

**ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL**

**INVESTIGATION OF THE MECHANICAL BEHAVIOUR OF SUPERALLOY  
BOLTED JOINTS UNDER EXTREME THERMAL CONDITIONS FOR  
AEROSPACE APPLICATIONS**



**M.Sc. THESIS**

**Tahsin TOPAL**

**Department of Mechanical Engineering**

**Mechanical Design Programme**

**JUNE 2026**



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**Tahsin TOPAL  
(503181230)**

**Department of Mechanical Engineering**

**Mechanical Design Programme**

**Thesis Advisor: Prof. Dr. Cevat Erdem İMRAK**

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**İSTANBUL TEKNİK ÜNİVERSİTESİ ★ LİSANSÜSTÜ EĞİTİM ENSTİTÜSÜ**

**HAVACILIK UYGULAMALARINDAKİ SÜPER ALAŞIM CIVATALI  
BAĞLANTILARIN ZORLU TERMAL KOŞULLAR ALTINDA MEKANİK  
DAVRANIŞININ İNCELENMESİ**

**YÜKSEK LİSANS TEZİ**

**Tahsin TOPAL  
(503181230)**

**Makina Mühendisliği Anabilim Dalı**

**Konstrüksiyon Programı**

**Tez Danışmanı: Prof. Dr. Cevat Erdem İMRAK**

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Tahsin TOPAL, a M.Sc. student of İTÜ Graduate School student ID 503181230, successfully defended the thesis/dissertation entitled “INVESTIGATION OF THE MECHANICAL BEHAVIOUR OF SUPERALLOY BOLTED JOINTS UNDER EXTREME THERMAL CONDITIONS FOR AEROSPACE APPLICATIONS”, which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

**Thesis Advisor :**      **Prof. Dr. Cevat Erdem İMRAK** .....  
Istanbul Technical University

**Jury Members :**      **Assoc. Prof. Dr. C. Oktay AZELOĞLU** .....  
Yıldız Technical University

**Assis. Prof. Dr. Adem CANDAŞ** .....  
Istanbul Technical University

**Date of Submission : 4 June 2026**  
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*To my spouse and family,*



## FOREWORD

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(Makina Mühendisi)



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## ABBREVIATIONS

|                 |                                                                     |
|-----------------|---------------------------------------------------------------------|
| <b>AMS</b>      | : Aerospace Material Specification                                  |
| <b>AMS 5663</b> | : Aerospace Material Specification 5663 (Inconel 718 specification) |
| <b>ANSYS</b>    | : ANSYS (Finite Element Analysis software suite)                    |
| <b>CAD</b>      | : Computer-Aided Design                                             |
| <b>EB</b>       | : Electron Beam                                                     |
| <b>FEA</b>      | : Finite Element Analysis                                           |
| <b>HAZ</b>      | : Heat-Affected Zone                                                |
| <b>HCF</b>      | : High-Cycle Fatigue                                                |
| <b>IN718</b>    | : Inconel 718 (Nickel-based superalloy)                             |
| <b>LCF</b>      | : Low-Cycle Fatigue                                                 |
| <b>MS</b>       | : Margin of Safety                                                  |
| <b>NDT</b>      | : Non-Destructive Testing                                           |
| <b>NLGEOM</b>   | : Geometric Nonlinearity (Large-Deflection option in ANSYS)         |
| <b>SWT</b>      | : Smith–Watson–Topper (Fatigue parameter)                           |
| <b>TIG</b>      | : Tungsten Inert Gas (Welding)                                      |
| <b>TLC</b>      | : Transverse Load Capability                                        |
| <b>TMF</b>      | : Thermo-Mechanical Fatigue                                         |
| <b>UNS</b>      | : Unified Numbering System                                          |
| <b>UT</b>       | : Ultrasonic Testing                                                |
| <b>VDI</b>      | : Verein Deutscher Ingenieure                                       |
| <b>VDI 2230</b> | : Systematic Calculation of High Duty Bolted Joints                 |



## SYMBOLS

|               |                                                  |
|---------------|--------------------------------------------------|
| $\rho$        | : Density [kg/m <sup>3</sup> ]                   |
| $E$           | : Young's modulus [GPa or N/mm <sup>2</sup> ]    |
| $\nu$         | : Poisson's ratio [-]                            |
| $\alpha$      | : Coefficient of thermal expansion [1/°C]        |
| $k$           | : Thermal conductivity [W/(m·°C)]                |
| $C_p$         | : Specific heat capacity [J/(kg·°C)]             |
| $\mu$         | : Coefficient of friction [-]                    |
| $D$           | : Bolt nominal diameter [mm]                     |
| $d$           | : Nominal diameter (in equations) [mm]           |
| $p$           | : Thread pitch [mm]                              |
| $A_t$         | : Tensile stress area of bolt [mm <sup>2</sup> ] |
| $A_s$         | : Stress area [mm <sup>2</sup> ]                 |
| $l_g$         | : Grip length [mm]                               |
| $t$           | : Flange thickness (single member) [mm]          |
| $D_b$         | : Bolt head diameter [mm]                        |
| $D_h$         | : Flange hole diameter [mm]                      |
| $d_h$         | : Hole diameter (including chamfer) [mm]         |
| $D_{bc}$      | : Bolt circle diameter [mm]                      |
| $D_r$         | : Rabbet diameter [mm]                           |
| $l_r$         | : Rabbet engagement/contact length [mm]          |
| $L_{lig}$     | : Flange ligament length [mm]                    |
| $t_{flange}$  | : Flange thickness (ligament definition) [mm]    |
| $L_{rb}$      | : Rabbet net-section engagement length [mm]      |
| $t_{rb}$      | : Rabbet thickness [mm]                          |
| $T$           | : Applied tightening torque [N·m]                |
| $K$           | : Nut factor [-]                                 |
| $F_p$         | : Target preload [N]                             |
| $F_{initial}$ | : Initial preload [N]                            |
| $F_{final}$   | : Operating-temperature preload [N]              |

|                         |                                                          |
|-------------------------|----------------------------------------------------------|
| $\Delta F_p$            | : Preload loss ( $F_{initial} - F_{final}$ ) [N]         |
| $\Delta F_{th}$         | : Thermally induced preload change [N]                   |
| $F_{b,t}$               | : Total bolt tensile force (from reactions) [N]          |
| $F_{b,max}$             | : Maximum bolt/constraint reaction force [N]             |
| $F_{CL}$                | : Clamping/contact normal force between flange faces [N] |
| $F_{ext}$               | : External separating force resultant [N]                |
| $F_x$                   | : Force component in x-direction [N]                     |
| $F_y$                   | : Force component in y-direction [N]                     |
| $F_z$                   | : Force component in z-direction [N]                     |
| $L_r$                   | : Radial shear load [N]                                  |
| $L_t$                   | : Tangential/transverse shear load [N]                   |
| $F_{shear,total}$       | : Resultant shear load [N]                               |
| $F_{friction,capacity}$ | : Maximum transferable shear by friction [N]             |
| $F_{r,assy}$            | : Rabbet radial reaction (assembly) [N]                  |
| $F_{r,op}$              | : Rabbet radial reaction (operation) [N]                 |
| $k_b$                   | : Axial stiffness of bolt [N/mm]                         |
| $k_m$                   | : Axial stiffness of clamped members [N/mm]              |
| $k_{single}$            | : Effective stiffness of a single flange member [N/mm]   |
| $k_{total,assembly}$    | : Total series stiffness (assembly) [N/mm]               |
| $k_{total,op}$          | : Total series stiffness (operation) [N/mm]              |
| $\Phi$                  | : Joint stiffness constant [-]                           |
| $\delta_{initial}$      | : Total displacement equivalent (assembly) [mm]          |
| $\delta_{final}$        | : Total displacement equivalent (operation) [mm]         |
| $\delta_{embed}$        | : Embedding displacement [mm]                            |
| $\Delta L_{thermal}$    | : Differential thermal elongation [mm]                   |
| $T_{op,bolt}$           | : Operating bolt temperature [°C]                        |
| $T_{op,flange}$         | : Operating flange temperature [°C]                      |
| $\Delta T_b$            | : Bolt temperature change [°C]                           |
| $\Delta T_f$            | : Flange temperature change [°C]                         |
| $\Delta T$              | : Temperature difference [°C]                            |
| $\alpha_b$              | : Bolt CTE [1/°C]                                        |
| $\alpha_f$              | : Flange CTE [1/°C]                                      |
| $E_b$                   | : Bolt Young's modulus [N/mm <sup>2</sup> ]              |
| $E_m$                   | : Member Young's modulus [N/mm <sup>2</sup> ]            |

|                        |                                                                            |
|------------------------|----------------------------------------------------------------------------|
| $A_b$                  | : Effective bolt area in thermal relation [mm <sup>2</sup> ]               |
| $\sigma_y$             | : 0.2% yield strength [MPa]                                                |
| $\sigma_{UTS}$         | : Ultimate tensile strength [MPa]                                          |
| $\sigma_{vm}$          | : Von Mises equivalent stress [MPa]                                        |
| $\sigma_b$             | : Bolt axial stress ( $\sigma_b = F_b / A_t$ ) [MPa]                       |
| $\sigma_{b,assy}$      | : Bolt axial stress (assembly) [MPa]                                       |
| $\sigma_{b,op}$        | : Bolt axial stress (operation) [MPa]                                      |
| $\sigma_z$             | : Linearized normal stress component [MPa]                                 |
| $\sigma_{z,max}$       | : Maximum normal stress [MPa]                                              |
| $\sigma_{z,min}$       | : Minimum normal stress [MPa]                                              |
| $\sigma_{bend,FEA}$    | : Bending stress component derived from FEA [MPa]                          |
| $M_b$                  | : Bolt bending moment [N·mm]                                               |
| $I_{thread}$           | : Area moment of inertia at thread-root section [mm <sup>4</sup> ]         |
| $c_{thread}$           | : Distance from neutral axis to outer fiber (thread-root section) [mm]     |
| $\sigma_{bend,thread}$ | : Thread-root bending stress [MPa]                                         |
| $\sigma_{tensile}$     | : Mean axial tensile stress component [MPa]                                |
| $\sigma_{max}$         | : Maximum fiber stress ( $\sigma_{tensile} + \sigma_{bend,thread}$ ) [MPa] |
| $A_{min}$              | : Minimum stress area [mm <sup>2</sup> ]                                   |
| $A_{br}$               | : Bearing/contact area under the bolt head [mm <sup>2</sup> ]              |
| $\sigma_{br}$          | : Bearing (crushing) stress [MPa]                                          |
| $\sigma_{br,assy}$     | : Bearing stress (assembly) [MPa]                                          |
| $\sigma_{br,op}$       | : Bearing stress (operation) [MPa]                                         |
| $\sigma_{f,bend}$      | : Flange ligament bending stress [MPa]                                     |
| $I_f$                  | : Area moment of inertia of flange ligament section [mm <sup>4</sup> ]     |
| $c_{flexure}$          | : Distance from neutral axis to outer fiber (flexure) [mm]                 |
| $M_f$                  | : Resultant bending moment [N·mm]                                          |
| $M_x$                  | : Bending moment component about x-axis [N·mm]                             |
| $M_y$                  | : Bending moment component about y-axis [N·mm]                             |
| $M_z$                  | : Bending moment component about z-axis [N·mm]                             |
| $\sigma_{rb,bend}$     | : Rabbet net-section bending stress [MPa]                                  |
| $I_{rb}$               | : Area moment of inertia of rabbet section [mm <sup>4</sup> ]              |
| $\sigma_{r,br}$        | : Rabbet bearing (crushing) stress [MPa]                                   |
| $\sigma_{r,br,assy}$   | : Rabbet bearing stress (assembly) [MPa]                                   |

|                        |                                                                   |
|------------------------|-------------------------------------------------------------------|
| $\sigma_{r,br,op}$     | : Rabbet bearing stress (operation) [MPa]                         |
| $A_{r,sec}$            | : Effective rabbet contact area (sector model) [mm <sup>2</sup> ] |
| $K_t$                  | : Stress concentration factor [-]                                 |
| $\sigma_{nominal,max}$ | : Maximum nominal equivalent stress [MPa]                         |
| $\sigma_{nominal,min}$ | : Minimum nominal equivalent stress [MPa]                         |
| $\sigma_{max,elastic}$ | : Local maximum stress (fully elastic assumption) [MPa]           |
| $\sigma_{min,elastic}$ | : Local minimum stress (fully elastic assumption) [MPa]           |
| $\Delta\sigma_{local}$ | : Local stress range at thread-root [MPa]                         |
| $\sigma_{max,elastic}$ | : Stabilized maximum stress (post-shakedown) [MPa]                |
| $\sigma_{min,stab}$    | : Stabilized minimum stress (post-shakedown) [MPa]                |
| $\sigma_a$             | : Stress amplitude [MPa]                                          |
| $\sigma_m$             | : Mean stress [MPa]                                               |
| $\sigma_{f'}$          | : Fatigue strength coefficient [MPa]                              |
| $b$                    | : Fatigue strength exponent [-]                                   |
| $N_f$                  | : Cycles to failure [cycle]                                       |
| $2N_f$                 | : Reversals to failure [reversal]                                 |
| $\Delta\epsilon/2$     | : Strain amplitude [-]                                            |
| $\epsilon_{f'}$        | : Fatigue ductility coefficient [-]                               |
| $c_{fatigue}$          | : Fatigue ductility exponent [-]                                  |

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# **INVESTIGATION OF THE MECHANICAL BEHAVIOUR OF SUPERALLOY BOLTED JOINTS UNDER EXTREME THERMAL CONDITIONS FOR AEROSPACE APPLICATIONS**

## **SUMMARY**

The effort to increase thermal efficiency in modern aviation propulsion systems brings with it extremely aggressive operating conditions that push the structural limits of engine components. Bolted flange connections located in the hot regions of gas turbine engines are not merely simple connecting elements. They are dynamic thermodynamic interfaces that must maintain sealing and rigidity at critical temperatures such as 700°C. This thesis investigates in-depth the thermomechanical behavior of a two-flange bolted connection group made of Inconel 718 superalloy under demanding service conditions. The literature generally presents simplified strength approaches. This study utilizes analytical methods based on standards such as VDI 2230 and NASA-STD-5020B. Furthermore, a hybrid methodology providing cross-validation with high-precision 3D Finite Element Analysis (FEA) performed using ANSYS software has been adopted.

The phenomenon of pre-stress relaxation, often overlooked in standard design processes, is one of the focal points of this research. The analyses revealed significant losses in pre-load strength during operation. This is largely due to the decrease in the elastic modulus of the materials along with different amounts of thermal expansion. FEA simulations predicted a pre-stress loss of 18.65% due to thermal effects alone. Furthermore, this value was calculated as 20.38% in analytical hand calculations, and these results largely coincide. This consistency supports the validity of the numerical model, but also reveals that a significant strength loss occurs in the design even before long-term creep effects come into play.

The structural evaluation of the connection also highlights the risks of relying on uniaxial stress controls. The bolt appears to have a high safety margin of 117% under pure tensile load. However, when the bending moments created by flange rotation are taken into account, this margin drops to around 40% at the thread roots. This demonstrates how critical the "prying action" is in high-temperature designs. Furthermore, analyses determined that the flange net cross-section is the weakest link in the structure with a safety margin of 19.15%. While secondary alignment features such as rabbets were designed with an overly safe margin of over 1000%, indicating suitability for optimization studies in these areas.

Finally, the fatigue life of the connection element was also calculated in this study. In this section, the strain-life approach was evaluated using the Smith-Watson-Topper (SWT) damage parameter. The material was assessed to exhibit local yielding or "shakedown" behavior during the initial assembly and warm-up cycles. In subsequent operating cycles, the stresses were found to stabilize within elastic limits. The calculated life of 73.4 million cycles confirms that the connection achieves a very long fatigue life, well beyond typical service requirements, as long as separation does not occur. The sealing function was further confirmed through a deliberately severe worst-

case scenario, in which two of the twenty bolts were assumed lost and the remainder installed at the minimum permissible preload. Even then, the joint retained a separation margin of 1359%, showing that flange opening is effectively impossible within the operating envelope. A complementary transverse-load check showed that interface friction resists the maneuver-induced shear with a margin of 101.67%, confirming that the connection acts as a pure friction-grip joint in which the bolts are never exposed to direct shear. These findings provide a validated design methodology that bridges the gap between static theory and the dynamic reality of gas turbines.



# **HAVACILIK UYGULAMALARINDAKİ SÜPER ALAŞIM CİVATALI BAĞLANTILARIN ZORLU TERMAL KOŞULLAR ALTINDA MEKANİK DAVRANIŞININ İNCELENMESİ**

## **ÖZET**

Havacılık ve uzay teknolojilerindeki durdurulamaz ilerleme, modern gaz türbini motorlarının termal verimlilik sınırlarını her geçen gün daha da yukarı taşımaktadır. Mühendislik dünyasındaki bu verimlilik arayışı, kaçınılmaz olarak motorun sıcak bölgelerinde (hot-section) görev yapan bileşenlerin maruz kaldığı çalışma sıcaklıklarını 700°C ve üzerine çıkarmaktadır. Bu ekstrem termal ortam, motorun modüler mimarisini bir arada tutan ve yapısal bütünlüğü sağlayan cıvatalı flanş bağlantıları için benzeri görülmemiş zorluklar doğurmaktadır. Geleneksel endüstriyel makinelerin aksine, bir uçak motorundaki flanş bağlantısı, sadece statik yükleri taşıyan pasif bir eleman değil; farklı termal genleşme, modül düşüşü ve dinamik uçuş yükleri altında sızdırmazlığı korumak zorundadır. Aynı zamanda hayati öneme sahip dinamik bir termodinamik sistemdir. Bu tez çalışması, Inconel 718 (IN718) süper alaşımı kullanılarak tasarlanmış kritik bir flanş bağlantısının, bahsi geçen agresif işletme koşulları altındaki termo-mekanik davranışını, yapısal bütünlüğünü ve yorulma ömrünü tüm yönleriyle aydınlatmayı hedeflemektedir.

Mevcut literatür incelendiğinde, cıvatalı bağlantıların analizi konusunda genellikle iki uç yaklaşımın benimsendiği görülmektedir: Ya aşırı basitleştirilmiş analitik el hesapları ya da doğrulanmamış karmaşık sayısal modeller. Bu çalışma, söz konusu metodolojik boşluğu doldurmak adına "hibrit" bir doğrulama stratejisi üzerine inşa edilmiştir. Çalışmanın analitik ayağında, cıvatalı bağlantıların mekaniği konusunda küresel kabul gören VDI 2230 ve NASA-STD-5020B standartları temel alınarak detaylı rijitlik ve mukavemet hesaplamaları yapılmıştır. Ancak bu yöntemler, doğaları gereği lineer varsayımlara dayanır. Aynı zamanda gaz türbinlerinin kaotik termal gradyanlarını tam olarak modelleyemediği için, çalışma ANSYS Workbench ortamında kurulan yüksek hassasiyetli bir 3D Sonlu Elemanlar Analizi (FEA) ile desteklenmiştir. Sayısal modelde, hesaplama zamanını optimize etmek adına 60 derecelik bir sektör model kullanılmıştır. Ancak kritik gerilme yığılması bölgeleri olan dış dipleri ve flanş geçiş radyuslarında ağ yapısı (mesh) özellikle sıkılaştırılarak analiz hassasiyeti maksimize edilmiştir. Analiz süreci; montaj (assembly), ön yükleme (preloading), basınçlandırma ve termal yükleme olmak üzere birbirini takip eden dört aşamalı, doğrusal olmayan bir kurguyla gerçekleştirilmiştir.

Araştırmanın en kritik bulgularından biri, cıvata ön gerilmesinin (preload) termal etkiler altındaki evrimi ile ilgilidir. Bir cıvatalı bağlantının yorulma ömrünü ve sızdırmazlık kapasitesini belirleyen temel parametre, montaj sırasında uygulanan kenetleme kuvvetinin işletme sırasında ne kadarının korunabildiğidir. Yapılan analizler, henüz herhangi bir dış mekanik yük veya uzun süreli sünme (creep) etkisi devreye girmeden bile, sadece sistemin 700°C sıcaklığa ulaşmasının ciddi bir kuvvet kaybına yol açtığını göstermiştir. Bu kaybın temel mekanizması, Inconel 718

malzemesinin elastisite modülünün yüksek sıcaklıkta oda sıcaklığına göre yaklaşık %20 oranında düşmesidir. Ayrıca cıvata ile flanş arasındaki termal genleşme farklarının yarattığı uyumsuzluktur. Sayısal analizler (FEA), işletme koşullarında ön gerilmenin %18,65 oranında azaldığını öngörürken, analitik yöntemlerle hesaplanan kayıp oranı %20,38 seviyesindedir. Tamamen farklı iki metodolojiden elde edilen sonuçlar arasında %2'nin altında bir sapma tespit edilmiştir. Bu yüksek tutarlılık, elde edilen sayısal modelin güvenilirliğini kanıtlamaktadır. Aynı zamanda termal gevşemenin (thermal relaxation) tasarım aşamasında mutlaka bir "tasarım faktörü" olarak tork değerlerine yansıtılması gerektiğini net bir şekilde ortaya koymaktadır.

Bağlantının yapısal dayanımı incelendiğinde, çalışmanın belki de en önemli mühendislik derslerinden biri ile karşılaşılmaktadır. Bu, basitleştirilmiş tek eksenli gerilme kontrollerinin yetersizliğidir. Cıvatalar sadece eksenel çekme yükleri (axial tensile stress) açısından değerlendirildiğinde, %117,36 gibi oldukça güvenli bir güvenlik marjına sahip görünmektedir. Bu sonuç, ilk bakışta tasarımın çok güvenli olduğu yanılgısını yaratabilir. Ancak, FEA sonuçları derinlemesine incelendiğinde, iç basınç etkisiyle flanşların dışa doğru dönme eğilimi gösterdiği ve bu durumun cıvata gövdesinde ciddi bir eğilme momenti (bending moment) yarattığı tespit edilmiştir. Literatürde "manivela etkisi" (prying action) olarak bilinen bu fenomen hesaba katılmıştır. Ardından cıvata dış diplerindeki maksimum fiber gerilmesi (maximum fiber stress) hesaplandığında, güvenlik marjının dramatik bir şekilde %39,83 seviyesine düştüğü görülmüştür. %117'den %40'a inen bu keskin düşüş, havacılık gibi güvenlik sınırlarının dar olduğu sektörlerde, flanş dönmesinin ve eğilme gerilmelerinin ihmal edilmesinin ne kadar tehlikeli sonuçlar doğurabileceğini somut bir veriyle kanıtlamaktadır.

Sistemin bütünsel yapısal analizi, tasarımın "zayıf halkasını" da ortaya çıkarmıştır. Beklenilenin aksine, en kritik bölge cıvatalar değil, flanşın kendisidir. Özellikle cıvata delikleri arasında kalan ve "ligament" olarak adlandırılan net kesit alanı, flanşın bükülme direncini belirleyen ana unsurdur. Bu bölgedeki eğilme gerilmeleri analiz edildiğinde, güvenlik marjının %19,15 seviyesinde olduğu hesaplanmıştır. Bu değer pozitif olduğu için tasarım güvenlidir, ancak diğer tüm bölgelere kıyasla en düşük marja sahiptir. Dolayısıyla yapının limitlerini belirlemektedir. Buna tam tezat oluşturacak şekilde, flanşların merkezlemesini sağlayan "rabbet" (faturalı geçme) bölgesi ve cıvata başı altındaki ezilme (bearing) yüzeyleri incelendiğinde, %1000 ve üzeri gibi mühendislik açısından "aşırı güvenli" (over-designed) kabul edilebilecek marjlar tespit edilmiştir. Bu bulgu, gelecekteki tasarım iyileştirme çalışmalarında rabbet kalınlığının veya flanş dış çapının azaltılarak ciddi bir ağırlık tasarrufu sağlanabileceğine dair tasarımcılara net bir optimizasyon rotası çizmektedir.

Bir gaz türbini flanş bağlantısının olmazsa olmazı, her koşulda sızdırmazlığı sağlaması ve flanşların birbirinden ayrılmasını (separation) önlemesidir. Bu kritik fonksiyonu doğrulamak adına, tezin kapsamında "En Kötü Durum" (Worst-Case) senaryosu çalışılmıştır. Bu senaryo, işletme sırasında olası bir hata sonucu 20 cıvata 2'sinin tamamen koptuğunu veya gevşediğini öngörmektedir. Ayrıca montaj torkunun da toleransların en alt sınırında (minimum preload) uygulandığını varsaymaktadır. Bu kadar ekstrem bir senaryoda dahi, bağlantının dış yükler altında ayrılmaya karşı %1359 gibi devasa bir güvenlik marjına sahip olduğu hesaplanmıştır. Benzer şekilde, motorun manevra yüklerinden kaynaklanan kesme kuvvetlerine karşı, flanş yüzeyleri arasındaki sürtünme direnci kontrol edilmiş (Transverse Load Capability - TLC) ve %101,67'lik bir marjla bağlantının kaymaya karşı güvenli olduğu doğrulanmıştır. Bu sonuçla birlikte flanşların birbiri üzerinde asla kaymayacağı doğrulanmıştır. Bu

sonuçlar, cıvataların işletme ömrü boyunca asla doğrudan kesme yüküne maruz kalmayacağını ve bağlantının tam bir "sürtünme etkili" (friction-grip) birleşim olarak çalışacağını garanti etmektedir.

Son olarak, bu çalışmanın kapsamlı analizlerinden biri olan yorulma ömrü değerlendirmesidir. Smith-Watson-Topper (SWT) hasar parametresi kullanılarak gerinim esaslı (strain-life) bir yaklaşımla gerçekleştirilmiştir. Yüksek sıcaklık ve yüksek ortalama gerilme koşulları, malzemelerin yorulma davranışını karmaşıklaştıran faktörlerdir. Analizler, montaj sırasındaki ilk sıkma işlemi ve hemen ardından gelen termal yüklemeye birlikte, diş diplerindeki gerilme yığılması bölgelerinde yerel bir akma (plastik deformasyon) yaşandığını göstermiştir. Ancak bu durum, malzemenin "shakedown" (uyum sağlama) mekanizması sayesinde stabil bir hale gelmektedir. Yani malzeme, ilk çevrimde yaşadığı bu küçük plastik şekil değişiminden sonra kendini toparlamakta ve motorun sonraki tüm çalışma döngülerinde (başlatma, uçuş, durdurma) tamamen elastik sınırlar içinde kalmaktadır. Elastik bölgeye oturan bu kararlı gerilme döngüsü üzerinden yapılan hesaplamalar, cıvatanın yorulma ömrünün yaklaşık 73,4 milyon çevrim olduğunu göstermektedir. Havacılık motorları için tipik ömür gereksinimlerinin çok üzerinde olan bu değer, cıvatanın teorik olarak sonsuz ömür (infinite life) bölgesinde çalıştığını ve termal döngülerden kaynaklı bir metal yorgunluğu riskinin bulunmadığını doğrulamaktadır.

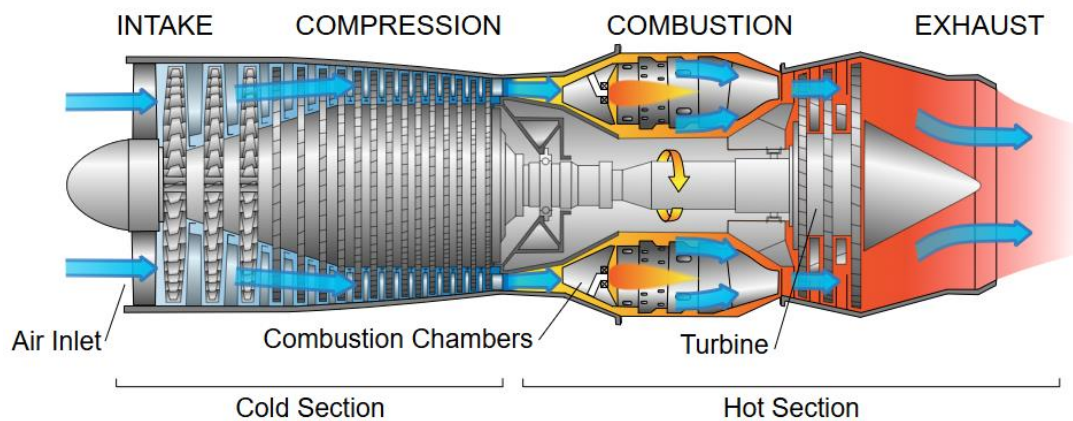
Özetle bu tez çalışması, 700°C gibi zorlu bir termal sınırdaki çalışan havacılık bağlantılarının tasarımında, statik ders kitabı varsayımlarının yetersiz kaldığını göstermektedir. Termal modül düşüşü, diferansiyel genleşme ve flanş dönmesi gibi lineer olmayan etkilerin tasarımın kaderini belirlediğini ortaya koymuştur. Analitik ve sayısal yöntemlerin birbirini %98'in üzerinde bir uyumla doğruladığı bu hibrit yaklaşım, gelecekteki gaz türbini motor tasarımları için güvenilir, optimize edilebilir ve doğrulanmış bir mühendislik çerçevesi sunmaktadır.



## 1. INTRODUCTION

### 1.1 Gas Turbine Engines in Aerospace Applications

Modern gas turbine engines aim to increase turbine inlet temperatures to achieve higher thermal efficiency and thrust-to-weight ratios [1]. This strategic methodology creates extremely demanding conditions for engine hot-section components, which must operate reliably at temperatures approaching  $700^{\circ}\text{C}$ . Bolted joints are a fundamental component of engine architecture. They ensure structural integrity while also facilitating maintenance, since they can be disassembled and reassembled. These challenging conditions also make the design of bolted joints that provide safe and long-lasting operation extremely challenging. This development is based on the need to increase efficiency, reduce fuel consumption, lower emissions, and improve reliability [2]. Following these developments, companies have made significant profits and increased their market share in this sector. These goals have led to changes in aerodynamic and structural design criteria and pioneered the emergence of more complex, advanced technology engines. Figure 1.1 provides a station-based schematic of a representative gas turbine.



**Figure 1.1 :** Schematic diagram of a modern gas turbine engine [3].

In terms of the efficiency of gas turbine engines, the thermodynamic efficiency of the Brayton cycle is a critical factor [4]. Turbine inlet temperature significantly impacts this efficiency. High temperatures at the turbine inlet improve fuel economy and

overall aircraft performance. This increases thermal efficiency and specific thrust values. Therefore, new generation gas turbine engines are designed with combustion chambers and turbine components capable of operating at high temperatures. Components operating under these conditions must be cooled efficiently. Despite advanced cooling systems, the metallic components of modern aviation engines can reach temperatures up to 1000 °C or even exceed this temperature.

To reach high operating temperatures, nickel-based superalloys, thermal barrier coatings, and complex internal cooling systems are widely used [5]. These technologies have significantly increased the thermal resistance of engine components. But this has also brought new challenges such as structural integrity, life, and manufacturability.

Gas turbine engines consist of various modules and each module has its own unique function [6]. Examples of these systems include the fan, low and high-pressure compressors, combustion chamber, high and low-pressure turbines, and exhaust system. Different modules are shown in Figure 1.1. Each module has its own specific design challenges. These modules are connected with different types of interfaces. Bolted joints, flanges, splines, and other mechanical features are the main connection elements used in module assembly. Bolted joints are one of the most common connection elements used in aero engines [7]. Bolted flange connections keep the structure secure and allow the engine to be disassembled, repaired, and reassembled with the same or new parts. This is why they are frequently preferred. The proper functioning of bolted connections is of direct importance in terms of the reliability and long-term costs of propulsion systems in aviation. Gas turbine engines consist of modular structures due to the strict maintenance and certification standards in aviation applications. Aviation engines require frequent inspection and part replacement throughout their service life. Other industrial machinery does not require this level of maintenance and part replacement. Bolts are directly important for the proper functioning of connections, the reliability of propulsion systems in aviation, and long-term costs.

Increasing temperatures in modern engines expose bolted connections to the most demanding conditions. Bolted connections in contact with high temperatures are threatened in their mechanical performance by factors such as high thermal conditions, sudden temperature changes, and repeated thermal stress cycles. As engine designs

approach their thermal limits, the durability of bolted connections becomes more than just an additional feature; it becomes a critical element of the design. Significant reductions in strength and service life occur.

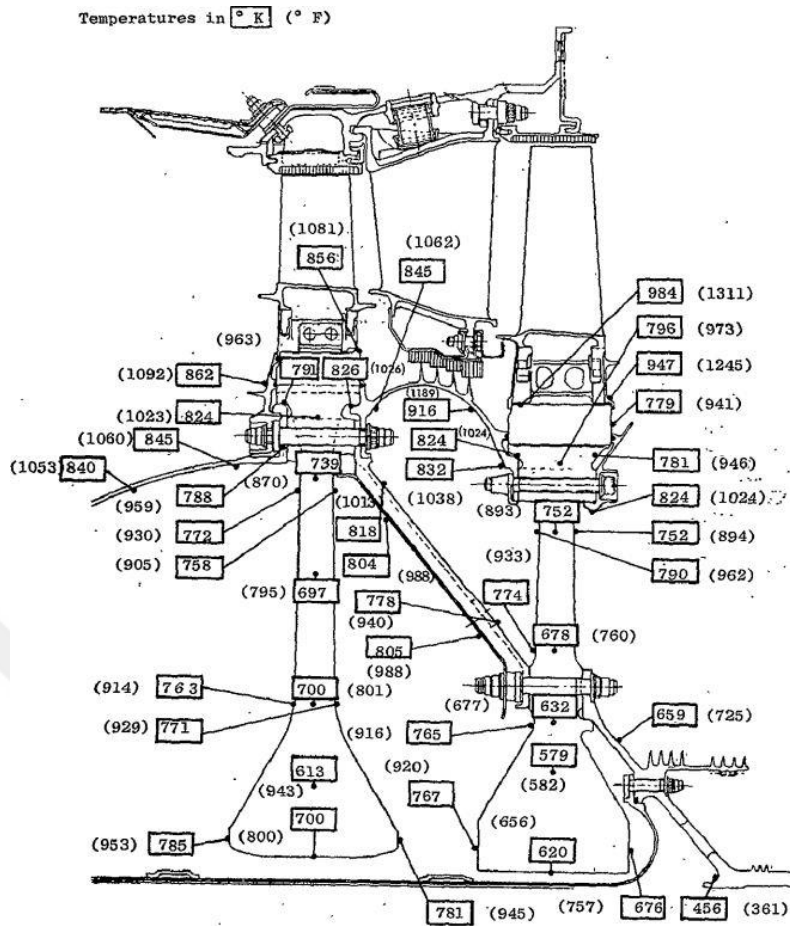
## **1.2 Thermo-Mechanical Environment of Gas Turbine Structures**

Gas turbine structures operate under combined thermal, mechanical, and dynamic loadings. Internal pressure, centrifugal forces, aerodynamic loads, vibration, and limited thermal expansion act together. The load condition is not constant throughout the engine; it also changes throughout the operating cycle. Therefore, connection points and fasteners encounter a challenging and time-dependent environment.

Thermal loading is often the primary factor because the temperature field is inherently non-homogeneous. Hot gas path regions are exposed to very high gas temperatures, while adjacent structural parts are cooled by internal airflow, external convection, and local insulation. This leads to temperature gradients within the components and, more importantly, across the interfaces. The resulting thermal stresses can be similar to mechanically induced stresses in hot section regions [8]. Therefore, treating the structure as isothermal may underestimate interface-related effects. Figure 1.2 illustrates a representative non-uniform temperature field in a hot-section parts.

Transient operation further increases the severity of thermal loading. Starting and shutting down cause rapid heating and cooling. Different components respond at different rates due to differences in thermal mass, hot gas exposure, and cooling efficiency. This mismatch causes additional stresses at the interfaces.

There are various sources of mechanical loading. Fuselage pressure creates circumferential and axial stresses that must be carried by the fuselage and transmitted through the connections. Rotating components create centrifugal forces transmitted through shafts, discs, and supports. Aerodynamic forces on blades and propellers create fluctuating loads that can trigger vibration over a wide frequency range [9]. These fluctuations are important for connection elements because small alternating loads can add to high average preloading and contribute to high cyclic fatigue over many cycles.



**Figure 1.2 :** Temperature distribution of the CF6 engine hot section [10].

Design requirements are not limited to normal operation. Engines must also maintain structural integrity under abnormal events, for example rotor imbalance, bird strikes, and blade detachment. These events are tested in certification tests, and the engine must withstand these conditions during these tests [11]. These type of scenarios create large transient forces transmitted through the fuselage and bolted connections. Therefore, connection performance under such events is critical not only for durability but also for safety.

The combined thermomechanical state generates complex stresses in engine structures. Multiaxial loading, local bending, and stress concentrations develop in regions where geometric discontinuities exist. Temperature-dependent material properties further affect stiffness, strength, and load sharing between bolts and contacting elements. Under these conditions, simplified uniaxial approaches cannot represent connection behavior with sufficient accuracy and require detailed investigation.

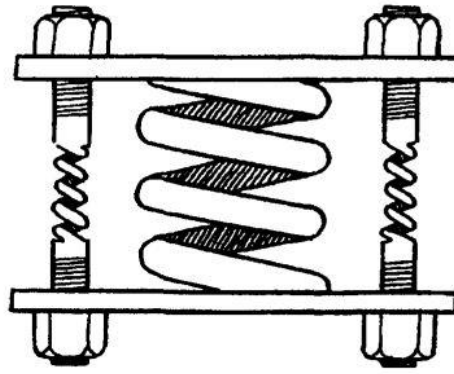
High-temperature material response plays a decisive role. Elastic modulus and strength decrease with temperature, fatigue behavior changes, and time-dependent mechanisms become more significant [12]. Although high-temperature alloys are selected to maintain capacity, their properties continue to vary considerably with temperature. Therefore, structural analyses must utilize temperature-dependent material data to produce physically meaningful predictions.

In this environment, bolted connections act as critical load paths between engine modules [7]. Any deterioration in connection performance (preload loss, shear, overdeformation, or fatigue damage) can reduce the structural margin. Therefore, a realistic assessment of the coupling behavior under combined thermomechanical loading is essential for the safe and efficient operation of the engine.

### **1.3 Structural Role of Bolted Joints in Gas Turbine Engines**

Modern gas turbine engines are designed modularly to facilitate necessary lifecycle maintenance (including inspection and overhaul). In this context, bolted connections represent much more than just auxiliary hardware. They function as critical structural fasteners, determining load transfer across module interfaces and ensuring geometric stability and seal integrity. Whether located in rotating components such as disc flanges or stationary housing elements, these interfaces face far stricter design constraints than those found in general industrial machinery [13].

Structurally, the performance of a bolted connection depends more on the precise application and maintenance of preload than on the tensile strength of the bolt itself. During assembly, the elongation of the bolt creates a compressive clamping force at the flange interface. This, in turn, creates a frictional force that resists separation and displacement. Consequently, the reliability of the connection depends on how well this clamping force is maintained under service conditions [14]. Field engineering practices demonstrate that these preloading objectives are not based solely on theoretical static calculations [15]. Instead, this requires validation, particularly against environmental factors such as the interaction of thermal cycling, vibration, and corrosion. The classical bolt–member spring model used to explain preload and load sharing is schematically summarized in Figure 1.3.



**Figure 1.3 :** Bolted joint spring model [16].

In the context of gas turbines, these connections must simultaneously meet three competing structural demands. First, they must ensure pressure retention; they must be robust enough to prevent leakage even when internal pressures distort the shell. Here, loss of contact pressure not only increases the risk of leakage but can also dangerously shift the load path directly onto the bolts. Second, they must maintain the geometric accuracy (alignment and rigidity) of the engine. Even microscopic slippage or rotations at the flange interface can alter end clearances and reduce aerodynamic performance. Finally, they must withstand severe dynamic loading spectra, including tensile, shear, and bending. Given the cyclical nature of these loads, the design must explicitly account for events such as microslippage and partial unloading to mitigate fatigue damage.

Therefore, contemporary design methodologies examine the bolted connection in conjunction with other fasteners rather than in isolation. This system-level approach is indispensable in high-temperature environments. At gas turbine operating temperatures, differential thermal expansion and modulus distortion can significantly alter stiffness ratios and load distributions. This can directly threaten the separation allowance and service life of the coupling.

#### **1.4 Challenges of High-Temperature Bolted Joints**

In high-temperature environments, it is a mistake to treat bolted connections simply as standard mechanical components because the challenges they face are fundamentally different and far more aggressive. We are not dealing with isolated problems here. Instead, there is a complex, almost chaotic interplay between material degradation, differential thermal expansion, and the extremely sensitive nature of preloading [17].

Consequently, relying on conventional design assumptions for aerospace gas turbine engines is not only optimistic but often dangerous.

Instantaneous degradation of mechanical properties is also a factor to consider. Operating temperatures reaching up to 700°C cause radical changes in the material behavior of the parts. These changes are clearly visible in material testing. Under these conditions, even superalloys like Inconel 718 experience a sharp decrease in elastic modulus (rigidity) and yield strength [18,19]. This situation is not insignificant for bolts because the loss of rigidity directly weakens the compressive force generated during assembly. Since most safety margins are calculated using room temperature data, they do not account for this 'softening' and leave the connection vulnerable to separation.

The stability of the preload is constantly battling against differential thermal expansion. In a gas turbine, the bolt and flange rarely heat up at exactly the same rate; they have different thermal masses and see different flow paths. Ultimately, thermal mismatch occurs [20]. If the flange expands faster, it forces the bolt to stretch beyond its yield point. If the bolt expands faster, the compressive force can effectively disappear. This struggle is most intense during engine start-stop transitions where thermal gradients peak.

Fatigue is also a significant threat. These connections not only remain immobile; they are subjected to cyclic loads (vibrations, pressure increases) as well as high average thermal stresses [21,22]. While standard models predict fatigue life based on room temperature, the reality of high heat introduces 'friction' and 'microslip'. Surface oxidation reduces hardness, allowing even microscopic movements at the interface to rapidly lead to cracks [23]. These cracks propagate much faster than standard models predict [24].

Additionally, the idea that the load is shared equally among all bolts is a theoretical construct and is never actually observed. Flange flexibility and manufacturing variations create an uneven load distribution during assembly [25]. Some bolts act as 'weak links' when they carry significantly more load than nearby bolts. Analyzing this behavior is quite difficult with simple hand calculations. Understanding this behavior is only possible through complex analysis.

Finally, we must consider the practical nightmare of assembly. High heat destroys thread lubricants, altering friction coefficients and making preloading a guesswork. It also leads to thread jamming during disassembly, making maintenance difficult. To address this, this thesis proposes a methodology that goes beyond linear assumptions, taking into account the nonlinear, temperature-dependent reality of these critical components.

### **1.5 Limitations of Existing Design Approaches for High-Temperature Bolted Joints**

Applying standard mechanical design rules to high-temperature aerospace components is not only difficult; it is fundamentally a flawed methodology. While current standards provide a solid foundation for general-purpose machinery, they hit a wall when confronted with the thermodynamic realities of a gas turbine [26]. The conditions here are aggressive. Therefore, relying solely on conventional methodologies is rarely sufficient for jet engines.

The fundamental failure of these conventional models is that they are based on 'static' room-temperature assumptions. In a standard handbook calculation, material properties—specifically Elastic Modulus ( $E$ ) and Yield Strength ( $\sigma_y$ )—are treated as constants [27]. These are fixed numbers. But physics requires otherwise. As metal temperatures rise toward 700°C, these properties do not remain constant. They degrade. And they do so in a nonlinear, unpredictable way.

Ignoring this temperature-dependent stiffness reduction is dangerous [25]. It creates a false sense of security. Safety margins calculated on paper simply disappear during actual operation. A design that appears robust at 20°C may approach plastic collapse at working height. Therefore, a new and more dynamic approach is needed.

A second, equally critical flaw is the assumption of 'uniform temperature'. Many analytical models treat the entire connection as a single thermal block. In reality, a gas turbine connection is a landscape of severe thermal gradients. The bolt, flange, and housing typically operate at different temperatures due to exposure to varying thermal masses and coolant flows. This differential thermal expansion is not a minor side effect; It is a fundamental driver of stress [20]. It actively modifies preload and contact pressure in ways that simplified, uniform temperature models cannot predict.

Furthermore, the concept of 'Preload' is often misunderstood in conventional design. Standard protocols treat preload as a constant, static number obtained during assembly [28]. Preload is a dynamic variable in high-temperature environments. The preload on the bolt is affected by phenomena such as material relaxation, thermal expansion, and surface displacement. Because of these, it constantly fluctuates. Since conventional methods do not account for this 'preload loss', they cannot accurately evaluate the behavior of the connection.

Fatigue analysis also suffers from similar blind spots. Most guidelines rely on S-N curves obtained from tests performed at room temperature, assuming linear elasticity [29]. This ignores aggressive mechanisms that are active at high temperatures and accelerate crack formation, such as surface oxidation and microstructural changes. Using a room-temperature fatigue model for a hot turbine casing is scientifically invalid and leads to nonconservative life estimates.

Finally, Finite Element Analysis (FEA) offers a way to model these complexities, but it comes with its own risks. There is a general tendency to view numerical results as absolute truth, but this is not an accurate perspective. However, an FEA model is only as good as its boundary conditions and material inputs. Complex FEA simulations, if interpreted without verification through analytical reasoning or physical testing, can mask fundamental design flaws. Furthermore, they can create a 'black box' illusion of accuracy. Therefore, industry needs a hybrid methodology: one that grounds advanced numerical simulation in fundamental mechanical principles while explicitly accounting for the chaotic thermal reality of the engine.

## **1.6 Motivation, Objectives, and Scope of the Present Thesis**

The fundamental impetus for this thesis is the relentless increase in thermal intensity in propulsion systems used in modern aerospace vehicles. As gas turbine engineers raise operating temperatures to increase efficiency, the load falls heavily on structural interfaces, particularly bolted flange connections. These connections are the structural backbone of the engine; their failure is not an option. However, despite their prevalence, there is a critical blind spot in our practical understanding of how they behave under extreme thermomechanical stress. Conventional design manuals, effective for standard machines, are often inadequate when attempting to predict the

total chaos of temperature-dependent stiffness loss, differential expansion, and fatigue loading in the aerospace environment [17].

Therefore, this study aims to eliminate these uncertainties by rigorously evaluating the structural integrity of a representative high-temperature flange assembly. The analysis focuses on a specific, industry-relevant configuration: two mating flanges secured with twenty M10×1.25 aerospace-standard 12-point bolts forged from Inconel 718 [18,19]. This setup is not random; It reflects compact, high-performance architectures found in real service and provides a reliable basis for research.

The analytical core of this work focuses on the instantaneous thermo-elastic response. By deliberately excluding long-term creep behavior, we isolate and amplify the effects of temperature on three specific variables: elastic-plastic deformation, preload retention, and fatigue life [21]. By filtering out creep, we can more precisely examine how thermal gradients and stiffness degradation directly attack the safety margins of the connection during operational cycles.

The studies in this thesis adopt a dual methodology for validation. First, analytical calculations based on modified fundamental stiffness principles are used to account for temperature-sensitive material data [25]. This provides a 'physics-based' understanding. Second, we use high-accuracy Finite Element Analysis (FEA) to capture complex contact pressure distributions and local thermal stress concentrations that are overlooked by hand calculations. The main aim here is not just to generate color graphs, but to provide a comparison between analytical predictions and numerical results. Identifying where these two methods diverge reveals the limitations of traditional design principles and highlights where safety factors can be misleading.

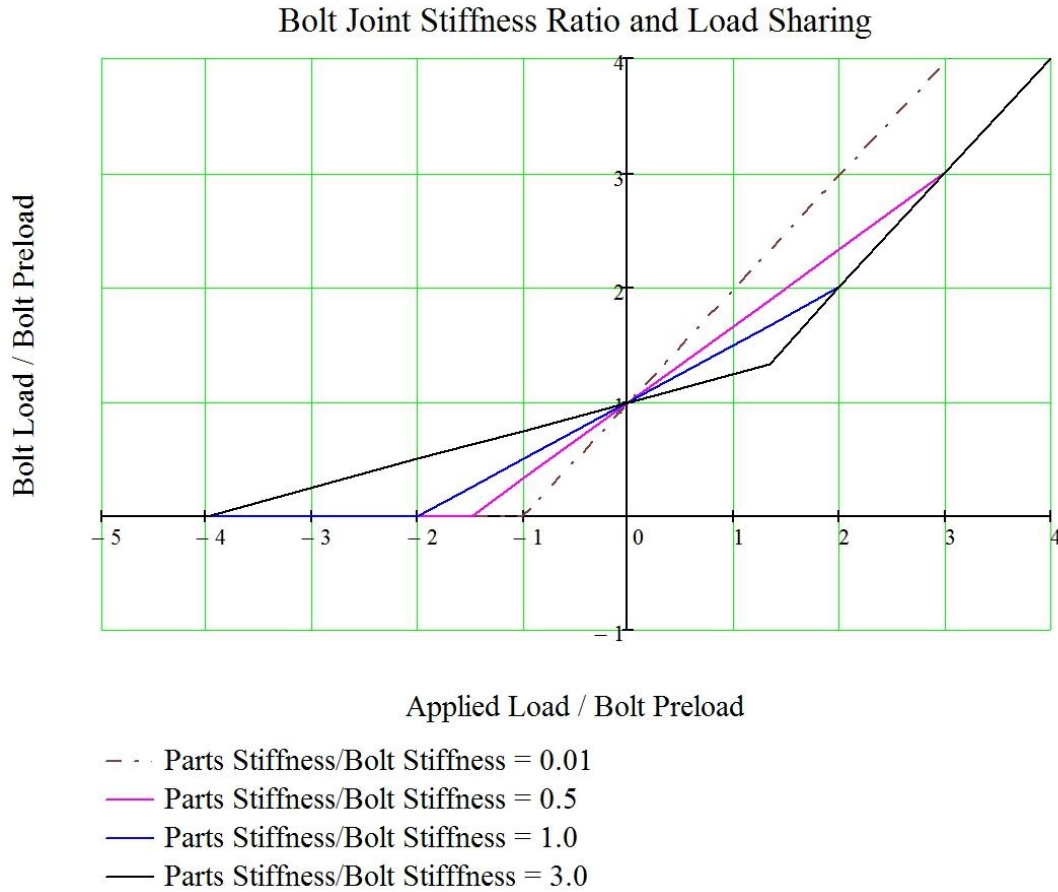
In summary, this research is limited to the structural and fatigue mechanics of metallic aerospace connections under high-temperature conditions. By bypassing manufacturing variability and non-metallic alternatives, it provides clear and actionable information for gas turbine design engineers. The ultimate goal is to bridge the gap between simplified textbook theory and the harsh reality of high-temperature operation, providing a validated reference for safer, more optimized engine structures.

## 2. LITERATURE REVIEW

### 2.1 Bolted Joint Fundamentals and Mechanical Behavior

Bolted joints, which are fundamental to engineering infrastructure, are not merely fasteners. They are the primary and most commonly used elements of modular design. Their ability to provide a secure yet demountable connection makes them indispensable in aerospace gas turbines where maintenance protocols require frequent module inspection and replacement [4,30]. Unlike permanent welds, these connections offer the structural adaptability necessary for safety-critical overhauls. The structural heart of these connections is 'preloading'. By elastically stretching the bolt during assembly, a compressive clamping force is created. Mechanically, this relationship is usually approximated by the short-form equation  $T = K \cdot F \cdot d$ , where  $T$  is the applied torque,  $F$  is the desired preload,  $d$  is the nominal diameter, and  $K$  is the 'nut factor', typically ranging from 0.15 to 0.20 depending on lubrication conditions. From a structural mechanics perspective, the connection behaves like two competing spring systems. The critical design parameter is the stiffness ratio, often defined as the Load Factor ( $\Phi$ ). In rigid aerospace flange designs, this factor is designed to remain low, typically between  $\Phi = 0.2$  and  $0.3$ . This allows the flange to absorb 70-80% of the load change when external tensile loads are applied, while the bolt only feels a small fraction of it. This protective effect is vital for high-cycle fatigue life and prevents the fastener from being exposed to dangerous stress amplitudes [27,31,32]. Effect of the joint stiffness ratio over the preload distribution can be seen in Figure 2.1.

However, achieving this precise preloading is physically challenging and torque tightening is inherently inefficient due to parasitic friction. According to Bickford's general distribution, approximately 50% of the input torque is consumed by friction under the bolt head and 40% is lost to thread friction. This results in only about 10% of the applied energy remaining to actually tension the bolt and generate the compressive force [30].



**Figure 2.1 :** Effect of the joint stiffness ratio over the preload distribution [33].

Consequently, torque-controlled tightening is extremely sensitive to surface conditions and often results in a preload distribution of  $\pm 25\%$  or more [28]. In contrast, more advanced methods such as hydraulic tensioning can reduce this distribution to  $\pm 5\text{-}10\%$ , but they present packing challenges in compact motors. Operational reality further complicates these static values. In multi-bolt flange connections, analytical models generally assume a uniform load distribution ( $1/N$ ). However, manufacturing tolerances and flange fluctuation create 'hard zones' where local load concentrations can occur, with certain bolts carrying up to 1.5 times the nominal load [34]. Furthermore, preloading is not static; it decreases. Immediately after tightening, surface roughness flattens (embedding), typically resulting in a 2% to 5% loss in initial stress within the first few thermal cycles [25]. Therefore, the design challenge is not only to achieve the target preload but also to protect it against thermal relaxation.

Consequently, the goal is to prevent connection separation. When the external load exceeds the compression limit, load transfer shifts from the friction clutch directly to the bolt stress, causing increased stress amplitudes. Therefore, aviation standards

require sufficient preloading to maintain a compression margin (typically a safety factor of 1.2 to 1.5 against separation) under all flight maneuvers.

## 2.2 Elevated Temperature Effects on Bolted Joints

The design of bolted connections for the hot sections of an aerospace gas turbine requires a fundamental departure from the static assumptions governing room-temperature mechanics. In these regions, where operating temperatures frequently rise toward 700°C, a bolted connection ceases to be a passive structural element. Instead, it behaves like a dynamic thermodynamic system. Consequently, relying on standard ambient temperature design criteria [4,35] is not only insufficient but also creates a blind spot that can mask critical failure modes. The most pressing structural loss is the degradation of material stiffness. As temperature increases, the Elastic Modulus ( $E$ ) of high-strength superalloys (such as Inconel 718) does not remain constant; it degrades nonlinearly, typically decreasing by approximately 15% to 20% at 700°C compared to room-temperature values. This is not a negligible decrease. This fundamentally alters the stiffness ratio ( $\Phi$ ) of the connection, which determines how external loads are shared between the bolt and flange. As the flange becomes more flexible (less rigid) due to heat, its ability to protect the bolt from external fatigue loads decreases. Therefore, even if the external mechanical force remains constant, the stress amplitude to which the bolt is subjected increases solely due to this thermal softening [27,36].

At the same time, the safety margin against yielding erodes. The Yield Strength ( $\sigma_y$ ) of the fastener material decreases with stiffness. A preloading level that seems conservative at 20°C can dangerously bring the bolt close to its yield point at operating temperatures. If the combined thermal and mechanical stresses exceed this reduced yield limit, the bolt undergoes local plastic deformation. The result is serious: after cooling, the bolt does not return to its original length, resulting in an irreversible loss of compressive force [28]. Therefore, the applicable design limits must be derived strictly from the 'hot' material properties. However, the most aggressive stress factor in these assemblies is usually Differential Thermal Expansion. Since bolts and flanges often have different thermal expansion coefficients ( $\alpha$ ) or different thermal masses, they rarely expand at the same rate. This mismatch creates an internal thermal force ( $\Delta F_{th}$ ) which can be approximately expressed by the basic relationship:

$$\Delta F_{th} = E_b \cdot A_b \cdot (\alpha_f \cdot \Delta T_f - \alpha_b \cdot \Delta T_b) \quad (2.1)$$

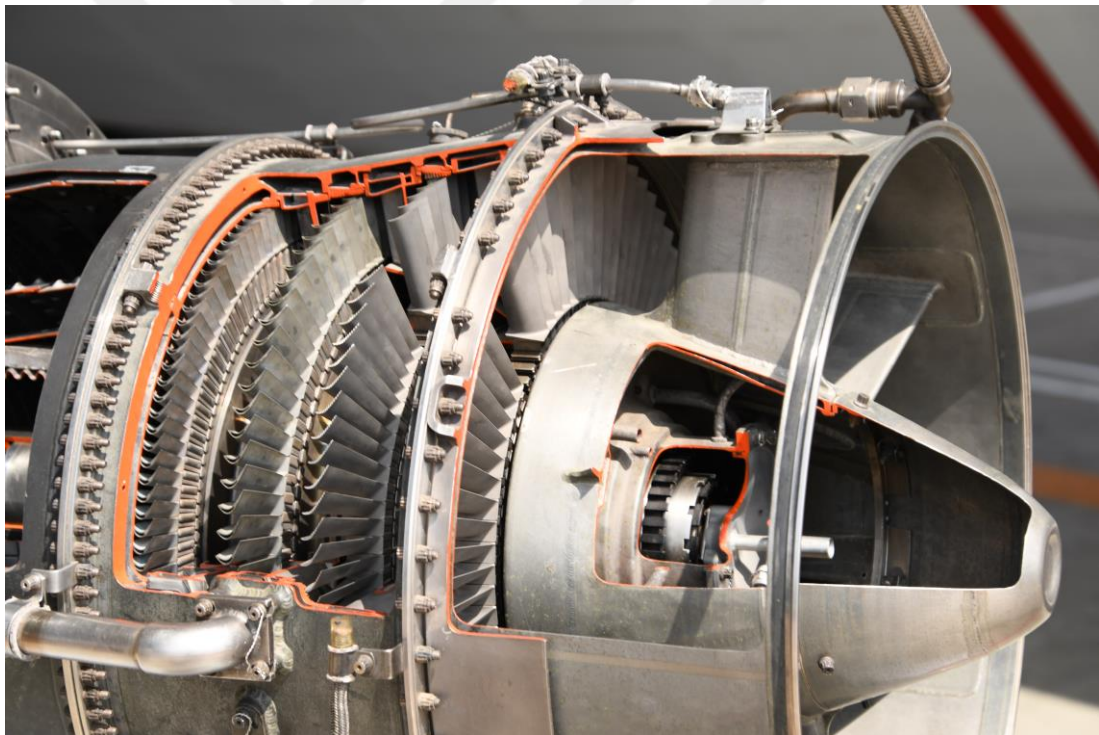
Here,  $E_b$  and  $A_b$  represent the modulus and area of the bolt, while  $\Delta T$  represents the temperature change for the flange ( $f$ ) and bolt ( $b$ ). The physics here is a tug-of-war: if the flange expands more aggressively than the bolt, it forces the fastener into tension and creates a large increase in stress. Conversely, if the bolt expands faster, the connection loses compression and leads to leakage or separation [35]. In a real engine environment, these thermal differences are not spatially homogeneous and create a complex, time-dependent stress map. These effects are most severe during 'transient' operations, especially when the engine is started and stopped. Unlike steady-state navigation, transient conditions create severe thermal gradients where thin flanges heat up rapidly, while larger bolts lag behind thermally. This transient mismatch leads to Low Cycle Fatigue (LCF) damage accumulation, typically generating the highest stress peaks seen throughout the entire flight cycle [4]. A static analysis that ignores these transient peaks essentially misses the most damaging part of the motor's life.

Finally, high temperatures alter the tribological conditions at the contact interface. Surface oxide layers alter the coefficient of friction, making torque-stress relationships unpredictable. Furthermore, even without long-term creep, instantaneous elastic-plastic redistribution occurs. Localized yielding at tooth roots or bearing surfaces can instantly alter the load path [36-38]. Therefore, a robust design methodology cannot address these factors in isolation; it must integrate temperature-dependent material data, transient thermal analysis, and validated safety margins to navigate this nonlinear environment [25,28].

### 2.3 Bolted Joints in Aerospace Gas Turbine Engines

In the challenging architecture of aerospace gas turbines, bolted connections serve a far more critical function than simpler connections. They are the primary joint elements of the engine's modular design strategy. Wide range of bolted joint usage can be seen in Figure 2.2. A modern propulsion system is not a monolithic block. It consists of a series of different functional modules that must be securely joined to transmit thrust, but remain fully demountable for lifecycle maintenance. These modules are: compressor, combustion chamber, turbine [4,35]. This contradiction creates a serious engineering conflict. The connection must possess enormous

structural rigidity to withstand flight loads, but it must also allow for rapid demounting during overhaul without damaging the coupling surfaces. Unlike stationary industrial machines where weight is negligible, aerospace connections must provide this reliability under strict weight penalties. Every gram counts. The operational environment is hostile for these connections. A typical hot-section connection is subjected to three simultaneous loads. First, it withstands static axial stress resulting from high internal pressures (containment load). Second, it resists dynamic shear loads resulting from rotor imbalance. Thirdly, it withstands high-frequency vibration resulting from combustion instability [13]. In addition to this mechanical chaos, there is also thermal stress. Because fuselage flanges and fittings typically heat at different rates, the connection constantly struggles against its own geometry. This differential expansion creates a complex stress state that standard room temperature design rules cannot predict.



**Figure 2.2 :** Bolted joint usage in gas turbine engines - JMSDF US-1A T64-IHI-10E turboprop engine [39].

Seal integrity is also an absolute necessity. In high-pressure turbine stages, the bolted connection acts as the last barrier against hot gas leakage. Here, the loss of compressive force is catastrophic. It not only causes a loose part; it allows superheated gas to escape, burning adjacent components and compromising the structural rigidity of the fuselage [35]. Consequently, aviation philosophy treats connection separation as a 'failure

condition'. Engineers must calculate sufficient preloading to maintain positive contact pressure under all flight maneuvers, including peak transient loads [28].

To manage these forces, industry relies on multi-bolted circumferential flanges. This circular arrangement also provides redundancy. If a single bolt loses its preload, the load is redistributed to adjacent bolts. This is preventing bolts from sudden failure. However, this safety net hides a complexity: uneven load distribution. While idealized models assume each bolt carries an equal share ( $1/N$ ), real-world flange flexibility creates 'hard spots'. Empirical data show that certain bolts in these hard regions can carry up to 30% more load than average, becoming hidden fatigue points [25,34].

Hardware selection is quite important. Aerospace engines rarely use coarse-threaded bolts. Instead, they generally use fine-pitched threads (e.g., M10×1.25). The physics behind this choice is measurable. The Tensile Stress Field ( $A_s$ ) of a bolt is defined by the standard formula:

$$A_s = 0.7854 \cdot \left(d - \frac{0.9382}{p}\right)^2 \quad (2.2)$$

Here,  $d$  is the diameter and  $p$  is the pitch. By reducing the pitch ( $p$ ), we effectively increase the stress field ( $A_s$ ). Typically, a fine-threaded aviation bolt offers 10% to 15% more load-carrying capacity than a coarse-threaded equivalent of the same diameter. In addition, the shallower helix angle of fine threads provides superior resistance to vibration-induced loosening [27,30].

Bolt head design is also determined by strict constraints. The space to accommodate this surface is generally narrow. Therefore, 12-point (double hexagonal) heads are used as standard. Unlike the standard hexagonal head, the 12-point design allows for higher torque application in confined spaces where large wrenches cannot fit. More importantly, it directly contributes to the motor's thrust-to-weight ratio targets by reducing the head mass.

Material selection is also subject to strict rules. Standard carbon steels oxidize rapidly at high temperatures such as 700°C. Therefore, industry uses precipitation-hardened nickel-based superalloys such as Inconel 718. These materials have optimum properties for use in gas turbines. They maintain a significant yield strength even at very high temperatures and have enough ductility to withstand thermal growth without

cracking [18,40]. However, their reduced ductility at high temperatures requires precise design tolerances to avoid brittle failure modes.

Finally, vibration management is critical. Gas turbines produce broadband vibration spectrums that can loosen unsecure couplings. Since polymer-based locking inserts will melt instantly in the hot section, aviation couplings rely on positive mechanical interlocking. Safety wire, washer washers, or deformed threaded lock nuts are mandatory [28]. From a certification standpoint, these couplings are 'Safety Critical'. Fatigue life and separation margins must be verified not only by calculation but also by rigorous physical testing to ensure survival over thousands of flight cycles [37].

## **2.4 A Comparative Review of Joining Technologies: Bolting, Welding, and Riveting**

Engineering design is essentially the effort to find the optimum to a need. The choice of a joining technology, such as bolting, welding, or riveting, is rarely a simple binary choice between 'good' and 'bad'. Instead, it is a complex multivariate optimization problem. We need to balance structural efficiency with manufacturing costs. We need to compare material compatibility with long-term service life [41]. In the context of aerospace gas turbines, where safety margins are narrow and maintenance cycles are rigorous, this decision matrix becomes more complex. Every choice has a specific penalty.

Welding represents the pinnacle of structural continuity. It creates a flawless interface by joining materials at the molecular level. This feature offers a significant weight advantage by eliminating the need for heavy flanges and fasteners. Mechanically, a welded connection is also superior in terms of rigidity because it is a connection devoid of geometric discontinuities that cause stress concentrations [42]. Modern processes such as Electron Beam (EB) or TIG welding can successfully weld advanced superalloys such as Inconel 718 and Titanium. However, welding has a critical operational drawback: permanence. It creates an irreversible bond. In a modular gas turbine engine, this is a logistical disadvantage. Internal components, such as turbine blades, require regular borescope inspection and replacement. If the housing is sealed with welding, access is impossible without destructive cuts.

Furthermore, welding metallurgy involves hidden risks. Intense thermal cycling creates a 'Heat Affected Zone' (HAZ) adjacent to the weld seam. In this HAZ zone, the grain structure of the base metal often becomes coarse. This reduces its mechanical properties and fatigue resistance. Also, verifying the integrity of a weld is an expensive process. While bolted connections can be visually inspected, this process is not applicable to welded connections. A weld requires Non-Destructive Testing (NDT) methods such as Ultrasonic Testing (UT) or Radiographic inspection. In order to detect internal porosity or lack of fusion. This adds complexity and cost to the manufacturing process.

Riveting is an entirely different matter. Historically, it formed the backbone of early aviation and civil structures such as the Eiffel Tower. It relies on the large plastic deformation of a ductile pin to create a mechanical lock [43]. It is resistant to shear force and immune to loosening. However, riveting is largely obsolete for modern high-temperature gas turbines. The laws of physics do not allow it. Riveting requires the fastener to be ductile enough to deform during installation. However, aviation bolts for hot sections are manufactured from precipitation-hardened superalloys, specifically designed for high yield strength, not ductility. They are too hard to rivet. Furthermore, rivets cannot produce the large, controllable compressive forces required to seal high-pressure flanges against gas leakage [44].

This leaves bolting as the undisputed standard for critical engine interfaces. What sets bolting apart is not only its strength but also its 'reversibility'. It is the only mature technology that allows a high-pressure system to be repeatedly assembled and disassembled without disrupting its basic structure [45]. For an engine that can be overhauled five or six times during its life cycle, this ease of service is undeniable. Bolts can be designed from ultra-high-strength materials to create enormous compressive loads through preloading, effectively transforming a multi-part assembly into a rigid unit [44].

However, this capability comes at a structural cost. To use a bolt, we must drill a hole. In fracture mechanics, a circular hole in a plate acts as a stress concentrator. It disrupts the stress flow and typically increases the local stress at the hole edge by three times ( $K_t \approx 3.0$ ) compared to the nominal stress. This makes the flange inherently susceptible to fatigue crack formation. In addition, unlike permanent welds, bolted connections are dynamic; they can spontaneously loosen under vibration or relax under

thermal cycling. Therefore, the engineering challenge in bolting is not just creating the connection; The goal is to manage these stress concentrations and maintain preload integrity over thousands of working hours [45]. Comparison detail of these different type of joints can be seen in Table 2.1.

**Table 2.1** : Comparison of the different type of joints [42].

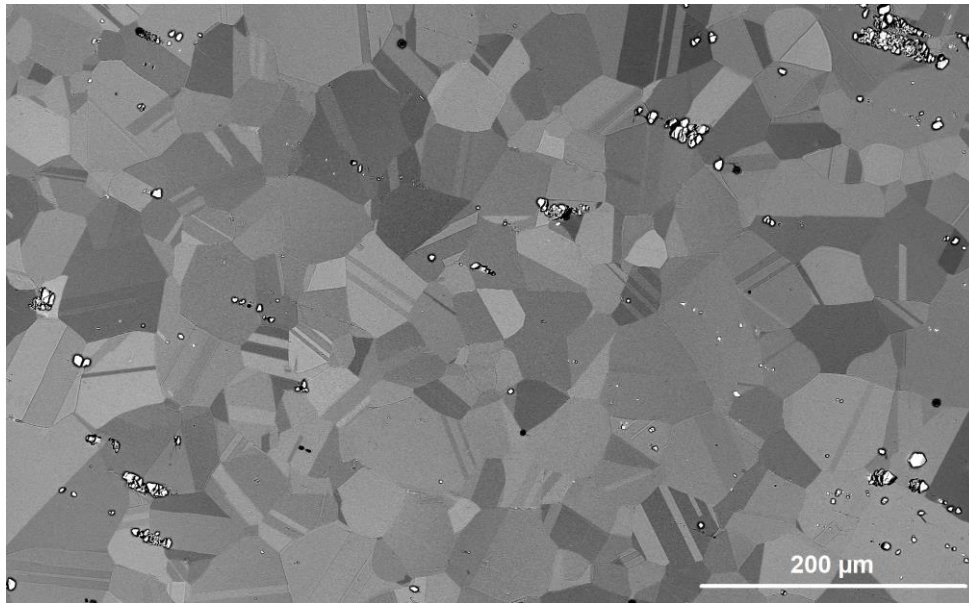
| Feature                | Bolted Joint                                   | Welded Joint                                     | Riveted Joint                            |
|------------------------|------------------------------------------------|--------------------------------------------------|------------------------------------------|
| Permanence             | Removable / Reusable                           | Permanent                                        | Semi-Permanent                           |
| Relative Strength      | High to Very High                              | Very High (often equals base material)           | Moderate to High                         |
| Rigidity               | Moderate to High                               | Very High                                        | Moderate                                 |
| Maintenance/Repair     | Excellent (designed for disassembly)           | Poor (destructive removal)                       | Fair (rivets can be drilled out)         |
| Heat Application (HAZ) | No                                             | Yes (Heat-Affected Zone created)                 | No (typically cold-formed)               |
| Inspection Difficulty  | Easy (visual, torque audit)                    | Difficult (requires NDT for internal flaws)      | Easy (visual inspection of formed head)  |
| Vibration Resistance   | Fair (requires locking features)               | Excellent                                        | Excellent                                |
| Typical Applications   | Machine assembly, structures, engines, flanges | Building structures, pipelines, pressure vessels | Historical structures, aircraft fuselage |

Data synthesized from refecence [42].

## 2.5 High-Temperature Fastener Materials with Emphasis on Inconel 718

In the hot section of a gas turbine engine, the quality of a fastener is only as good as its metallurgy. Choosing the right material is not just a detail; it is the first line of defense against structural failure. The bolts here operate in a harsh environment. They face high preloading levels, cyclic thermal shock, and corrosive oxidation gases. Under these conditions, conventional alloy steels are useless. They rapidly lose their strength and oxidize instantly [18,46]. As a result, the aerospace industry has turned to nickel-based superalloys as the only viable solution. Among these, Inconel 718 stands out as the industry standard. It is the basic material for gas turbine fasteners. Because it offers a rare balance between strength, ductility, and manufacturability. Its mechanical performance is not accidental. It is designed with a specific 'precipitation hardening' mechanism. When heat-treated, the alloy forms compatible Gamma-double-prime ( $\gamma'' - Ni_3Nb$ ) and Gamma-prime ( $\gamma' - Ni_3(Al, Ti)$ ) phases within the nickel matrix. This microstructural reinforcement locks the grain boundaries and allows Inconel 718 to maintain its high yield strength up to approximately 650°C–700°C. This

temperature range is ideal for many combustion chamber and turbine housing connections [40]. Microstructure of the IN718 shown in Figure 2.3.



**Figure 2.3 :** Microstructure of the IN718 [47].

Maintaining the yield strength ( $\sigma_y$ ) is a critical parameter. In a bolted connection, the preload is entirely dependent on the elastic stress of the connecting element. If the material yields (softens) due to heat, this elastic stress is converted into permanent plastic deformation. The compressive load disappears. Inconel 718 has an advantage here because its yield strength decreases much more slowly with temperature compared to other alloys. This means we can maintain high preloading levels during operation without exceeding the plastic limit [36,40]. This capability is essential to prevent joint separation.

Fatigue resistance is another determining factor. Studies consistently show that Inconel 718 outperforms many other high-temperature materials in both Low Cycle Fatigue (LCF) and High Cycle Fatigue (HCF) regimes [24,36]. A significant advantage is its relatively low notch sensitivity. Threaded fasteners are, by definition, full of notches (stress concentrations). Some superalloys are brittle and crack easily at the thread roots. Inconel 718 is more forgiving. Its high strength is combined with sufficient ductility to redistribute local stresses. This makes it the default choice for critical fatigue-prone areas such as turbine bodies and exhaust frames [18].

Then there is the manufacturing issue. It is difficult to manufacture bolts from a superalloy. Many high-strength alloys (such as Waspaloy or Udimet) are extremely

difficult to machine and forge. They crack during production. Inconel 718 is different. In solution annealing, it has properties suitable for forging. It can be machined into complex thread forms before hardening by final aging heat treatment. This machining flexibility is a major logistical advantage. It reduces scrap rates and provides consistent mechanical properties in thousands of bolts [18].

However, Inconel 718 is not perfect material. It has a metallurgical rate limit. When used for extended periods at temperatures near its maximum threshold (approximately 700°C), the microstructure can become unstable. Reinforcing  $\gamma$  precipitates can transform into the stable Delta phase ( $\delta$ ). This transformation is detrimental. It kills the ductility and strength of the material [18]. This is a creep-dominant phenomenon. Designers must be cautious. If the application involves sustained holding at peak temperatures, significant design margins must be applied. For cyclical gas turbine applications, this is manageable, but remains a valid concern. Tribology, the science of friction and wear, is also critical. Nickel alloys tend to 'wear' (cold weld) when compressed together under high pressure. If an Inconel bolt is compressed into an Inconel flange without lubrication, it will likely seize. To prevent this, aerospace fasteners use special high-temperature coatings. Silver plating is the industry standard. Silver acts as a solid lubricant, reducing the coefficient of friction and preventing metal-to-metal contact. Aviation regulations mandate these coatings to ensure consistent torque-stress relationships and prevent tooth damage [28].

Finally, we must also consider environmental attack. Inconel 718 forms a corrosion-resistant, stable, passive oxide layer. This is vital. Surface oxidation can serve as a crack initiation site for fatigue. Compared to lower-grade alloys, 718's ability to maintain surface integrity in the hot gas path is a significant safety advantage [18].

In summary, the selection of Inconel 718 is a calculated engineering decision. While exotic alloys may offer slightly higher temperature limits, they often come with exorbitant costs or manufacturing problems. Inconel 718 captures the 'ideal spot.' It offers proven reliability, a large operational history database, and a predictable failure mode [40,46]. This material remains the most suitable candidate for high-temperature bolted connections examined in this thesis. Details material properties of IN718 can be seen in Table 2.2.

**Table 2.2 : Material properties of IN718.**

| Property                       | Symbol         | Unit                 | Room Temperature (~21°C) | Operating Temperature (~700°C) | Change (%) |
|--------------------------------|----------------|----------------------|--------------------------|--------------------------------|------------|
| Density                        | $\rho$         | kg/m <sup>3</sup>    | 8190                     | 7960                           | -2.8%      |
| Elastic Modulus                | $E$            | GPa                  | 200                      | 162                            | -20.6%     |
| 0.2% Yield Strength            | $\sigma_y$     | MPa                  | 1034                     | 938                            | -9.3%      |
| Ultimate Tensile Strength      | $\sigma_{UTS}$ | MPa                  | 1275                     | 1015                           | -20.4%     |
| Thermal Expansion Coeff.       | $\alpha$       | 10 <sup>-6</sup> /°C | 13.0                     | 15.5                           | +19.2%     |
| Isotropic Thermal Conductivity | $k$            | W/(m·°C)             | 11.4                     | 25.0                           | +119%      |
| Specific Heat                  | $C_p$          | J/(kg·°C)            | 435                      | 610                            | +40.2%     |
| Elongation at Break            | $\epsilon_f$   | %                    | 12                       | 20                             | +66.7%     |
| Poisson's Ratio                | $\nu$          | -                    | 0.29                     | 0.29                           | 0%         |

## 2.6 Analytical and Numerical Design Methodologies for High-Temperature Bolted Joints

In aviation gas turbines, reliable structural design is not a single calculation; it is a layered cost-benefit analysis. Designing bolted connections for high temperatures requires detailed calculations. It necessitates a hybrid methodology that bridges the gap between classical analytical mechanics and advanced computational simulation. Traditional manual calculations are indispensable for preliminary sizing and validation, but they have limitations and are quite inadequate for detailed design. These manual calculations account for relatively few variables. The operating environment of gas turbine engines, however, is highly dynamic and involves many variable conditions. Therefore, advanced numerical simulations are used to simulate thermal gradients and nonlinear material behavior. This dual approach is not only 'recommended' but essential for verifying static strength under the harsh conditions of flight. The foundation of any bolted connection is preloading. This axial stress is the heart of the assembly. It is not a random number. The initial compressive force must be sufficient to keep the flanges compressed under external service loads and prevent the catastrophic triad of separation, leakage, and fatigue relaxation. In aerospace practice, we don't aim for 'tight enough'. We aim for precision. Typically, the target preload ( $F_p$ ) is set between 65% and 90% of the tensile strength ( $S_p$ ) of the bolt

material. This aggressive target ensures the connection remains closed during assembly without causing accidental loosening of the bolt [27].

Determining the input torque required to achieve this preload is governed by the short-form relation:

$$T = K \cdot d \cdot F_p \quad (2.3)$$

Here,  $T$  is the applied torque,  $d$  is the nominal diameter, and  $K$  is the 'nut factor', a variable representing friction. Although the formula seems simple, the situation is actually complex. In aerospace applications, engineers must explicitly account for preload loosening due to differential thermal expansion, a factor often overlooked in standard industrial calculations [48].

Once tightened, the connection becomes a load-sharing system. The distribution of external tensile loads is determined by the 'Joint Stiffness Constant' ( $\Phi$ ), defined as follows:

$$\Phi = \frac{k_b}{k_b + k_m} \quad (2.4)$$

Here,  $k_b$  is the bolt stiffness and  $k_m$  is the stiffness of the clamped elements. This ratio indicates how much of the external load the bolt actually 'feels'. A low  $\Phi$  value, typically designed to be between 0.10 and 0.25, is the target. This allows large flanges to absorb most of the external load and protect the bolt from fatigue [17]. Calculating  $k_m$  is not easy; it requires estimating the pressure cones within the flange material using the methods described in VDI 2230 [25,27].

The strength assessment continues by calculating safety margins. This safety margin calculation allows us to make a final judgment about the mechanical strength of the design. By interpreting this numerical value, it can be checked whether the strength is safe. The total axial stress ( $\sigma_t$ ) and shear stress ( $\tau$ ) acting on the bolt stress field ( $A_s$ ) are used. The total stress calculation is made with these stresses. And control is provided using the Von Mises yield criterion. Then, the Safety Margin (MS) is calculated to ensure compliance [49]:

$$MS = \frac{\sigma_{yield}}{\sigma_{vm}} - 1 \quad (2.5)$$

A positive MS is mandatory. Standards such as NASA-STD-5020B ensure reliability even under worst-case loading by setting strict minimums for these values [28]. Beyond the bolt itself, we must also check the flange. The bearing stress ( $\sigma_b$ ) under the bolt head should not exceed the material's compressive limit, otherwise 'embedding' will occur and lead to preload loss. Furthermore, shear resistance is critical for connections carrying shear loads. The friction force ( $F_f = \mu \cdot F_p \cdot n$ ) must overcome the lateral loads to prevent the connection from slipping, especially in high-vibration regions [50].

To perform these calculations, engineers rely on two basic standards: VDI 2230 and NASA-STD-5020B. VDI 2230 is the systematic European standard. It is rigorous in calculating stiffness and load factors. However, it has a blind spot: it assumes constant temperatures. Applying this to a turbine body at 700°C requires significant manual adjustment of the material data. In contrast, NASA-STD-5020B is specifically designed for spaceflight equipment. It is more conservative. It explicitly demands the analysis of 'separation margins' and mandates checking the bolts in terms of interaction effects (tensile + shear) to prevent failure in complex stress states [28].

However, analytical methods have limitations. They rely on simplifying assumptions—concentric loading, uniform temperature, and linear elasticity. A gas turbine does not conform to any of these. It is a chaotic environment of thermal gradients and local plasticity. This is where the 'Hybrid Approach' becomes necessary. Analytical methods define bolt size and safety margins. Then Finite Element Analysis (FEA) comes into play. FEA determines the stress concentrations in the first contacting thread. It reveals the thermal distortion that manual calculations miss [51]. By comparing the FEA results with analytical predictions, the designer can iterate through the geometry and ensure that the final design is not only mathematically correct but also physically sound.

## **2.7 Service Life, Durability, and Fatigue Mechanics of High-Temperature Bolted Joints in Aerospace Gas Turbines**

### **2.7.1 A functional concept of durability and lifespan**

In the rigorous field of aerospace engineering, defining durability is not as simple as counting cycles to fracture. In gas turbine applications in particular, the service life of a bolted connection extends far beyond basic fracture mechanics. This is a system-level characteristic. Unlike a monolithic turbine blade where failure is signaled by a crack, a bolted connection can fail while the bolt itself remains structurally sound. The definition of failure here is 'functional'. If the connection leaks hot gas, loses its rigidity, or allows flange separation, it has reached the end of its service life [17]. Therefore, durability testing should evaluate the assembly not merely as an isolated fastener, but as an integrated tribological system. We must analyze the bolt, nut, compressed flanges, and operating environment as a single, interacting unit.

### **2.7.2 The crucial mechanics of preload and load transfer**

The life of a bolted connection depends on one fundamental variable: Initial Preloading. The mechanics here are not intuitive but are critical. A significant preload acts as a shield. By compressing the connection components, we ensure that most of the external cyclic loads are absorbed by the flange (decompression) instead of the bolt (stress) [25]. This 'shield effect' is key to fatigue resistance.

According to conventional standards such as VDI 2230, high preloading significantly reduces the Stress Amplitude ( $\sigma_a$ ) to which the fastener is subjected. This is vital. Fatigue damage primarily results from alternating stress, not static stress. However, there is a trade-off. Increasing preloading inevitably increases the Mean Stress ( $\sigma_m$ ) in the bolt material. Conventional fatigue criteria such as Goodman or Soderberg lines show that high mean stress reduces fatigue capacity. However, in the specific context of a bolted connection, physics supports preloading. The benefit of suppressing the stress amplitude ( $\sigma_a$ ) far outweighs the disadvantage of higher mean stress [52]. Therefore, maintaining preloading near the yield limit is considered essential for fatigue resistance.

### **2.7.3 Mechanisms of preload degradation and joint separation**

Although preloading is our primary defense, it is not a permanent state. It is a decaying entity. It decays over numerous operational cycles. Immediately after compression, microscopic irregularities on mating surfaces flatten under pressure. This phenomenon, known as 'embedding', leads to elongation loss and subsequent stress reduction. Furthermore, at high temperatures, creep relaxation and elastic-plastic redistribution mechanisms actively attack the compressive force [53].

This degradation leads to the ultimate failure mode: Joint Separation. If the preload falls below the external load threshold, the flanges separate. This is catastrophic. The load transfer mechanism changes instantaneously. When the compressive force is exhausted, the bolt has to carry 100% of the external cyclic load. The result is a massive increase in stress amplitude – often an increase of orders of magnitude. This aggressively accelerates crack propagation. Consequently, the airworthiness regulations for aircraft are strict. They require that sufficient preload be maintained to prevent separation under any possible flight conditions [54,55].

### **2.7.4 Complex fatigue interaction: high cycle fatigue, low cycle fatigue, and thermal cycling**

Gas turbines do not present a clean, single-source fatigue environment. Instead, the links are confronted with a chaotic superposition of regimes. First, there is High Cycle Fatigue (HCF). This results from high-frequency aerodynamic vibration during steady-state operation and rapidly accumulates over millions of cycles. Second, there is Low Cycle Fatigue (LCF). This is due to large stress-strain changes that occur during major transient events—engine starting, launching, and stopping [56].

Then there's the hidden danger: Thermo-Mechanical Fatigue (TMF). Even without mechanical loads, temperature fluctuations cause damage. Since bolts and flanges often have different thermal masses or coefficients of expansion ( $\alpha$ ), they expand at different rates. These cyclical thermal strains alter the internal stress distribution. Over time, TMF causes fatigue damage that standard mechanical analysis cannot predict [57].

### **2.7.5 Material deterioration and localized stress concentration**

Heat attacks the material itself. In nickel-based superalloys like Inconel 718, hardness is temperature-dependent. As the engine heats up, the Elastic Modulus decreases. This decrease alters how the material responds to strain [18]. But the real danger is geometric. A threaded fastener is, by definition, a series of notches. Fatigue cracks almost always begin at the root of the first contacting thread. Because the load distribution is not homogeneous. Experimental data show that the first thread typically carries 30% to 34% of the total load, creating a significant stress concentration factor ( $K_t$ ). At high temperatures, the reduced yield strength allows localized plastic deformation to occur at this thread root, even if the nominal stress in the bolt body is safe. This localized plastic deformation initiates fracture. Environmental factors such as high-temperature oxidation accelerate the process by attacking these grain boundaries [58]. Therefore, nominal stress calculations are often misleading. Finite Element Analysis (FEA) is necessary to resolve these localized stress states for accurate life estimation [59].

### **2.7.6 Design philosophy and considerations for maintenance**

Given these uncertainties, the aerospace industry is pessimistic. We predominantly use the 'Safe Life' design philosophy. Threaded fasteners are difficult to inspect; a crack hidden within a nut is not visible to the naked eye. Therefore, we do not rely on 'Damage Tolerance' (finding cracks before fracture). Instead, components are decommissioned after a certain number of hours, long before the statistical probability of failure increases [55].

Finally, we must take the human factor into account. Maintenance affects longevity. Disassembling and reassembling a link changes the tribological conditions – especially the coefficient of friction. This leads to a distribution in the resulting preload. A robust design must be tolerant to this variability and ensure reliability not only on day one but throughout the entire service life of the engine [17].

## **2.8 FEA in High-Temperature Bolted Joint Design for Aerospace Applications**

While analytical standards like VDI 2230 and NASA-STD-5020B provide a solid foundation for preliminary sizing, they hit a wall when confronted with the thermodynamic chaos of a gas turbine. Manual calculations assume uniform

temperatures, linear material behavior, and idealized concentric loading. A real engine does not conform to any of these assumptions. Consequently, Finite Element Analysis (FEA) has evolved from a validation tool to a primary design tool. It allows engineers to solve complex thermo-mechanical interactions that closed-form equations cannot capture [25,28].

### **2.8.1 Model fidelity and simulation technique**

The first challenge in FEA is the trade-off between computational cost and model accuracy. Simplicity is attractive but dangerous for a bolted connection. Simplified 1D beam elements or 'spider' kinematic connections are computationally inexpensive, but they are blind to local effects. Research consistently shows that a full 3D solid model is indispensable in high-temperature fatigue prediction [60,61]. Because fatigue cracks start at microscopic stress concentrations, especially at the root of the first contacting thread. A beam element cannot see the thread root. Therefore, in industrial applications, the bolted threaded region is often modeled with a dense mesh. And 'pre-stress elements' are often used in software such as ANSYS or Abaqus to apply the compressive force before any external load is applied [61]. This ensures that the simulation starts from a physically accurate state.

### **2.8.2 Contact mechanics and nonlinear behavior**

There may be unpredictable details at the contact points. A bolted connection is essentially a connection with multiple interfaces: bolt head to flange, nut to flange, and thread to thread. These are not welded connections; they are frictional contacts. Under load, they can slip, separate, or change their stiffness. Modeling this requires advanced contact algorithms – typically the 'Penalty Method' or 'Lagrange Multipliers' – to ensure non-penetration while allowing relative movement. Friction is critical here. The standard coefficient for dry steel-superalloy contacts is approximately 0.15 - 0.20. If the model assumes a bonded contact, it artificially stiffens the joint, masks potential separation modes, and underestimates bolt stress [51]. Furthermore, the 'lift-off action' of flange bending that keeps the bolt open is a nonlinear geometric effect completely overlooked by linear analytical methods.

### 2.8.3 Thermo-mechanical coupling

At 700°C, thermal and mechanical loads cannot be analyzed separately. The simulation must be a 'Combined Thermo-Mechanical Analysis'. The workflow is typically sequential: First, the thermal analysis solves the heat transfer equations to generate the temperature map ( $\nabla T$ ) of the joint. Second, this temperature domain is mapped as a load to the structural model. This step is quite important because it captures 'Differential Thermal Expansion'. In the specific case of this thesis, the Inconel 718 bolt and flange expand at different rates. If the flange expands faster, it stretches the bolt and increases the preload. If the bolt expands faster, the compressive force is lost. FEA quantifies this precise change in preload ( $\Delta F_{th}$ ), something that manual calculations can only estimate with broad assumptions.

### 2.8.4 Estimating fatigue life: strain-life approach

For high-pressure turbine bolts, stress-based (S-N) fatigue life estimation is often insufficient because the material frequently exhibits local yielding at the thread roots. Instead, the 'Stress-Life' ( $\varepsilon - N$ ) method is preferred. This approach uses the Coffin-Manson relation:

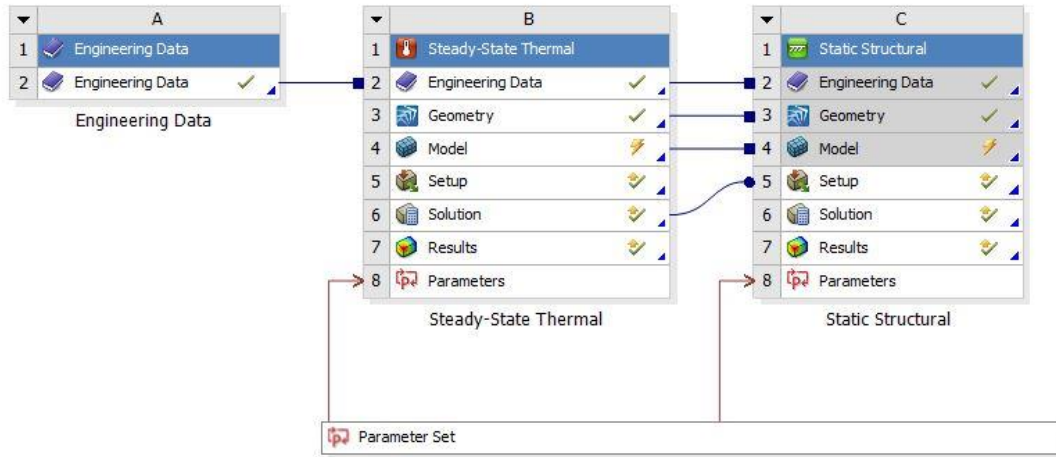
$$\frac{\Delta \varepsilon}{2} = \frac{\sigma'_f}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \quad (2.6)$$

Here, the total stress amplitude ( $\Delta \varepsilon/2$ ) is the sum of the elastic and plastic stress components. The precise stress history is extracted from the hot spot (usually the first thread) of the FEA model. Then, using this stress, life is calculated with much greater confidence than using the nominal stress alone. The life calculated here refers to the number of cycles until crack formation ( $N_f$ ) [36]. This allows for a precise assessment of Low Cycle Fatigue (LCF) during engine start and stop transitions. Consequently, the best design methodology is a hybrid approach.

### 2.8.5 Integrating fea with design methodology

Analytical methods (VDI 2230) provide the starting point: bolt size, target preload, and basic safety margins. Then, FEA acts as a 'stress test'. It reveals hidden risks: thermal degradation, contact separation, local plasticity. Based on the FEA results, the design can be iteratively improved. As a suggested solution, flange stiffness is changed or preload is increased to prevent loosening. In this way, design engineers ensure that

the final connection is not only mathematically correct but also operationally robust [51]. This synergy of classical mechanics and modern simulation is the gold standard for aerospace structural integrity. Figure 2.4 outlines the coupled thermo-mechanical FEA workflow adopted in this thesis.



**Figure 2.4 : FEA model workflow.**

## 2.9 Literature Gaps and Motivation of the Present Study

### 2.9.1 Literature gaps

The current literature in the aerospace field is undoubtedly extensive. We have a significant data library regarding the fundamental mechanics of bolted connections [27,30,31]. Similarly, the specific metallurgy of superalloys such as Inconel 718 is well documented. We have robust evidence confirming its suitability for hot-section connection elements [18,40]. However, a critical gap remains. While these issues are well understood individually, there is a striking lack of research that integrates them into a unified system. The mechanical behavior of an aircraft flange at high temperatures is not simply the sum of its parts. It is a complex, nonlinear interaction. Unfortunately, current studies often fail to capture this synergy.

The first and most dangerous gap lies in the way temperature is handled. The vast majority of current research examines bolted connections either at ambient temperature or under simplified, steady-state thermal conditions. When high-temperature effects are addressed, they are usually dealt with through linear approaches. Many analytical models rely on room temperature stiffness data. They simply apply linear "temperature reduction factors" to simulate the heat effect. This

approach is fundamentally flawed for gas turbine applications. It assumes that material degradation is predictable and linear, whereas it is not. At temperatures approaching 700°C, the reduction in stiffness and the onset of creep create nonlinear instabilities. Simple reduction factors cannot predict this complex behavior [25,28]. Consequently, the actual mechanical performance of these connections near their thermal limits is not adequately characterized.

A second major deficiency concerns the "Preload Blind Spot". We know that preloading is critical for the survival of the connection. We also know that it loosens over time. However, contemporary research often isolates the causes. Studies usually examine embedding, loosening, or vibration separately [37]. They rarely examine the collective attack. In a working engine, a bolt does not experience these events individually. It faces a compound loss mechanism. Thermal expansion fights against stiffness degradation. At the same time, vibration promotes relaxation. The current literature isolates these variables to simplify the mathematics. However, this isolation creates a blind spot regarding the cumulative effect on link integrity. In aviation, where separation margins are very thin, relying on isolated studies is not safe.

Thirdly, there is a significant "Geometric Gap". The literature loves simplicity. Numerous experimental and numerical studies rely on single-bolt test specimens or idealized concentric cylinders [31,36]. While these are useful for basic materials science, they are structurally misleading. A real gas turbine engine uses multi-bolt circumferential flanges. Single-bolt models cannot capture "group effects". They ignore the uneven load distribution caused by flange flexibility, manufacturing tolerances, and lift effect [25,34]. By ignoring structural cross-interaction between adjacent fasteners, current studies cannot represent the asymmetric reality of an operational engine.

Furthermore, fatigue research remains largely binary. Studies often focus only on High Cycle Fatigue (HCF) or Low Cycle Fatigue (LCF). However, aerospace gas turbine bolts are subjected to the "Overlapping" of both. They are subjected to the high-frequency vibration (HCF) of the rotor dynamics. On top of this, the large thermal transitions of the flight cycle (LCF) are added [24]. The interaction between these regimes has not been adequately reported. Thermal stress determines the average stress level for vibrational fatigue. Consequently, fatigue tests that examine only a single stress mode may not accurately reflect the actual service life of the component.

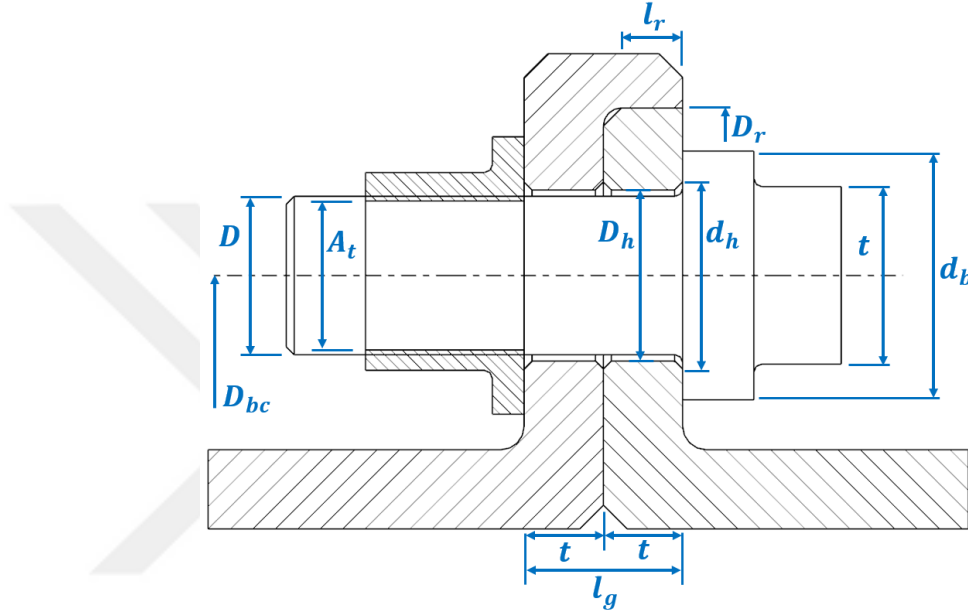
Finally, in terms of design methodology, there is a gap between theory and simulation. We have analytical standards such as VDI 2230. These are fast but rigid [25,28]. There is Finite Element Analysis (FEA). This is detailed but computationally expensive. There is little published research that bridges the gap between these in practice. Industry needs a "Hybrid Framework". We need a method that uses high-precision finite element analysis (FEA) to validate and calibrate faster analytical hand calculations for specific high-temperature scenarios. This integration is currently missing.

### **2.9.2 Motivation for this study**

This thesis directly addresses the identified gaps in the literature. It aims to go beyond the simplified, isolated analyses of the past. The primary objective is to perform a comprehensive mechanical evaluation of a realistic high-temperature aerospace connection. We focus on a representative multi-bolted flange configuration fixed with M10×1.25 aerospace standard fasteners. We target a challenging environment above 700°C. By aligning conventional bolted connection theory with complex thermo-mechanical finite element analysis, this study aims to quantify interactions neglected by others. The ultimate goal is to establish a validated systematic approach. We aim to better predict preload retention, stress distribution, and fatigue life, thereby improving the structural safety of future gas turbine engines.

### 3. JOINT GEOMETRY AND ANALYTICAL METHODOLOGY

The analysis focuses on a representative two-flange bolted joint geometry, as shown in Figure 3.1 [62,63].



**Figure 3.1 :** Geometric schematic of the analysed bolted joint assembly.

The key geometric and material parameters used for the analytical calculations are defined in Table 3.1.

**Table 3.1 :** Key Geometric and Material Parameters.

| Parameter                         | Symbol   | Value | Unit            |
|-----------------------------------|----------|-------|-----------------|
| Bolt Nominal Diameter             | $D$      | 10    | mm              |
| Bolt Tensile Area                 | $A_t$    | 61.2  | mm <sup>2</sup> |
| Grip Length                       | $l_g$    | 10    | mm              |
| Single Flange Thickness           | $t$      | 5     | mm              |
| Bolt Head Diameter                | $D_b$    | 15.7  | mm              |
| Applied Torque                    | $T$      | 65    | Nm              |
| Rabbit Engagement Length          | $l_r$    | 3.85  | mm              |
| Rabbit Diameter                   | $D_r$    | 351   | mm              |
| Flange Hole Diameter              | $D_h$    | 10.8  | mm              |
| Flange Hole Diameter with Chamfer | $d_h$    | 11.8  | mm              |
| Bolt Circle Diameter              | $D_{bc}$ | 165   | mm              |
| Nut Factor                        | $K$      | 0.20  | -               |

### 3.1 Analytical Stiffness and Preload Calculation

To validate the subsequent Finite Element Analysis (FEA), we must first establish a solid analytical foundation. We need a theoretical reference point. Therefore, this work adheres strictly to the established methodologies specified in VDI 2230 and NASA-STD-5020B.

The mechanical behavior of the preloaded connection is modeled as a 'bulk parameter system'. Effectively, the physics are simplified to parallel-arranged linear springs. In this theoretical model, the bolt is idealized as a tensile spring with axial stiffness  $k_b$ . Conversely, the clamped flanges function as a compression spring with a stiffness denoted as  $k_m$  [14,17,25,27,37,62,63].

The structural performance of the connection—especially in terms of fatigue life and separation resistance—is entirely dependent on load sharing. How are external tensile loads distributed among these components? This distribution is governed by the Connection Stiffness Constant (C). This parameter represents the ratio of the bolt's rigidity to the total rigidity of the entire assembly.

#### 3.1.1 Bolt axial stiffness( $k_b$ )

The axial rigidity of a bolt is determined by its elastic deformation under tension. In standard analytical practice, a bolt is typically modeled as a 'spring series' that sums the different rigidities of the unthreaded body and the threaded region. However, the specific fastener configuration in this study simplifies the physics. Since the bolt is fully threaded along its entire length ( $l_g$ ), the calculation is reduced to that of a smooth cylinder. This is simply defined by the tensile stress area ( $A_t$ ) and modulus of elasticity ( $E_b$ ) at the operating temperature, in order to eliminate the need for complex geometric segmentation [17,25,27,62,63].

The governing equation is:

$$k_b = \frac{A_t \cdot E_b}{L_g} \quad (3.1)$$

Calculation Parameters:

To accurately reflect the high-temperature environment, the material properties of Inconel 718 at 700°C are utilized [64]:

- Tensile Stress Area ( $A_t$ ): 61.2mm<sup>2</sup> [62,63]
- Modulus of Elasticity ( $E_b$ ) @ 700°C: 158,900N/mm<sup>2</sup>
- Grip Length ( $L_g$ ): 10mm

Step-by-Step Calculation:

First, the relevant geometric and material parameters are substituted into the stiffness equation:

$$k_b = \frac{61.2\text{mm}^2 \times 158,900\text{N/mm}^2}{10\text{mm}}$$

Solving for the numerator, which represents the product of the effective cross-sectional area and the material stiffness:

$$k_b = \frac{9,724,680\text{N}}{10\text{mm}}$$

Performing the final division by the grip length yields the axial stiffness of the bolt [17,25,27]:

$$k_b = 972,468\text{N/mm}$$

This calculated value serves as the reference stiffness for the bolt element in the analytical model [62,63].

### 3.1.2 Flange axial stiffness ( $k_m$ )

The pressure distribution within the flange material is not homogeneous. Therefore, calculating the rigidity of interlocking elements is more complex than calculating the rigidity of the bolt. As the bolt head applies pressure, the stress spreads outward into the flange, creating a barrel-shaped stress field. To approximate this behavior analytically, the pressure-cone (or frustum) method is widely employed, as detailed in VDI 2230 [17,25,27,62,63].

This method models the effective load-bearing volume as a hollow frustum of a cone expanding at a half-angle  $\alpha$  (typically 30°). Since the joint consists of two identical flange layers clamped together, the total member stiffness  $k_m$  is derived by calculating

the stiffness of a single flange ( $k_{single}$ ) and then treating the two flanges as springs connected in series [62,63].

The stiffness of a single flange ( $k_{single}$ ) is given by the VDI 2230 formulation [25]:

$$k_{single} = \frac{0.577 \cdot \pi \cdot E_m \cdot d}{\ln\left[\frac{(1.155 \cdot t + D - d)(D + d)}{(1.155 \cdot t + D + d)(D - d)}\right]} \quad (3.2)$$

Calculation Parameters:

- Modulus of Elasticity ( $E_m$ ): 158,900N/mm<sup>2</sup>
- Nominal Bolt Diameter ( $d$ ): 10mm
- Washer Face/Head Diameter ( $D$ ): 15.7mm
- Single Flange Thickness ( $t$ ): 5mm

Step-by-Step Calculation:

Step A: Calculate the Numerator (Pre-factor)

The numerator represents the material and geometric scaling factor.

$$\text{Numerator} = 0.577 \times \pi \times 158,900 \times 10 \approx 2,880,635$$

Step B: Calculate the Logarithmic Argument

The term inside the natural logarithm accounts for the geometric expansion of the stress cone through the thickness of the flange.

$$\begin{aligned} \text{Arg} &= \frac{(1.155 \times 5 + 15.7 - 10)(15.7 + 10)}{(1.155 \times 5 + 15.7 + 10)(15.7 - 10)} \\ \text{Arg} &= \frac{(5.775 + 5.7)(25.7)}{(5.775 + 25.7)(5.7)} = \frac{(11.475)(25.7)}{(31.475)(5.7)} \\ \text{Arg} &= \frac{294.9075}{179.4075} \approx 1.6438 \end{aligned}$$

Step C: Calculate Single Flange Stiffness

Using the derived numerator and logarithmic term, the stiffness of a single 5mm flange is found:

$$k_{single} = \frac{2,880,635}{\ln(1.6438)} = \frac{2,880,635}{0.4970} \approx 5,796,046 \text{N/mm}$$

Step D: Calculate Total Member Stiffness ( $k_m$ )

Because the joint comprises two identical flanges stacked in series, the total equivalent stiffness is half that of a single layer:

$$k_m = \frac{k_{single}}{2} = \frac{5,796,046}{2} = 2,898,023 \text{ N/mm}$$

### 3.1.3 Joint constant ( $C$ )

The Joint Constant ( $C$ ), also referred to as the stiffness ratio or load factor, is a dimensionless parameter that defines the proportion of an external tensile load that is transferred to the bolt. A lower value of  $C$  is generally desirable in design, as it indicates that the majority of the external load acts to relieve the compressive force on the flanges rather than increasing the tensile stress in the bolt, thereby enhancing fatigue resistance [17,25,27,62,63].

The governing equation is:

$$C = \frac{k_b}{k_b + k_m} \quad (3.3)$$

Step-by-Step Calculation:

Substituting the stiffness values calculated in the previous sections:

$$C = \frac{972,468}{972,468 + 2,898,023}$$

Calculating the total denominator (system stiffness):

$$C = \frac{972,468}{3,870,491}$$

Performing the final division yields the joint constant [17,25,27]:

$$C = 0.2512$$

This result implies that approximately 25.1% of the external load will be carried by the bolt, with the remaining 74.9% reducing the clamping force.

### 3.2 Torque-Tension and Preload ( $F_p$ )

The initial clamping force or preload ( $F_p$ ) is the most critical parameter in bolted connection design. In practice, this force is generated by applying a specific torque ( $T$ ) during assembly. The relationship between this input torque and the resulting stress is governed by the standard short-form equation. Importantly, this formula is based on the nut factor ( $K$ ), an empirical constant that sums up the complex friction interactions on the thread and bearing surfaces to a manageable value [14,17,65].

The governing equation is:

$$F_p = \frac{T}{K \cdot d} \quad (3.4)$$

Calculation Parameters:

- Applied Torque ( $T$ ): 65 Nm [14,17]
- Nut Factor ( $K$ ): 0.20 (Assumed for lubricated conditions) [14,17,65]
- Nominal Diameter ( $d$ ): 0.01 m (converted from 10 mm)

Step-by-Step Calculation (Nominal Preload) [14,17]:

$$F_{nom} = \frac{65}{0.20 \times 0.01}$$
$$F_{nom} = \frac{65}{0.002} = 32,500\text{N} = 32.5\text{kN}$$

#### 3.2.1 Uncertainty calculation

##### 3.2.1.1 Maximum preload case

Maximum preload was calculated in order to obtain highest preload on the bolt during the assembly. This maximum preload results from assembly uncertainty and should be used in calculations in some specific cases.

$$F_{max} = F_{nom} \times (1 + 0.20) \quad (3.5)$$

$$F_{max} = 32.5 \times 1.20 = 39.0\text{kN}$$

### 3.2.1.2 Minimum preload case

Minimum preload was calculated in order to obtain lowest preload on the bolt during the assembly. This minimum preload results from assembly uncertainty and should be used in calculations in some specific cases.

$$F_{min} = F_{nom} \times (1 - 0.20) \quad (3.6)$$

$$F_{min} = 32.5 \times 0.80 = 26.0\text{kN}$$





## **4. NUMERICAL ANALYSIS METHODOLOGY**

### **4.1 Introduction to Finite Element Analysis Strategy**

The analytical methods detailed in the previous section provide a fundamental basis for connection sizing. However, they are based on idealized assumptions. These methods generally fail to capture the nonlinear behavior of complex assemblies under combined thermomechanical loads. To overcome these limitations, a three-dimensional Finite Element Analysis (FEA) was performed using ANSYS Workbench ver. 19.2 [62,63].

The primary objective of this simulation is specific. It aims to elucidate complex nonlinear contact mechanics and quantify the relaxation of bolt preload due to thermal gradients. It also determines the localized stress-strain history necessary for subsequent Low Cycle Fatigue (LCF) assessment. The analysis adopts a combined approach. It integrates thermal body loads with nonlinear structural arrays to simulate the harsh service environment of a gas turbine engine operating at 700°C [14,17,62,63].

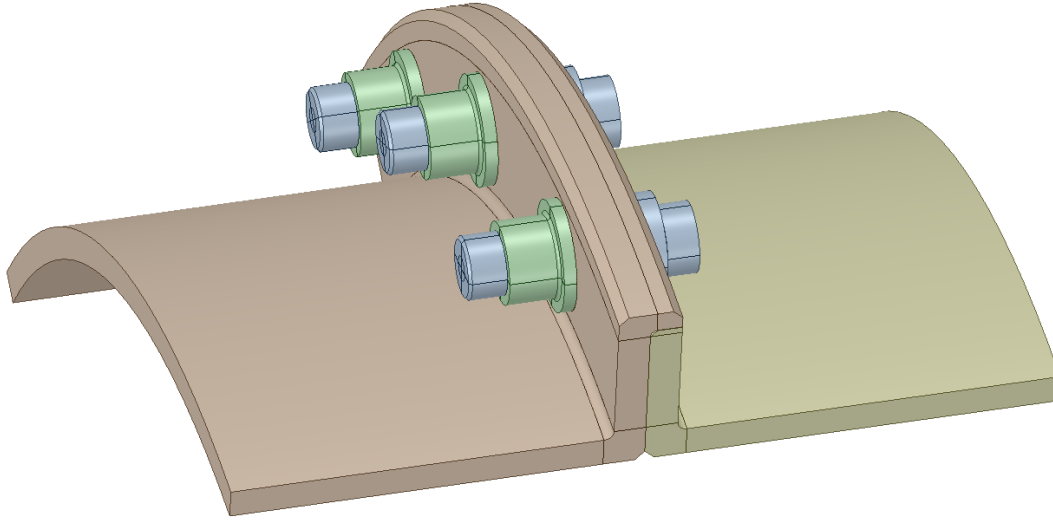
### **4.2 Geometric Modeling and Sub-Structuring**

The physical assembly corresponds to a critical IN718 bolted flange connection located within the engine housing. Simulating the entire 360-degree structure is generally unnecessary and costly. Therefore, to optimize computational efficiency without sacrificing the accuracy of stress results, a sub-configuration technique using rotational periodicity was employed [63,64].

#### **4.2.1 Sector model definition**

The entire assembly was not modeled. Instead, a 60° rotational symmetry sector was isolated for analysis. This sector is geometrically symmetrical and contains three full bolts. This is sufficient to capture the representative mechanical behavior of the repeating flange pattern. The logic here is strategic: reducing the model size allows us

to allocate computational resources more effectively. It facilitates a significantly higher mesh density in critical regions—particularly at bolt bodies and flange corners—and enables accurate resolution of steep stress gradients [62,63]. Created sector model can be seen in Figure 4.1.



**Figure 4.1 :** 60° rotational symmetry sector model geometry.

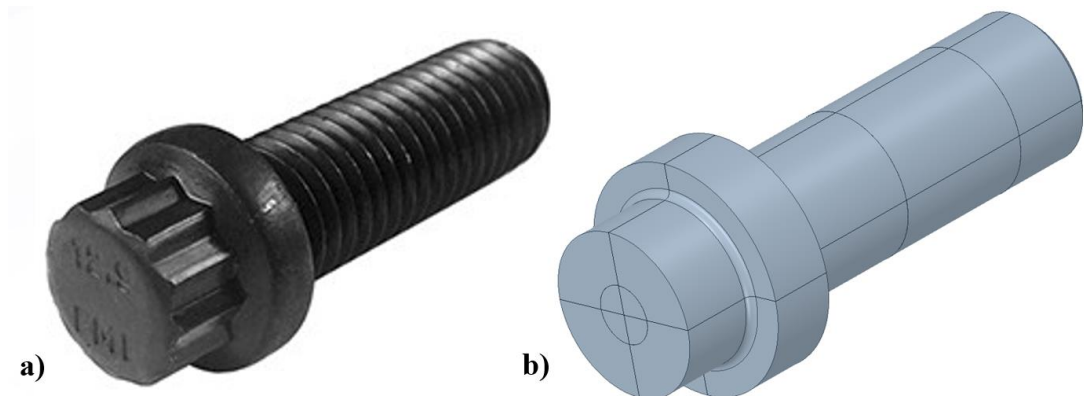
#### **4.2.2 Geometric simplifications**

Achieving a structured hexagonal mesh, the gold standard for accuracy, requires certain geometric simplifications. Raw CAD models are rarely simulation-ready. Therefore, the following modifications were implemented to ensure numerical stability without affecting the physical state:

First, Thread Representation was addressed. The detailed helical geometry of bolt and nut threads was simplified to flat cylindrical surfaces. Because explicitly modeling each thread in a bolted assembly is computationally very costly. It often creates artificial stress singularities and requires an excessive number of elements. Instead, load transfer mechanics are simulated using coupled contact formulations. This is a standard practice in macroscopic structural analysis and allows us to capture the stiffness of the thread geometry without computational noise [62,63,66]. Figure of the real bolt and simplified geometry of the bolt is shown in Figure 4.2.

Second, Feature Suppression was applied. Small cosmetic details such as small coupling radii and chamfers outside the primary load path were removed. These features contribute little to structural stiffness but are notorious for causing 'thin

elements'—poorly shaped elements that disrupt numerical convergence. Suppressing them provides a solid solution [62,63].



**Figure 4.2 :** a) Figure of the real bolt with threads b) The simplified 3D geometric model of the bolt.

#### 4.2.3 Discretization and mesh generation

Transforming a continuous geometric model into a discrete domain is never a trivial task. There is a calculated balance between computational efficiency and the absolute necessity of high-accuracy stress resolution. In this study, the decomposition process was performed in the ANSYS Workbench environment. Given the critical nature of stress states at the bolt-flange interface, a general automated mesh generation proved insufficient. Instead, a strategic combination of global dimensioning checks and aggressive local refinement was used to generate a high-quality mesh [62,63].

The mesh is built to resolve bearing interfaces under head and nut, flange-to-flange contact edges, and geometric transitions where stress gradients are expected. A global element size is used to keep the model uniform. Local refinements are introduced at contact edges and fillets to control peak stress sensitivity. Mesh definition:

- Element type: SOLID185
- Element order: Linear
- Meshing method: Sweep method
- Global sizing: 1.4 mm
- Local refinement: applied near bearing surfaces, flange contact edges, fillets, and geometric transitions
- Additional controls: edge sizing applied to improve smooth transitions on flange radii and on bolt/nut geometric changes

- Geometric preparation: bolts, nuts, and flanges were simplified where appropriate, and the bodies were partitioned (“chunked”) into sweepable volumes to enable sweep meshing consistently across all components
- Total nodes: 85,657
- Total elements: 67,582

#### 4.2.3.1 Element method and quality

To ensure structural accuracy—especially for the resolution of contact stresses—hexagonal elements were rigorously prioritized over quadrilateral elements. Although quadrilateral meshes are easier to generate, they generally exhibit artificial stiffness in bending. Therefore, only the 'Sweep Method' was used for flange sections and bolt bodies. To make this possible, the geometry required manual intervention. Complex parts were subjected to a 'Decomposition Process' where they were divided into simpler, sweepable sub-volumes. This preparatory phase of division was crucial. It enables the creation of structured hexagonal layers by ensuring topological consistency between the source and target faces. These structured grids provide superior convergence behavior compared to unstructured patches. Consequently, the Multi-Region method was suppressed to ensure the solver strictly adhered to the sweep algorithm [62,63].

Element quality metrics (ANSYS report values):

- Aspect ratio (min/avg/max): 1.079 / 1.6932 / 39.605
- Skewness (avg/max): 0.13834 / 0.94277
- Jacobian ratio (min/avg/max): 1 / 1.3046 / 5.2305
- Element quality (min/avg/max):  $9.38 \times 10^{-2}$  / 0.88918 / 0.99566
- Orthogonal quality (avg/min): 0.9631 /  $5.7227 \times 10^{-2}$

Mesh quality was evaluated using ANSYS metrics because frictional contact and pretension introduce localized nonlinearities that can amplify the numerical influence of poorly shaped elements. The reported average values indicate a generally well-shaped mesh suitable for nonlinear contact analysis, while the localized worst-case values justify the explicit reporting of quality metrics for transparency and repeatability, particularly in contact-sensitive stress evaluations.

#### 4.2.3.2 Global discretization strategy

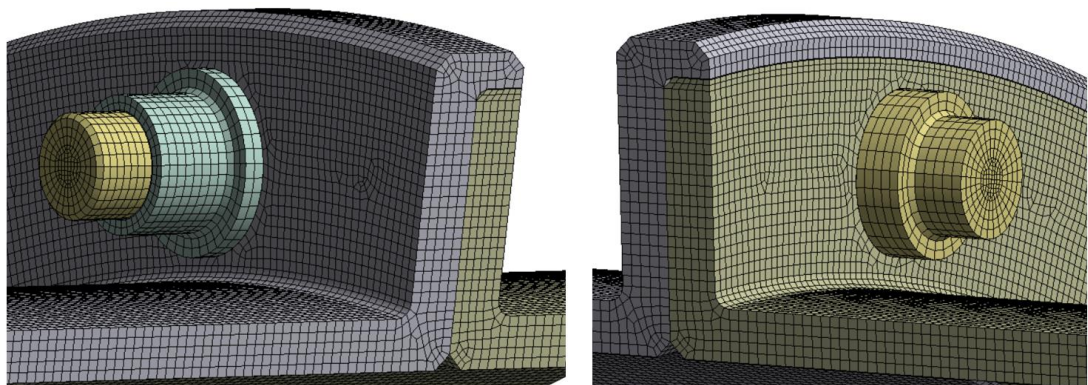
A 'Body Sizing' control was applied to the primary components to create a uniform basis. A maximum element size of 1.5 mm was applied for both flange bodies and connecting elements. This global sizing is a strategic choice: it allows the mass deformation of the joint to be captured with sufficient resolution while preventing the formation of excessively distorted (high aspect ratio) elements in non-critical regions [62,63].

#### 4.2.3.3 Local refinement and edge sizing

However, global sizing alone is insufficient. Due to the high stress concentrations found in bolted connections, a uniform network cannot accurately predict fatigue life. Therefore, localized 'Edge Sizing' controls were applied to improve the network in critical regions. The improvement strategy targeted specific physical phenomena:

**Bolt and Nut Interfaces:** The edges and bearing surfaces of the bolt body were intensively improved. This is necessary to accurately capture the steep slopes of contact pressure and shear transfer.

**Corner and Radius Regions:** Transition regions and corners under the bolt head were subjected to intensive edge sizing. Capturing the curvature is indispensable here, as these are primary regions of stress concentration factors ( $K_t$ ) that drive Low Cycle Fatigue (LCF) damage mechanisms [62,63,67]. Mesh details of the bolt, nut and flanges can be seen in Figure 4.3. Also mesh details of the whole assembly is shown in Figure 4.4.



**Figure 4.3 :** Mesh details of the bolt, nut and flanges.

Targeted Edge Sizing was used in conjunction with the sweeping method. The resulting mesh structure enables the precise calculation of the strain range ( $\Delta\epsilon$ ) required for subsequent strain-life fatigue assessment.

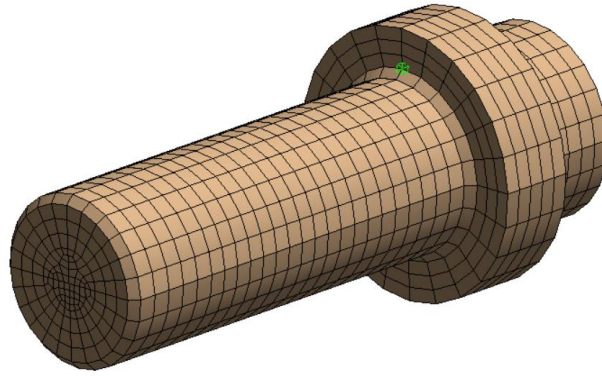


**Figure 4.4 :** Detailed view of the finite element mesh of whole assembly.

#### **4.2.3.4 Mesh convergence study**

In finite element analysis (FEA), a critical consideration is the trade-off between model fidelity and computational cost. The accuracy of computed localized stresses—particularly in highly stressed regions such as the thread roots of a preloaded bolted joint—is heavily dependent on the spatial discretization of the computational domain. Fasteners are inherently vulnerable due to the severe stress concentrations at the thread roots and the head-to-shank fillet. To ensure the reliability of the subsequent thermo-mechanical and static strength predictions for the IN718 bolt operating at 700°C, a systematic mesh convergence study was performed prior to the final structural assessment.

The convergence investigation focused on the most critical stress concentration region: the first engaged thread root. This location is established in the literature as a primary site for peak stresses and potential structural failure under the severe thermo-mechanical loading experienced in gas turbine applications. The maximum equivalent von Mises stress was monitored at a specific reference node situated exactly at the notch root of the bolt while iteratively refining the local element size. 3D view of the IN718 bolted joint highlighting the refined mesh zone and the specific nodal location at the thread root Radius can be seen in Figure 4.5.



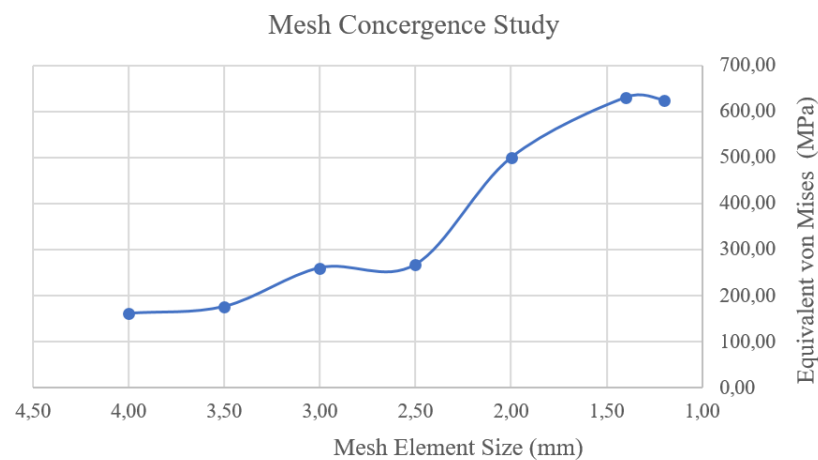
**Figure 4.5 :** Detailed view of the localized mesh refinement at the bolt thread root, indicating the exact nodal location monitored during the convergence study.

To objectively evaluate spatial convergence, the local element size at the critical thread region was systematically reduced from a coarse baseline of 4.00 mm down to a highly refined 1.20 mm. The corresponding equivalent von Mises stress values recorded at the reference node are summarized in Table 4.1.

**Table 4.1 :** Mesh Convergence Study Results at the Critical Thread Root.

| Mesh Size<br>(mm) | Equivalent von Mises Stress<br>(MPa) | Relative Change<br>(%) |
|-------------------|--------------------------------------|------------------------|
| 4.00              | 161.00                               | -                      |
| 3.50              | 176.00                               | 9.32%                  |
| 3.00              | 260.00                               | 47.73%                 |
| 2.50              | 267.00                               | 2.69%                  |
| 2.00              | 499.00                               | 86.89%                 |
| 1.40              | 630.00                               | 26.25%                 |
| 1.20              | 625.00                               | -0.79%                 |

The line graph shown in Figure 4.6 demonstrating the asymptotic convergence curve of the equivalent von Mises stress with respect to mesh refinement.



**Figure 4.6 :** Asymptotic behavior of the equivalent von Mises stress with respect to mesh refinement.

As observed in the extracted data, the calculated stress exhibits extreme sensitivity to mesh density at coarser element sizes. The severe fluctuations in stress—most notably the 86.89% surge when transitioning from 2.50 mm to 2.00 mm, and the 26.25% increase from 2.00 mm to 1.40 mm—clearly indicate that the coarser meshes completely fail to capture the steep stress gradients inherent to the geometry of the thread root notch. A high-fidelity 3D solid model is strictly necessary for the level of accuracy required to predict these localized stresses.

However, as the element size is reduced beyond 1.40 mm, the stress values begin to exhibit true asymptotic behavior. Refining the mesh size from 1.40 mm to 1.20 mm results in a slight stress reduction from 630.00 MPa to 625.00 MPa. The relative difference between the 1.40 mm and 1.20 mm mesh iterations is merely -0.79%. In the context of computational mechanics, a variance of less than 1% indicates that the solution has stabilized and the discretization error has been sufficiently minimized.

The plateauing of the stress response at 1.40 mm confirms that the mesh has sufficiently converged. Further refinement to 1.20 mm offers no significant gain in accuracy but would exponentially increase the computational cost and solution time, which is a critical consideration for the highly non-linear, thermo-mechanically coupled simulations required for this study. Therefore, a local mesh size of 1.40 mm at the critical thread root regions was selected as the optimum configuration. This ensures the high-fidelity stress extraction necessary for the subsequent structural integrity and static strength assessments, while maintaining a computationally efficient and robust finite element model.

#### **4.2.4 Contact mechanics and interface definitions**

The mechanical integrity of a bolted connection is primarily determined by the interaction between the interlocking parts. The ANSYS model includes nonlinear contact definitions to simulate the physical behavior of the connection components.

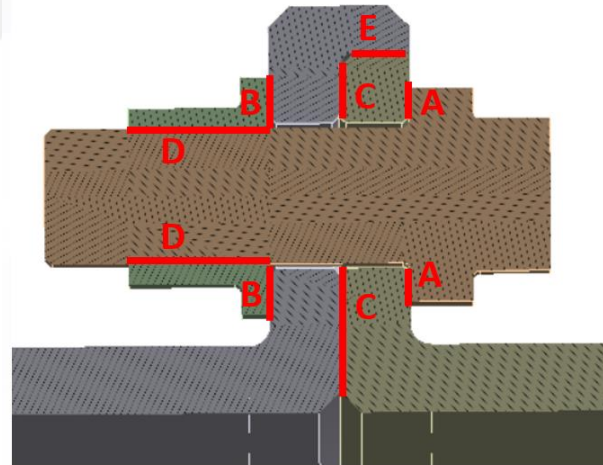
##### **4.2.4.1 Frictional interfaces**

To simulate the real-world behavior of the joint, Frictional Contacts were defined between the following surfaces [66]:

- **Flange-to-Flange Interfaces:** This definition allows the flanges to slide relative to one another (shear resistance). And crucially, to separate (gap opening) if the external tensile loads exceed the clamping force.
- **Bolt Head/Nut to Flange Faces:** This captures the effects of micro-slip and embedding under the fastener heads.
- **Friction Coefficient ( $\mu$ ):** A coefficient of friction of  $\mu = 0.15$  was applied to these interfaces. This value is consistent with standard engineering assumptions for lubricated IN718 aerospace fasteners [62-64].

#### 4.2.4.2 Bonded interfaces

The Bonded Contact formulation was used for the simplified interface between bolt threads and nut threads. This assumption treats the threaded connection as a perfectly rigid connection. It also transfers loads directly between the bolt body and the nut without allowing relative thread slippage or stripping [62,63,66]. All contacts are represented by letters and this visualization can be seen in Figure 4.7. Also contact types and friction coefficients are shown in Table 4.2



**Figure 4.7 :** Visualization of the contact regions represent with red letters.

**Table 4.2 :** Contact types and friction coefficients.

| Contact Name | Type       | Friction Coeff. |
|--------------|------------|-----------------|
| A            | Frictional | 0.15            |
| B            | Frictional | 0.15            |
| C            | Frictional | 0.15            |
| D            | Bonded     | -               |
| E            | Frictional | 0.15            |

#### **4.2.5 Loads and boundary conditions**

Real engine operation is not static; it is a sequence of events. Therefore, the analysis is structured as a multi-step nonlinear simulation. Loads are not unloaded all at once. Instead, they are applied sequentially over 4 different Time Steps. This chronological approach is vital to capturing the path-dependent nature of friction and plasticity [62,63].

To accurately replicate reality, Finite Element Analysis uses a multi-step quasi-static approach. This sequential strategy is essential. This strategy allows us to capture the change in contact state, the activation of friction, and the complex interplay between mechanical and thermal stresses [62,63]. The time schedule is divided as follows:

##### **4.2.5.1 Assembly (interference fit)**

The first stage focuses on the geometry. The solver resolves the 'int fit' or penetrations between the contacting surfaces. There is no external load on the connection in this step. This establishes a stable contact equilibrium. It provides a mathematically consistent starting point that simulates the physical alignment of the components before a wrench touches the bolt [14,17].

##### **4.2.5.2 Bolt preload**

Once the contact is stable, the bolt is tightened. The ANSYS 'Bolt Preload' element is used to apply a specific axial force to the shaft. This creates the initial compressive force on the flange. In this step, the elements forming the connection become locked together by the preload force [14,17].

##### **4.2.5.3 Pressure loading**

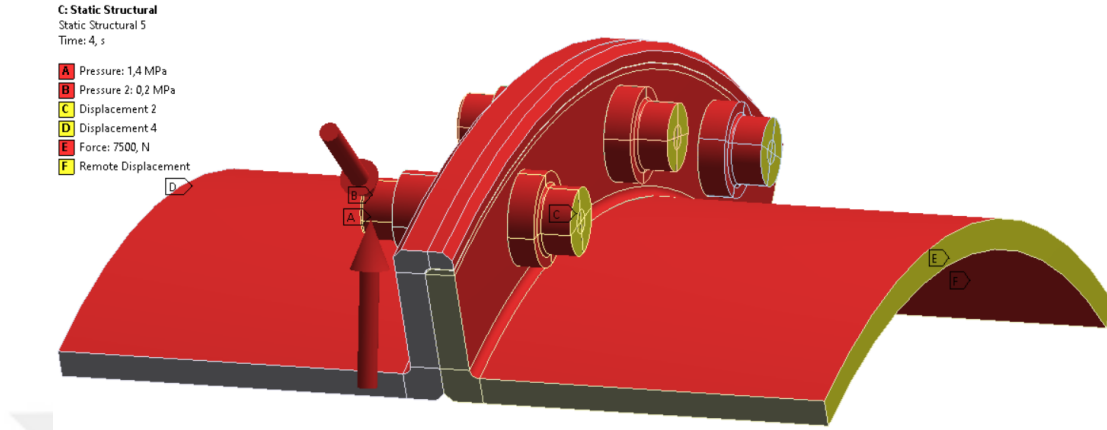
Next, mechanical loads are applied. An internal compressive load, representing the gas path, is applied to the inner surfaces. This creates tensile ring stresses and attempts to separate the flanges. By applying this before heat, we can isolate the mechanical rigidity of the joint under purely mechanical conditions [14,17,62,63].

##### **4.2.5.4 Thermal loading & global reaction**

The final step exposes the challenging environment. A temperature range corresponding to 700°C is applied to the structure. This causes thermal expansion. Since the bolt and flange can expand at different rates, significant thermal stresses

arise. The fully-coupled solution represents the final equilibrium state: the superposition of preloading, compressive stress and thermal expansion [14,17].

All of these boundary conditions are shown in Figure 4.8.



**Figure 4.8 :** Boundary conditions setup and applied loads and boundary conditions.

#### 4.2.6 Material model and environmental conditions

The validity of numerical results depends on the accurate description of material behavior and the appropriate representation of geometric properties. In this study, a simplified geometric approach combined with a thermomechanical material model was used.

##### 4.2.6.1 Material constitutive model

The material for the flange and bolt components is Inconel 718 (UNS N07718). Material is in the solution-treated and aged condition (AMS 5663). The material behavior in the Finite Element Analysis (FEA) is defined using a temperature-dependent Linear Elastic Isotropic model [64].

- **Elastic Properties:** The simulation uses the temperature-dependent Modulus of Elasticity ( $E$ ) and Poisson's ratio ( $\nu$ ). As stated in the material data, the stiffness of Inconel 718 decreases by approximately 20.6% at the operating temperature of 700°C [62-64].
- **Thermal Expansion:** The coefficient of thermal expansion ( $\alpha$ ) is defined as a function of temperature. It helps to capture the differential expansion between the bolt and the flange, which is the primary driver of thermal loading [62,63].

#### 4.2.6.2 Geometric simplification and stress interpretation

To optimize computational efficiency, the bolt threads are not explicitly modeled in the FEA geometry. The bolts are represented as smooth cylinders with an equivalent tensile stress area ( $A_t$ ). Consequently, stress results obtained from FEA are interpreted as nominal stresses rather than local peak stresses [62,63].

- **Linearity and Post-Processing:** The structural analysis assumes linear elastic behavior. Since the physical notch effect of the threads is not present in the mesh, the stress concentrations at the thread roots are not captured directly by the solver.
- **Fatigue Correction:** For the Low Cycle Fatigue (LCF) life prediction, these nominal elastic stresses are post-processed analytically. A theoretical Stress Concentration Factor ( $K_t$ ) is applied to obtain the realistic stress value on the thread geometry [67].

#### 4.2.6.3 Environmental and thermal boundary conditions

The analysis simulates the steady-state operating environment of a gas turbine casing [62,63].

- **Thermal Loading:** A global operating temperature of 700°C is applied.
- **Thermal Gradient:** A thermal gradient was simulated to represent heat transfer through the hot gas path. A radial temperature gradient ranging from 700°C at the inner flange diameter to 690°C at the outer diameter was applied. This gradient is critical for simulating the effects of thermal bending and the resulting change in bolt preload [14,17,62,63].

#### 4.2.7 Solution methodology

The numerical solution is executed using the ANSYS Mechanical solver. Crucially, the 'Large Deflection' (NLGEOM) setting was explicitly enabled. For any analysis involving contact, this setting is not optional; it is mandatory. Because the stiffness of the system is not constant. It changes abruptly as contact interfaces flip between 'open' and 'closed' states. Large Deflection accounts for these non-linear stiffness updates and significant geometric changes. The solver iterates at each substep, hunting for equilibrium until strict force and displacement convergence criteria are fully satisfied, ensuring the result is physically stable [62,63].

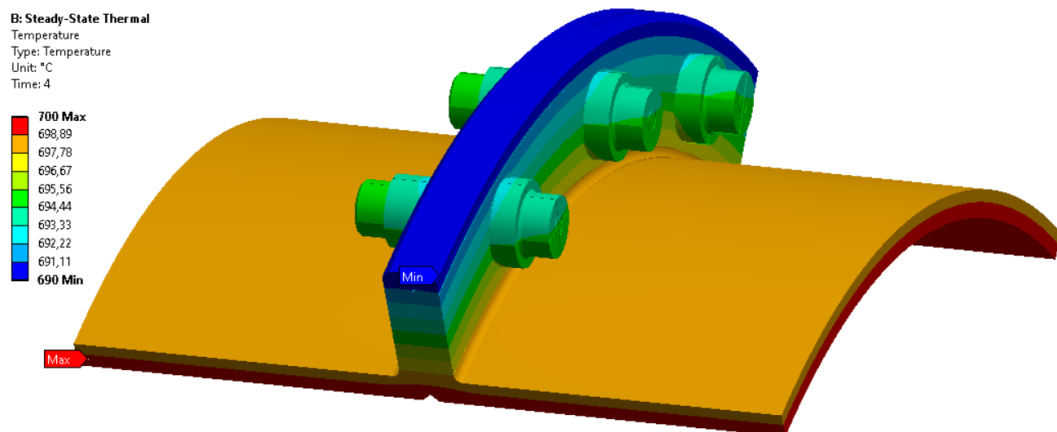
## 5. RESULTS AND DISCUSSION

### 5.1 Finite Element Analysis Results and Preload Verification

This chapter comprehensively presents the results of the thermo-mechanical Finite Element Analysis (FEA) of the flange modeled in this thesis. The research is systematic. We analyze the findings in three main areas: temperature distribution map, structural stress evolution from assembly to operation, and quantitative evaluation of the bolt preload holding ratio. However, validation of the simulation data is required. Therefore, the FEA results are rigorously compared with previously established analytical calculations. This comparison validates the numerical model against fundamental mechanical design principles [14,17,64].

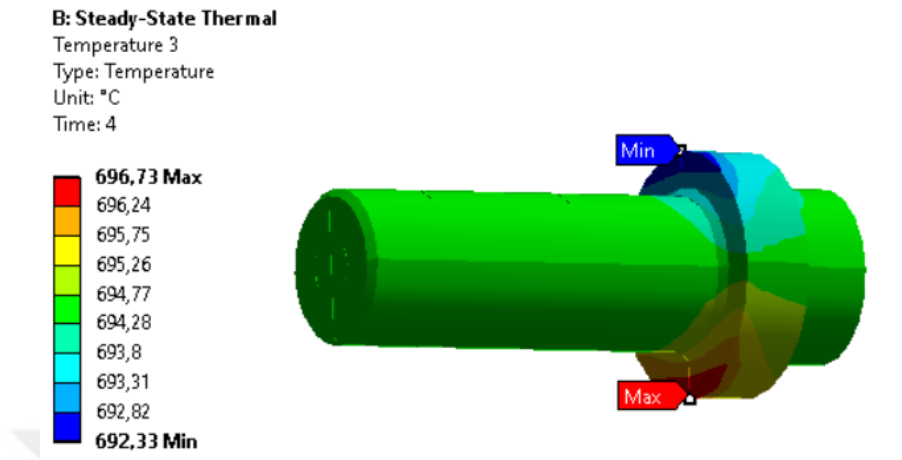
#### 5.1.1 Steady-state thermal analysis

The combined analysis begins with the determination of the temperature field. This is the starting point because temperature is the primary driver of both thermal expansion and material property degradation. Steady-state thermal analysis generates a precise distribution map. This result is not merely visual; it is applied as 'Body Load' in the subsequent structural simulation and imposes thermal reality along a 60° sector in the operating condition [62,63]. Result of the steady state thermal analysis is shown in Figure 5.1. This shows the color map ranging from approx. 690°C to 698°C.



**Figure 5.1 :** Steady-State Temperature Distribution contour plot.

As illustrated in Figure 5.2, the model predicts a relatively uniform temperature distribution with a minimal gradient. The bolt body operates at approximately 696.73°C, while the flange regions stabilize at 692.33°C [62,63].



**Figure 5.2 :** Temperature distribution of the bolt in operating condition.

Despite the thermal gradient ( $\Delta T$ ) between the bolt and the flange is small ( $< 5^{\circ}\text{C}$ ), the absolute temperature magnitude is critical. At these elevated temperatures, the Elastic Modulus ( $E$ ) of Inconel 718 drops significantly (from  $\approx 200$  GPa at room temperature to  $\approx 159$  GPa at  $700^{\circ}\text{C}$ ). This reduction in stiffness, combined with the coefficient of thermal expansion ( $\alpha$ ), fundamentally alters the stress equilibrium of the joint [64].

### 5.1.2 Structural stress distribution

The structural response of the joint was evaluated at two critical time steps: the Assembly State (Room Temperature, Preload Only) and the Operating State ( $700^{\circ}\text{C}$ , Preload + Pressure + Boundary Force) [14,17,62,63].

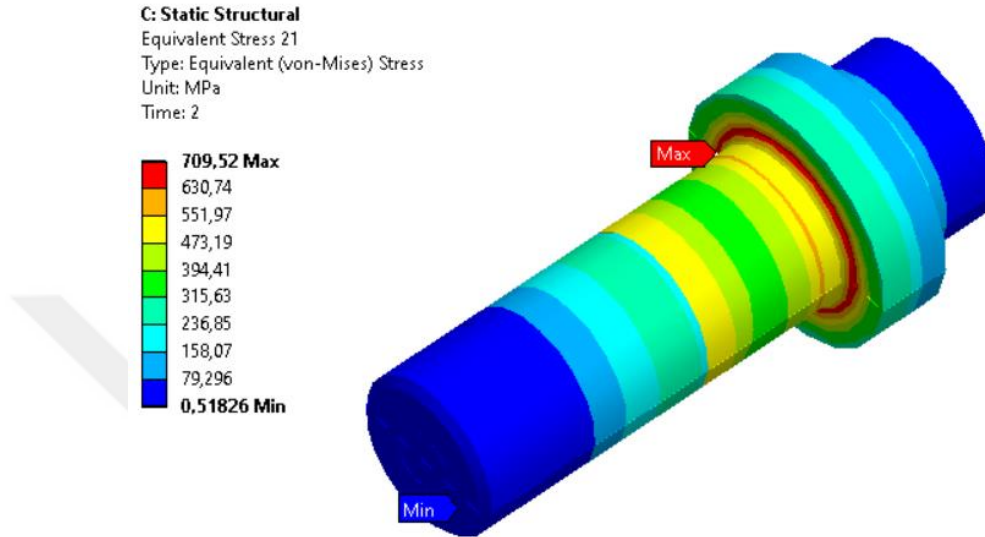
#### 5.1.2.1 Assembly state (time = 2s)

In the initial assembly step, the bolts are pretensioned with the target preload of 32,500 N. The Equivalent (Von-Mises) stress distribution reflects the mechanics of a classic preloaded joint [14,17].

- **Bolts:** The maximum stress is concentrated in the bolt body and at the corner radius below the bolt head, corresponding to the applied tensile load.

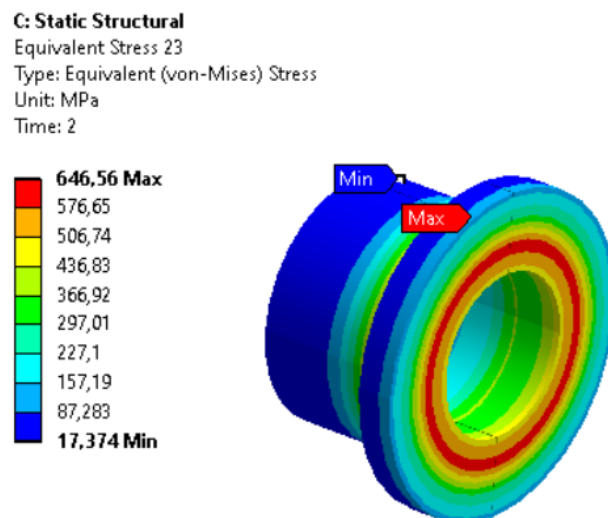
- **Flanges:** Compressive stress cones radiating from the bolt head bearing surfaces toward the flange body are clearly visible. Stress levels in the net cross-section of the flange remain well below the elastic limit of IN718 at room temperature [64].

Equivalent Von-Mises stress (MPa) contour plot of the bolt at the assembly condition is shown in Figure 5.3.



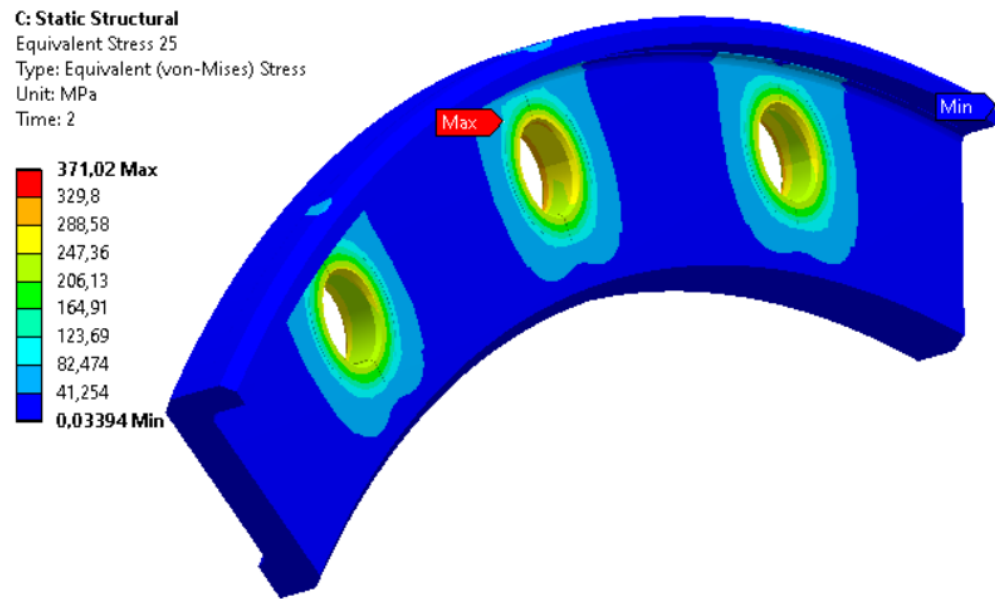
**Figure 5.3 :** Equivalent Von-Mises stress (MPa) results of the bolt (time = 2s).

Equivalent Von-Mises stress (MPa) contour plot of the nut at the assembly condition is shown in Figure 5.4.



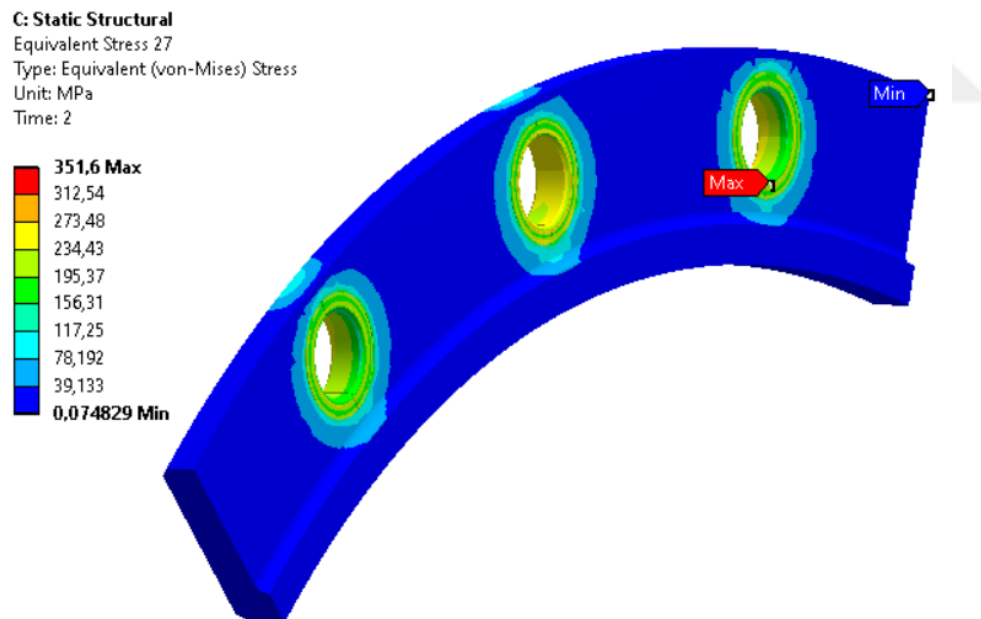
**Figure 5.4 :** Equivalent Von-Mises stress (MPa) results of the nut (time = 2s).

Equivalent Von-Mises stress (MPa) contour plot of the Flange1 at the assembly condition is shown in Figure 5.5.



**Figure 5.5 :** Equivalent Von-Mises stress (MPa) results of the Flange 1 (time = 2s).

Equivalent Von-Mises stress (MPa) contour plot of the Flange2 at the assembly condition is shown in Figure 5.6



**Figure 5.6 :** Equivalent Von-Mises stress (MPa) results of the Flange 2 (time = 2s).

#### 5.1.2.2 Operating state (time = 4s)

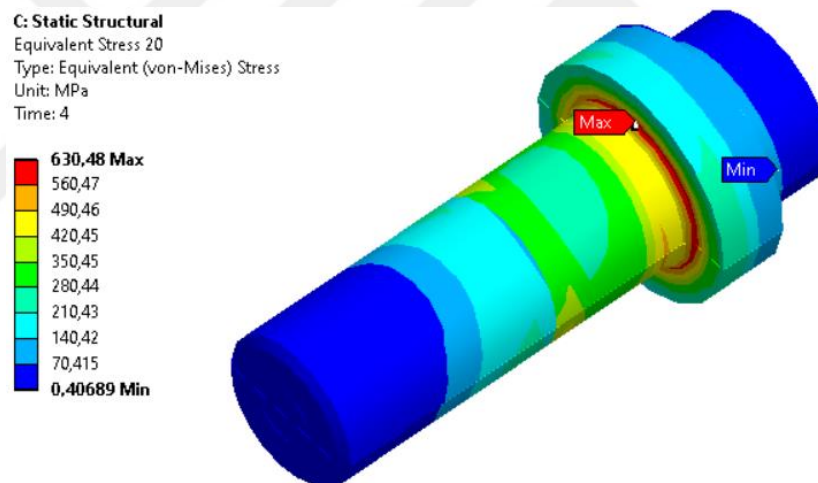
With the application of thermal body loads combined with an internal pressure of 1.4 MPa, the internal stress state changes significantly. Figure 5.7 presents a visual

representation of the global Equivalent von Mises stress distribution under these specific operating conditions.

Here, we observe a seemingly contradictory phenomenon. We might naturally expect the internal pressure creating axial thrust to shear the bolts and increase the stress. However, the opposite happens. The overall stress magnitude in the bolts actually decreases compared to the initial assembly state.

This counterintuitive behavior is how physics works. It is a direct consequence of 'thermal softening'. The dramatic decrease in the Young's Modulus at point (  $700^{\circ}\text{C}$  ) indicates that the material has become less rigid. Consequently, the force required to maintain the current bolt elongation decreases, leading to a net loosening of the connection.

Equivalent Von-Mises stress (MPa) contour plot of the bolt at the operating condition is shown in Figure 5.7.

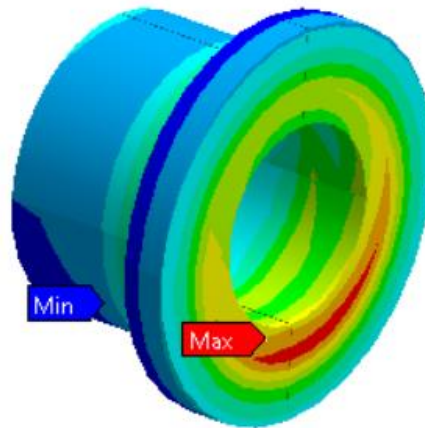
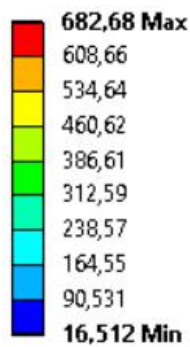


**Figure 5.7 :** Equivalent Von-Mises stress (MPa) results of the bolt (time = 4s).

Even at the reduced operating stress, the peak value remains concentrated at the first engaged thread root, where the change in section creates a stress concentration. This location stays the most highly stressed point of the bolt and therefore governs the fatigue assessment performed later in this chapter.

Equivalent Von-Mises stress (MPa) contour plot of the nut at the operating condition is shown in Figure 5.8.

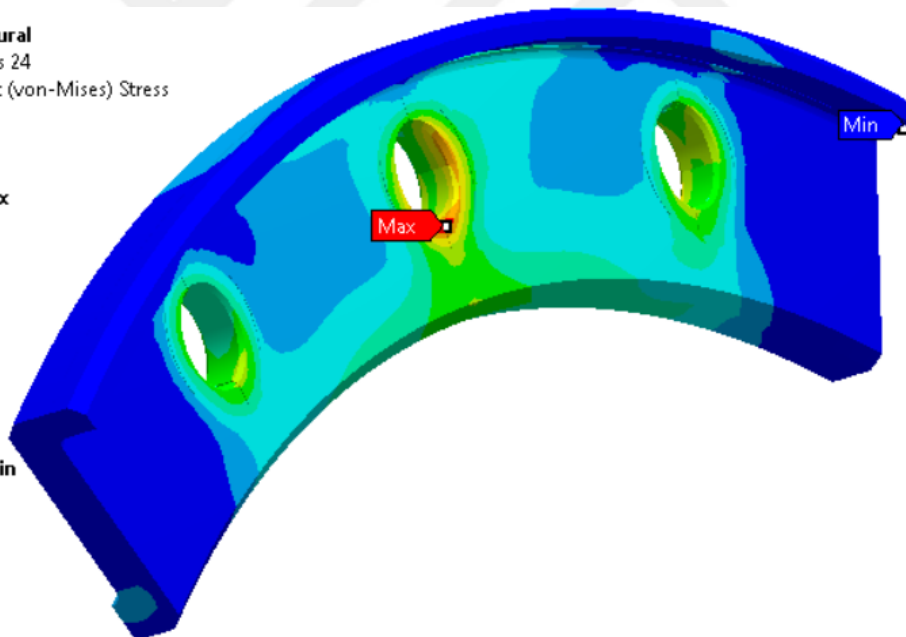
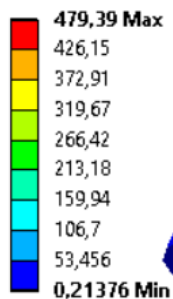
**C: Static Structural**  
 Equivalent Stress 22  
 Type: Equivalent (von-Mises) Stress  
 Unit: MPa  
 Time: 4



**Figure 5.8 :** Equivalent Von-Mises stress (MPa) results of the nut (time = 4s).

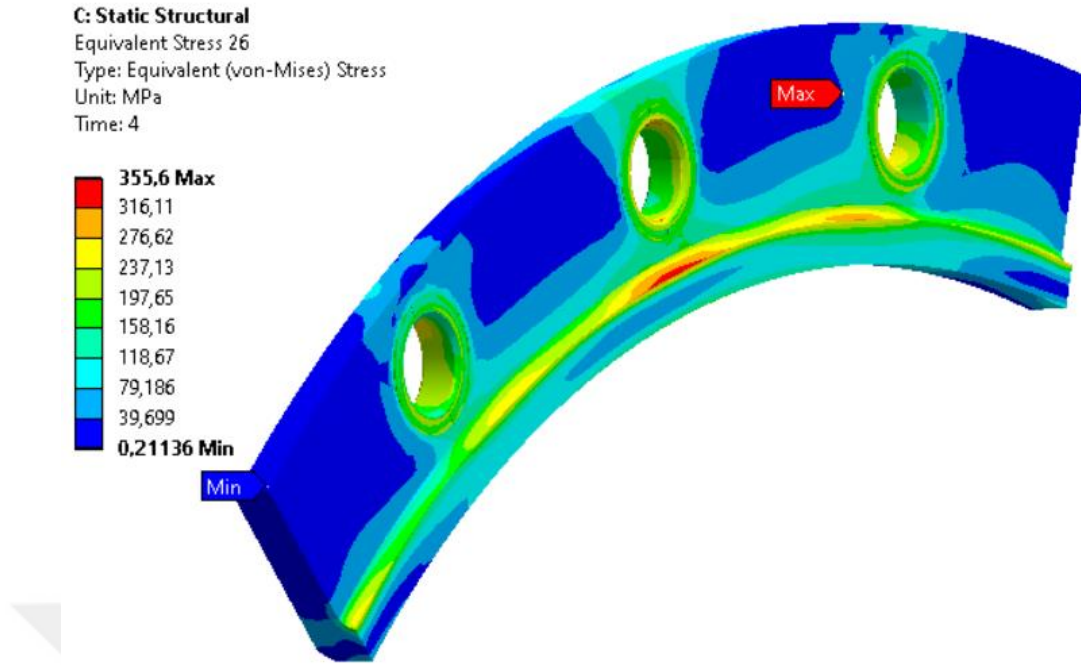
Equivalent Von-Mises stress (MPa) contour plot of the Flange1 at the operating condition is shown in Figure 5.9.

**C: Static Structural**  
 Equivalent Stress 24  
 Type: Equivalent (von-Mises) Stress  
 Unit: MPa  
 Time: 4



**Figure 5.9 :** Equivalent Von-Mises stress (MPa) results of the Flange 1 (time = 4s).

Equivalent Von-Mises stress (MPa) contour plot of the Flange2 at the operating condition is shown in Figure 5.10.



**Figure 5.10 :** Equivalent Von-Mises stress (MPa) results of the Flange 2 (time = 4s).

### 5.1.3 Preload loss investigation: analytical vs. numerical

One of the primary objectives of this thesis is to quantify the loss of bolt preload due to thermal effects. A significant reduction in clamping force can lead to joint separation, leakage, or fatigue failure. To validate the FEA results, an analytical "hand calculation" was performed using the stiffness methodology adapted from Shigley and VDI 2230 standards [14,25,27,62,63].

#### 5.1.3.1 Analytical model

The final preload ( $F_{final}$ ) at operating temperature is estimated analytically using Equation, which accounts for the change in joint stiffness and differential thermal expansion [14,17].

$$F_{final} = k_{total,op} \cdot \left[ \left( \frac{F_{initial}}{k_{total,assembly}} - \delta_{embed} \right) + L_g(\alpha_b \Delta T_b - \alpha_m \Delta T_m) \right] - \Delta F_{relax}$$

Instantaneous Operating Preload ( $F_{operating}$ )

Relaxation Loss

(5.1)

- $F_{initial}$  (Initial Preload @ 23°C): 32,500 N (Reference value from FEA) [14,17]

- $T_{assembly}$  (Assembly Temperature): 23°C
- $T_{op,bolt}$  (Bolt Operating Temp.): 695°C
- $T_{op,flange}$  (Flange Avg. Operating Temp.): 694.5°C
- $A_t$  (Bolt Tensile Stress Area): 61,2 mm<sup>2</sup> (per M10x1.25 standard) [62,63]
- $L_g$  (Grip Length): 10 mm
- $t$  (Single Flange Thickness): 5 mm
- $d$  (Bolt Nominal Diameter): 10 mm
- $D$  (Washer Face Diameter): 15.7 mm
- $E_{assembly}$  (Modulus of Elasticity @ 23°C): 200,000 N/mm<sup>2</sup>
- $E_{op}$  (Modulus of Elasticity @ 700°C): 158,900 N/mm<sup>2</sup>
- $\alpha$  (Mean Coeff. of Thermal Expansion):  $14.5 \times 10^{-6}$  1/°C

Here is the summary of the analytical calculations, formulas, components, and their respective values [62,63].

This calculation determines the instantaneous preload loss due to thermal effects and stiffness changes, assuming zero time-dependent relaxation (creep), to align with the FEA results [14,17].

#### 5.1.3.2 Calculation of assembly state (T = 23°C)

##### Bolt stiffness (assembly)

$$k_{b,assembly} = \frac{A_t \cdot E_{assembly}}{L_g} \quad (5.2)$$

$$k_{b,assembly} = \frac{61.2 \text{ mm}^2 \cdot 200,000 \text{ N/mm}^2}{10 \text{ mm}}$$

$$k_{b,assembly} = 1,224,000 \text{ N/mm}$$

- $A_t$  (Bolt Tensile Stress Area): 61,2 mm<sup>2</sup> (per M10x1.25 standard) [62,63]
- $E_{assembly}$  (Modulus of Elasticity @ 23°C): 200,000 N/mm<sup>2</sup>
- $L_g$  (Grip Length): 10 mm

### Flange stiffness (assembly)

$$k_{m,assembly} = k_{m,op} \cdot \left( \frac{E_{assembly}}{E_{op}} \right) \quad (5.3)$$

$$k_{m,op} = 2,898,023.15 \text{ N/mm}$$

$$k_{m,assembly} = 2,898,023.15 \text{ N/mm} \cdot \left( \frac{200,000}{158,900} \right)$$

$$k_{m,assembly} = 3,647,235.33 \text{ N/mm}$$

- $E_{assembly}$  (Modulus of Elasticity @ 23°C): 200,000 N/mm<sup>2</sup>
- $E_{op}$  (Modulus of Elasticity @ 700°C): 158,900 N/mm<sup>2</sup>

### Total system stiffness (assembly)

$$k_{total,assembly} = \left( \frac{1}{k_{b,assembly}} + \frac{1}{k_{m,assembly}} \right)^{-1} \quad (5.4)$$

$$k_{total,assembly} = \left( \frac{1}{1,224,000} + \frac{1}{3,647,235.33} \right)^{-1}$$

$$k_{total,assembly} = ((8.16993 \times 10^{-7}) + (2.74180 \times 10^{-7}))^{-1}$$

$$k_{total,assembly} = (1.091173 \times 10^{-6})^{-1}$$

$$k_{total,assembly} = 916,447 \text{ N/mm}$$

### Initial system elongation

$$\delta_{initial} = \frac{F_{initial}}{k_{total,assembly}} \quad (5.5)$$

$$\delta_{initial} = \frac{32,500 \text{ N}}{916,447.8 \text{ N/mm}}$$

$$\delta_{initial} = 0,03546 \text{ mm}$$

- $F_{initial}$  (Initial Preload @ 23°C): 32,500 N (Reference value from FEA) [14,17]

### 5.1.3.3 Calculation of operating state (T = 700°C)

#### Flange stiffness (operating)

$$k_{m,op} = 2,898,023.15 \text{ N/mm}$$

#### Bolt stiffness (operating)

$$k_b = \frac{A_t \cdot E_b}{l_g} \quad (5.6)$$

$$k_b = \frac{61,2 \text{ mm}^2 \cdot 158,900 \text{ N/mm}^2}{10 \text{ mm}}$$

$$k_b = 972,468 \text{ N/mm}$$

#### Total system stiffness (operating)

$$k_{total,op} = \left( \frac{1}{k_{b,op}} + \frac{1}{k_{m,op}} \right)^{-1} \quad (5.7)$$

$$k_{total,op} = \left( \frac{1}{972,468} + \frac{1}{2,898,023.15} \right)^{-1}$$

$$k_{total,op} = ((1.028311 \times 10^{-6}) + (0.345094 \times 10^{-6}))^{-1}$$

$$k_{total,op} = (1.373405 \times 10^{-6})^{-1}$$

$$k_{total,op} = 728,111.9 \text{ N/mm}$$

#### Differential thermal elongation

$$\Delta L_{thermal} = L_g \cdot \alpha \cdot (T_{op,bolt} - T_{op,flange}) \quad (5.8)$$

$$\Delta L_{thermal} = 10 \text{ mm} \cdot (14.5 \times 10^{-6} \text{ 1/}^\circ\text{C}) \cdot (695^\circ\text{C} - 694.5^\circ\text{C})$$

$$\Delta L_{thermal} = 10 \cdot (14.5 \times 10^{-6}) \cdot (0.5)$$

$$\Delta L_{thermal} = +0.0000725 \text{ mm}$$

This term captures the small thermal-expansion mismatch between the bolt and the clamped flanges, and feeds the final preload calculation that quantifies the clamp-force loss.

#### 5.1.3.4 Final preload and loss calculation

##### Final total elongation

$$\delta_{final} = \delta_{initial} + \Delta L_{thermal} \quad (5.9)$$

$$\delta_{final} = 0.035462 \text{ mm} + 0.0000725 \text{ mm}$$

$$\delta_{final} = 0.03553 \text{ mm}$$

##### Final preload (operating)

$$F_{final} = k_{total,op} \cdot \delta_{final} \quad (5.10)$$

$$F_{final} = 728,111.9 \text{ N/mm} \cdot 0.0355345 \text{ mm}$$

$$F_{final} = 25,873 \text{ N}$$

##### Total preload loss

$$\Delta F_p = F_{initial} - F_{final} \quad (5.11)$$

$$\Delta F_p = 32,500 \text{ N} - 25,873 \text{ N}$$

$$\Delta F_p = 6,626 \text{ N}$$

#### 5.1.3.5 Extraction of bolt loads from FEA

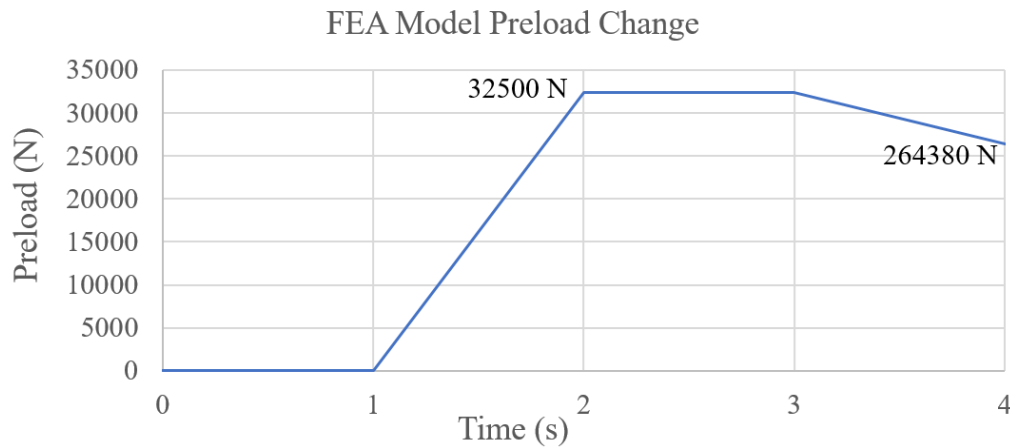
Defining how the bolt load history is extracted from the numerical model needs to be done before performing the comparative analysis. Unlike the analytical method which calculates the force according to the stiffness equations, the FEA solver calculates the nodal forces directly from the stress field [62,63].

To track the instantaneous compressive force throughout the loading history, a Force Reaction probe was defined across the entire bolt body (Bolt Body). This probe sums the nodal reaction forces in the axial direction (Z-axis) at each time step and provides the precise value of the total tensile load acting on the bolt [62,63].

Data points were extracted for two critical states to assess the preload evolution [14,17]:

- **Assembly state (time = 2):** The reaction force is recorded immediately after the "Bolt Pretension" step is completed and locked. This value represents the initial clamping force ( $F_{initial}$ ) at room temperature ( $23^{\circ}C$ ).
- **Operating state (time = 4):** The reaction force is recorded at the final operating step, where the joint is subjected to the combined effects of internal pressure (1.4 MPa), boundary reactions, and the steady-state thermal field ( $700^{\circ}C$ ). This value represents the residual preload ( $F_{final}$ ) acting on the joint under service conditions [14,17].

The setup for the Force Reaction probe and the resulting load history graph are illustrated in Figure 5.11. This results were extracted from bolt tensile area via "Force Reaction" command. This figure includes the time-history graph below it showing the force level at Time 2 (Preload) and the drop at Time 4 (Operating) [14,17].



**Figure 5.11 :** Preload change during operation.

The values obtained from this probe serve as the primary inputs for the comparison with the analytical hand calculations presented in the following section [62,63].

#### 5.1.3.6 Comparative results

The comparison summarized in Table 5.1, shows excellent agreement between the analytical model and the FEA results.

**Table 5.1 :** Comparison of Analytical vs. FEA Preload Loss ( $700^{\circ}C$ ) [14,17].

| Method           | Preload @ $23^{\circ}C$ | Preload @ $700^{\circ}C$ | Preload Drop (N) | Preload Drop (%) |
|------------------|-------------------------|--------------------------|------------------|------------------|
| Analytical Model | 32500                   | 25873                    | 6626             | 20,38            |
| FEA Result       | 32500                   | 26438                    | 6062             | 18,65            |

Both methods confirm a critical preload loss of 18-20% due to thermal effects alone, before considering long-term stress relaxation [14,17].

## 5.2 Structural Integrity Margin and Life Calculations

Following the validation of the thermal model and the quantification of preload evolution, the structural integrity of the bolted joint was assessed against critical failure modes. The margins of safety were calculated based on the forces and moments extracted from the validated Finite Element Analysis (FEA) model [14,17,62,63].

### 5.2.1 Bolt tensile stress assessment ( $\sigma_b$ )

The primary criterion for the static strength of the bolt is the average tensile stress across the bolt shank. To prevent permanent deformation, the tensile stress must remain within the elastic region of the material. Therefore, the calculated stress is compared against the 0.2% Yield Strength ( $S_y$ ) of Inconel 718 at the respective temperature of the load case [64].

The assessment was conducted for two distinct and critical loading scenarios [62,63]:

- **Assembly condition (24°C):** This situation evaluates the bolt under "Maximum Preload". In this step, the uncertainty (scatter) in torque application is taken into account and it is checked whether the bolt will deform during assembly before service loads are applied [14,17].
- **Operating condition (700°C):** This case evaluates the bolt under "Nominal Preload" combined with external pressure and thermal loads. The criteria for this case utilize the degraded material properties of IN718 at the elevated operating temperature [14,17,64].

The average bolt tensile stress,  $\sigma_b$ , is calculated using the standard axial stress formula:

$$\sigma_b = \frac{F_{b,t}}{A_t} \quad (5.12)$$

Where:

- $F_{b,t}$  is the total tensile load acting on the bolt (extracted from FEA reaction forces).

- $A_t$  is the tensile stress area of the bolt. For the M10 fastener used in this study,  $A_{min} = 61.2 \text{ mm}^2$  [62,63].

The Margin of Safety (MS) is defined as [62,63]:

$$MS(\%) = \left[ \left( \frac{S_y}{\sigma_b} \right) - 1 \right] \times 100 \quad (5.13)$$

#### 5.2.1.1 Assembly condition (T = 23°C)

For the assembly case, a conservative approach is taken by assuming the Maximum Preload condition. Considering a tightening accuracy scatter (typically  $\pm$  values associated with torque wrench application), the bolt load is higher than the nominal target [14,17].

From the analysis data:

- Maximum Bolt Tensile Load ( $F_{b,t}$ ): 39,149 N
- Tensile Stress Area ( $A_t$ ): 61.2 mm<sup>2</sup> [62,63]
- Yield Strength of IN718 at 23°C: 1034 MPa [64]

Calculating the stress:

$$\sigma_{b,assy} = \frac{39149 \text{ N}}{61.2 \text{ mm}^2} = 639.69 \text{ MPa}$$

Calculating the Margin of Safety:

$$MS_{assy} = \left[ \left( \frac{1034}{639.69} \right) - 1 \right] \times 100 = 61.64\%$$

#### 5.2.1.2 Operating condition (T = 700°C)

For the operating case, the load represents the Nominal Preload adjusted for thermal relaxation and external pressure loads as determined in the thermo-mechanical FEA (Time Step 4). The material limit is significantly reduced due to the high temperature [14,17,62,63]. Nominal bolt tensile loads were extracted from tensile area of the bolt and shown in Figure 5.12.

From the analysis data:

- Nominal Bolt Tensile Load ( $F_{b,t}$ ): 26,410 N

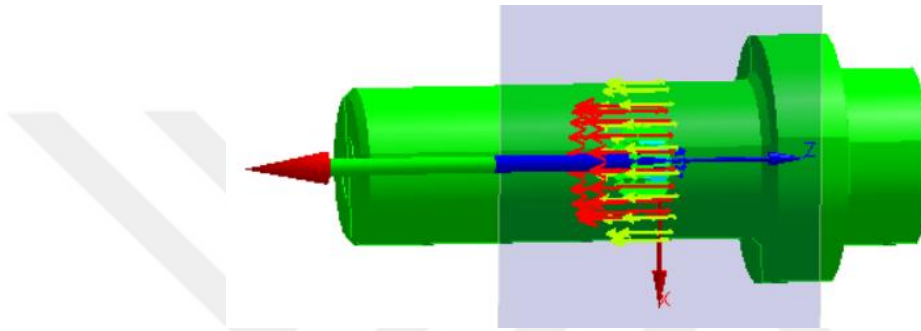
- Tensile Stress Area ( $A_t$ ): 61.2 mm<sup>2</sup> [62,63]
- Yield Strength of IN718 at 700°C: 938 MPa [64]

Calculating the stress:

$$\sigma_{b,op} = \frac{26410 \text{ N}}{61.2 \text{ mm}^2} = 431.54 \text{ MPa}$$

Calculating the Margin of Safety:

$$MS_{op} = \left[ \left( \frac{938}{431.54} \right) - 1 \right] \times 100 = 117.36\%$$



**Figure 5.12 :** Load extraction from the tensile area of the bolt.

### 5.2.2 Bolt maximum fiber stress assessment ( $\sigma_{max}$ )

While the average tensile stress provides a baseline for static strength, it does not account for the bending moments induced by the deformation of the joint. In a flanged assembly, the internal pressure causes the flanges to rotate (prying action), and the differential thermal expansion at 700°C can induce lateral shifts. These mechanisms create a bending moment ( $M_b$ ) in the bolt shank, leading to a non-uniform stress distribution across the cross-section.

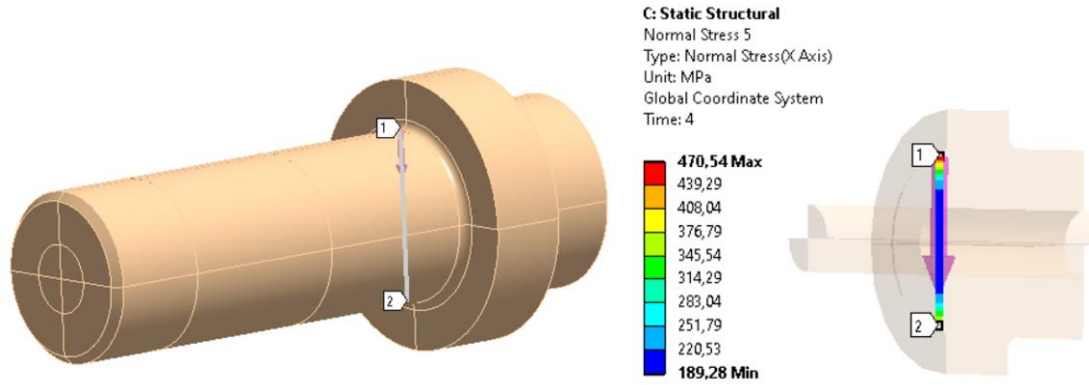
To ensure structural integrity, the Maximum Fiber Stress—defined as the superposition of the axial tensile stress and the maximum bending stress—must be evaluated. This assessment is performed specifically for the Operating Condition (700°C with Nominal Preload) [14,17,62,63].

#### 5.2.2.1 Derivation of bending moment from FEA

Since the FEA model utilizes a simplified cylindrical geometry ( $D_{model} = 10 \text{ mm}$ ) for the bolt thread region, the bending moment must be derived from the linearized stress distribution calculated by the solver.

The Normal Stress ( $\sigma_z$ ) values were extracted from the nodes at the outer fibers of the bolt shank cross-section located immediately below the head-to-shank fillet. Detail of the path was used for the calculation of the normal stress is shown in Figure 5.13.

- Maximum Normal Stress ( $\sigma_{z,max}$ ): 470.54 MPa
- Minimum Normal Stress ( $\sigma_{z,min}$ ): 189.28 MPa



**Figure 5.13 :** Normal stress (MPa) path and plot along the bolt thread root area.

The bending stress component within the FEA model ( $\sigma_{bend,FEA}$ ) is half of the stress differential across the section:

$$\sigma_{bend,FEA} = \frac{\sigma_{z,max} - \sigma_{z,min}}{2} = \frac{470.54 - 189.28}{2} = 140.63 \text{ MPa} \quad (5.14)$$

Using the geometric properties of the modeled 10mm cylinder, the acting Bending Moment ( $M_b$ ) is calculated:

Model Radius ( $c_{model}$ ): 5 mm

Model Moment of Inertia ( $I_{model}$ ):  $\frac{\pi \cdot D^4}{64} = \frac{\pi \cdot 10^4}{64} \approx 490.87 \text{ mm}^4$

$$M_b = \frac{\sigma_{bend,FEA} \cdot I_{model}}{c_{model}} \quad (5.15)$$

$$M_b = \frac{140.63 \text{ MPa} \cdot 490.87 \text{ mm}^4}{5 \text{ mm}} \approx 13,805 \text{ Nmm}$$

### 5.2.2.2 Stress evaluation on critical thread section

Although the moment was derived from the 10mm shank, the structural check must be performed on the weakest section of the fastener: the Thread Root. Therefore, the

calculated moment ( $M_b$ ) is applied to the effective properties of the M10 thread [62,63].

Geometric Properties of M10 Thread Section:

- Effective Radius ( $c_{thread}$ ): 4.188 mm
- Moment of Inertia ( $I_{thread}$ ): 241.6 mm<sup>4</sup>

The corrected bending stress at the thread root ( $\sigma_{bend,thread}$ ) is:

$$\sigma_{bend,thread} = \frac{M_b \cdot c_{thread}}{I_{thread}} = \frac{13805 \text{ Nmm} \cdot 4.188 \text{ mm}}{241.6 \text{ mm}^4} = 239.30 \text{ MPa} \quad (5.16)$$

The Total Maximum Fiber Stress ( $\sigma_{max}$ ) is the sum of the average tensile stress (calculated in Section 4.2.1) and the thread bending stress:

$$\sigma_{max} = \sigma_{tensile} + \sigma_{bend,thread} \quad (5.17)$$

$$\sigma_{max} = 431.54 \text{ MPa} + 239.30 \text{ MPa} = \mathbf{670.84 \text{ MPa}}$$

### 5.2.2.3 Margin of safety calculation

The calculated maximum fiber stress is compared against the yield strength of IN718 at the operating temperature (700°C) [64].

- Criterion ( $S_{y,700^\circ C}$ ): 938 MPa
- Calculated Stress ( $\sigma_{max}$ ): 670.84 MPa

The Margin of Safety is calculated as [62,63]:

$$MS_{fiber}(\%) = \left[ \left( \frac{S_y}{\sigma_{max}} \right) - 1 \right] \times 100 \quad (5.18)$$

$$MS_{fiber} = \left[ \left( \frac{938 \text{ MPa}}{670.84 \text{ MPa}} \right) - 1 \right] \times 100 = 39.83\%$$

### 5.2.2.4 Summary of maximum fiber stress results

The analysis yields a positive Margin of Safety of 39.83% against yielding at the critical maximum fiber location. Admittedly, this figure is significantly lower than the calculated pure tensile margin of +117.36%. However, it remains robust enough to ensure structural integrity.

This discrepancy teaches us a important lesson. It highlights the critical importance of including bending moments in high-temperature flange design. Ignoring the so-called 'prying action' is dangerous. Doing so would have resulted in a non-conservative—and potentially misleading—life estimation.

### 5.2.3 Flange net section bending stress assessment ( $\sigma_{f,bend}$ )

Mechanically, the situation in a bolted connection is dynamic. Internal pressure creates an axial thrust force that attempts to separate the flanges. However, the bolts resist this and effectively act as support points. As a result, this constraint creates a significant bending moment across the flange ring. The limiting factor for structural integrity—the 'weak connection'—is the Net Cross-Section. Specifically, this refers to the bonding area located between the bolt holes. This narrow region represents the minimum cross-sectional area available to counteract the resulting bending forces [62,63].

To assess this critical failure mode, a detailed analytical check was performed for both Flange 1 and Flange 2 under the Operating Condition (700°C with Nominal Preload) [14,17,62,63].

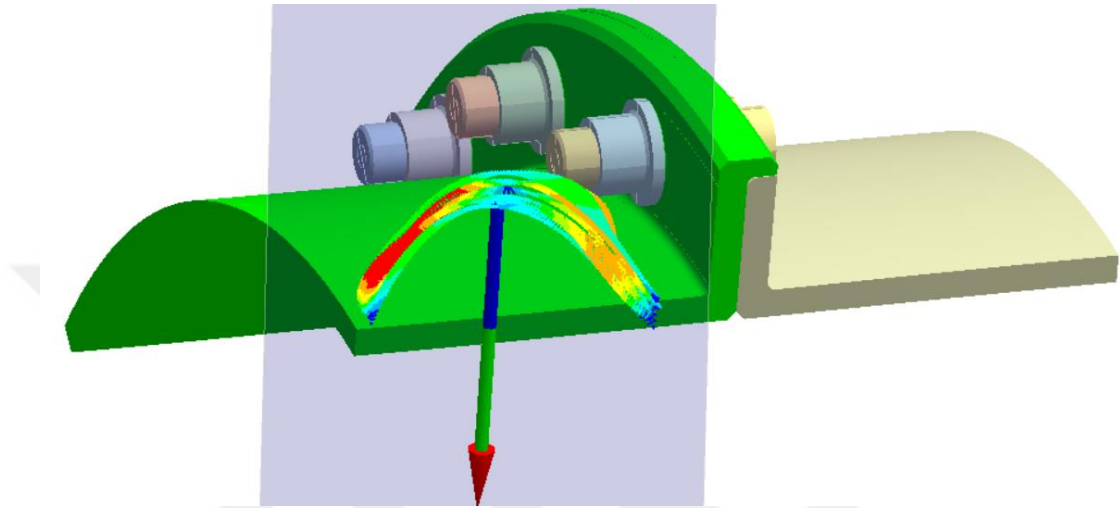
#### 5.2.3.1 Methodology: moment extraction and stress calculation

Unlike simple analytical beam models, the 3D FEA model captures the complex, multi-axial loading of the flange. To utilize this high-fidelity data, the following procedure was employed to determine the effective bending stress [62,63]:

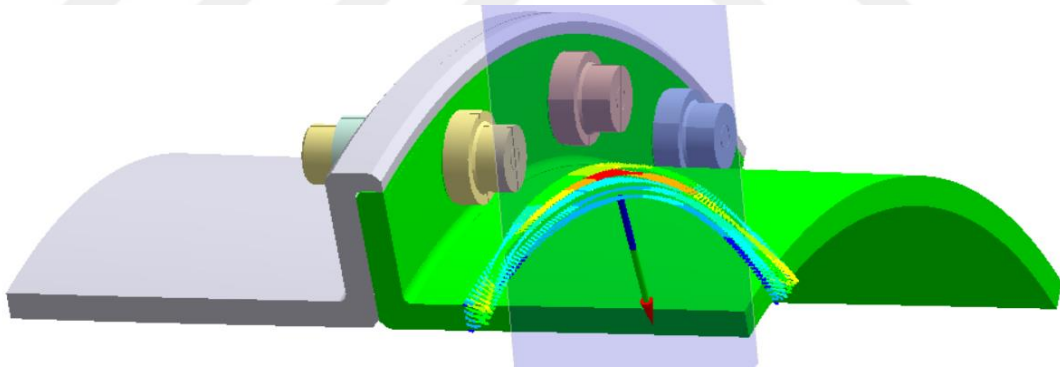
- **Force Reaction Extraction:** A section plane was defined in the FEA model at the transition edge near the flange fillet. The nodal reaction forces ( $F_x, F_y, F_z$ ) acting on this section were extracted [62,63]. These forces were extracted from Flange1 and Flange2 sections and shown in Figure 5.14 and Figure 5.15 respectively.
- **Moment Calculation at Centroid:** These reaction forces were transferred to the geometric centroid of the ligament area. The moments ( $M_x, M_y, M_z$ ) created by these forces were calculated based on their respective orthogonal distances (lever arms) from the centroid.

- Resultant Bending Moment ( $M_f$ ): Since the loading induces bending in multiple planes, the resultant moment vector ( $M_f$ ) was calculated to represent the total bending severity [62,63]:

$$M_f = \sqrt{M_x^2 + M_y^2 + M_z^2} \quad (5.19)$$



**Figure 5.14 :** Force extraction from Flange 1 section.



**Figure 5.15 :** Force extraction from Flange 2 section.

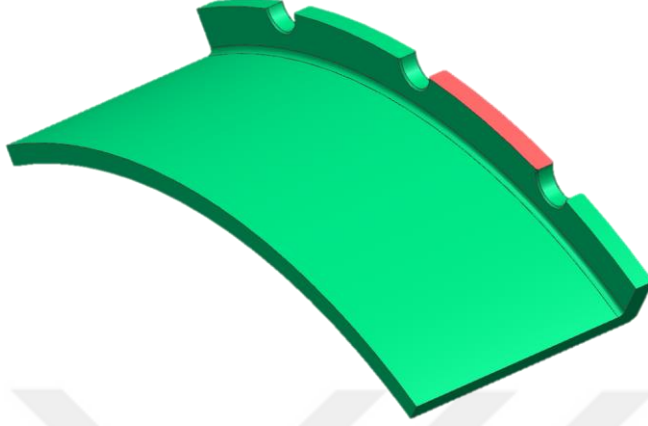
### 5.2.3.2 Geometric properties of the ligament

The bending stress is calculated based on the net cross-sectional properties of the flange ligament:

- Ligament Length ( $L_{lig}$ ): 41.0358 mm
- Flange Thickness ( $t_{flange}$ ): 5 mm
- Area Moment of Inertia ( $I_f$ ):

$$I_f = \frac{L_{lig} \cdot t_{flange}^3}{12} \approx 427.46 \text{ mm}^4 \quad (5.20)$$

Flange net section ligament is shown Figure 5.16 with red dashed region.



**Figure 5.16 :** Net section of flange ligament (dashed with red color).

#### 5.2.3.3 Assessment of flange 1

Flange 1 exhibits the highest loading due to the asymmetric nature of the joint geometry. The extracted moments and resulting stress are as follows [62,63]:

Component Moments:

- $M_x = 10,269.00 \text{ Nmm}$
- $M_y = 66,512.50 \text{ Nmm}$
- $M_z = 532.96 \text{ Nmm}$

Step 1: Calculation of Resultant Moment ( $M_f$ )

$$M_f = \sqrt{(-10269)^2 + (66512.5)^2 + (532.96)^2} = 67,302.67 \text{ Nmm}$$

Step 2: Calculation of Net Section Bending Stress ( $\sigma_{f,bend}$ )

The bending stress is calculated using the flexure formula  $\sigma = \frac{M \cdot c}{I}$ . The distance to the net section fiber ( $c$ ) is taken as the full flange thickness (5 mm). to account for the complex stress state and shear effects in the short ligament. This is a conservative approach [62,63].

Substituting the values:

$$\sigma_{f,bend} = \frac{M_f \cdot c}{I_f} = \frac{67302.67 \text{ Nmm} \cdot 5 \text{ mm}}{427.46 \text{ mm}^4}$$

$$\sigma_{f,bend} = 787.25 \text{ MPa}$$

Step 3: Margin of Safety Calculation

Comparing this stress against the yield strength of IN718 at 700°C ( $S_y = 938 \text{ MPa}$ ) [64]:

$$MS_{f1} = \left[ \left( \frac{938 \text{ MPa}}{787.25 \text{ MPa}} \right) - 1 \right] \times 100 = 19.15\%$$

#### 5.2.3.4 Assessment of flange 2

Flange 2 experiences a different load distribution. Following the same methodology:

Resultant Moment ( $M_f$ ):

$$M_f = \sqrt{(49135.50)^2 + (16432.08)^2 + (274.14)^2} = 51,811.06 \text{ Nmm}$$

Net Section Bending Stress ( $\sigma_{f,bend}$ ):

$$\sigma_{f,bend} = \frac{51811.06 \cdot 5}{427.46} = 606.04 \text{ MPa}$$

Margin of Safety:

$$MS_{f2} = \left[ \left( \frac{938 \text{ MPa}}{606.04 \text{ MPa}} \right) - 1 \right] \times 100 = 54.78\%$$

#### 5.2.3.5 Summary of flange bending results

The structural result is clear. The analysis calculated a safety margin of 19.15% for this criterion. Flange 1 can be identified as the limiting structural component. This is lower than the observed tensile stress margin for the bolt under tension (+117%). Although the flange margin is significantly lower, it remains positive. This confirms that the flange bond can withstand the operating loads without succumbing to plastic collapse. Consequently, this result underlines an important design lesson: geometry—particularly flange thickness and bond width—plays a critical role in the success of high-temperature bond design [62,63].

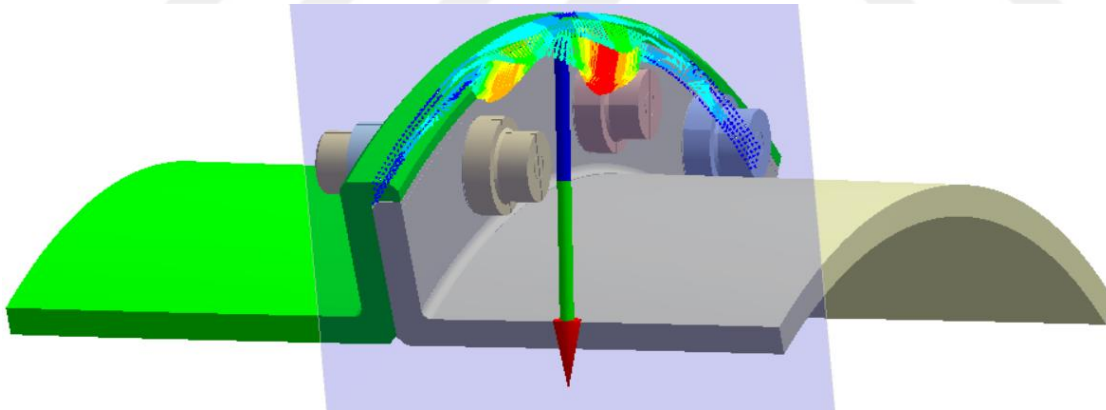
#### 5.2.4 Rabbet net section bending stress assessment

The "rabbet" (or shear lip) is a critical geometric feature designed to facilitate the alignment of the mating flanges and, in some designs, to assist in carrying shear loads. Although the primary axial and bending loads are borne by the flange arm and bolts, the rabbet feature can experience localized bending due to thermal distortion and internal pressure causing relative radial shifts between the casings.

To ensure this feature does not fail under the Operating Condition (700°C, Nominal Preload), a structural check was performed on the critical net section of the rabbet [14,17,62,63].

##### 5.2.4.1 Methodology: moment reaction extraction

Unlike the flange arm calculation, where forces were converted to moments, the moment acting on the rabbet was extracted directly using a Moment Reaction probe in ANSYS. Force extraction from ANSYS model can be seen in Figure 5.17. A specific cross-section was defined at the root of the rabbet geometry (the interface between the rabbet lip and the main flange body). The solver computed the summed nodal moments acting on this cut plane about the global coordinate axes [62,63].



**Figure 5.17** : Force extraction from rabbet section.

##### 5.2.4.2 Geometric properties of the rabbet section

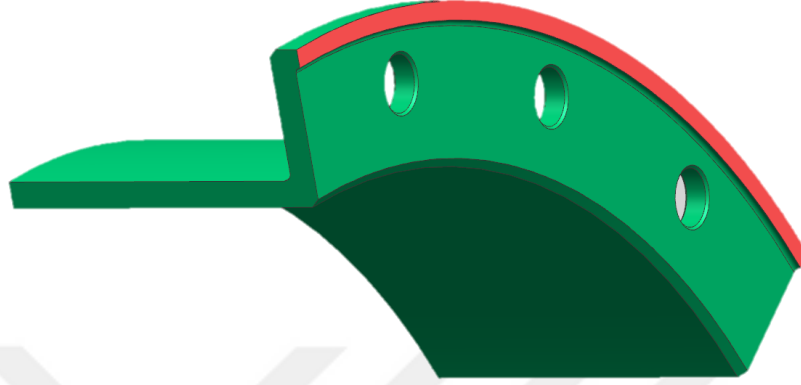
The structural capacity of the rabbet is defined by its rectangular cross-section at the root:

- Engaged Length ( $L_{rb}$ ): 175.5 mm (Arc length of the sector)
- Thickness ( $t_{rb}$ ): 3.5 mm

- Area Moment of Inertia ( $I_{rb}$ ):

$$I_{rb} = \frac{L_{rb} \cdot t_{rb}^3}{12} = \frac{175.5 \cdot (3.5)^3}{12} \approx 627.05 \text{ mm}^4$$

Rabbet net section is shown Figure 5.18 with red dashed region.



**Figure 5.18 :** Net section of rabbet bending area (dashed with red color).

#### 5.2.4.3 Resultant moment and stress calculation

The moment components extracted from the FEA model at the operating step are:

Component Moments:

- $M_x = 19.00 \text{ Nmm}$
- $M_y = 1131.50 \text{ Nmm}$
- $M_z = 5505.50 \text{ Nmm}$

Resultant Moment ( $M_f$ ):

$$M_f = \sqrt{M_x^2 + M_y^2 + M_z^2} \quad (5.19)$$

$$M_f = \sqrt{(19.00)^2 + (1131.50)^2 + (5505.50)^2} = 5,620.60 \text{ Nmm}$$

Bending Stress Calculation:

The bending stress is calculated using the standard flexure formula. Consistent with the conservative methodology applied to the flange analysis, the distance to the extreme fiber ( $c$ ) is taken as the full thickness ( $t_{rb} = 3.5 \text{ mm}$ ) to account for potential stress concentrations and shear effects at the root fillet [62,63].

$$\sigma_{rb,bend} = \frac{M_f \cdot c}{I_{rb}} = \frac{5620.60 \text{ Nmm} \cdot 3.5 \text{ mm}}{627.05 \text{ mm}^4}$$

$$\sigma_{rb,bend} = 31.37 \text{ MPa}$$

#### 5.2.4.4 Margin of safety calculation

The calculated stress is compared against the yield strength of IN718 at 700°C [64].

- Criterion ( $S_y$ ): 938 MPa

Margin of Safety:

$$MS_{rb} = \left[ \left( \frac{938 \text{ MPa}}{31.37 \text{ MPa}} \right) - 1 \right] \times 100 = 2,889.87\%$$

#### 5.2.4.5 Summary of rabbet structural results

The rabbet feature exhibits an exceptionally high Margin of Safety (+2889%). This figure confirms that the component is structurally robust. It is certainly not a limiting factor in the overall joint design. The observed low stress levels indicate a clear load path. The heavy lifting—specifically the primary bending loads—is effectively managed by the main flange body and the bolted connection. Consequently, the rabbet is free to serve its primary function of alignment without sustaining significant structural deformation.

#### 5.2.5 Bolt/flange crush stress assessment ( $\sigma_{br}$ )

In high-performance bolted joints, bearing stress is a critical parameter. This criterion is caused from the compressive force generated at the interface between the fastener and the flange face. If the contact pressure becomes excessive, it leads to localized yielding of the flange material. The consequences are severe. Embedding permanently shortens the grip length, leading to a direct and often catastrophic loss of bolt preload [14,17,62,63].

To prevent this specific failure mode, a Crush Stress (or Bearing Stress) assessment was performed. The analysis focuses strictly on the interface under the Bolt Head. The reason is geometric. The effective contact area under the bolt head is narrower than the area under the flanged nut. Since force is constant, this smaller area results in significantly higher localized pressures, making it the critical location for analysis [62,63].

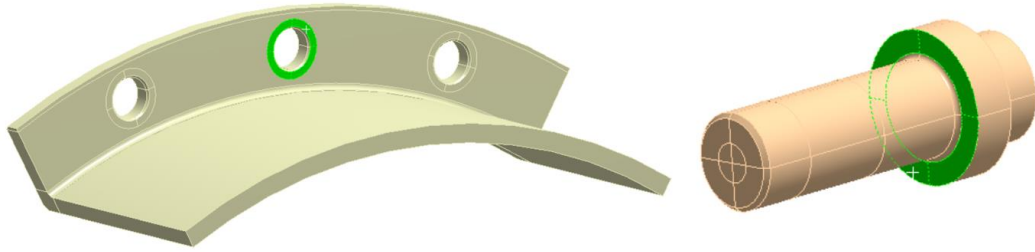
#### 5.2.5.1 Methodology: contact force extraction

The compressive force acting on the flange face was extracted directly from the FEA model to ensure accuracy [62,63].

Probe Definition: A "Force Reaction" probe was used specifically to the Contact Surface between the underside of the bolt head and the mating flange face [62,63]. These crush forces were extracted from crush stress areas on the bolt and the Flange2. These areas are shown in Figure 5.19 and dashed with green.

Loading Conditions:

- Assembly Condition (24°C): Evaluated at the Maximum Preload to represent the worst-case crushing force during tightening [14,17].
- Operating Condition (700°C): Evaluated at the Nominal Preload (post-relaxation) to assess the margin against the reduced yield strength of the material at high temperature [14,17].



**Figure 5.19** : Crush stress areas on the bolt and the Flange 2 (dashed with green).

#### 5.2.5.2 Calculation of bearing area ( $A_{br}$ )

The effective bearing area is defined by the annular region between the bolt head diameter ( $D_w$ ) and the hole chamfer diameter ( $D_h$ ). Using the geometric parameters from the design:

Bolt Head Diameter ( $D_w$ ): 15.70 mm

Hole/Chamfer Diameter ( $D_h$ ): 11.80 mm

$$A_{br} = \frac{\pi}{4} \cdot (D_w^2 - D_h^2) \quad (5.21)$$

$$A_{br} = \frac{\pi}{4} \cdot (15.70^2 - 11.80^2) \approx 84.23 \text{ mm}^2$$

### 5.2.5.3 Assembly condition assessment

Under the maximum assembly preload, the reaction force extracted from the FEA model is 39,149 N [14,17].

Crush Stress Calculation:

$$\sigma_{br,assy} = \frac{F_{b,max}}{A_{br}} = \frac{39149 \text{ N}}{84.23 \text{ mm}^2} = 464.77 \text{ MPa}$$

Margin of Safety:

Compared against the yield strength of IN718 at Room Temperature ( $S_y = 1034 \text{ MPa}$ ) [64]:

$$MS_{br,assy} = \left[ \left( \frac{1034 \text{ MPa}}{464.77 \text{ MPa}} \right) - 1 \right] \times 100 = 122.48\%$$

### 5.2.5.4 Operating condition assessment

Under operating conditions ( $700^\circ\text{C}$ ), the reaction force decreases to 26,410 N due to thermal relaxation.

Crush Stress Calculation:

$$\sigma_{br,op} = \frac{F_{b,nom}}{A_{br}} = \frac{26410 \text{ N}}{84.23 \text{ mm}^2} = 313.53 \text{ MPa}$$

Margin of Safety:

Compared against the yield strength of IN718 at  $700^\circ\text{C}$  ( $S_y = 938 \text{ MPa}$ ) [64]:

$$MS_{br,op} = \left[ \left( \frac{938 \text{ MPa}}{313.53 \text{ MPa}} \right) - 1 \right] \times 100 = 199.17\%$$

### 5.2.5.5 Summary of crush stress results

The analysis yields substantial safety margins for both load cases ( $> 120\%$ ). This confirms that the compressive stresses under the bolt head are quite below the yield limit of the flange material. Consequently, the structural integrity of the contact interface is preserved. The risk of 'embedding'—and the subsequent secondary preload loss associated with it—is rendered negligible for this design configuration [14,17].

### 5.2.6 Rabbet crush stress assessment ( $\sigma_{r,br}$ )

The rabbet acts as the primary radial locating feature for the flanges. Its mission is to ensure strict concentricity between the mating flanges. Bolts handle the axial clamping and the rabbet interface often bears the brunt of radial loads transmitted through the joint.

However, at elevated temperatures, the physics gets complicated. Differential radial expansion between the inner and outer flanges can induce significant contact pressures at this interface—a phenomenon known as 'thermal interference'. To ensure that these pressures do not cause local plastic deformation, a Crush Stress assessment was performed. This is vital because any yielding here would permanently degrade the alignment accuracy of the engine casing [62,63].

#### 5.2.6.1 Methodology: radial contact force extraction

The contact pressure is derived from the total radial reaction force acting on the rabbet interface.

FEA Probe: A "Force Reaction" probe was defined on the specific contact face of the rabbet. The probe sums the radial force components acting on the 60° sector model [62,63]. These forces were extracted from rabbet area shown in Figure 5.20.

Load Cases:

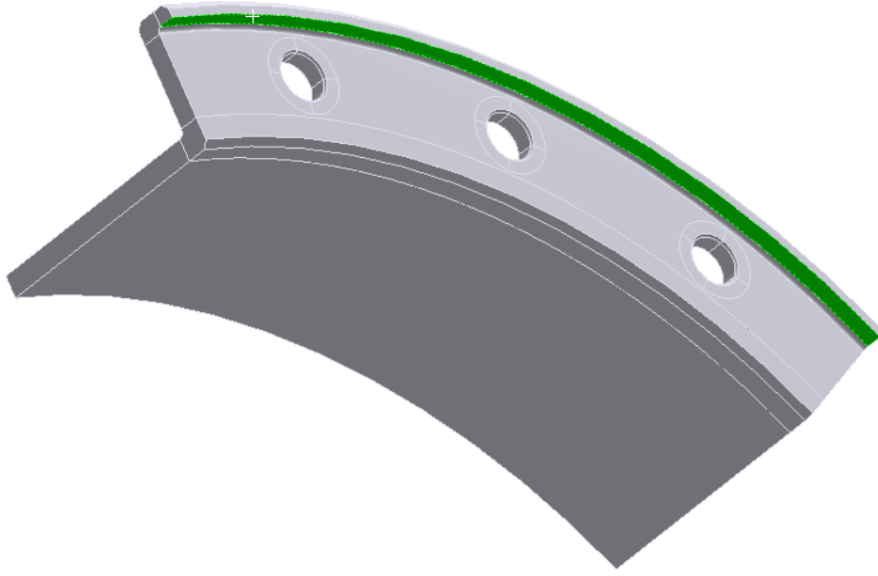
- Assembly Condition: Assessed under Maximum Preload to capture the initial contact pressure due to tightening mechanics [14,17].
- Operating Condition: Assessed at 700°C. It is observed that the radial load increases in this state, indicating that the thermal expansion creates a tighter fit (increased interference) at the rabbet interface.

#### 5.2.6.2 Geometric properties and contact area

The bearing stress is calculated over the effective contact area of the modeled sector:

- Rabbet Diameter ( $D_r$ ): 351.00 mm
- Contact Length ( $l_r$ ): 3.85 mm
- Sector Angle: 60° (or 1/6 of the ring)

Rabbet crush area on the Flange1 is shown in Figure 5.20 and dashed with green.



**Figure 5.20 :** Rabbet crush area on the Flange 1 (dashed with green).

The effective contact area for the 60° sector ( $A_{r,sec}$ ) is:

$$A_{r,sec} = \frac{\pi \cdot D_r \cdot l_r}{6} = \frac{\pi \cdot 351 \cdot 3.85}{6} \approx 707.49 \text{ mm}^2 \quad (5.22)$$

### 5.2.6.3 Assembly condition assessment

Under the maximum assembly preload, the radial contact force is 4,696.60 N [14,17].

Crush Stress Calculation:

$$\sigma_{r,br,assy} = \frac{F_{r,assy}}{A_{r,sec}} = \frac{4696.60 \text{ N}}{707.49 \text{ mm}^2} = 6.64 \text{ MPa}$$

Margin of Safety:

Compared against the yield strength of IN718 at Room Temperature ( $S_y = 1034 \text{ MPa}$ ) [64]:

$$MS_{assy} = \left[ \left( \frac{1034 \text{ MPa}}{6.64 \text{ MPa}} \right) - 1 \right] \times 100 = 15,478\%$$

### 5.2.6.4 Operating condition assessment

Under operating conditions (700°C), the radial force increases to 5,792.70 N due to thermal expansion effects.

Crush Stress Calculation:

$$\sigma_{r,br,op} = \frac{F_{r,op}}{A_{r,sec}} = \frac{5792.70 \text{ N}}{707.49 \text{ mm}^2} = 8.19 \text{ MPa}$$

Margin of Safety:

Compared against the yield strength of IN718 at 700°C ( $S_y = 938 \text{ MPa}$ ) [64]:

$$MS_{op} = \left[ \left( \frac{938 \text{ MPa}}{8.19 \text{ MPa}} \right) - 1 \right] \times 100 = 11,357\%$$

#### 5.2.6.5 Summary of rabbet crush stress results

The rabbet feature provides high margins of safety against compressive yielding (>11,000%). The stress levels (<10 MPa) are negligible compared to the material limits. This analysis confirms that the rabbet will maintain its dimensional stability and alignment function during the operational envelope. Rabbet is working with no risk of plastic deformation even under the increased radial loads induced by thermal expansion.

#### 5.2.7 Transverse load capability (tlc) margin assessment

In high-performance bolted connections, the bolt bodies are not intended to directly carry shear loads. Instead, the primary mechanism for transmitting transverse loads is friction at the flange interfaces (joining surfaces). Ideally, the connection remains locked. However, if external shear loads exceed this friction capacity, the connection begins to slip. Then, the flange holes slide and press against the bolt bodies. Subsequently, the bolts are subjected to direct shear stress, which is an undesirable outcome in bolted connection design. This slip condition indicates a critical failure in the design. This study investigates the aforementioned slip condition.

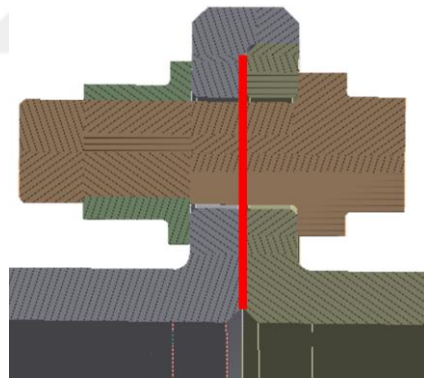
To ensure the joint maintains its non-slip integrity, the Transverse Load Capability (TLC) was assessed [62,63].

##### 5.2.7.1 Methodology: worst-case friction assessment

The assessment is performed on a per-bolt basis, scaling the global sector forces to represent the load carried by a single effective fastener pitch (an 18° segment of the full 360° ring) under the most conservative conditions for slip:

- Loading Condition: Operating Condition (700°C) is selected, as thermal relaxation reduces the clamping force [62,63].

- **Preload Assumption:** The Minimum Preload is utilized. Unlike the strength checks which assumed nominal or maximum loads, the slip check assumes the lower bound of the preload scatter (due to torque inaccuracies and relaxation) to determine the minimum available friction force [14,17].
- **Bolt Loss Scenario:** The assessment assumes that 2 bolts are lost (failed or completely loosened) out of the total 20-bolt assembly. This requires the remaining 18 bolts to maintain the seal, significantly reducing the available total clamping force.
- **Force Extraction:** The contact forces were extracted directly from the FEA model using probes on the flange-to-flange interface. This interface load extraction surface is shown in Figure 5.21.
  - **Axial Contact Load ( $F_{CL}$ ):** The total compressive normal force remaining between the flanges.
  - **Shear Loads ( $L_r, L_t$ ):** The reaction forces in the Radial and Transverse (Tangential) directions.



**Figure 5.21 :** Interface loads extraction surface.

#### 5.2.7.2 Resultant shear force calculation

The joint is subjected to simultaneous shear loads in orthogonal directions. The total external shear force acting to slip the joint ( $F_{shear,total}$ ) is the vector resultant of the radial and transverse components extracted from the FEA:

- Radial Shear Load ( $L_r$ ): 83.28 N
- Transverse Shear Load ( $L_t$ ): 1,032.99 N

$$F_{shear,total} = \sqrt{L_r^2 + L_t^2} \quad (5.23)$$

$$F_{shear,total} = \sqrt{(-83.28)^2 + (-1032.99)^2} = 1,036.34 \text{ N}$$

#### 5.2.7.3 Frictional capacity calculation

The resistance to slip is governed by the Coulomb friction law. A conservative friction coefficient ( $\mu$ ) of 0.12 was selected for this margin check to account for potential surface smoothing or lubrication effects at high temperatures, providing a lower bound than the 0.15 used in the structural model [62,63,66].

- Axial Contact Load ( $F_{CL}$ ): 17,416.89 N
- Friction Coefficient ( $\mu$ ): 0.12

$$F_{friction, capacity} = \mu \cdot F_{CL} = 0.12 \cdot 17416.89 \text{ N} = 2,090.03 \text{ N} \quad (5.24)$$

#### 5.2.7.4 TLC margin calculation

The Transverse Load Capability Margin quantifies the excess friction capacity relative to the applied shear.

$$TLC(\%) = \left( \frac{F_{friction, capacity}}{F_{shear, total}} - 1 \right) \cdot 100 \quad (5.25)$$

$$TLC(\%) = \left( \frac{2090.03 \text{ N}}{1036.34 \text{ N}} - 1 \right) \cdot 100 = 101.67\%$$

#### 5.2.7.5 Summary of transverse load capability

The analysis yields a positive margin of 101.67%, indicating that the available frictional resistance is more than double the applied shear load, even under the worst-case assumption of minimum preload and a conservative friction coefficient. This confirms that the joint will operate as a true friction-grip connection, with no relative slip between the flanges, thereby protecting the bolts from direct shear stresses and preventing fretting damage [14,17,62,63].

### 5.2.8 Joint separation margin assessment

In gas turbine housings, especially those containing high-pressure fluids, a fundamental requirement for bolted flange connections is to prevent connection separation. Separation occurs when the external tensile loads acting on the connection exceed the compression capacity provided by the bolts. This leads to a loss of contact (clearance) between the flange faces, which can cause leakage and rapidly increase bolt fatigue [62,63].

To rigorously verify the design integrity, a Separation Margin was calculated under a "Worst-Case" scenario. This scenario combines three conservative assumptions [62,63]:

- Operating Condition (700°C): Accounting for thermal relaxation of the preload [14,17].
- Minimum Preload: Assuming the lower bound of assembly torque scatter [14,17].
- Bolt Loss Scenario: The assessment assumes that 2 bolts are lost (failed or completely loosened) out of the total 20-bolt assembly. This requires the remaining 18 bolts to maintain the seal, significantly reducing the available total clamping force.

#### 5.2.8.1 Methodology: calculation of critical and external loads

The assessment is performed on a per-bolt basis, scaling the global sector forces to represent the load carried by a single effective fastener pitch (an 18° segment of the full 360° ring)

#### 5.2.8.2 Critical separation load ( $P_0$ ):

The critical load is defined as the external force required to reduce the interface contact pressure to zero. It is calculated based on the residual clamping force acting on the joint during operation [62,63].

Using the classic joint diagram relationship:

$$P_0 = \frac{F_{cl}}{1 - C} \quad (5.26)$$

Where:

$F_{cl}$  is the Axial Contact Load per bolt pitch. Extracting the total contact force from the model (64,507 N for the 60° sector) and scaling it to the full ring, then dividing by the remaining effective bolts, yields a conservative local contact load of 17,417 N per bolt location for this check.

$C$  is the Joint Stiffness Constant ( $\phi$ ), calculated analytically as 0.2512 [62,63].

$$P_0 = \frac{17417 \text{ N}}{1 - 0.2512} = 23,260 \text{ N}$$

### 5.2.8.3 External separation load ( $F_{ext}$ ):

The net external force attempting to separate the flanges was derived by analyzing the force balance on the flange geometry [62,63].

Force Components: The resultant forces acting on the flange interface were extracted in global Cartesian coordinates. These values represent the net separating force per bolt pitch. These net separating loads were calculated using the loads extracted from Flange1 and Flange2. Flange1 force extraction details are shown in Figure 5.22 and Flange2 force extraction details are shown in Figure 5.23

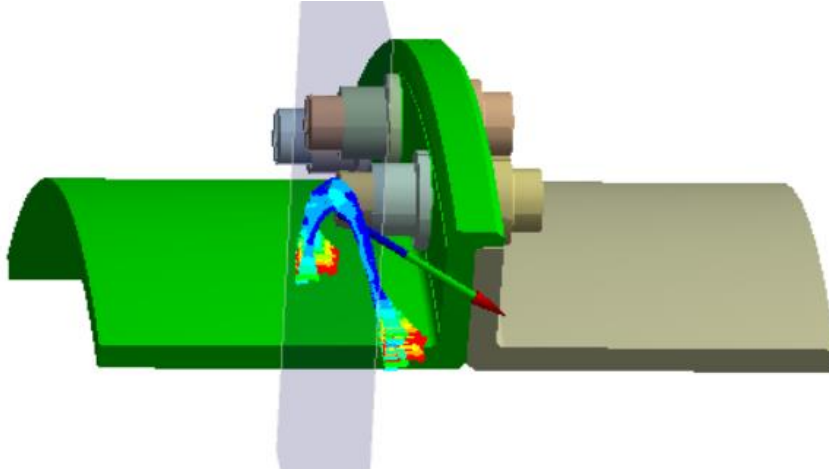
- Axial Component ( $F_x$ ): 19.13 N
- Tangential Component ( $F_y$ ): 498.54 N
- Radial Component ( $F_z$ ): 1,513.50 N

Resultant External Load ( $F_{ext}$ ):

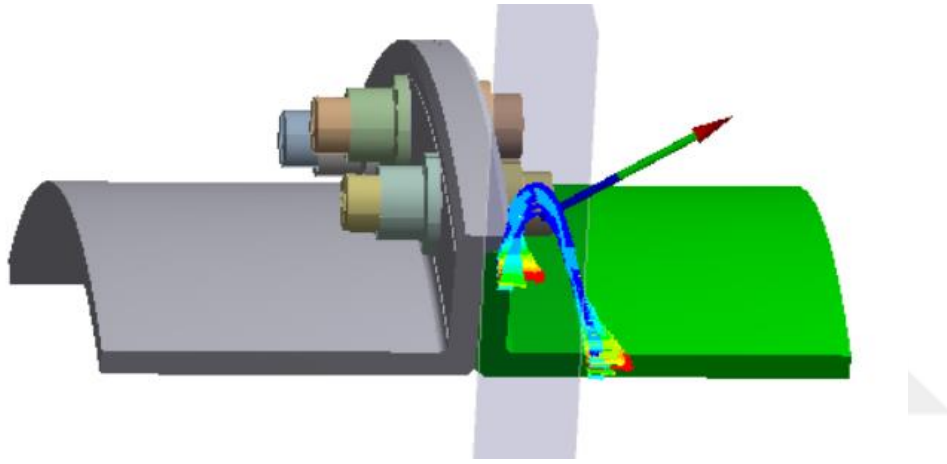
$$F_{ext} = \sqrt{F_x^2 + F_y^2 + F_z^2} \quad (5.27)$$

$$F_{ext} = \sqrt{(19.13)^2 + (498.54)^2 + (1513.50)^2} = 1,594 \text{ N}$$

The radially outward force dominates the resultant, while the axial and tangential contributions are small. The combined separating load per bolt therefore stays well below both the critical separation load and the residual clamp force, which is why the separation margin reported next is so large.



**Figure 5.22 :** Separation load extraction from Flange 1.



**Figure 5.23 :** Separation load extraction from Flange 2.

#### 5.2.8.4 Margin of safety calculation

The Separation Margin is calculated by comparing the critical load capacity ( $P_0$ ) to the applied external load ( $F_{ext}$ ).

$$MS_{sep} = \left[ \left( \frac{P_0}{F_{ext}} \right) - 1 \right] \times 100 \quad (5.28)$$

$$MS_{sep} = \left[ \left( \frac{23260 \text{ N}}{1594 \text{ N}} \right) - 1 \right] \times 100 = 1,359.56\%$$

#### 5.2.8.5 Summary of separation results

Even under the extreme "Bolt Loss" scenario with minimum preload, the joint exhibits a high safety margin of 1,359.56% against separation. This result confirms that the external separating forces created by pressure and thermal loads are negligible

compared to the high clamping force of the joint. The design is exceptionally robust against leakage and separation-induced fatigue [14,17].

### 5.2.9 Bolt fatigue life assessment

The final and another critical step in the structural verification is the prediction of the bolt's fatigue life. Given the high operating temperature ( $700^{\circ}\text{C}$ ) and the presence of cyclic thermal loads (Engine Start-Up  $\leftrightarrow$  Steady State Operation), the bolt is sensitive to fatigue failure.

The FEA model utilizes a simplified cylindrical geometry for the bolt shank to optimize mesh quality. Because of this simplification, stress concentration effects of the thread roots are not explicitly captured in the global stress results. To avoid these misleading stress results for fatigue life calculation, a local strain-life approach is employed analytically, applying a Stress Concentration Factor ( $K_t$ ) to the nominal stresses extracted from the FEA [67].

#### 5.2.9.1 Methodology: analytical strain-life approach

The assessment follows a "Shakedown" analysis logic suitable for components that experience local yielding during the first cycle. But stabilize into elastic behavior for subsequent cycles. The **Smith-Watson-Topper (SWT)** damage parameter is used to account for the significant tensile mean stress present in the preloaded bolt [14,17,68].

#### 5.2.9.2 Definition of loading and material parameters

The analysis inputs are derived from the validated FEA model (Von-Mises Equivalent Stresses) and the material literature for Inconel 718 [64].

FEA Nominal Stresses (Bolt Shank):

- $\sigma_{nominal,max} = 709 \text{ MPa}$  (Nominal stress at assembly)
- $\sigma_{nominal,min} = 630 \text{ MPa}$  (Nominal stress during operation)

Note: The stress decreases during operation due to thermal relaxation, defining the cyclic range.

Design Parameters:

- Stress Concentration Factor ( $K_t$ ): 2.5 (Standard assumption for rolled threads) [67].

IN718 Material Properties (700°C) [64]:

- Yield Strength ( $\sigma_y$ ): 938 MPa
- Fatigue Strength Coefficient ( $\sigma_f$ ): 1476 MPa [60]
- Fatigue Strength Exponent ( $b$ ): -0.0840

### 5.2.9.3 Theoretical local stress calculation

First, the local stresses at the thread root are calculated assuming purely elastic material behavior to determine if yielding occurs.

Theoretical Max Stress:

$$\sigma_{max,elastic} = \sigma_{nom,max} \cdot K_t = 709 \text{ MPa} \cdot 2.5 = 1772.5 \text{ MPa} \quad (5.29)$$

Theoretical Min Stress:

$$\sigma_{min,elastic} = \sigma_{nom,min} \cdot K_t = 630 \text{ MPa} \cdot 2.5 = 1575 \text{ MPa} \quad (5.30)$$

### 5.2.9.4 Plasticity and shakedown check

The theoretical maximum stress (1772.5 MPa) significantly exceeds the material yield strength (938 MPa).

Conclusion: The material cannot support this elastic stress. Localized plastic deformation will occur at the thread root during the initial assembly tightening. However, the bolt will "shakedown" (stabilize) if the cyclic stress range is within the elastic limit [14,17].

### 5.2.9.5 Determination of the stabilized stress cycle

The stress range ( $\Delta\sigma$ ) remains constant and is driven by the external load fluctuation.

Local Stress Range:

$$\Delta\sigma_{local} = (\sigma_{nom,max} - \sigma_{nom,min}) \cdot K_t \quad (5.31)$$

$$\Delta\sigma_{local} = (709 - 630) \cdot 2.5 = 79 \cdot 2.5 = 197.5 \text{ MPa}$$

Since the range (197.5 MPa) is significantly less than the yield strength (938 MPa), the subsequent cycles will be fully elastic. The stress cycle effectively "shifts" downwards due to the initial plasticity.

Stabilized Maximum Stress ( $\sigma_{max,stab}$ ): Capped at the yield limit.

$$\sigma_{max,stab} = \sigma_y = 938 \text{ MPa}$$

Stabilized Minimum Stress ( $\sigma_{min,stab}$ ):

$$\sigma_{min,stab} = \sigma_{max,stab} - \Delta\sigma_{local} \quad (5.32)$$

$$\sigma_{min,stab} = 938 - 197.5 = 740.5 \text{ MPa}$$

#### 5.2.9.6 Calculation of fatigue parameters

For the life prediction, the amplitude and mean stress of this new stabilized cycle are required.

Stress Amplitude ( $\sigma_a$ ):

$$\sigma_a = \frac{938 - 740.5}{2} = 98.75 \text{ MPa}$$

Mean Stress ( $\sigma_m$ ):

$$\sigma_m = \frac{938 + 740.5}{2} = 839.25 \text{ MPa}$$

#### 5.2.9.7 Life prediction (SWT method)

The Smith-Watson-Topper (SWT) equation is employed to predict life ( $N_f$ ) under high mean stress conditions [68]:

$$\sigma_{max,stab} \cdot \sigma_a = (\sigma'_f)^2 \cdot (2N_f)^{2b} \quad (5.33)$$

Substituting the values:

$$938 \cdot 98.75 = (1476)^2 \cdot (2N_f)^{2(-0.0840)}$$

Calculating Damage Parameter (Left Side):

$$92,627.5$$

Calculating Material Resistance (Right Side):

$$2,178,576 \cdot (2N_f)^{-0.168}$$

Solving for  $2N_f$  (Reversals to Failure):

$$(2N_f)^{-0.168} = \frac{92,627.5}{2,178,576} = 0.042517$$

$$2N_f = (0.042517)^{\frac{1}{-0.168}} = (0.042517)^{-5.952}$$

$$2N_f \approx 146,854,825 \text{ Reversals}$$

Cycles to Failure ( $N_f$ ):

$$N_f = \frac{2N_f}{2} \approx 73,427,412 \text{ Cycles}$$

### 5.2.9.8 Conclusion of fatigue analysis

The calculated fatigue life is approximately 73.4 Million Cycles.

This result demonstrates that although the bolt experiences local yielding during the first assembly (shakedown), the subsequent operational stress range ( $\Delta\sigma_{local} = 197.5 \text{ MPa}$ ) is small enough that it corresponds to a very high fatigue life for the Inconel 718 material, well beyond the required service cycles. The high mean stress is sustainable because the alternating stress amplitude is low. The design is therefore considered safe for High Cycle Fatigue (HCF) and exceeds typical gas turbine engine life requirements (e.g., 20,000 - 50,000 cycles) by a substantial margin [62-64].

## 5.3 Results Summary

Detailed summary of the structural integrity margin and life calculations can be seen in Table 5.2.

As Table 5.2 shows, every limit state returns a positive margin of safety. The governing case is the net-section bending of Flange 1 at 19.15 %, making flange bending the limiting mode rather than any bolt failure, while the remaining checks and the 73.4-million-cycle fatigue life retain large reserves.

**Table 5.2 :** Summary of structural integrity margin and life calculations.

| <b>Criterion<br/>(Margin)</b>                | <b>What it controls</b>                         | <b>Condition / Preload<br/>Case</b> | <b>Margin (%)</b> |
|----------------------------------------------|-------------------------------------------------|-------------------------------------|-------------------|
| Bolt tensile stress margin                   | Gross axial yielding of bolt shank              | Assembly / Max Preload              | 61.64             |
| Bolt tensile stress margin                   | Gross axial yielding of bolt shank              | Operating / Nom Preload             | 117.36            |
| Bolt max fiber stress margin                 | Combined tension + bending in bolt              | Operating / Nom Preload             | 39.83             |
| Flange net-section bending margin (Flange 1) | Flange yielding in bending at critical section  | Operating / Nom Preload             | 19.15             |
| Flange net-section bending margin (Flange 2) | Flange yielding in bending at critical section  | Operating / Nom Preload             | 54.78             |
| Bearing / crush margin                       | Local compressive yielding at bearing/contact   | Assembly / Max Preload              | 122.48            |
| Bearing / crush margin                       | Local compressive yielding at bearing/contact   | Operating / Nom Preload             | 199.17            |
| Rabbet net-section bending margin            | Rabbet section yielding in bending              | Operating / Nom Preload             | 2889.87           |
| TLC margin                                   | Slip resistance (friction vs transverse demand) | Operating / Min Preload             | 101.67            |
| Separation margin                            | Interface opening resistance                    | Operating / Min Preload             | 1359.56           |
| Bolt Fatigue Life                            | Bolt Fatigue Life                               | Operating / Nom Preload             | 73.4M Cycles      |



## 6. CONCLUSION AND FUTURE WORKS

### 6.1 Conclusions

The research revealed the following specific results regarding the performance of the connection.

It was found that one of the dominant causes of stress generation on the parts is thermal effects. The physics of differential thermal expansion between elements, combined with modulus distortion at high temperatures (700 °C), resulted in significant preload relaxation.

The Finite Element Analysis (FEA) model predicted a loss of 18.65% in compressive strength due to thermal effects alone.

Importantly, this result closely matches analytical hand calculations which predicted a loss of 20.38%. The difference between the preload loss obtained from the hand calculation and the FEA model is less than 2%. The reliability of the numerical model is confirmed by this relatively small difference. Furthermore, this highlights the importance of considering thermal relaxation during initial torque determination, a crucial design parameter for bolted connections.

Safety Margin (MS) assessments allowed several specific bottlenecks in the design to be identified.

With respect to bolt strength, the results show that the bolt will maintain its structural integrity. However, a significant change was also observed. The pure tensile margin is high (117.36%). Subsequent calculations included bending moments, reducing this margin to 39.83% at the maximum fiber position. This highlights that ignoring flange rotation in simplified 1D analyses can lead to unconservative results.

Regarding flange stiffness, Flange 1 was identified as the limiting structural component. The net cross-section shows the lowest safety margin against bending yield strength (+19.15%). Consequently, future weight optimization studies related to flange thickness should be conducted with extreme caution.

In contrast, the rabbet and bearing surfaces showed excessive margins (>1000%). This indicates that they are not failure-critical and are the most suitable candidates for weight reduction.

The connection also demonstrates quite adequate stability against leakage and relative movement.

In terms of separation, even in the worst-case scenario (assuming loss of 2 bolts and minimum preloading), a significant separation margin of +1359% was calculated for the connection.

With respect to slip, the Transverse Load Capacity (TLC) margin is +101.67%. This high TLC margin calculation indicates that there will be no slip between the flanges. This shows that the friction forces are sufficient. The sufficient friction force indicates that slip, which would directly expose the bolts to shear, will not occur.

The Low Cycle Fatigue (LCF) assessment shows that the bolt operates with a very large fatigue-life margin. Although localized yielding ('shakedown') occurs at the thread roots during the first assembly cycle, the subsequent operating stress range ( $\Delta\sigma_{nom} = 79$  MPa) is sufficiently low.

The Smith-Watson-Topper (SWT) method predicted a lifespan of approximately 73.4 million cycles.

This result confirms that the design is safe against fatigue failure, provided that the initial preloading prevents link separation, which would otherwise significantly increase the cyclic stress amplitude.

## **6.2 Recommendations for Future Work**

This study has established a solid foundation for characterizing the behavior, strength, and service life of bolted connections. In addition, specific areas require more detailed investigation. To improve prediction accuracy and achieve a more optimal design, several steps have been identified.

The analyses completed in this study addressed high-temperature effects (primarily the change in Elastic Modulus at 700°C). However, Inconel 718 is not immune to time-dependent physics. At long service intervals (e.g., >10,000 hours), creep relaxation becomes a decisive factor. Future research requires the application of a Creep-Stress

material model open to finite element analysis (FEA). This step is crucial for simulating the gradual loss of preload throughout the entire engine overhaul cycle, rather than simply capturing instantaneous thermal equilibrium.

Numerical models, regardless of their accuracy, require an experimental basis. A targeted physical testing campaign is proposed to validate the Finite Element Analysis (FEA) findings. Such a campaign would equip a prototype test flange with specialized high-temperature strain gauges, apply controlled heating cycles up to 700 °C to measure flange bending and bolt stress relaxation in real time, and perform tests for separation, slippage, and leakage under the assumption that some of the flanges considered in this study are lost.

The scope of this thesis was limited to static thermomechanical loads. In reality, gas turbine engines act as significant sources of dynamic excitation. Consequently, a modal analysis is necessary. The aim is to verify that the natural frequencies of the assembly do not match the excitation sequences of the engine. Such resonance can trigger High Cycle Fatigue (HCF) mechanisms, which are different from the thermal cycles analyzed here.

Structural evaluation revealed excessive safety margins ( $>1000\%$ ) in the groove and bearing areas. These findings indicate that the component is overdesigned in some respects. A topological optimization study can be conducted in this regard. By selectively reducing the groove thickness or flange outer diameter, significant weight savings, a priority in aerospace engineering, can be achieved. This can be done without compromising the structural integrity of the limiting inner flange.



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## **CURRICULUM VITAE**

**Name Surname : Tahsin Topal**

### **EDUCATION:**

- **B.Sc.** : 2019, Istanbul Technical University, Mechanical Engineering

### **PROFESSIONAL EXPERIENCE AND REWARDS:**

- 2021-Present TEI TUSAS Engine Industries, Mechanical Design Engineer.
- 2019-2021 GE Aerospace, Mechanical Design Engineer.

### **PUBLICATIONS, PRESENTATIONS AND PATENTS ON THE THESIS:**

- **Topal T.**, Candaş A., İmrak C. E., 2025: Investigation of the mechanical behaviour of superalloy bolted joints under extreme thermal conditions for aerospace applications. 9th International Conference on Engineering Technologies (ICENTE'25), 20-22 November 2025, Konya, Türkiye.