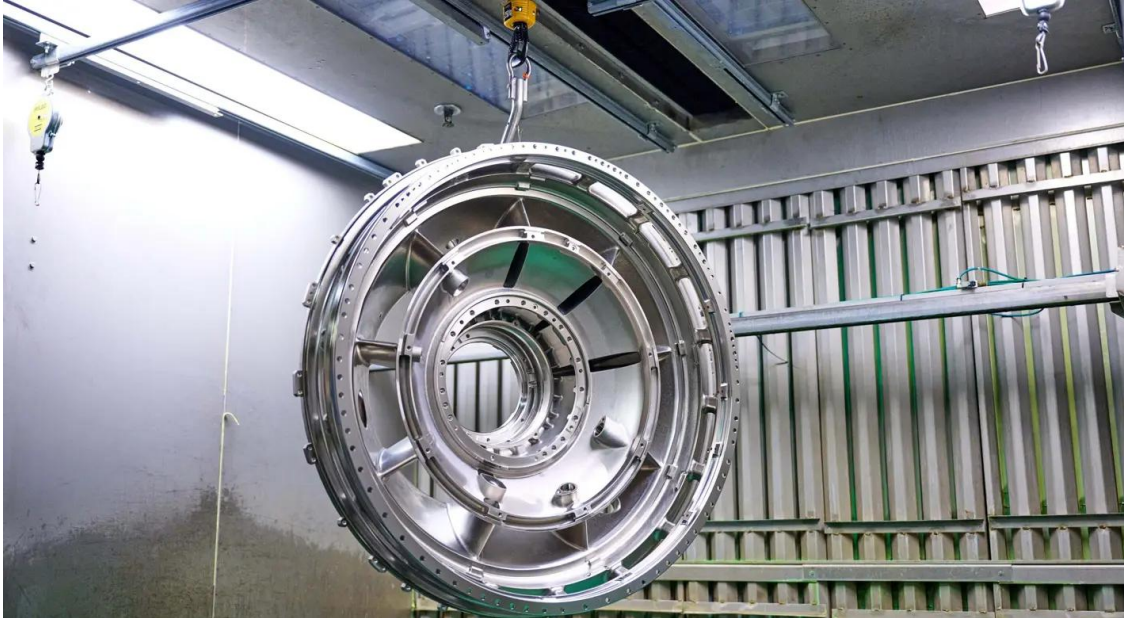




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Structural Evaluation of Defects in Bolt Hole Threads

Master's Degree Thesis Report

Master's thesis in MSc Product Development

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Gothenburg, Sweden 2026

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MASTER THESIS REPORT 2026

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Department of Industrial and Material Science

CHALMERS UNIVERSITY OF TECHNOLOGY
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Abstract

Threaded bolted joints are a critical fastening method in aerospace engineering, used to connect structural components under demanding mechanical and thermal conditions. In many of the assemblies manufactured at GKN Aerospace, bolts are installed directly into tapped holes machined into large, high-value components — without the use of nuts. This thesis focuses on a component called the Intermediate Case (IMC), produced from cast titanium alloy Ti-6Al-4V (Ti-64). The bolts are fabricated from Inconel 718 (IN-718), with a thread specification of .3125-24 UNJF-3A (Unified National Joint Fine). Because the threaded holes are integral to the parent components, any damage or defect renders replacement extremely difficult and costly.

Thread defects can arise at various stages of a component's lifecycle, including manufacturing, installation, maintenance, and handling. Currently, GKN Aerospace evaluates such defects using a conservative approach in which the defect is assumed to behave as a pre-existing crack. Crack propagation and fatigue life analyses are subsequently performed using linear elastic fracture mechanics. While this method ensures structural safety, it frequently leads to the rejection of components that may still be structurally serviceable, resulting in unnecessary material waste, increased costs, and extended maintenance downtime.

This thesis aims to develop a more realistic and less conservative methodology for evaluating defects in bolt hole threads. The approach involves explicitly incorporating defect geometry into finite element models rather than relying on simplified stress concentration factors and crack assumptions. Numerical simulations are carried out using software tools including Siemens NX and ANSYS.

The study encompasses a systematic classification of defects, followed by targeted simulation and evaluation for each defect. The findings are intended to support more informed maintenance decision-making at GKN Aerospace, helping to distinguish between components that require removal from service and those that can safely continue in operation.

The outcomes of this work are expected to contribute to improved safety assessment practices while reducing unnecessary conservatism in defect evaluation, thereby minimising operational costs and material waste in aerospace maintenance.

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ABDUL BASIT MOHAMED HABIBULLAH

Gothenburg, 2026

Acronyms

CAD = Computer-Aided Design

D1–D6 = Defect 1 to Defect 6

FEA = Finite Element Analysis

FEM = Finite Element Method

IMC = Intermediate Case

IN-625 = Inconel 625

IN-718 = Inconel 718

ISO = International Organization for Standardization

mm = Millimetre

MPa = Megapascal

N = Newton

Ref. = Reference model (without defect)

SC30 = Surface Crack 30

STEP = Standard for the Exchange of Product Data

Tet10 = Ten-node quadratic tetrahedral element

Ti-64 = Titanium alloy Ti-6Al-4V

UNJF = Unified National Joint Fine

List of Symbols

K_t = Stress concentration factor

E = Young's modulus

ν = Poisson's ratio

μ = Coefficient of friction

F_t = Tangential contact force

F_n = Normal contact force

σ = Normal stress

ε = Normal strain

σ_{xx} = Normal stress in the x-direction

σ_{yy} = Normal stress in the y-direction

σ_{zz} = Normal stress in the z-direction

τ_{xy} = Shear stress acting on the x-face in the y-direction

τ_{yz} = Shear stress acting on the y-face in the z-direction

τ_{xz} = Shear stress acting on the x-face in the z-direction

ε_{ij} = Strain tensor component in directions i and j; where i and j are coordinate directions

σ_{ij} = Stress tensor component in directions i and j; where i and j are coordinate directions

δ_{ij} = Kronecker delta

σ_{kk} = Trace of the stress tensor

σ_e = von Mises equivalent stress

σ_1 = First principal stress (maximum principal stress)

σ_2 = Second principal stress (intermediate principal stress)

σ_3 = Third principal stress (minimum principal stress)

S_y = Yield strength of the material

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1. Introduction

Structural Evaluation of Defects in Bolt Hole Threads

1.1 Background

Threaded bolted joints are one of the most fundamental fastening methods in aerospace engineering. They are used extensively to connect structural components that must sustain significant mechanical loads, high temperatures, and cyclic operational stresses throughout their service life. Despite their small physical size relative to the assemblies they connect, threaded fasteners are considered safety-critical elements — even minor damage to thread geometry can compromise joint integrity and, in the most severe cases, contribute to engine failure or catastrophic structural accidents.

At GKN Aerospace, several engine and structural assemblies rely on bolts installed directly into tapped holes machined into primary load-bearing components, without the use of nuts. The key components in these assemblies are described below and shown in Figure 1.1:

Intermediate Case (IMC): Manufactured from the titanium alloy Ti-6Al-4V (Ti-64), a material widely used in aerospace structures for its excellent strength-to-weight ratio and corrosion resistance.

Bracket: Manufactured from the nickel-based superalloy Inconel 625 (IN-625), chosen for its high strength, excellent fatigue resistance, and suitability for elevated-temperature service environments. Furthermore, the bracket can be manufactured from alternative materials, such as steel; however, Inconel 625 was selected for this thesis. It should be noted that the bracket is not the primary focus of this study.

Bolts: Fabricated from the nickel-based superalloy Inconel 718 (IN-718), a material noted for its exceptional tensile strength, creep resistance, and retention of mechanical properties at high temperatures. The bolts conform to a thread specification of .3125-24 UNJF-3A, a Unified National Joint Fine (UNJF) thread form that provides high fatigue resistance and precise dimensional control, as required for aerospace fastening applications.

Because these threaded holes are integral features of large, high-value components, any damage or defect renders repair or replacement extremely difficult and costly. Thread defects can arise at multiple stages of a component's lifecycle, including during manufacturing (e.g. incorrect machining or tap geometry), installation (e.g. improper bolt torque or cross-threading), and routine maintenance operations (e.g. repeated assembly and disassembly cycles).

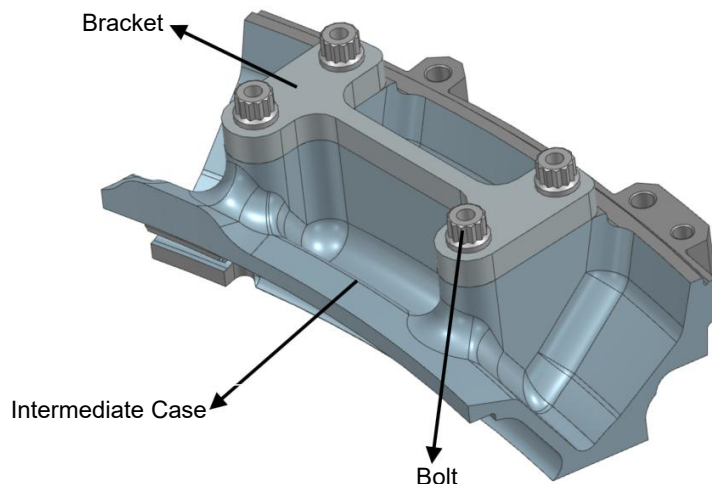


Figure 1.1. Detailed view of the assembly section (IMC, bracket, and bolts).

1.2 Problem Statement

The current approach adopted by GKN Aerospace for evaluating bolt hole thread defects is rooted in a conservative set of fracture mechanics assumptions. In this method, the observed defect — regardless of its actual geometry or severity — is assumed to have already developed into a crack-like discontinuity, and subsequent analyses are conducted on this basis. Representative examples of the types of thread damage encountered in practice are shown in Figure 1.2 [1].

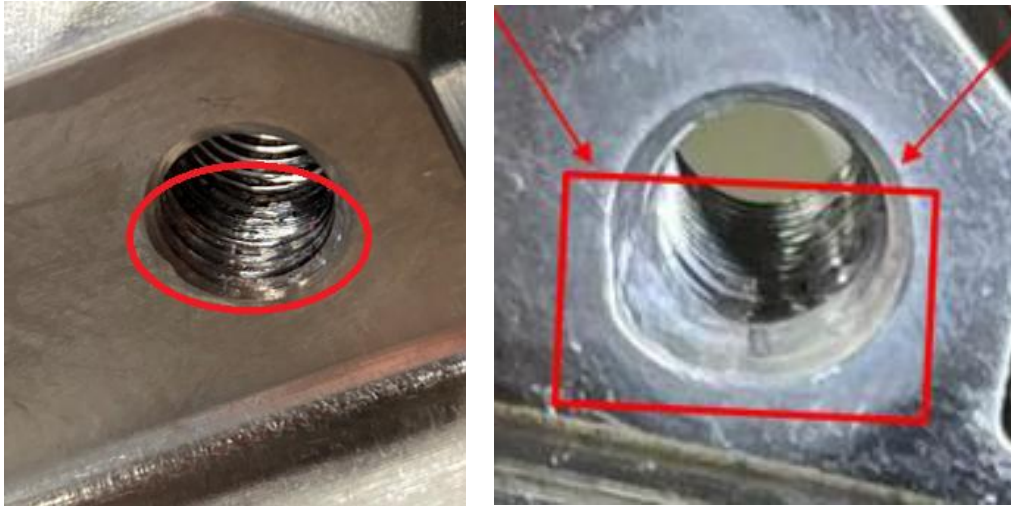


Figure 1.2. Examples of damaged bolt hole threads observed at GKN Aerospace.

The current evaluation procedure follows a series of clearly defined steps. First, a linear elastic finite element analysis (FEA) is conducted to determine the von Mises stress distribution in the vicinity of the defect, without explicitly modelling the defect geometry in the simulation. A stress concentration factor (K_t) is then applied to estimate the localised peak stress at the defect site.

Since linear elastic analysis cannot account for local material yielding and characteristically over-predicts peak stresses, the Neuber correction is subsequently applied to obtain a more realistic estimate of the elastic-plastic stress state. The corrected stress obtained through this method is then compared against the allowable stress limit, defined by the applicable plastic strain criterion, to confirm that an adequate safety margin is maintained.

While this methodology reliably ensures structural safety, it introduces significant and unnecessary conservatism into the evaluation process. The assumption that all defects behave as cracks, together with the use of linear elastic analysis without explicit defect modelling, might lead to the rejection of components that may in reality remain fully serviceable. This results in unnecessary material waste, increased operational costs, and extended maintenance downtime. There is therefore a clear engineering need for a more physically realistic and efficient defect evaluation approach.

1.3 Aim of the Thesis

The overarching aim of this thesis is to develop a physically realistic and less conservative methodology for evaluating structural defects in bolt hole threads within aerospace components. This involves moving away from simplified crack-assumption-based approaches — relying on K_t factors and the Neuber correction applied to undamaged models — and towards methods that explicitly represent the defect geometry within the finite element model, thereby capturing the actual stress state and fatigue behaviour with greater accuracy.

The thesis further aims to investigate the applicability of different numerical analysis tools — including ANSYS for FEA and NASGRO for fracture mechanics and fatigue life prediction — to a range of defect types and geometries. By systematically comparing these approaches, the work seeks to identify the most suitable evaluation method for each class of defect, balancing predictive accuracy against practical engineering applicability.

Ultimately, this thesis aims to deliver improved engineering guidance to GKN Aerospace for the assessment of bolt hole thread defects, supporting better-informed maintenance decisions and contributing to reductions in component scrappage and associated costs.

1.4 Research Questions

The following research questions guide the scope and direction of this thesis:

1. What types of geometric defects occur in bolt hole threads within the GKN Aerospace components, and how can they be systematically classified for finite element evaluation?
2. How does explicitly modelling defect geometry in a finite element model affect the predicted von Mises stress distribution compared to the defect-free reference case, and what does this reveal about the structural acceptability of each defect?
3. Which stress metric — linear elastic von Mises stress, nonlinear plastic stress, or equivalent plastic strain — provides the most reliable and consistent basis for comparing defect severity in threaded bolt hole simulations, and why?

1.5 Scope and Limitations

This thesis focuses specifically on structural defects in bolt hole threads within the assembly context described above, comprising the Ti-64 IMC, IN-625 bracket, and IN-718 bolts with thread specification .3125-24 UNJF-3A. The evaluation looks at how the defects affect the structural strength of the joint, using FEA with the threads modelled explicitly.

The thesis does not examine strategies for repairing or reconditioning threads following a defect assessment, nor does it consider the effect of manufacturing tolerances on nominal thread geometry beyond their role in defect formation. The scope is confined to the numerical and analytical evaluation of identified defect types and does not extend to experimental testing or physical specimen preparation unless such data is made available through GKN Aerospace.

2. Literature Review and Defect Overview

2.1 Finite Element Analysis — A Brief Literature Review

The Finite Element Method is a well-established numerical technique used to analyse engineering problems involving stress, deformation, and heat transfer. The basic idea behind the method is to divide a complex structure into a large number of small, simple pieces called elements. The behaviour of each element is described mathematically, and all the elements are then connected together to represent the full structure. By solving the equations for all the elements at once, it is possible to calculate how the structure responds to loads and boundary conditions. Because of its flexibility, FEM can be applied to almost any shape or loading situation, which has made it one of the most widely used tools in mechanical and structural engineering [2, 3].

FEM has been used extensively to study bolted and threaded connections. One of the main questions that researchers have tried to answer is how the load from a bolt is shared between the different thread turns in a joint. Studies have shown that the load is not shared equally — the first few engaged threads tend to carry a much larger portion of the total load compared to the threads further along [4]. This uneven load distribution means that certain parts of the thread are under much higher stress than others, which is important when trying to understand where a threaded joint is most likely to fail.

Over the years, the level of detail in FEM models of threaded connections has increased considerably. Early models were often two-dimensional or used simplified geometries that did not fully capture the shape of the thread. As computers became more powerful, it became possible to build full three-dimensional models that represent the helical geometry of the thread more accurately [5]. These more detailed models give better results, especially when studying localised effects such as stress concentrations at the thread root or the influence of contact between the bolt and the tapped hole.

The contact between the bolt thread and the hole thread is an important part of modelling a threaded joint. When a bolt is tightened, the surfaces press against each other and the pressure between them changes depending on the load and geometry. Several studies have looked at how this contact behaviour affects the overall stress in the joint and have shown that it needs to be modelled carefully to get accurate results [6]. Software tools such as ANSYS Mechanical, which is used in this thesis, include dedicated contact modelling. Literature on defects in threads was searched for online, but no reliable literature was found.

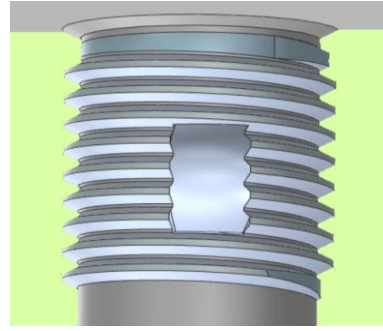
2.2 Overview of Defects Analysed

This thesis investigates six different types of defects that can occur in bolt hole threads. These defects were selected because they represent common problems found in manufacturing and during the service life of a threaded connection. Each defect was modelled using computer-aided design (CAD) and then used as the basis for FEA. Below is a brief description of each defect along with images showing how it looks in reality and how it was represented in the CAD model.

Defect 1 - Missing Piece of Thread Material: In this defect, a small section of the thread inside the hole has broken away, leaving a visible gap. As shown in the photograph, one localised area of the thread is damaged and a piece of material is no longer present. This kind of damage can occur during machining, assembly, or due to an impact on the part. In the CAD model, this is represented as a rectangular block of material removed from one section of the thread turns.



Actual defect



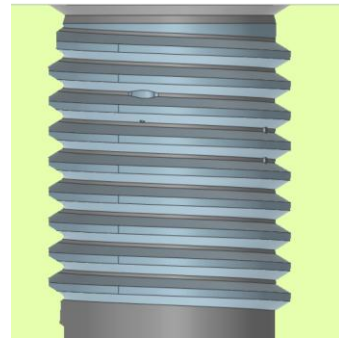
CAD representation

Figure 3.1 – Defect 1: actual photograph (left) and CAD model (right).

Defect 2 - Breaks on the Thread Flanks: This defect involves visible cracks on the surface of the thread. As seen in the photograph, there are two areas where breaks are present in the thread face. These breaks sit on the side surfaces of the thread and change how contact is made between the bolt and the hole. In the CAD model, this is represented as a small U shaped slot cut into the surface of one of the thread flanks.



Actual defect



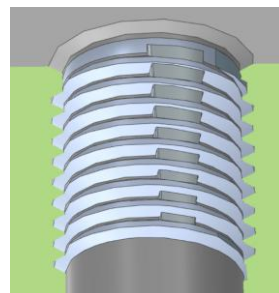
CAD representation

Figure 3.2 – Defect 2: actual photograph (left) and CAD model (right).

Defect 3 - Flattened Thread Crests: In this defect, the tips of the threads are worn down and no longer keep their original shape. The photograph shows a region of the thread where the tops of the thread turns appear visibly flatter than the rest. This kind of wear can happen due to repeated contact or improper engagement with the bolt, or during the machining process. The CAD model shows a section where the thread crests have been replaced with flat tops.



Actual defect



CAD representation

Figure 3.3 – Defect 3: actual photograph (left) and CAD model (right).

Defect 4 - Large Chunk of Material Missing at Thread Entry: This defect is more significant than Defect 1. A large portion of material at the top of the thread hole is missing, leaving a big open gap at the entry of the thread. The photograph clearly shows that a considerable amount of material has broken away from one side of the hole. In the CAD model, this is represented as a large spherical void at the top of the threaded region.

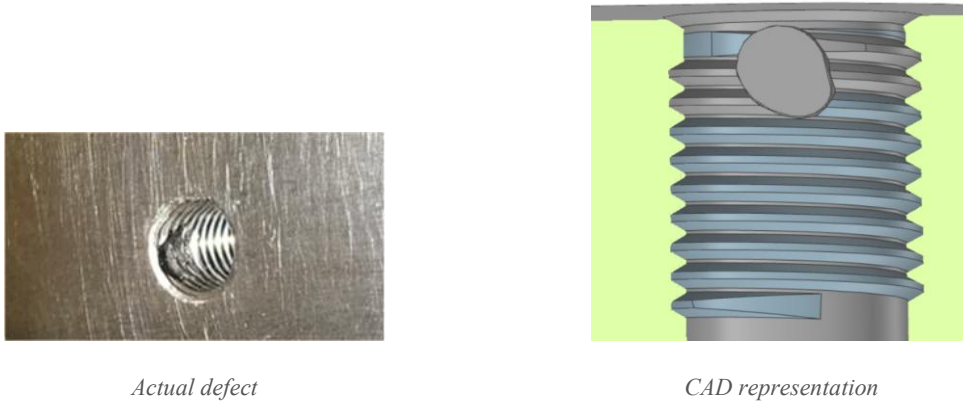
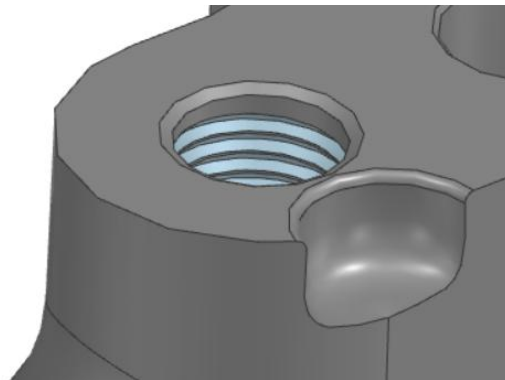


Figure 3.4 – Defect 4: actual photograph (left) and CAD model (right).

Defect 5 — Blended-Out Surface Around the Thread Hole: In this defect, the area of the part surface immediately around the thread hole has been smoothed out or deformed. Instead of having a flat or clearly defined edge at the hole entry, the surface has been blended out. This happens when the surface material is damaged by contact during manufacturing or assembly and deforms rather than breaking off cleanly. The deformed material is then blended out. No physical photograph was available for this defect; the CAD model showing the blended surface is shown below.



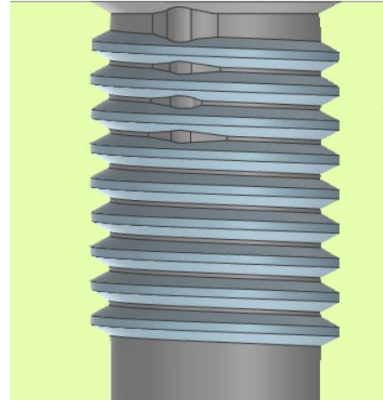
CAD representation (no actual photograph available)

Figure 3.5 – Defect 5: CAD model of blended-out surface around the thread hole.

Defect 6 - Small Breaks on Multiple Thread Flanks: This defect consists of small breaks appearing on the face of several thread turns. Unlike Defect 2 where cracks are concentrated in one area, here the breaks are spread across a number of threads. The photograph shows cracks visible on the thread surface at the top of the hole. In the CAD model, these are shown as small diamond-shaped slots on the faces of several thread turns.



Actual defect



CAD representation

Figure 3.6 – Defect 6: actual photograph (left) and CAD model (right).

3. Theoretical Background

Before getting into the numerical modelling side of things, it is necessary to establish a clear theoretical foundation that underpins the FEA performed in this study. This chapter covers the key mechanical and structural concepts that directly govern how bolted connections behave — particularly when geometric defects such as voids or corrosion damage are present at the bolt hole region. The topics discussed here, ranging from basic thread geometry to nonlinear material plasticity, were all actively referenced during the modelling process in ANSYS Mechanical.

It is worth noting that bolted joints are among the most widely used fastening methods in structural and mechanical engineering. Their apparent simplicity can be deceiving — the mechanics of thread engagement, contact, and pretension interact in ways that become especially sensitive when defects are introduced. Understanding these interactions at a theoretical level makes it far easier to interpret simulation results and spot where things go wrong.

3.1 Thread Terminology

The geometry of a threaded fastener is defined by a specific set of parameters, each of which plays a role in determining load-carrying capacity and stress distribution. Figure 3.1 [7] describes the basic terms. The key dimensional quantities are described below and illustrated in Figure 3.2 [8].

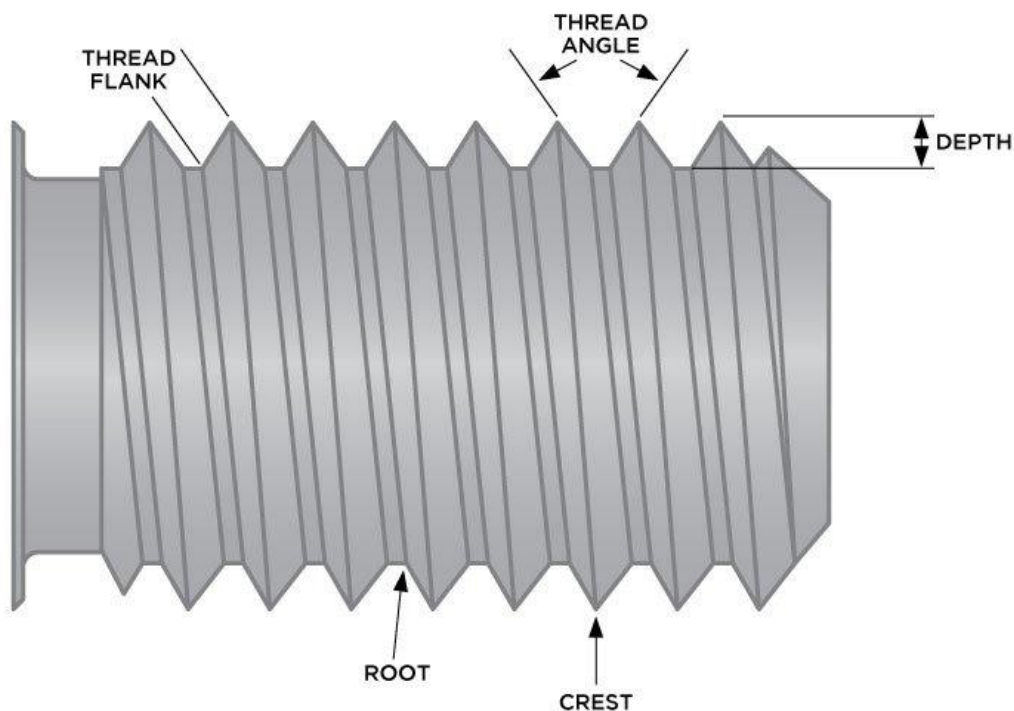


Figure 3.1. Thread geometry diagram – basic terminology.

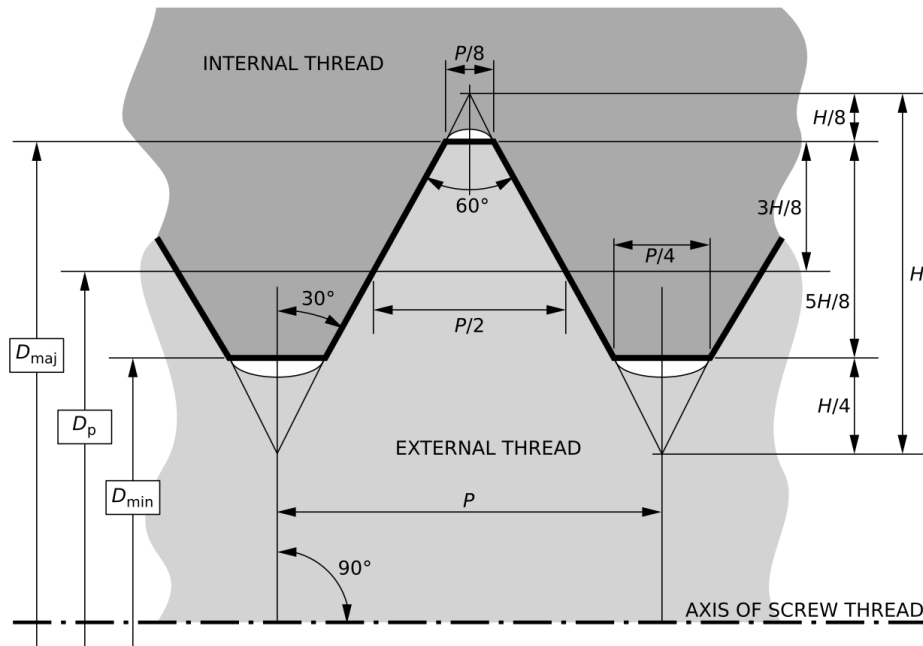


Figure 3.2. Thread geometry illustration – dimensional parameters.

Table 3.1 – Summary of Thread Terminology

Term	Symbol	Definition
Major Diameter	D_{maj}	Largest diameter of the thread (outer diameter of bolt)
Minor Diameter	D_{min}	Smallest diameter of the thread (root diameter)
Pitch Diameter	D_p	Diameter at which tooth and space widths are equal
Pitch	P	Axial distance between adjacent thread crests
Lead	l	Axial advance per one full revolution
Thread Angle	α	Included angle between flanks (60° for ISO metric)

The pitch diameter D_p is arguably the most important of these dimensions for stress calculations, as it represents the effective load-transferring contact surface between bolt and nut threads. For ANSYS, the geometry was modelled using these precise UNJF dimensions to ensure the contact and stress results reflected real-world fastener behaviour.

Thread root geometry is of particular interest in this project. The stress concentration at the thread root is well documented in the literature, and any defect — such as a crack initiating at the root — can significantly alter the local stress field. Shigley and Mischke suggest that the stress concentration factor K_t at the thread root can range from 2.5 to 4.0 for standard metric profiles, depending on the root radius.

3.2 Thread Forces

The mechanics of thread engagement are often explained using the inclined plane analogy, where the thread helix is 'unwrapped' into a ramp. This simplification, while not perfectly accurate, captures the essential force relationships and is widely used in textbook treatments of bolted joint mechanics.

When a torque is applied to the bolt head, it generates a wrench force at the effective radius (pitch diameter / 2). This force acts along the inclined plane of the thread helix. The forces on the inclined surface are then resolved, taking into account the friction on both the thread flanks and the bolt-head bearing surface.

In practice, a large proportion of the applied torque — typically 50 to 60 percent — is consumed by friction under the bolt head, with only around 30 to 40 percent contributing to thread torque, and the remainder being used to generate clamping force. This is why the tightening efficiency of bolted joints is so sensitive to lubrication conditions and surface finish.

The thread friction force acts tangentially along the thread helix and directly affects the shear stress state in the thread root region. When combined with the axial tension and bending from thread load distribution, this creates a multiaxial stress state that needs to be properly captured in the ANSYS simulation. Thread load distribution is notoriously non-uniform, with the first engaged thread carrying a disproportionately large share of the load.

3.3 Pretension

Pretension, or bolt preload, is the axial clamping force induced in the bolt when it is tightened. It is one of the most critical parameters in bolted joint design, yet also one of the most difficult to control precisely in practice. Getting the preload right determines whether the joint stays clamped under service loads, whether fatigue life is adequate, and whether the connected components are adequately compressed to prevent slip or fretting.

In ANSYS, the pretension was applied using the Bolt Pretension feature available in Mechanical, which introduces a cut plane in the bolt shank and applies the specified preload as force. This approach correctly captures the elastic interaction between bolt and joint members during the loading sequence. The pretension step was completed before any external loading was applied, following the recommended two-step analysis procedure.



Figure 3.3. Pretension loading procedure in ANSYS.

A key result of pretension is the introduction of residual compressive stress in the clamped joint material and residual tensile stress in the bolt. When external tensile loads are subsequently applied, these modify the bolt and joint loads according to the joint diagram, which has significant implications for fatigue life — especially at bolt holes where stress concentration factors amplify local peak stresses.

3.4 Stress-Strain Fundamentals

A solid understanding of stress and strain is fundamental to any structural simulation. Here the basic relationships are briefly reviewed, focusing on aspects most relevant to this study — particularly the three-dimensional stress state around a bolt hole and the conditions under which yielding might occur in the vicinity of a defect.

In a general three-dimensional body, the state of stress at a point is defined by a symmetric second-order tensor with six independent components $\{\sigma_{xx}, \sigma_{yy}, \sigma_{zz}, \tau_{xy}, \tau_{yz}, \tau_{xz}\}$. For linear isotropic elastic materials, the constitutive relationship is given by the generalised Hooke's Law [9]:

$$\varepsilon_{ij} = \frac{1 + \nu}{E} \sigma_{ij} - \frac{\nu}{E} \delta_{ij} \sigma_{kk} \quad (1)$$

where E is Young's modulus, ν is Poisson's ratio, δ_{ij} is the Kronecker delta, and σ_{kk} is the trace of the stress tensor (first stress invariant).

The von Mises yield criterion [10] is the most commonly used failure criterion for ductile metals, and was adopted in this study for evaluating yielding risk around the bolt hole defect. It states that yielding initiates when the von Mises equivalent stress σ_e reaches the uniaxial yield strength S_y :

$$\sigma_e = \sqrt{\frac{1}{2}[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]} \geq S_y \quad (2)$$

where σ_1 , σ_2 and σ_3 are the principal stresses. This criterion is independent of the geometry and takes into account that the hydrostatic stress component has no effect on yielding.

Stress concentration at bolt holes is a well-established phenomenon. For an infinite plate with a circular hole under uniaxial tension, the theoretical stress concentration factor $K_t = 3.0$ (Kirsch's solution [11]). For non-circular defects or cracks at the hole boundary, this factor can be considerably higher. Neuber's rule [12] was used in the previous method to relate the local elastic stress concentration to the actual elasto-plastic notch stress, which is important for accurate fatigue life estimation.

3.5 Contact Mechanism

Contact modelling is arguably the most nuanced part of setting up a bolted joint simulation in ANSYS Mechanical [13]. In this study, multiple surfaces interact simultaneously — bolt thread flanks engage nut thread flanks, the bolt head presses against the bearing surface of the upper plate, the clamped plates compress against each other, and the bolt shank potentially contacts the hole wall under lateral loading. Each of these contact pairs behaves differently in terms of how much relative motion is allowed and whether the surfaces can separate. Choosing the wrong contact type for any one of these pairs can lead to results that are either overly stiff or unrealistically compliant, both of which would skew the stress distributions around the defect region.

3.5.1 Bonded Contact

Bonded contact is the simplest and most computationally efficient type. It prevents all relative motion — both normal separation and tangential sliding — between the two surfaces, effectively treating the interface as if it were welded or glued. Because it introduces no nonlinearity (the contact state never changes during the solution), it is processed in a single linear pass and does not require iterative contact detection.

In this study, bonded contact was initially considered for the thread flanks in a preliminary simplified model to verify the mesh and loading setup before introducing the full nonlinear contact treatment. However, bonded contact overestimates joint stiffness because it does not allow the microslip that always occurs at real thread flanks under load. It was therefore replaced by frictional contact in the final model.

3.5.2 Frictionless Contact

Frictionless contact enforces the normal contact condition (no penetration, surfaces can separate) but applies no tangential resistance — it is equivalent to setting the friction coefficient $\mu = 0$. It is a nonlinear contact type because the contact area can change as the load changes, requiring iterative detection of which nodes are active (in contact) versus inactive (separated).

This contact type was not used in the primary simulation for this project.

3.5.3 Rough Contact

Rough contact is the opposite limit of frictionless — it enforces zero tangential slip (effectively infinite friction coefficient) while still allowing normal separation. Once two surfaces come into contact in the rough model, they stick permanently and cannot slide relative to each other. This is equivalent to Coulomb friction with $\mu \rightarrow \infty$.

This contact type was not used in the primary simulation for this project, because real bolt thread interfaces do exhibit finite slip under load, and modelling them as rough would significantly overestimate the thread root shear stress.

3.5.4 Frictional Contact

Frictional contact is the most physically realistic and also the most computationally demanding of the ANSYS contact types. It uses Coulomb's friction model, which distinguishes between two contact states: stick (when the tangential contact force $|F_t| < \mu F_n$) and slip (when $|F_t| = \mu F_n$ and the surfaces slide).

The governing Coulomb condition at each contact integration point is:

$$|F_t| \leq \mu F_n \rightarrow \text{no relative slip} \quad (3)$$

$$|F_t| = \mu F_n \rightarrow \text{surfaces slide} \quad (4)$$

Frictional contact was assigned to three interface pairs in this study: the bolt-head flange surface ($\mu = 0.2$), Bolt-Bracket ($\mu = 0.1$), and bolt thread–IMC ($\mu = 0.1$). The friction coefficient values were selected from the ranges documented in the GKN internal reference document [14].

4. Research Methodology

The study began by identifying a clear problem that needed to be investigated. Defining the problem at the start was important because it helped set the direction for everything that followed, making sure that the work carried out was focused and relevant. Once the problem was established, a literature review was conducted to gain a better understanding of what has already been studied and reported in this area. This involved reading through various academic papers, journals, and textbooks related to the topic. The literature review was a crucial step because it provided a strong foundation for the methodology, helped avoid repeating mistakes made in previous studies, and also gave useful guidance on how similar simulations have been approached by other researchers.

After completing the literature review, the next step was to identify all defects reported in threaded holes. This was done by reviewing the drawing and noting down all drawing item numbers related to the bolt holes. After identifying these numbers, they were searched in GKN's non-conformance portal to retrieve all associated non-conformance reports of IMC. In total, 191 non-conformances were identified for the bolt hole threads. These were classified into three categories: lack of material (material removal around, but not on, the threads), thread damage (the primary focus of this study), and casting defects. Following this classification, the six most common and relevant defects were selected for further analysis in this study.

Then the next step was to build the CAD model. The CAD model is essentially a three-dimensional digital representation of the component or structure being studied. At this stage, the model was created without any defects, as the primary goal of Phase 1 was to establish and validate a reliable baseline model before introducing any complexities. Getting the geometry right at this point was important because any errors in the CAD model would carry through into the simulation and affect the accuracy of the results.

With the CAD model ready, the simulation was then set up within the FEA software. This stage involved several important decisions that directly influenced the quality and reliability of the results. One of the key tasks was selecting the correct contact surfaces. This means identifying and defining which surfaces of the model interact with each other during the simulation, so that the software can correctly calculate the forces and stresses at those interfaces. Incorrectly defined contact surfaces can lead to unrealistic results, so careful attention was given to this step.

Another critical aspect of the setup was choosing an appropriate mesh size. Meshing is the process of dividing the entire model into a large number of small elements, and the size of these elements determines how detailed the calculations will be. A finer mesh generally gives more accurate results but requires significantly more computational time and resources. To find the right balance, a mesh convergence study was carried out, where the mesh was gradually refined until the results stopped changing significantly, confirming that the chosen mesh size was appropriate. In addition to this, the correct boundary conditions were applied to replicate the real-world loading and support conditions, and material properties were assigned based on values found in the literature.

Once the simulation setup was complete, the simulation was run and the results were analysed. At this point, a validation check was performed to determine whether the model was behaving as expected. This was done by comparing the simulation results against experimental data or analytical solutions available from the literature. If the results did not match well enough, the simulation setup was reviewed and adjusted, and the simulation was run again. This loop continued until the model was considered validated and the results were in good agreement with the reference data.

Once the baseline model was successfully validated, the study moved into Phase 2, which focused on simulating the effects of defects. A notable feature of this phase, as shown in the flowchart, is that two separate defect simulation streams were carried out independently and in parallel. Each stream corresponded to a different category or type of defect being investigated, which allowed the individual effects of each defect type to be studied in a systematic and controlled manner without interference between the two cases.

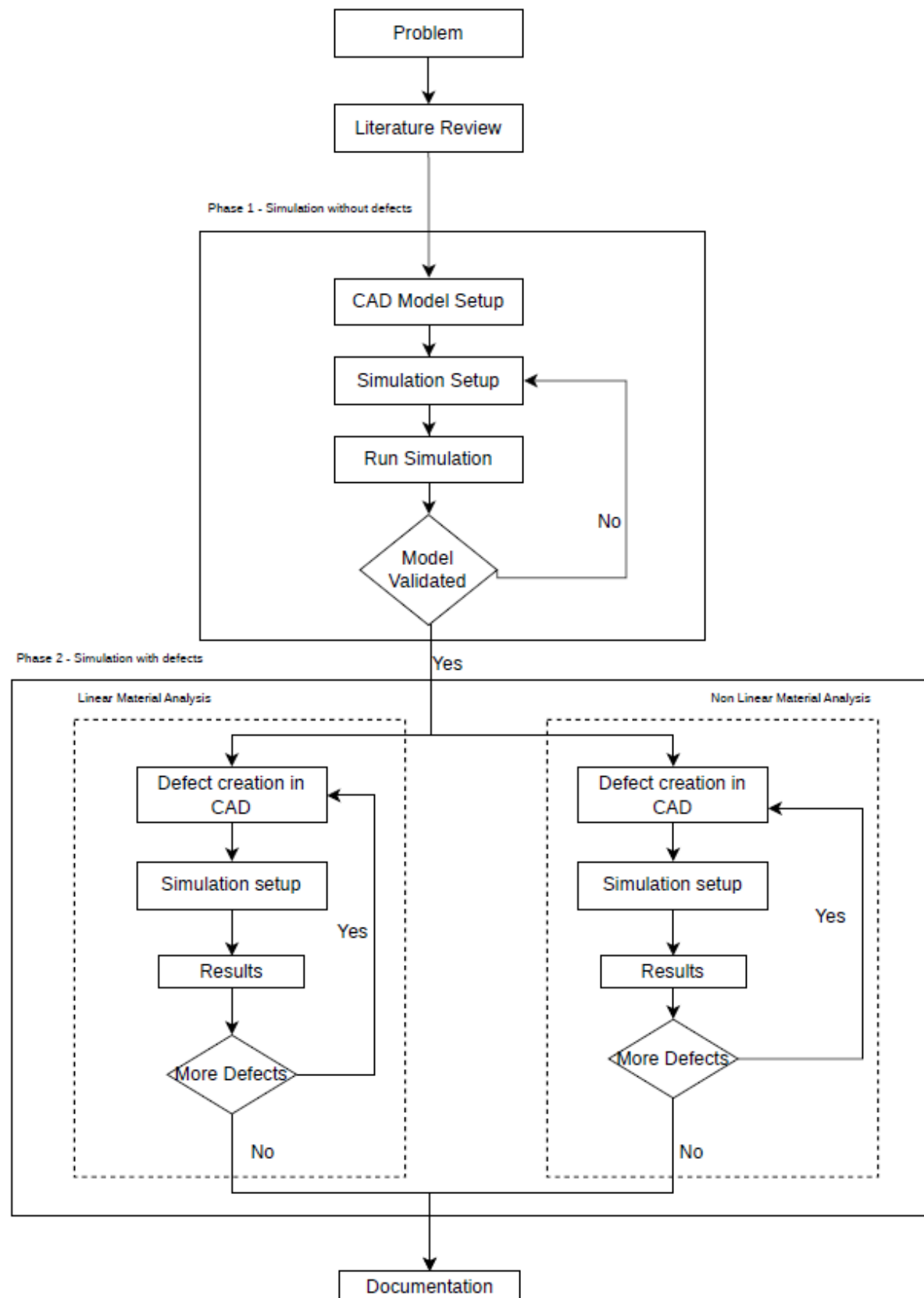


Figure 4.1. Simulation methodology flowchart.

Within each stream, defects were introduced one at a time into the CAD model. After each defect was created, the simulation was set up again using the same validated parameters from Phase 1. However, adjustments were made where necessary to account for the changes in geometry introduced by the defect, particularly in terms

of the contact surface definitions and the local mesh refinement around the defect region, where stress concentrations are typically expected. The simulation was then run and the results for that particular defect were recorded and analysed. Once this was completed, the process looped back to create the next defect, and the simulation was repeated. This iterative cycle continued within each stream until all the defect cases had been fully simulated.

After both defect simulation streams had been completed and all results had been gathered, the findings from every simulation case were brought together and documented. The documentation stage involved organising the results in a clear and structured manner, which forms the basis for the final analysis, discussion, and conclusions of the study. Having a thorough record of all the simulation cases, including the setup parameters and results for each defect, is essential for ensuring that the work is transparent and can be reproduced or reviewed by others in the future.

5. Finite Element Analysis Setup

5.1 CAD Geometry

A small portion of the IMC (H-Boss) was selected for simulation in this master's thesis, as modelling the entire IMC would be computationally prohibitive. A specific H-Boss was chosen as the first component for investigation due to the absence of external loading, making it a suitable starting point for establishing the simulation methodology before progressing to load cases.

The CAD geometry was developed using Siemens NX, and the complete assembly comprising the H-Boss, bracket, and bolts was used for the analysis, as illustrated in Figure 5.1. Based on the non-conformance reports, the identified defects were explicitly modelled into the threaded holes of the IMC. The geometry was then exported in STEP (Standard for the Exchange of Product Data) format and imported into ANSYS for simulation.

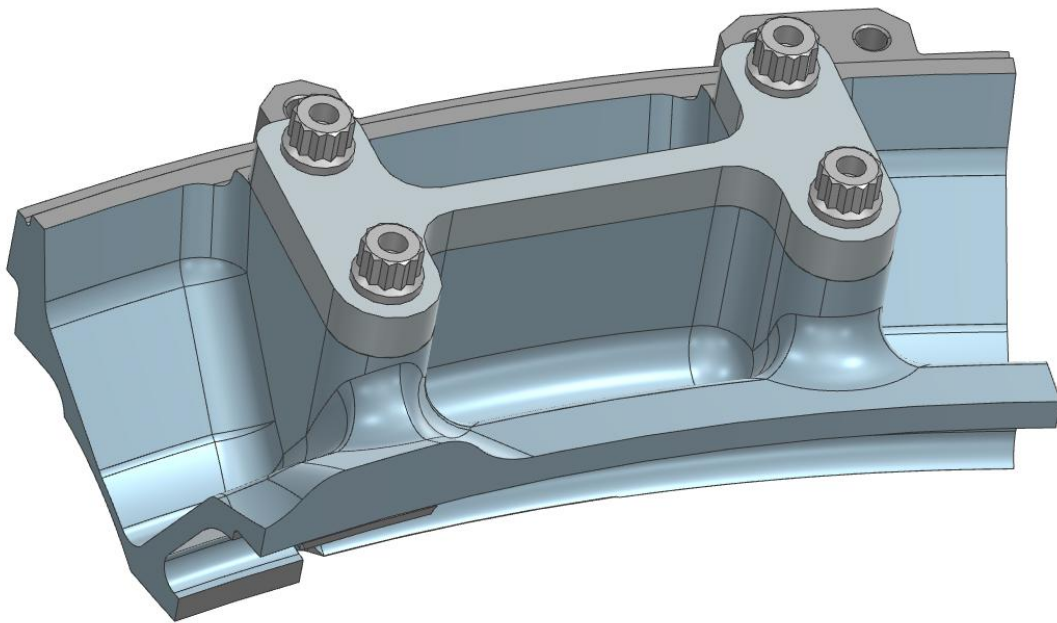


Figure 5.1. Detailed view of H-Boss assembly CAD geometry.

Prior to the analysis, a modification was made to the CAD geometry, particularly to the bolt threads. The final thread of the bolt contained a sharp edge which, if left unmodified, could lead to stress singularities in the simulation. Therefore, the final thread was given a smooth finish to eliminate this effect and ensure reliable results, as shown in Figure 5.2.

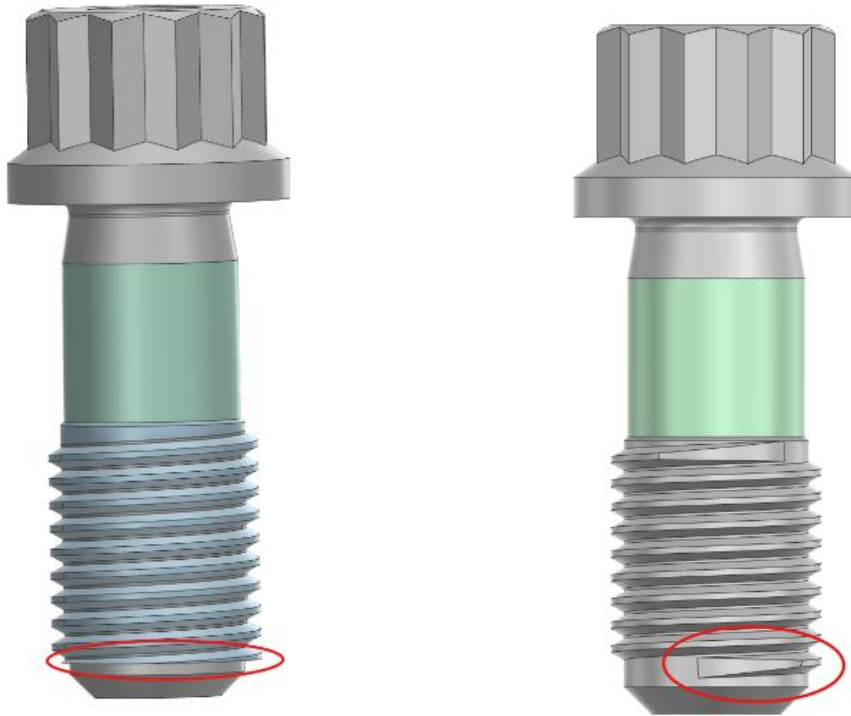


Figure 5.2. (a) Thread with sharp finish; (b) Thread with smooth finish.

5.2 Mesh

In FEA, the accuracy of the results depends on the density of the mesh. A mesh convergence study was therefore conducted to determine the optimal mesh configuration — one that produces results independent of further refinement, while minimising computational expense. As the mesh is progressively refined, the solution is expected to converge toward a true value, and convergence is considered achieved when further refinement yields negligible change in the quantity of interest.

The mesh was generated using quadratic tetrahedral elements (Tet10). Initially, a body sizing of 1 mm was applied to the IMC, bracket, and bolt, while a face sizing of 0.1 mm was applied to the thread surfaces of both components. The mesh was subsequently refined to a body sizing of 0.7 mm and a face sizing of 0.07 mm. A comparison of the peak stress values between the two configurations revealed a difference of only 0.01%, confirming that the solution had converged. To balance accuracy with computational efficiency, the final mesh adopted a body sizing of 1 mm and a face sizing of 0.1 mm.

5.3 Boundary Conditions

As per Saint-Venant's Principle [15], the stress effects of a load become negligible at a distance approximately equal to the largest dimension of the loaded region. Applying this principle, the scope of the analysis was restricted to the H-boss region of the IMC, as this constitutes the primary area of interest. The full IMC geometry was therefore reduced accordingly, and fixed support boundary conditions were assigned to the cut surfaces to replicate the structural constraints that would otherwise be provided by the removed material. A bolt pre-tension of 19,862 N was applied to the model as specified in [16]. The resulting boundary conditions are presented in Figure 5.4.

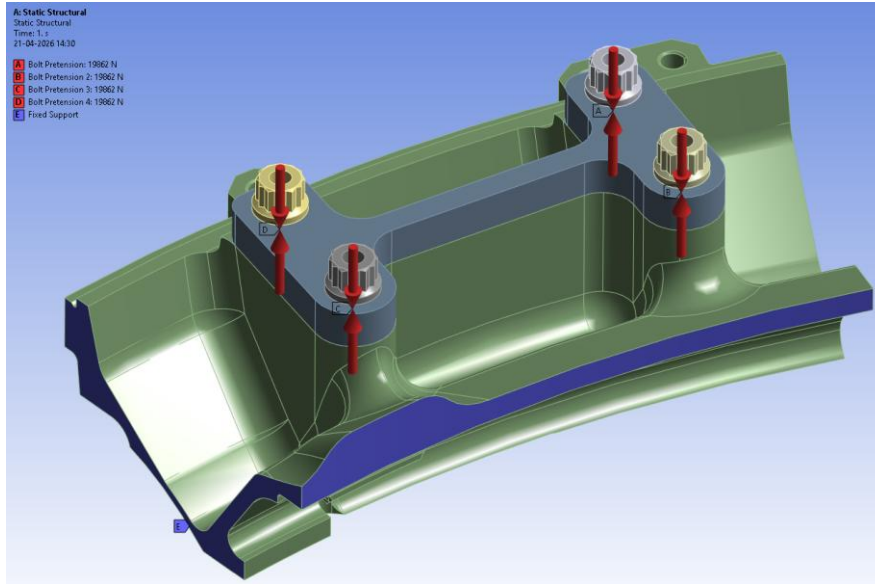


Figure 5.4. Boundary conditions applied to the H-Boss model.

Following the initial simulation, the stress distribution plots were examined to assess the extent of the stress field generated by the bolt pre-tension. The results indicated that the induced stresses remained localised in each bolt and showed no significant interaction with the adjacent bolt regions. This behaviour is consistent with Saint-Venant's Principle, which was subsequently applied for a second time to further simplify the model to a single-bolt configuration. Given that the stress states associated with individual bolts are largely independent of one another, this geometric simplification resulted in a considerable reduction in computational time, while preserving the accuracy of the results within the region of interest. The final geometry and boundary conditions are presented in Figure 5.5.

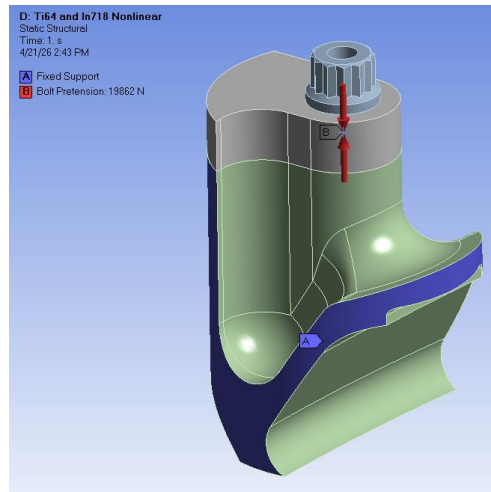


Figure 5.5. Final single-bolt geometry and boundary conditions.

5.4 Contact

Since the threads are explicitly modelled in the CAD geometry, a frictional contact condition was defined between the IMC and the bolt, with the IMC assigned as the target body and the bolt as the contact body. The contact was restricted to the load-bearing flank of the threads, as illustrated in Figure 5.6. This is because, under bolt pretension and external loading, force transfer occurs primarily through one flank of the thread due to the thread geometry and the axial direction of the applied load.

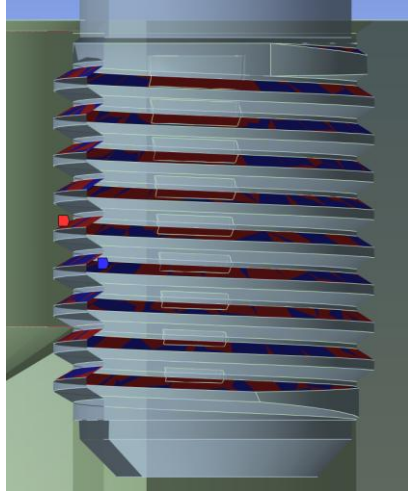


Figure 5.6. Contact surfaces between bolt and IMC.

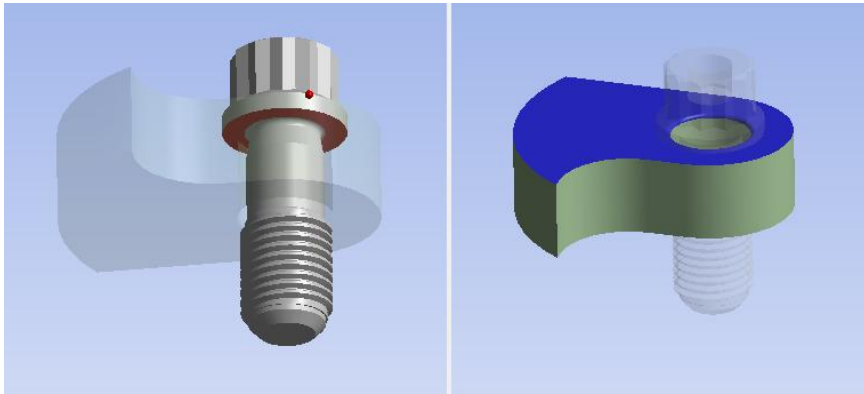


Figure 5.7. Contact surfaces between bolt and bracket.

Frictional contact was also applied to all interfaces between the bolt and bracket, and between the bracket and the IMC, as shown in Figures 5.7 and 5.8. The Coulomb friction model was adopted to characterise the tangential behaviour at these interfaces. The friction coefficients assigned to each contact pair were sourced from [14] and are summarised in Table 5.1.

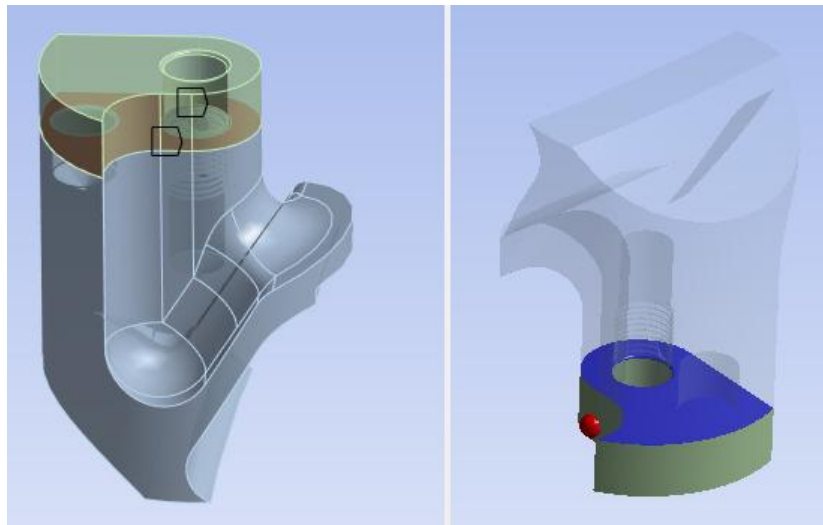


Figure 5.8. Contact surfaces between IMC and bracket.

Table 5.1. Friction Coefficient Values.

Contact Bodies	Friction
IMC and Bolt	0.1
Bracket and Bolt	0.1
Bracket and IMC	0.2

5.5 Material Data

Simulations were carried out using two sets of material data: linear elastic material data and non-linear elasto-plastic material data.

5.5.1 Linear Elastic Material

The material behaves predictably and proportionally. When stress is applied, the strain response follows a straight-line relationship. This behaviour holds if the material stays within its elastic limit, governed by Hooke's Law:

$$\sigma = E\varepsilon \quad (5)$$

where σ is the normal stress, E is Young's modulus and ε is the normal strain.

5.5.2 Non-Linear Plastic Material

The material does not follow the proportional relationship. The stress-strain curve is curved, meaning the response changes depending on the magnitude of the applied load. This can occur due to material properties, large deformations, or plastic behaviour beyond the yield point. The nonlinear material data was imported into ANSYS from GKN Aerospace's internal material library.

6. FEA Results and Analysis

This chapter presents the FEA results for the reference model (without defect) and all six defect cases. The comparison between each defect model and the reference is made using the linear elastic stress contour plots. The focus area for each defect is stated clearly, based on where the critical assessment should be made.

6.1 Analysis Approach and Justification

Earlier iterations of this analysis considered two additional metrics alongside the linear stress contours: the nonlinear plastic stress and the equivalent plastic strain. However, following discussions with the project supervisor, it was decided that these metrics should not be used as the primary basis for comparison. The reasons are as follows.

The nonlinear peak stress is not a useful differentiator between defect cases because, in a nonlinear model, the material yields and the stress is capped near the yield strength of Ti-64 (823–866 MPa) [17] for all defect cases. This means almost all defects produce a similar peak nonlinear stress value, making it impossible to rank them by severity using this metric alone.

The equivalent plastic strain is heavily dependent on the mesh size, especially at defect tips where a stress singularity exists. At a singularity, the plastic strain theoretically increases without limit as the mesh is refined. This means any plastic strain value extracted at the defect tip is not a reliable, mesh-independent quantity and should not be used to compare defects unless a full mesh convergence study has been completed.

For these reasons, the linear elastic stress contour is used as the primary comparison tool. The linear model does not cap at yield, so the peak stress values clearly reflect the severity of the stress concentration created by each defect. Comparing the linear stress contour of each defect model against the reference contour shows directly how much the defect changes the stress distribution in the critical regions. The plastic strain plots are included in each section as supporting information only.

The focus area for each defect is defined based on the defect location and type:

- For Defects 1, 2, and 6: the main focus is the stress distribution in the region of the first two engaged threads. The defect itself sits in a zone of lower stress so it is not the primary concern.
- For Defects 3, 4, and 5: the assessment covers both the first two thread region and the stress at the defect location itself, as these defects are in or near higher-stress zones.

6.2 Reference Case: Without Defect

The reference analysis was carried out on the defect-free geometry to establish the baseline stress distribution. All defect cases are compared against these results.

6.2.1 Linear Elastic Stress

The reference model was analysed using a linear elastic material model to establish the baseline stress distribution. The contour plot in Figure 6.1 shows the von Mises stress across the full thread engagement length, with stress values labelled at selected nodes. The first thread carries the highest stress (1964.5 MPa at the peak zone), with the second thread showing 816.15 MPa. Beyond that, the stress drops considerably: 138.29 MPa at the third thread node and 77.715 MPa at the fourth. This rapid decrease shows that the connection transfers the majority of the load through the first two threads, which is the expected behaviour for this type of threaded joint.

These selected node values form the reference baseline against which all defect cases are compared.

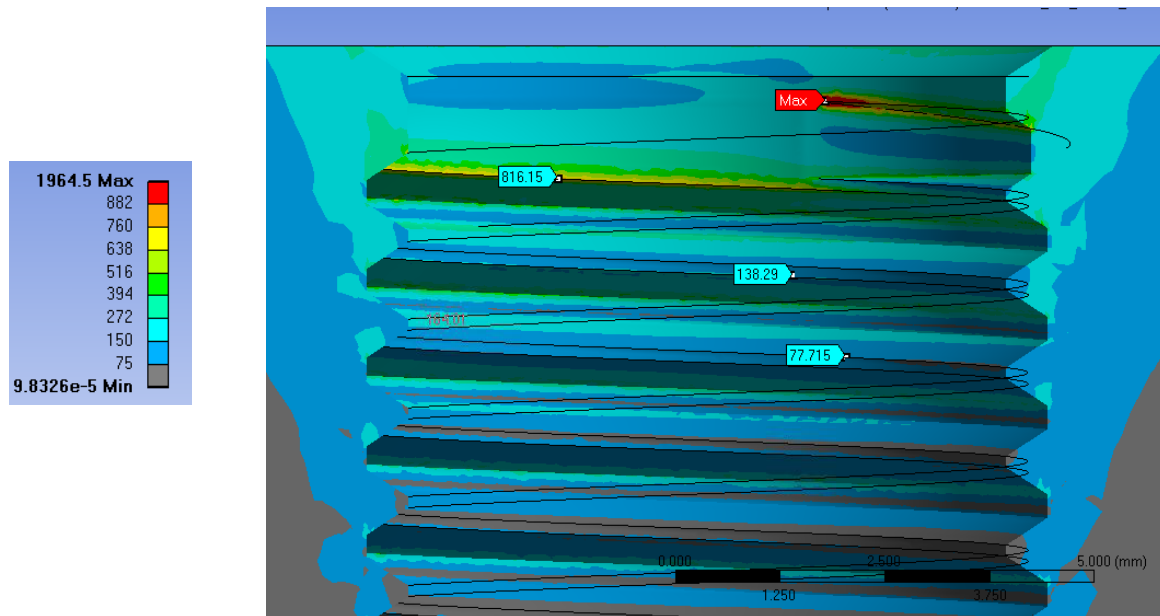


Figure 6.1. von Mises stress contour – Without Defect, Linear Elastic Analysis (Max: 1964.5 MPa).

6.2.2 Nonlinear Plastic Stress

Shown here for reference context only. The nonlinear model gave a peak of 882.66 MPa. As discussed in Section 6.1, this value is not used for comparison between defect cases.

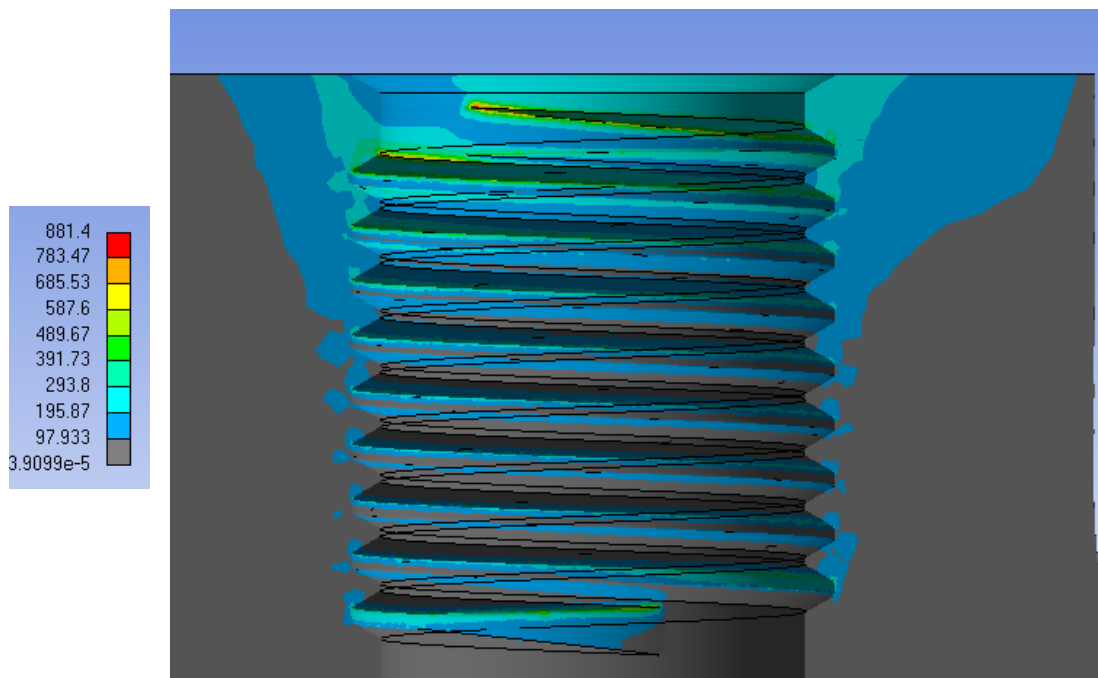


Figure 6.2. von Mises stress contour – Without Defect, Nonlinear Plastic Analysis (Max: 881.4 MPa).

6.2.3 Plastic Strain (Reference)

Shown for reference only. The reference plastic strain of 1.8% occurs at the first thread root even without any defect, confirming that some localised yielding is inherent to this connection under the applied load.

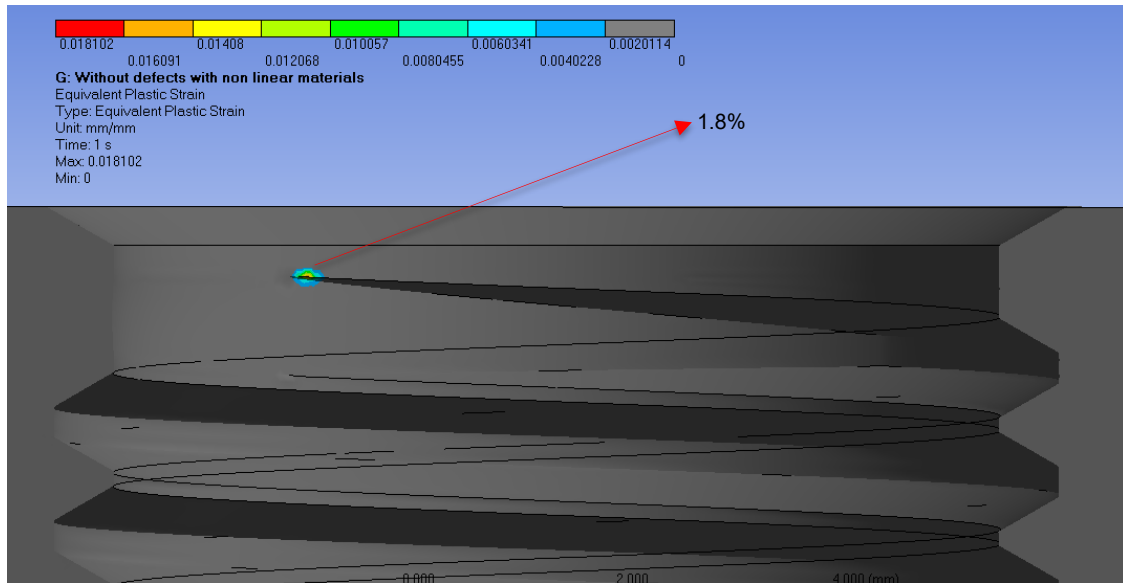


Figure 6.3. Equivalent Plastic Strain – Without Defect (Max: 1.8%).

6.3 Defect 1

Focus area: Stress in the region of the first two engaged threads. The defect (hole) is in a lower-stress zone and is not the primary concern.

6.3.1 Linear Elastic Stress

The linear elastic analysis of Defect 1 gave a peak von Mises stress of 1959.6 MPa, shown in Figure 6.4. Selected node values were recorded at the same thread locations as the reference to allow a direct comparison.

Table 6.1 compares the stress values at the selected nodes between the reference model and Defect 1.

Table 6.1. Selected node stress comparison between the reference model and Defect 1.

Location	Reference (MPa)	Defect 1 (MPa)	Difference
1st thread (max zone)	1964.5	1959.6	-0.25%
2nd thread node	816.15	813.02	-0.38%
3rd thread node	138.29	142.87	+3.3%
4th thread node	77.715	89.815	+15.6%
Defect area node	N/A	100.8	—

Starting from the top of the thread engagement region, the stress at the first thread zone is essentially the same in both models – 1964.5 MPa in the reference and 1959.6 MPa in Defect 1, a difference of less than 0.3%. Moving to the second thread, the values are again almost identical at 816.15 MPa and 813.02 MPa respectively. These first two thread nodes are the most structurally important as they carry the largest share of the applied load. The near-identical values confirm that Defect 1 has no meaningful effect on load transfer in this critical region.

Further down, at the third thread node, the reference reads 138.29 MPa while Defect 1 reads 142.87 MPa – a small increase of about 3.3%. At the fourth thread node, the reference is 77.715 MPa and Defect 1 is 89.815 MPa, a difference of about 15.6%. Although this percentage looks larger, the absolute stress values at these locations are very low in both cases and well below the yield strength of Ti-64 (823–866 MPa). The slight increase in stress at the lower threads is likely because the defect void removes some load-carrying material in that region, causing the threads below to pick up a marginally higher share of the load.

The defect area node itself reads 100.8 MPa. This is a low value, sitting well within the blue/light-blue zone of the colour scale. The defect is not drawing significant stress to itself, which is consistent with a large void being located in a lower-stress region of the connection.

Looking at the overall contour patterns in both figures, they show the same characteristic behaviour. Stress is highest at the first engaged thread and decreases progressively down the engagement length. The colour distribution, the shape of the stress bands, and the spacing of the contour lines are visually very similar between the two models.

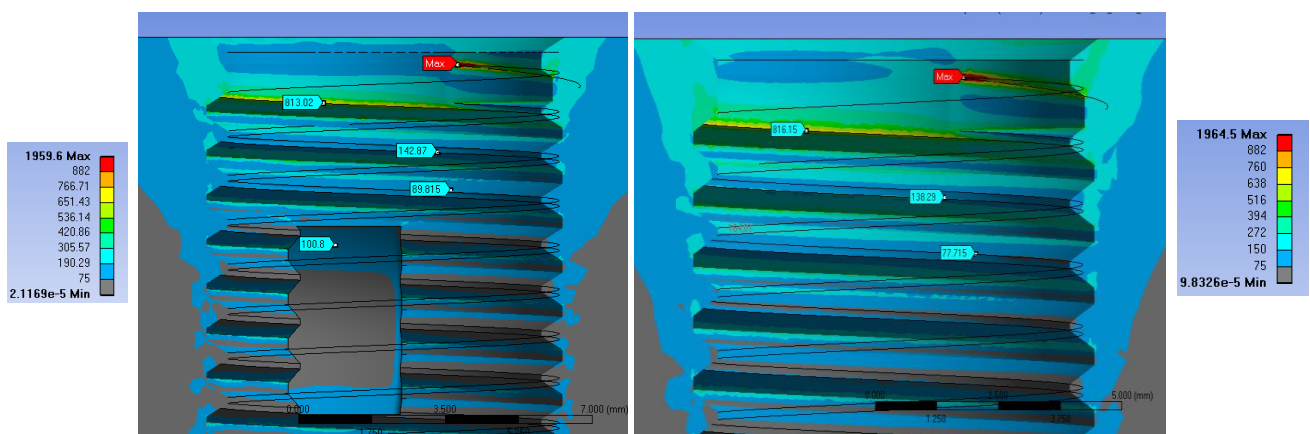


Figure 6.4. von Mises Linear stress contour – Defect 1 vs Defect Free.

6.3.2 Plastic Strain (Supporting Information)

Note: Plastic strain is shown for context only and is not used as a basis for assessment – see Section 6.1.

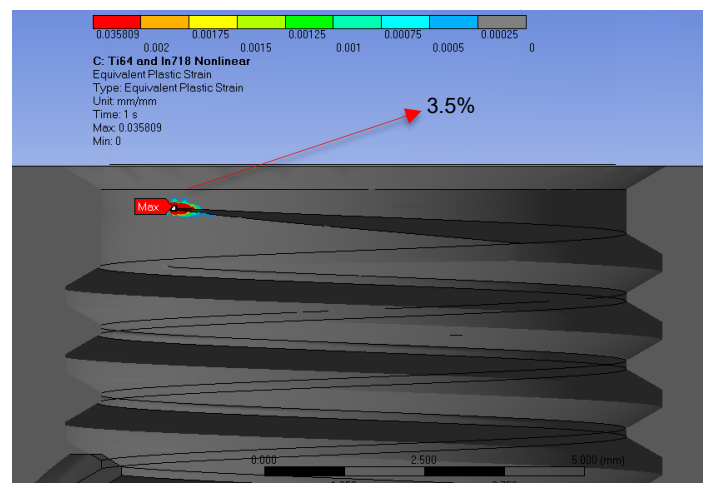


Figure 6.5. Equivalent Plastic Strain – Defect 1 (shown for reference only of max 3.5%).

6.3.3 Conclusion – Defect 1

The node stress values in the first two threads – the primary focus area for this defect – are within 0.4% of the reference values. The small increases observed at the lower thread nodes are at stress levels that are structurally insignificant. The defect area itself is under low stress (100.8 MPa), confirming the void is not acting as a significant stress concentration in this loading condition. The overall stress distribution pattern is not changed by the presence of Defect 1.

Assessment: ACCEPTABLE

6.4 Defect 2

Focus area: Stress in the region of the first two engaged threads. The defect itself is not the primary concern.

Table 6.2. Selected node stress comparison between the reference model and Defect 2

Location	Reference (MPa)	Defect (MPa)	Difference
2nd thread node	816.15	626.84	-23.2%
3rd thread node	138.29	137	-0.9%
4th thread node	77.715	78.878	+1.5%
Defect area – node A	N/A	138.75	—
Defect area – node B	N/A	106.39	—
Defect area – node C	N/A	110.73	—
Defect area – node D	N/A	88.602	—

6.4.1 Linear Elastic Stress

In Figure 6.6, the second thread node for Defect 2 reads 626.84 MPa, which is 23% lower than the reference value of 816.15 MPa. This reduction is a positive finding – the defect has not increased the stress at this critical location. At the third thread, the value is 137 MPa, essentially the same as the reference (138.29 MPa, -0.9%). The fourth thread reads 78.878 MPa against 77.715 MPa in the reference, a negligible difference of +1.5%.

Figure 6.7 shows the stress in the area immediately around the defect. The two circled zones highlight the defect location. The node values within these zones range from 88.602 MPa to 138.75 MPa, with a higher reading of 272.04 MPa at the mid-region node. All of these values are well below the yield strength of Ti-64. The defect is not acting as a significant stress raiser, and the surrounding material remains at low stress levels.

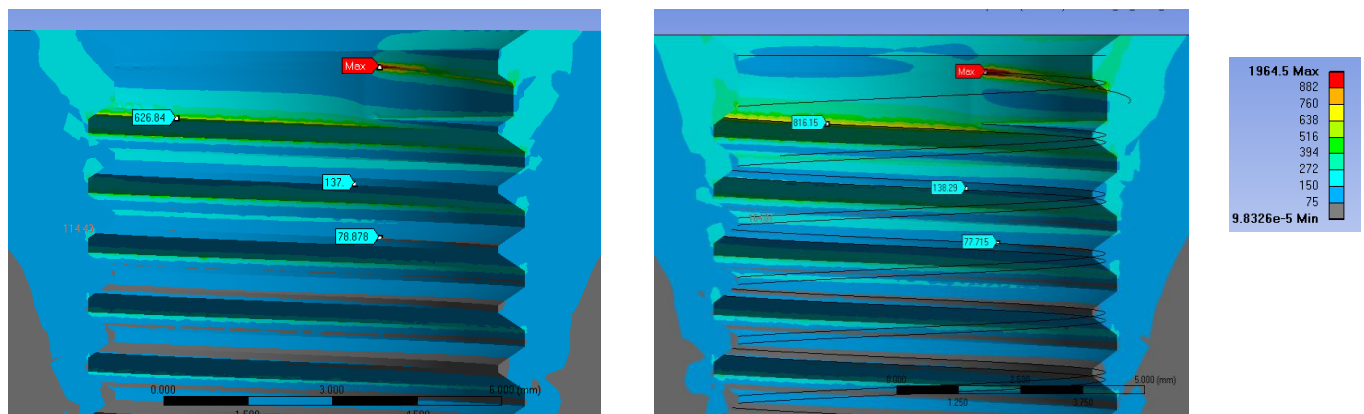


Figure 6.6. von Mises Linear stress comparison – Defect 2 vs Defect Free.

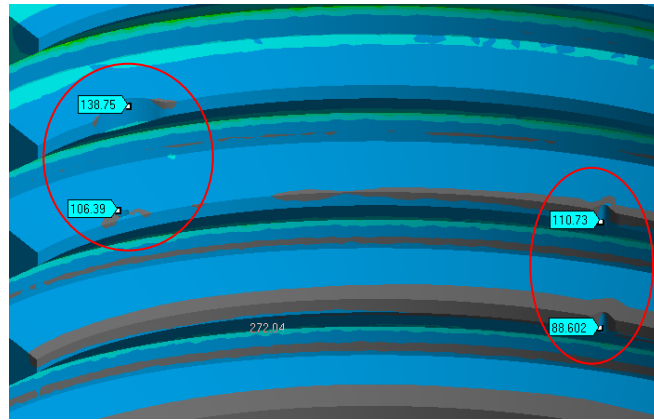


Figure 6.7. von Mises Linear stress around the defect.

6.4.2 Plastic Strain (Supporting Information)

Note: Plastic strain is shown for context only and is not used as a basis for assessment – see Section 6.1.

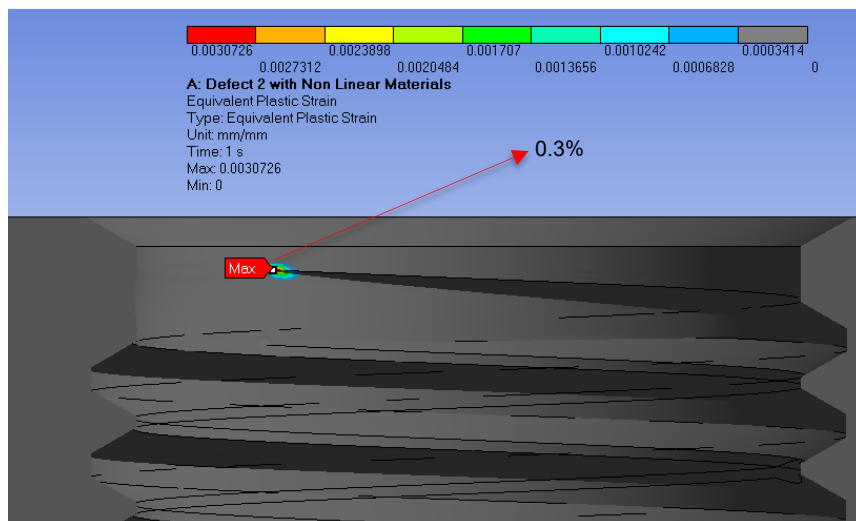


Figure 6.8. Equivalent Plastic Strain of Max 0.3% – Defect 2 (shown for reference only).

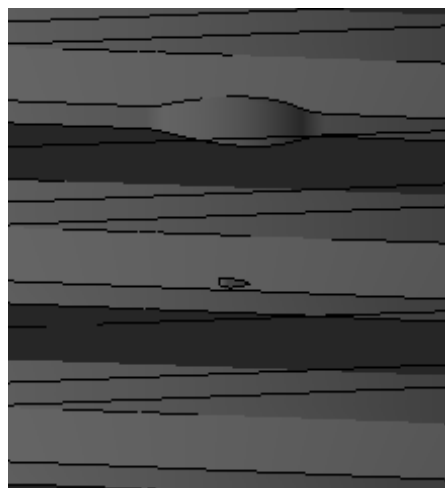


Figure 6.9. Plastic Strain around Defect 2 – zoomed view (shown for reference only).

6.4.3 Conclusion – Defect 2

The second thread node is 23% lower than the reference, which is a better result. Threads 3 and 4 match the reference closely. There is no indication of an unusual stress concentration that would make this defect a concern. A lower peak stress compared to the defect-free case is a positive result.

Assessment: ACCEPTABLE

6.5 Defect 3

Focus area: Stress in the region of the first two engaged threads and the stress at the defect location itself.

Table 6.3. Selected node comparison – Defect 3. Values at the defect blocks compared against the reference at the same thread level.

Location	Reference (MPa)	Defect (MPa)	Difference
2nd thread level – at defect block	816.15	1562.8	+91.5%
2nd thread level – defect block (2)	138.29	958.67	+17.5%
2nd thread level – defect block (3)	138.29	474.88	-41.8%
3rd thread node (away from defect)	138.29	136.87	-1.0%
4th thread node (away from defect)	77.715	75.829	-2.4%

6.5.1 Linear Elastic Stress

Defect 3 is the most severe case in this study. At the first defect block, the node reads 1562.8 MPa against a reference of 816.15 MPa at the same thread level — an increase of 91.5%. This alone would be a serious concern. However, the second and third defect blocks show a drastic increase. The reference value at their equivalent positions in the thread engagement region is 138.29 MPa. The node values at those blocks are 958.67 MPa and 474.88 MPa respectively — increases of 593% and 243% above what the undisturbed thread experiences at that level. These very large increases show that the removed material is forcing the load to concentrate dramatically in the remaining thread ligament, producing stress levels that are completely inconsistent with normal thread behaviour at those positions.

Moving to the threads on the undisturbed side of the connection, away from the defect blocks, the picture is different. The 3rd thread node reads 136.87 MPa against a reference of 77.715 MPa — a 76% increase, suggesting some additional load redistribution is reaching even the undamaged threads. The 4th thread node reads 75.829 MPa, essentially the same as the reference (77.715 MPa, -2.4%), indicating the stress returns to near-normal levels by that point.

The overall picture is that Defect 3 creates extreme localised stress concentrations at the defects themselves, with the most severe case being 958.67 MPa at a location where the reference only expects 138.29 MPa. This is nearly 7 times the reference stress at that position and exceeds the yield strength of Ti-64 (823–866 MPa) by a significant margin.

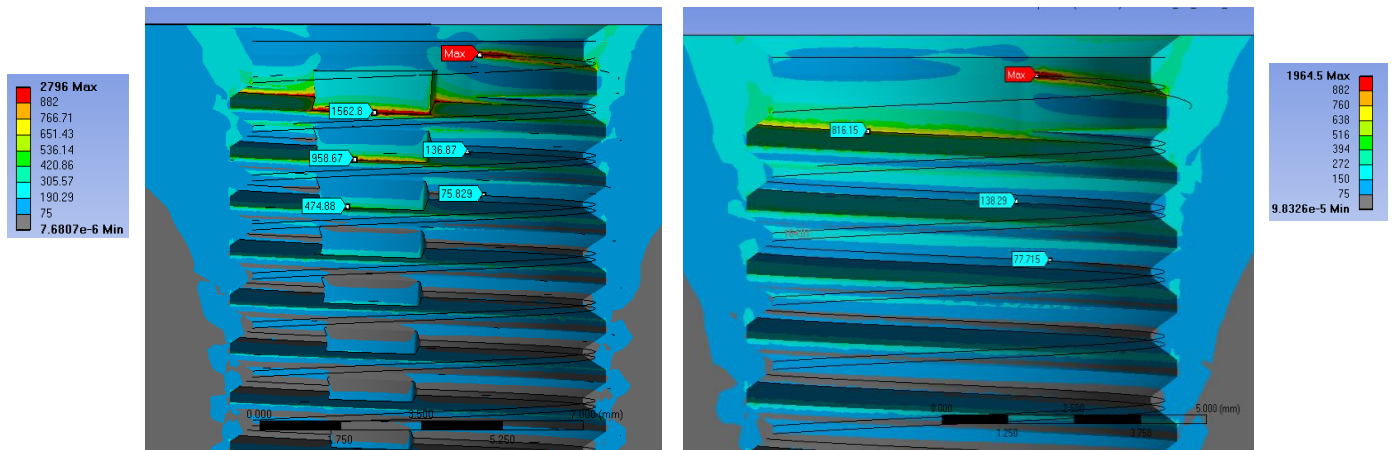


Figure 6.10. von Mises Linear stress comparison – Defect 3 vs Defect Free.

6.5.2 Plastic Strain (Supporting Information)

Note: The nonlinear strain plot is included here as additional context. As discussed in Section 6.1, it is not the primary comparison metric.

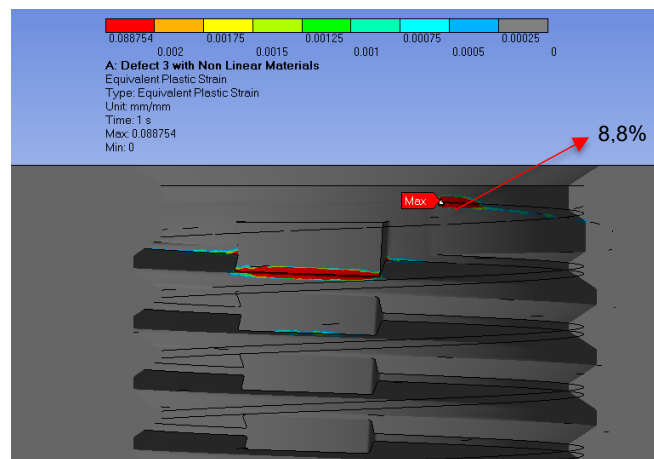


Figure 6.11. Equivalent Plastic Strain of Max 8.8% – Defect 3 (shown for reference only).

6.5.3 Conclusion – Defect 3

The comparison confirms that the defect blocks are experiencing stress levels between 243% and 593% above what the undisturbed thread carries at the same positions. The removed material has fundamentally altered the load transfer path through the connection. This defect cannot be accepted in service.

Assessment: NOT ACCEPTABLE

6.6 Defect 4

Focus area: Stress in the region of the first two engaged threads and the stress at the defect location itself.

Table 6.4. Selected node comparison – Defect 4 with reference

Location	Reference (MPa)	Defect (MPa)	Difference
Near-max zone (1st thread)	670.2	1115.2	+66.4%
2nd thread – left of void	816.15	854.81	+4.7%
3rd thread – left of void	138.29	672.43	+386.2%
Void perimeter node	N/A	993.27	—
Lower thread (away from void)	77.715	125.45	+61.4%

6.6.1 Linear Elastic Stress

Defect 4 involves a large circular void (visible in the contour with a green circle outline) positioned within the thread engagement zone. The void significantly changes how stress flows through the connection, as the load has to travel around it rather than through it.

The node nearest to the maximum stress zone reads 1115.2 MPa – this is at the top thread area where the load first enters the connection. The second thread node to the left of the void reads 854.81 MPa, which is close to but slightly above the reference value of 816.15 MPa at that level (+4.7%). More notably, the node at what would be the third thread level, on the left side adjacent to the void, reads 672.43 MPa. The reference at the third thread is only 138.29 MPa, so this is a major elevation in stress, caused by the load being forced to detour around the void. The node directly on the void perimeter reads 993.27 MPa, further confirming the elevated stress concentration around the void boundary.

The node further below the void region reads 125.45 MPa, which is close to the reference at the equivalent position, suggesting the stress returns to near-normal levels once the load is past the void.

The void, because of its size and position in the thread engagement zone, is creating a significant redistribution of stress that goes well beyond what the reference model experiences at the same locations.

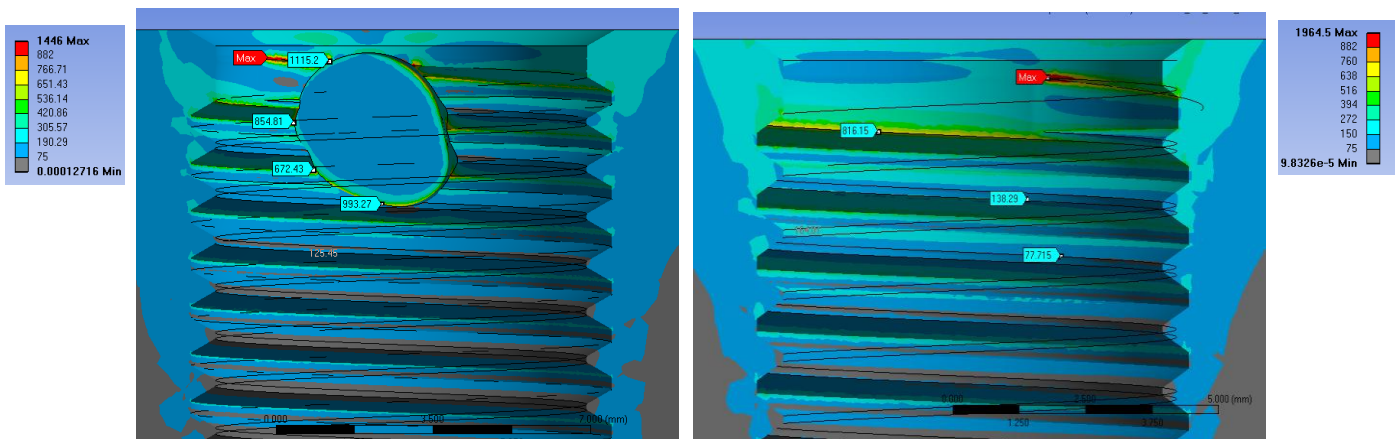


Figure 6.12. von Mises Linear stress comparison – Defect 4 vs Defect Free.

6.6.2 Plastic Strain (Supporting Information)

Note: Plastic strain is shown for context only and is not used as a basis for assessment – see Section 6.1.

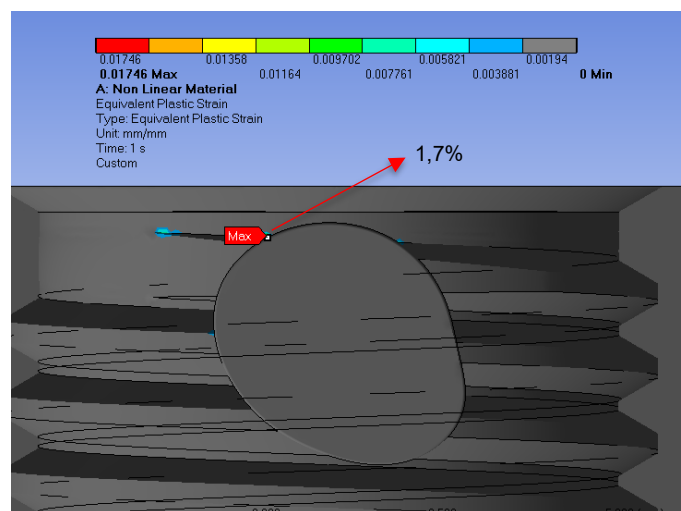


Figure 6.13. Equivalent Plastic Strain of max 1.7% – Defect 4 (shown for reference only).

6.6.3 Conclusion – Defect 4

The large circular void is positioned within the load-carrying thread engagement zone, forcing the stress to redistribute around it. Node values adjacent to and on the void perimeter are substantially higher than the reference at the same locations (993.27 MPa at the void perimeter, 672.43 MPa at the 3rd thread level). Further assessment of the void size and its position is recommended before a definitive acceptance decision is made. Performing a crack propagation analysis would be the ideal next step.

Assessment: BORDERLINE – The stress is below the reference but the local concentration at the defect site requires further assessment

6.6.4 Crack Propagation Analysis (NASGRO)

The linear elastic stress results alone were not sufficient to reach a final acceptance decision for Defect 4. While the stress values at the first and second thread nodes remained close to the reference, the void perimeter node reached 993.27 MPa and the adjacent third thread node reached 672.43 MPa — levels that represent a significant redistribution of load around the void. These elevated values raised the question of whether the defect could act as an initiation site for crack growth under operational loading. To address this, crack propagation analysis was carried out using NASGRO.

NASGRO is a fracture mechanics software tool used to evaluate crack growth behaviour under fatigue loading conditions. The surface crack model SC30 was selected, which represents a semi-elliptical surface crack in a finite-width plate subjected to a stress gradient through the thickness. This model was considered appropriate for the geometry and loading conditions associated with Defect 4. The following input values were used for the analysis:

- Stress gradient: extracted directly from the ANSYS finite element model at the defect location
- Thickness, $t = 5$ mm (from ANSYS geometry)
- Width, $W = 30$ mm (from ANSYS geometry)
- Crack offset, $B = 1$ (assumption)
- Initial flaw size, $a_0 = 0.79$ mm

Both the thermal condition and the mechanical loading condition were considered in the analysis, consistent with the in-service loading environment of the IMC. The crack geometry used in NASGRO is illustrated in Figure 6.14, showing the SC30 surface crack model with its relevant dimensional parameters.

The NASGRO analysis was used to assess whether the initial flaw of 0.79 mm would grow to a critical size under the applied loading cycles, or whether it would remain stable throughout the expected service life. The results of the crack propagation analysis, combined with the FEA stress data and the minimum wall thickness assessment, form the basis for the final acceptance decision on Defect 4. Given the borderline nature of the FEA results and the proximity of the void to the primary load-carrying threads, the outcome of the NASGRO analysis is a critical input to that decision.

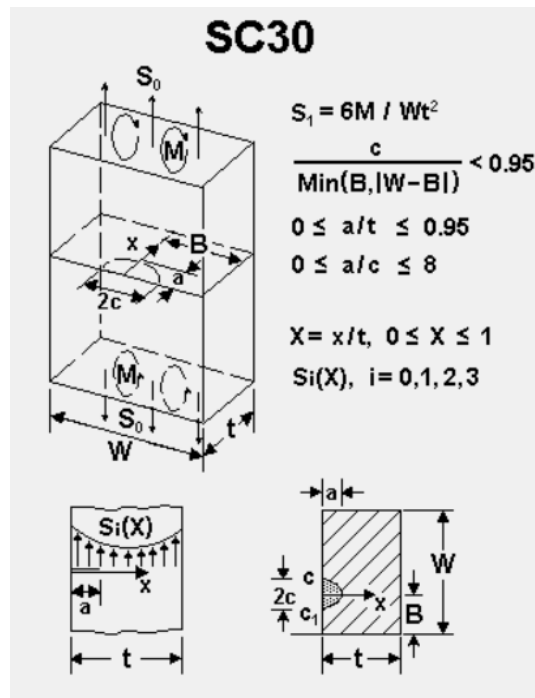


Figure 6.14. Crack Geometry.

6.7 Defect 5

Focus area: Stress in the region of the first two engaged threads and the stress at the defect location itself.

Table 6.5. Selected node comparison – Defect 5 with reference

Location	Reference (MPa)	Defect (MPa)	Difference
2nd thread node	816.15	773.6	-5.2%
3rd thread node	138.29	214.97	+55.4%
4th thread node	77.715	147.11	+89.3%
Defect area – upper node	N/A	751.19	—
Defect area – lower node	N/A	437.33	—

6.7.1 Linear Elastic Stress

The second thread node for Defect 5 reads 773.6 MPa, slightly lower than the reference of 816.15 MPa (−5.2%). At first glance this might appear acceptable. However, moving to the third thread, the value increases to 214.97 MPa compared to the reference of 138.29 MPa – an increase of 55.5%. At the fourth thread, the value is 147.11 MPa against 77.715 MPa in the reference – an increase of 89.3%. These are significant increases that show the defect is changing how load is distributed along the engagement length, with more stress being carried by the lower threads than in the reference case.

Figure 6.15 shows the stress in the area around the defect itself. The upper node at the defect reads 751.19 MPa, which is approaching but below the yield strength of Ti-64 (823–866 MPa). The lower defect node reads 437.33 MPa. The defect appears to be a blend that reduces the wall thickness of the connection at that location.

The minimum wall thickness at the defect location is an important design consideration for this defect type. Even if the stress values are below yield, a significant reduction in wall thickness means less material is

available to carry the load, and under fatigue conditions, this thinner section could be the initiation site for cracking. The acceptability of this defect cannot be determined on stress alone and must also consider the remaining wall thickness relative to the minimum permissible design value.

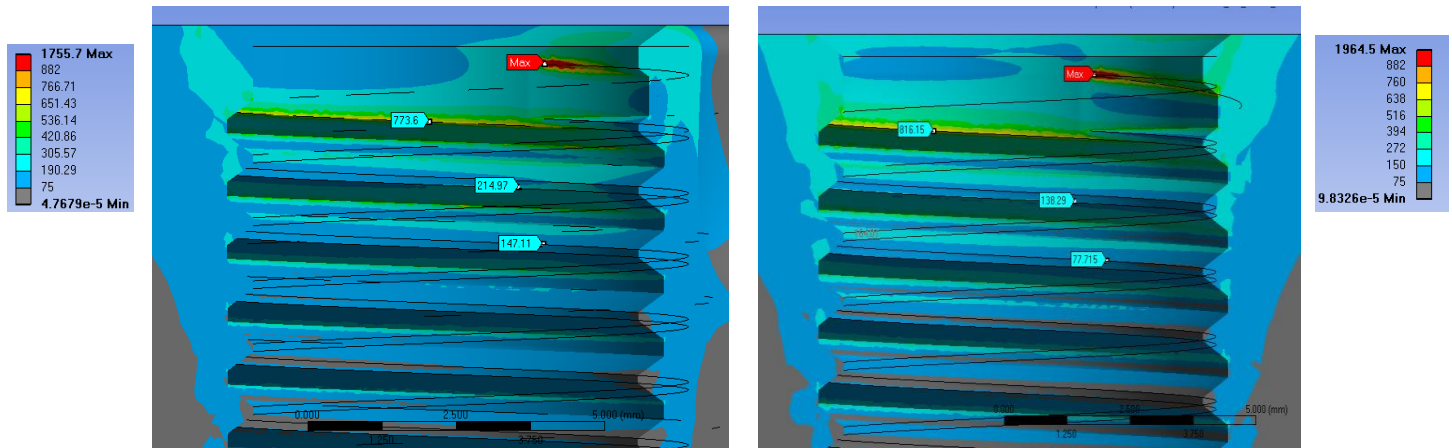


Figure 6.15. von Mises Linear stress comparison – Defect 5 vs Defect Free.

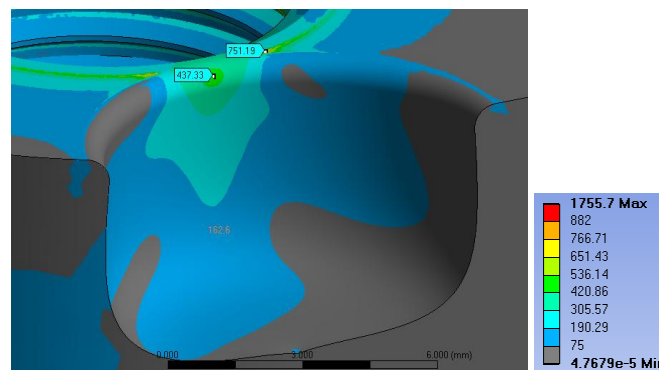


Figure 6.16. Stress distribution around the Defect 5 location.

6.7.2 Plastic Strain (Supporting Information)

Note: Plastic strain is shown for context only and is not used as a basis for assessment – see Section 6.1.

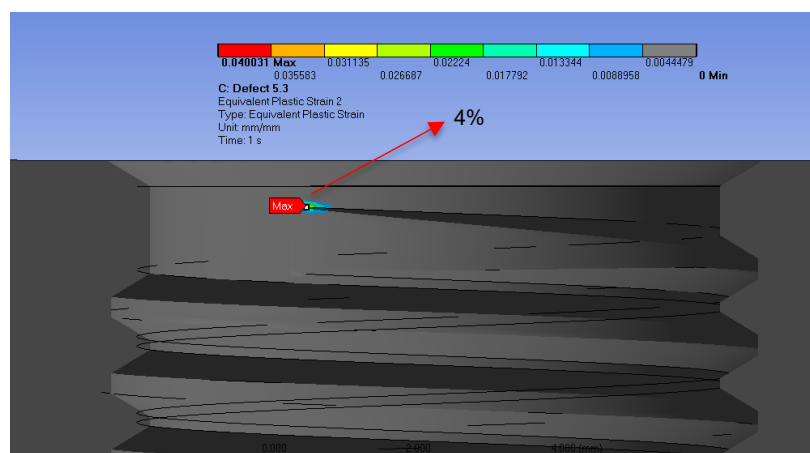


Figure 6.17. Equivalent Plastic Strain of max 4% – Defect 5 (shown for reference only).

6.7.3 Conclusion – Defect 5

The 3rd and 4th thread stresses are notably elevated compared to the reference (+55% and +89% respectively). The defect area stress (751.19 MPa) approaches the yield strength. The minimum remaining wall thickness at the defect location must be evaluated against design limits before an acceptance decision can be made.

Assessment: BORDERLINE – The stress is below the reference, but the minimum wall is an important aspect to be considered

6.8 Defect 6

Focus area: Stress in the region of the first two engaged threads. The defect itself is not the primary concern.

Table 6.6. Selected node comparison – Defect 6.

Location	Reference (MPa)	Defect (MPa)	Difference
2nd thread node	816.15	604.23	-26.0%
3rd thread node	138.29	150.4	+8.8%
4th thread node	77.715	86.059	+10.7%
Defect area – node A	N/A	505.06	—
Defect area – node B	N/A	242.94	—
Defect area – node C	N/A	240.46	—
Defect area – node D	N/A	144.42	—

6.8.1 Linear Elastic Stress

The second thread node for Defect 6 reads 604.23 MPa, which is 26% lower than the reference value of 816.15 MPa. This is a favourable result for the primary concern area. At the third thread, the value is 150.4 MPa compared to the reference of 138.29 MPa, an increase of 8.8%. The fourth thread reads 86.059 MPa against 77.715 MPa in the reference, an increase of 10.7%. Both differences at threads 3 and 4 are small in absolute terms and the values remain well below the yield strength.

The right-hand image shows the stress in the area around the defect. The highest reading in this zone is 505.06 MPa at the top of the defect region, followed by 242.94 MPa and 240.46 MPa at two mid-level nodes, and 144.42 MPa at the lower node. All these values are below the yield strength of Ti-64. The defect area is experiencing elevated stress relative to the surrounding background material, but none of the node values are at a level that would suggest the defect is critically stressed.

The contour pattern in the thread engagement region is broadly similar to the reference, with the same characteristic rapid decrease in stress going down the thread engagement length.

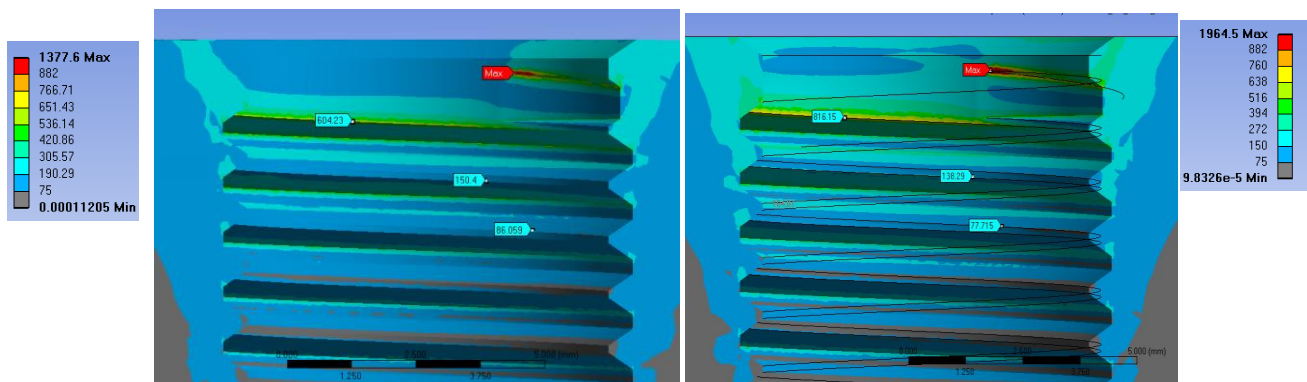


Figure 6.18. von Mises Linear stress comparison – Defect 6 vs Defect Free.

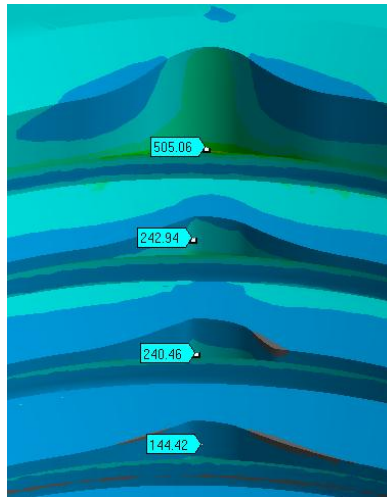


Figure 6.19. Stress distribution around the Defect 6 location.

6.8.3 Plastic Strain (Supporting Information)

Note: Plastic strain is shown for context only and is not used as a basis for assessment – see Section 6.1.

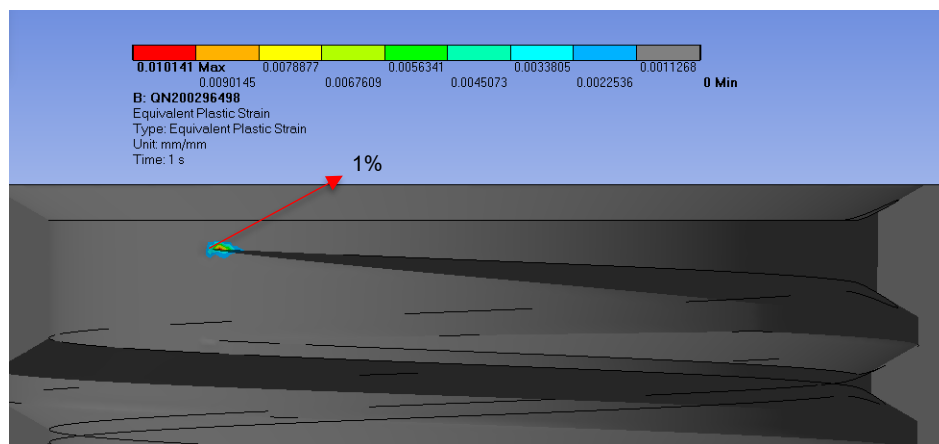


Figure 6.20. Equivalent Plastic Strain of max 1% – Defect 6 (shown for reference only).

6.8.4 Conclusion – Defect 6

The linear stress for Defect 6 is about 30% lower than the reference. The first two thread region shows a very similar stress pattern to the reference, and the stress around the defect area is not significantly elevated. Defect 6 does not make the stress condition of the connection worse than the defect-free case.

Assessment: ACCEPTABLE

7. Discussion and Conclusion

This chapter brings together the results from Chapter 6 and draws overall conclusions about the acceptability of each defect based on the linear stress contour comparison with the reference model.

7.1 Summary of Results

Table 7.1 below summarises the linear elastic stress at the selected thread nodes for each defect alongside the reference and states the overall verdict for each case.

Table 7.1. Summary of linear stress comparison and acceptability assessment for all defect cases.

Defect	2nd Thread (MPa)	3rd Thread (MPa)	4th Thread (MPa)	Verdict
Ref.	816.15	138.29	77.715	REFERENCE
D1	813.02	142.87	89.815	ACCEPTABLE
D2	626.84	137	78.878	ACCEPTABLE
D3	1562.8	136.87	75.829	NOT ACCEPTABLE
D4	854.81†	672.43	125.45	BORDERLINE
D5	773.6	214.97	147.11	BORDERLINE
D6	604.23	150.4	86.059	ACCEPTABLE

7.2 Discussion

7.2.1 Defect 1 and Defect 4 — Effect of Position on Void-Type Defects

Defects 1 and 4 are both void-type defects, but their outcomes are very different, and the reason comes down entirely to where each void is located in the connection.

Defect 1 is a large material void positioned in the lower section of the connection, below the main thread engagement zone. The selected node values in the first two threads (813.02 MPa and the reference-level readings further down) show that the void is not in the load path. The defect area node itself reads only 100.8 MPa, confirming the void sits in a region that barely participates in load transfer. Because the void is not in the way of the load, the connection behaves almost identically to the reference, and the defect is acceptable.

Defect 4, by contrast, is also a void — but a large circular one positioned directly within the thread engagement zone. The load has no choice but to travel around it, and this shows clearly in the node values. The void perimeter reads 993.27 MPa and the adjacent thread node reads 672.43 MPa, compared to just 138.29 MPa at the equivalent undisturbed thread position in the reference. The void is forcing significant stress redistribution across the engagement zone, which is why the assessment is borderline.

These two defects together illustrate a key principle: the geometry of a defect matters, but its position in the load path matters more. A large void away from the load path has virtually no effect on the connection. The same type of void inside the load path creates a significant stress redistribution problem that cannot be ignored.

7.2.2 Defect 2 and Defect 6 — Similar Defect Types, Consistent Results

Defects 2 and 6 show a notable similarity that is worth drawing out. Both produce a peak linear stress in a similar range (1382 MPa for Defect 2 and 1377.6 MPa for Defect 6), and both show a reduction in the second thread node stress compared to the reference (626.84 MPa for Defect 2 and 604.23 MPa for Defect 6, reductions of 23% and 26% respectively). At the third and fourth thread nodes, both defects track close to the reference values.

Both defects also have a "stress around the defect" region that can be assessed from the zoomed contour views. For Defect 2, the nodes in the circled defect zones range from 88.602 MPa to 272.04 MPa. For Defect 6, the equivalent nodes range from 144.42 MPa to 505.06 MPa. Defect 6 shows somewhat higher local stress at the defect than Defect 2, with the upper defect node reaching 505.06 MPa. However, even this higher value remains well below the yield strength of Ti-64, and the thread engagement region for both defects behaves similarly to the reference.

The consistency between these two results gives additional confidence in the acceptable verdict for both. Despite having different defect geometries and being at different positions in the connection, both produce the

same overall pattern: a reduced load at the second thread compared to the reference, near-reference behaviour at threads 3 and 4, and a defect area that is stressed but not critically so. This suggests that both defects are of a type and scale that does not significantly disrupt the load transfer through the connection.

7.2.3 Defect 3

Defect 3 is the most severe case in this study. The defect blocks are positioned directly within the thread engagement zone, which is the primary load-carrying region of the connection. The node values at the defect locations clearly show the impact — 1562.8 MPa at the first block where the reference carries 816.15 MPa, and 958.67 MPa and 474.88 MPa at the second and third blocks where the reference carries only 138.29 MPa. These represent increases of 91%, 593%, and 243% respectively. The removed material forces the remaining thread ligament to carry a load far beyond what it would experience in an intact connection, producing stress levels that exceed the yield strength of Ti-64 at two of the three block nodes.

On the undisturbed side of the connection, away from the blocks, the thread nodes return to near-reference values, showing the damage is localised. However, the severity at the block locations alone is sufficient to conclude the defect is not acceptable. The stress state in the critical engagement zone has been fundamentally altered, and the component must be repaired before entering service. Any defect with more material removal throughout the thread must be carefully assessed, as it is likely to lead to unacceptable results.

7.2.4 Defect 5

Defect 5 is a blended-type defect that reduces the wall thickness at a localised cross-section. The second thread node reads 773.6 MPa, slightly below the reference of 816.15 MPa, which appears reassuring at first. However, the third and fourth thread nodes are 55% and 89% above the reference respectively, showing that load is redistributing further down the engagement length to compensate for the reduced cross-section at the defect. The connection is relying on more threads than usual to carry the applied load.

The stress at the defect area reaches 751.19 MPa, which is close to the yield strength of Ti-64. This leaves a limited margin, and under repeated loading this location is a potential fatigue initiation site. The minimum remaining wall thickness at the defect cross-section is an additional factor that must be checked against the design limit before a final decision can be made. On stress data alone the assessment is borderline, and acceptance should not be confirmed until that wall thickness check is complete and an additional crack propagation analysis has been carried out.

7.3 Conclusion

This thesis aimed to develop a more realistic and less conservative way of evaluating defects in bolt hole threads. Instead of assuming that every defect behaves as a crack and using a stress concentration factor (K_t) together with the Neuber correction, the defect geometry was modelled directly in the FEA model and compared with a defect-free reference model. The conclusions are given below for each of the three research questions.

Research question 1 – types and classification of defects. In total, 191 non-conformances were found for the bolt hole threads of the IMC. They were divided into three categories: lack of material, thread damage and casting defects. Thread damage was the focus of this thesis, and the six most common and relevant defects were selected: a missing piece of thread material (Defect 1), breaks on the thread flanks (Defect 2), flattened thread crests (Defect 3), a large chunk of material missing at the thread entry (Defect 4), a blended-out surface around the thread hole (Defect 5) and small breaks on multiple thread flanks (Defect 6). Each defect was modelled in CAD with a simple representative shape, so that it could be evaluated with FEA. The location of the defect was also used for the classification. Defects 1, 2 and 6 are in lower-stress zones, so the main check

for them is the stress in the first two engaged threads. Defects 3, 4 and 5 are in or near higher-stress zones and therefore had to be checked at the defect location as well.

Research question 2 – effect of the explicit defect geometry on the von Mises stress. Modelling the defect geometry made it possible to see how much each defect changes the stress compared with the reference model. In the reference model the first two threads carry most of the load (816.15 MPa at the second thread node), and the stress drops quickly to 138.29 MPa and 77.715 MPa at the third and fourth thread nodes. Defects 1, 2 and 6 did not increase the stress in the first two threads: the second thread node was 813.02 MPa for Defect 1, 626.84 MPa for Defect 2 and 604.23 MPa for Defect 6, and the stress around the defects stayed below the yield strength of Ti-64 (823–866 MPa). These three defects are therefore acceptable. Defect 3 is not acceptable. The node at the first defect block reached 1562.8 MPa, which is 91.5% higher than the reference at the same thread level, and two of the three block nodes were above the yield strength of Ti-64. Defects 4 and 5 are borderline. In Defect 4 the void forces the load to go around it, which gives 993.27 MPa at the void perimeter and 672.43 MPa at the third thread level, compared with 138.29 MPa in the reference. In Defect 5 the second thread node was lower than the reference (773.6 MPa), but the third and fourth thread nodes were 55% and 89% higher, and the stress at the defect (751.19 MPa) is close to the yield strength.

The comparison of Defect 1 and Defect 4 also shows that the position of a defect in the load path is more important than the type of defect. Both are voids, but only Defect 4, which lies inside the thread engagement zone, changed the stress distribution significantly. Since Defects 1, 2 and 6 did not increase the stress in the first two threads, not all defects need to be handled with the conservative crack assumption, and components with defects like these may be kept in service. This is in line with the aim of reducing unnecessary scrapping, although the acceptance here is based on the comparison with the defect-free reference model only.

Research question 3 – most reliable stress metric. The linear elastic von Mises stress at selected nodes was the most reliable and consistent metric for comparing the severity of the defects. The nonlinear stress is limited by yielding, so the peak value is close to the yield strength of Ti-64 in all cases and cannot be used to rank the defects. The equivalent plastic strain depends strongly on the mesh size at the defect tips, where stress singularities exist, so it is not reliable without a full mesh convergence study. Even the reference model without any defect showed a plastic strain of 1.8% at the first thread root. The linear elastic stress is not limited by yielding, so it shows the size of the stress concentration and allows a direct node-by-node comparison with the reference model. The nonlinear stress and the plastic strain were therefore only used as supporting information.

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