

T.R.
GEBZE TECHNICAL UNIVERSITY
GRADUATE SCHOOL

**HIGH CYCLE FATIGUE LIFE ASSESSMENT OF GAS
TURBINE ENGINE ANNULAR COMBUSTOR**

AHMET ONUR ŞAHİN

**A THESIS OF MASTER OF SCIENCE
DEPARTMENT OF ENERGY TECHNOLOGIES
IN APPLIED PROPULSION SYSTEM DESIGN &
ENGINEERING FOR AEROSPACE TECHNOLOGIES
PROGRAM**

ADVISOR: PROF. DR. MEHMET ALİ ARSLAN

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T.C.
GEBZE TEKNİK ÜNİVERSİTESİ
LİSANSÜSTÜ EĞİTİM ENSTİTÜSÜ

GAZ TÜRBİNLİ MOTOR ANNULAR YANMA
ODASINDA YÜKSEK ÇEVİRİMLİ YORULMA ÖMÜR
DEĞERLENDİRMESİ

AHMET ONUR ŞAHİN

YÜKSEK LİSANS TEZİ
ENERJİ TEKNOLOJİLERİ ANABİLİM DALI
HAVACILIK VE UZAY TEKNOLOJİLERİNDE
UYGULAMALI İTKİ SİSTEMİ TASARIM
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DANIŞMAN: PROF. DR. MEHMET ALİ ARSLAN

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ABSTRACT

Vibration is one of the reasons for the most critical failure modes in the design processes and life management of gas turbine aviation engines. Vibration causes high cycle fatigue in parts, which can lead to part loss, engine loss, and most importantly, loss of life. It is of great importance to evaluate the effect of vibration on the parts that the engine will be exposed to throughout its service life and to make it safe. In this context, the Flame Tube part, one of the gas turbine engine components, is studied. This part, which is freer than other parts due to its design nature, is heavily affected by vibration. By examining high cycle fatigue methods, the Goodman method was chosen and the life of the areas with the highest stresses due to vibration in critical locations of the part was evaluated. This study examines the high cycle fatigue life evaluation process of the Flame Tube part in detail and points out the applicability of this method to other parts within the engine. The required material data for the calculations were obtained from tests and life evaluation was carried out using these data. The free condition vibration characteristics of the part model created using finite element analysis were verified by comparing it with the hammer test results of the real experimental part. The results obtained from this study were evaluated from various aspects and future studies were explored to improve the method.

Key Words: Fatigue, High Cycle Fatigue, Stress, Finite Element Analysis, Life Assessment

ÖZET

Gaz türbinli havacılık motorlarının tasarım süreçlerinde ve ömür yönetiminde en kritik hata modlarının yaşanma sebeplerinden biri de titreşimdir. Titreşim, parçalarda yüksek çevrimli yorulmaya neden olarak parça kaybına, motor kaybına ve en önemlisi can kaybına sebep olabilir. Motorun servis ömrü boyunca maruz kalacağı titreşimin parçalara olan etkisinin değerlendirilmesi ve güvenli hale getirilmesi büyük önem taşımaktadır. Bu bağlamda, bu çalışmada gaz türbinli motor bileşenlerinden biri olan Flame Tube (Yanma Odası) parçası ele alınmıştır. Tasarım doğası gereği diğer parçalara göre daha serbest olan bu parça, titreşimden yoğun bir şekilde etkilenmektedir. Yüksek çevrimli yorulma metotları incelenerek Goodman metodu seçilmiş ve parçanın kritik lokasyonlarındaki titreşim sebepli en yüksek streslere sahip bölgelerin ömür değerlendirmesi yapılmıştır. Bu çalışma, Flame Tube parçasının yüksek çevrimli yorulma ömrü değerlendirmesi sürecini detaylı bir şekilde incelemekte ve bu metodun motor içerisindeki diğer parçalar için de uygulanabilirliğine işaret etmektedir. Hesaplamalar için gerekli malzeme verileri oluşturulmuş ve bu veriler kullanılarak ömür değerlendirmesi gerçekleştirilmiştir. Sonlu elemanlar analizi kullanılarak oluşturulan parça modelinin serbest koşuldaki titreşim karakteristiği, gerçek deneysel parçanın çekiç testi sonuçları ile karşılaştırılarak doğrulanmıştır. Bu çalışma ile elde edilen sonuçlar çeşitli açılardan değerlendirilmiş ve metodun geliştirilebilmesi için gelecekte yapılacak çalışmalar belirlenmiştir.

Anahtar Kelimeler: Yorulma, Yüksek Çevrimli Yorulma, Gerilme, Sonlu Elemanlar Analizi, Ömür Değerlendirmesi

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LIST OF SYMBOLS AND ABBREVIATIONS

σ_a	: Alternating Stress
N	: Cycle (Life)
S	: Cyclic Stress Amplitude
E	: Elastic Modulus
S_{eqv}	: Equivalent Von-Mises Stress
ε'_f	: Fatigue Ductility Constant
c	: Fatigue Ductility Exponent
N_f	: Fatigue Life
σ'_f	: Fatigue Strength Coefficient
b	: Fatigue Strength Exponent
Hz	: Frequency
S_1	: Maximum Principal Stress
σ_{max}	: Maximum Stress
σ_m	: Mean Stress
σ_{min}	: Minimum Stress
ε_n	: Nominal Strain
ν	: Poisson's Ratio
ε	: Strain
ε_a	: Strain Amplitude
σ	: Stress
σ_a	: Stress Amplitude
A	: Stress Amplitude Ratio
K_t	: Stress Concentration Factor
σ_r	: Stress Range
R	: Stress Ratio
ε_t	: True Strain
<i>HCF</i>	: High Cycle Fatigue
<i>LEFM</i>	: Linear Elastic Fracture Mechanics
<i>LCF</i>	: Low Cycle Fatigue
PF	: Participation Factor
RoEM	: Ratio of Effective Mass to Total Mass
Temp	: Temperature

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1. INTRODUCTION

1.1 Gas Turbine Engines and Main Components

A gas turbine is a compact and lightweight power plant, capable of generating substantial energy output. As a result, it has become increasingly popular in the power industry. The aerospace industry has been a leader in the development of gas turbine technology, with a focus on designing engines that are highly reliable, perform well, and can be started and operated flexibly during flights.

In the last century, various gas turbine engines were developed to enhance the capabilities of aircraft. These included turbojets, turboshafts, and turboprops, all of which were based on the principles of gas turbines. The competition among aircraft engine manufacturers has spurred the development of more powerful yet fuel-efficient engines.

A gas turbine is comprised of several main components, including a compressor, a combustion chamber, and a turbine.

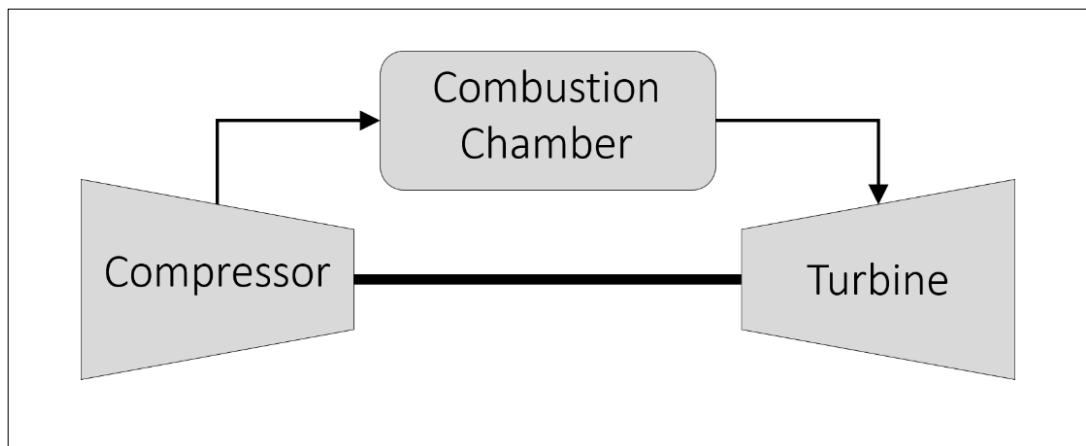


Figure 1.1: Gas turbine engine main components.

These components work together to compress air, mix it with fuel and ignite it, and then use the resulting hot gas to spin the turbine and generate mechanical power.

1.2 Combustion Chamber and Types

The combustion chamber, a pivotal element in a gas turbine engine, undertakes the challenging responsibility of burning substantial quantities of fuel, delivered through specialized fuel spray nozzles, in tandem with air provided by the compressor. The intricacies of this combustion process are managed to release heat in a controlled manner, prompting the expansion and acceleration of the air. The ultimate objective is to produce a seamless flow of uniformly heated gas, ensuring that the turbine operates optimally under diverse conditions. In essence, the combustion chamber serves as the linchpin in the engine's intricate system, where the harmonious interaction of fuel, air, and controlled heat release is fundamental for its reliable and efficient performance.

[1].

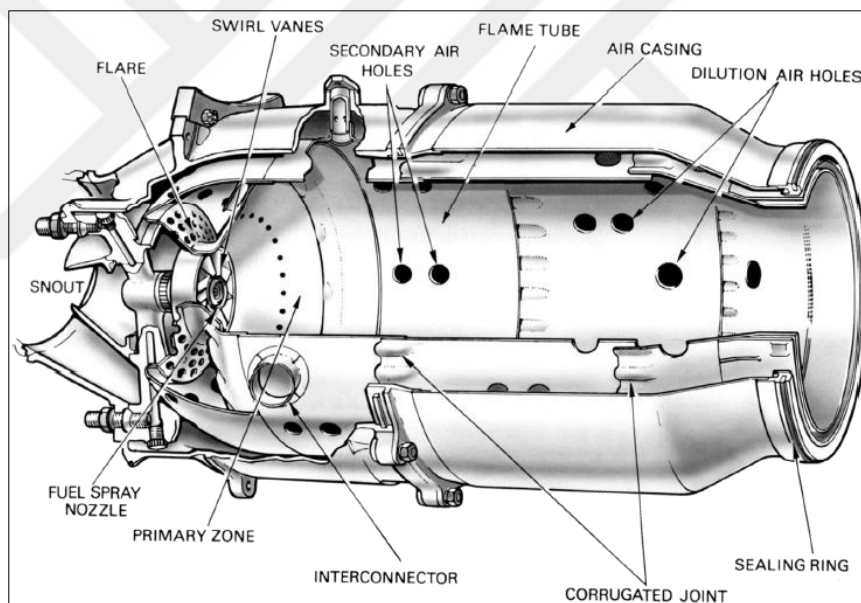


Figure 1.2: General View of Combustion Chamber [2].

There are three (3) main types of combustion used in gas turbine: the tubular (can), the tubo-annular (cannular), and the annular combustion chamber.

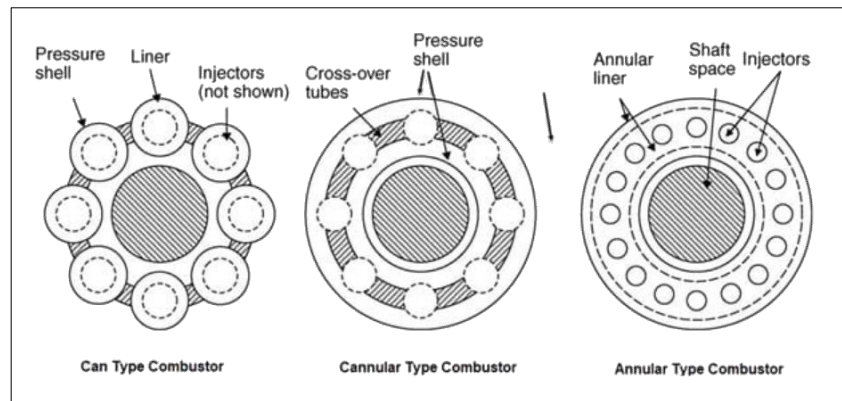


Figure 1.3: Main Types of Combustors [2].

Tubular (Can):

The first aircraft engines made use of Tubular (or Can) combustors. The chambers are arranged surrounding the engine, and air delivered by the compressor is guided through ducts to enter each individual chamber. See **Figure 1.3 [1]**. As previously explained, the airflow moves through the snout of the flame tube and also between the tube and the outer casing [1]. The distinct flame tubes are interconnected, ensuring uniform pressure across all of them and facilitating the spread of combustion around the flame tubes during engine startup.

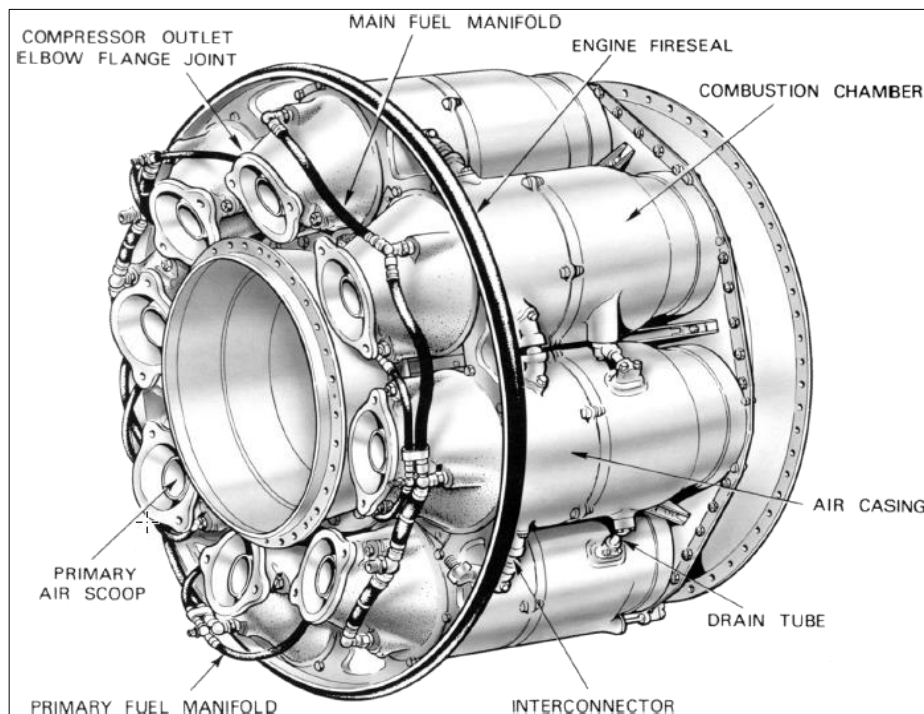


Figure 1.4: Multiple (Tubular, Can) [2].

Tubo-annular (Cannular):

A set of tubular liners, typically ranging from 6 to 10, is organized within a solitary annular casing, as depicted in **Figure 1.5** [2]. This concept aims to merge the compactness of an annular chamber with the structural integrity provided by a tubular chamber [3]. The exit flow from the tubo-annular typically exhibits a more consistent temperature distribution, enhancing the performance of the turbine section. Additionally, this design eliminates the necessity for each chamber to be equipped with its individual igniter.

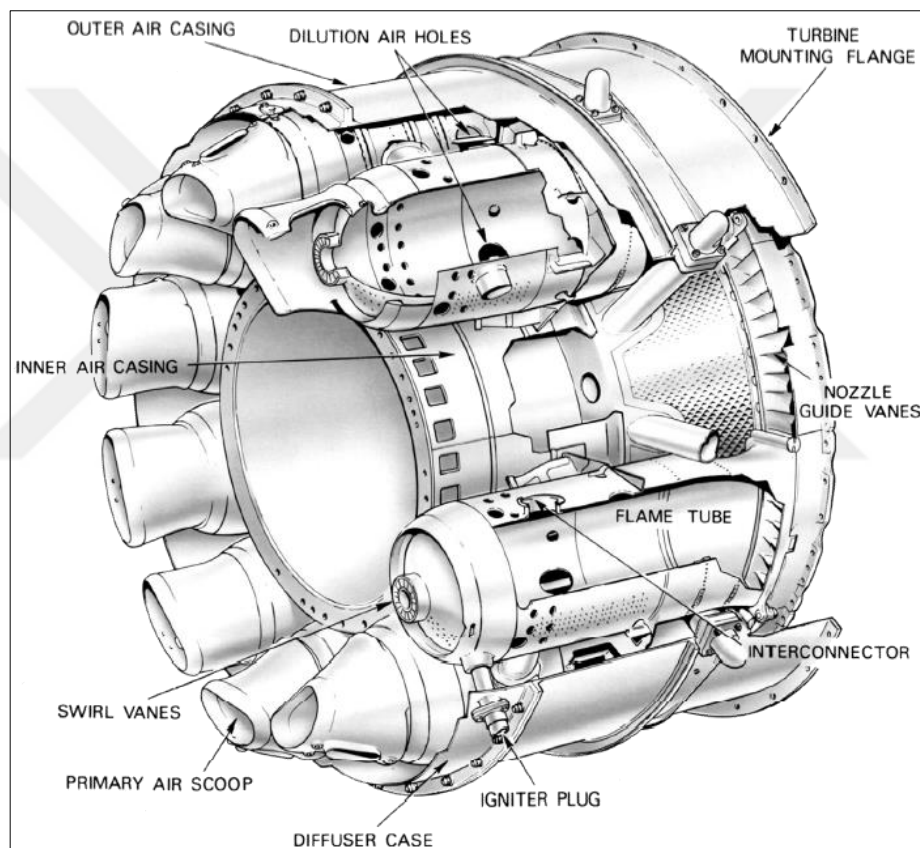


Figure 1.5: Turbo-Annular [2].

Annular:

In this configuration, an annular liner is installed concentrically within an annular combustion casing. In numerous aspects, it represents an optimal chamber design due to its streamlined aerodynamics, resulting in a compact unit with lower pressure losses compared to other combustor configurations [3]. However, its primary drawback arises from the substantial buckling load exerted on the outer liner [3]. **Figure 1.4.**

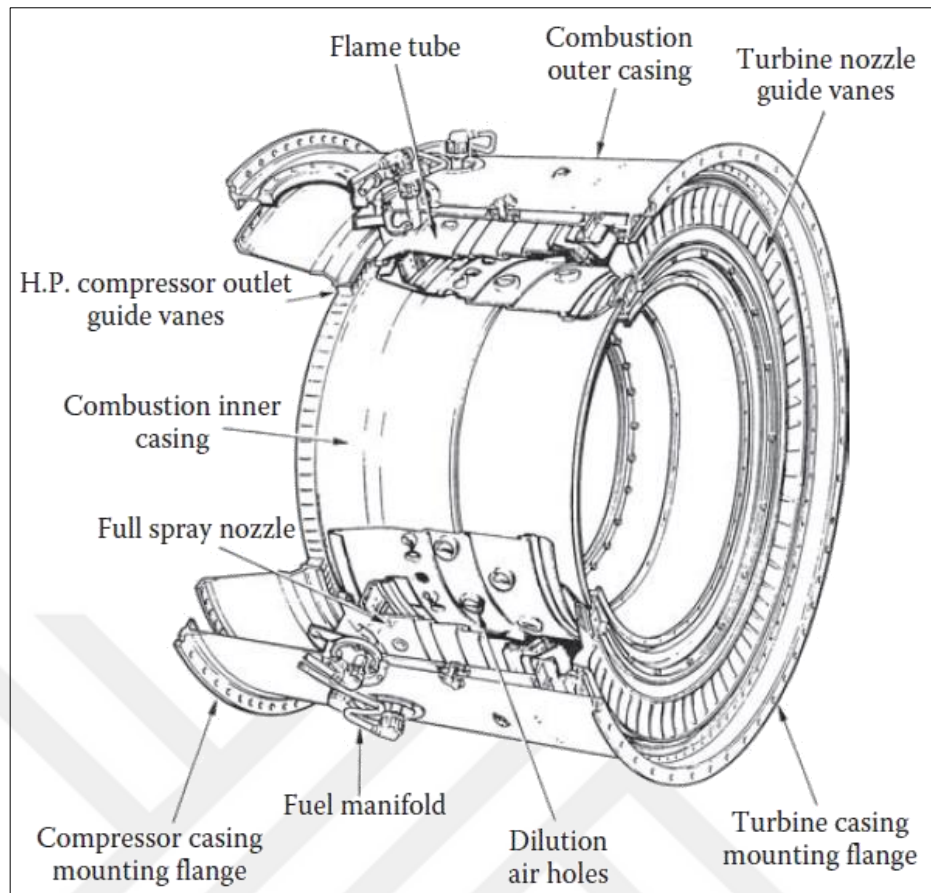


Figure 1.6: Annular Combustion Chamber [2].

1.3 What is High Cycle Fatigue?

The High Cycle Fatigue (HCF) is a type of fatigue that occurs in materials after undergoing a large number of repeated loads or stresses. This type of fatigue is commonly identified by a very high number of stress cycle, often in the millions or billions, before failure occurs. In contrast to low cycle fatigue, which typically involves fewer cycles but higher stress levels, high cycle fatigue is more likely to occur in materials that are subjected to relatively low levels of stress over long periods of time.

High Cycle Fatigue (HCF) is typically characterized by elevated frequencies, low amplitudes, cyclic behavior approximating elasticity, and a high number of cycles. Positioned at the right end of a traditional stress-life curve or a Wöhler diagram. HCF occurs where the cycle count is generally too extensive to acquire a statistically

meaningful dataset. This circumstance poses a challenge in accurately characterizing material behavior with a notably high level of confidence [4].

1.4 Structural Overview of the Thesis

The thesis initiates by delving into a comprehensive exploration of the cyclic fatigue phenomenon through a literature review in Chapter 2. The chapter begins by providing some historical background on fatigue, fatigue life in aviation, and structural integrity.

Next, the structural integrity and mechanical behavior of materials is explained, followed by a definition of mechanical behavior of materials and an explanation of the parameters that affect fatigue life. The chapter then discusses fatigue failure on components, including the need for a good understanding of stress cycles, low cycle fatigue (LCF), high cycle fatigue (HCF) and mean stress correction curves.

In chapter 3, a general description of the Flame Tube is introduced. Information is given about the materials used in the assembly. To facilitate a comparison of the material's mechanical properties in various forms, **Table 3.1** displays the mechanical properties at room temperature. In the subsequent sections, the finite element analysis performed in order to create the necessary outputs for the life evaluation is explained in detail. The result of the static analysis performed towards the end of the chapter are given. Based on this static analysis model, a model analysis was run that can simulate the real-life working conditions of the part. Finite element modal analysis results were examined, and Campbell diagram was generated. Harmonic analysis was performed for each of the critical modes determined by the Campbell diagram. With harmonic analysis, alternating stresses occurred by critical modes on the part were determined.

In chapter 4, how to obtain the material data required to evaluate the life of the part is explained. A high cycle fatigue testing system is described. S-N Curve and Goodman Diagram belonging to material of the critical regions have been normalized and presented. In the rest of this section, an impact hammer test was performed on the Flame Tube in order to verify the finite element analysis model.

In the last section, the impact hammer test results and free-free modal analysis results are compared in the first title. With this comparison, validation of the real Flame Tube part with Flame Tube's finite element analysis model was ensured. Continuing through this section, alternating stresses obtained from harmonic analysis, mean stresses

obtained from static analysis, and the goodman curve obtained from material high cycle fatigue testing were used to evaluate the life of Flame Tube. The results were evaluated and recommendations were made to improve the method.



2. LITERATURE SURVEY

2.1 Historical Background of Fatigue

The origin of fatigue can be traced back to Albert, a Royal Hannoverian civil servant specializing in mines. In 1837, he unveiled the inaugural results of fatigue tests in Clausthal. Albert's focus extended beyond mere material examination, as he ingeniously designed a testing apparatus for the conveyor chains that had previously failed in the Clausthal mines. Notably, he conducted tests on real components, not limiting his analysis to materials alone. Recognizing the substantial expense involved in importing hemp rope as the sole replacement for chains during that era, Albert's innovation of the wire rope emerged as a pivotal development, arguably overshadowing the significance of those initial fatigue tests.

In 1842, Rankine, renowned for his contributions to thermodynamics through the "Rankine process," delved into the fatigue characteristics of railway axles. His proposal involved forging axles with an expanded hub diameter and generous radii to minimize disruption to the grain flow during machining. Concurrently, York undertook experiments specifically focused on railway axles.

In 1853, the Frenchman Morin addressed the fatigue-related concerns in his book, where he detailed accounts from two engineers overseeing horse-drawn mail coaches. An early illustration of the "safe life" design methodology emerged as they recommended replacing the coaches' axles after covering 60,000 km. Additionally, for axles in other mail coaches, a meticulous inspection was advised after 70,000 km, with any identified cracks addressed through "fire-welding." Interestingly, observations highlighted that these cracks predominantly manifested at points of section transitions.

Contrary to the widely held belief attributing the term "fatigue" to Morin, it was actually first introduced by the Englishman Braithwaite in 1854. Braithwaite, however, credits Mr. Field with coining the term [5].

The initiation of fatigue investigation, particularly the adoption of fatigue life curves, is often mistakenly attributed to Wöhler. Wöhler wasn't the initial researcher to explore fatigue, and he presented fatigue life data using tables rather than curves. It's crucial to note, however, that this acknowledgment doesn't diminish his deserved recognition.

One of his significant contributions includes introducing the concept of an endurance limit and emphasizing that stress range, rather than peak stress, serves as a pivotal governing parameter [6].

Wöhler designed a test setup in the 1850s that involved two symmetrically connected railway shafts that were bent at a constant speed with a constant spring force until fatigue damage occurred. This setup is illustrated in **Figure 2.1**.

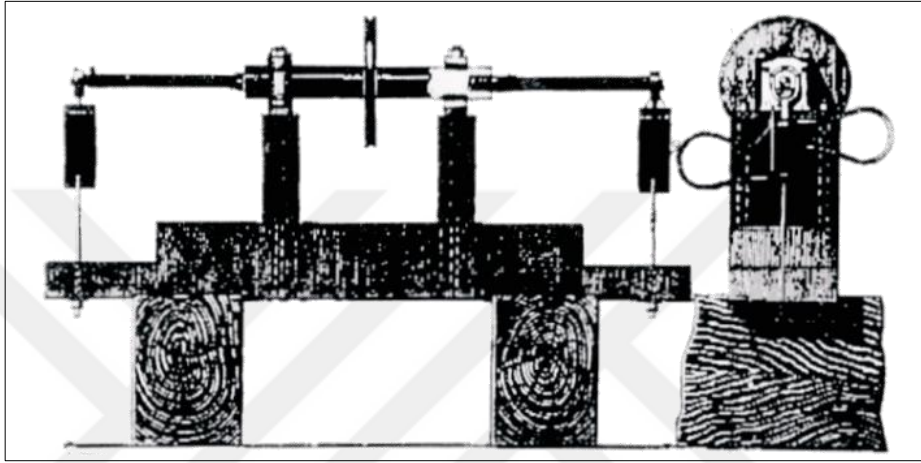


Figure 2.1: Wöhler's Rotating Bending Fatigue Test [7].

The groundbreaking studies he conducted in the 1860s remained unparalleled throughout the 19th century. Half a century later, when Basquin articulated the well-known log-log correlation between stress amplitude σ_a and fatigue life N_f as $\sigma_a = A(N_f)^b$, he drew upon Wöhler's fatigue life data.

Subsequently, the expression was modified to rely on the count of reversals [6]:

$$\sigma_a = \sigma'_f (2N_f)^b \quad (1.1)$$

σ_a : Stress Amplitude

σ'_f : Fatigue Strength Coefficient

N_f : Fatigue Life

b: Fatigue Strength Exponent

During the 1950s, fatigue and fracture gained significance, leading to an accelerated focus on mechanics studies. Post-war, technological strides made during World War

II found applications in aviation and shipbuilding. Notably, the United States embraced welding techniques in the construction of troop transport ships, departing from the slower riveting method, thus streamlining the manufacturing process. The transition temperature from ductile to brittle behavior in structural steel at that time was approximately 20 °C [8]. Liberty ships experienced a total of 145 structural failures, leading to either the loss of the vessel or the hull becoming perilously weakened. Additionally, 694 ships encountered significant fractures that necessitated urgent repairs [8].

The reliability assessment of aircraft stands out as a highly successful application of LEFM (linear elastic fracture mechanics). Notably, in 1953 and 1954, the loss of two aircraft resulted from a fast fracture originating from a fatigue crack that had grown from an astro-navigation window. One of the initial applications of LEFM was to provide an explanation for this rapid fracture [8]. A confluence of factors initiated crack propagation, subsequently leading to fracture caused by metal fatigue near the cabin fuselage window. Ultimately, this sequence of events resulted in the crash of the aircraft [9]. Subsequent to **Figure 2.2** [9], the schematic representation depicts the location of the crack in the Comet aircraft.

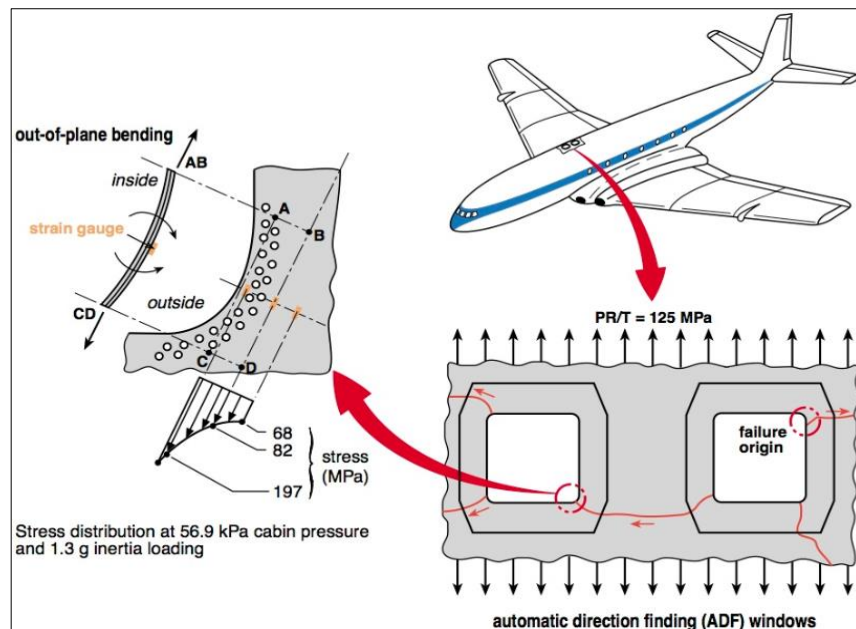


Figure 2.2: Comet accident – Origin of the crack [9].

In 1954, Manson and Coffin outlined the behavior of metallic materials subjected to inelastic cyclic strain amplitudes, presenting a four-parameter formula. This

formulation continues to be applied in the current LCF (Low Cycle Fatigue) domain and for predicting fatigue life based on the local concept.

Moreover, in the 1980s and 1990s, substantial transformations in various facets of fatigue design were ascribed to advancements in computer technology. These encompassed the development of software for diverse fatigue life (durability) models and improvements in simulating actual loadings under conditions of variable amplitude, involving specimens, components, or full-scale structures. Consequently, this integration significantly increased the incorporation of field testing within the laboratory. The advent of Integrated CAE, incorporating finite element analysis, dynamic simulation, and life prediction models, gave rise to the concept of confining testing to component durability rather than utilizing it for developmental purposes. The twenty-first-century objective has shifted towards enhanced digital prototyping with reduced reliance on physical testing in fatigue design [10].

2.2 Structural Integrity

The aim of structural integrity is to avert devastating failures throughout the operational life of a component. This assurance is maintained through two primary criteria: strength (static) and service life.

The static strength criterion guarantees that the hardware possesses adequate load capacity for the maximum load generated in a single load cycle. This can be assessed in the design phase using nominal material properties and nominal geometrical parameters. Nevertheless, despite its apparent suitability, static strength tests do not assess the load carrying capacity of the hardware when subjected to initial and/or cumulative damage during service.

Therefore, implications arise regarding service life requirements, considering factors such as usage duration and the mechanisms of induced damage during operation. This involves specifying not only the rate at which damage accumulates but also determining the residual strength of the component. Life (service) testing fundamentally establishes a safe operational timeframe, incorporating multiple intervention opportunities to minimize the probability of a devastating failure during operation [11].

2.3 Materials Mechanics

This section provides definitions for fundamental terms in the mechanics of materials, along with a recapitulation of basic formulations associated with these terms.

Mechanical behavior refers to the way materials respond to external forces or loads. It is the study of how materials deform and break when subjected to various types of stress. This includes the study of elasticity, plasticity, and fracture. The mechanical behavior of a material can be characterized by its strength, stiffness, ductility, toughness, and other properties. Comprehending the mechanical behavior of materials is crucial for the design of structures and machines that are both safe and reliable.

In mechanics of materials, strength refers to the ability of a material to withstand a load or external force without breaking or failing. Different materials have different strengths, and the strength of a particular material depends on its composition, structure, and other factors. There are several types of strength that are commonly used to characterize the mechanical behavior of materials, including tensile strength, compressive strength, shear strength, and yield strength. The strength of a material is an important factor in determining its suitability for different applications. It is best described as "Strengths are the magnitudes of stresses at which something of interest occurs, such as the proportional limit, 0.2 percent-offset yielding, or fracture. In many cases, such events represent the stress level at which loss of function occurs." [12].

Stress is a measure of the internal forces within a body that arise as a response to external forces applied to it. It is calculated as the force per unit area of the material and is typically measured in units of pounds per square inch (PSI) or pascals (Pa). Stress can be either tensile, compressive, or shear, depending on the direction of the applied forces and the resulting deformation of the material. The normal stress, represented by σ , is the resistance uniformly distributed over the normal section per unit area. (2.3)

$$\sum F = 0 \quad (2.1)$$

$$F - \sigma A = 0 \quad (2.2)$$

$$\sigma = \frac{F}{A} \quad (2.3)$$

An external force applied perpendicular to the surface area induces normal stress, while shear stress is generated by a force applied parallel to the surface area.

Stress requires a dual subscript definition, with the initial subscript denoting the normal direction to the applied force area, while the second subscript specifies the direction of the applied force. Normal stress is represented when the same subscript is applied twice, whereas shear stress, alternatively denoted as τ , utilizes mixed subscripts to signify the directions of force.

The tensor notation is shown in equation (2.4).

$$\sigma_{ij} = \begin{bmatrix} \sigma_{xx} & \sigma_{yx} & \sigma_{zx} \\ \sigma_{xy} & \sigma_{yy} & \sigma_{zy} \\ \sigma_{xz} & \sigma_{yz} & \sigma_{zz} \end{bmatrix} \quad (2.4)$$

During a tensile test on a machine, as the applied force (F) increases, the length of the specimen also increases. The incremental change in length dl is directly proportional to the corresponding increase in force dF . In the equation, the normalized increase in length is specified (2.5).

$$d\varepsilon = \frac{dl}{l} \quad (2.5)$$

The finite form, known as *true strain*, is obtained by integrating from the initial length, L_o , to the current length, L , (2.6).

$$\varepsilon = \int dL/L = \ln(L/L_o) \quad (2.6)$$

In numerous engineering applications, a simplified form of strain referred to as engineering strain is commonly employed. It is described as:

$$\epsilon_n = \epsilon_e = \frac{\Delta l}{l_0} = \frac{l_1}{l_0} - 1 \quad (2.7)$$

$$e = \Delta L / L_0 \quad (2.8)$$

For small strains, such as those typically found in metals (which are generally less than 0.005), the engineering (nominal) strain and true strain are almost identical during elastic deformation. However, as the deformation becomes plastic, the difference between the true strain and engineering (nominal) strain increases. The relationship between engineering strain and true strain during elastic deformation can be described using equation (2.9).

$$\epsilon_t = \ln(1 + \epsilon_e) \quad (2.9)$$

In the elastic region, the stress and strain of metals can be considered proportional, as described by Hooke's Law. This law, formulated by Robert Hooke through experiments in 1678, states that the relationship between stress and strain in metals can be represented by the equation shown in (2.10).

$$E = \sigma / \epsilon \quad (2.10)$$

Young's Modulus (E) also referred to as the modulus of elasticity or elastic modulus, quantifies the stiffness of a material. Although it is relatively constant, the value of Young's Modulus can be affected by heat and mechanical treatments.

In uni-axial tension, where the force is applied in the direction of the stress, the ratio of stress to strain provides Young's modulus. Uni-axial tension also induces lateral strains, as denoted in equation (2.11). Poisson's ratio, ν , shows the intention of an isotropic solid to shrink in two directions, when it is sustained under the force applied in the third direction. In the presence of a general stress state ($\sigma_x, \sigma_y, \sigma_z$) resulting in

the strain e_x , Poisson contractions in σ_y and σ_z are illustrated as depicted in (2.12), with stress σ_x contributing as described in (2.13).

$$e_x = e_z = -\nu e_x \quad (2.11)$$

$$e_x = -\nu \sigma_y / E \quad \text{and} \quad e_x = -\nu \sigma_z / E \quad (2.12)$$

$$e_x = \sigma_x / E \quad (2.13)$$

Hooke's law can be expressed in general terms as follows:

$$e_x = (1 / E)[\sigma_x - \nu(\sigma_x + \sigma_z)] \quad (2.14)$$

Principal stresses represent the normal stresses that emerge when all shear stresses become nonexistent. They are the main or dominant stresses in a material. It is always feasible to identify a set of axes (1, 2, 3) on a body that establishes the principal stresses. The principal stresses, σ_1 , σ_2 , and σ_3 , can be characterized as the three roots of equation (2.15).

$$\sigma_p^3 - I_1 \sigma_p^2 - I_2 \sigma_p - I_3 = 0 \quad (2.15)$$

Where;

$$I_1 = \sigma_{xx} + \sigma_{yy} + \sigma_{zz} , \quad (2.16)$$

$$I_2 = \sigma_{yz}^2 + \sigma_{zx}^2 + \sigma_{xy}^2 - \sigma_{yy}\sigma_{zz} - \sigma_{zz}\sigma_{xx} - \sigma_{xx}\sigma_{yy} , \quad (2.17)$$

$$I_3 = \sigma_{xx}\sigma_{yy}\sigma_{zz} + 2\sigma_{yz}\sigma_{zx}\sigma_{xy} - \sigma_{xx}\sigma_{yz}^2 - \sigma_{yy}\sigma_{zx}^2 - \sigma_{zz}\sigma_{xy}^2 \quad (2.18)$$

The Von-Mises stress, extensively employed in design as a yield criterion, can be calculated by using the principal stresses. The stress level should be kept below the yield stress of the material, indicating the initiation of plastic deformation. Von-Mises stress takes into account all of the principal stresses ($\sigma_1, \sigma_2, \sigma_3$) and is expressed as:

$$\sigma_v = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2} \right]^{1/2} \quad (2.19)$$

2.4 Factors Affecting Fatigue Life

The standard fatigue condition for stress-life behavior typically includes fully reversed ($R=-1$) bending or axial loading, employing small, clean (unnotched) specimens. However, the reference fatigue condition is subject to several influencing factors, leading to potential variations in fatigue life results for the same material. Important factors affecting fatigue life are outlined below [10].

- **Loading Type:** The type of loading is a critical factor influencing the strength of a part. The life of a part differs due to axial tensile stress, bending, torsion, or more than one type of combined loading. In addition, the load intensity, frequency, constant or fluctuating load also affect the fatigue life of the part.
- **Effect of Microstructure:** Grain structure and microstructural flaws have a substantial impact on fatigue life. Consider a scenario where a material has a larger grain size; in such cases, its fatigue limit might be lower than that of a material with a smaller grain size when exposed to cyclic loading at ambient temperature. In contrast, at elevated temperatures within the creep range of metals, materials characterized by a coarse grain structure typically showcase improved fatigue properties [13].
- **Processing Techniques:** Directional properties are introduced into materials through processing techniques like forging, rolling, and extrusion, resulting in grain orientation. Usually, fatigue life is increased in the oriented direction but decreased in the transverse direction. These defects act as stress concentrators and may potentially serve as sites for crack initiation. Various processes, including case hardening, surface coating, heat treatment, plating, cold and hot working etc., have

the potential to impact fatigue life. For instance, the manufacturing process might generate residual stresses, which are self equilibrium stresses existing in a material without external loads. In most cases, the presence of compressive residual stresses at external surfaces has a positive influence on fatigue life, whereas tensile residual stresses are disadvantageous [13].

- **Size Effect:** Under unnotched bending conditions, if the diameter or thickness of the specimen is <10 mm (0.4 in.), then the S-N fatigue behavior is reasonably independent of the diameter or thickness. For larger size, S-N fatigue resistance is decreased [10]. Therefore, the size of the part can affect the fatigue behavior. Also, depending on the design of a part, the stress concentration may be greater in one region. The sharp corner turns of the windows of the comet planes caused fatigue fracture as they created stress concentration in this region. For this reason, modern airplane windows are more circular in shape and less angular. A very high proportion of all fatigue errors start on the surface of the parts. Therefore, surface conditions are one of the most important factors that affect fatigue resistance. Scratches, dents, machining marks on the surface help reduce the fatigue resistance of the part by contributing to crack formation. Moreover, notches are well-known for affecting stress distribution and causing elevated localized stresses. The theoretical stress concentration factor, denoted as K_t , is a significant factor, as it intensifies the applied force in the concentrated region. This localized stress raiser is known to occasionally surpass the threshold stress for crack opening, thereby hastening the initiation of fatigue cracks [14].
- **Environmental Factors:** Material degradation commonly takes place in a corrosive environment, often revealing itself through visual signs like pitting or surface roughness. These notch-like areas serve as stress concentrators and typically become the initiation sites for cracks. Particularly crucial are locations where fatigue slip activity is occurring. A marginal variation in the electrochemical potential between the exterior and interior of the slip band can facilitate the corrosion process [10].
- **Frequency:** The relationship between frequency and the S-N behavior of metals is intricate, as it involves the combined effects of factors such as test temperature, corrosive environment, stress-strain sensitivity to strain rate, and the frequency itself. Generally, elevated temperature and corrosive environments, when considered individually, tend to negatively impact fatigue resistance. In cases

where higher test frequencies lead to specimen heating due to internal hysteresis damping, the increased temperature can obscure the genuine fatigue behavior at ambient temperature. This effect is particularly noteworthy for metals with lower strength [10].

2.5 Fatigue-Related Failures in Components

In the majority of practical scenarios, the stress on components tends to exhibit random fluctuations. When subjected to cyclic stress, these components often fail at much lower levels of stress than expected when subjected to constant stress. This is known as fatigue and is often the main cause of failure during operation.

Fatigue pertains to a material's capacity to endure repeated applications of loads or forces. When a material is exposed to cyclic loading, it may eventually experience failure as a result of the accumulation of microscopic cracks and defects. This phenomenon is known as fatigue failure. Fatigue is also defined as “*the process of progressive localized permanent structural change occurring in a material subjected to conditions that produce fluctuating stresses and strains at same point or points and that may culminate in cracks or complete fracture after a sufficient number of fluctuations.*” in ASTM E1823-21 [15].

Fatigue is a prevalent failure mode for numerous engineering materials, manifesting under diverse loading conditions such as tension and compression, and across various frequencies and amplitudes. Three essential factors contribute to fatigue: a maximum tensile stress of a significantly high value, a substantial variation or fluctuation in the applied stress, and a sufficiently large number of cycles of the applied stress [16].

2.5.1 Stress Curve Definition

There are several types of cyclic stress that can occur, including fully reversed cycles (where the maximum and minimum stresses are equal but have opposite signs) and repeated stress cycles (where there is a mean stress in addition to the maximum and minimum stresses). Irregular (random) cycles involve random loading during service. A diagram of a cyclic stress is presented in Figure 2.3. While it is typically depicted with sinusoidal waves, the actual shape of the wave is not significant. The frequency of the wave is not important as long as it allows the component sufficient time to

dissipate heat and return to a lower temperature. The term "constant amplitude fatigue cycle" refers to a basic type of cyclic stress.

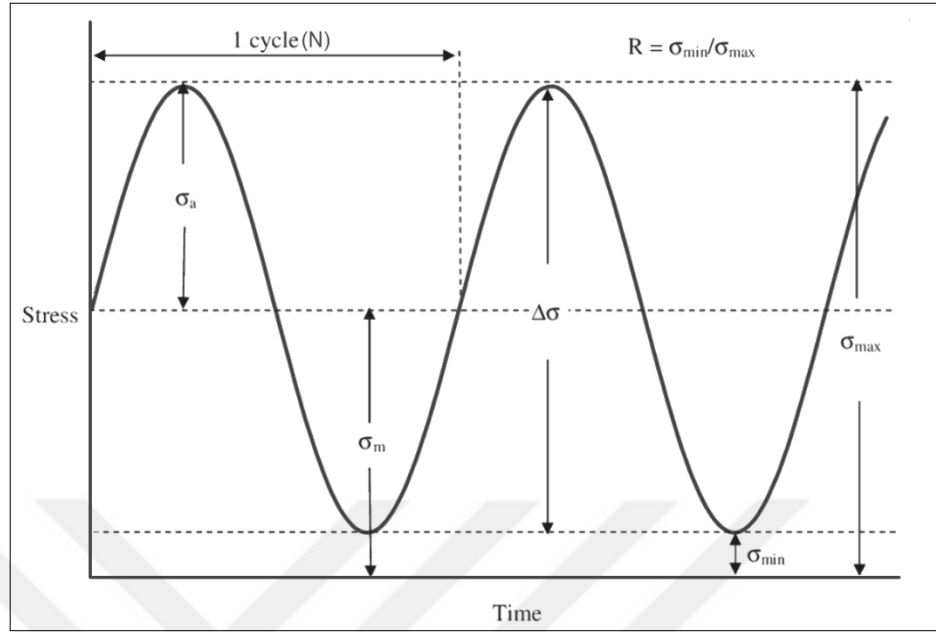


Figure 2.3: Schematic of Cyclic Stress.

Cycle (N): A cycle, also referred to as a period or load repetition, is the smallest unit of stress that is repeatedly applied to a part.

Maximum Stress (σ_{max}): The term "maximum stress" refers to the stress applied to a structure that has the highest absolute value. Tensile stresses are stated with positive (+) values whereas compressive stresses are expressed with negative (-) values.

Minimum Stress (σ_{min}): The term "minimum stress" refers to the applied stress having the lowest algebraic value.

Mean Stress (σ_m): The term "mean stress" refers to the arithmetic average of the maximum and minimum stresses.

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad (2.20)$$

Alternating Stress (σ_a): The alternating stress, also referred to as the stress amplitude, is defined as the difference between the peak stresses and the mean stress.

$$\sigma_a = \left| \frac{(\sigma_{max} - \sigma_{min})}{2} \right| \quad (2.21)$$

Stress Range (σ_r): The term "stress range" denotes the variance between maximum and minimum stresses.

$$\sigma_r = \sigma_{max} - \sigma_{min} \quad (2.22)$$

Stress Ratio (R): The R ratio is used for expressing the loading condition and it is the ratio of minimum and maximum stress.

$$R = \frac{\sigma_{min}}{\sigma_{max}} \quad (2.23)$$

Fully reversed ($R=-1$) loading and pulsating ($R=0$) loading are two prevalent loading conditions utilized in testing and for deriving fatigue properties, such as endurance limits. In the case of fully reversed loading, there is no mean component to the load, and the maximum tensile load and compressive load are equal. Conversely, for pulsating loading, the minimum stress is zero, resulting in a consistent tensile loading.

All the definitions elucidated in this section, along with various loading cases based on R ratios, are illustrated in **Figure 2.4 [17]**.

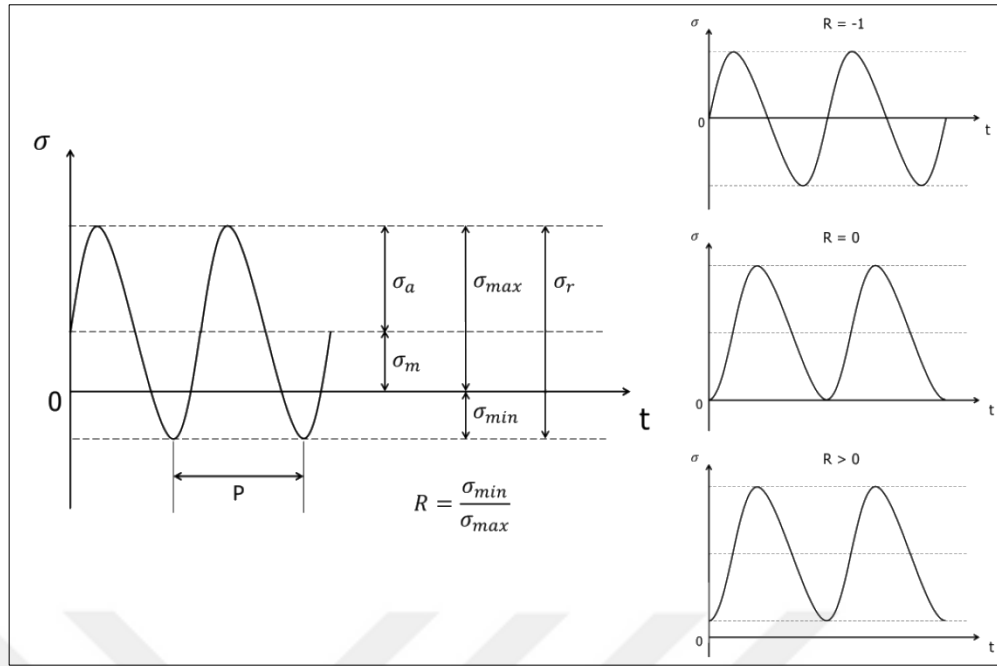


Figure 2.4: Definition of stress quantities for cyclic stresses [17].

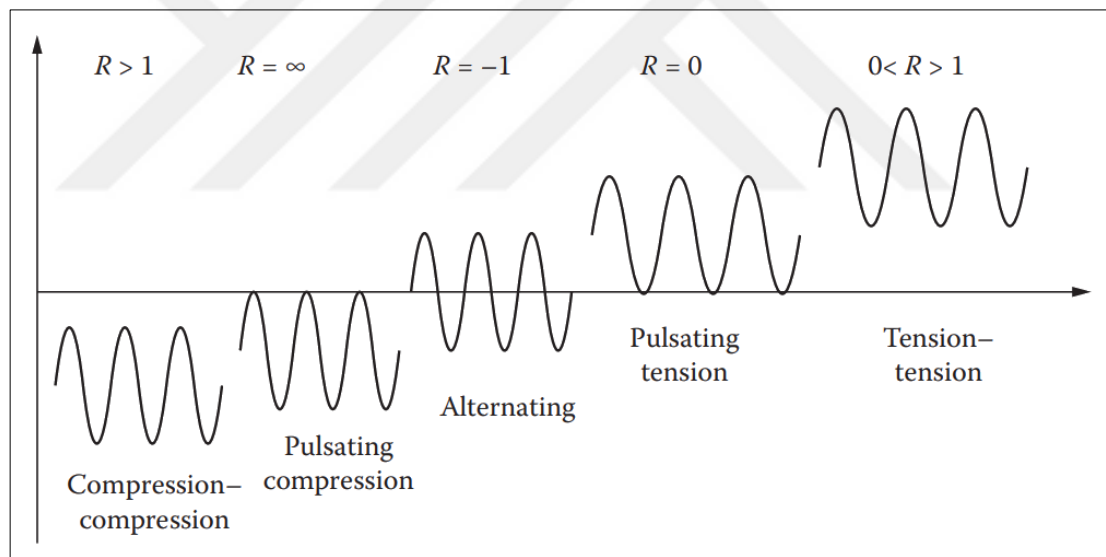


Figure 2.5: Stress Cycles [24].

Stress Amplitude Ratio (A): The stress amplitude ratio is a measure of the severity of cyclic loading on a material. It is defined as the ratio of the maximum alternating stress to the mean stress in a cyclic load and is typically expressed as a percentage.

The stress amplitude ratio is an important factor in predicting the fatigue life of a material, as it determines the amount of stress that the material is subjected to during each cycle of loading. Materials with a high stress amplitude ratio will generally have

a shorter fatigue life than those with a lower stress amplitude ratio, as the higher stress levels can cause more rapid crack growth and eventual failure.

$$A = \frac{\sigma_a}{\sigma_m} = \frac{1 - R}{1 + R} \quad (2.24)$$

2.5.2 Stress – Cycle Curve

Long before a microstructural understanding of fatigue processes emerged, engineers had already devised empirical methods to quantify fatigue processes and design against them. One crucial concept is the S-N diagram, where a constant cyclic stress amplitude $S(\sigma)$ is applied to a specimen, and the number of loading cycles N until failure is determined. Given that millions of cycles may be needed to induce failure at lower loading levels, the abscissa is typically plotted logarithmically [18].

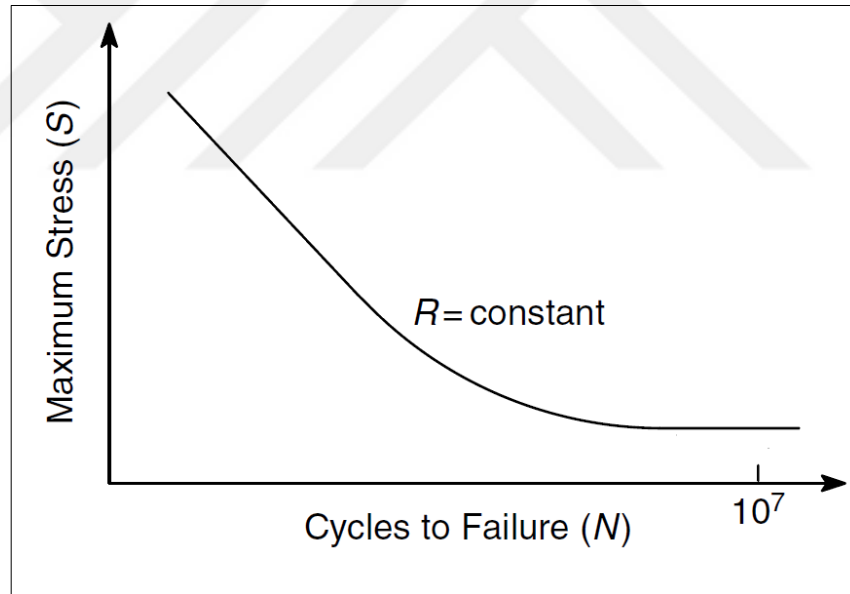


Figure 2.6: Schematic of S-N Curve [4].

At lower loading levels, millions of cycles may be required for the component to break. Therefore, the horizontal section on S-N curves is often plotted logarithmically [19]. After Wöhler's test studies, Basquin obtained the relationship shown in Section 1.1 using some of these data. In this relationship, "b" is referred to as the Basquin slope

(Fatigue Strength Exponent) and is the slope of the line plotted on the S-N log-log diagram.

$$b = \frac{-(\log \sigma - \log \sigma_0)}{(\log N_0 - \log N)} \quad (2.25)$$

$$\log N_0 - \log N = -\frac{1}{b} \log \left(\frac{\sigma}{\sigma_0} \right) \quad (2.26)$$

$$\log N = \log N_0 + \frac{1}{b} \log \left(\frac{\sigma}{\sigma_0} \right) \quad (2.27)$$

$$N = N_0 \left(\frac{\sigma}{\sigma_0} \right)^{1/b} \quad (2.28)$$

1/b is often expressed as “k”. Therefore, the equation takes the following form:

$$N = N_0 \left(\frac{\sigma}{\sigma_0} \right)^k \quad (2.29)$$

If the Basquin slope (b) and a stress value and the number of cycles corresponding to this stress value are known for calculating the fatigue life of a component, the number of cycles corresponding to any stress value can be calculated using the above relation.

2.5.3 Strain – Life Approach

The strain (ϵ) – life (N) method relies on the observation that, in numerous components, the material response in critical areas (such as notches) is contingent upon strain or deformation. When loads are minimal, stresses and strains exhibit a linear relationship. Therefore, in this range, load-controlled and strain-controlled test outcomes are interchangeable. For elevated load levels, within the low cycle fatigue (LCF) realm, the cyclic stress-strain response and material behavior are most accurately represented under strain-controlled conditions [20].

The Manson and Coffin equation, which is still used today, was developed in the 1950s and describes the relationship between plastic deformation and fatigue life.

$$\varepsilon_p = \varepsilon'_f (2N_f)^c \quad (2.30)$$

In this relation, “ ε_p ” is expressed as plastic strain amplitude, “ ε'_f ” is fatigue ductility constant, “ N_f ” is fatigue life of the part and “ c ” is fatigue ductility exponent.

The localized strain-life approach has been acknowledged as a valuable technique for assessing the fatigue life of a notched component. Both the American Society for Testing and Materials (ASTM) and the Society of Automotive Engineers (SAE) have endorsed guidelines and protocols for conducting strain-controlled tests and utilizing this data to forecast fatigue life [20].

Engineering components and structures are frequently exposed to cyclic loads, typically involving the existence of mean stress and mean strain. Several approaches have been devised to simulate the impact of mean stress on the fatigue characteristics of metals. It is well-established that mean stress significantly affects fatigue life, with tensile mean stress having a detrimental impact, while compressive mean stress is advantageous [21].

The traditional methods of evaluating the effect of mean stress on the S-N approximation are outlined in Section 2.6. The Morrow, Manson-Halford and Smith-Watson-Topper relations, which include the effect of mean stress in the ε -N approximation and are generally used in current calculations, are shown below.

Morrow,

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \quad (2.31)$$

Manson-Halford,

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^b + \varepsilon'_f \left(\frac{\sigma'_f - \sigma_m}{E} \right)^{c/b} (2N_f)^c \quad (2.32)$$

Smith-Watson-Topper,

$$\sigma_{max}\varepsilon_a = \sigma_{max}\frac{\Delta\varepsilon}{2} = \frac{(\sigma'_f)^2}{E}(2N_f)^{2b} + \varepsilon'_f\sigma'_f(2N_f)^{c+b} \quad (2.33)$$

In SWT, “ ε_a ” strain amplitude, “ σ'_f ” is fatigue resistance constant, “ ε'_f ” is the fatigue ductility constant, “ N_f ” is the fatigue life of the part, “ b ” is the fatigue strength exponent and “ c ” is the fatigue ductility exponent.

2.5.4 Comparison of LCF and HCF

Low Cycle Fatigue (LCF) is typically associated with failure occurring in fewer than 100,000 cycles, whereas High Cycle Fatigue (HCF) is linked to failure after than 100 million cycles. The conventional assumption is that the transition from LCF to HCF transpires when both elastic and plastic strains contribute equally to the total strain. Nevertheless, rather than focusing solely on the cycle count, it is more crucial to consider the extent of damage incurred during each load application. In the context of critical gas turbine components, the distinction between LCF and HCF often hinges on the source of the loading: LCF is primarily induced by pilot demands and the application of relatively large loads, while HCF is predominantly caused by vibration and the application of relatively small loads.

The high amplitude of cyclic stresses in LCF (low-cycle fatigue) leads to a higher accumulated strain energy per reversal than in HCF (high-cycle fatigue), resulting in a shorter time spent on crack initiation and a longer time spent on crack propagation.

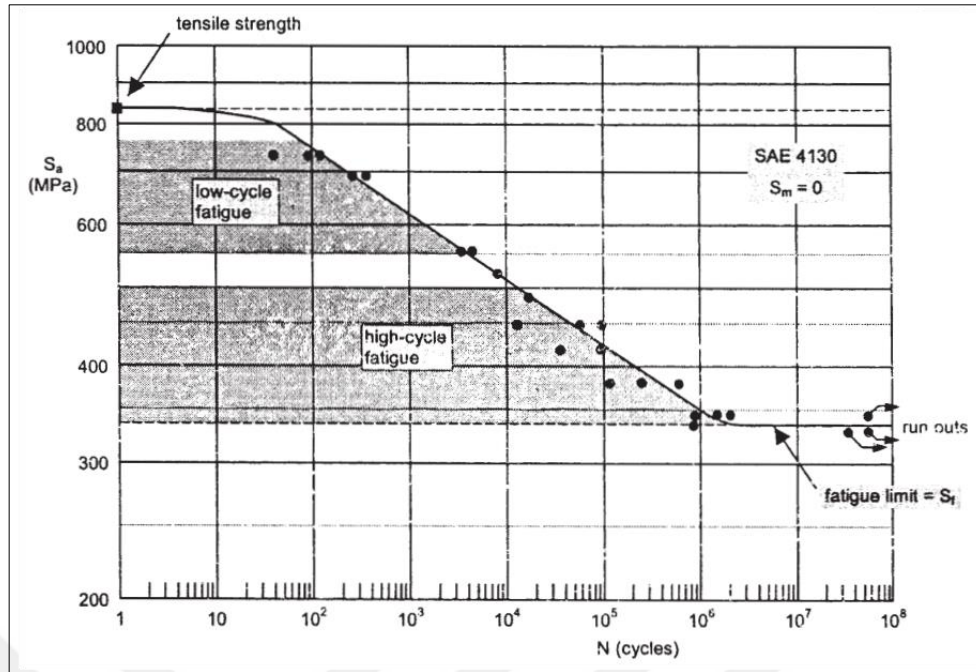


Figure 2.7: S-N Curve Example [22].

2.6 Mean Stress Correction Theroies

Mean stress correction is a method used to account for the effect of mean stress on the fatigue life of a material. Mean stress refers to the average stress that a material experiences over a period of time and can have a significant impact on the fatigue life of the material.

Mean stress correction curves are graphical representations of the relationship between mean stress and fatigue life for a particular material. These curves are used to predict the fatigue life of a material under different mean stress conditions and can be used to design components that will be subjected to cyclic loading.

Various methods exist for computing mean stress correction, such as the Goodman method, the Soderberg method, and the Gerber method. These approaches rely on experiments conducted with different R ratios and utilize permissible alternating and mean stress values to ascertain a safe zone where failure is not anticipated. Failure is expected in any method if the stress state surpasses the mean stress curve. The allowable alternating stress is typically considered as the endurance limit at 10^7 cycles obtained from fully reversed loading, while the allowable mean stress varies for each method [17].

In the Stress Life approach, when there are experimental data points at various mean stresses or r-ratios, mean stress can be directly incorporated through interpolation among material curves. In cases where experimental data is lacking, various empirical alternatives can be employed, including Gerber, Goodman, and Soderberg theories. These theories utilize static material properties (such as yield stress and tensile strength) in conjunction with S-N data to accommodate any mean stress effects [23].

Goodman,

$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_u} = 1 \quad (2.33)$$

Soderberg,

$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_y} = 1 \quad (2.33)$$

Gerber,

$$\frac{\sigma_a}{\sigma_e} + \left(\frac{\sigma_m}{\sigma_u}\right)^2 = 1 \quad (2.33)$$

The performance of most metals and alloys typically falls between Goodman's line and Gerber's parabola. Goodman's line closely aligns with experimental data for brittle materials but tends to be conservative for ductile materials. On the other hand, the Soderberg line is more conservative than the Goodman line but is more physically meaningful, as it sets the maximum possible mean stress equal to the yield stress [23].

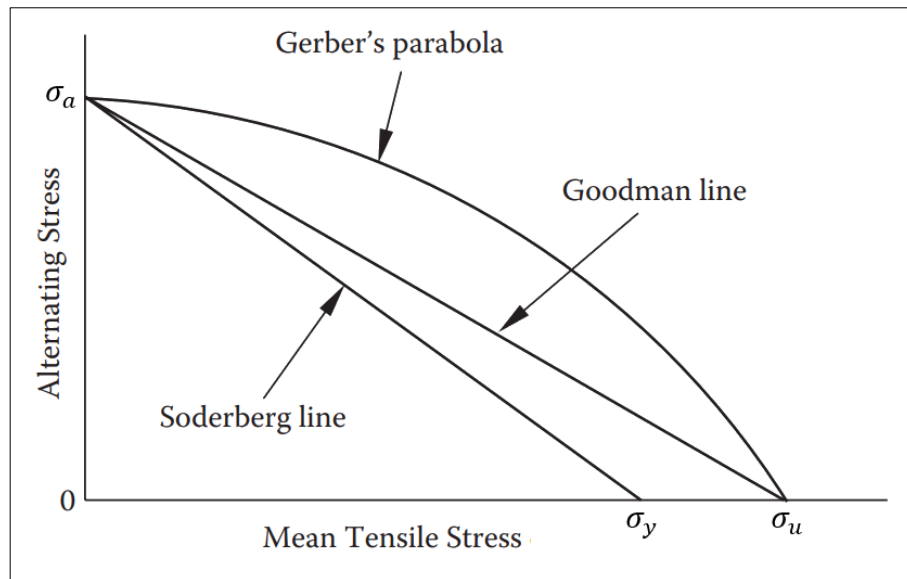


Figure 2.8: Comparison of Gerber, Soderberg, and Goodman laws.

In this thesis, assessment will be made by the Goodman method.

3. MATERIAL AND METHOD

This chapter provides an explanation of the geometry and FE (Finite Element) model, including details on geometrical adjustments, loading conditions, boundary conditions and meshing. The static and pre-stressed modal analyses results are presented, along with the creation of a campbell diagram to identify the critical locations of the HCF (High Cycle Fatigue). Critical frequencies are determined based on the diagram, and a harmonic analysis is performed. Finally, the life of the object is estimated using the Goodman methodology. In all images, the hole configuration is hidden because it is critical information.

3.1 Definition of Flame Tube

The Flame Tube is a radial structure that constitutes a part of the primary aerodynamic pathway within the engine.

A combustion chamber, also known as a flame tube, is a crucial part of a gas turbine engine. It's where fuel and air are combined, burned, and converted into hot gases that generate the turbine's power. The combustion chamber's design and construction are essential for the engine's efficiency and performance, withstanding high temperatures and pressures while ensuring complete fuel combustion and minimal emissions. Different combustion chambers/flame tubes are used based on the engine's application and performance needs.

The Flame Tube has four main parts: outer liner, inner liner, dome and aft dome with vaporization tubes. **Figure 3.2.**

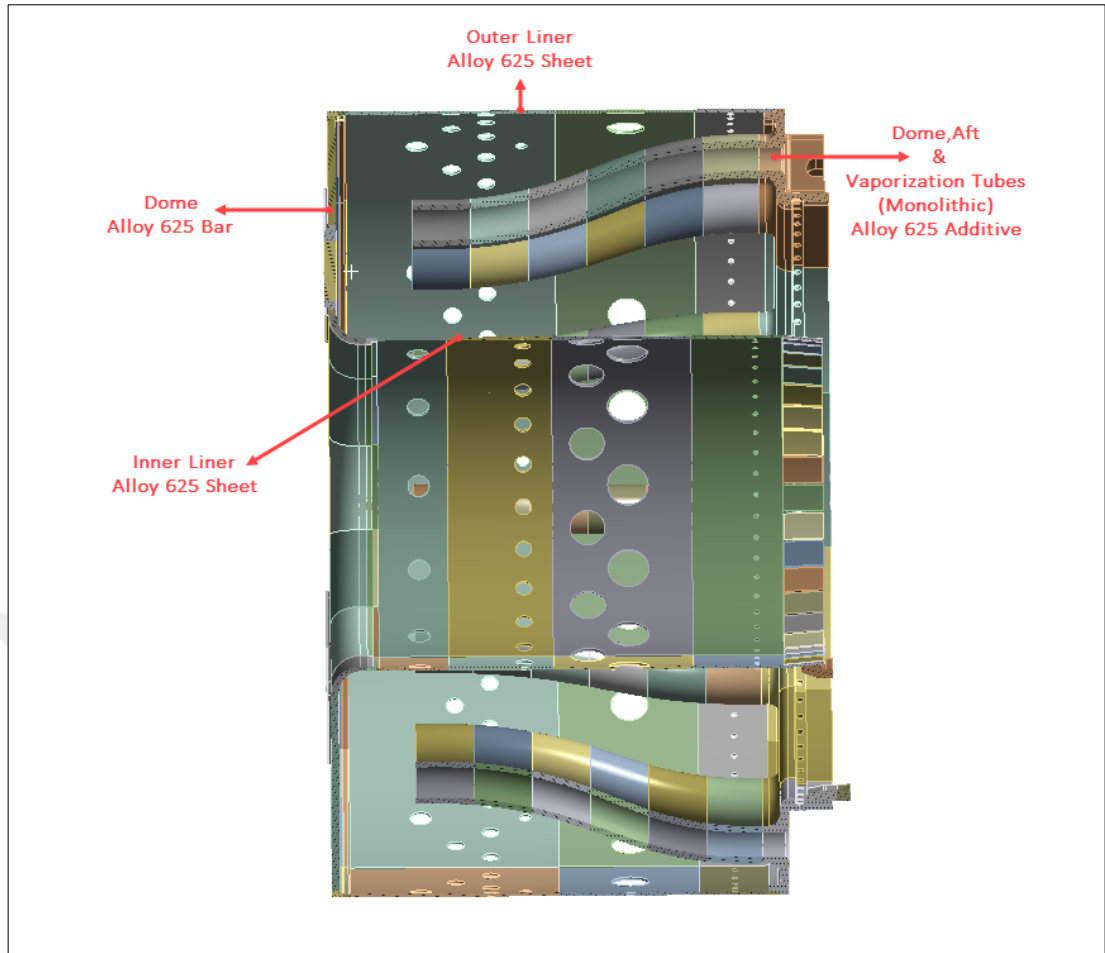


Figure 3.1: Small turbojet engine's Flame Tube main parts.

3.2 Material Properties

Due to adherence to corporate privacy policy and information security, the data resulting from material tests conducted at TEI's laboratory is considered proprietary company knowledge. As such, sharing or publicizing this data with external parties necessitates the company's consent. While material data is crucial for precise analysis, only normalized material data is disclosed in this section.

The Flame Tube is made of Alloy 625. The raw material is produced through bar, sheet, or additive manufacturing methods to accommodate various experimental trials. In this study, sheet, bar, and additive materials are used. The main elastic properties are provided in the following **Table 3.1**.

Table 3.1: Mechanical Properties of Alloy 625 Sheet, Bar and Additive

Material	Temp °C	Density [g/cm ³]	Elastic Modulus [GPa]	Poisson's Ratio	Yield Strength [MPa]	Tensile Strength [MPa]
			Normalized	Normalized	Normalized	Normalized
Alloy 625 Sheet	25	8.45	1	1	1	1
Alloy 625 Bar	25	8.45	1	1	0.79	0.92
Alloy 625 Additive	25	8.45	1	1	0.92	1.01

3.3 Preperation of Finite Element Analysis (FEA) Model in ANSYS WB

The Flame Tube is implemented as actual manufactured parts, ensuring that all components modeled with Ansys replicate the exact geometric details of the real counterpart. The parts are modeled to capture the genuine strength of the real parts.

3.3.1 CAD Model

The geometrical features of the Flame Tube are highly complex because of its function as a combustor, making it challenging to model using finite element modelling tools.

Therefore, the described procedure is followed:

- The components are designed using a CAD program and are precisely dimensioned, with every small feature included.
- Minor features, such as blends and chamfers, that are far enough from the area of interest and have no impact on stiffness, are removed.
- The model is divided into smaller sections to create a model using numerous brick elements, as described in the following section. Adequate space for transition elements is considered during this procedure.

Ansys WB has a distinct definition of mesh. Ansys WB stores all geometrical features in the FE model, meaning that elements and nodes remain connected to the solid geometry to allow for flexible load application. Certain regions allow for independent

meshing, with contact elements used to establish connections between elements situated on various converging surfaces.

The divided form of Flame Tube geometry designated for modeling in Ansys Workbench is shown in **Figure 3.3**.

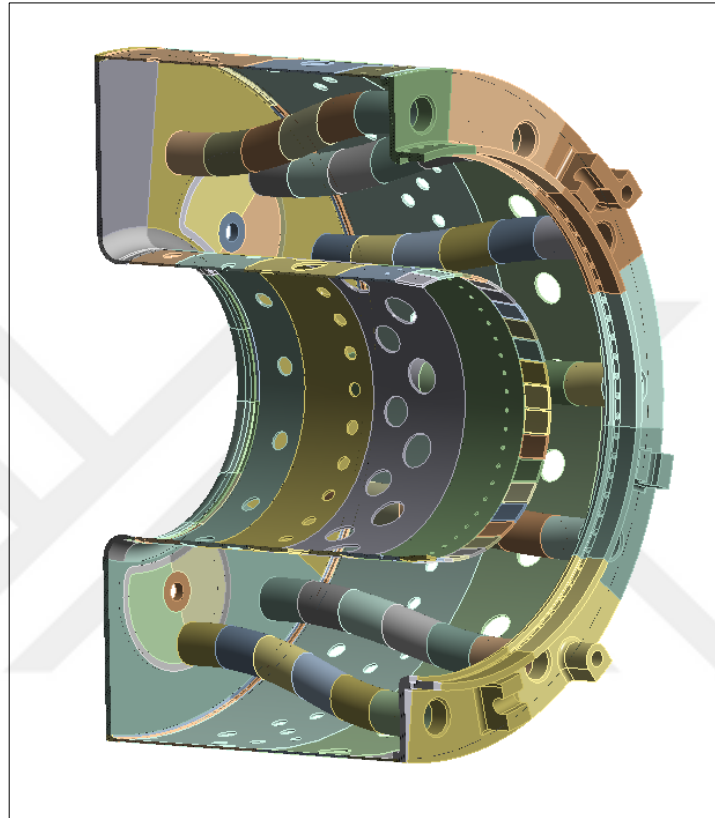


Figure 3.2: Chunked geometry of the Flame Tube (Cross-Section).

3.3.2 Mesh Model

Modeling the components, particularly the liners, in ANSYS WB is challenging because they are wide, lightweight components. If the total number of nodes is excessive, it can lead to difficulties, such as prolonged calculation time. To decrease the total element number and node number, the mesh size must be large enough, but not so large that it fails to reflect the actual geometry and model's stiffness. To accomplish this, the element size value is not uniform throughout the mesh.

This section outlines aspects pertaining to the mesh structure created for structural simulation and provides a general overview of mesh methodology. It highlights the

critical importance of maintaining a high level of mesh quality sensitivity to meet the convergence criteria for the analysis. Various element types are used in the finite element method for constructing the mesh structure.

Non-linear behavior is not examined in this study. The harmonic analysis method also takes a very long time. For this reason, Linear Solid 185 elements were used extensively to shorten the solution time.

Solid 185 is a mesh element with a cubic shape. *“SOLID185 is used for 3-D modeling of solid structures. It is defined by eight nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. The element has plasticity, hyperelasticity, stress stiffening, creep, large deflection, and large capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elastoplastic materials, and fully incompressible hyperelastic materials.”* [26].

Solid 185 element types are shown in **Figure 3.4**.

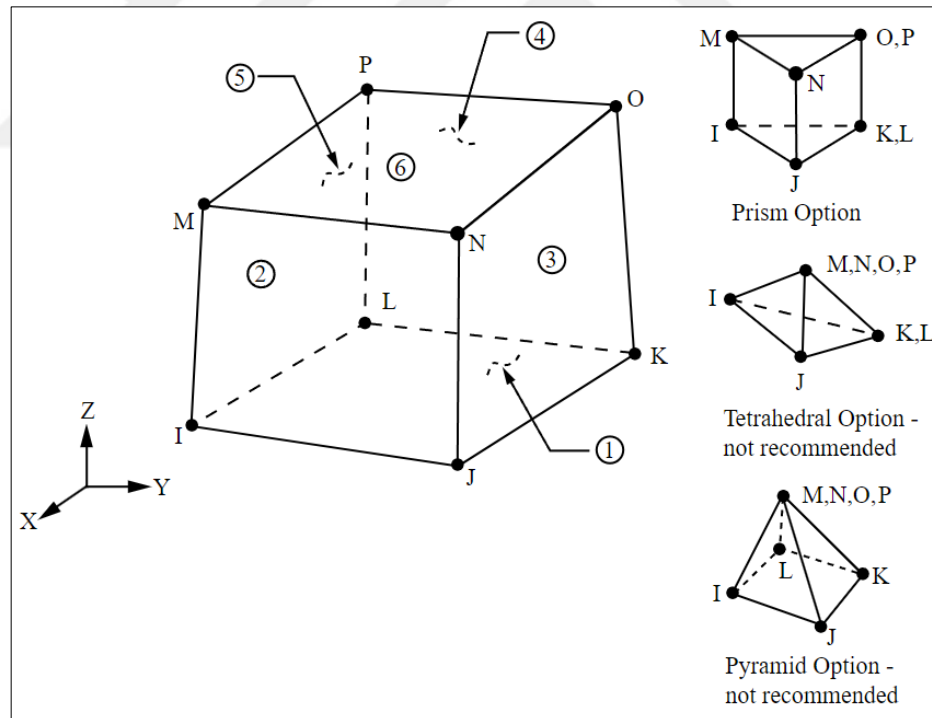


Figure 3.3: Solid 185 Homogenous Structural Element [26].

Meshed model shown in **Figure 3.4**. 453518 Nodes and 339259 Elements were used.

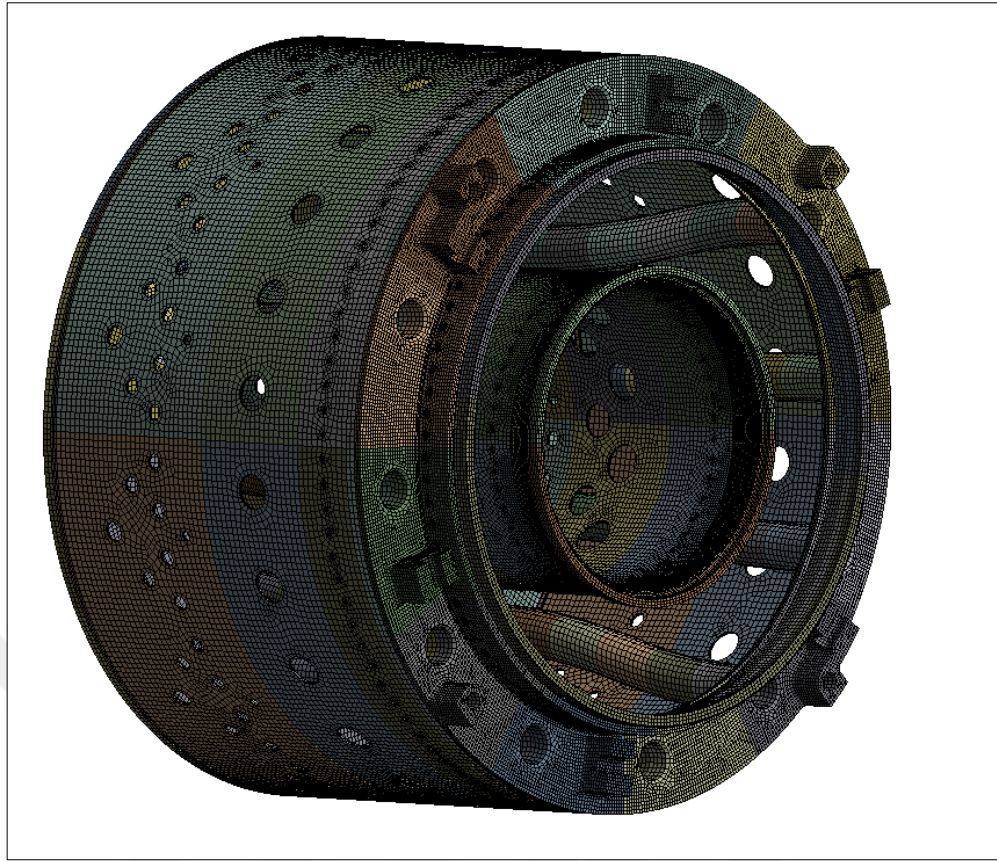


Figure 3.4: Mesh Image of Flame Tube Geometry.

3.3.3 Boundary Conditions

3D Flame Tube thermal outputs and CFD (Computational Fluid Dynamics) analysis outputs are imported into structural analysis. Bonded contact is specified for weld regions. Cut boundaries are defined at mechanical interfaces. To simulate real conditions, interface components were modeled. Cut boundaries obtained from whole engine analysis were taken from the cut surfaces. Frictional interfaces are defined as frictional contact. Tight fit areas are defined as interference fit contact.

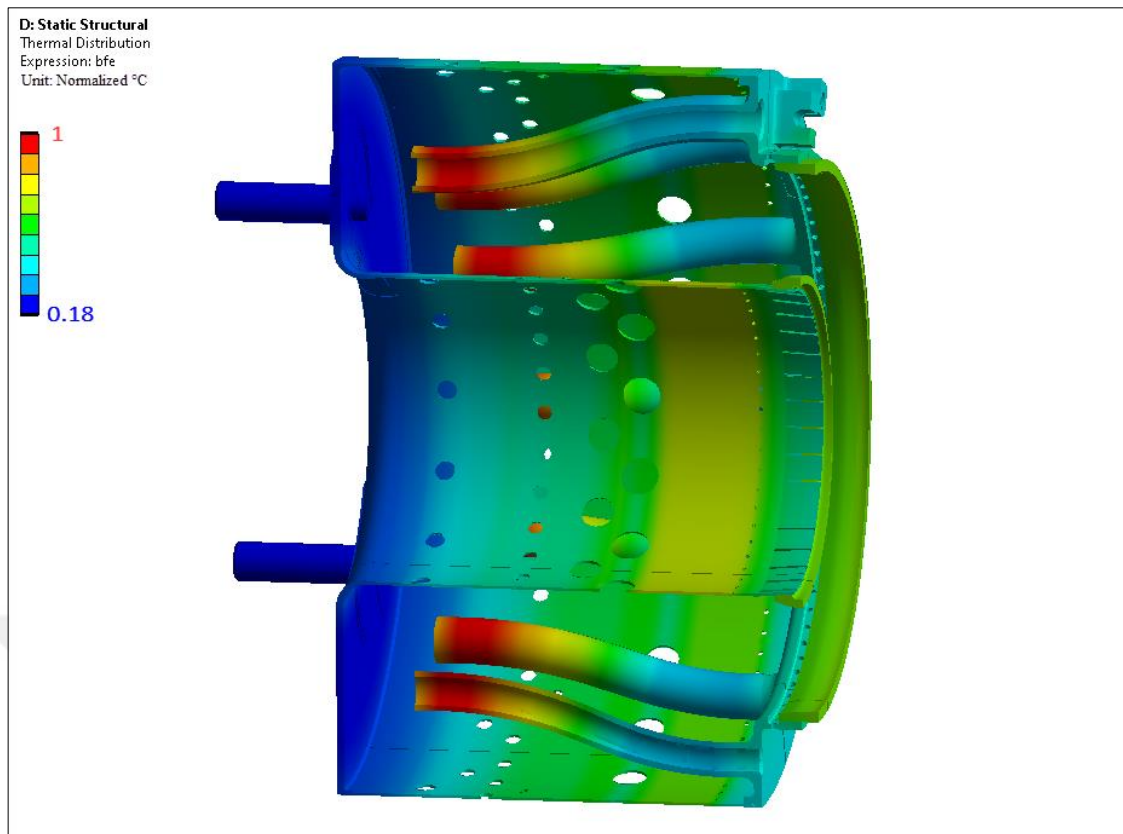


Figure 3.5: Thermal Analysis Input (Normalized).

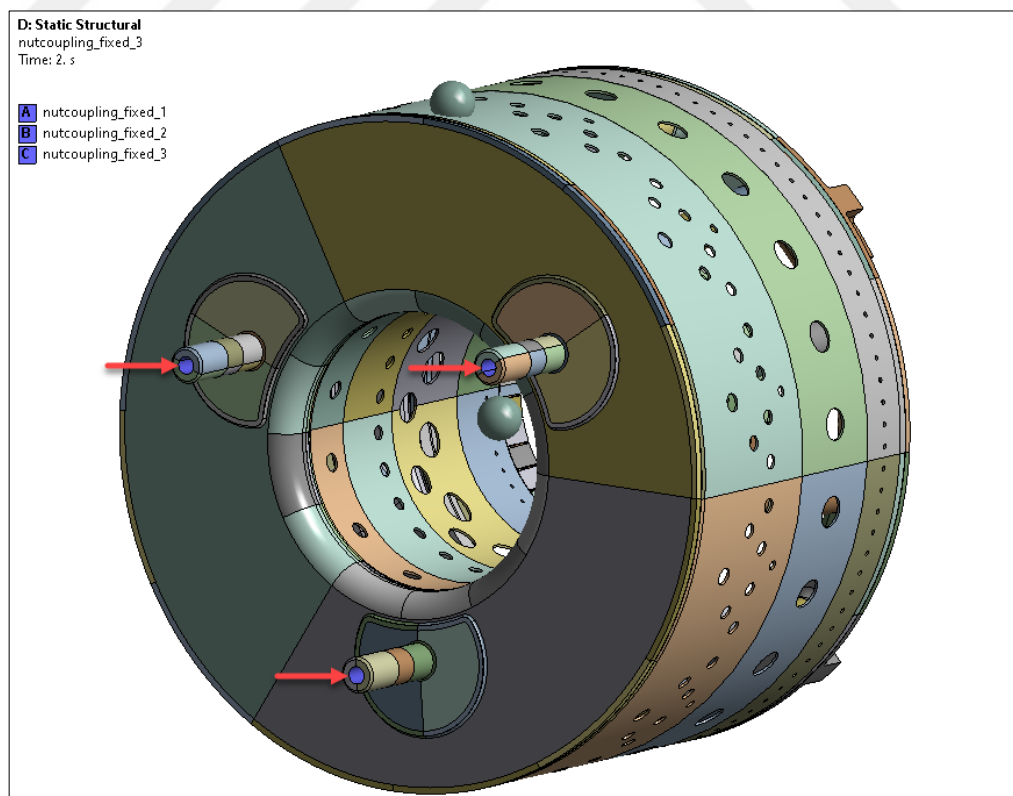


Figure 3.6: Fixed Supports on Mechanical Interface.

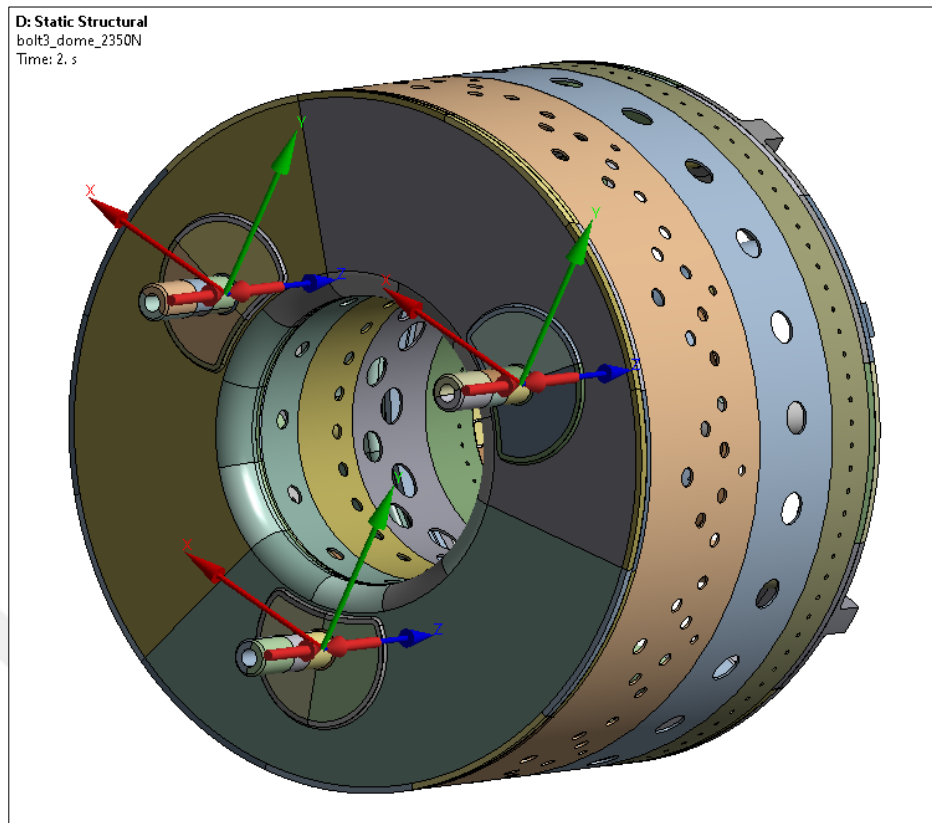


Figure 3.7: Bolt Pretensions on Nut Coupling-Flame Tube Interface.

As a result of the analysis made by the aerodynamic team, it was seen that there was not much pressure change along all surfaces. For this reason, average pressure values were applied to all flowpath surfaces **Figure 3.8**.

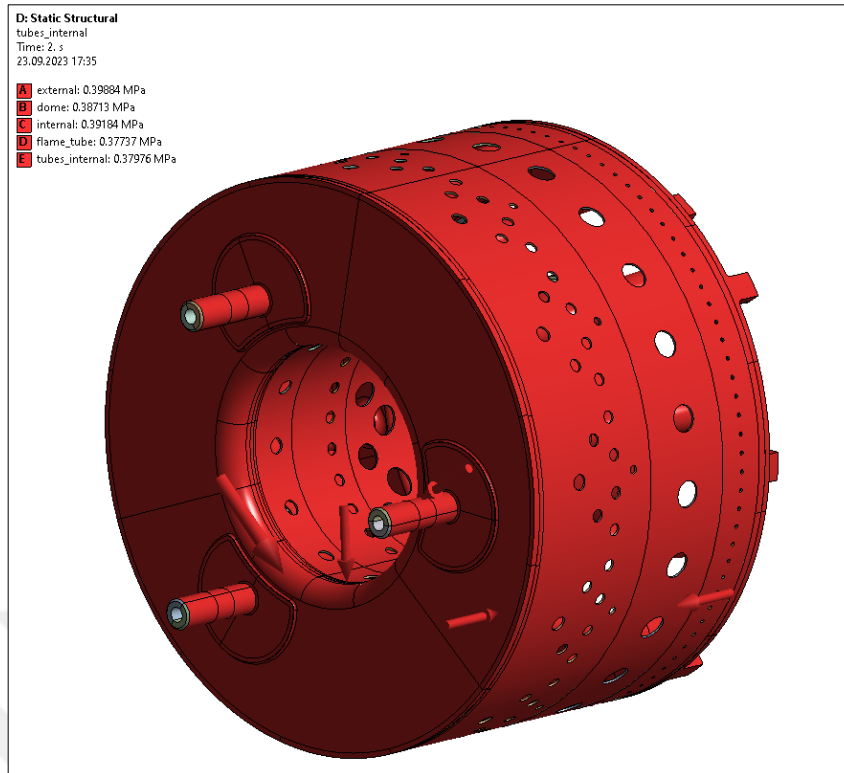


Figure 3.8: Pressure Definitions on Flame Tube Surfaces.

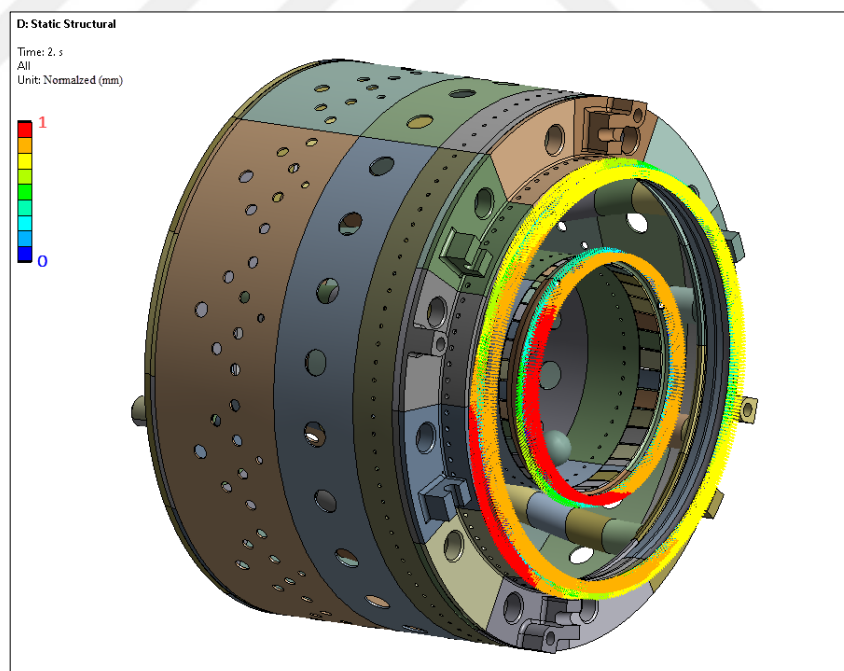


Figure 3.9: Cut Boundaries from Whole Engine Analysis (Normalized).

The parts in the Igniter interface and the Fuel Manifold parts are defined as point mass with deformable behaviour to the relevant Flame Tube interfaces.

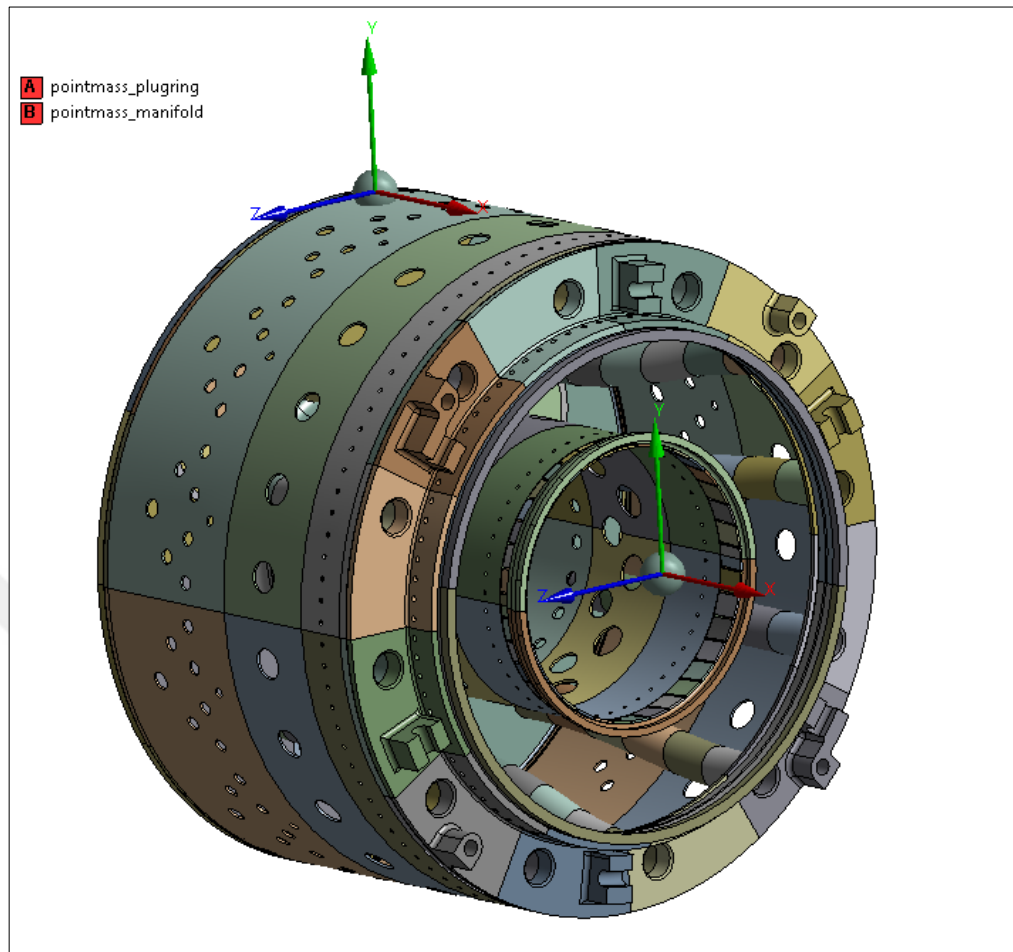


Figure 3.10: Point Mass Definition on Flame Tube.

3.4 Static Structural Analysis Results

This section delves into the examination of the results from the static model within ANSYS Workbench. Modal analysis and Harmonic analysis will be continued on this model. All results shown are normalized.

Total Deformation results are shown in **Figure 3.11**, **Figure 3.12**, and **Figure 3.13**.

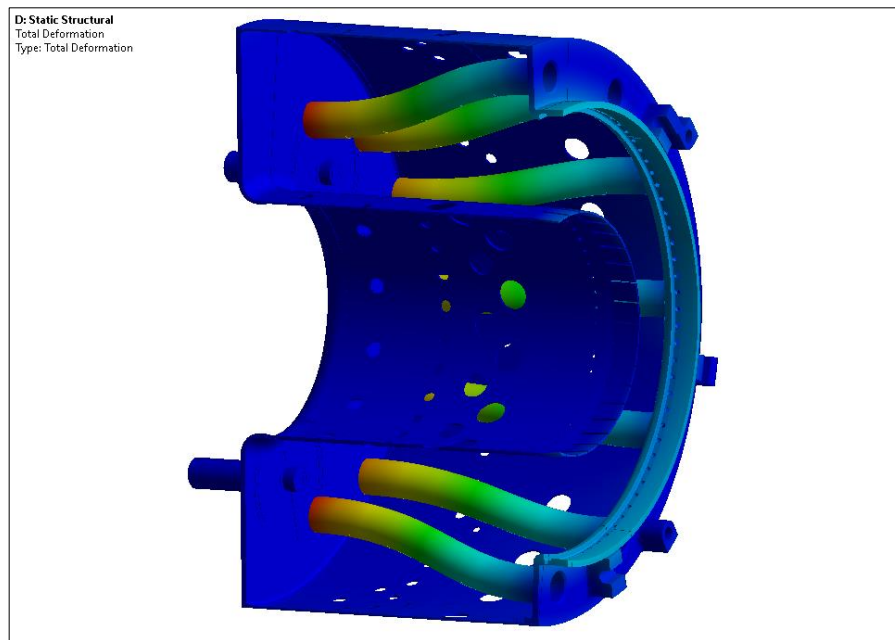


Figure 3.11: Total Deformation Plot (Cross-Sectional).

Table 3.2: Total Deformation Max & Min Results (Normalized)

ANSYS Workbench Static Structural		
Total Deformation	Max (normalized)	Min (normalized)
	1	-1,04

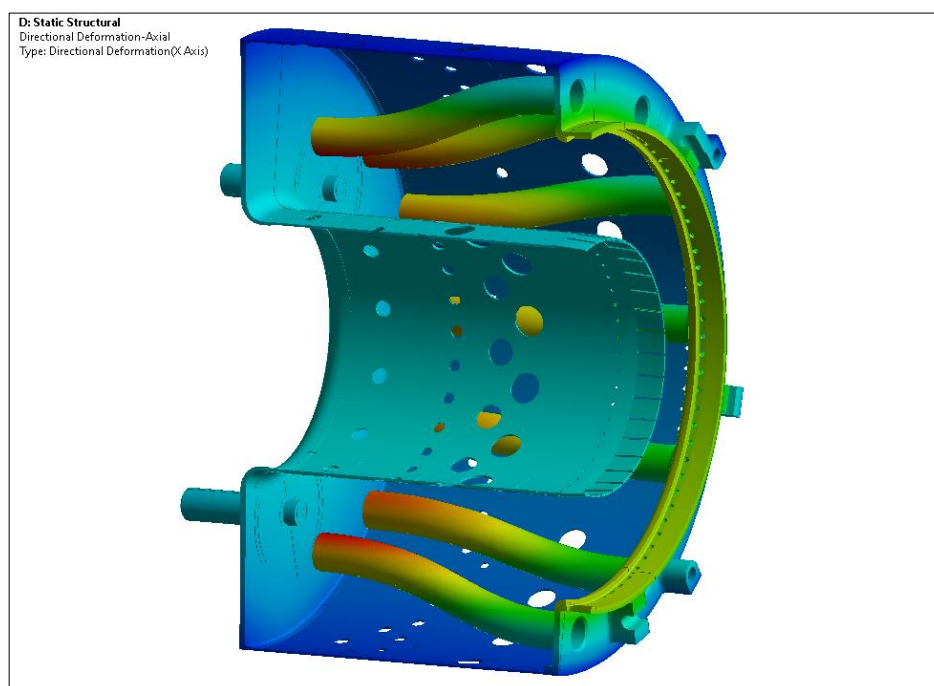


Figure 3.12: Directional Deformation Plot (Axial).

Table 3.3: Directional Deformation Axial Max & Min Results (Normalized)

ANSYS Workbench Static Structural		
Directional Deformation Axial	Max (normalized)	Min (normalized)
	0,35	-0,11

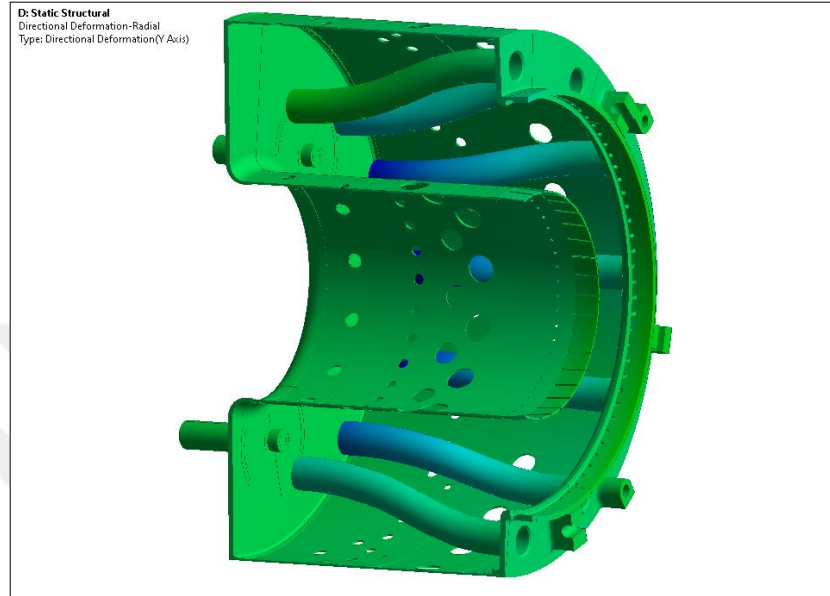


Figure 3.13: Directional Deformation Plot (Radial).

Table 3.4: Directional Deformation Radial Max & Min Results (Normalized)

ANSYS Workbench Static Structural		
Directional Deformation Axial	Max (normalized)	Min (normalized)
	1,09	-0,88

When the deformations are examined, it is seen that the vaporization tubes are more active than other regions due to their built-in structure.

In the continuation of the section, the stress distribution around vaporization tubes were investigated. The S_1 (Max. Principal Stress) and S_{eqv} (Equivalent Von-Mises Stress) distributions are shown in **Figure 3.14** and **Figure 3.15**.

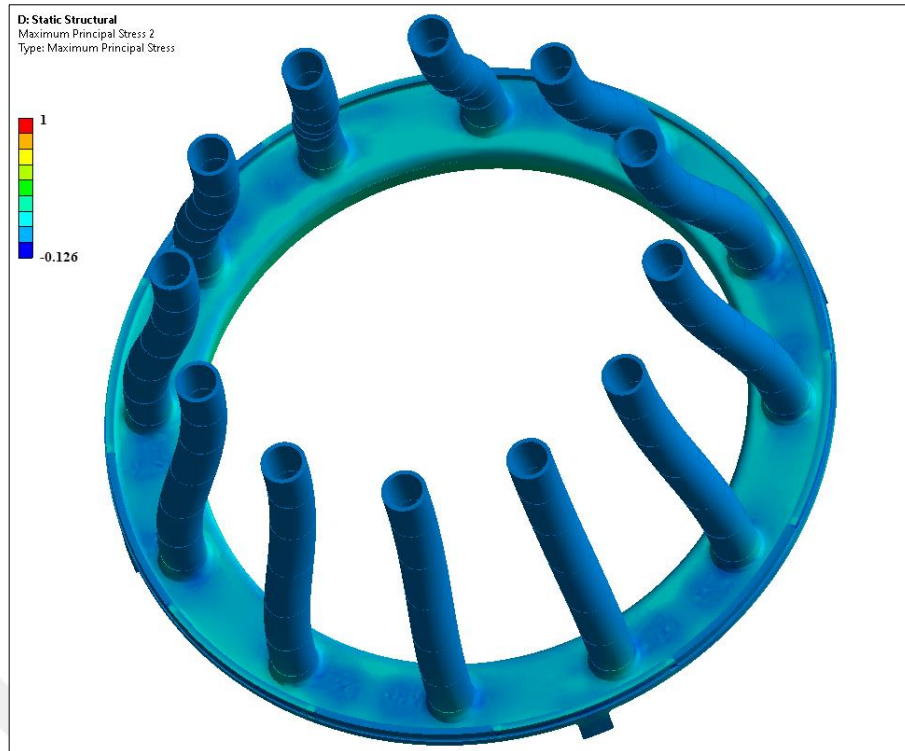


Figure 3.14: S1 (Max. Principal Stress) Distribution.

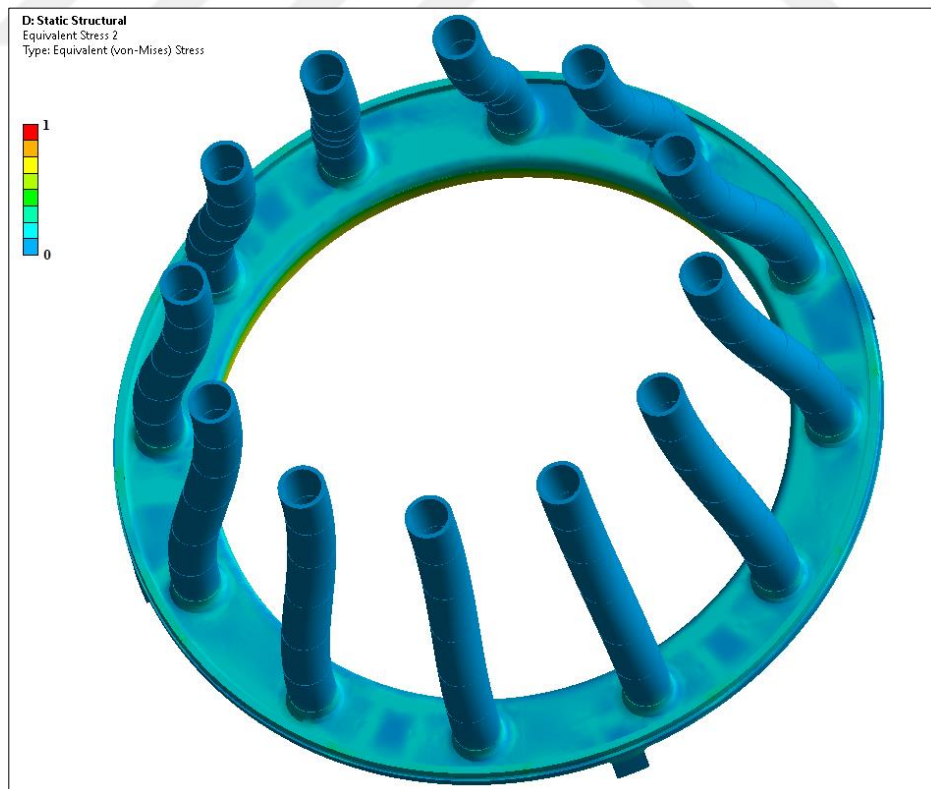


Figure 3.15: S_{eqv} (Equivalent Von-Mises Stress) Distribution.

3.5 Pre-Stressed Modal Analysis Results

Pre-stress modal analysis, also known as pre-stressed vibration analysis, is a technique used in structural engineering and mechanics to study the dynamic behavior of structures that have been subjected to pre-stress or initial forces before they experience any external loads such as interface loads, thermal loads and joints. This analysis helps in understanding how the presence of pre-stress affects the natural frequencies, mode shapes and overall dynamic character.

In traditional modal analysis, structures are assumed to be at free-free from any forces or stresses before analysis but in real life, many structures are often subjected to pre-stress during operation conditions.

In the following section, the results of the pre-stressed modal analysis conducted on the model, which underwent static analysis in the previous section.

Definition			
Pre-Stress Environment	Static Structural		
Pre-Stress Define By	Load Step		
Pre-Stress Loadstep	Last		
Reported Loadstep	Last		
Reported Substep	Last		
Reported Time	End Time		
Contact Status	Use True Status		
		Options	
		Max Modes to Find	200
		Limit Search to Range	Yes
		Range Minimum	0 (Normalized Frequency)
		Range Maximum	1 (Normalized Frequency)
		Spin Softening	Yes

Figure 3.16: Modal Analysis Options.

General mode shapes of vaporization tubes are shown in **Figure 3.17**.

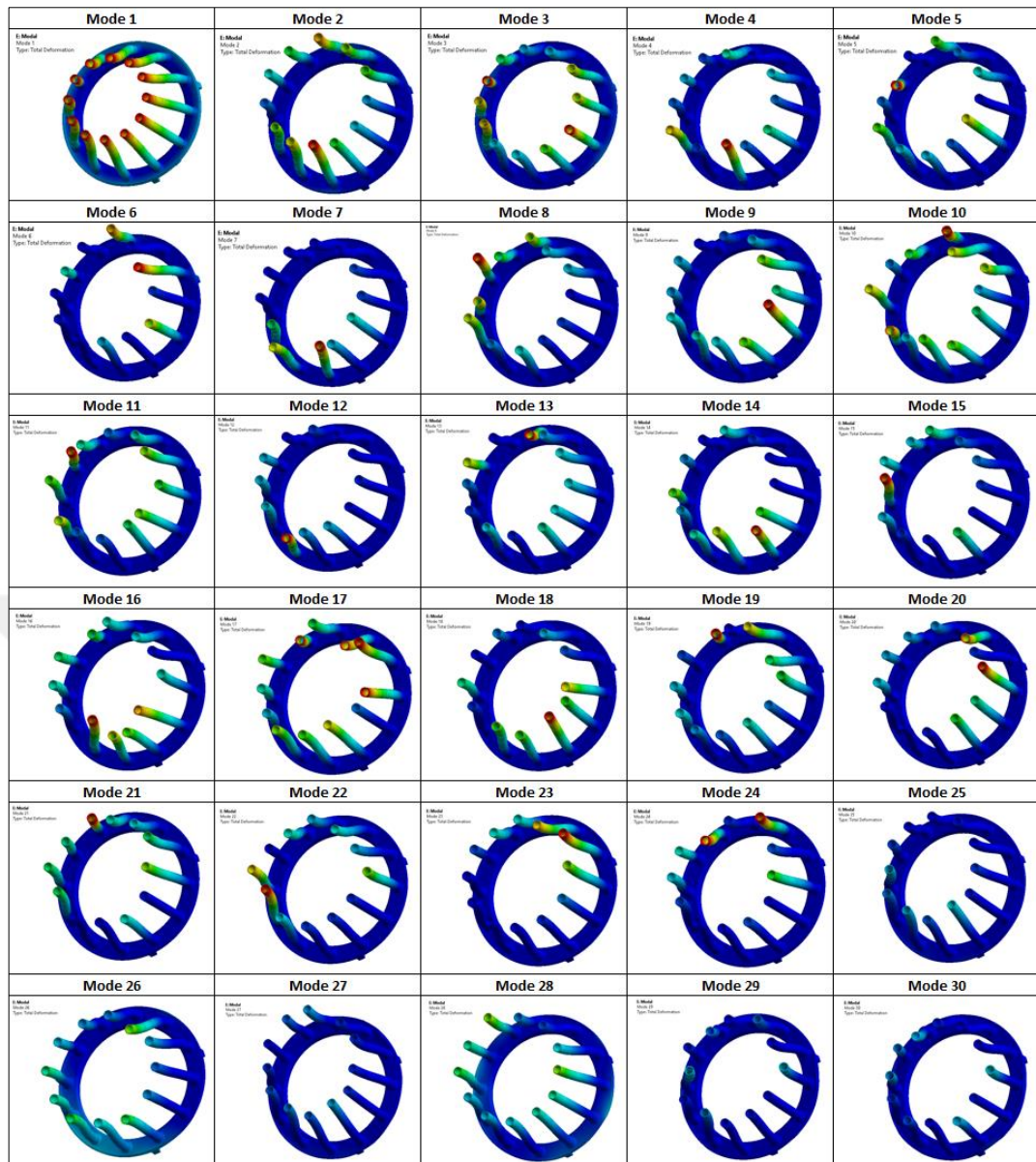


Figure 3.17: General Mode Shapes of Vaporization Tubes.

3.6 Campbell Diagram

The Campbell diagram provides a comprehensive or bird's-eye view of the potential vibration excitation in an operating system. It can be created based on machine design criteria or operational data. The X-axis represents the engine rotational speed, while the Y-axis represents the system frequency. The fan lines correspond to engine-order lines [25].

It is important to filter out the identified modes before creating the Campbell diagram. Not all of detected 33 resonance points are critical. The modes to be taken into consideration for evaluation should be determined based on the “Participation Factor (PF)” and the “Ratio of Effective Mass to Total Mass (RoEM)”. In this study, modes with a “Participation Factor” value above 10% or a “Ratio of Effective Mass to Total Mass” value above 2,5% have been included in the diagram.

Table 3.5: Eliminated Mode Chart (Normalized)

Mode	Normalized Frequency	PF	RoEM
2	0.3966	9.27	0.2130
3	0.3975	9.17	0.2085
8	0.5200	3.01	0.0226
9	0.5230	2.68	0.0172
10	0.5305	1.44	0.0051
13	0.5554	2.21	0.0088
15	0.5578	1.29	0.0067
16	0.5606	1.87	0.0077
17	0.5618	1.21	0.0028
18	0.5635	1.32	0.0043
20	0.5701	3.76	0.0340
21	0.5735	4.89	0.0473
22	0.5743	3.09	0.0378
23	0.5795	4.40	0.0369
24	0.5803	2.30	0.0586
25	0.7790	0.37	0.2110
26	0.8684	3.27	0.0203
27	0.9028	6.77	0.1200
28	0.9135	7.37	0.1024
29	0.9190	5.14	0.0528
30	0.9267	9.20	0.2189
0-1 Normalized Frequency Range		PF	> %10
		RoEM	> %2,5

The Campbell diagram created based on the eliminated modes are shown in **Figure 3.18**.

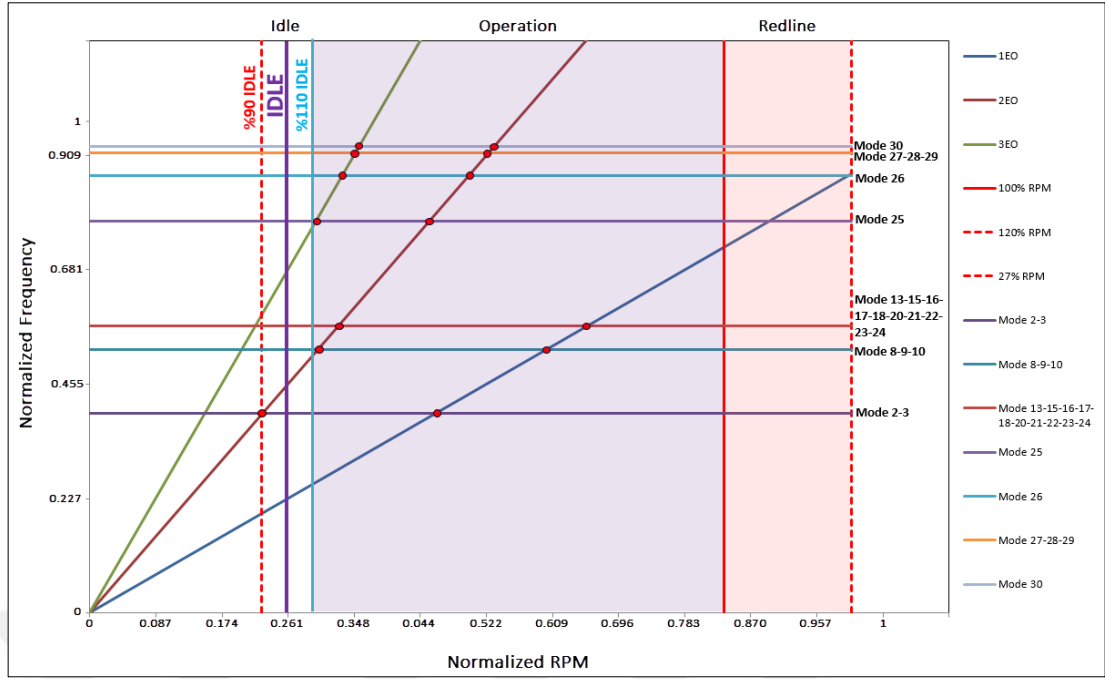


Figure 3.18: Campbell Diagram.

The 1st, 2nd and 3rd engine orders have been examined on the diagram. In terms of HCF, the most critical and prioritized modes to be evaluated are those that cross with the 1st engine order line. In the diagram, certain modes are depicted as a single line horizontally. The reason for this is that their frequencies are very close to each other, and they possess similar mode shapes but different phases. In essence, these can be considered as the same modes.

3.7 Harmonic Analysis and Results

The primary objective of performing harmonic analysis through the utilization of the Finite Element Method is to examine how a structure or system behaves when subjected to different harmonic loads or vibrations. This analysis seeks to ascertain how the system's elements react to harmonic stimulation and to pinpoint the patterns of forces, displacements and stresses that emerge due to these harmonic influences.

In the following section, modes that coincide with the 1st Engine Order are referred to as X1, X2 and X3, and they have been individually examined through harmonic analyses.

- The RPM value is normalized according to the max RPM value of the engine.

- The frequency value is normalized according to the maximum value of the frequency range in which the modal analysis is performed.
- All values plotted as MPa were normalized according to the maximum value for which the Goodman diagram of the material was created.

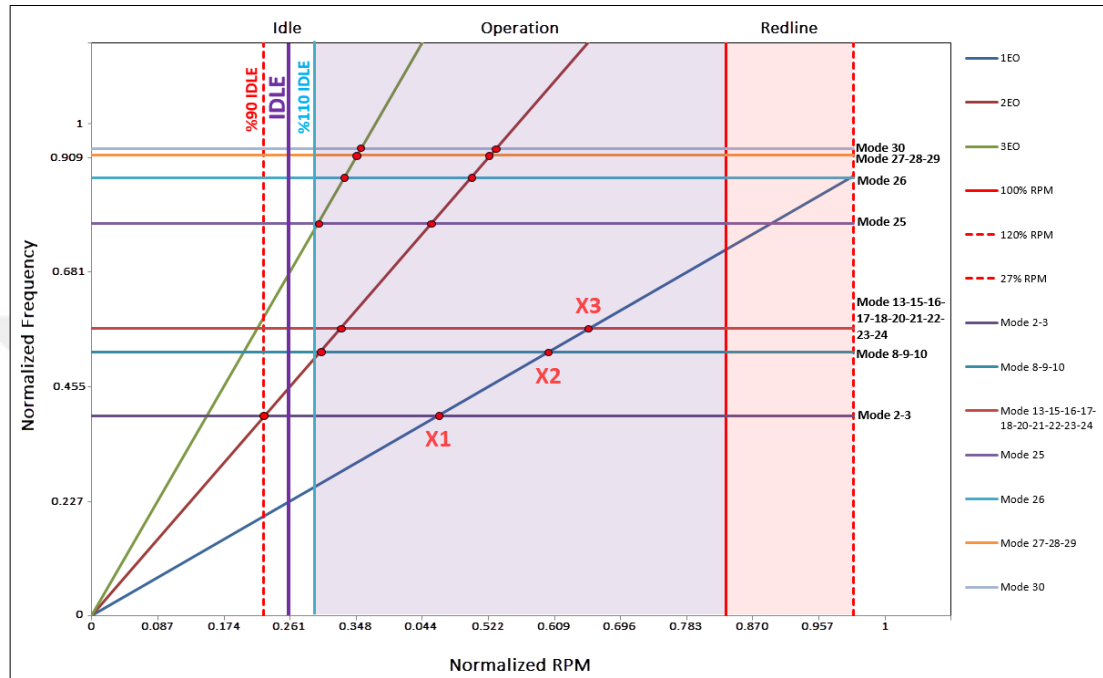


Figure 3.19: Harmonic Analysis Points.

Damping Controls		Definition	
Eqv. Damping Ratio From Modal	Yes	Base Excitation	No
☒ Damping Ratio	1,e-002	Define By	Components
Stiffness Coefficient Define By	Direct Input	Coordinate System	Global Coordinate System
☐ Stiffness Coefficient	0,	☐ X Component	0, mm/s ²
☐ Mass Coefficient	0,	☐ Y Component	98067 mm/s ²
		☐ Z Component	0, mm/s ²
		Suppressed	No

Figure 3.20: Analysis Settings.

- X1, Mode 2-3;

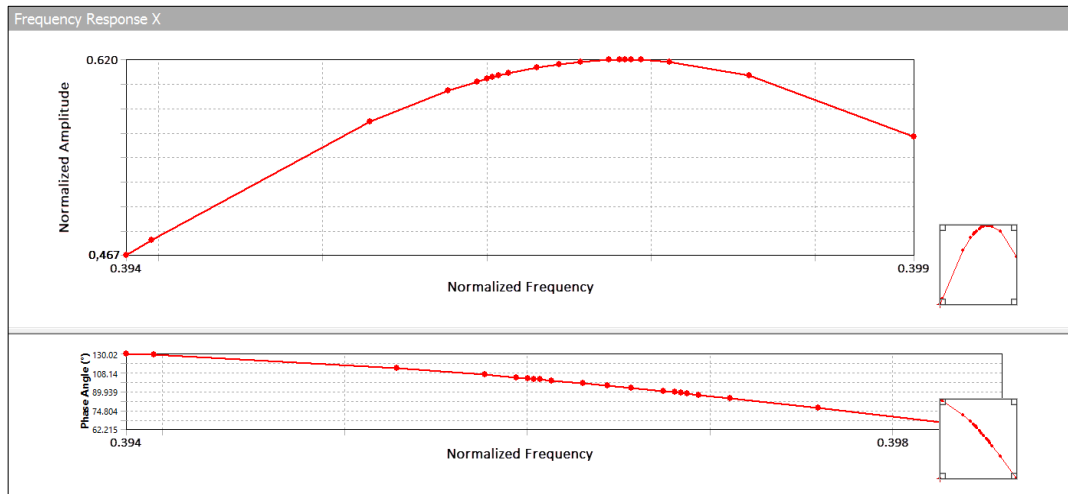


Figure 3.21: X1, Frequency Response X.

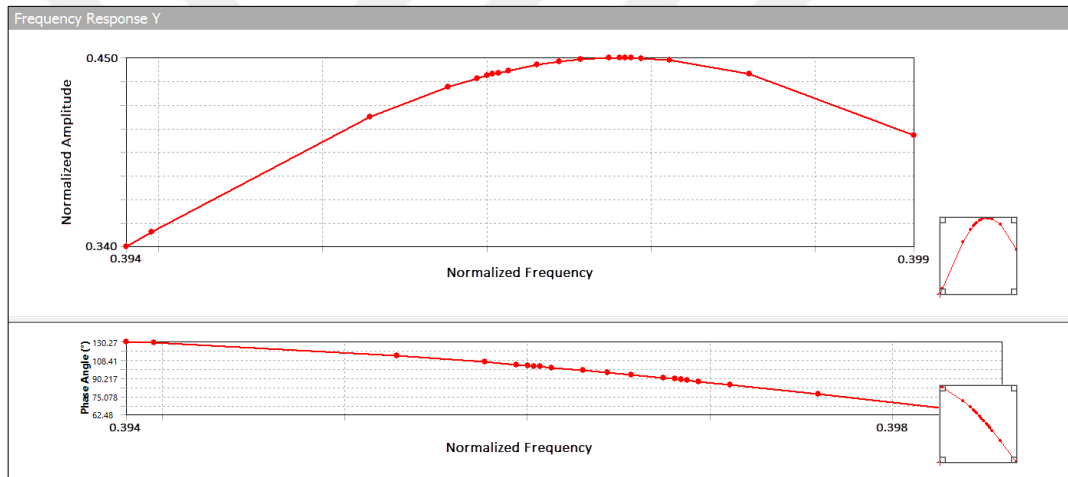


Figure 3.22: X1, Frequency Response Y.

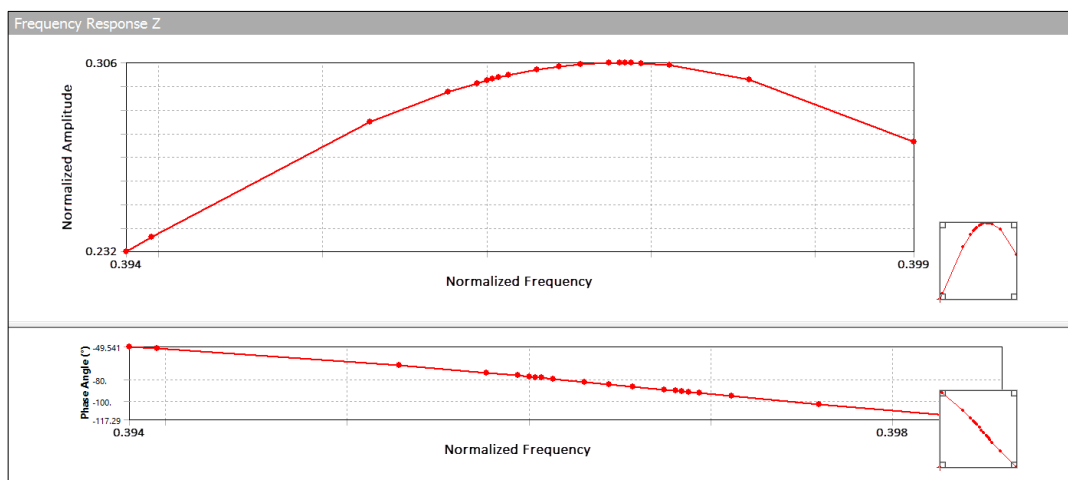


Figure 3.23: X1, Frequency Response Z.

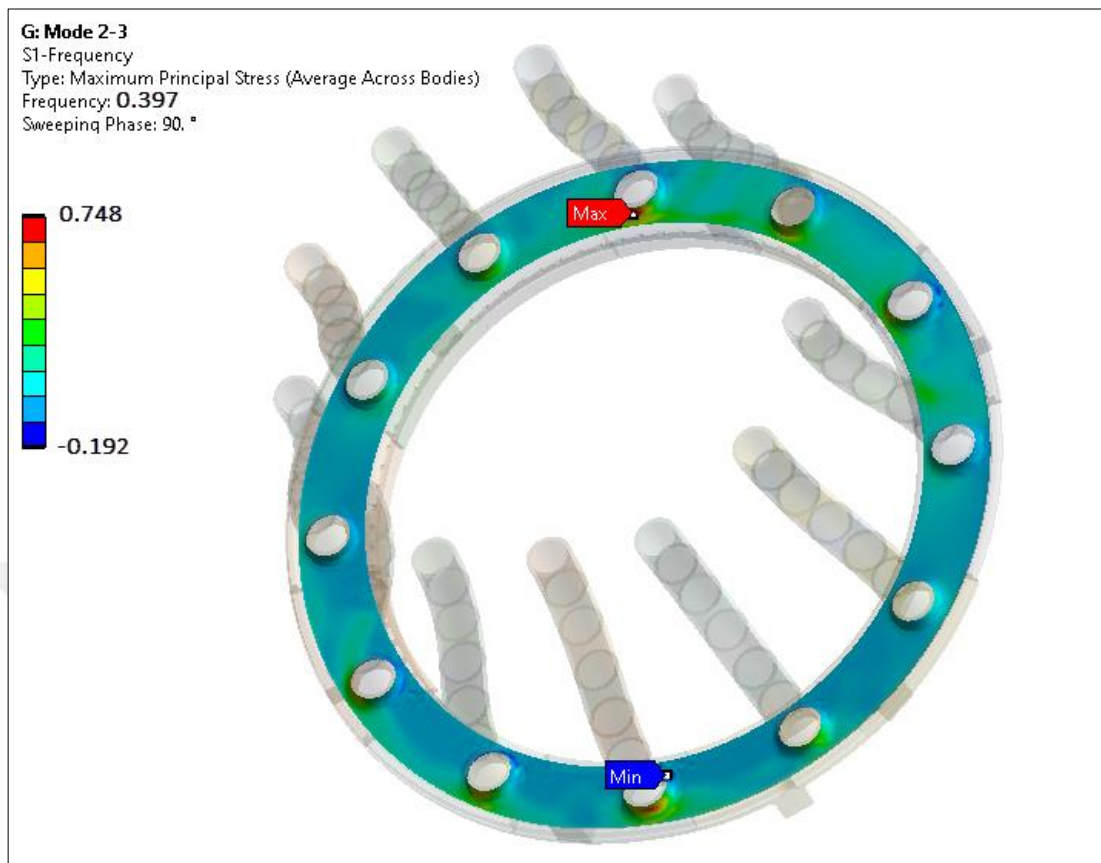


Figure 3.24: X1, Maximum Principal Strees Plot.

- **X2, Mode 8-9-10;**

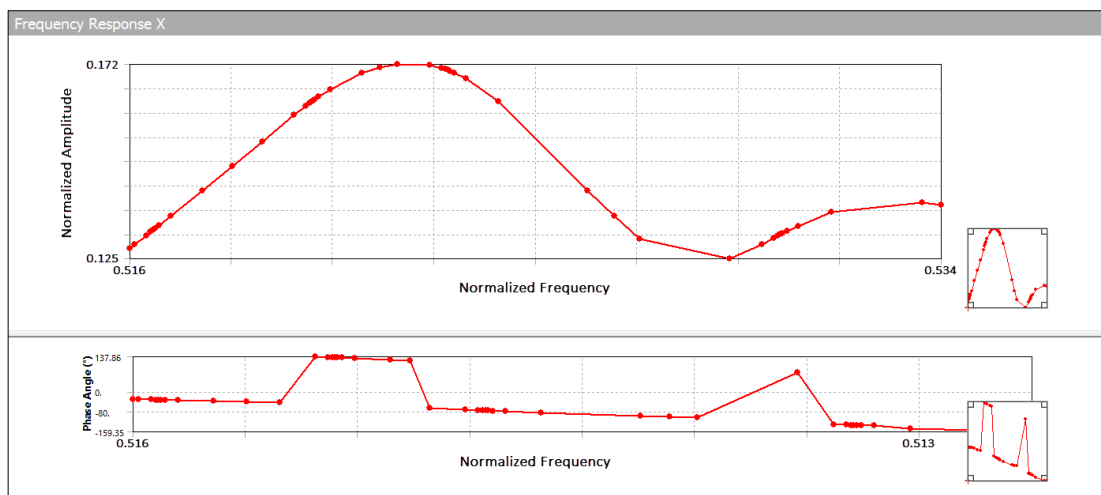


Figure 3.25: X2, Frequency Response X.

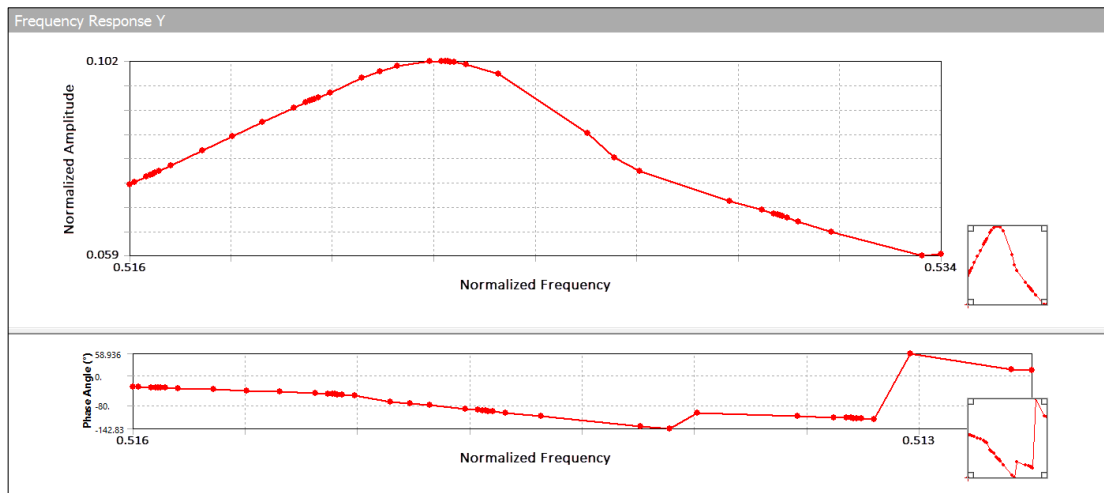


Figure 3.26: X2, Frequency Response Y.

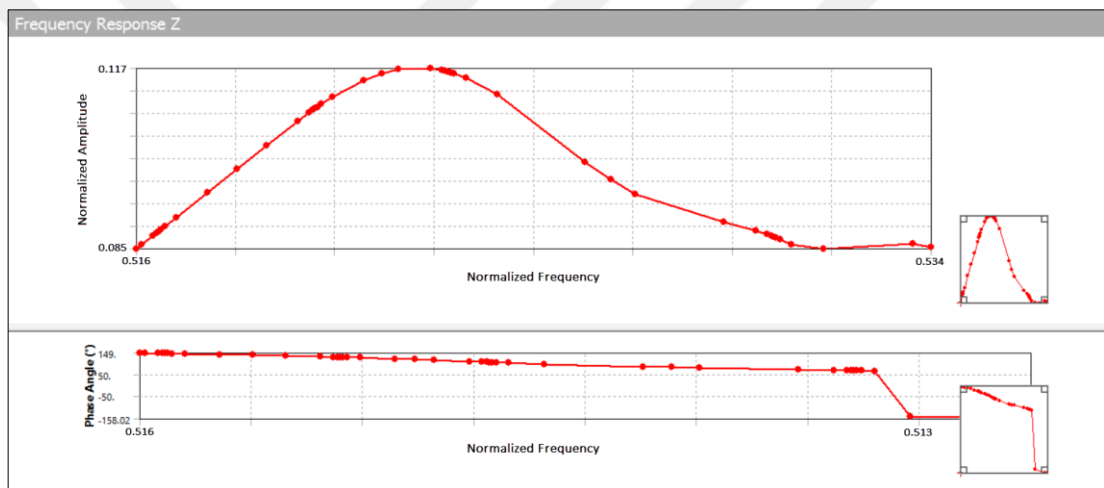


Figure 3.27: X2, Frequency Response Z.

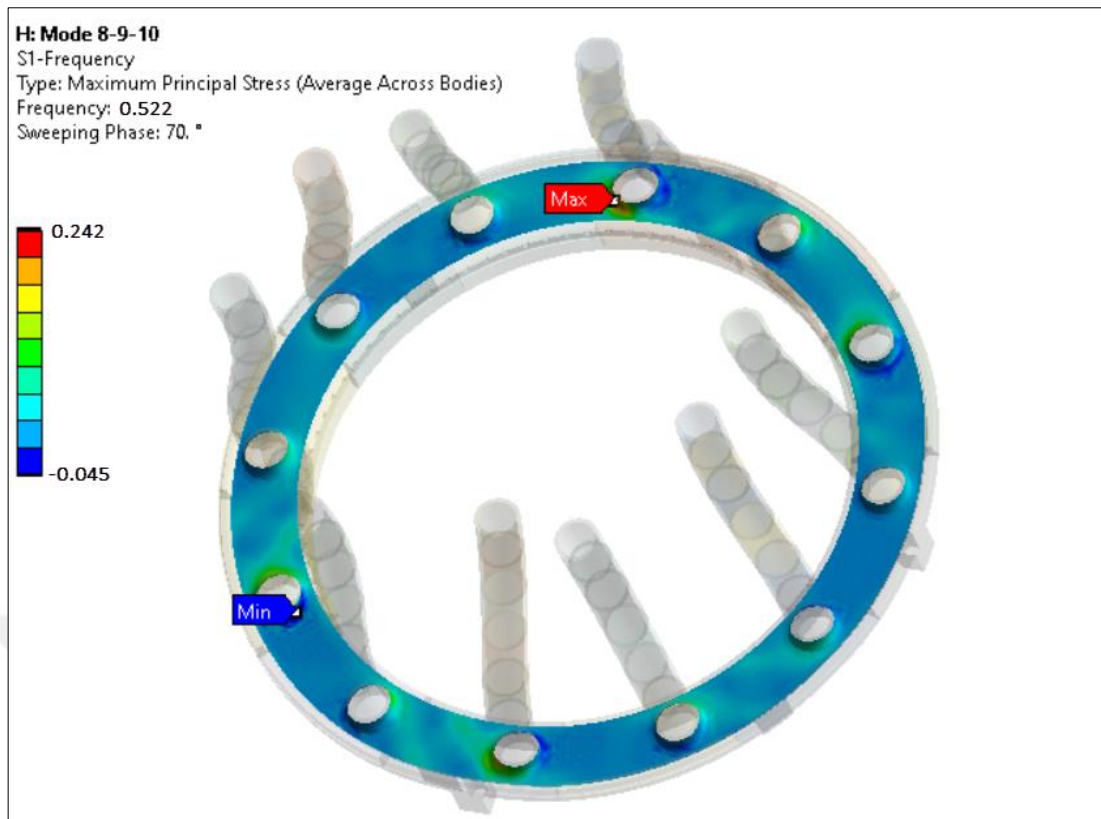


Figure 3.28: X2, Maximum Principal Strees Plot.

- **X3, Mode 13-15-16-17-18-20-21-22-23-24;**

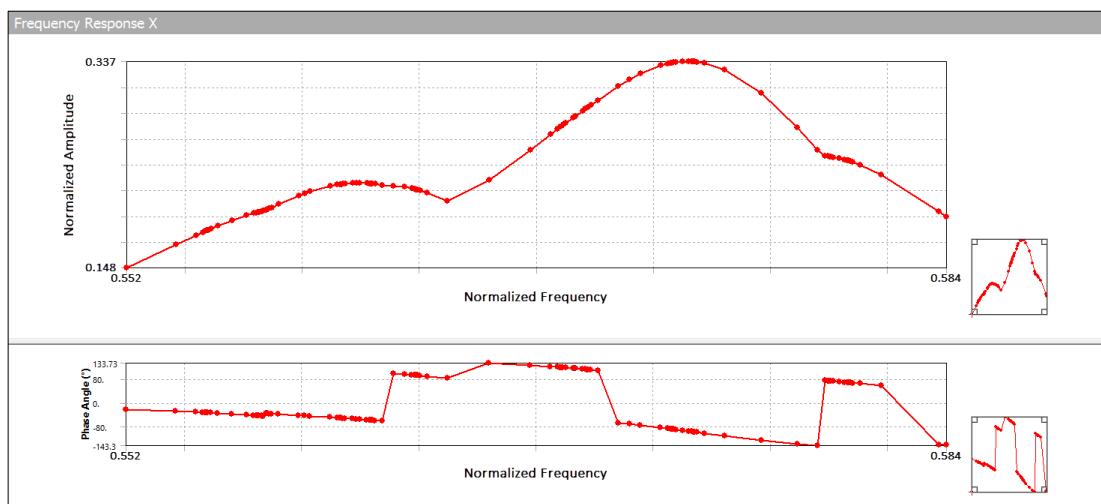


Figure 3.29: X3, Frequency Response X.

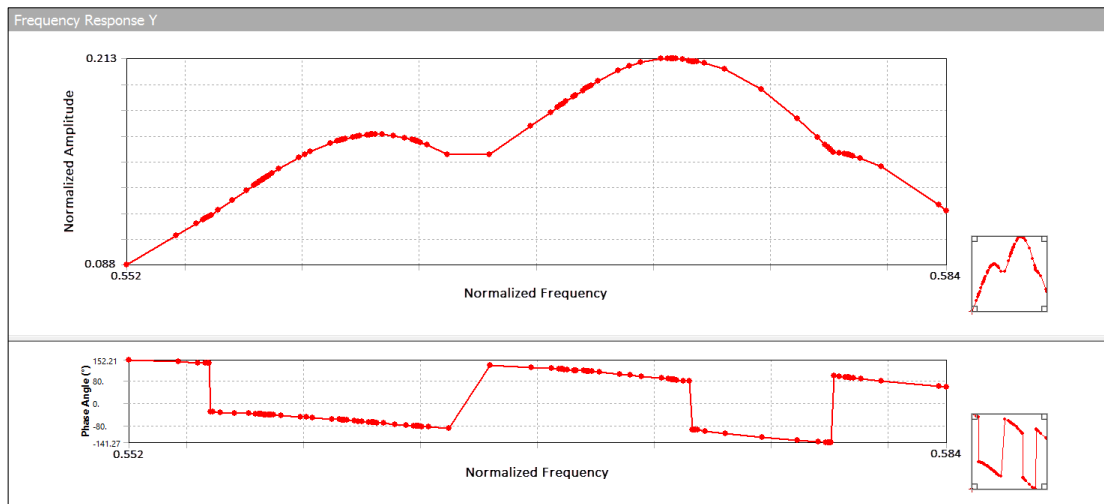


Figure 3.30: X3, Frequency Response Y.

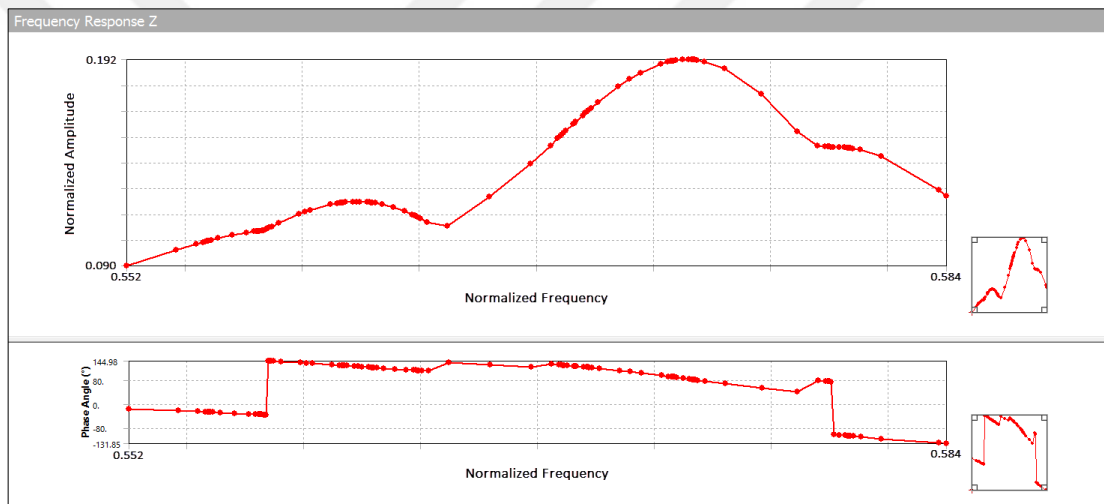


Figure 3.31: X3, Frequency Response Z.

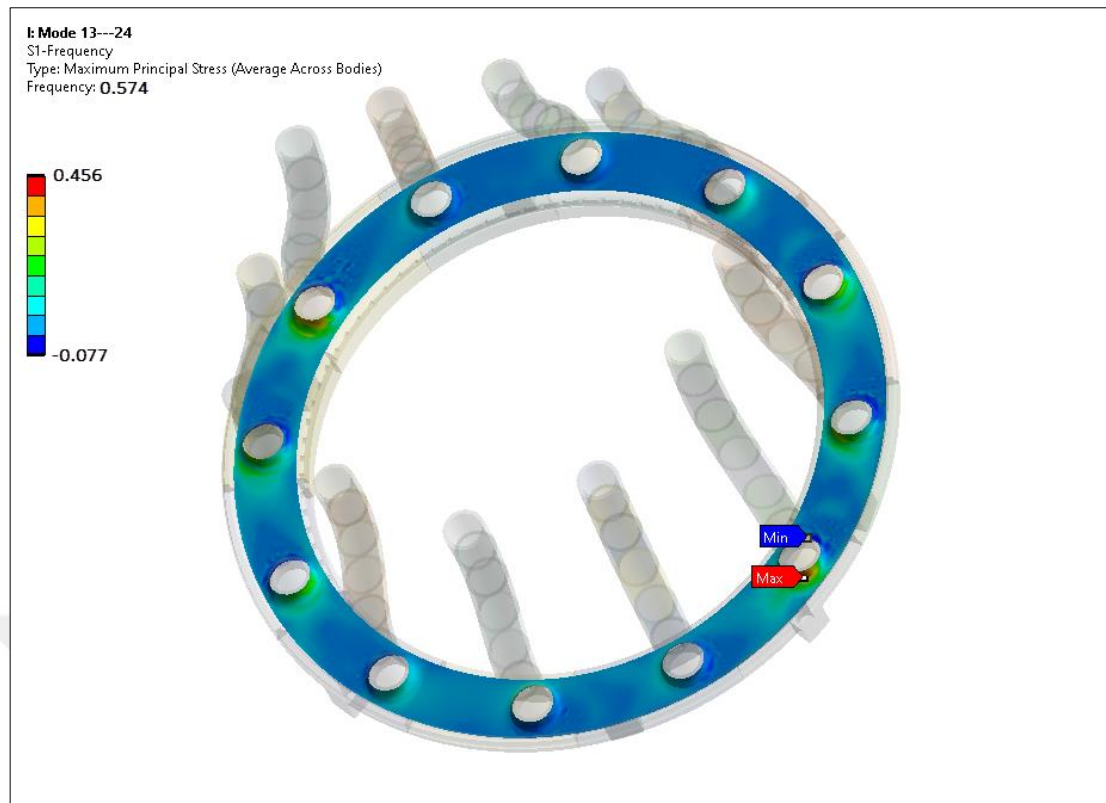


Figure 3.32: X3, Maximum Principal Strees Plot.

4. HCF TEST AND FLAME TUBE HAMMER IMPACT TEST

This chapter provides an understanding of the High Cycle Fatigue test setup and Hammer Impact test setup. The High Cycle Fatigue tests results are presented. On the other hand, Hammer Impact Test results also presented.

4.1 HCF Test

High cycle fatigue testing is a way to measure the durability of a material under repeated loading. By testing samples under different load or stress levels, engineers can create S-N curves that show how many cycles the material can withstand before it fails. This information can then be used to design components that are less likely to fail under fatigue loading.

- High Cycle Fatigue tests are typically performed in the range of 10^5 to 10^7 cycles.
- High Cycle Fatigue tests are more difficult than Low Cycle Fatigue test because they require more cycles.
- High Cycle Fatigue tests are more expensive than Low Cycle Fatigue tests because they are required more test material and more test equipment.

4.1.1 HCF Test Rig Setup and Procedure

In the domain of materials science and engineering, the High Cycle Fatigue test system holds a significant position. This methodological approach is pivotal in unravelling the fatigue characteristics of materials when subjected to repetitive cyclic loading.

In the continuation of the section, the High Cycle Fatigue test is examined under 4 main headings.

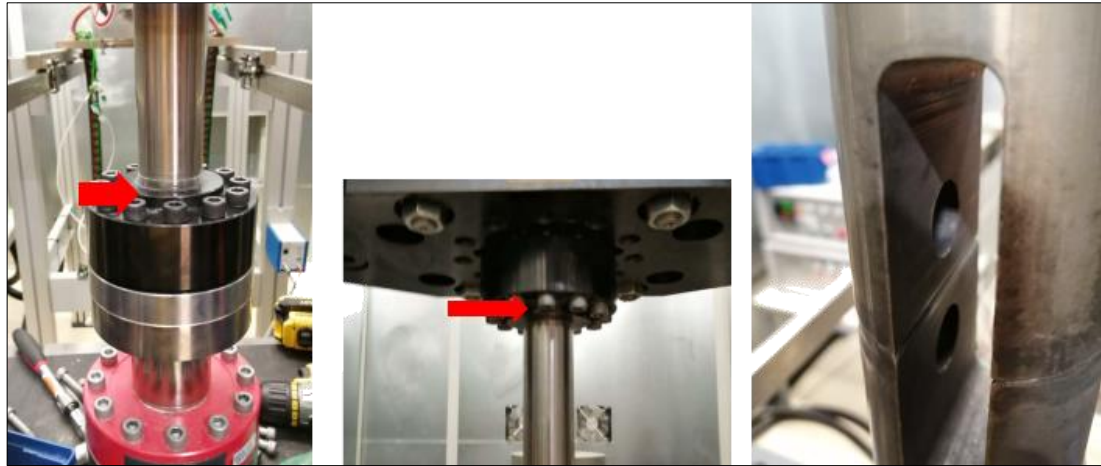


Figure 4.1: HCF Test Rig Setup.

Selection and Preparation of Test Specimens:

At the core of every HCF test system lies the careful selection and meticulous preparation of test specimens. The raw materials undergo a transformational process to shape them into specimens fit for testing. This metamorphosis involves intricate machining processes to ensure consistent geometry and surface quality. Rigorous measurements and dimensional evaluations are carried out to guarantee adherence to defined standards.

Quality inspection of specimens before testing shall include the following items:

- Visual Inspection
- Documentation Control
- Verification of Specimen Identification

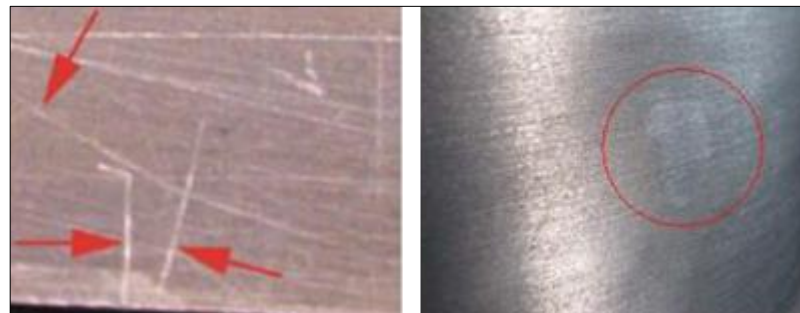


Figure 4.2: Surface Irregularities of Specimens.

Application of Load:

The essence of HCF testing revolves around the precise application of load. A dynamic loading system, intricately calibrated, orchestrates cyclic stresses that mimic real-world scenarios. The test specimen finds its place within a specially designed grip assembly. An oscillating actuator transmits forces through this assembly, subjecting the specimen to cyclic stress cycles. These stress levels resonate at specific frequencies, imitating the dynamic forces experienced during the material's actual service life.



Figure 4.3: Test Setup.

Monitoring and Control:

Maintaining a harmonious balance between monitoring and control mechanisms is paramount throughout the HCF testing process. Modern servo-hydraulic systems, enhanced by state-of-the-art software interfaces, regulate cyclic loading with unparalleled accuracy. Strain gauges, extensometers, and accelerometers act as vigilant observers, capturing the material's response to cyclic stresses. Real-time data acquisition constructs a detailed narrative of the material's performance, revealing its endurance capabilities and susceptibility to fatigue.



Figure 4.4: Temperature Control Unit and Screen.



Figure 4.5: Temperature Control Unit Configuration.

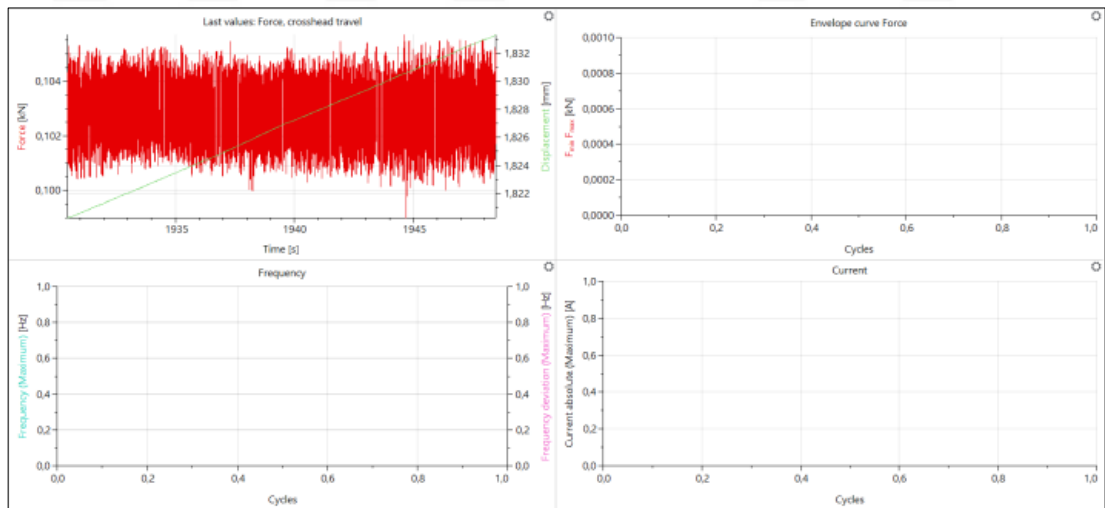


Figure 4.6: Run Test Screen / Test Monitoring.

Frequency and Stress Parameters:

Within this intricate ballet of materials and forces, frequency and stress parameters assume roles of significance. Frequencies are carefully chosen, often residing in the kilohertz range, mirroring the rapid oscillations encountered in practical applications.

Stress amplitudes, welding influence over material fatigue, are meticulously controlled to account for worst-case scenarios.

4.1.2 Results

In this section, S-N curve and Goodman Curve of Alloy 625 Additive material are shown. Relevant data was obtained in TEI materials laboratory. It has been normalized because it is confidential information belonging to the company.

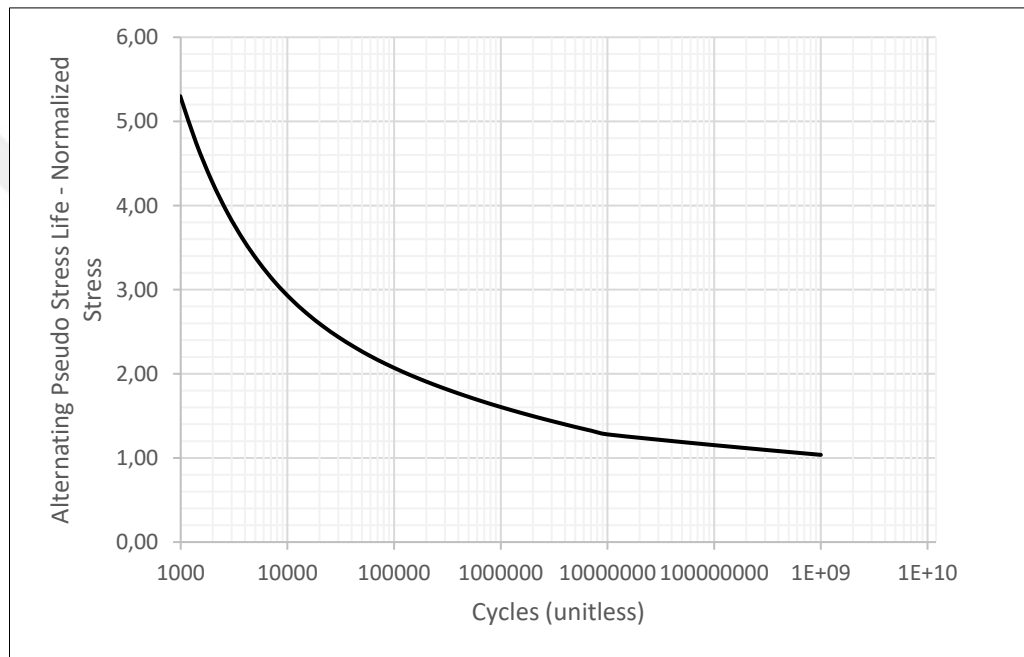


Figure 4.7: S-N Curve.

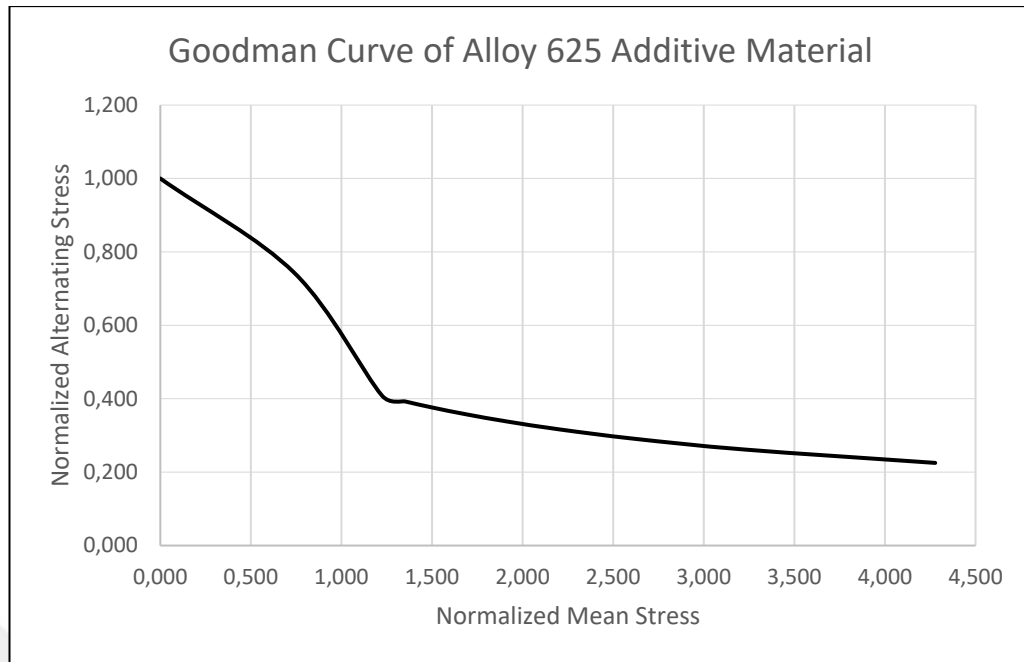


Figure 4.8: Goodman Curve.

4.2 Impact Hammer Test

An essential technique for evaluating the dynamic characteristics of structures and materials is the Impact Hammer test. It involves striking the material or structure with a tool like a hammer and observing the reaction to determine different mechanical properties.

4.2.1 Impact Hammer Test Setup and Procedure

The Impact Hammer testing procedure is described in below.

1. **Impact Hammer:** The Impact Hammer is the most important tool for this test setup. It is usually made up of a weight attached to a handle.
2. **Accelerometer:** A highly accurate accelerometer is attached to the material or structure at a specific location. This device measures the acceleration response during the impact.
3. **Data Acquisition System:** A powerful data collection and analysis system is needed to capture and study the acceleration data from accelerometer.

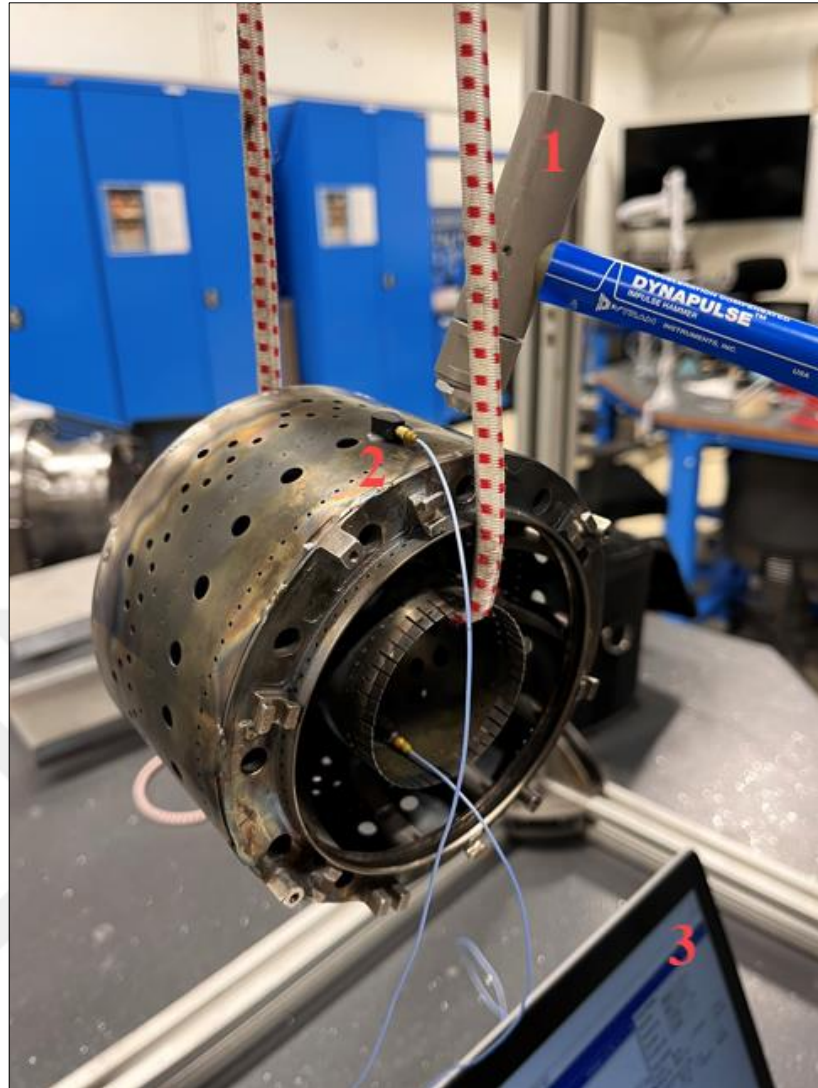


Figure 4.9: Impact Hammer, Accelerometer and Data Acquisition System.

The testing impact hammer modal test are as follows:

1. **Preparation:** The testing area was set up in a controlled environment to minimize external factors that could affect the measurement. It was ensured that the Flame Tube Assy was safely positioned and stable.
2. **Instrumentation:** The accelerometer was attached to specific points on the Flame Tube Assy using adhesive. The accelerometer was connected to the data acquisition system.
3. **Calibration:** Adjusted the Impact Hammer so that it strikes with a consistent force. Ensured that the accelerometer is properly configured and produces accurate readings.
4. **Impact:** The Impact Hammer was held vertically on the Flame Tube Assy at the designated impact point. The hammer was released to strike the Flame Tube, creating an impact load. The resulting acceleration data was recorded from the accelerometer to the data acquisition system.
5. **Repeat:** Multiple impact tests were performed at various locations on the Flame Tube Assy to obtain comprehensive data for analysis. Two separate setups were installed for the Dome and Liner parts. Accurate measurements were ensured throughout the testing process.

The test setup is shown in the **Figure 4.10**.

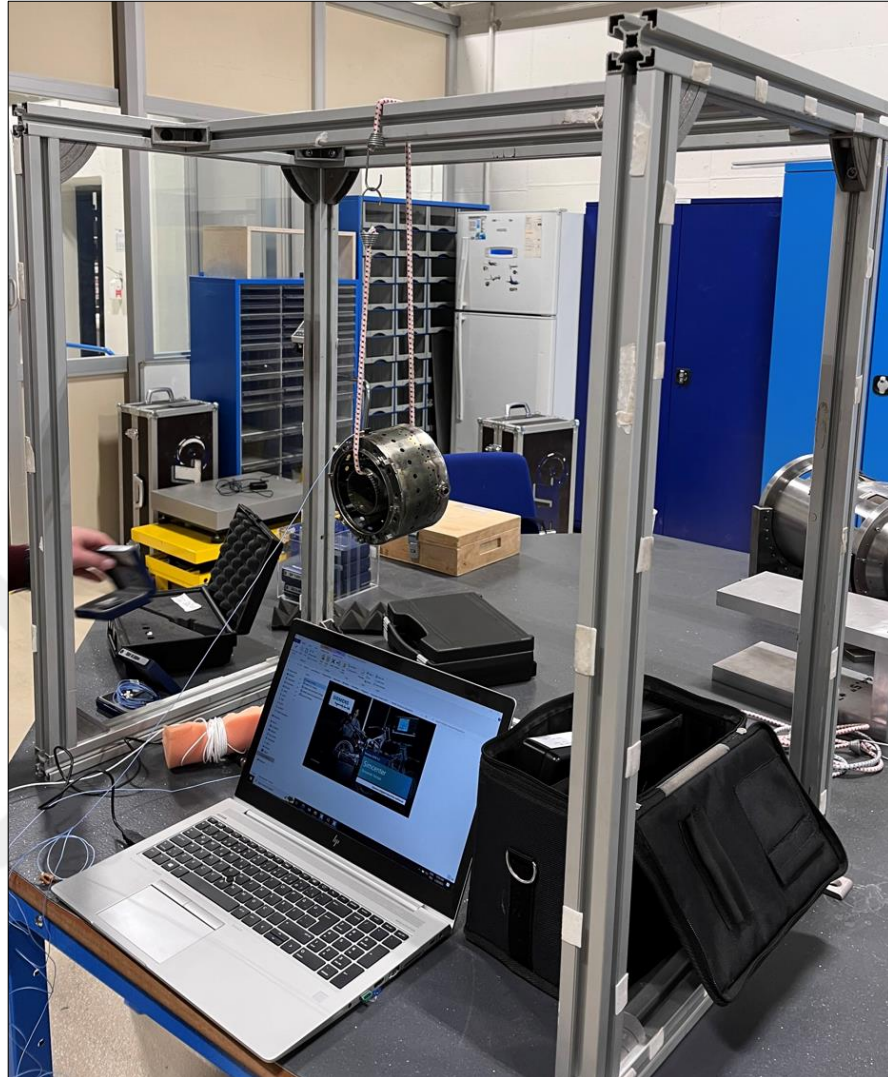


Figure 4.10: Impact Hammer Test Setup.

4.2.2 Impact Hammer Test Results

While performing the Impact Hammer test on the Flame Tube Assembly, data was collected from a hammer strike at a fixed point and from two points using accelerometers. In the first test, the strike was made on the Dome region, and data was collected from that region. In the second test, the strike was made on the Outer Liner, and data was collected from both the Inner and Outer Liners.

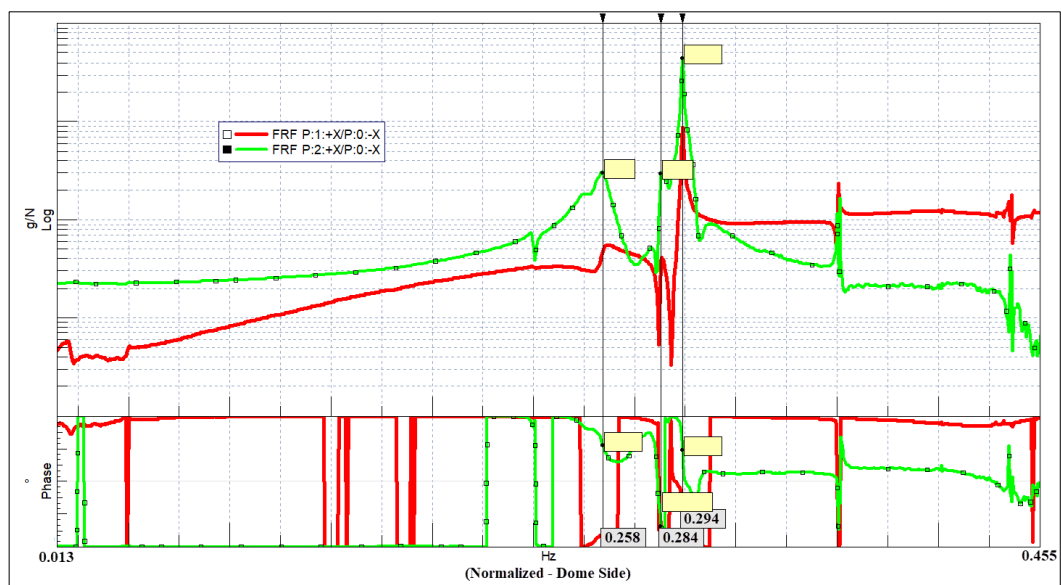


Figure 4.11: Test 1 (Dome).

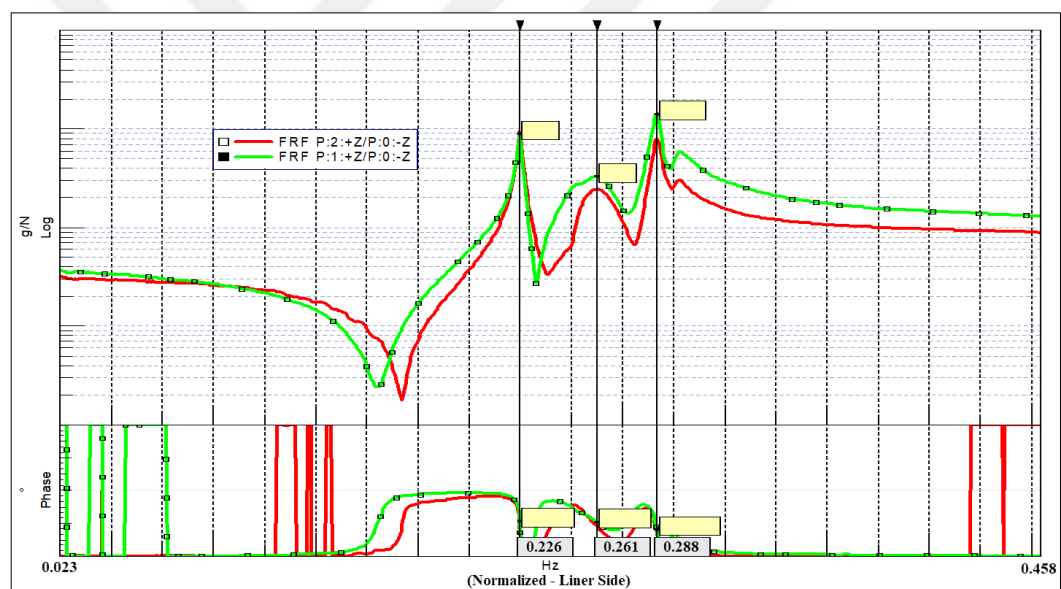


Figure 4.12: Test 2 (Liners).

5. VALIDATION AND RESULTS

In this section, for the validation of the modeled Flame Tube, the free-free modal analysis and impact hammer test results of the part were compared. In the continuation of the chapter, the harmonic analysis outputs obtained in the third chapter were used with the curves obtained from the high cycle fatigue test to calculate the high cycle fatigue life of the Flame Tube. At the end of the chapter, the results are summarized, suggestions are made, and the future works are described shortly.

5.1 Comparison of FEA Free-Free Modal Analysis Results and Hammer Impact Test Results

A free-free modal analysis model was established in order to compare the analysis model used for all calculations with the actual part produced. The modes detected by the hammer test were compared with the free-free modal analysis results. With this comparison, it is possible to get an idea about whether the analysis model simulates the real part or not.

Firstly, when the impact hammer test performed from the dome region (**Figure 5.1**) was examined, the expected mode with a frequency difference of 0.002-0.012 could be detected. It is shown in **Figure 5.2**.

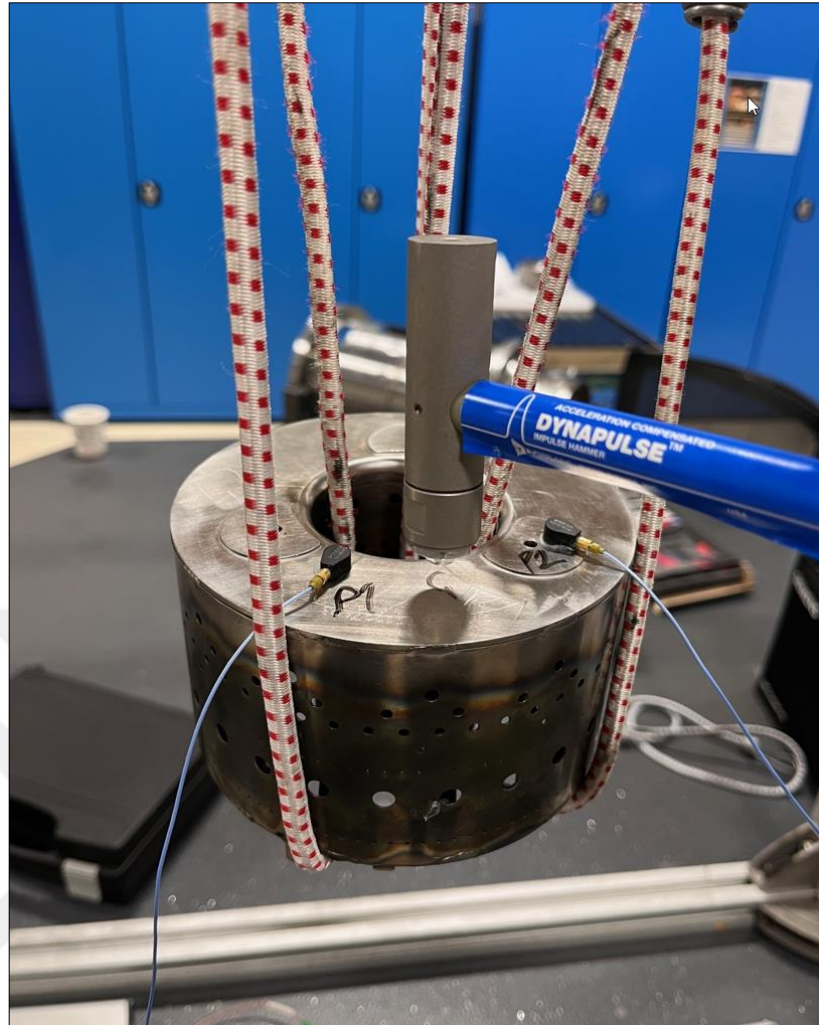


Figure 5.1: Test 1 – Impact from Dome Region.

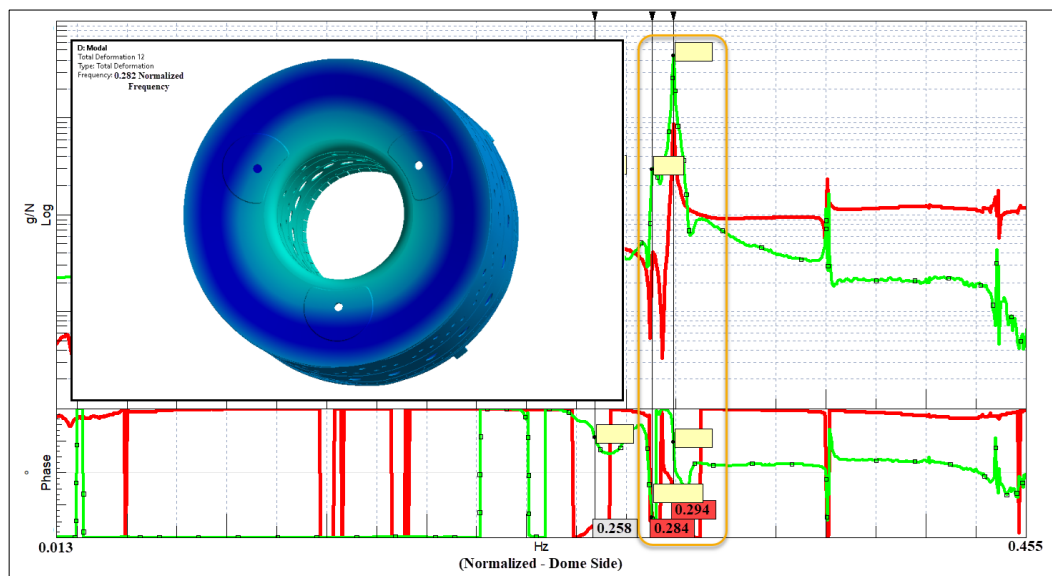


Figure 5.2: Test 1 – Expected Mode (Dome Region).

In continuation, when the impact hammer test performed from the liner region (Figure 5.3) was examined, the expected mode with a frequency difference of 0.001 could be detected. It is shown in **Figure 5.4**.

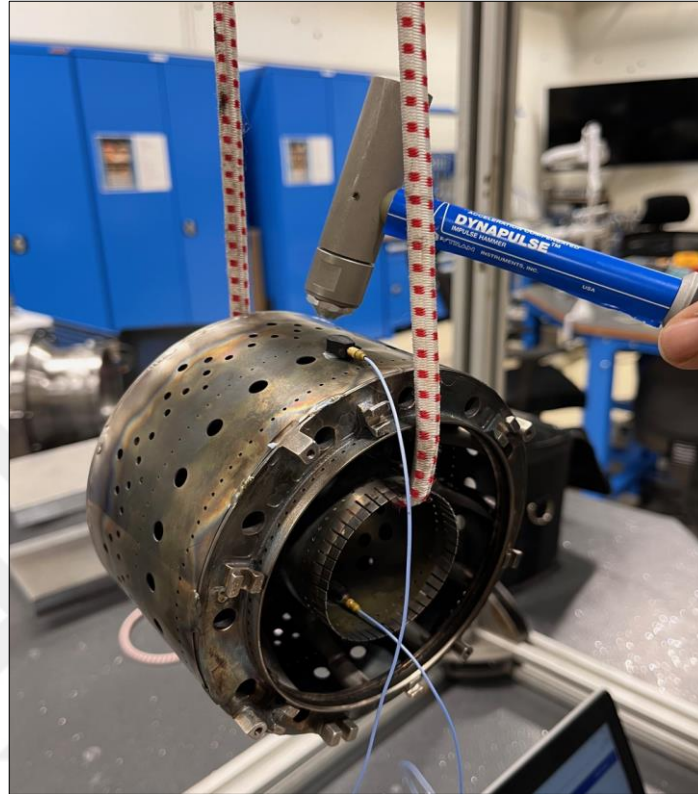


Figure 5.3: Test 2 – Impact from Liner Region.

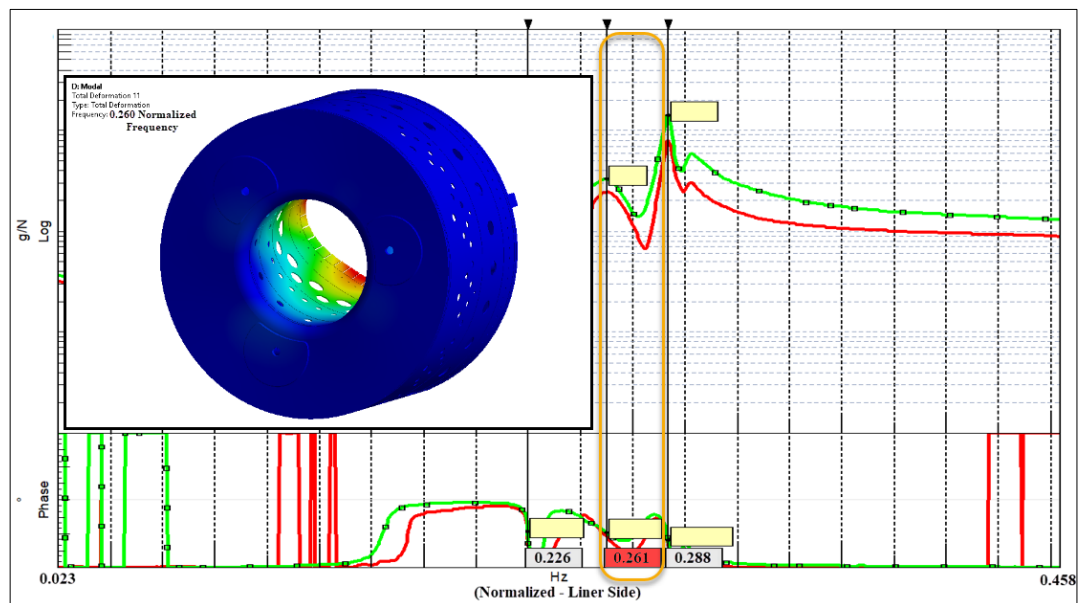


Figure 5.4: Test 2 – Expected Mode (Liner Region).

The comparison between the analysis model and the real part is specified in two modes. By increasing the number of accelerometers and hammering from different regions, all modes of the part and the phases of these modes can be obtained.

With this study, it is seen that the part in the analysis model can represent the real part. The dimensions used in modeling are the nominal dimensions of the part, but the real part contains dimensions from the manufacturing. The reason for the difference between the hammer test and analysis results is the actual condition of the real part.

5.2 Life Assessment

In this section, high cycle fatigue life evaluation was performed for X1, X2 and X3 modes. For the evaluation, the mean stress (σ_m) values from static analysis were read for the highest alternating stress (σ_a) nodes identified in harmonic analysis (**X1A, X2A, X3A**). Conversely, alternating stress (σ_a) values from harmonic analysis were read for the highest mean stress (σ_m) obtained from static analysis (**X1M, X2M and X3M**). In this way, a total of six critical regions (nodes) have been evaluated for modes X1, X2 and X3. Since the thermal distribution was uniform in the evaluated region, no assessment was required within this scope.

The labels X1A, X2A and X3A were used to assess the nodes with highest alternating stress in the X1, X2 and X3 modes. The nodes with highest mean stress are labeled X1M, X2M and X3M.

Stress values for a total of 6 points to be evaluated are shown in table.

Table 5.1: Summarized table for six critical points

Mod	Name	Node ID	σ_a	σ_m	Normalized σ_a & σ_m
X1	X1A	589795	0.748	0.538	
	X1M	573587	0.005	1.224	
X2	X2A	635216	0.242	0.156	
	X2M	573587	0.004	1.224	
X3	X3A	600974	0.456	0.169	
	X3M	573587	0.001	1.224	

The representation of points on the Goodman diagram is as follows.

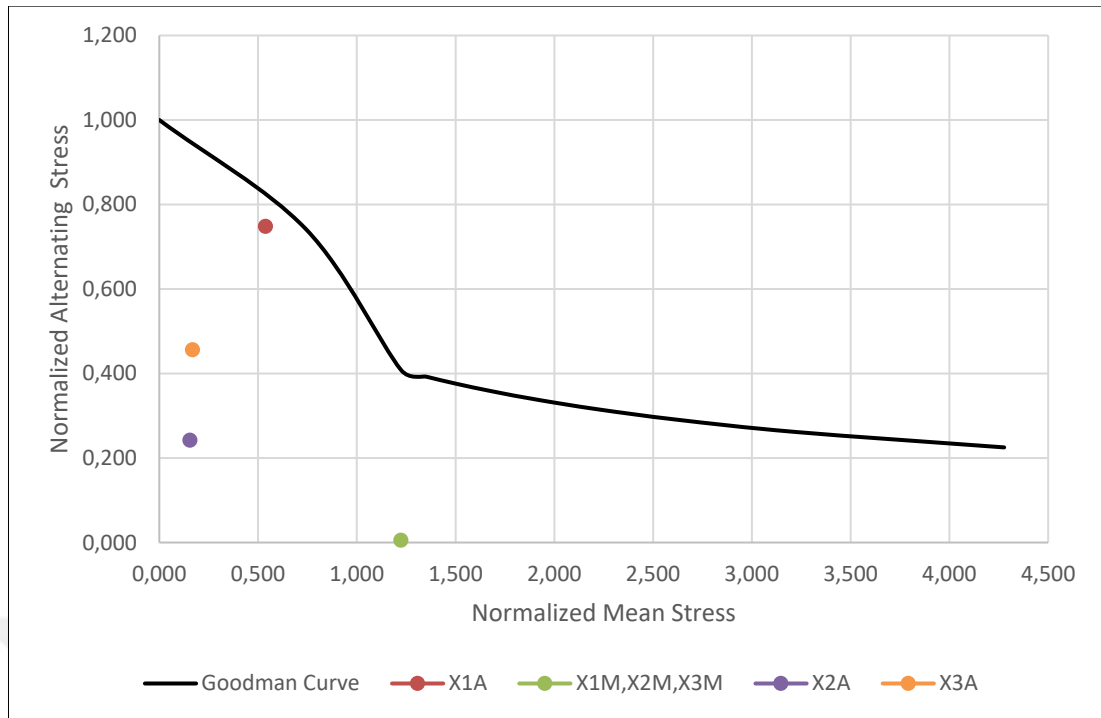


Figure 4.5: Goodman curve and six critical points.

When the results were examined, it was seen that the mentioned critical points remained at the bottom of the Goodman Curve. Staying under the curve means that these points are **infinite life** in terms of high cycle fatigue. Although the X1A point is close to the curve, it can be theoretically defined to have an infinite life when evaluated with the Goodman Curve, which is created by projecting from 10.000.000 cycle region of S-N curve using the **actual test data**.

6. CONCLUSION

In the course of this investigation, static structural analysis, modal analysis, harmonic analysis, and impact hammer tests were conducted to assess the vibration characteristics and high cycle fatigue of the Flame Tube within an experimental engine operating context.

Material data were generated for use in this evaluation. For the material data, many tests were carried out in the material test laboratory environment. A Goodman curve was produced in the 10^7 -cycle projection from the S-N Curve obtained as a result of these high cycle fatigue tests.

Pre-stressed finite modal analysis was performed to determine the HCF critical modes. Additionally, impact hammer tests were performed on the real Flame Tube to validate the modal analysis model. To compare with the results of these hammer impact tests, Free-Free finite element modal analysis was conducted. The modal behavior of the Flame Tube analysis model was examined in comparison to the real part, revealing very close natural frequencies with minimal variance.

The 33 modes obtained as a result of the pre-stressed finite element modal analysis were eliminated according to the “Participation Factor” and “Ratio of Effective Mass to Total Mass” criteria. As a result of this elimination, the remaining 21 modes were examined in the Campbell diagram. In the Campbell diagram, three critical modes are identified that coincide with the 1st Engine Order. Finite element harmonic analysis was performed for each of these modes (X1, X2, X3). Alternating stresses occurring in the critical region were determined by harmonic analysis. As can be seen in the study, six separate worst-case stress conditions were studied for three critical modes. The mean stresses (σ_m) at the nodes with the highest alternating stresses (σ_a) and the alternating stresses (σ_a) at the nodes with the highest mean stresses (σ_m) were read. The six critical situations (X1A, X2A, X3A, X1M, X2M, X3M) created in this way were evaluated on the Goodman diagram. As a result of the analysis, tests, and evaluations, it has been determined that the critical points have an **infinite life** beyond 10^7 cycles.

This study outlines a comprehensive method applicable to the determination of high cycle fatigue life for static components more broadly.

7. RECOMMENDATIONS AND FUTURE WORKS

The Flame Tube components undergo various types of loading and exposure to temperature variations. By examining the effects of temperature fluctuations on the part in more detail, we can examine how temperature changes impact high cycle fatigue life. This approach enables a more realistic evaluation of the Flame Tube under operational thermal conditions.

Validation can be achieved through additional testing with various experimental methods to enhance the methodology and analysis studies conducted in this master's thesis. This study focused on only limited areas (the three nodes with the highest stress due to vibration and the node with the highest operational stress). In the future, a method for evaluating all locations of the part will be developed. Furthermore, beyond this study under specific load conditions, an extended testing process can be arranged to understand the behavior of the part under different operating conditions. Tests conducted under different loads will provide a better understanding of the overall fatigue performance of the part.

Using this method, critical modes of different components in the engine can be investigated, providing insights into high cycle fatigue cycle times for gas turbine engine design.

Increasing the number of accelerometers will allow us to obtain all natural frequencies and phases of the part through the Impact Hammer Test, providing a more comprehensive verification.

Additionally, the high cycle fatigue behavior of the critical area of the part will be examined by performing a shaker test on the produced part. The theoretical study will be compared with the experimental study.

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BIOGRAPHY

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