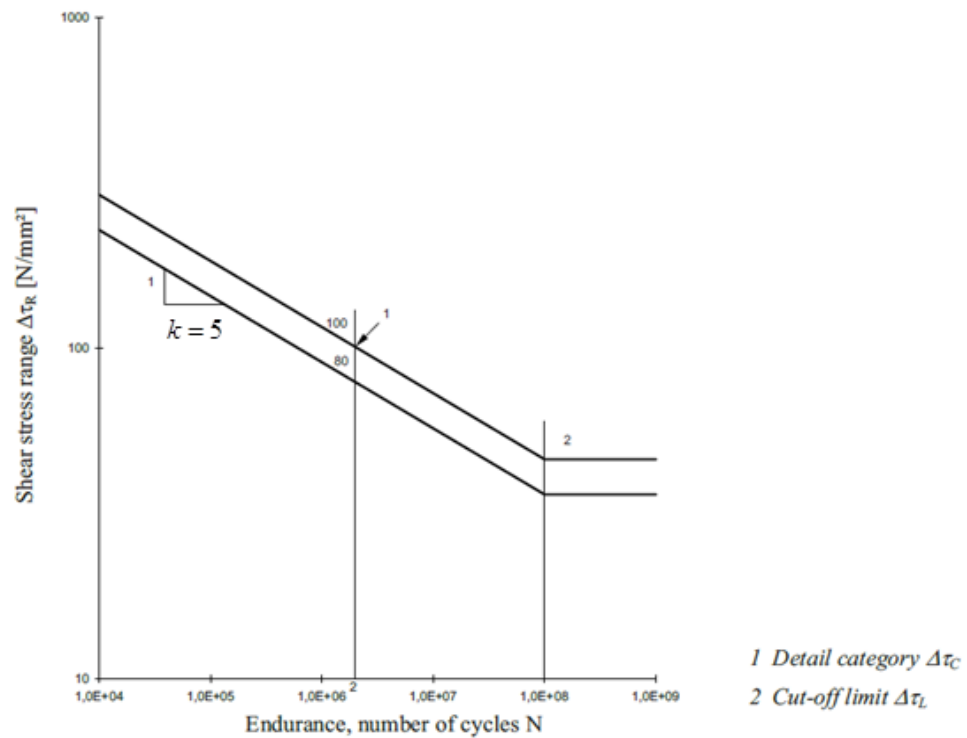


Figure 5.15: Fatigue strength of curves for shear stress ranges

According to EC3, in no-welded or stress relieved welded components , the mean stress influence on the fatigue strength may be taken into account by determining a

reduced effective stress range $\Delta\sigma_{E,2}$ in the fatigue assessment when part or all of the stress cycle in compressive.

$$\Delta\sigma = |\sigma_{\max}| + 0.6|\sigma_{\min}| \quad (5.69)$$

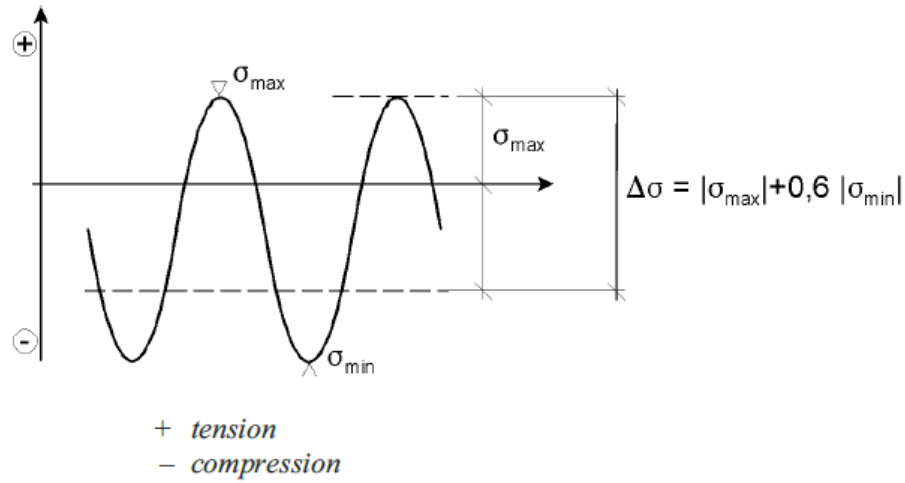


Figure 5.16: Modified stress range for non-welded or stress relieved details

5.7.7 Fatigue verification

It should be verified that under fatigue loading, in case of combined stress,

$$\left(\frac{\gamma_{Ff} \Delta\sigma_{E,2}}{\Delta\sigma_C / \gamma_{Mf}} \right)^3 + \left(\frac{\gamma_{Ff} \Delta\tau_{E,2}}{\Delta\tau_C / \gamma_{Mf}} \right)^5 \leq 1,0 \quad (5.70)$$

This verification also called as “calculated degree of utilization”.

6. FATIGUE ANALYSIS USING FINITE ELEMENT METHOD

Finite element models are used to analyse engineering components. The results may include stress, strains, temperatures, deflections and frequency responses. These results can be used for fatigue analysis. Also, in order to evaluate the fatigue analyse, basically stress or strain is needed depends on the approach.

Many fatigue analyses use the stress results from a linear elastic Finite Element Analysis (FEA). A linear analysis assumes a linear relationship between load and response. An elastic analysis assumes that the material stress-strain behaviour is linear elastic, with no allowance for yielding.

6.1 Terminology of Finite Element Analysis

The finite element method is a powerful numerical method which can be used to obtain approximate solutions to real world problems that involve complicated physics, geometry, and boundary conditions. Applications range from deformation and stress analysis of complex structures to field analysis of heat flux, fluid flow, magnetic flux [21,22].

In finite element method, a continuum domain Ω is firstly accepted as a collection of geometrically simple sub domains that are named as finite elements [22].

Each finite element Ω_e is viewed as an independent domain that refers to the geometric region over which the equations are solved. Second, over each subdomain the governing relationships are considered and then, are approximated by any of the traditional variational methods to obtain algebraic equations among the quantities of interest. Third, the relationships from all elements are assembled. Finally, taking into consideration the loading and constraints, the set of algebraic equation is solved. [21].

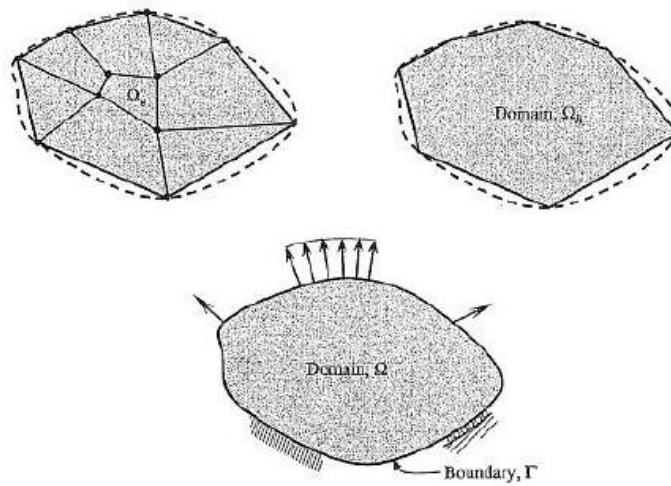


Figure 6.1: Collection of geometrically simple domains, assembled domain, applied boundary condition

For stress-deformation analysis of a body in equilibrium under external loading, the examination of the elements involves derivation of the load-deformation relationship. Primary concern is to find the distribution of displacement; therefore laws and principles which are used to derive load-deformation relationship are expressed in terms of displacement (u) [22].

Initial step in finite element analysis involves subdividing the body into a suitable number of small elements. The intersections of the sides of the elements are called nodes, while the interfaces between the elements named as nodal lines. The element type that should be used depends on the characteristics of the continuum and the chosen idealization [22].

Next, a pattern or shape for the distribution of the unknown quantity within an element that can be a displacement and/or stress for stress-deformation problems are chosen. The nodal points of the element provide strategic points for writing mathematical functions to describe the shape of the distribution of the unknown quantity over the domain of the element. A number of mathematical functions such as polynomials and trigonometric series can be used for this purpose, especially polynomials because of the ease and simplification they provide in the finite element formulation [22].

If u is the unknown displacement, an example polynomial interpolation function can be expressed as:

$$u = N_i u_i \quad i = 1, 2, \dots, m \quad (6.1)$$

where $u_1, u_2, u_3, \dots, u_m$ are the values of the unknowns at the nodal points and $N_1, N_2, \dots, N_m$ are the interpolation functions. The shape of pattern that is chosen, have to satisfy the conditions, laws, and principles of the problem at hand [22]

Energy methods are based on the idea of finding consistent states of bodies or structures associated with stationary values of a scalar quantity assumed by the loaded bodies. In engineering, usually this quantity is a measure of internal energy or work [22].

The other major alternative for formulating the finite element method is the method of weighted residuals (MWR). A number of schemes are employed under MWR, among which are collocation, subdomain, least squares, and Galerkin's methods. Galerkin's method is the most commonly used residual methods for finite element applications [22].

The MWR is based on minimization of the residual left after an approximate or trial solution substituted into the differential equations governing problem. As simple illustration, considering the following differential equation:

$$\frac{\partial^2 u^*}{\partial x^2} - \frac{\partial u^*}{\partial x} = f(x) \quad (6.2)$$

where u^* the unknown, x is the coordinate, t is the time, and $f(x)$ is the forcing function.

In mathematical notation, it can be written as

$$Lu^* = f \quad (6.3)$$

where L is the differential operator.

An approximate or trial function u for u^* is denoted as

$$u = \varphi_0 + \alpha_i \varphi_i \quad i = 1, 2, 3, \dots, n \quad (6.4)$$

Where $\varphi_1, \varphi_2, \varphi_3, \dots, \varphi_n$ are unknown functions chosen in such way to satisfy the homogeneous boundary conditions; φ_0 is chosen to satisfy the essential, geometric, or forced boundary conditions; and α_i are parameters or constants to be determined.

For convenience, the equation 6.4 is written as

$$u = \alpha_i \varphi_i \quad i = 1, 2, 3, \dots, n \quad (6.5)$$

If the approximate solution u is substituted into equation, a residual can be written as

$$\mathbf{R} \mathbf{x} = \mathbf{L} \mathbf{u} - \mathbf{f} \quad (6.6)$$

In MWR, the aim is to find an approximate solution u for u^* such that the residual is minimized. A number of schemes are available to minimize $R(x)$ Galerkin's method is one of them to derive finite element equations for a number of simple problems. Only brief information about the method will be given here [24].

Mathematically the idea of minimization can be expressed as

$$\int_D R x W_i x dx = 0 \quad i=1, 2, \dots, n \quad (6.7)$$

where D denotes the domain of a structure or body under consideration. W_i denotes weighting function which get various values with respect to different residual schemes. In the Galerkin's method, the weighting functions W_i are chosen from the basis functions used for constructing u^* [21,22].

Use of either energy methods or the method of weighted residuals the equations describing the behaviour of an element can be expressed as

$$\mathbf{k} \mathbf{q} = \mathbf{Q} \quad (6.8)$$

where $[k]$ is an element property matrix. $[q]$ is a vector of unknowns at the element nodes, and $[Q]$ is a vector of element nodal forcing parameters [22].

Once the element equations are established for a basic element, same equations are generated recursively for other elements. Equations for elements are added together to find global system of equations. This assembling process is based on the law of compatibility or continuity which requires that the body remain continuous. According to the law, the neighbouring points should remain in the neighbourhood of each other after the load is applied which means more simply that the displacements of two adjacent or consecutive points must have identical values. Depending on the type of the problem, the continuity conditions may be of various [24].

The assembled equations can be expressed in matrix notation as

$$\mathbf{K} \mathbf{r} = \mathbf{R} \quad (6.9)$$

Where $[\mathbf{K}]$ is an assemblage property matrix, $[\mathbf{r}]$ is an assemblage vector of nodal unknowns and $[\mathbf{R}]$ is an assemblage vector of nodal forcing parameters. For stress-deformation problems, these quantities are the assemblage stiffness matrix, nodal displacement vector, and the nodal load vector, respectively [22].

6.2 Meshing

As revealed before, a continuum domain Ω is firstly accepted as a collection of geometrically simple subdomains. In meshing stage, bodies are subdivided into a suitable number of small elements that form a grid named mesh. Information about how implanted atlantoaxial complex was meshed, will be given here in details immediately afterwards an overview of various element types.

Family, degrees of freedom, number of nodes, formulation and integration are five aspects to characterize element behaviour [23].

One of the major distinctions between different element families is the geometry type that each family assumes. The most commonly used element families in a stress analysis can be seen in Figure 6.2.

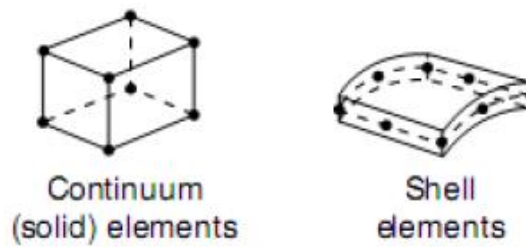


Figure 6.2: The most commonly used element families in a stress analysis

The unknown degrees of freedom are the fundamental variables calculated during the analysis. For a stress-displacement simulation the degrees of freedom are the translations and, for shell elements, also the rotations at each node [23].

Displacements or other degrees of freedom are calculated at the nodes of the element. At any other point in the element, the displacements are obtained by interpolating from the nodal displacements [23].

Elements that have nodes only at their corners, use linear interpolation in each direction and are often called linear elements or first order elements. Elements with midside nodes use quadratic interpolation and are often called as quadratic elements or second order elements. Modified triangular or tetrahedral elements with midside nodes use a modified second order interpolation Figure 6.3 [23].

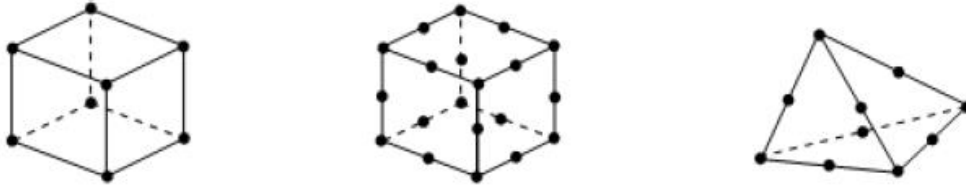


Figure 6.3: Linear hexahedral element (left), quadratic hexahedral element (middle), modified second order tetrahedral element (right)

An element's formulation refers to the mathematical theory used to define the element's behaviour. In the Lagrangian description of behaviour the element deforms with the material. In the alternative, Eulerian description elements are fixed in space as the material flows through them [23].

Numerical techniques integrate quantities over the volume of each element allowing complete generality in material behaviour. Using Gaussian quadrature for most elements, the material response at each integration point in each element is evaluated [23].

Second order elements provide higher accuracy than first order elements for smooth problems that do not involve impact, or severe element distortions. They capture stress concentrations more and are better for modelling geometric features [23].

For linear analysis the strain is defined as

$$\varepsilon = \text{sym} \left(\frac{\partial u}{\partial x} \right) \quad (6.10)$$

where u is the total displacement and X is the spatial position of the point under consideration in the original configuration.

Shell elements are used to model structures in which one dimension, the thickness, is significantly smaller than the other dimensions. Conventional shell elements use this

condition to discretize a body by defining the geometry at a reference surface. In this case the thickness is defined through the section property definition. Conventional shell elements have displacement and rotational degrees of freedom at nodes.

6.3 Numerical Fatigue Strength Assessment Points of Welds

Engineering structures may include a large number of components, therefore they include a large number of different types of welds. According to Fatigue strength assessment techniques to the welded areas, in this study local nominal approach, stress assessment point on the weld section has to be clarified. In this thesis a numerical nominal stress approach method has been used to calculate the fatigue strength of welded joints of the structures.

According to both numerical (Finite Element Method) and experimental techniques (strain-gage measurement) the stress assessment points of nominal stress approach can be determined in followed methods. As mentioned before as an area of this study numerical methods like FEM have been used and explanations about the stress assessment points of welded sections have been made on the basis of numerical approach.

All assessments are usually based on stress values which are either written separately for each element by the finite element code or which are calculated from displacement fields. Consecutively, assessment points are assigned to each node of an element, for which the numerical weld strength assessment analysis is performed. Assessment points are equivalent to possible points of crack initiation. For shell elements, the number of assessment points are two, one on the top of the element and one on the bottom of the element. Additional assessment points at the weld toe or root points are added if the shell element is connected to other elements by a weld. Figure 6.5 shows a T-joint, the corresponding shell element based model and the possible crack initiation points.

For a weld assessment, all element strips adjacent to the weld are examined separately. Figure 6.5 illustrates this by showing three different weld situations along with the potential points of crack initiation:

- Assessment :Web
- Assessment: Left Flange

- Assessment : Right Flange

According to the nominal stress assessment approach, in numerical fatigue analysis an offset value has to be implemented as extracting the stress values for welded joints. With an offset value, distance from the weld node to stress extractions points can be determined. Two effects can be treated this way:

- Elimination of mesh size effects at singular points. (Figure 6.5) a typical distance for avoiding singularities is 1.5 times the thickness.
- Correction of stresses due to high stress gradient factors and the real distance between shell node and weld toe.

When using the offset method for stress extraction, stress extraction points are also transformed into the local weld coordinate systems.

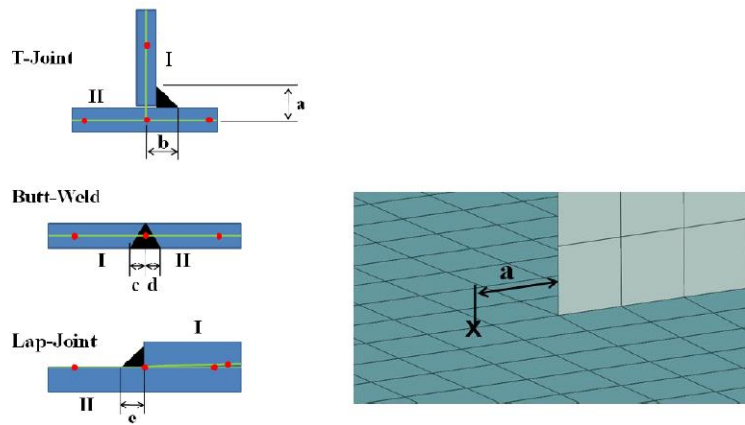


Figure 6.5: Distances of weld nodes to assessment points with FE representation

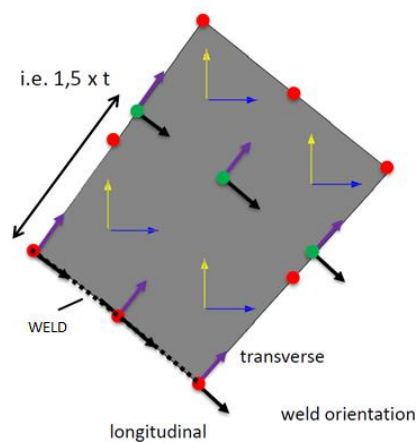


Figure 6.6: Distances of weld nodes to assessment points

6.4 An Illustrative Basic Assessment Problem According To FKM Guideline

Up to now, a detailed information about the types of welded joints and fatigue strength assessment of welded joints have been given according to three different approaches, FKM, DVS1612 and EC3.

In this section of thesis, numerical and analytical calculation of a given weldment will be compared pursuant to FKM guideline.

In Figure 6.7 T joint welded simple problem can be seen:

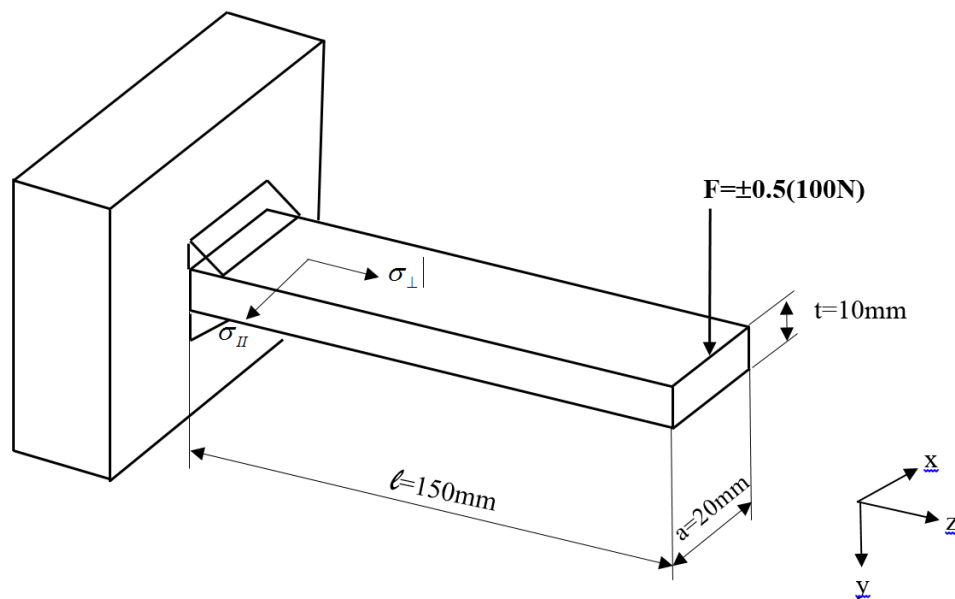


Figure 6.7: T-joint welded problem

where F is the shear force applying on tip of the beam.

Material properties of the beam can be taken as following in Table 6.1.

Table 6.1: Material properties of the given problem in Figure 6.7

E (Modulus of Elasticity)	210000 MPa
ν (Poisson's Ratio)	0.3
R_E (Yield Stress)	355 Mpa (St 52)

6.4.1 Analytical calculation

Sectional moment of inertia of the given problem is to be calculated as follow:

$$I_x = \frac{at^3}{12}$$

where I is the sectional moment of inertia of the system and I is found as:

$$I_x = \frac{20 \cdot 10^3}{12} \cong 1666.7 \text{ mm}^4$$

Within the scope of this thesis, the local nominal stress approach has been used for evaluating the fatigue strength of weldments.

The local nominal normal stress around the weld is to be calculated from 1.5*thickness far from the root of the weld.

$$l_{eff} = 150 - (1.5 \cdot 10) = 135 \text{ mm}$$

where l_{eff} is the effective length for calculating the nominal stress.

The local nominal stresses around the weldment in the directions is calculated as follow:

$$\sigma_{\perp} = \frac{F \cdot l_{eff}}{I_x} \cdot \frac{t}{2} = \frac{100 \text{ N} \cdot 135 \text{ mm}}{1666.7 \text{ mm}^4} \cdot 5 \text{ mm} = 40.50 \text{ MPa}$$

According to these values found above, amplitudes and the mean values of stress components on top of the welding plate is:

$$\sigma_{\perp, \max} = 0.5 \cdot (40.50 \text{ MPa}) = 20.25 \text{ MPa}$$

$$\sigma_{\perp, \min} = -0.5 \cdot (40.50 \text{ MPa}) = -20.25 \text{ MPa}$$

$$\sigma_{\perp, \text{mean}} = 0 \text{ MPa}$$

Design Parameters

FAT classes for individual stress components can be seen in Figure 6.8.

FAT class conversion factor for axial stress is:

$$f_{FAT,\sigma_{\perp}} = 0,5 \cdot \left(\frac{2 \cdot 10^6}{5 \cdot 10^6} \right)^{\frac{1}{3}} = 0,37$$

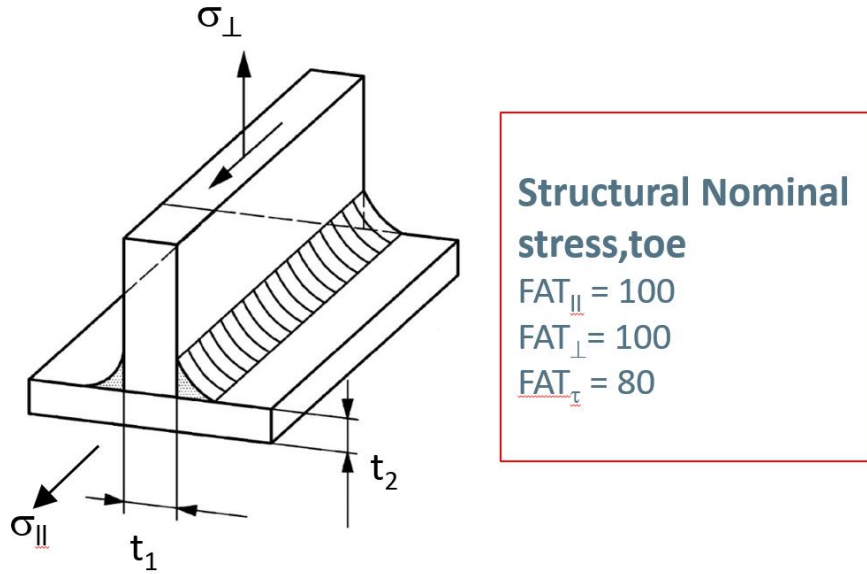


Figure 6.8: FAT classes of toe of double-sided T-joint weld

Component Strength

Component fatigue limit for completely reversed stress can be found as:

$$\sigma_{WK,\perp} = 100 \cdot 0,37 \cdot 1,1 = 37 \text{ MPa}$$

Component Fatigue Limit

To be able to calculate the component fatigue limit, residual stress factor and mean stress sensitivity have to be clarified.

Low residual stress factor is taken as 1,54,

$$K_{E,\sigma} = 1,54$$

And mean stress sensitivity is taken as 0,30,

$$M_{\sigma} = 0,30$$

In order to calculate the components fatigue limit, mean stress effect has to be included. In the problem, R, stress ratio) is constant and equals to -1. According to given parameters, the relevant area is field II in Haigh diagram.

The mean stress effect $K_{AK,\sigma}$ is taken as 1.

The components fatigue limit is calculated as:

$$\sigma_{AK,\perp} = 1.1,54.37 = 56,98MPa$$

Component Variable Amplitude Fatigue Strength

Component variable amplitude fatigue strength is evaluated with using model I constant amplitude S-N curve. For infinite life, variable amplitude strength factor is:

$$K_{BK,\sigma} = \left(\frac{5.10^6}{5.10^6} \right)^{\frac{1}{3}} = 1$$

Component variable amplitude fatigue strength equals to the component fatigue limit for infinite life.

Degree of Utilization

According to given and calculated parameters, the individual degree of utilization of the weld can be calculated as:

$$a_{BK,\perp} = \left(\frac{20,25}{56,98} \right) = 0,35 \leq 1$$

The degree of utilization of weld joint fatigue strength is 35 %. The assessment of the fatigue strength has been successful.

6.4.2 Numerical approach

In this section, the problem solved analytically in section 6.4 is solved numerically by commercial software packages.

Model, including meshing, material and sectional properties, is prepared in Hypermesh 13. The FE model is solved with ABAQUS 6.13 and finally the fatigue strength assessment according to FKM guideline is evaluated by using LIMIT software package.

FE Model of the Problem

Finite element representation of the problem is modelled in Hypermesh 13.0. Shell modelling technique is used due to thickness of the beam in Figure 6 is relatively small when compared to the other dimensions.

Shell modelling technique is used due to thickness of the beam in Figure 6 is relatively small when compared to the other dimensions.

FE representation of the problem can be seen in Figure 6.9.

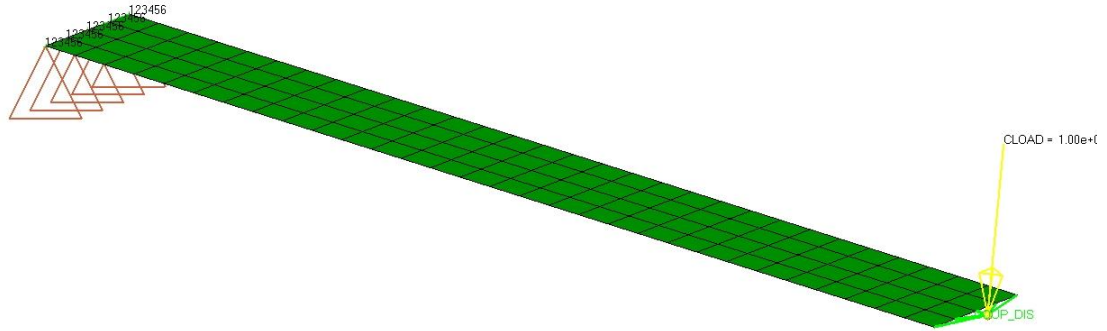


Figure 6.9: FE model of the given problem

The model includes 156 nodes and 121 elements. The left side of the beam is encastered. A load of 100N amplitude is applied at the right tip of the beam. The static analysis results of the beam can be seen in Figure 6.10.

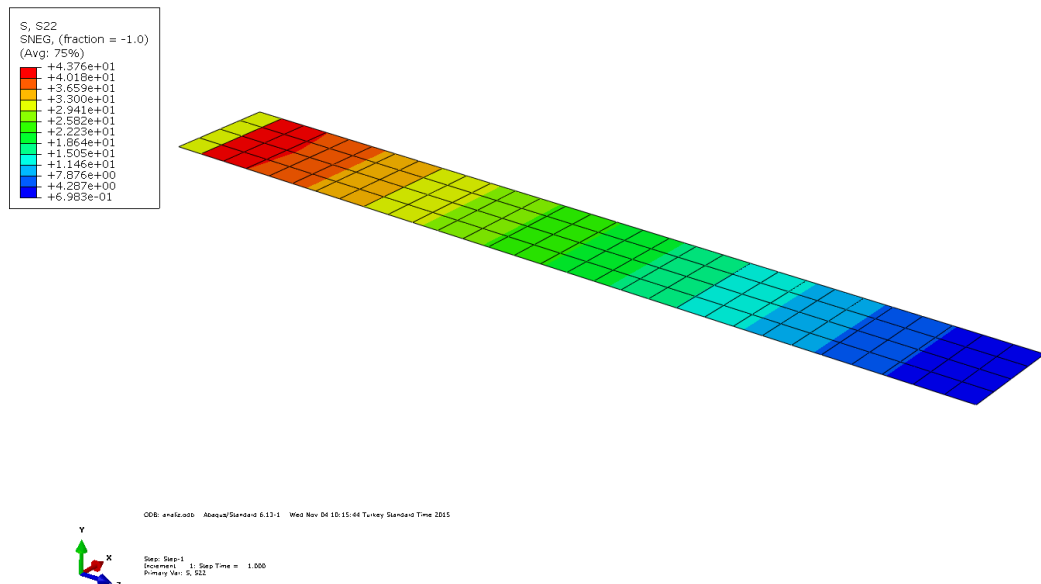


Figure 6.10: Numerical static analysis results (axial stress, $\sigma_{\perp}$)

Also you can see the 3D representation of the FE model, with sheet thickness representation, in Figure 6.11.

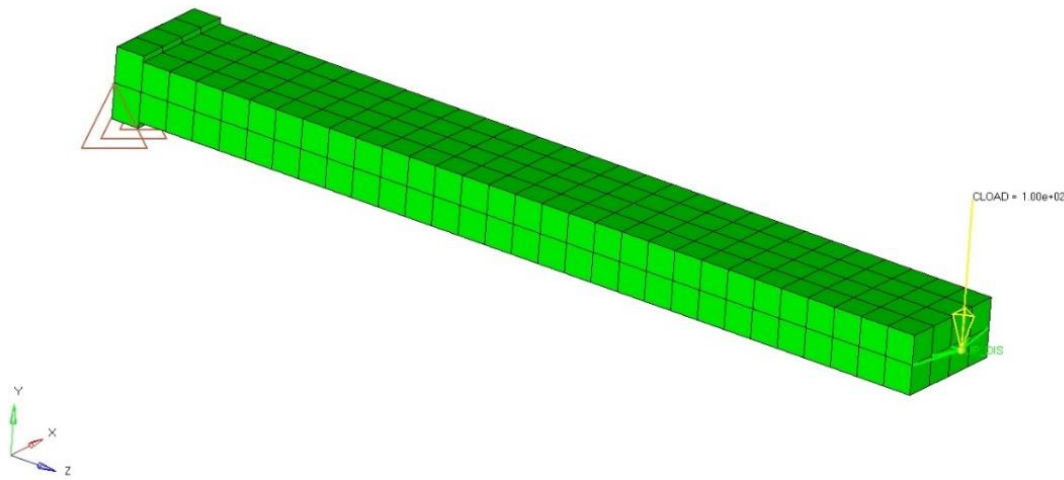


Figure 6.11: 3D representation of the FE model

In Figure 6.12 the detailed numerical weld configuration of the model is introduced in LIMIT software.

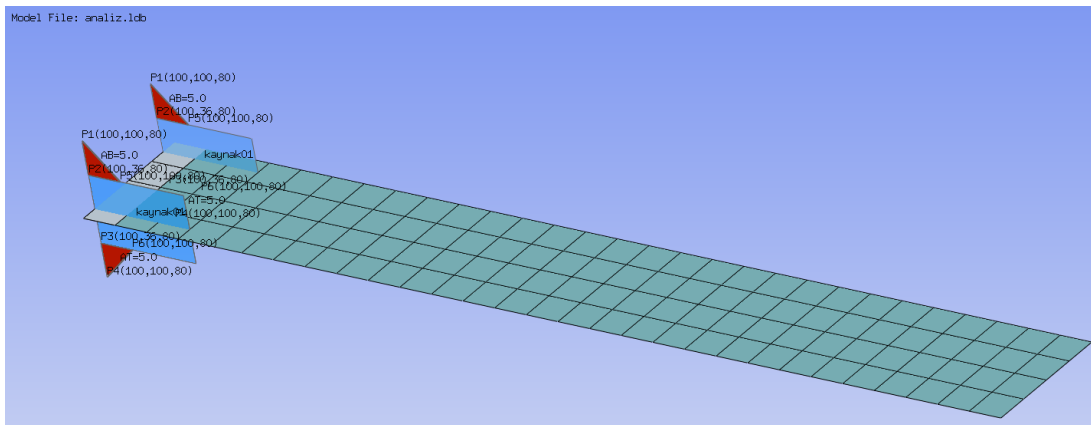


Figure 6.12: Detailed numerical weld configuration of the FE model

After introducing the weld configuration, FAT classes, loading scenarios and material properties has been assigned to the numerical model. According to the related loading conditions and FAT classes of the weldment the numerical fatigue strength assessment result can be seen in Figure 6.13.

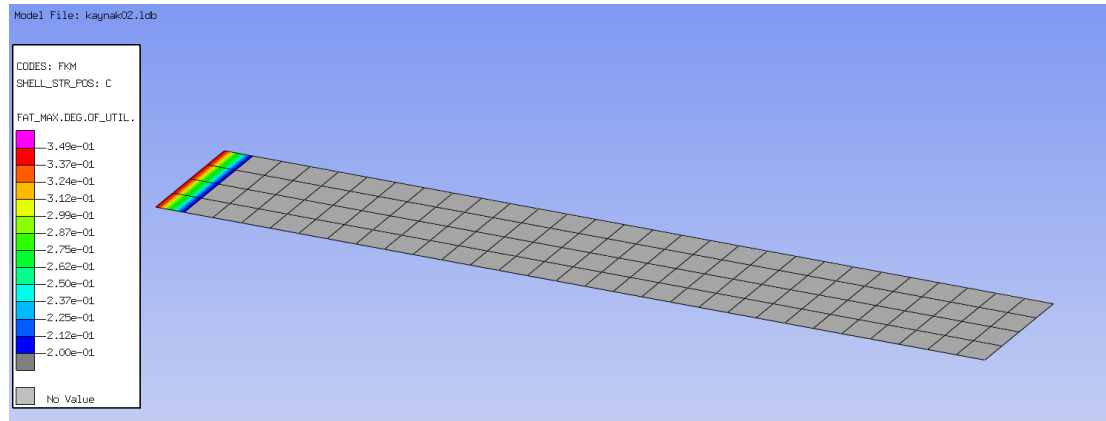


Figure 6.13: Numerical fatigue strength assessment results of the model

According to numerical analysis results maximum degree of utilization of the welded zone is found 34,9%. The assessment of the fatigue strength has been successful.

When compared to analytically found DOU and numerically found DOU of the weldments, the results are nearly same. The fatigue strength assessment of the welded zone is successful in both methods.

In analytic calculation section, the nominal stress approach was mentioned. The local nominal stress approach is also implemented to LIMIT software. Stress extraction points on the model can be seen in Figure 6.14.

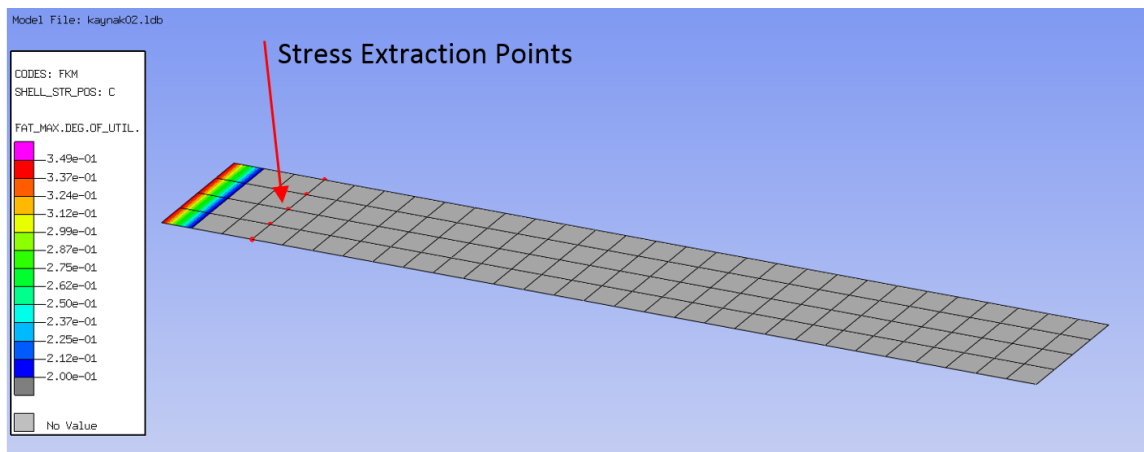


Figure 6.14: Stress extraction points of the numerical model

7. APPLICATION EXAMPLE: WELD FATIGUE ANALYSIS OF A “Rail Vehicle” USING DIFFERENT CODES

The aim of this thesis is to simulate the fatigue strength assessment of welded zones of the mechanical structures numerically. Up to now, a detailed information about the weld types, stress concepts of welded zones and international fatigue strength assessment codes have been introduced. In this section, a numerical application example of a rail vehicle is introduced within scope of static analysis and fatigue strength assessment of its welded zones. Fatigue strength assessment of welded zones of rail vehicle is evaluated with three aforementioned international codes, FKM, DVS1612, and EC3.

7.1 Introduction

A full scale freight wagon is expected to run under highly dynamic loads due to operating conditions, as well as structural and /or infrastructure-based excitations. Running dynamics of a train including rail/wheel interactions and train longitudinal dynamics exert impact-like loads on the chassis which may cause undesired and unexpected fatigue failure of wagons. Welded joints of the chassis are, thus, the most critical areas compared to main structural supporting girders and have to ensure the desired fatigue conditions foreseen, such as those stated in EN 12663 [20].

In this section of thesis, the welded joints of a full scale freight wagon, including mostly continuous and discontinuous seam welds have been modelled and evaluated against fatigue limits of base material and components according to three international codes, FKM, DVS1612, EC3.

The analytical fatigue strength evaluation studies have been conducted using LIMIT software in conjunction with ABAQUS and HYPERMESH software packages. The structural static model of the wagon have been modelled in HYPERMESH with detailed welding joints. The final model approximately includes 2000 seam welds with different structural properties. After the static requirements of the wagon are fulfilled according to EN 12663, the welds of the chassis have been analyzed in both global and

detailed manner according to international fatigue strength assessment codes with different cyclic dynamic loading scenarios in LIMIT software package.

7.2 Methods

The analysis ran have been designed as three fold. The first step is to model all the welded areas with different element properties and arranging element sets. And the next step is to analyze the FEM model with different loading scenarios with the right boundary condition. Numerically applying the right boundary condition to a full scale freight wagon is the key to a successful results compared to physical world. The last and the most important part in accordance with this thesis can be divided into two main categories. The first one is to determine the critical welds with global weld fatigue assessment. The second assessment is to analyze the critical welded areas with full details.

7.3 Geometry

The geometrical views of the design are provided in Figure 7.1. The view was taken from the finite element environment, and when compared to standard freight wagon models, it can be clearly seen the detailed modeling techniques of welded areas. The net weight of the chassis and the number of the numerical components, number of the finite elements and nodes are provided in Table 7.1.

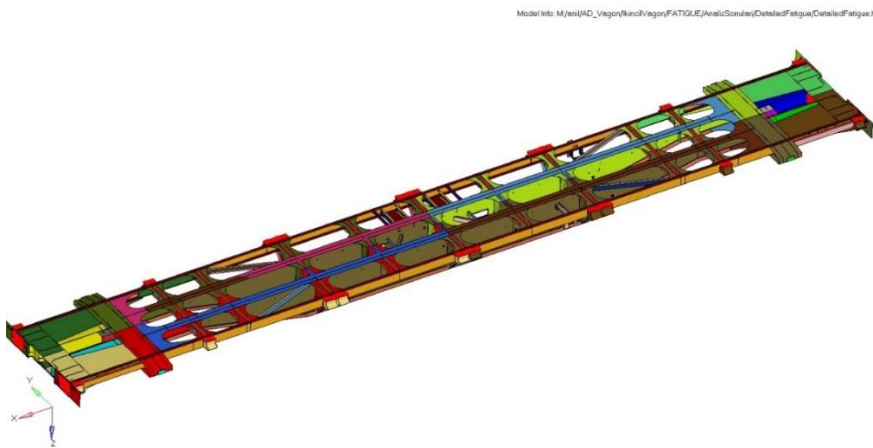


Figure 7.1: General view of the freight wagon model

During the evaluation studies, only the bare chassis has been taken into consideration as the main point of interest, and the geometry of the containers were not geometrically modelled but their mechanical based properties like weights were taken into account.

Table 7.1 Numerical data for FEM model of the freight wagon

Net weight(tonne)	# of components	# of elements	# of weld elements	# of nodes
9.070	2450	947,974	41,231	942,972

7.4 Static Analysis of The Wagon Chassis

The static loading conditions are derived from EN 12663 requirements for the freight wagon. The summary of the load cases applied are:

- i) Gravitational effect of the chassis mass
- ii) Gravitational effects of the chassis and the container masses
- iii) Axial loads at buffer/coupler level

The load ranges are considered as container loads applied at locks as point loads over the contact surfaces for all scenarios. Thus, 5 different freight loading conditions have been investigated. Load application points can be seen in Figure 7.2. The material parameters used are as of common steel type of ST-52 is introduced in Table 7.2.

Table 7.2: Material properties of the wagon chassis

E (modulus of elasticity)	ν (Poisson's ratio)	ρ (density)
210000 MPa	0.3	7850 kg/m^3

For keeping the tables and legends tidy, a shortcut name for each scenario was generated. The list of codes is provided in the following Table 7.3. All analyses were linear elastic.

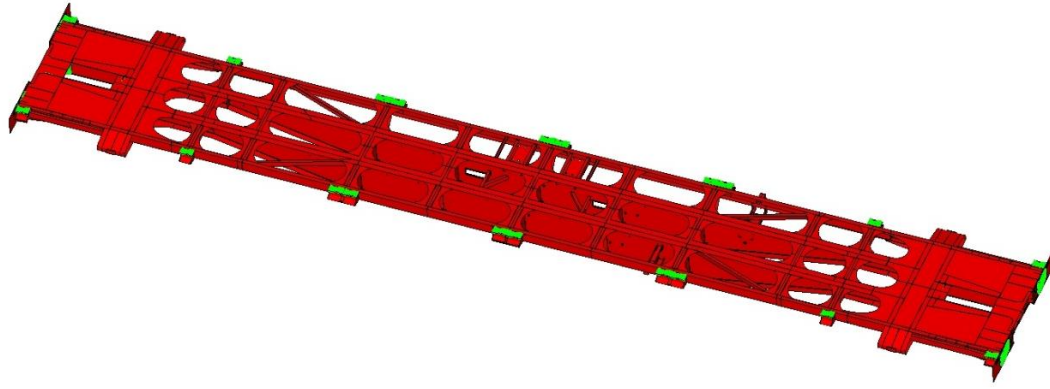


Figure 7.2: Example of loads applied to freight wagon chassis

Table 7.3: Loading scenarios applied on the static analysis

ACRONYM	PAYLOAD DEFINITION
C1	2x20'' Containers on Wagon @32.0 tonnes
C2	2x30'' Containers on Wagon @32.0 tonnes
C3	3x20'' Containers on Wagon @23.3 tonnes
C4	1x20'' Container at the Center of Wagon @32.0 tonnes
C5	1x40'' Container at the Center of Wagon @32.00 tonnes

The numerical model preparation and the meshing processes were completed in HYPMERSH 13 environment based on the CAD model. The view of the mesh is introduced in Figure 7.3.



Figure 7.3: Example mesh view of the wagon chassis

After the mesh and loading points has been prepared the numerical analysis solver process and the results post processing procedures were completed in ABAQUS 6.13 environment.

An example of static analysis result on FEM environment can be seen in Figure 7.4.

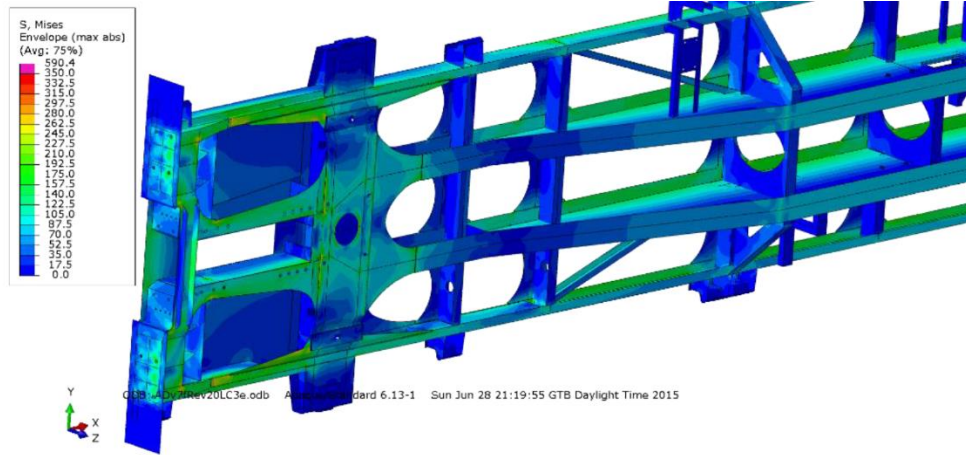


Figure 7.4: Example numerical static analysis result
Von Mises stress distribution for scenario C3 with symmetric buffer compr. 2000KN

7.5 Experimental Tests Applied

The experimental tests have been executed at Turkish Coach Industry Corporation (TÜVASAŞ) test bench by ITU team. Strain gauges and displacement transducers are applied to different critical locations on the freight wagon chassis.

The payload distributions and the all static loading scenarios have been executed in accordance with the EN 12663 standard.

The structural integrity of the freight wagon have been checked using FEM before the tests. Thus strain gages have been distributed on the chassis according to critical points observed in FEA. Depending on the stress state of the chassis according to FEA, either 3-element rosette type or single element type strain gauges were applied to different critical positions.

An example critical positions of the wagon executed according to Finite Element Results is seen in Figure 7.5.

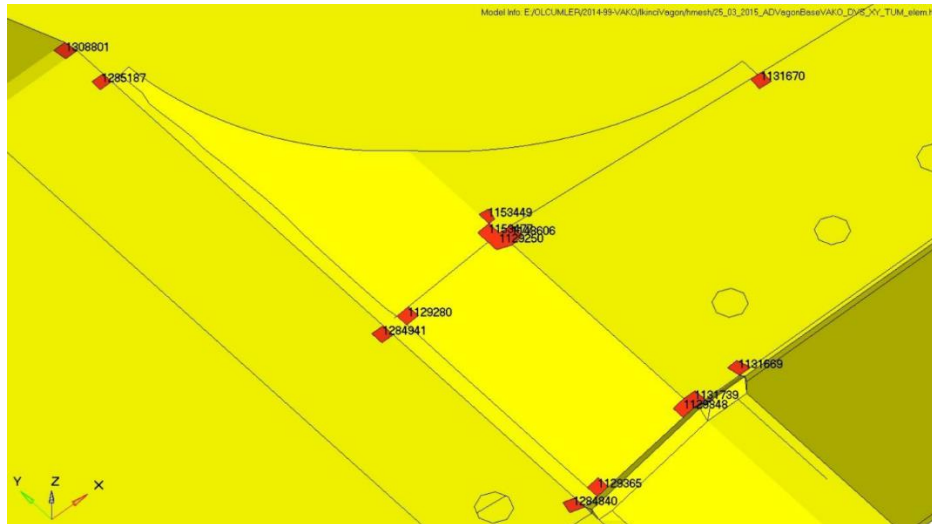


Figure 7.5: Example critical strain gage locations of the wagon

The comparison has been executed between the Finite Element Analysis results and the Experimental Analysis Results. An example of the experimental stress analyze is also introduced in Table 7.4.

According to FEA and experimental stress analysis, the results match up successfully. Thus fatigue strength assessment can be done with numerical stress analysis results.

7.6 Finite Element Analysis of the Chassis Aimed at Weld Fatigue Assessment

Fatigue loading conditions of freight wagon according to EN 12663 has been determined. Implementation the tare weight of the chassis and the container weights from physical world to numerical model has been performed. The finite element analysis has been performed according to two main inertial loads: first scenario is performed arising from the gravitational effects and the second one is of lateral (centrifugal) effects.

Each of the static analysis were conducted via ABAQUS 6.13 Finite Element Software on HP DL 585 G7 4x16 core/256GB RAM computer available at ITU Mechanical Engineering Faculty, Laboratory of Biomechanics' & Strength of Materials. The boundary conditions are applied as if the chassis is lying on a bogie.

Table 7.4: Example experimental stress analysis result
Von Mises stress stresses in related gages

	Event	1	2	3	4	5	6	7	9	10
Gage	Zone	Rotation of Chassis for 2g Weight Measurement R1	Rotation of Chassis for 2g Weight Measurement R2	Lifting of Chassis for Weight Measurement R1	Lifting of Chassis for Weight Measurement R2	Lifting of Chassis for Weight Measurement R3	3x20 Container Loading-Unloading R1	3x20 Container Loading-Unloading R2	3x20 Container Loading R3	3x20 Container Exceptional Loading R1
R01	1	0.0	0.0	0.0	0.0	0.0	13.5	13.5	13.5	4.6
	2	4.0	3.3	13.4	15.3	15.0	10.6	2.4	4.6	3.7
	3	0.4	0.1	2.2	0.9	2.1	14.2	14.7	4.6	3.7
R02	1	0.0	0.0	0.0	0.0	0.0	3.6	3.6	3.6	159.1
	2	45.7	46.4	41.6	44.8	44.5	158.4	162.4	159.0	203.2
	3	0.1	0.4	3.4	2.8	0.9	4.1	4.2	159.0	203.2
R03	1	0.0	0.0	0.0	0.0	0.1	3.7	3.7	3.7	25.1
	2	23.6	23.7	21.4	23.6	22.1	27.5	25.5	25.1	33.1
	3	0.1	0.1	3.1	1.6	0.8	4.1	4.6	25.1	33.1
R04	1	0.0	0.0	0.0	0.0	0.0	4.0	4.0	4.0	21.9
	2	11.1	11.2	7.2	7.7	8.0	22.3	21.3	21.9	28.0
	3	0.1	0.1	0.8	0.7	1.0	4.2	4.5	21.9	28.0
R05	1	0.0	0.0	0.0	0.0	0.0	16.6	16.6	16.6	109.3
	2	50.8	50.9	30.2	32.7	34.1	103.3	107.8	109.3	134.8
	3	0.1	0.0	3.4	2.4	0.4	18.6	17.8	109.3	134.8
R06	1	0.1	0.1	0.1	0.2	0.2	6.3	6.3	6.3	19.8
	2	2.4	2.4	6.0	5.5	4.5	24.2	20.6	19.8	21.5
	3	0.2	0.1	8.3	6.8	6.6	5.3	6.3	19.8	21.5
R07	1	0.0	0.0	0.0	0.0	0.0	9.7	9.7	9.7	67.5
	2	26.8	26.7	15.5	16.7	17.7	66.6	66.5	67.5	83.7
	3	0.0	0.1	1.4	1.1	0.2	10.3	10.3	67.5	83.7

An example FEA result view of the chassis under the gravitational effect can be seen in Figure 7.6.

For keeping the tables and figures tidy, from now on gravitational affects is given with “z” suffix and the lateral (centrifugal) effect is given with “y” suffix.

Aforementioned before, weld fatigue strength assessment of mechanically stressed components, has been done in two stages. First stage is to analyze all welds in a structure with low degree welding qualities, in order to capture all the critical welded sections according to loading conditions. And the second stage is to analyze critical welded sections that has been captured in the first stage, with full details and quality conditions same to that of the physical structure.

Also all the weld elements of the freight wagon chassis model can also be seen in Figure 7.8.

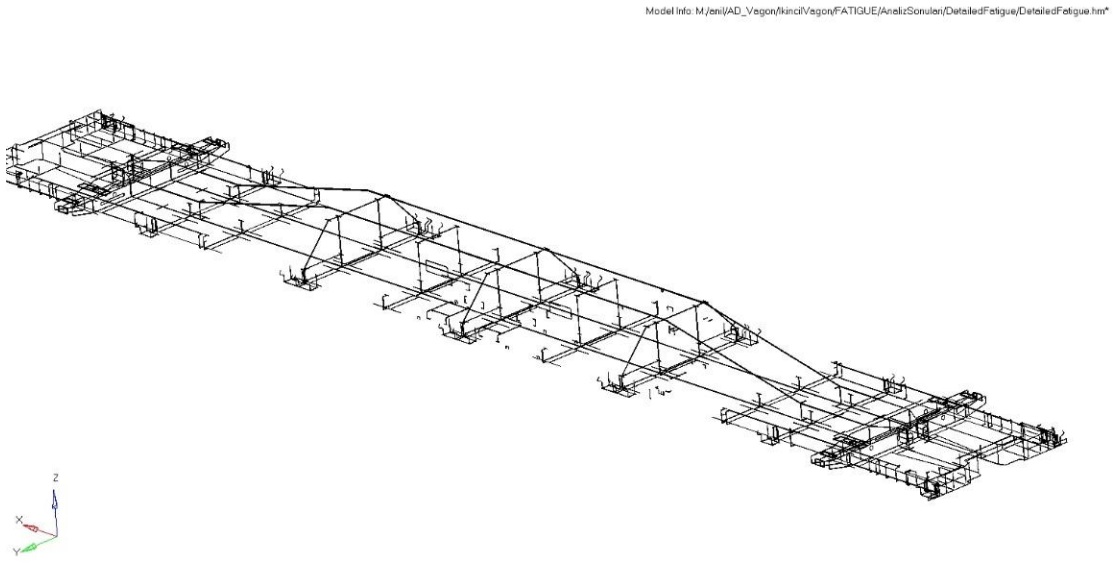


Figure 7.8: Weld elements of the freight wagon chassis

7.8 RESULTS

Fatigue strength assessment of the chassis has been executed using three aforementioned codes. First fatigue strength assessment according to FKM Guideline is introduced.

7.8.1 FKM Guideline results

Weld global fatigue analysis has been performed with FAT classes in Figure 7.9 according to FKM Guideline.

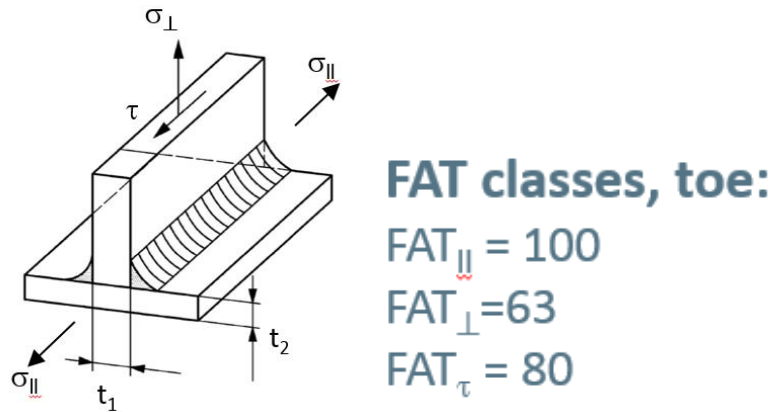


Figure 7.9: FAT classes for weld global fatigue analysis according to FKM Guideline

The numerical material utilization outcomes (a quantity which is inverse of the safety margin for this case) have been obtained at specific weld elements of the model. One of the worst locations belonging to C2z loading scenario is seen in Figure 7.10.

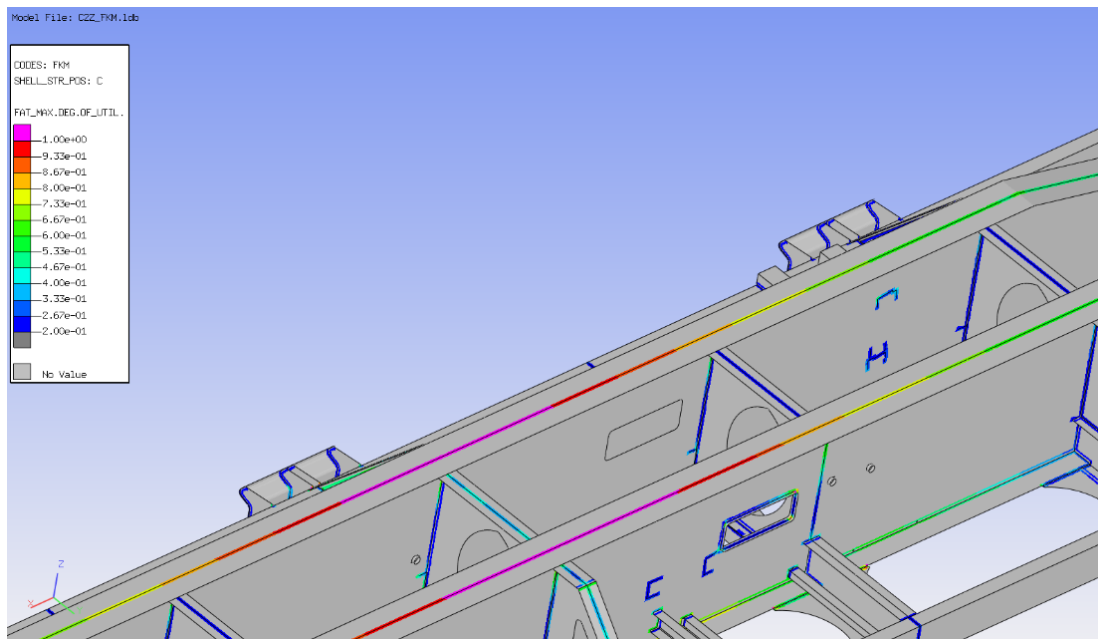


Figure 7.10: Weld global analysis result according to FKM Guideline: C2z loading scenario

An example view of results obtained from the global weld fatigue analyses with C3z loading scenario has been visually illustrated in Figure 7.11.

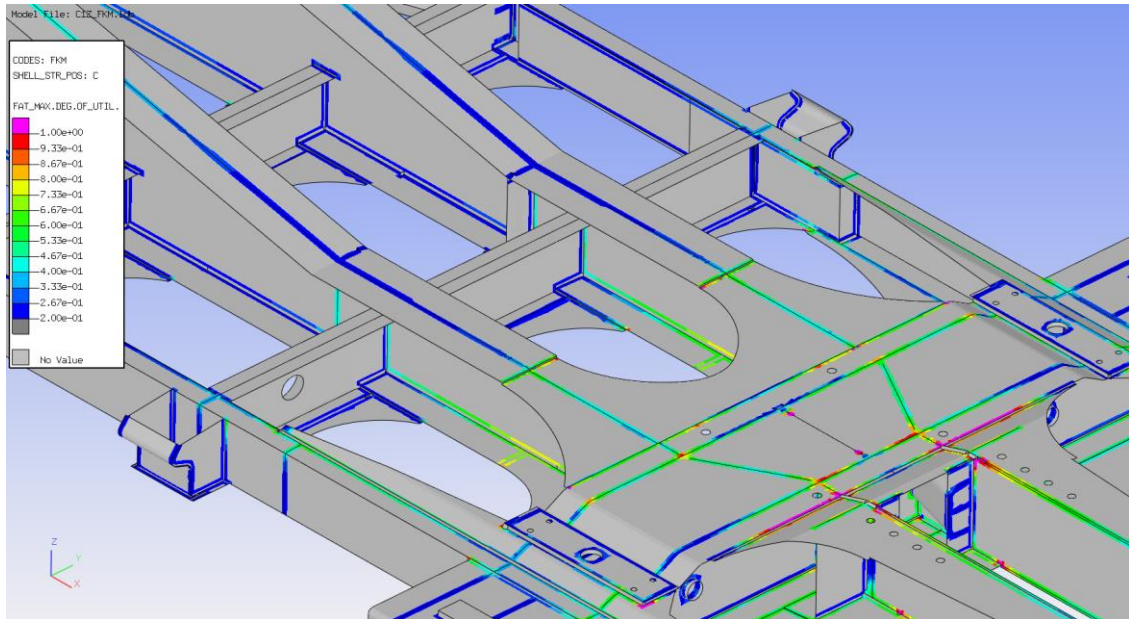


Figure 7.11 Weld global analysis result according to FKM Guideline: C1Z loading scenario

Ten weld global analysis are executed regarding gravitational and centrifugal effects. After weld global analysis according to FKM guideline has been completed, the critical welded areas are determined.

According to critical welds spotted in weld global analysis, the detailed configuration of these welds are modelled in LIMIT software with full detailed FAT classes and properties.

An example view of one of these critical welded area, with detailed properties can be visually seen in Figure 7.12.

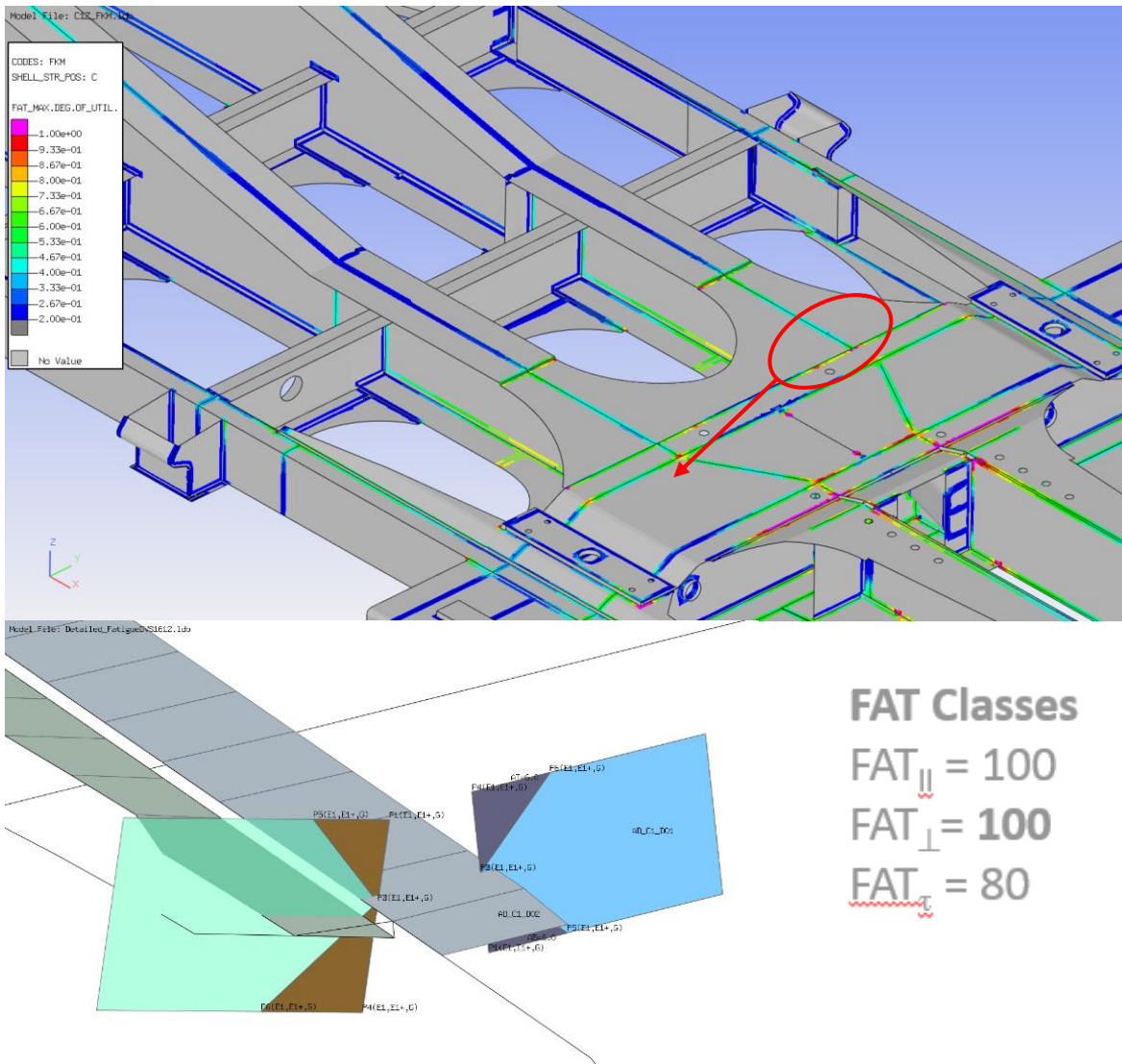


Figure 7.12: Detailed configuration of critical weld with fat classes according to FKM Guideline

Four critical welds has been spotted via weld global analysis. Here, one of the most critical ones' result is visualized. All weld global analysis and detailed critical weld analysis can be found in APPENDIX A according to aforementioned loading scenarios.

According to critical welds spotted in weld global analysis, the detailed configuration of these welds are modelled in LIMIT software with full detailed FAT classes and properties.

In Figure 7.13 an example result of critical weld analysis is illustrated.

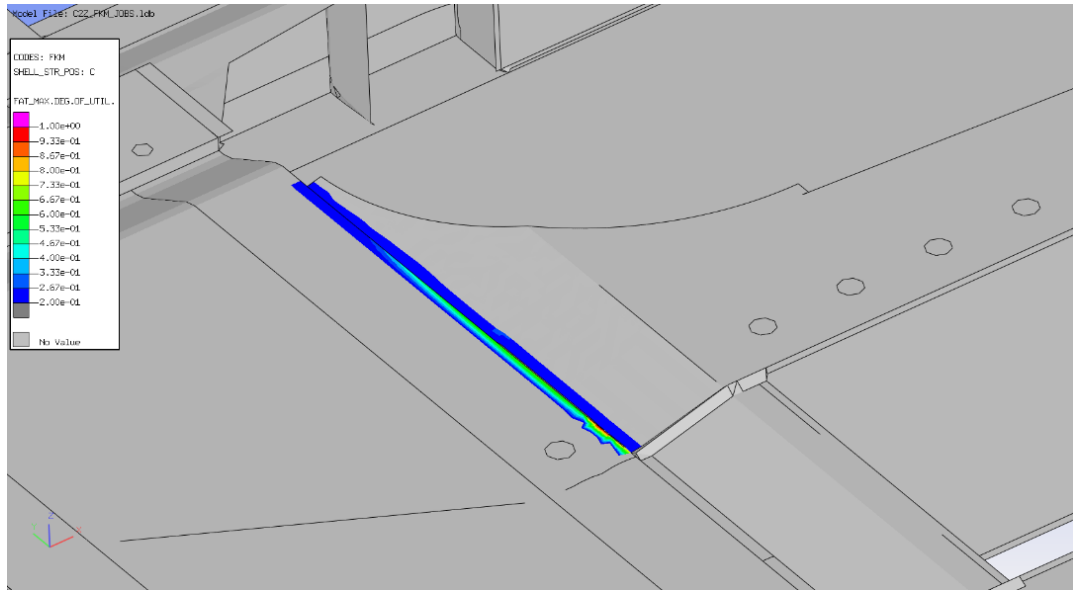


Figure 7.13: Detailed weld fatigue analysis result according to FKM Guideline: C2z loading scenario

7.8.2 DVS1612 Results

Weld global fatigue analysis have been performed with low class weld qualities, illustrated in Figure 7.14 according to DVS1612.

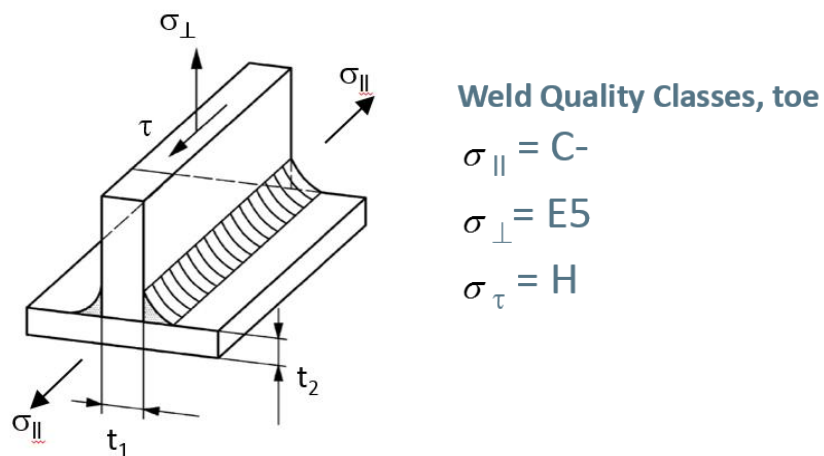


Figure 7.14: Weld global quality parameters according to DVS1612

The numerical material utilization outcomes (a quantity which is inverse of the safety margin for this case) have been obtained at specific weld elements of the model. One of the worst locations belonging to C1z loading scenario is seen in Figure 7.15.

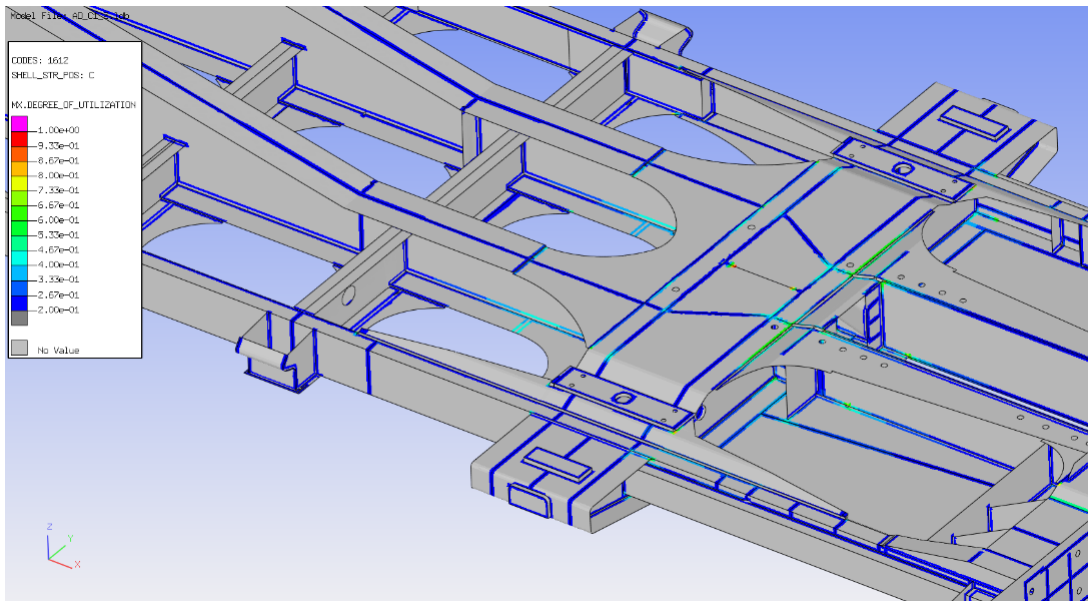


Figure 7.15: Weld global analysis result according to DVS1612: C1z loading scenario

An example view of results obtained from the global weld fatigue analyses with C2z loading scenario has been visually illustrated in Figure 7.16.

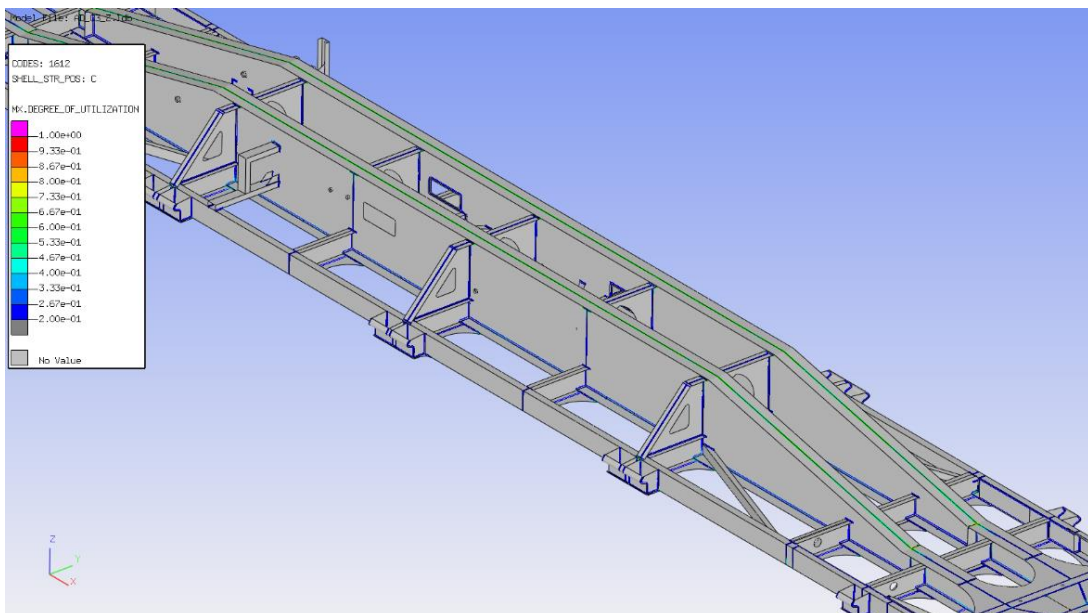


Figure 7.16: Weld global analysis result according to DVS1612: C3z loading scenario

Ten weld global analysis have been executed according to DVS1612 and 4 different critical welded areas have been determined.

One of the most critical welded areas can visually illustrated in Figure 7.17 with detail weld configuration and weld qualities according to DVS 1612.

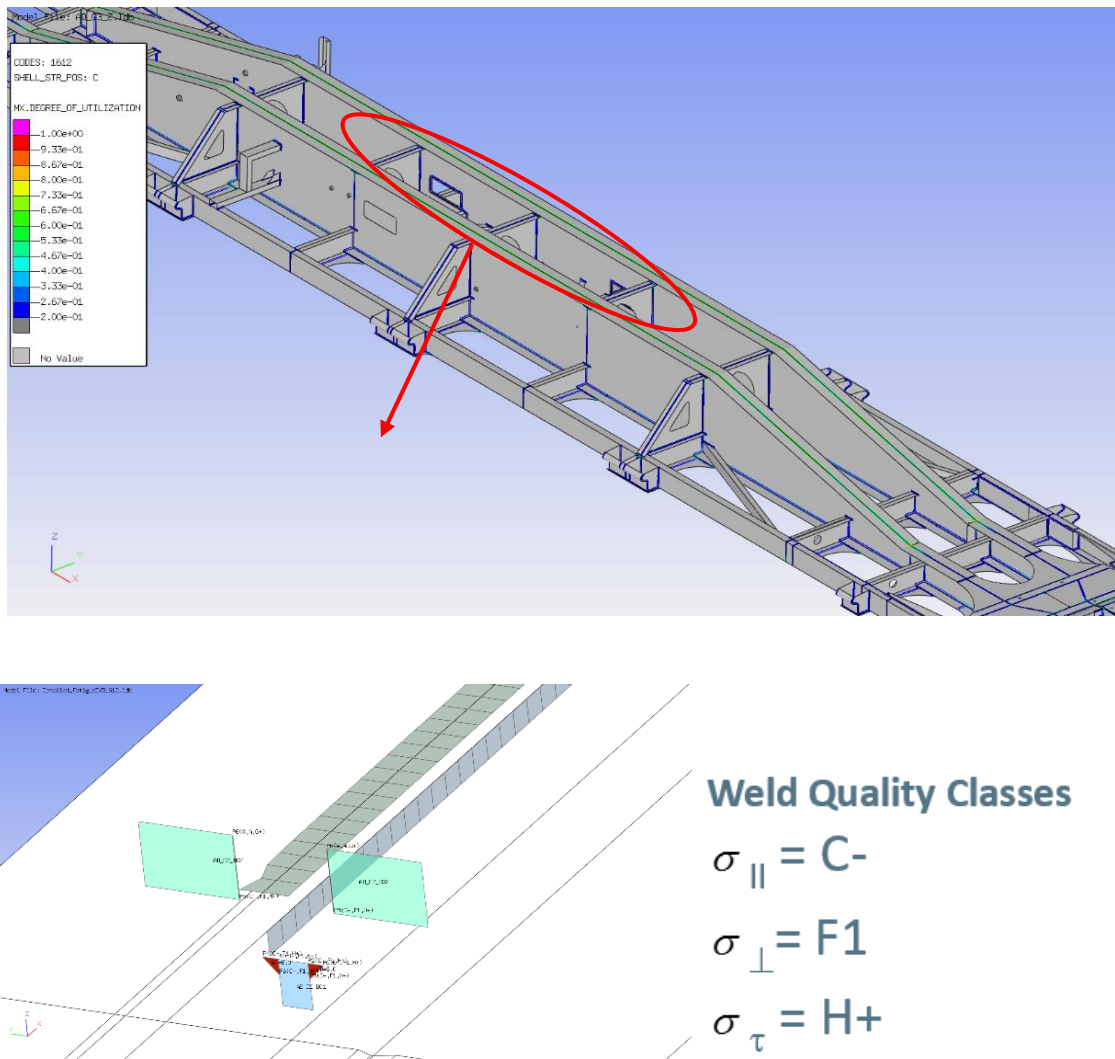


Figure 7.17: Detailed configuration of critical weld with quality classes according to DVS1612 Guideline

Detailed weld fatigue analysis result of the welded area given in Figure 7.17 has been visually illustrated in Figure 7.18.

All weld global analysis results and detailed weld fatigue analysis results can be found in APPENDIX A.

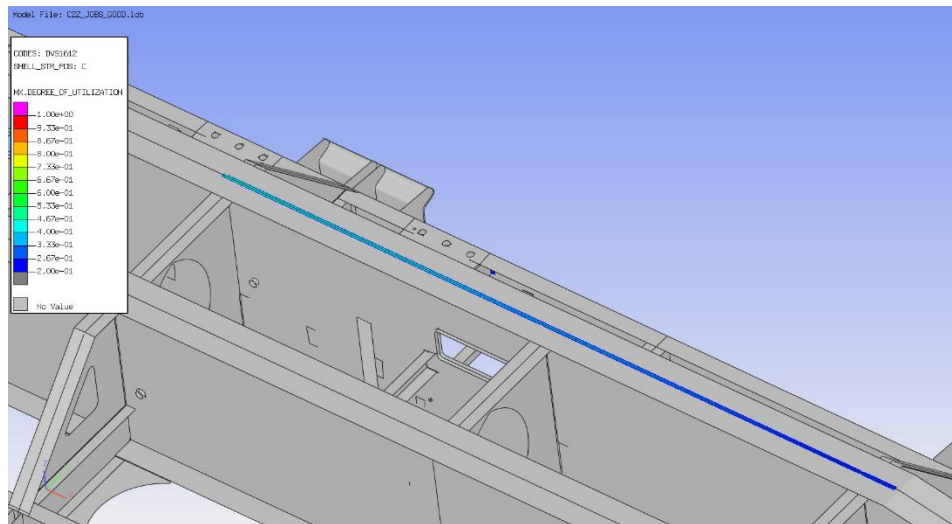


Figure 7.18: Detailed weld fatigue analysis result according to DVS1612 : C2z loading scenario

7.8.3 EN 1993-1-9 (Eurocode 3) results

Weld global fatigue analysis has been performed with FAT classes in Figure 7.19 according to EN 1993-1-9.

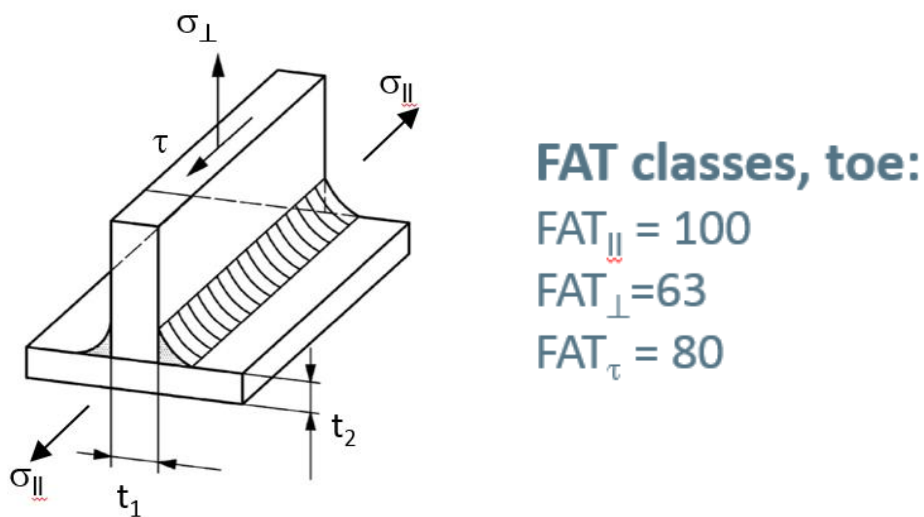


Figure 7.19: FAT classes for weld global fatigue analysis according to Eurocode 3

The numerical material utilization have been obtained at specific weld elements of the model. One of the worst locations belonging to C2y loading scenario is seen in Figure 7.20.

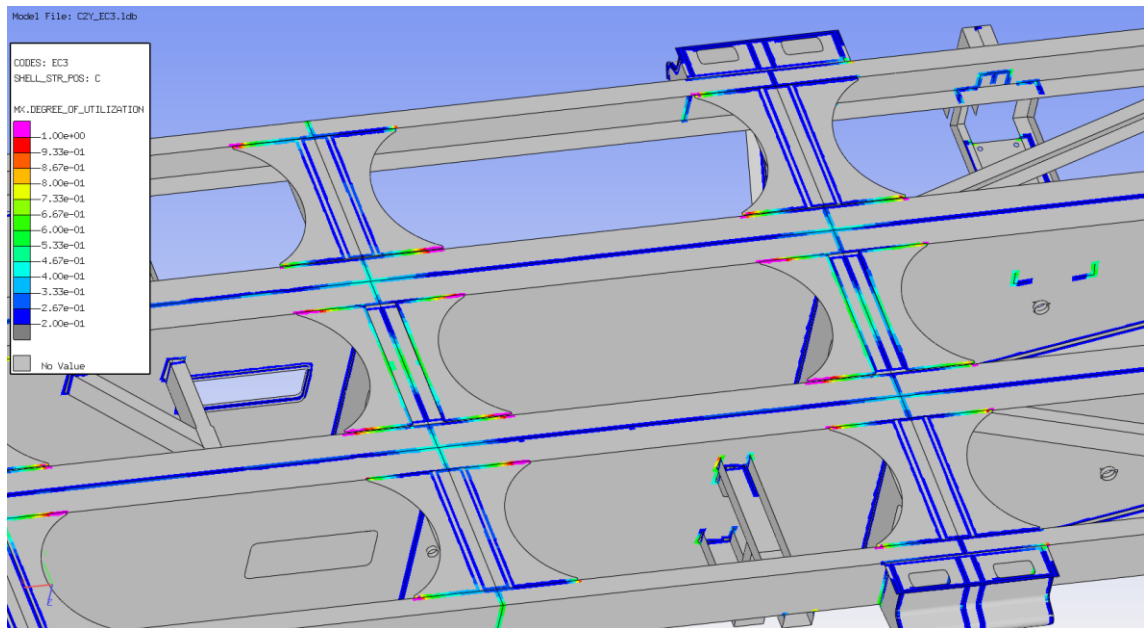


Figure 7.20: Weld global analysis result according to Eurocode 3: C2y loading scenario

Ten weld global analysis are executed regarding gravitational and centrifugal effects. After weld global analysis according to Eurocode 3 has been completed, the critical welded areas are determined.

According to critical welds spotted in weld global analysis, the detailed configuration of these welds are modelled in LIMIT software with full detailed FAT classes and properties.

An example view of one of these critical welded area, with detailed properties can be visually seen in Figure 7.21.

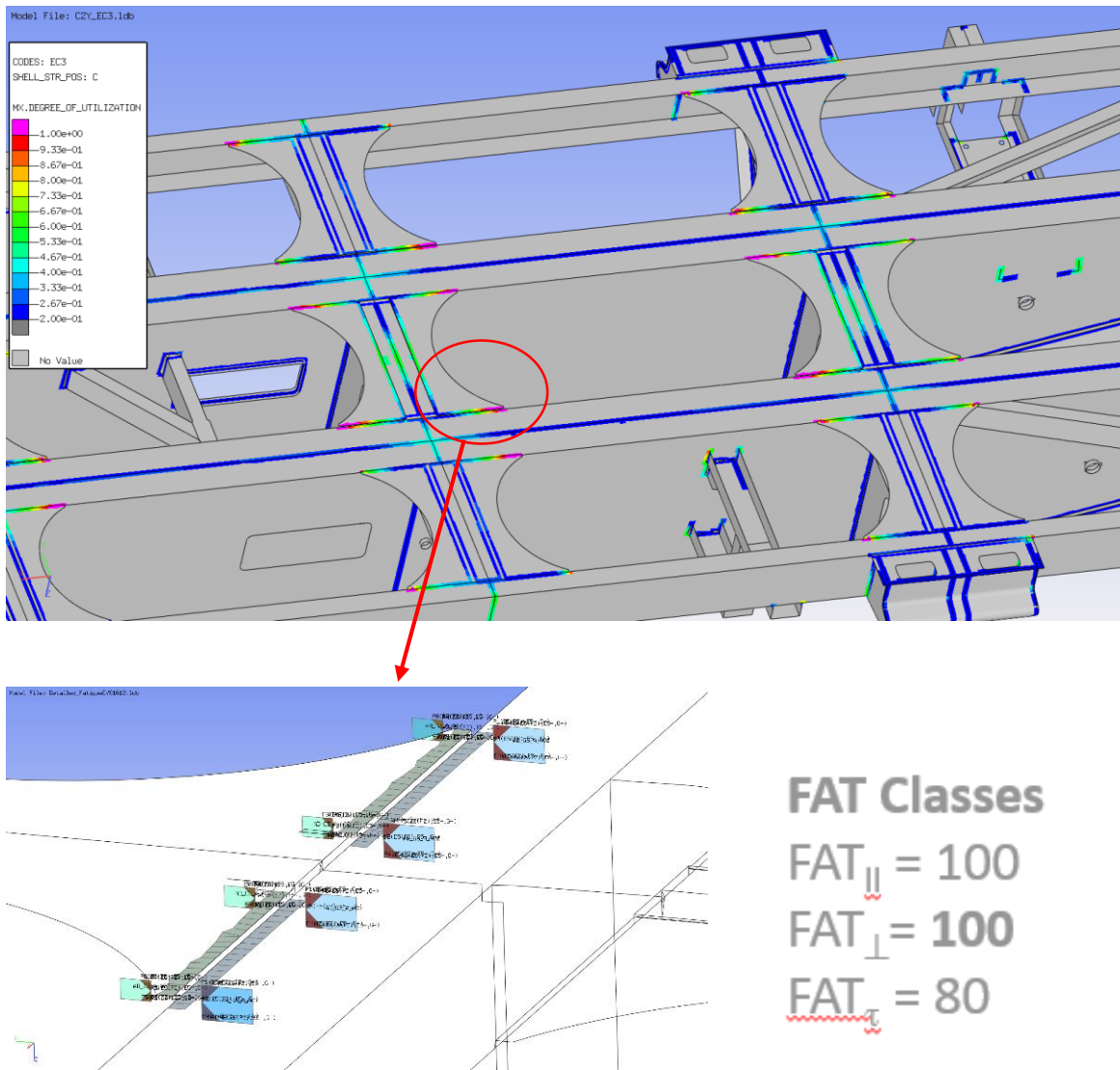


Figure 7.21: Detailed configuration of critical weld with fat classes according to FKM Guideline

Four critical welds have been spotted via weld global analysis. Here, one of the most critical ones' result is visualized. All weld global analysis and detailed critical weld analysis can be found in APPENDIX A according to aforementioned loading scenarios.

In Figure 7.22 an example result of critical weld analysis is illustrated.

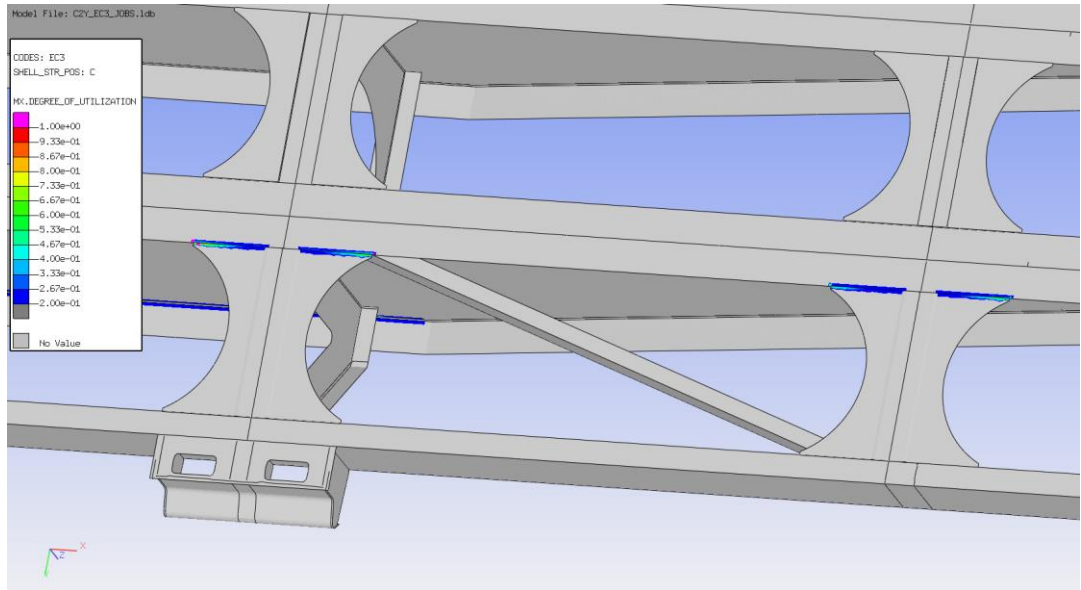


Figure 7.22: Detailed weld fatigue analysis result according to FKM Guideline: C2y loading scenario

Weld fatigue strength assessment of the freight wagon chassis has been done with three different international fatigue strength assessment codes with LIMIT software package.

All FEA analysis result of the chassis, weld global fatigue strength assessment analysis and detailed weld fatigue analysis results of the chassis according to three different codes can be found in APPENDIX A.

8. CONCLUSION

Engineering structures are expected to work under dynamic loads due to operating conditions, as well as structural and infrastructure-based excitations. Especially mechanical engineering structures, like railway vehicles, are expected to run under highly impact like conditions. And these loading conditions, may causes undesired and unexpected fatigue failures of the structure. Welded joints of the structures are, thus, the most critical areas when compared to main structural supporting girders and have to ensure the desired fatigue conditions.

Throughout this study it has been exposed that the yet unclear nature of fatigue, which is even more uncertain when it comes to fatigue on welded structures. Welded components may have complex state of stresses, complicated geometries and a great variety of defects introduced during welding process.

Many theories have been developed along the years to account different factors and their influence on fatigue behaviour of materials. Comments and comparisons between these theories and international codes, have been performed with the aim to provide the engineers/ academicians a better understanding of regulation from the industry point of view. In this sense, most important findings about fatigue behaviour and the fatigue strength assessment of welded structures are published in international codes, as also examined in this thesis, FKM, DVS1612, EC3.

Within scope of this thesis, aforementioned three international weld fatigue strength assessment codes have been studied both theoretically and with the industrial example of freight wagon. All of these three codes have been applied to assess the fatigue strength to welded areas of the wagon chassis in the same manner. First the global weld fatigue assessment has been performed in order to capture all the possible critical areas in the structure and secondly the detailed weld fatigue strength assessment in order to model as in real life structure and to see the more realistic results. While FKM and EC3 uses FAT classes to assign the welding quality of the critical areas, DVS1612 has its own quality parameters with MKJ diagrams. And also, while FKM Guideline and EC3 standards may have been applied all engineering structures, DVS 1612

standard should only be applied to rail vehicles. When dealing with rail vehicle structures as freight wagons, passenger coaches and locomotives, DVS 1612 can be thought as a benchmark code.

There are two ways in order to decrease the degree of utilization in the critical welded areas in the structures. First one is to improve welding quality via peening, toe grinding, heat treatment. These methods improve the welding quality, here in this thesis FAT classes in FKM and EC3 and quality classes in DVS 1612, but none of these methods can change the stress conditions in the structure. The second one, to alter the design in order to modify the critical section with additional part(s). However, welding an extra support to the structure can lead to a worse situation, since adding a structural support to the structure means adding an extra weldments to the structure, and if the stress state around the critical area is not well studied, this second approach may result in a more critical condition.

As a result of this study, many researchers and engineers are focusing on the static and dynamic behaviour of the mechanically stressed structures' main girders and supports. This study shows the importance and necessity of examining the welded areas of engineering structures. Examining these most critical sections in the structures require great experience and requires to have a good understanding of stress state. Well studied fatigue strength assessment of welded areas in the structure may prevent an undesired and unexpected fatigue failure

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APPENDIX A

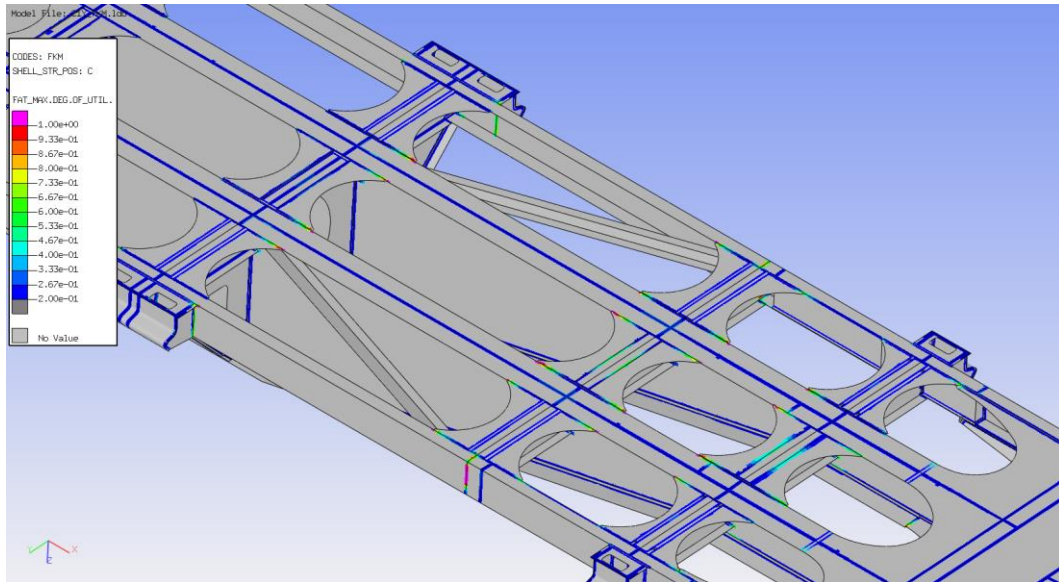


Figure A.1: Weld global fatigue analysis result-FKM-C1Y

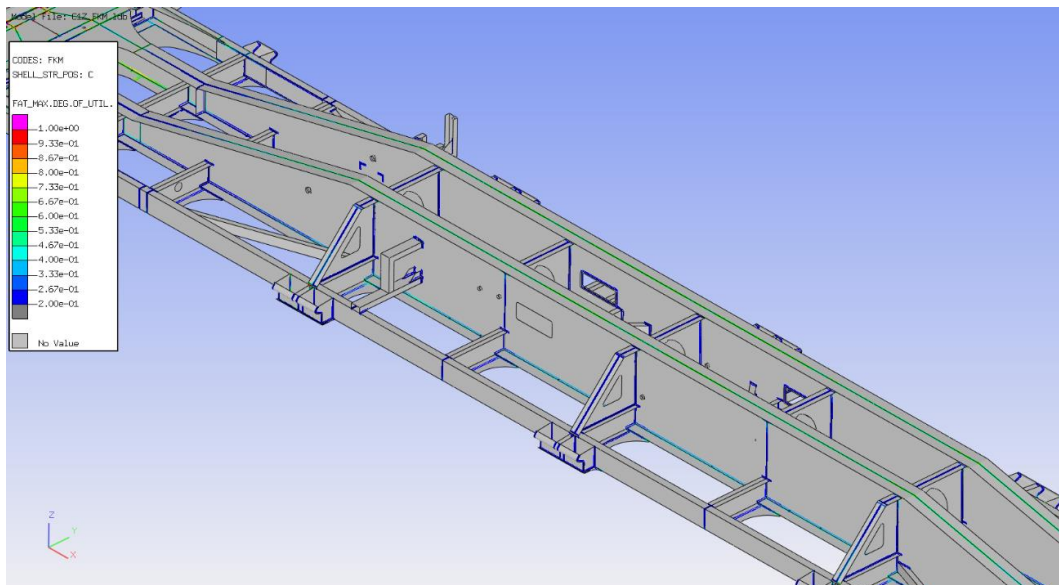


Figure A.2: Weld global fatigue analysis result-FKM-C1Z

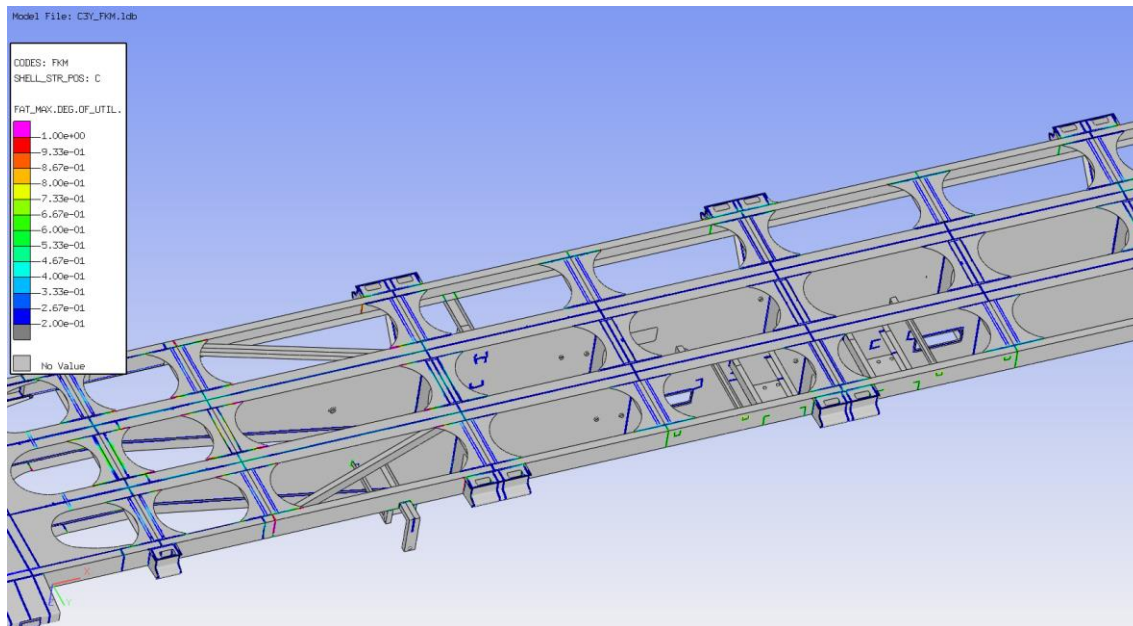


Figure A.3: Weld global fatigue analysis result-FKM-C2Y

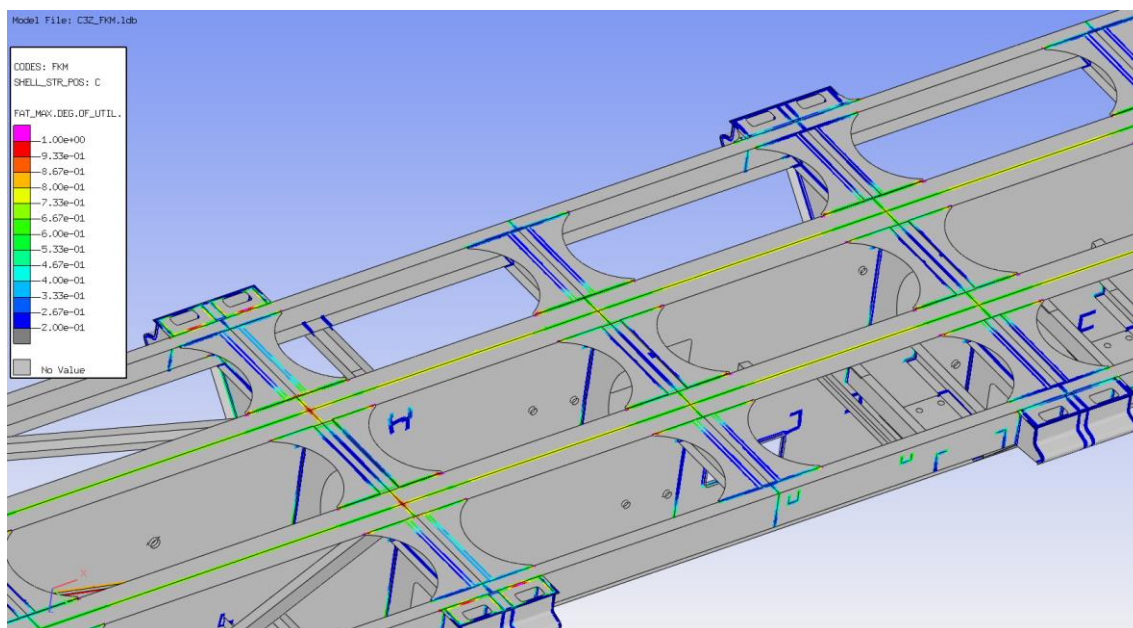


Figure A.4: Weld global fatigue analysis result-FKM-C2Z

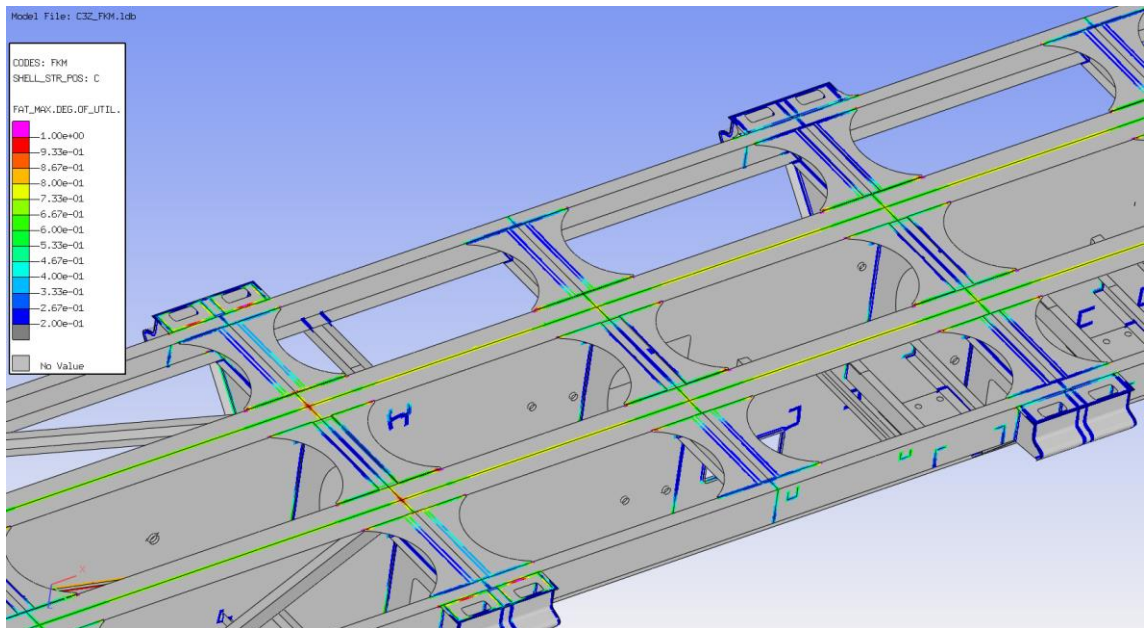


Figure A.5: Weld global fatigue analysis result-FKM-C3Y

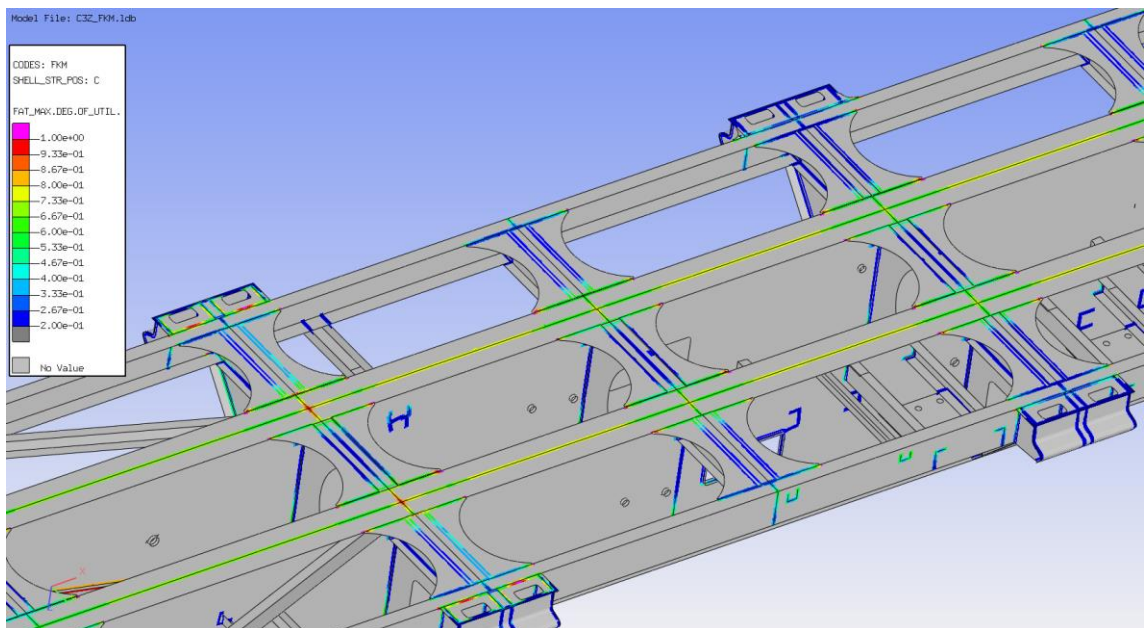


Figure A.6: Weld global fatigue analysis result-FKM-C3Z

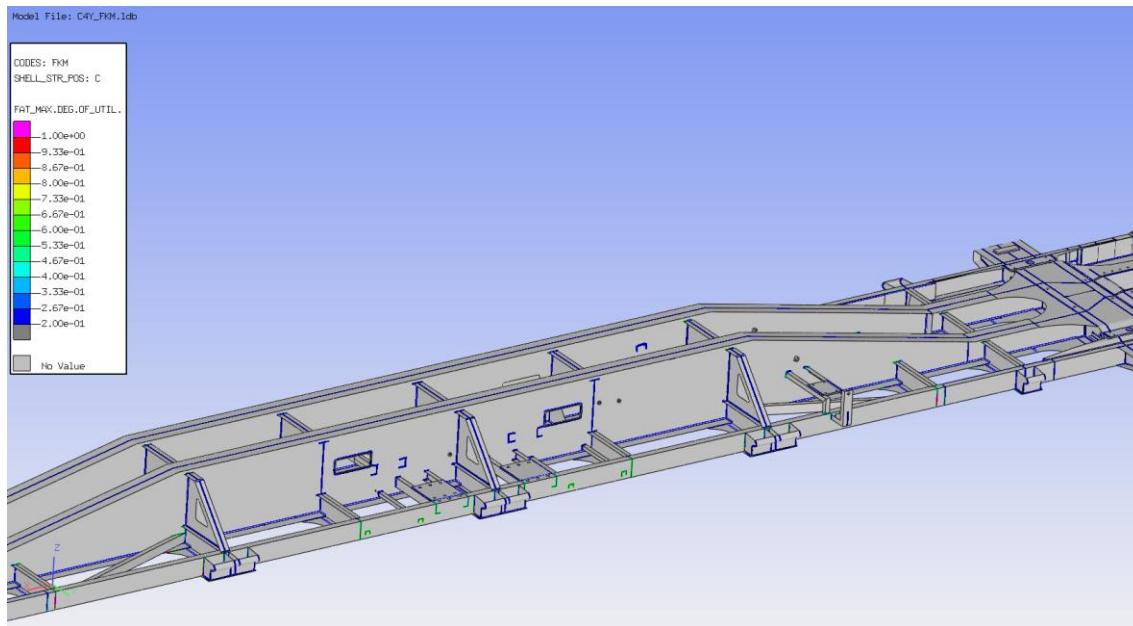


Figure A.7: Weld global fatigue analysis result-FKM-C4Y

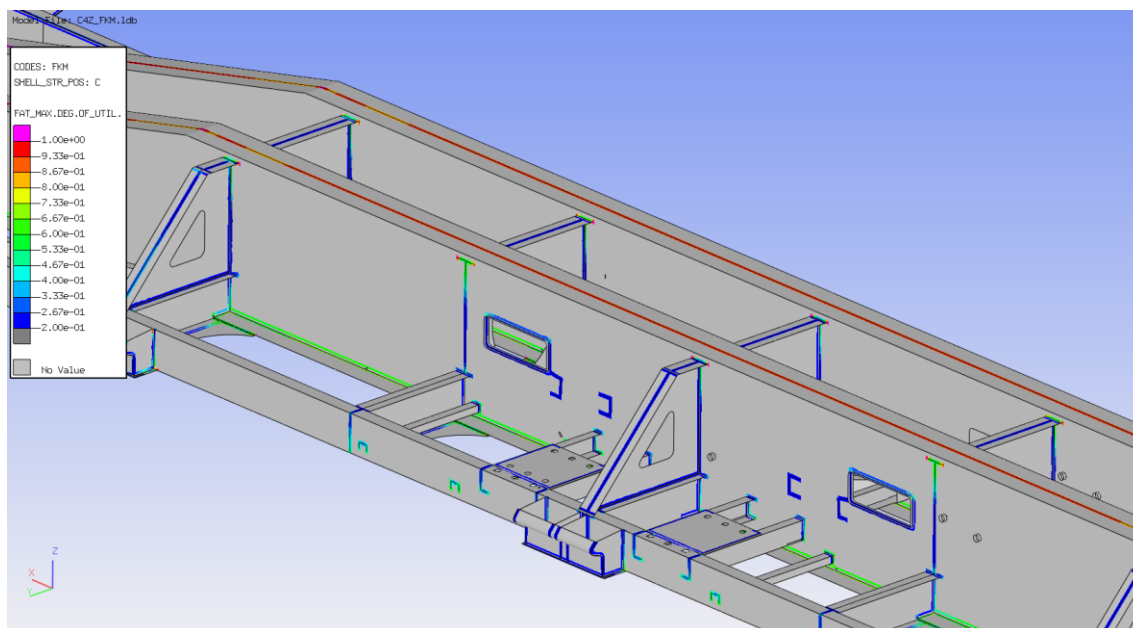


Figure A.8: Weld global fatigue analysis result-FKM-C4Z

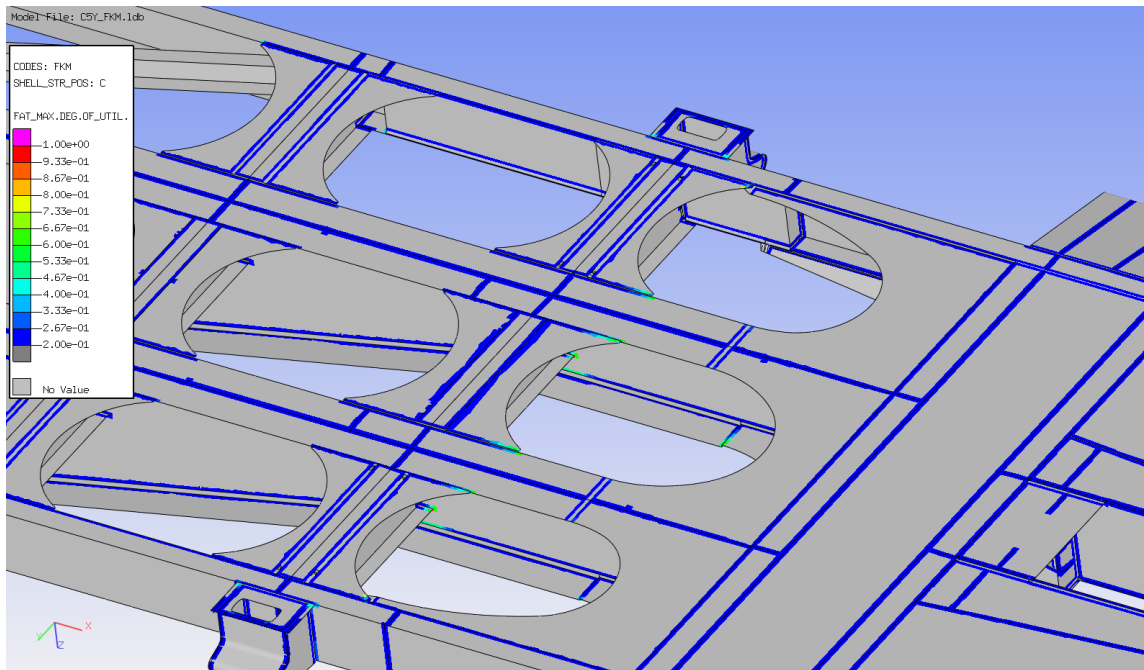


Figure A.9: Weld global fatigue analysis result-FKM-C5Y

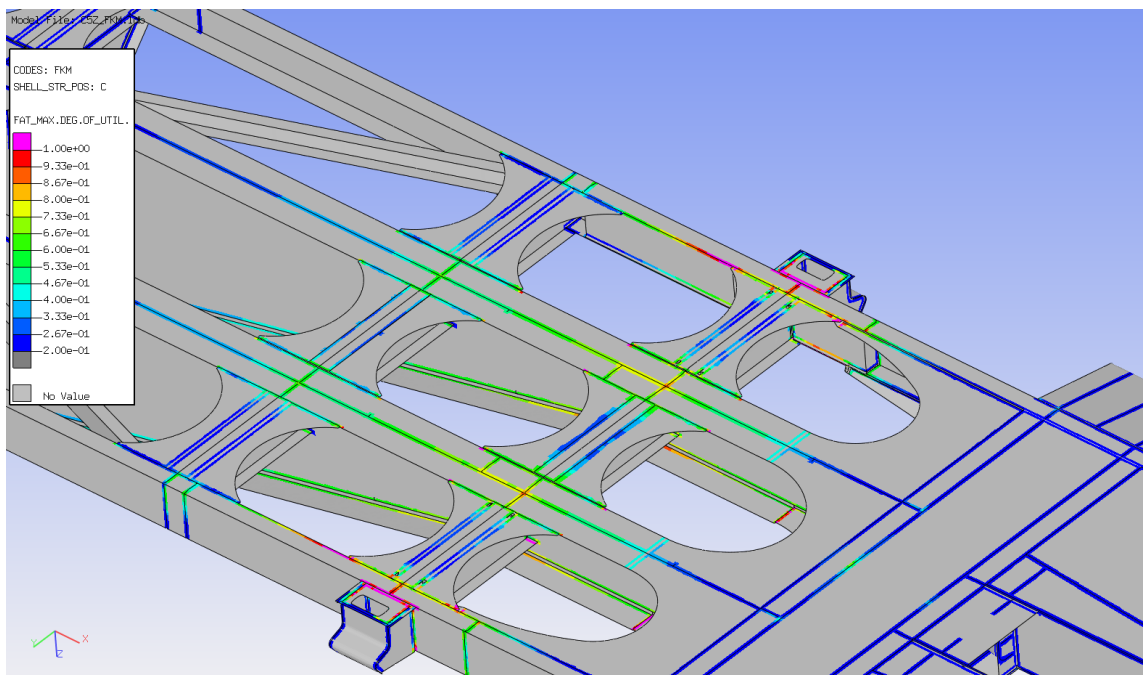


Figure A.10: Weld global fatigue analysis result-FKM-C5Z

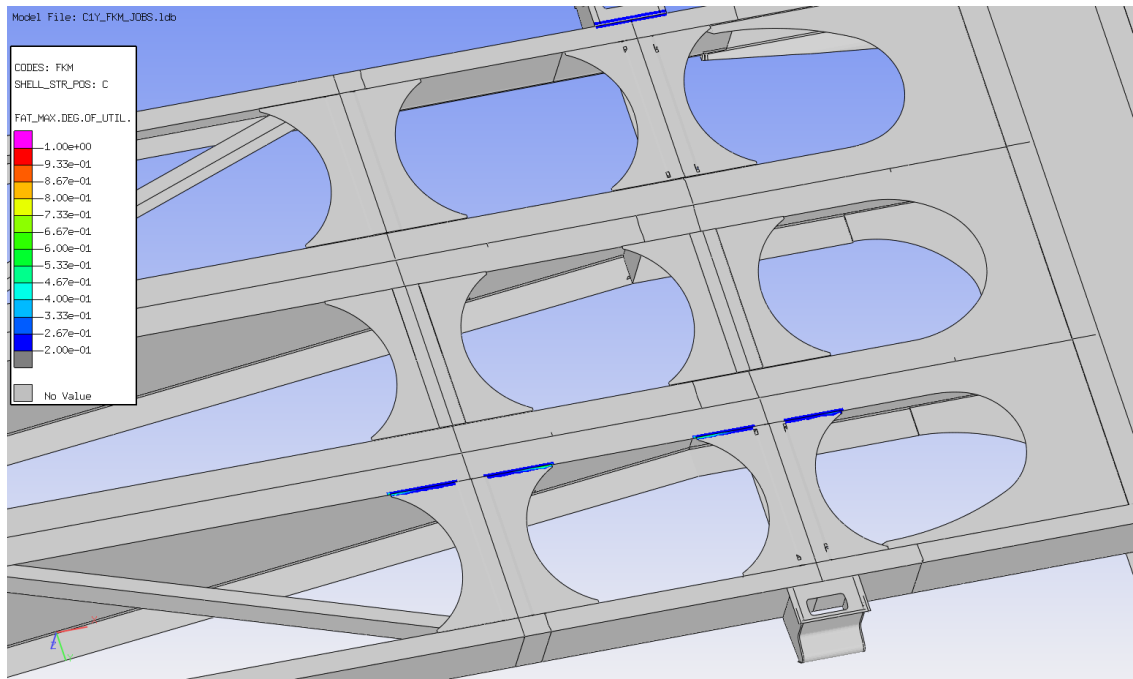


Figure A.11: Detail weld fatigue analysis result worst locations-FKM-C1Y

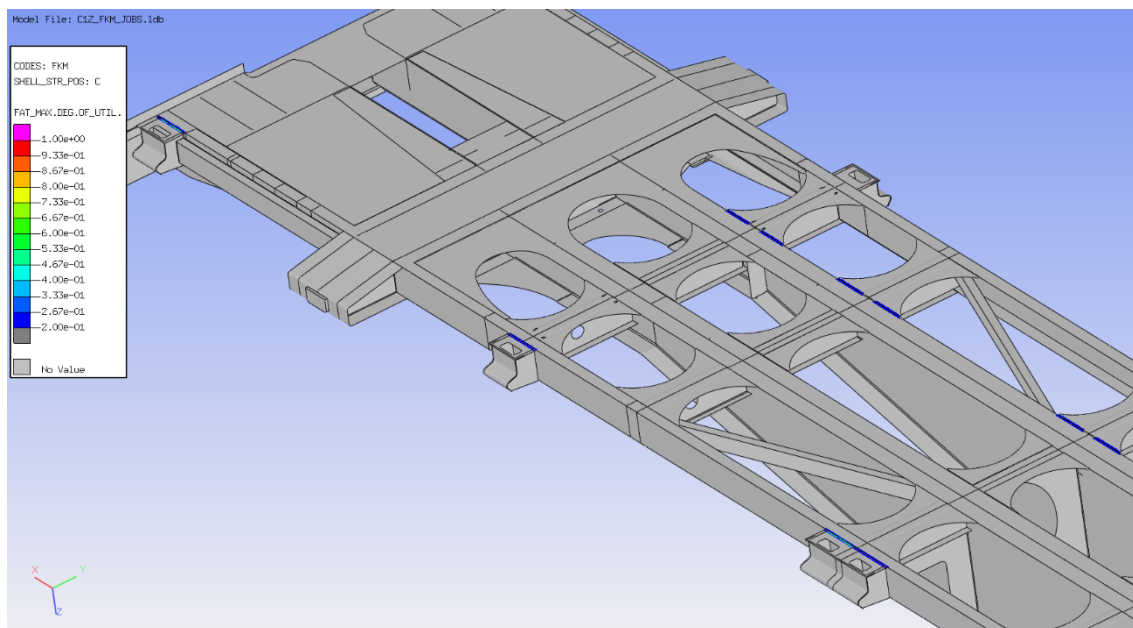


Figure A.12: Detail weld fatigue analysis result worst locations-FKM-C1Z

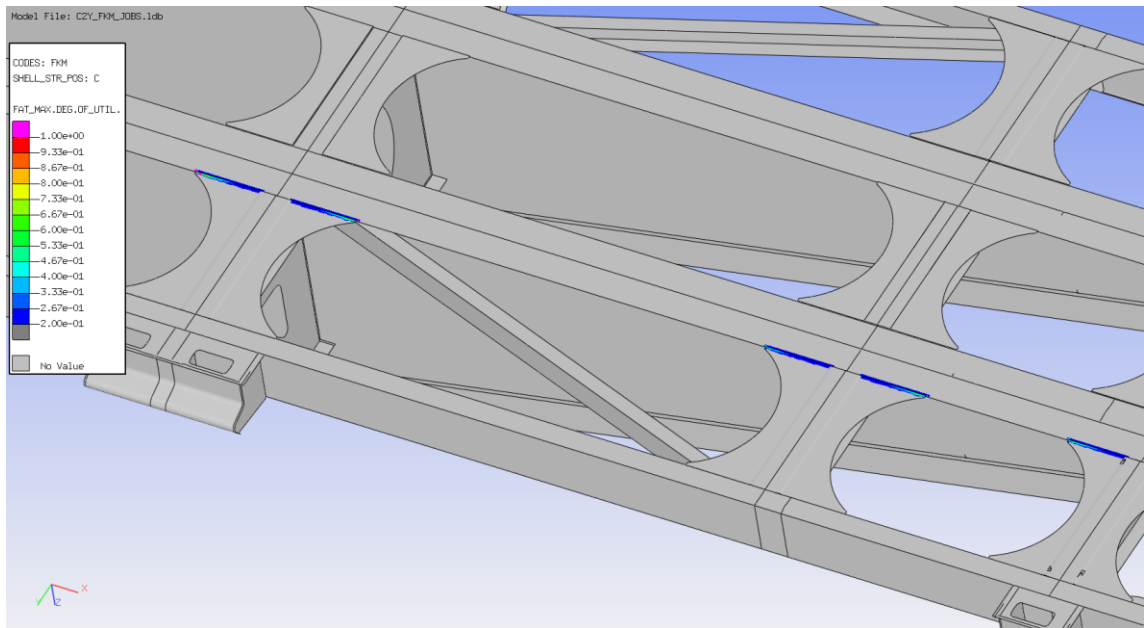


Figure A.13: Detail weld fatigue analysis result worst locations-FKM-C2Y

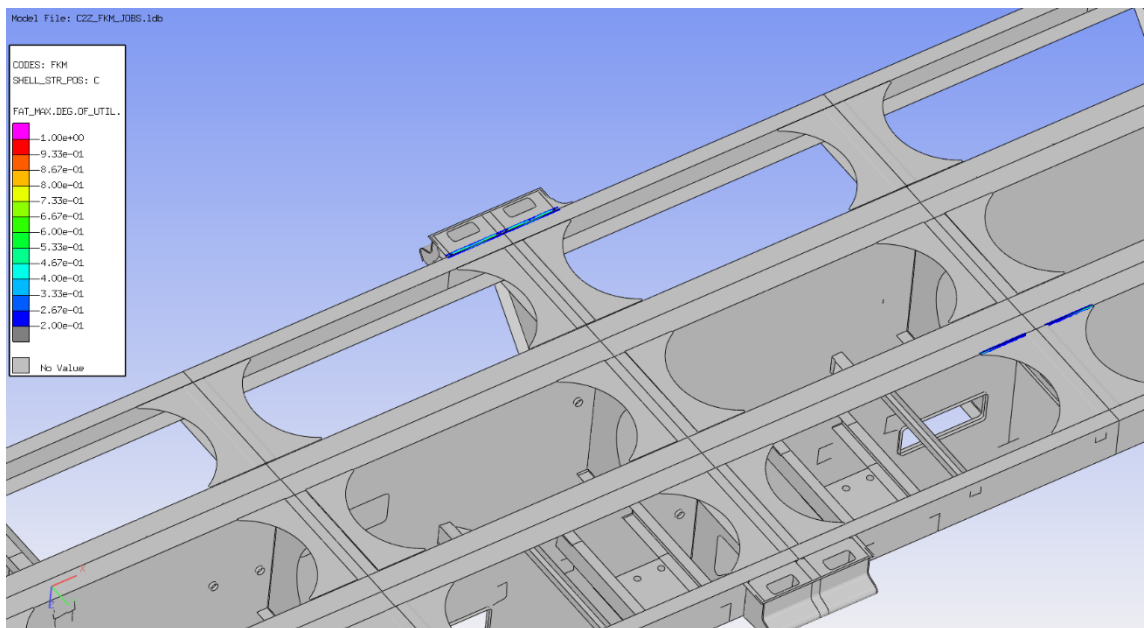


Figure A.14: Detail weld fatigue analysis result worst locations-FKM-C2Z

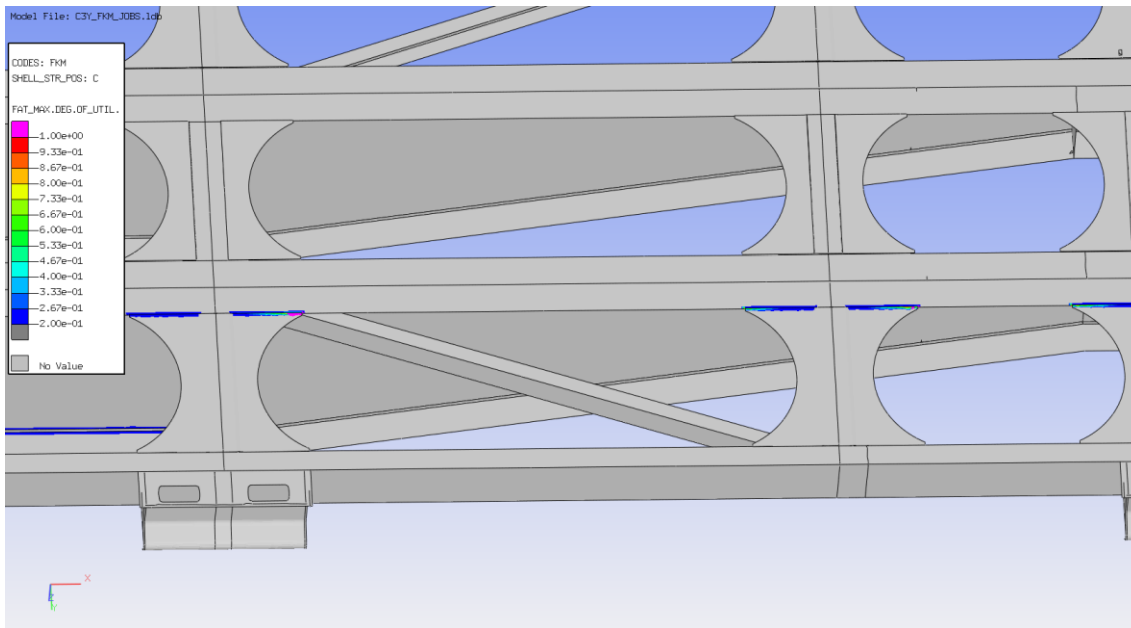


Figure A.15: Detail weld fatigue analysis result worst locations-FKM-C3Y

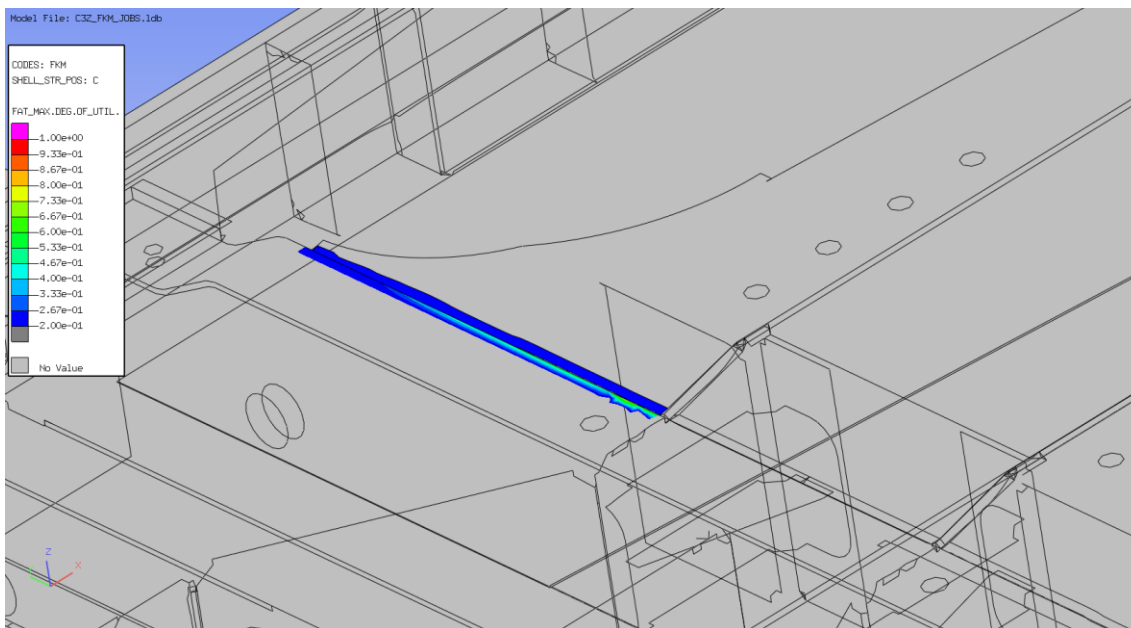


Figure A.16: Detail weld fatigue analysis result worst locations-FKM-C3Z

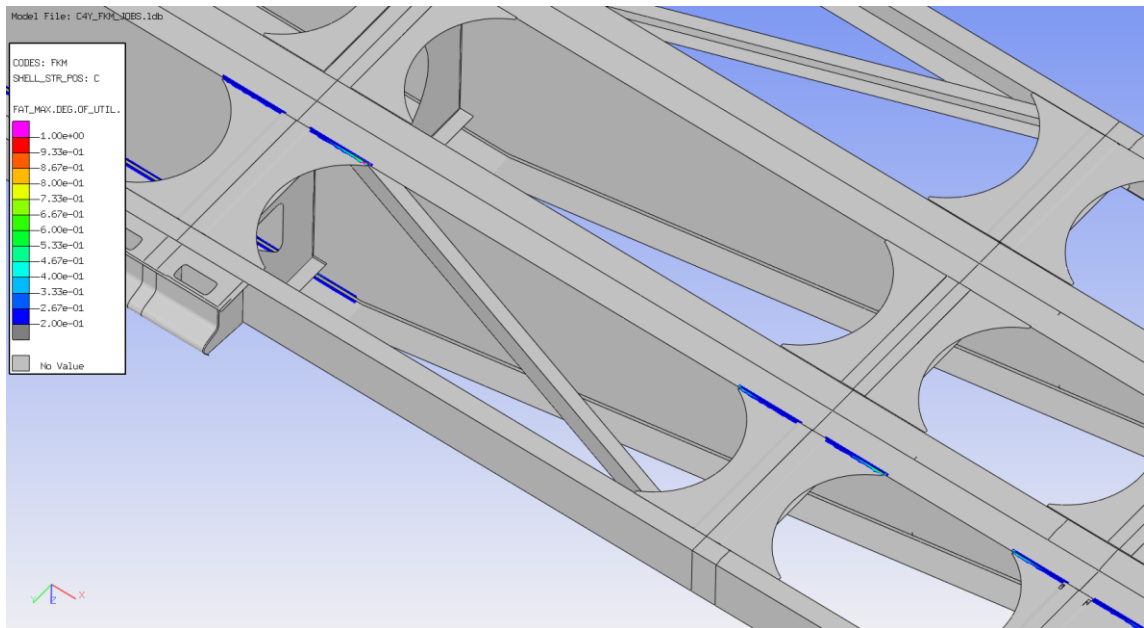


Figure A.17: Detail weld fatigue analysis result worst locations-FKM-C4Y

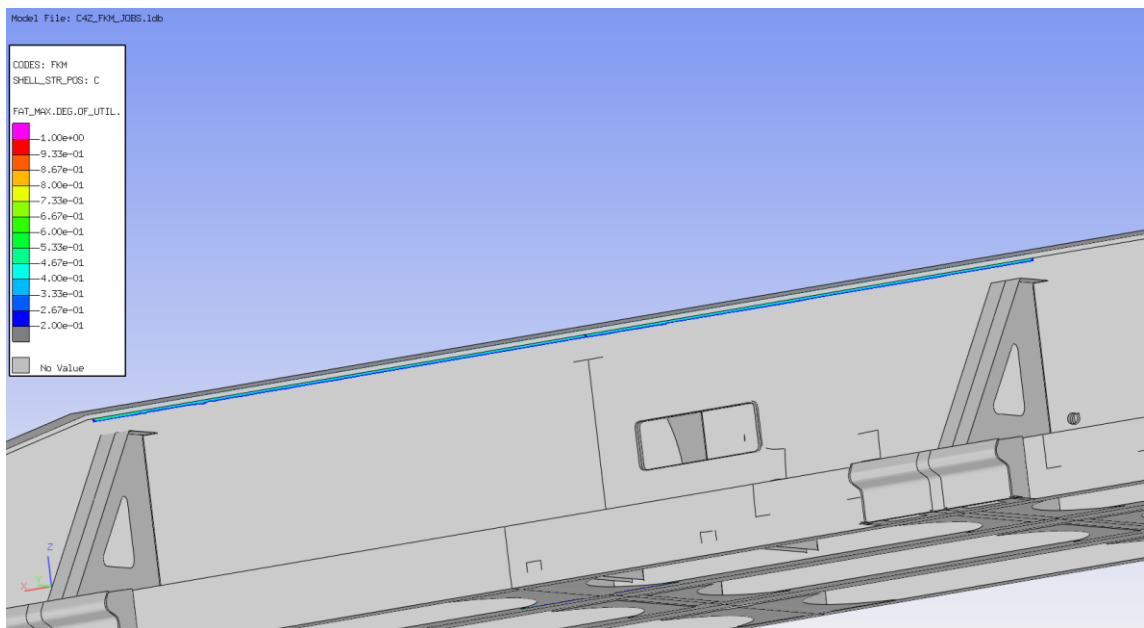


Figure A.18: Detail weld fatigue analysis result worst locations-FKM-C4Z

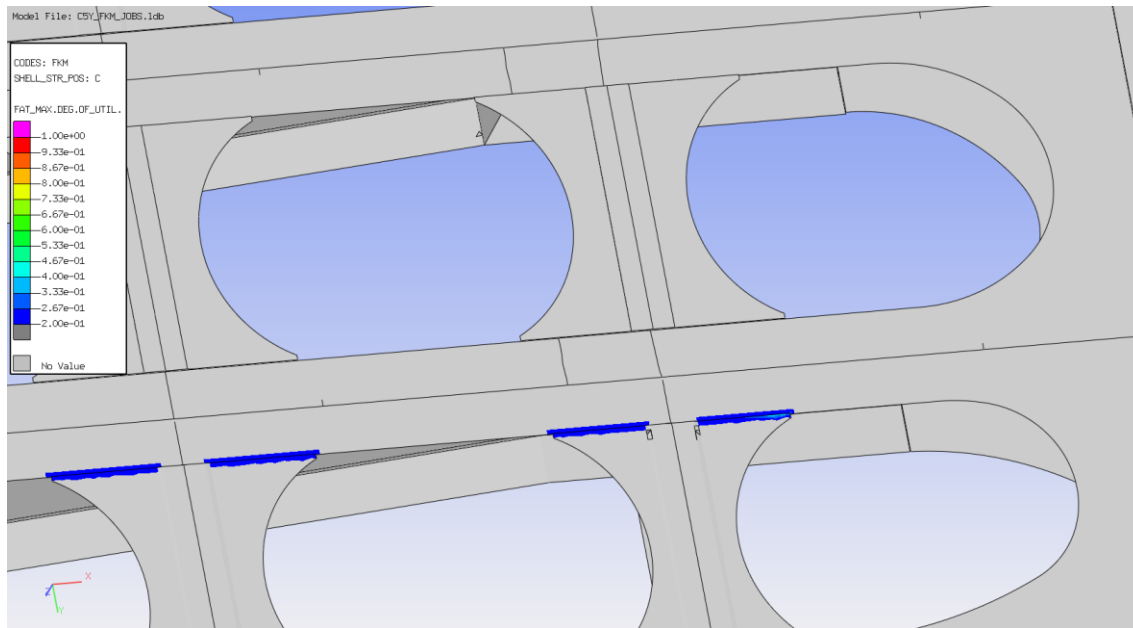


Figure A.19: Detail weld fatigue analysis result worst locations-FKM-C5Y

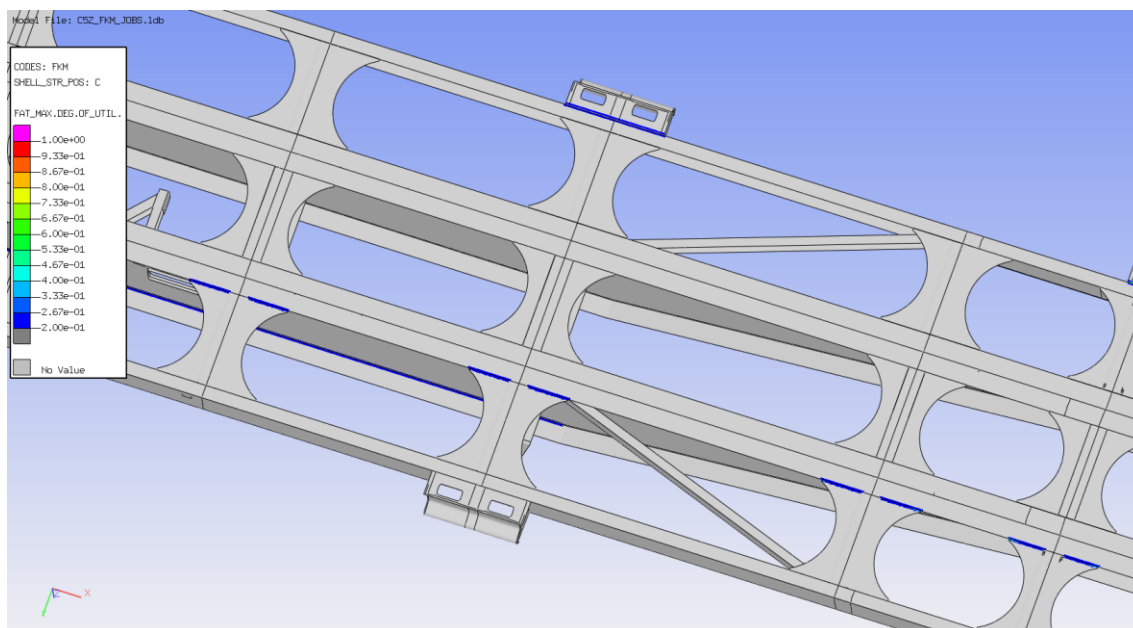


Figure A.20: Detail weld fatigue analysis result worst locations-FKM-C5Z

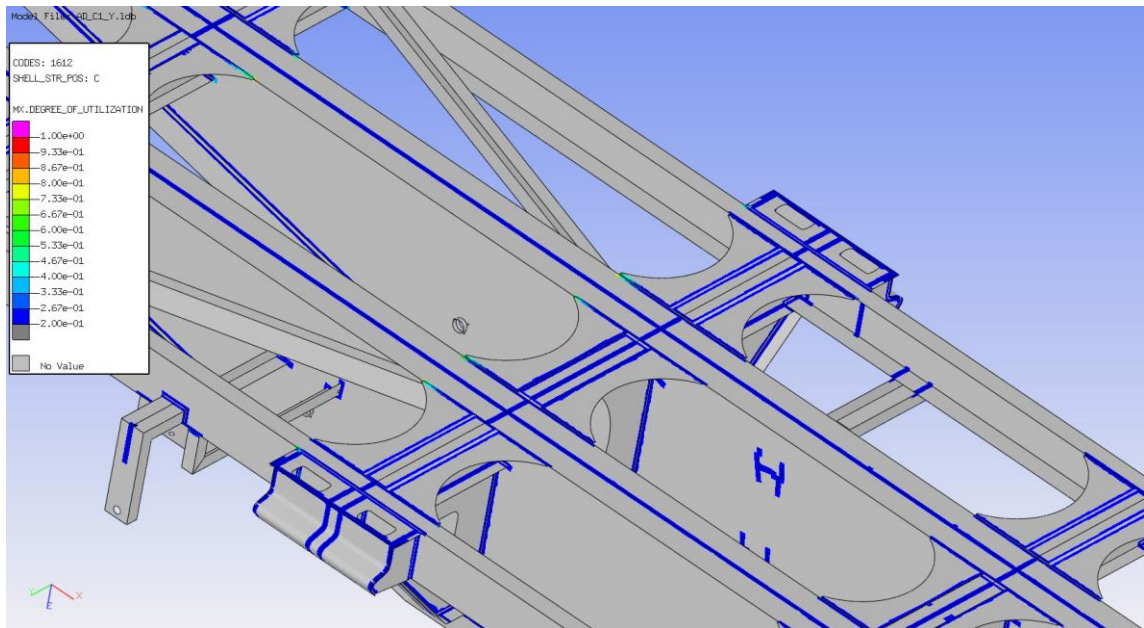


Figure A.21: Weld global fatigue analysis result-DVS1612-C1Y

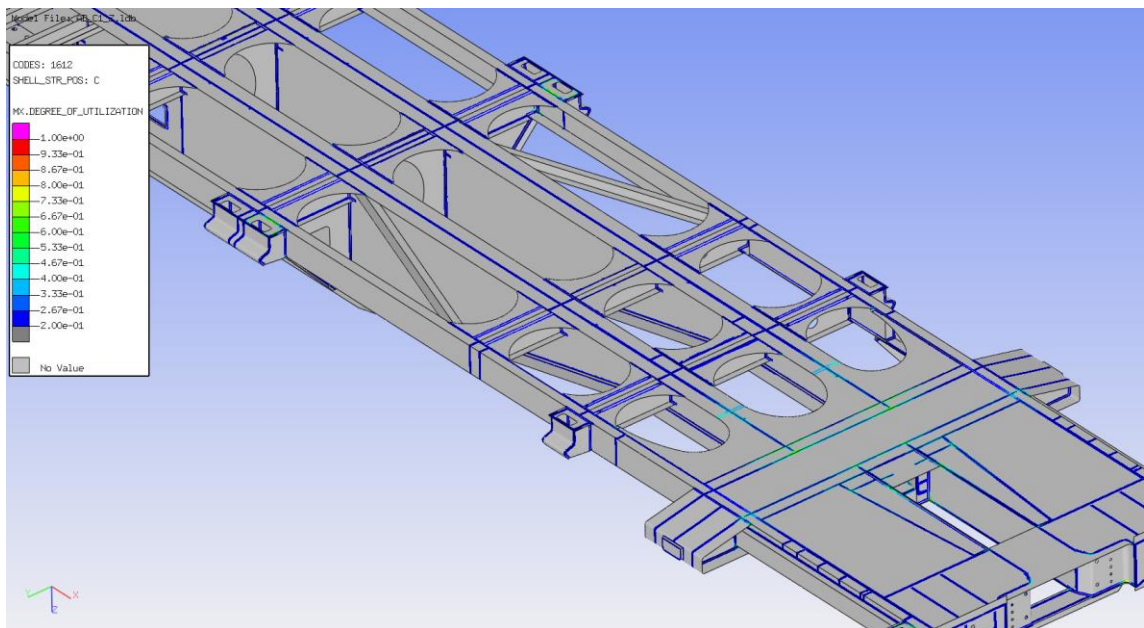


Figure A.22: Weld global fatigue analysis result-DVS1612-C1Z

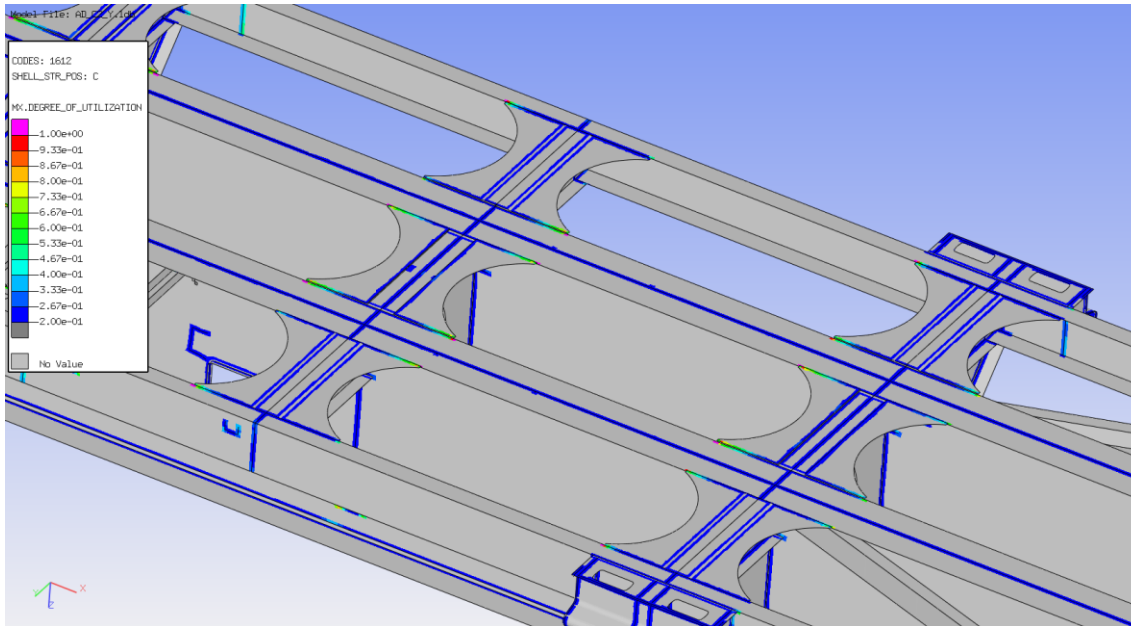


Figure A.23: Weld global fatigue analysis result-DVS1612-C2Y

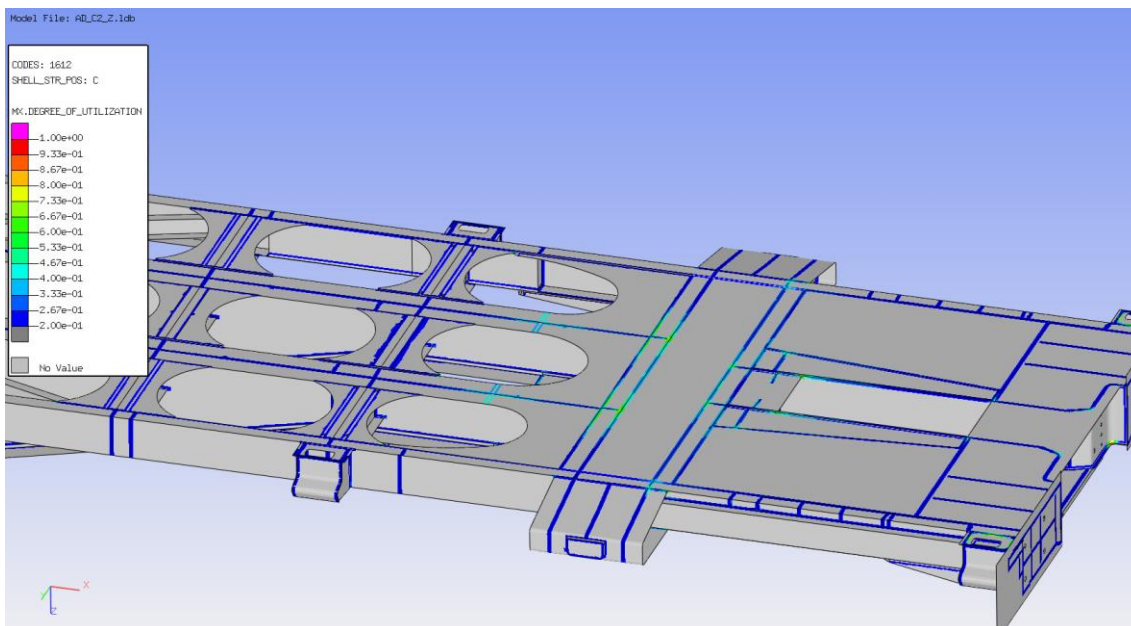


Figure A.24: Weld global fatigue analysis result-DVS1612-C2Z

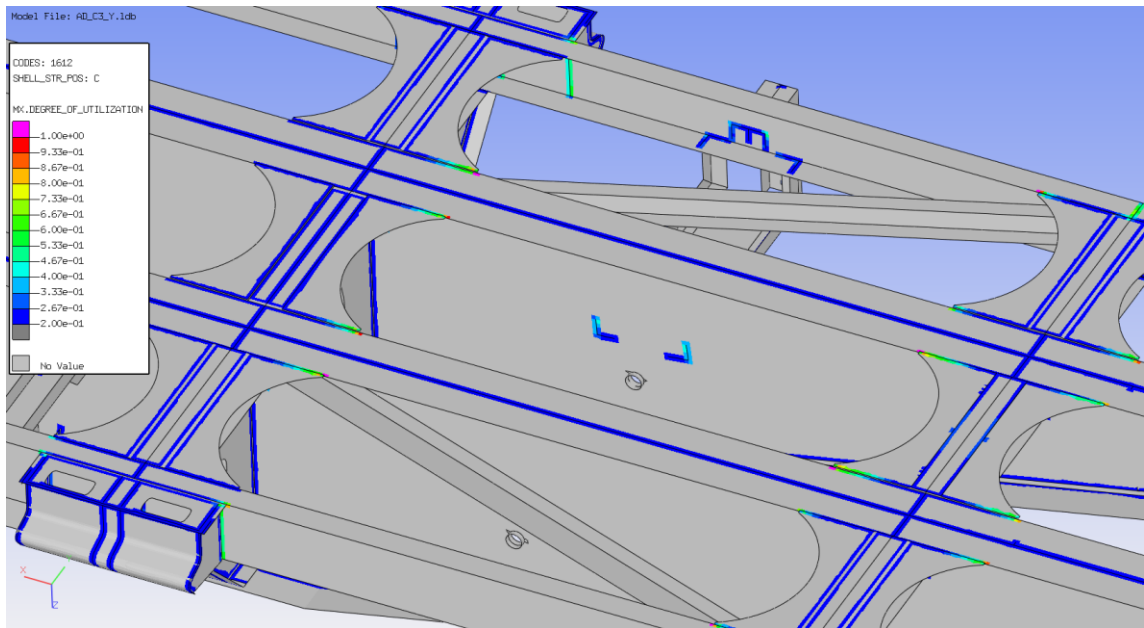


Figure A.25: Weld global fatigue analysis result-DVS1612-C3Y

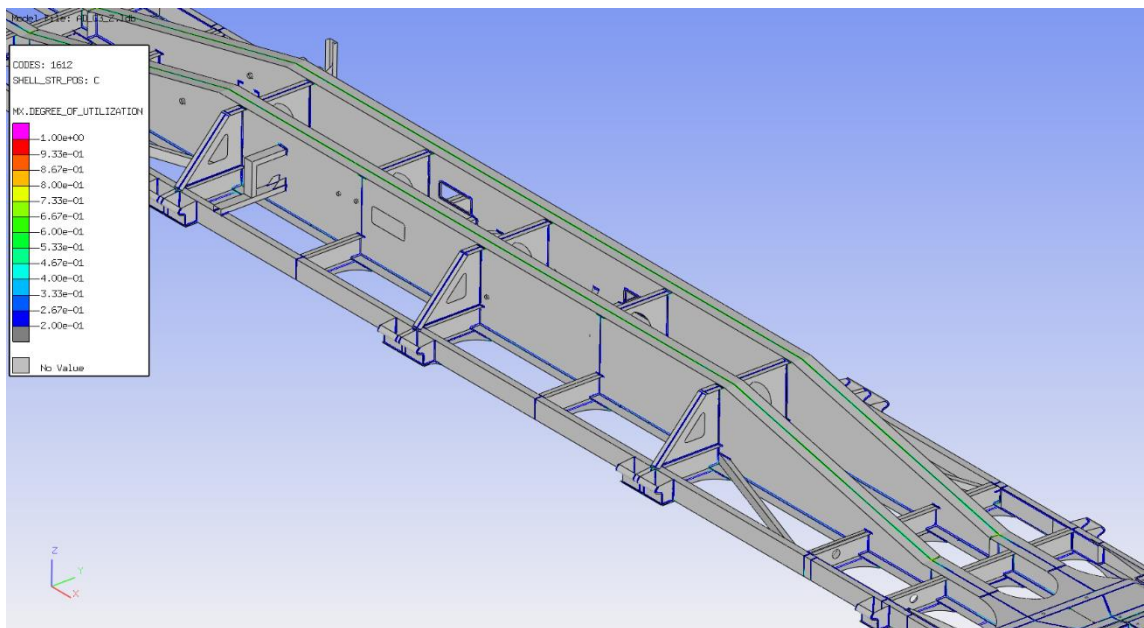


Figure A.26: Weld global fatigue analysis result-DVS1612-C3Z

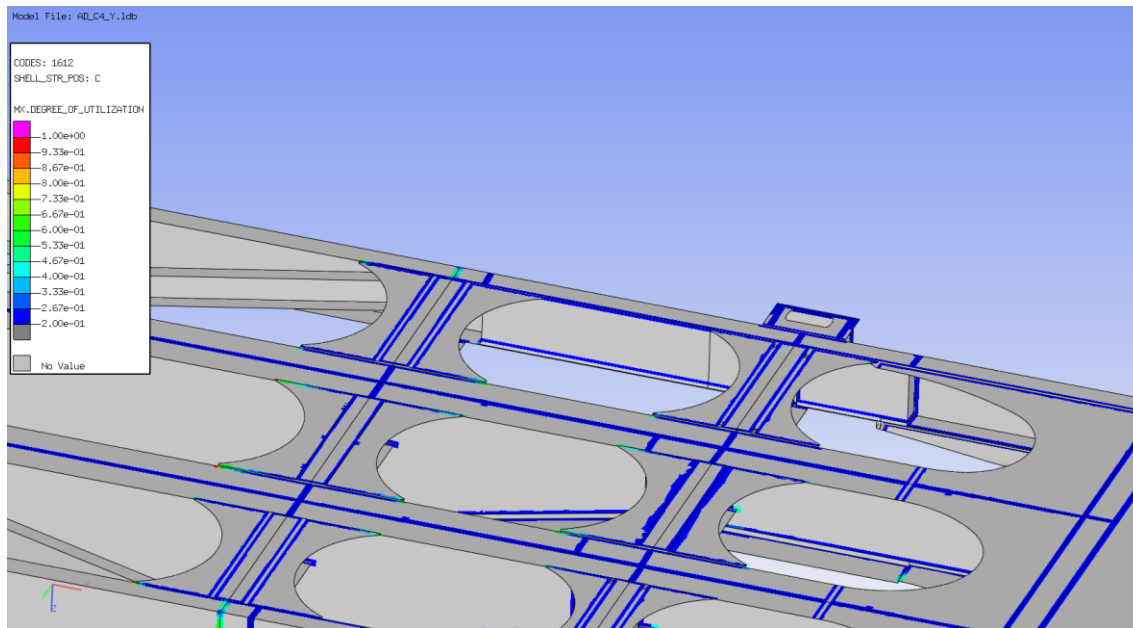


Figure A.27: Weld global fatigue analysis result-DVS1612-C4Y

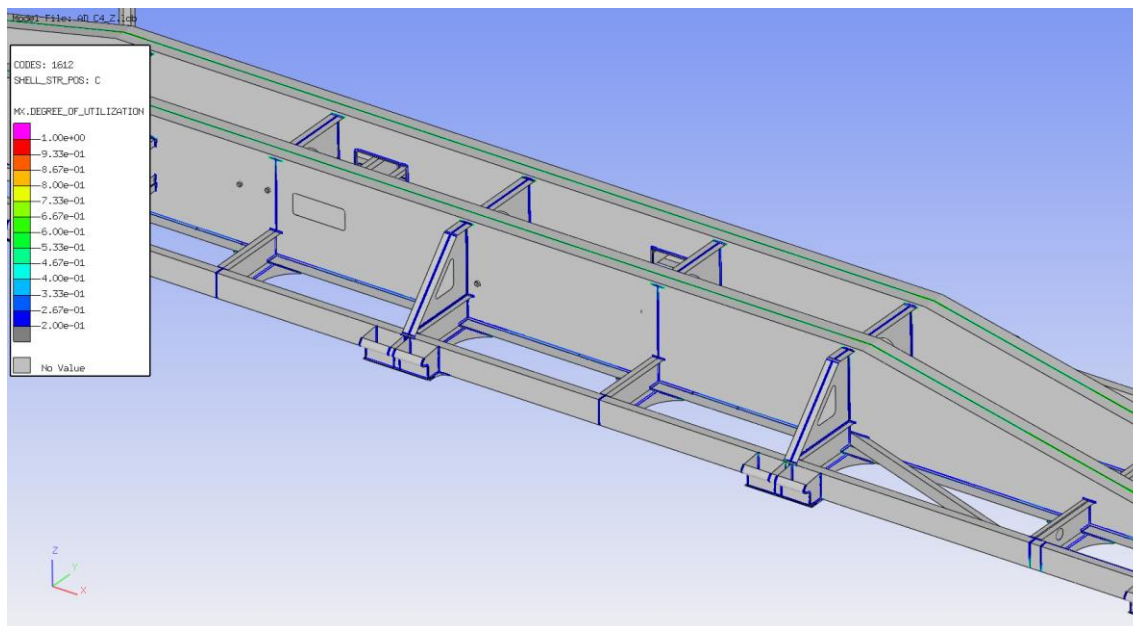


Figure A.28: Weld global fatigue analysis result-DVS1612-C4Z

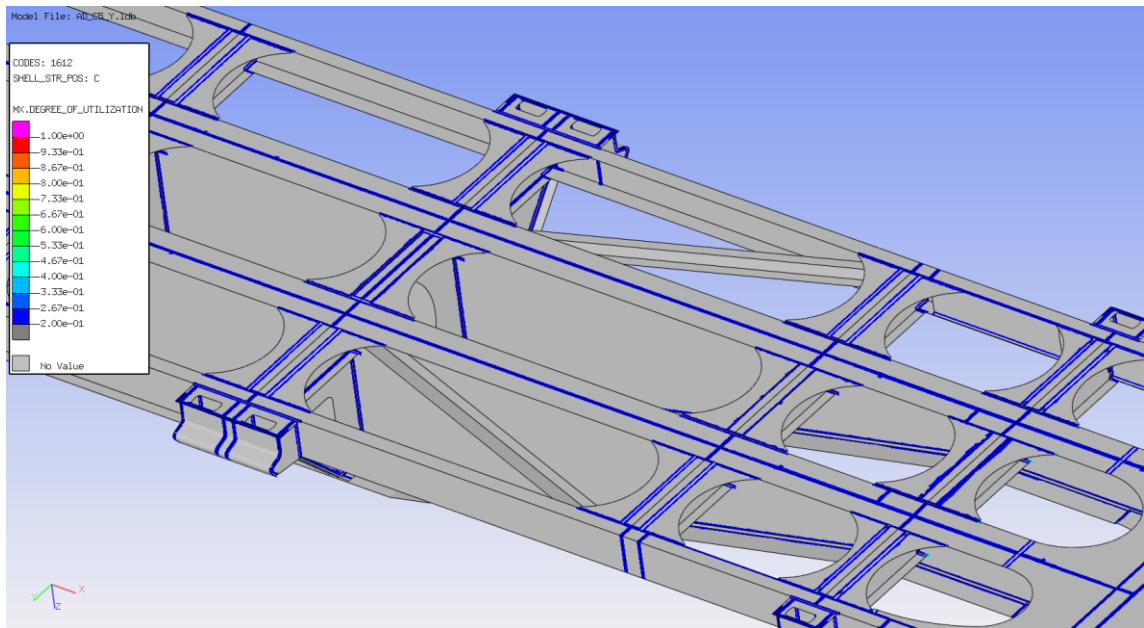


Figure A.29: Weld global fatigue analysis result-DVS1612-C5Y

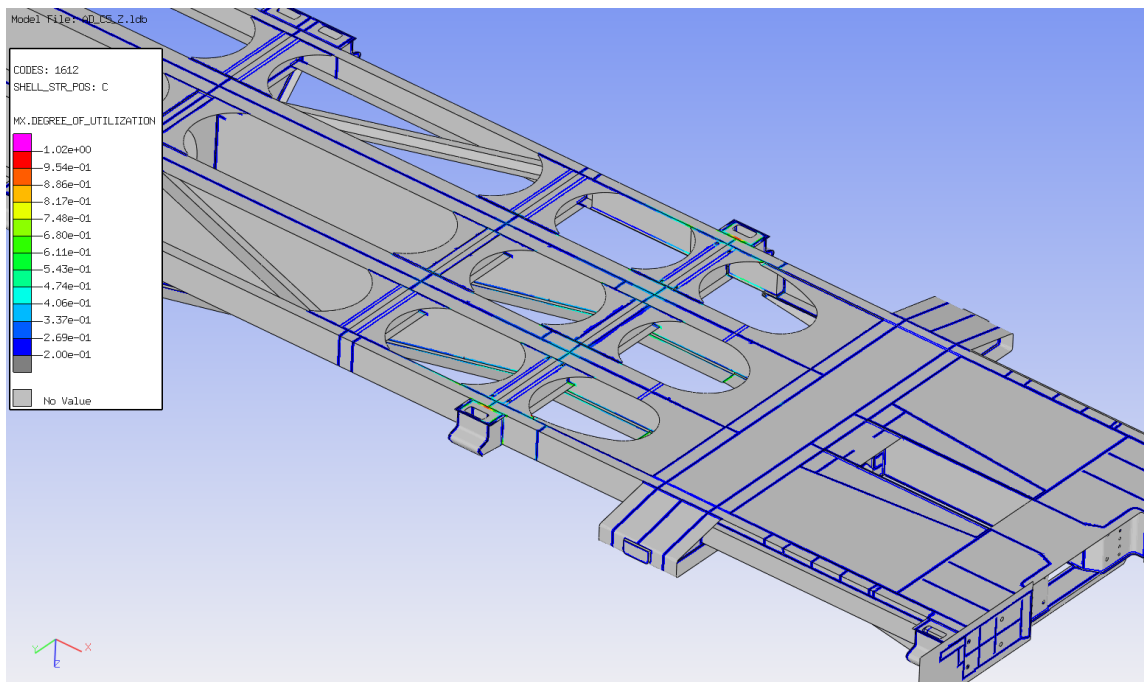


Figure A.30: Weld global fatigue analysis result-DVS1612-C5Z

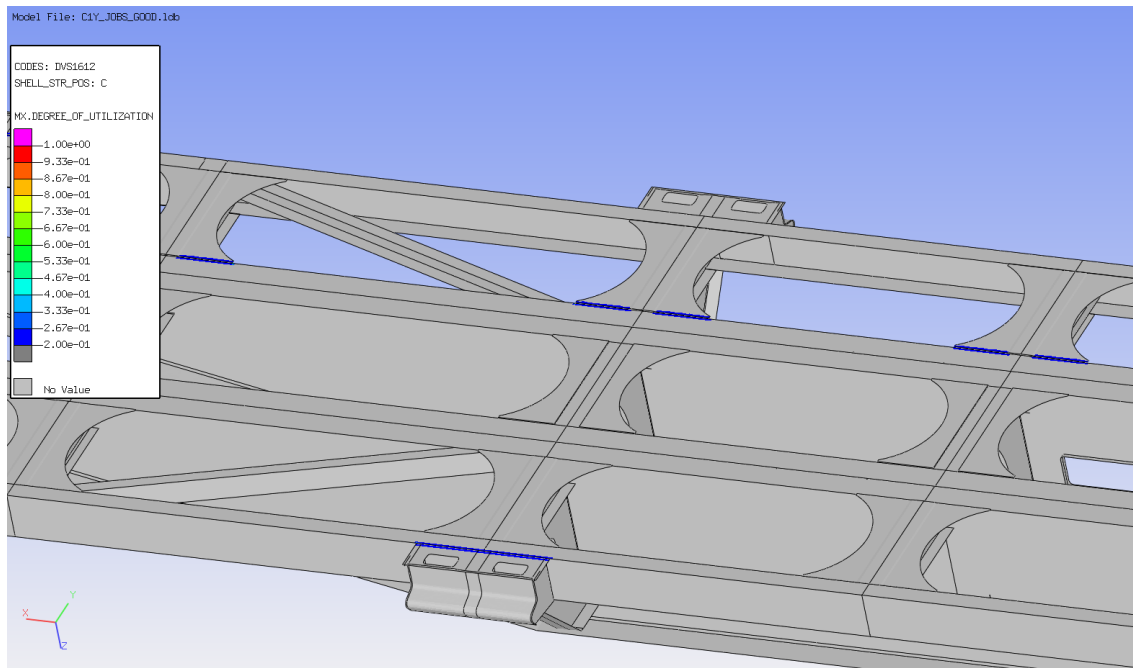


Figure A.31: Detail weld fatigue analysis result worst locations-DVS1612-C1Y

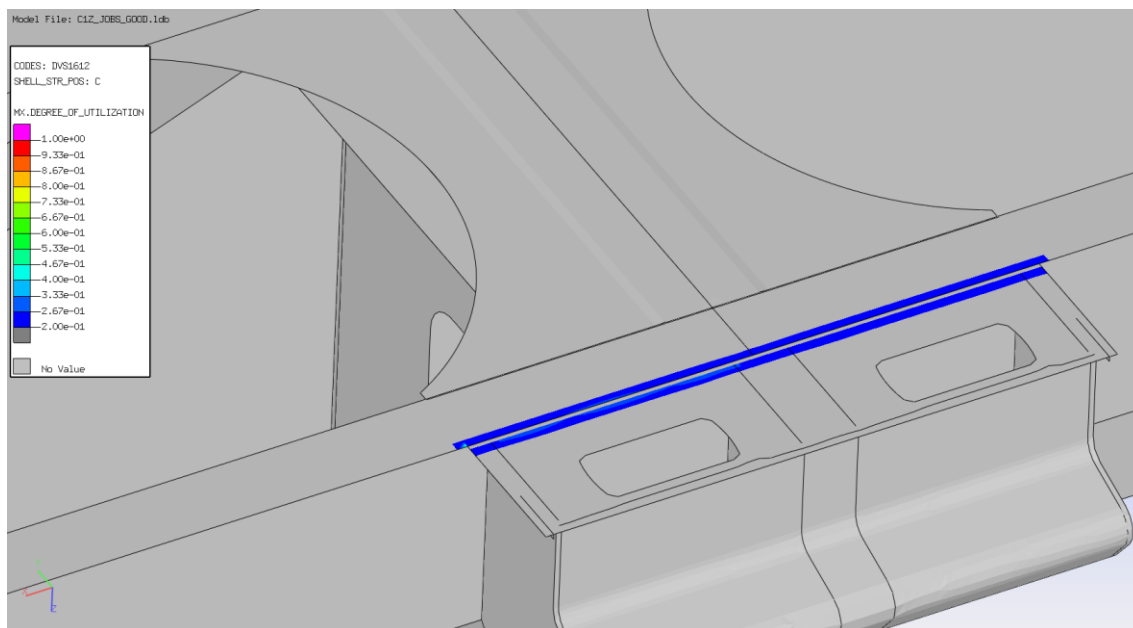


Figure A.32: Detail weld fatigue analysis result worst locations-DVS1612-C1Z

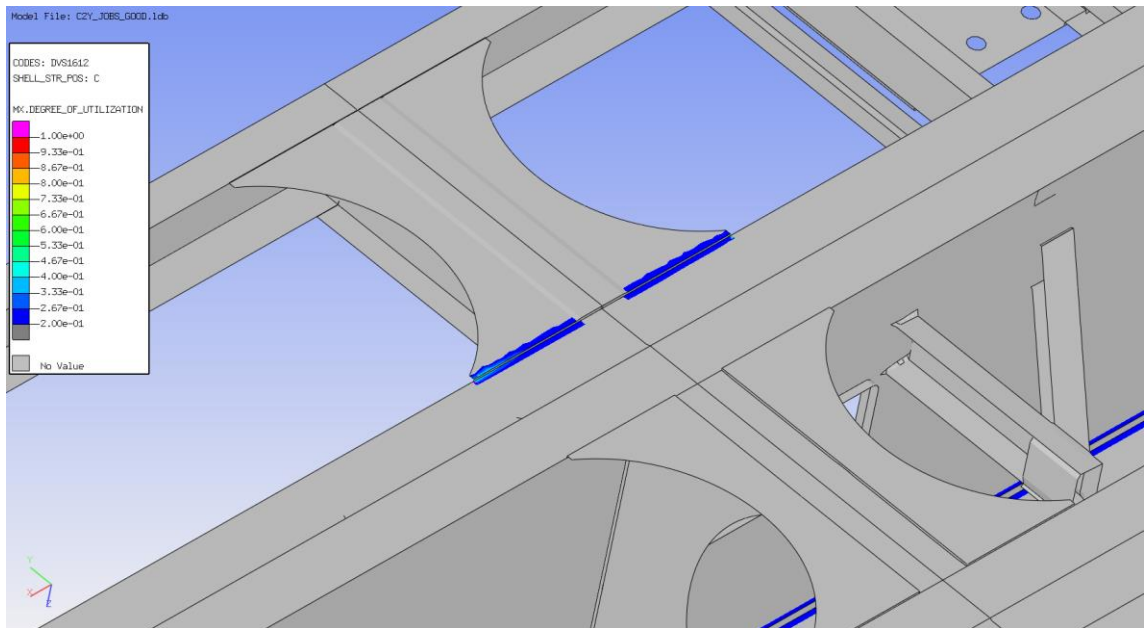


Figure A.33: Detail weld fatigue analysis result worst locations-DVS1612-C2Y

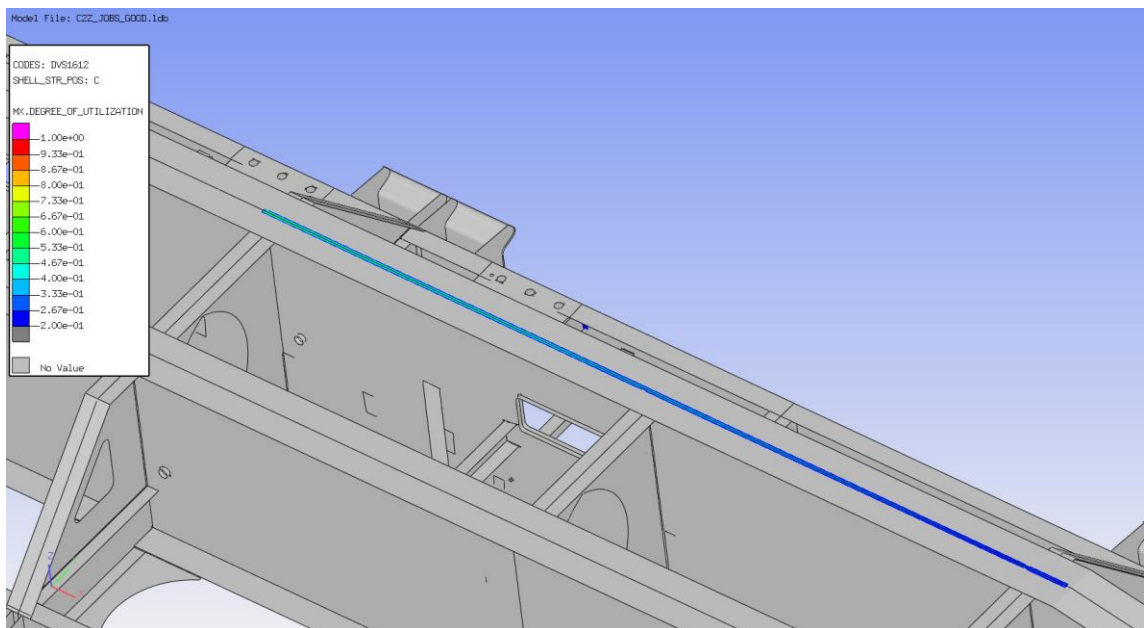


Figure A.34: Detail weld fatigue analysis result worst locations-DVS1612-C2Z

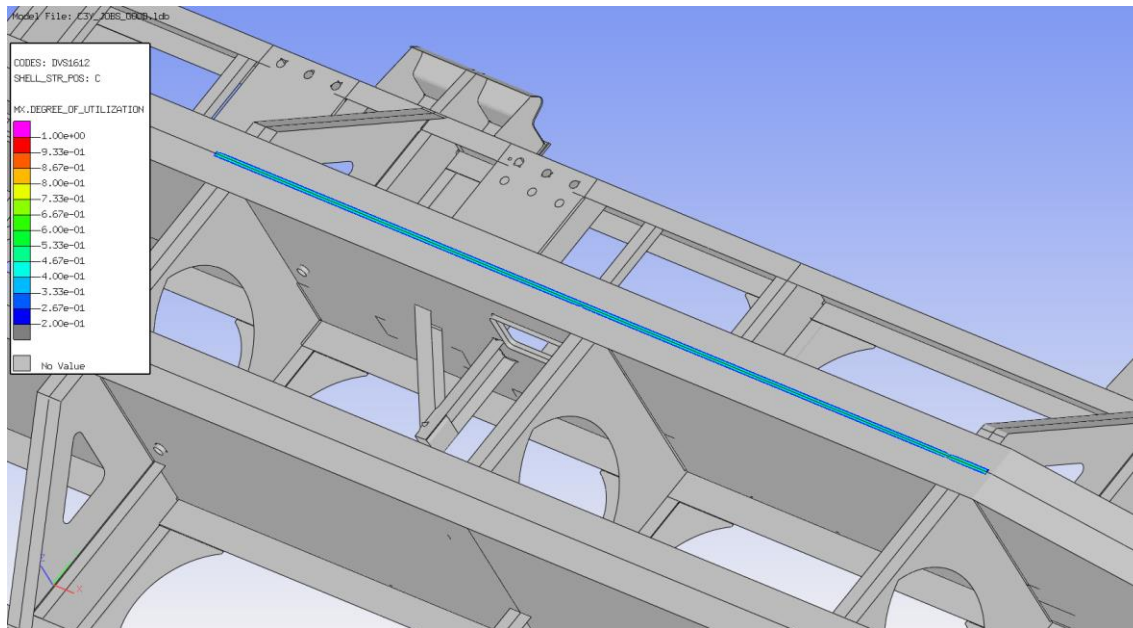


Figure A.35: Detail weld fatigue analysis result worst locations-DVS1612-C3Y

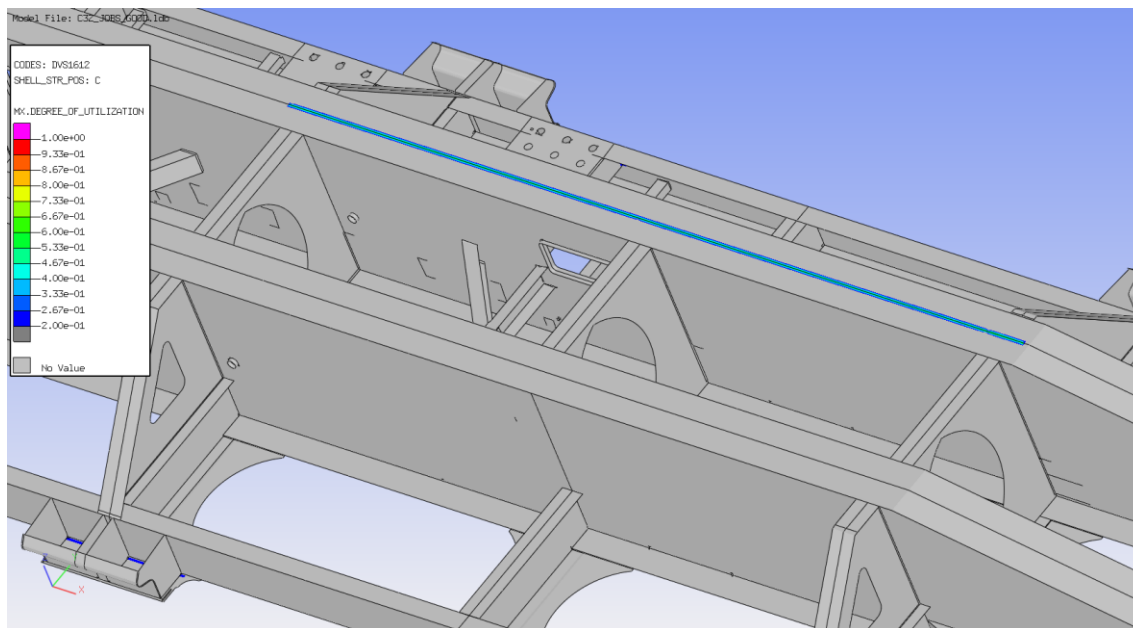


Figure A.36: Detail weld fatigue analysis result worst locations-DVS1612-C3Z

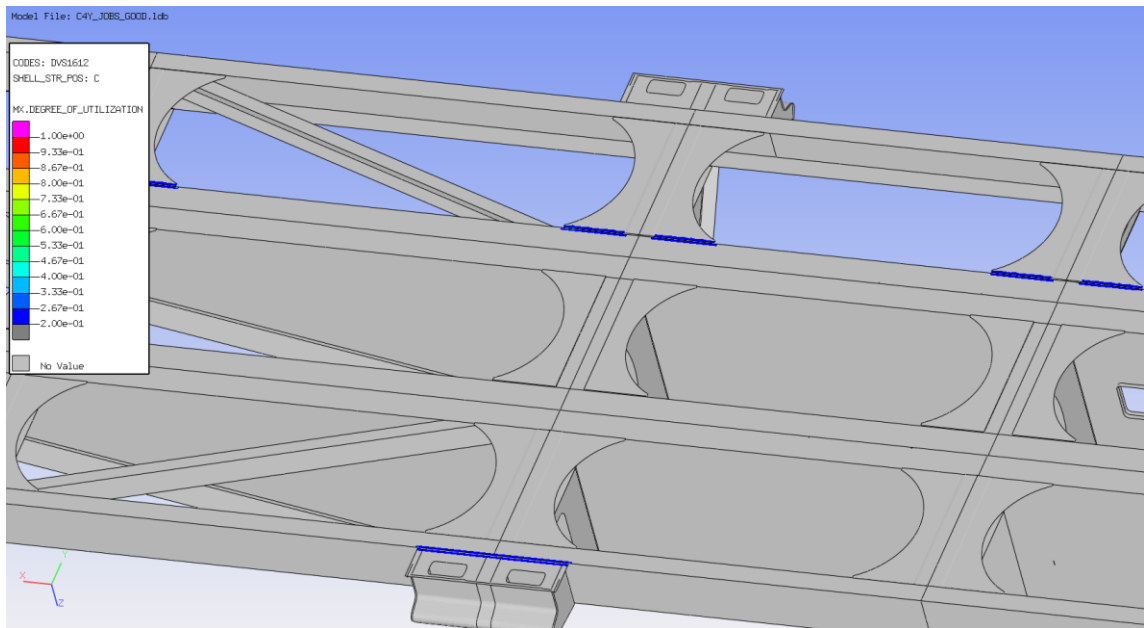


Figure A.37: Detail weld fatigue analysis result worst locations-DVS1612-C4Y

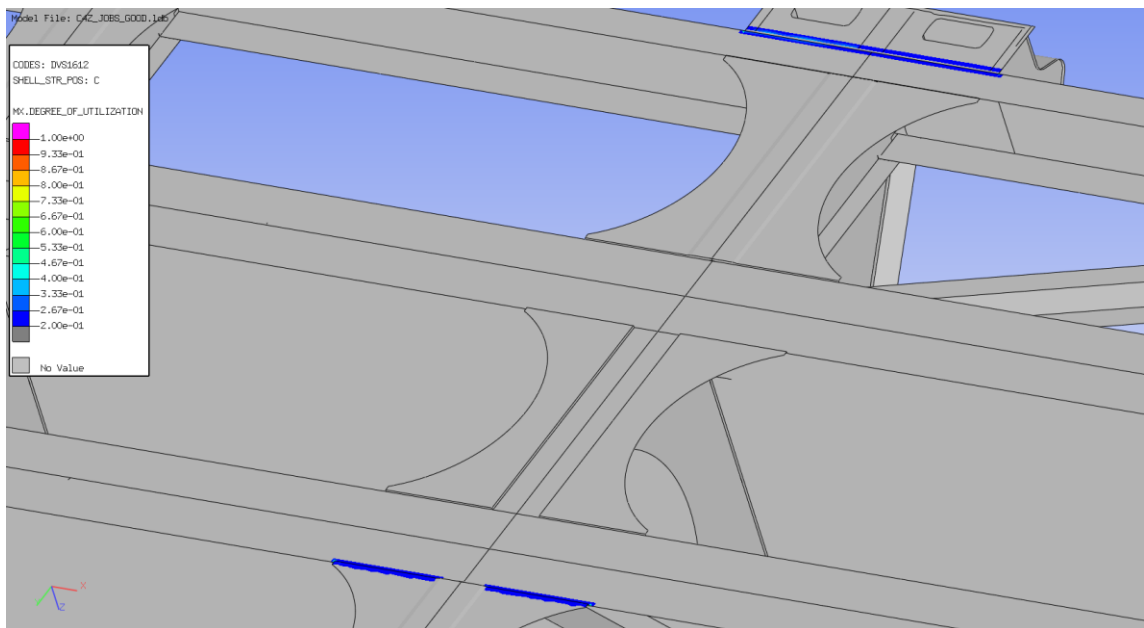


Figure A.38: Detail weld fatigue analysis result worst locations-DVS1612-C4Z

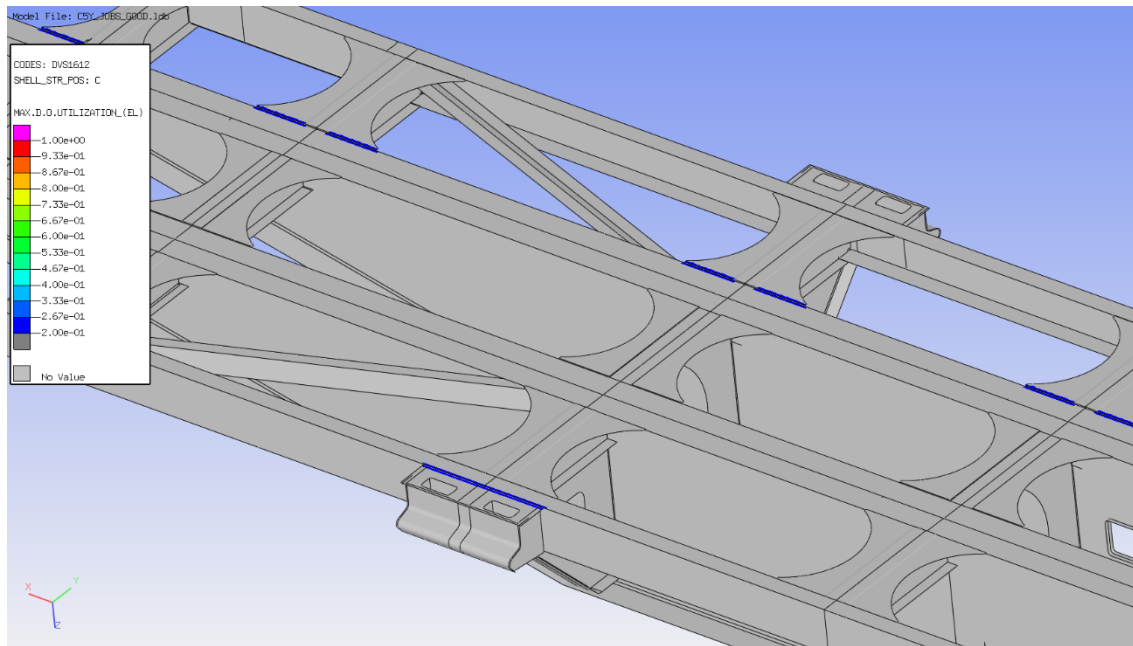


Figure A.39: Detail weld fatigue analysis result worst locations-DVS1612-C5Y

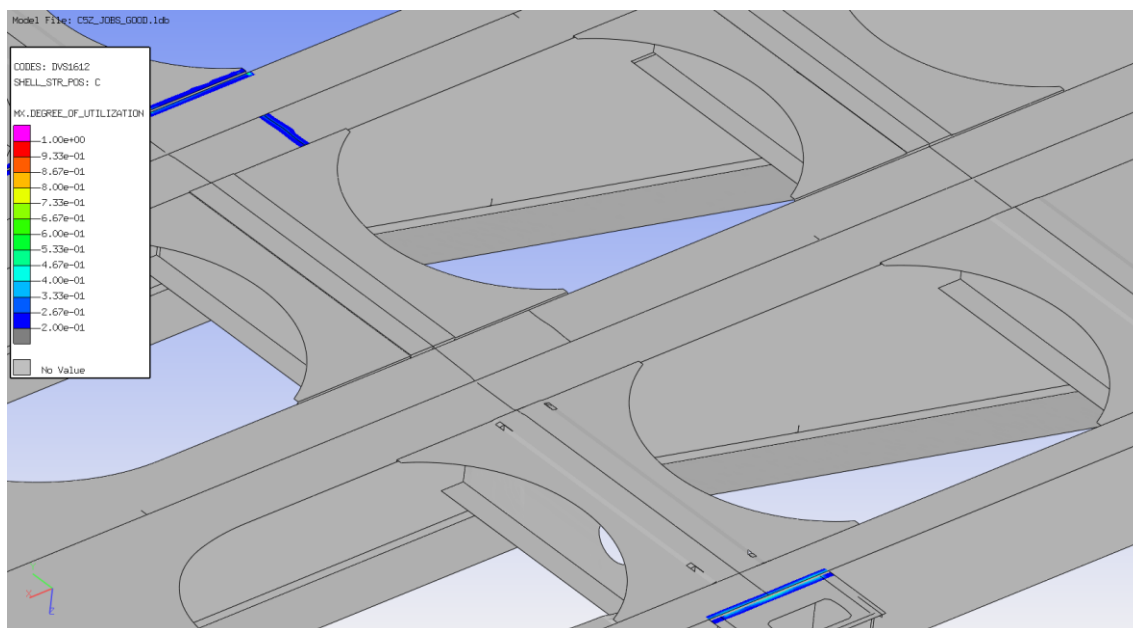


Figure A.40: Detail weld fatigue analysis result worst locations-DVS1612-C5Z

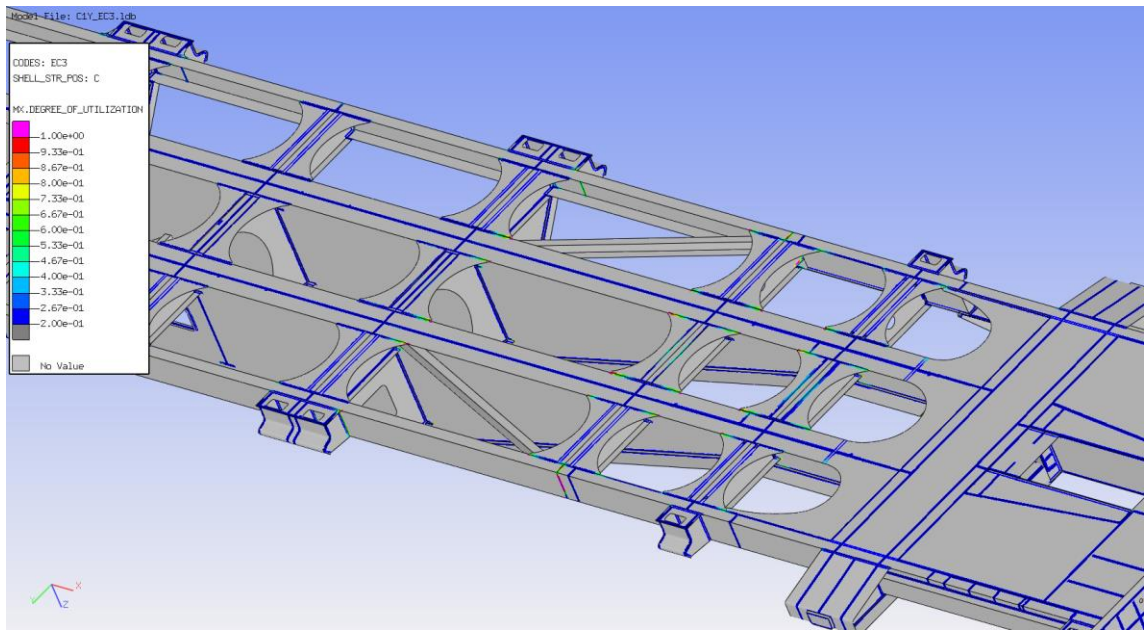


Figure A.41: Weld global fatigue analysis result-EC3-C1Y

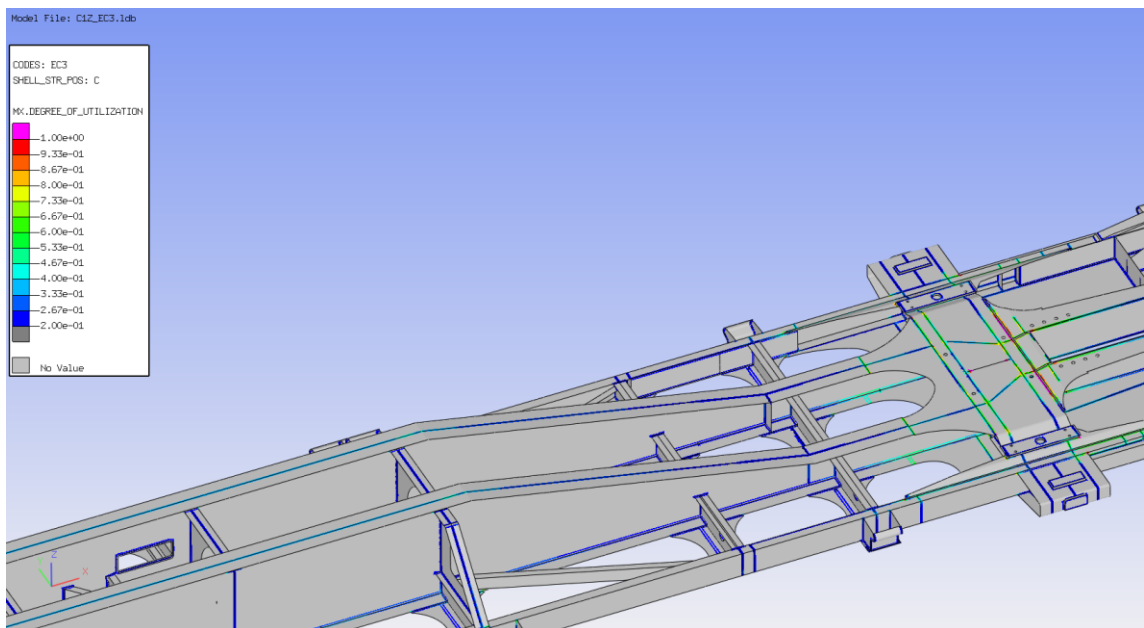


Figure A.42: Weld global fatigue analysis result-EC3-C1Z

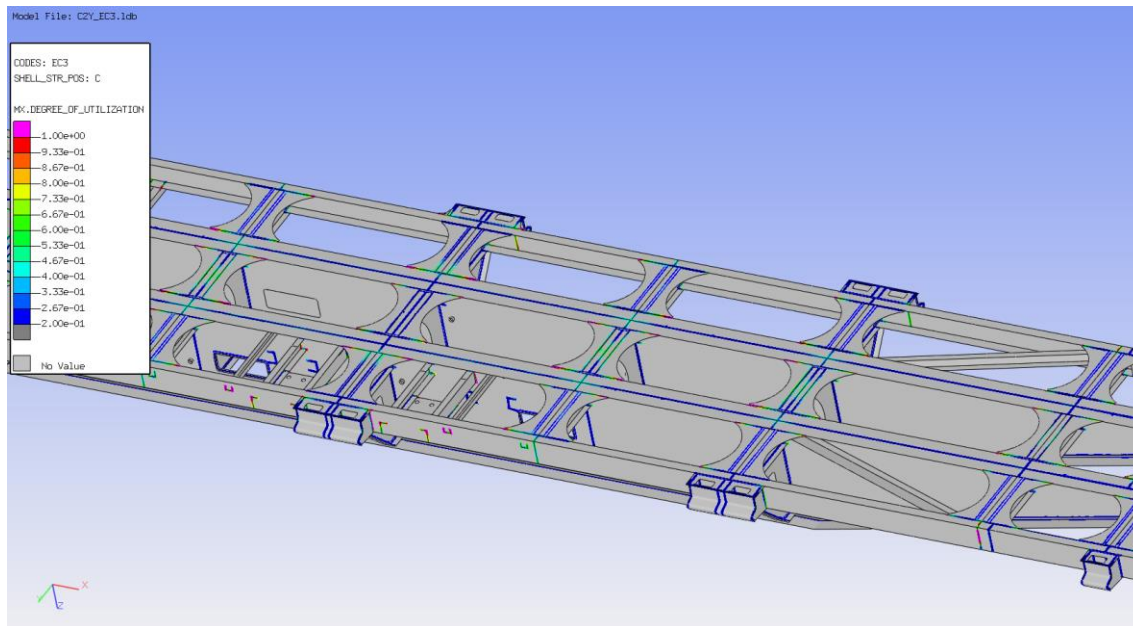


Figure A.43: Weld global fatigue analysis result-EC3-C2Y

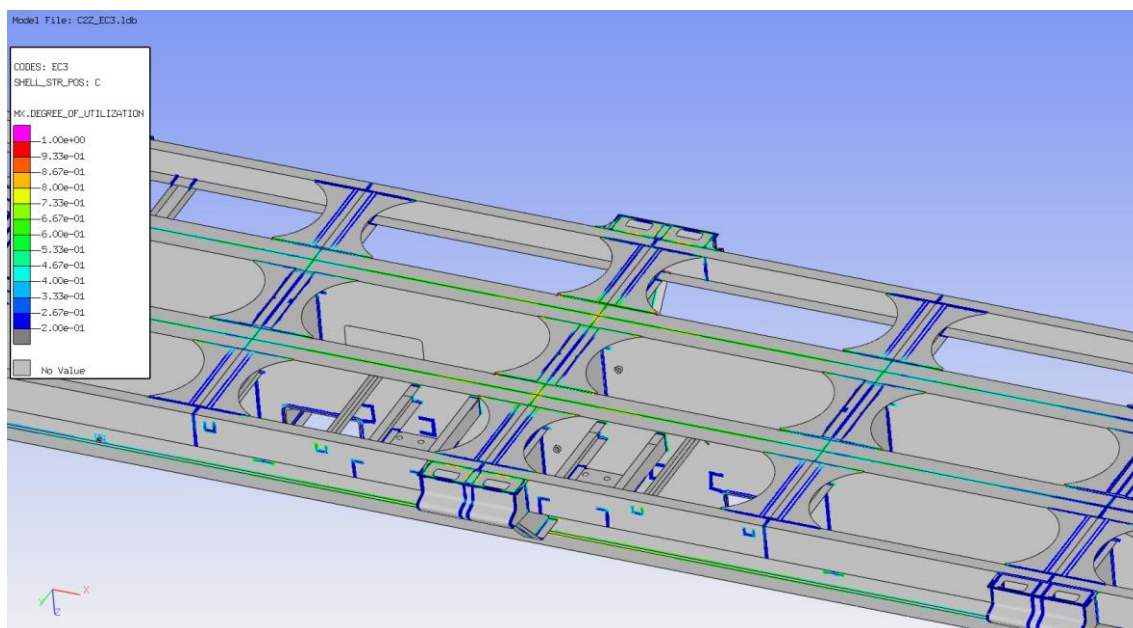


Figure A.44: Weld global fatigue analysis result-EC3-C2Z

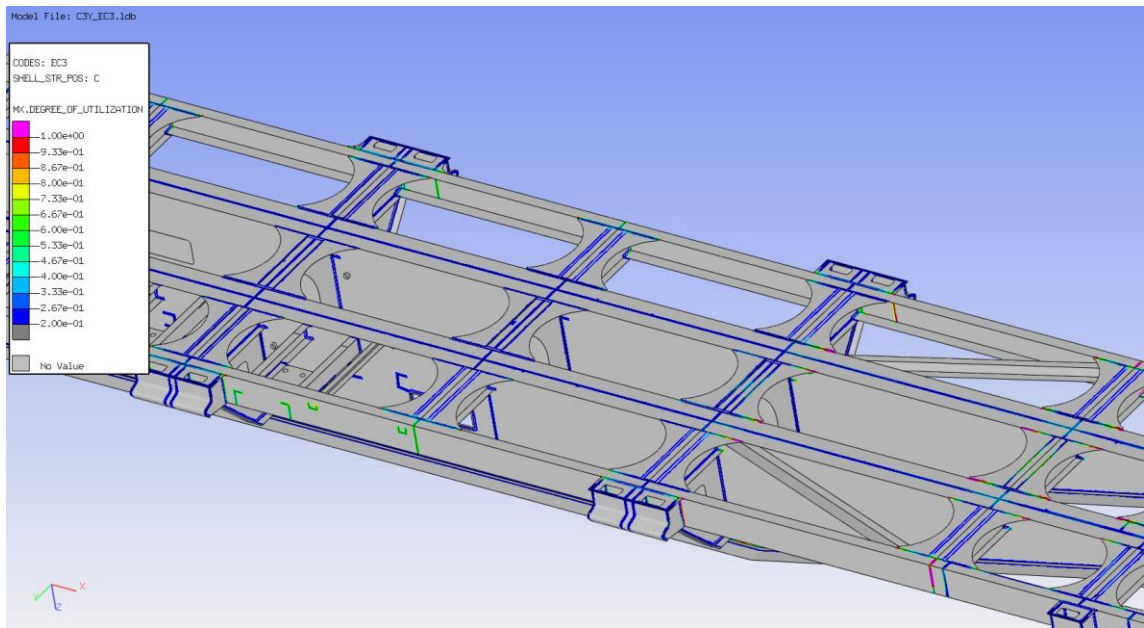


Figure A.45: Weld global fatigue analysis result-EC3-C3Y

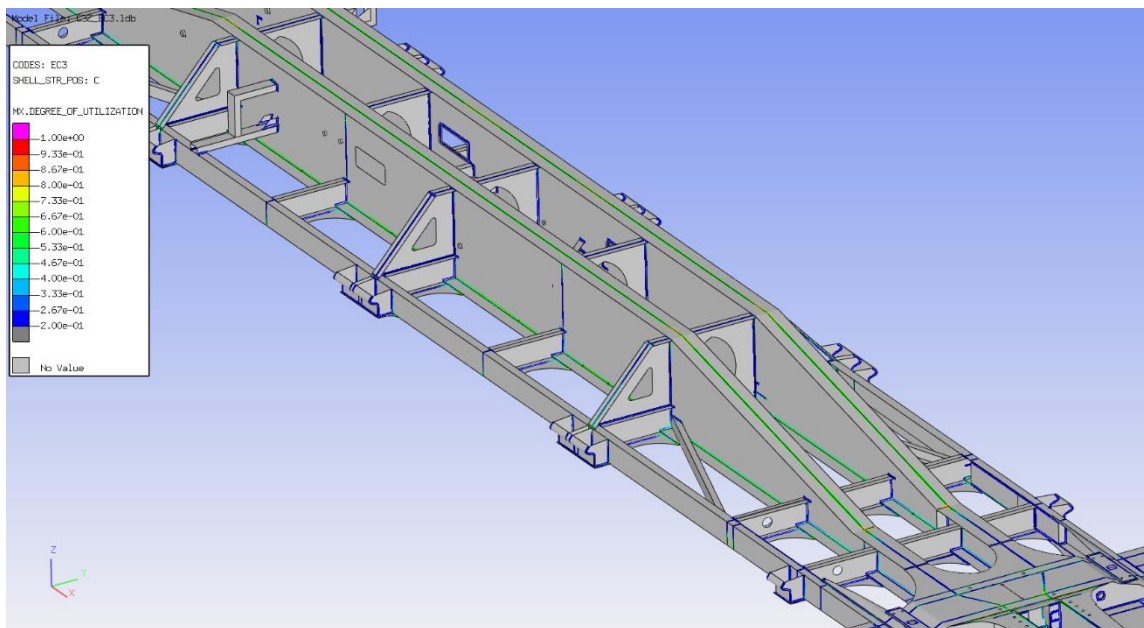


Figure A.46: Weld global fatigue analysis result-EC3-C3Z

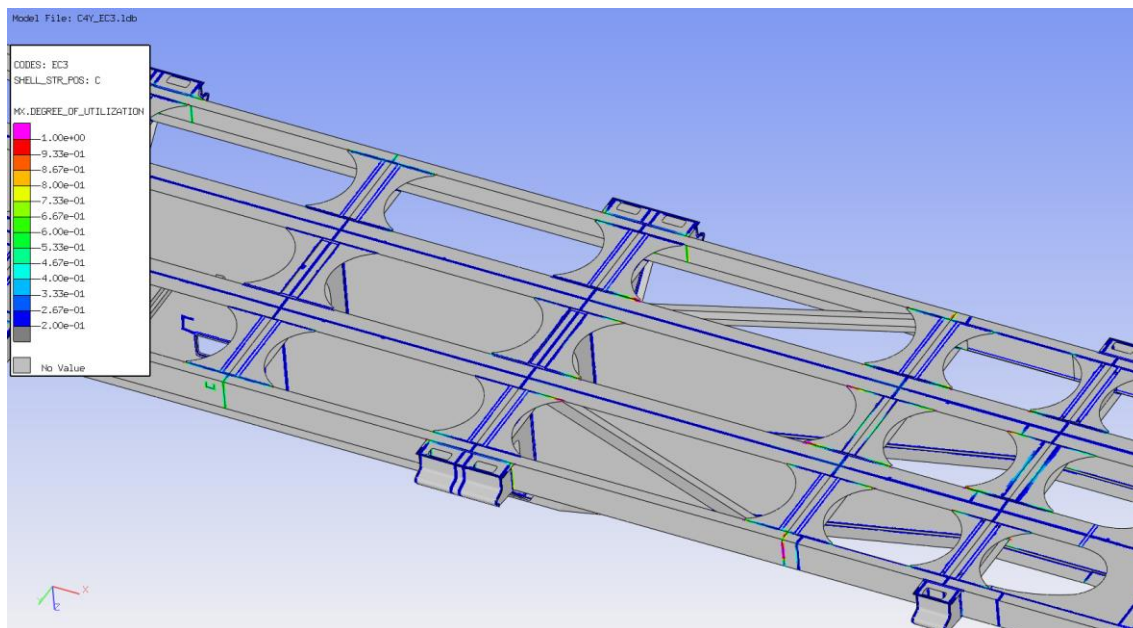


Figure A.47: Weld global fatigue analysis result-EC3-C4Y

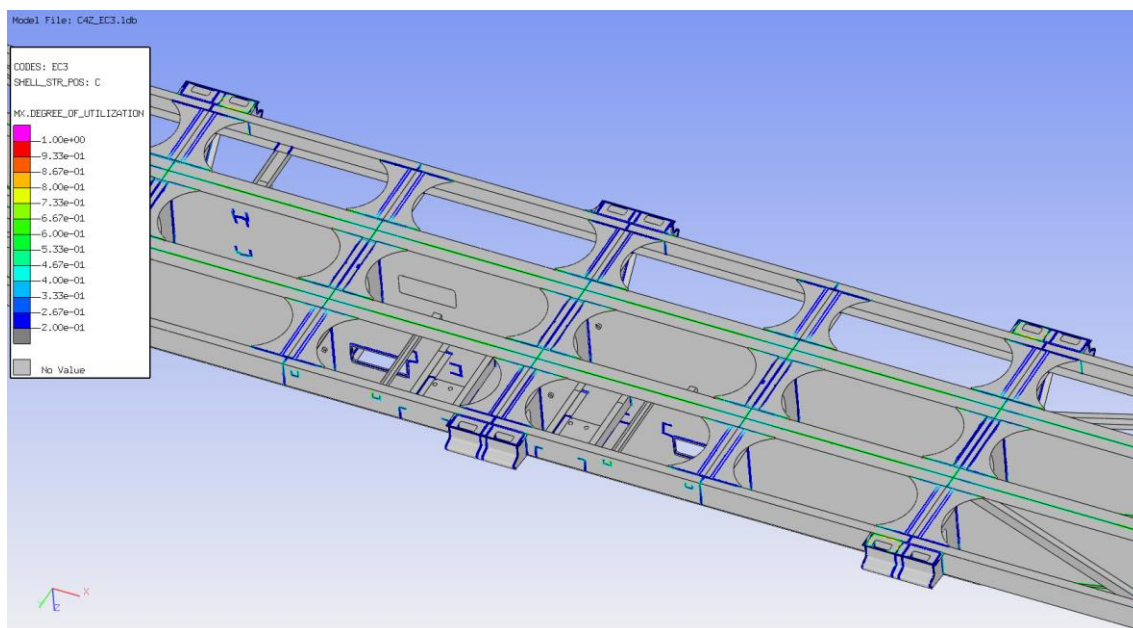


Figure A.48: Weld global fatigue analysis result-EC3-C4Z

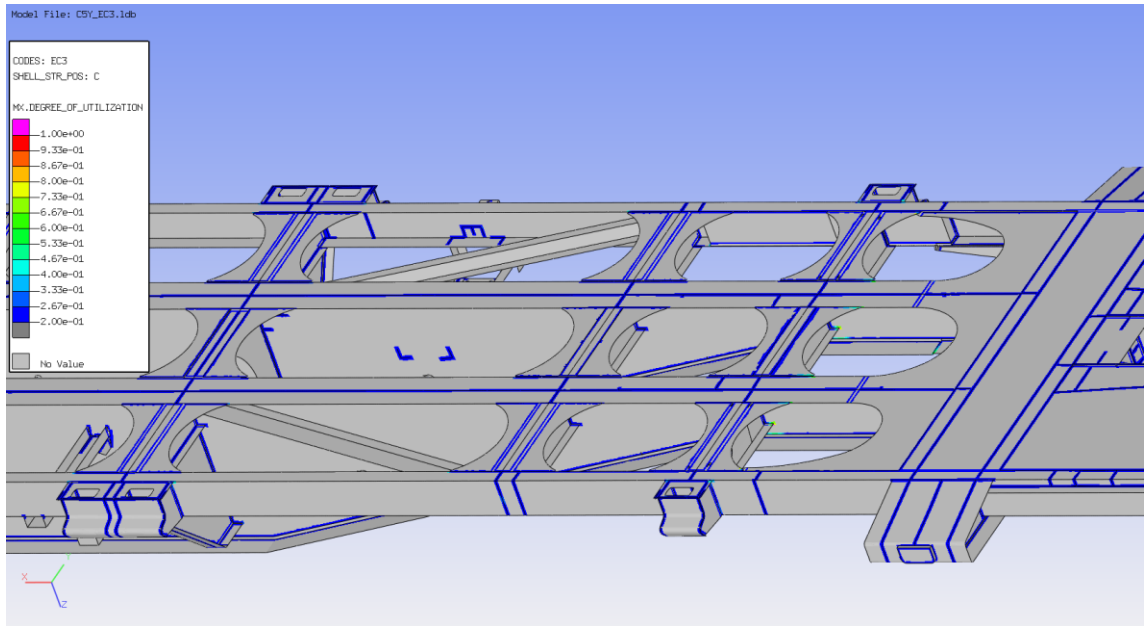


Figure A.49: Weld global fatigue analysis result-EC3-C5Y

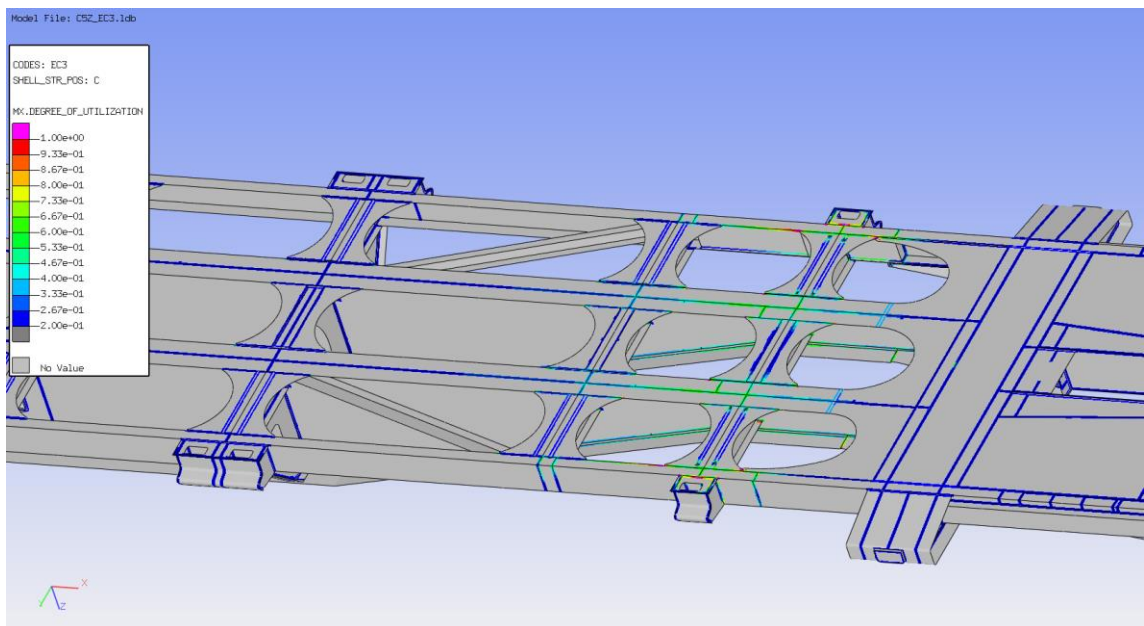


Figure A.50: Weld global fatigue analysis result-EC3-C5Z

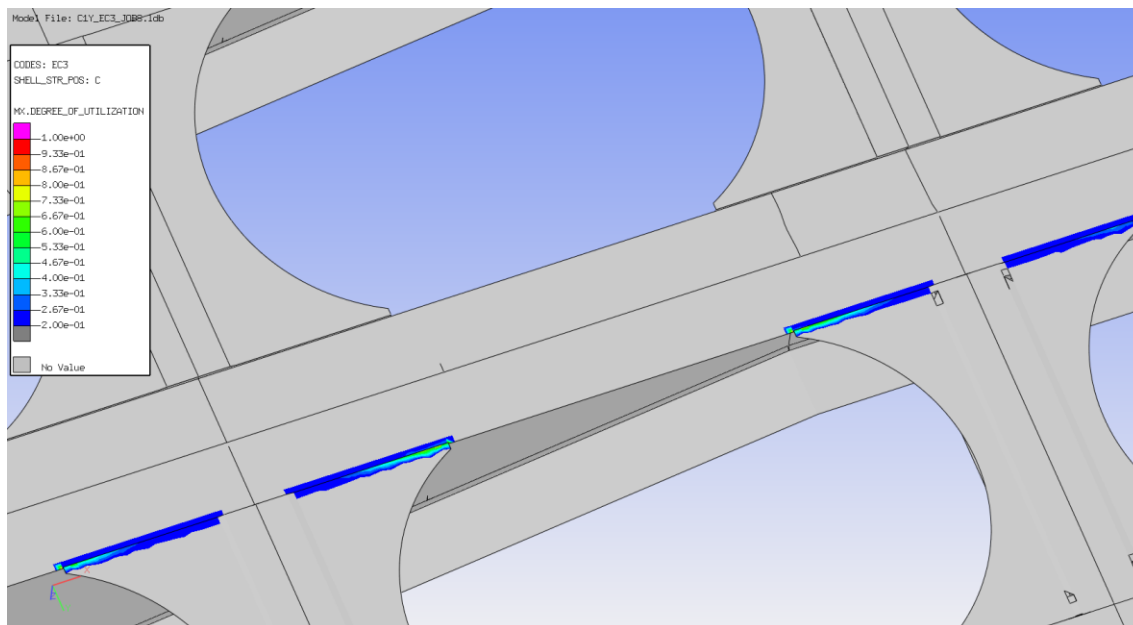


Figure A.51: Detail weld fatigue analysis result worst locations-EC3-C1Y

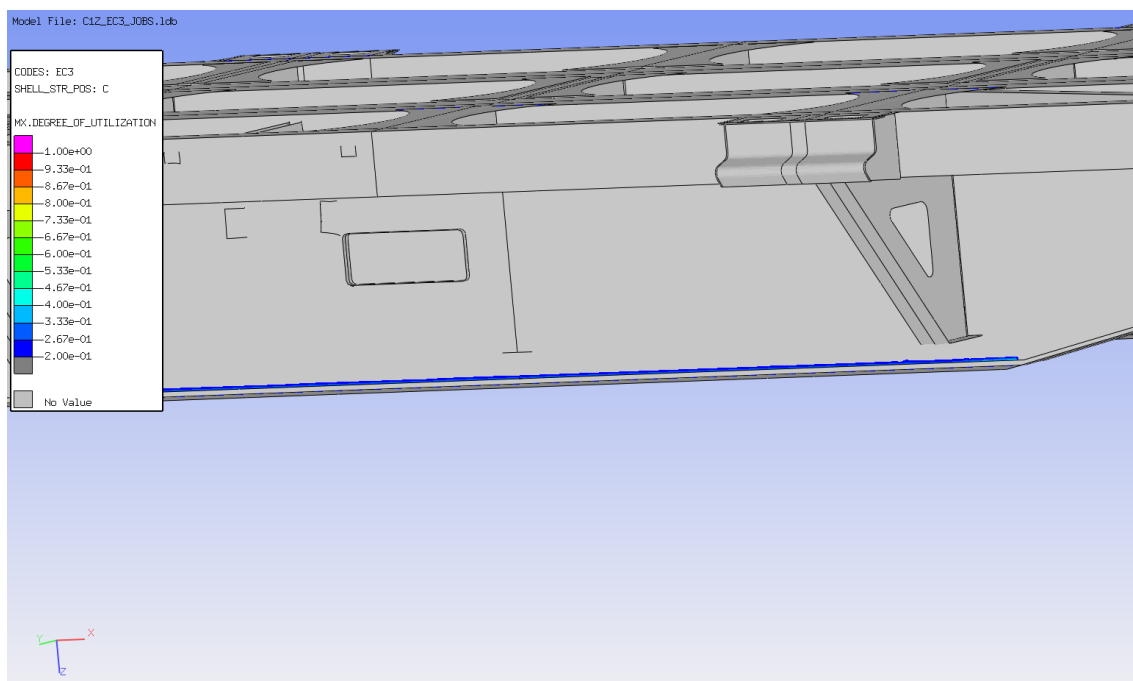


Figure A.52: Detail weld fatigue analysis result worst locations-EC3-C1Z

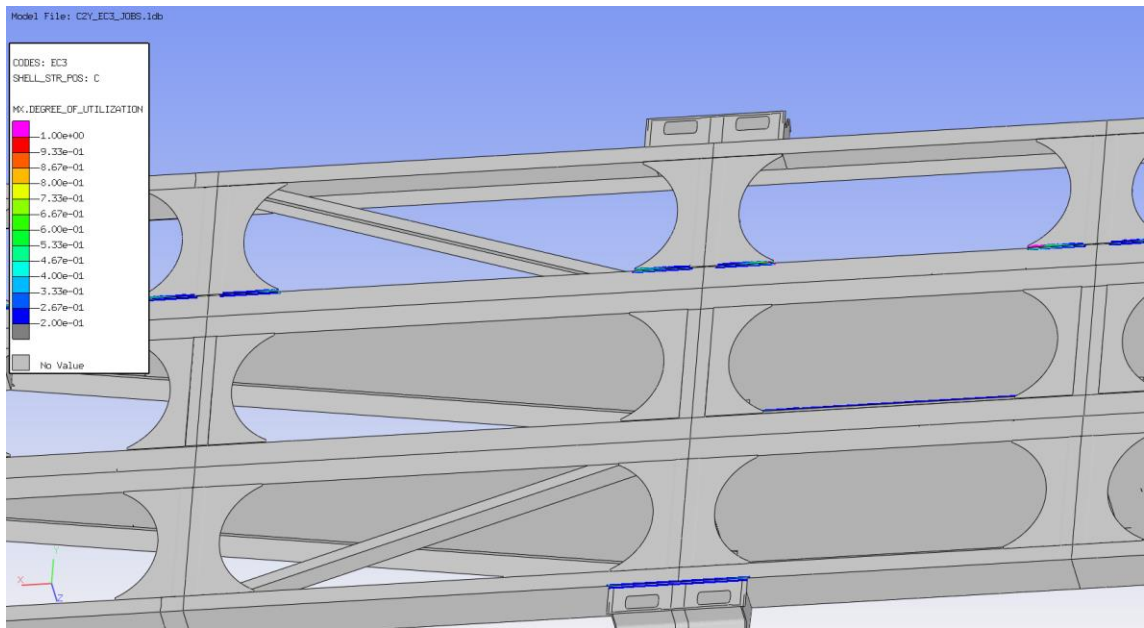


Figure A.53: Detail weld fatigue analysis result worst locations-EC3-C2Y

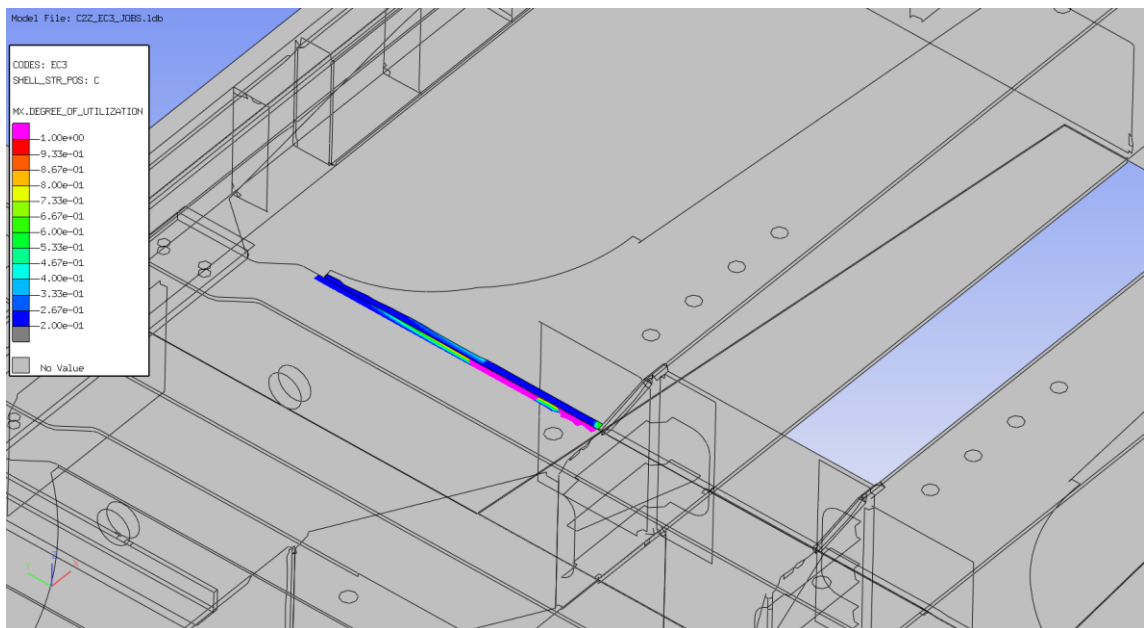


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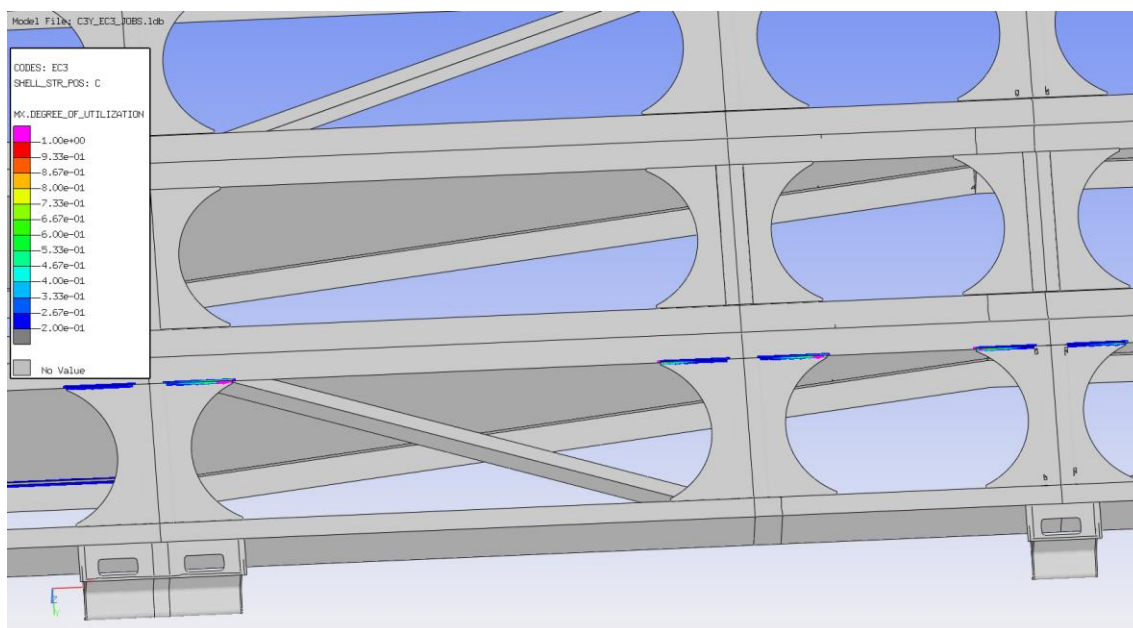


Figure A.55: Detail weld fatigue analysis result worst locations-EC3-C3Y

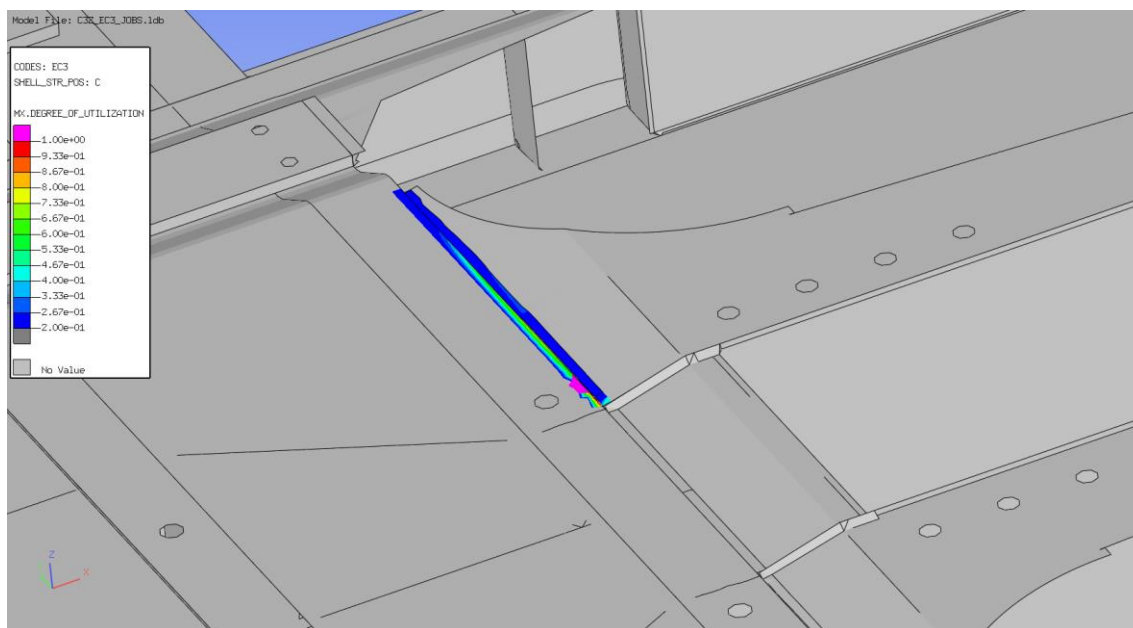


Figure A.56: Detail weld fatigue analysis result worst locations-EC3-C3Z

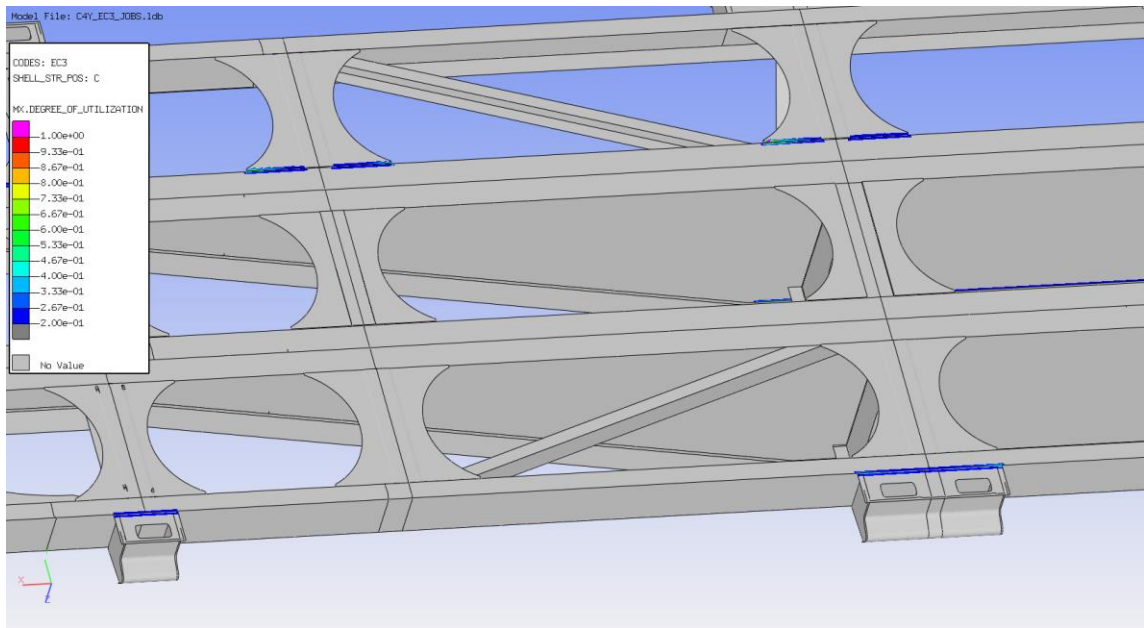


Figure A.57: Detail weld fatigue analysis result worst locations-EC3-C4Y

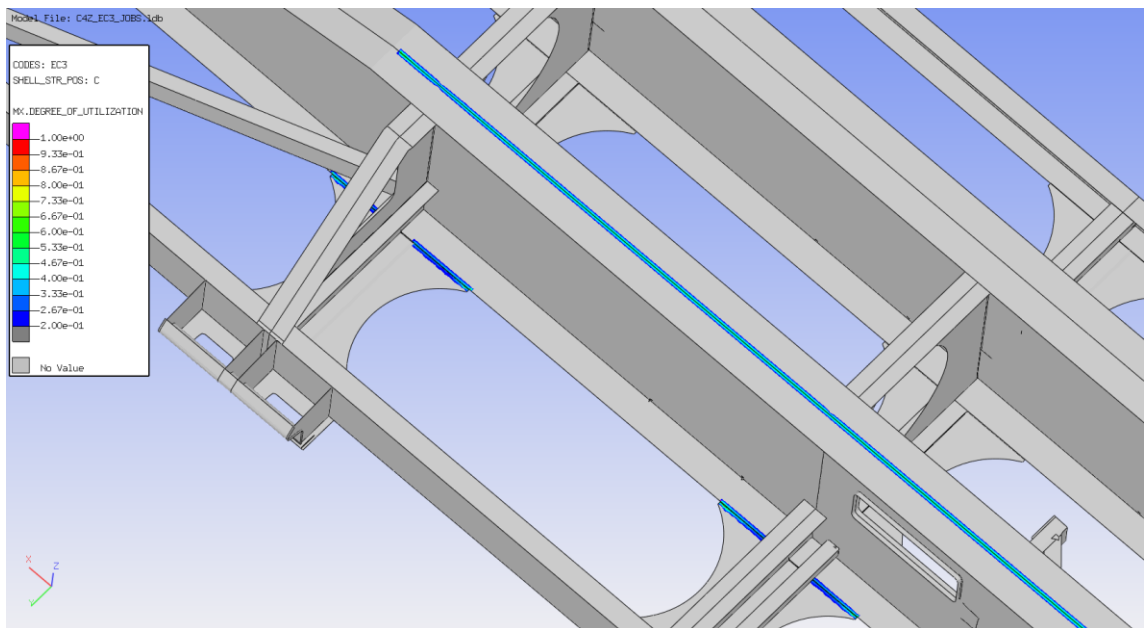


Figure A.58: Detail weld fatigue analysis result worst locations-EC3-C4Z

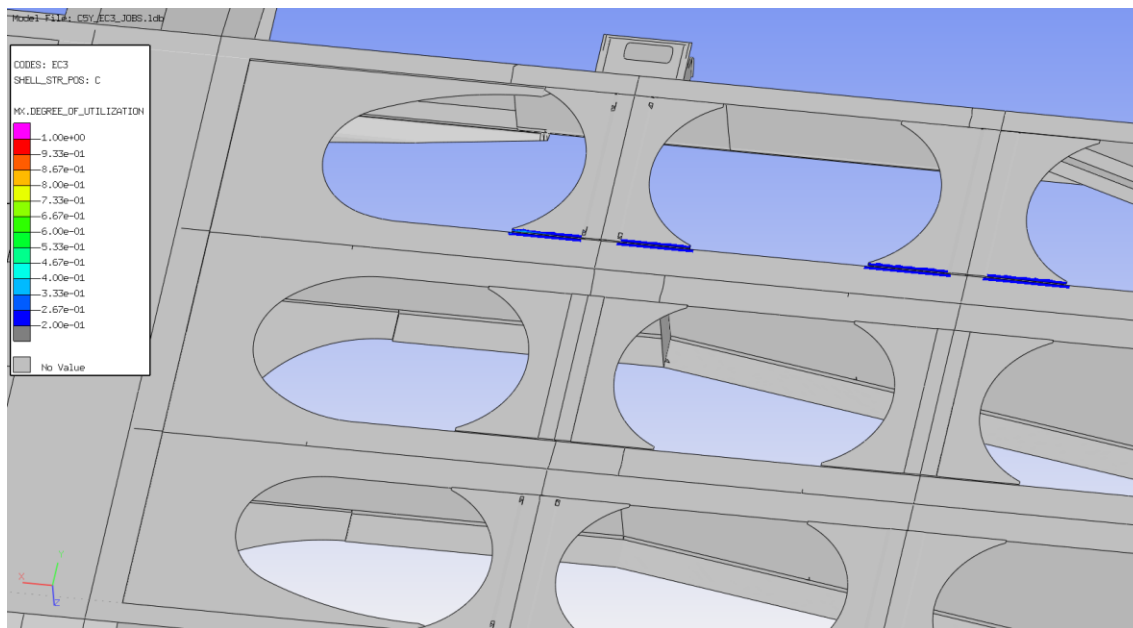


Figure A.59: Detail weld fatigue analysis result worst locations-EC3-C5Y

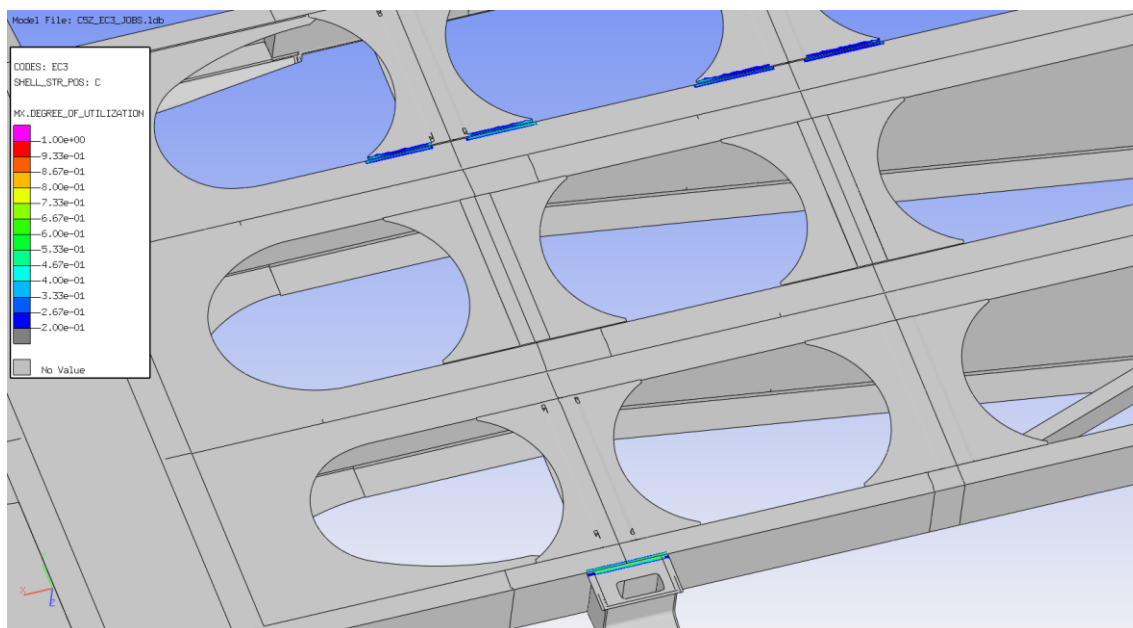


Figure A.60: Detail weld fatigue analysis result worst locations-EC3-C5Z

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- 2012-2014 Stress Analysis Project Assistant, Laboratory of Strength of Material's, Faculty of Mechanical Engineering, I.T.U.

LIST OF PUBLICATIONS

International Publications

Numerical Fatigue Evaluation Of A Full Scale Passenger Rolling Stock's Welded Joints According to FKM Guideline

Anıl ÖZDEŞ, Emin SÜNBÜLOĞLU, Ergün BOZDAĞ, Nikolaus FRIEDL

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The Numerical Investigation Of Running Safety and Fatigue Analysis Of a Freight Wagon

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National Conferences

Yolcu Vagonu Gövdesi Kaynaklı Bağlantılarında Yorulma Analizi

Anıl ÖZDEŞ, Emin SÜNBÜLOĞLU, Ergün BOZDAĞ

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