PREDICTION OF FATIGUE LIFE ON LOWER SUSPENSION ARM SUBJECTED TO VARIABLE AMPLITUDE LOADING

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Dedicated to my parents

My supervisor

My friends

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ABSTRACT

This project focuses on the finite element based fatigue life prediction of lower suspension arm subjected to variable amplitude loading using different fatigue methods. Objectives of this project are to predict fatigue life of the lower suspension arm using stress-life and strain-life methods, to investigate the effect of the mean stress and to identify the suitable material for the suspension arm. The lower suspension arm was developed using computer aided design software. The finite element modeling and analysis were performed utilizing the finite element analysis code. The finite element analysis was performed using MSC.NASTRAN code using the linear elastic approach. In addition, the fatigue life analysis was performed using the stress-life and strain-life approach subjected to variable amplitude loading. The three types of variable amplitude are considered including positive mean loading (SAETRN), compressive mean loading (SAESUS) and zero mean loading (SAEBKT). It can be seen that TET10 mesh and maximum principal stress were captured the maximum stress. From the fatigue analysis, Goodman method is predict the conservative result when subjected to SAETRN and SAESUS loading while SWT method is applicable in SAEBKT loading. Stress-life is capable to give higher fatigue life when subjected to SAEBKT while strain-life method is applicable to give higher fatigue life when subjected to SAETRN and SAESUS. From the material optimization, 7175-T73 aluminum alloy is suitable material of the lower suspension arm.

ABSTRAK

Projek ini mengfokuskan unsur terhingga berdasarkan jangka hayat lesu “lower suspension arm” di bawah pengaruh jenis bebanan amplitud berubah yang menggunakan kaedah jangka hayat lesu yang berlainan. Objektif projek ini ialah meramalkan jangka hayat lesu “lower suspension arm” menggunakan kaedah kehidupan tekanan dan kehidupan regangan, menyiasat kesan daripada tegasan min dan menentukan bahan yang sesuai untuk “lower suspension arm”. “Lower suspension arm” distrukturkan dengan perisian lukisan bantuan komputer. Pengesahan model unsur dan analisis unsur dibangunkan menggunakan unsur terhingga berdasarkan kod analisis lesu. Analisis unsur terhingga dijalankan dengan kod MSC.NASTRAN menggunakan pendekatan elastic linear. Dengan itu, analisis jangka hayat lesu dijalankan menggunakan kaedah kehidupan tekanan dan kehidupan regangan dibawah pengaruh jenis bebanan amplitud berubah. Tiga jenis bebanan amplitud berubah diambil kira termasuk bebanan min positif (SAETRN), bebanan min mampatan (SAESUS) dan bebanan min sifar (SAEBKT). Jaringan TET10 dan prinsip tekanan maksimum diambil kira untuk analisis ini dan kawasan kritikal yang dititikberatkan ialah pada nod (8408). Daripada keputusan yang didapati, pembetulan tegasan Goodman menjangkakan kaedah yang konsevertif apabila dikenakan bebanan SAETRN dan SAESUS, manakala kaedah Smith-Watson Topper (SWT) boleh diguna pada bebanan SAEBKT. Kehidupan tekanan mampu meningkatkan hayat lesu apabila dikenakan bebanan SAEBKT, manakala kehidupan regangan dapat dipakai untuk meningkatkan hayat lesu apabila dikenakan bebanan SAETRN dan SAESUS. Daripada pengoptimuman bahan, aluminum alloy 7175-T73 ialah bahan yang paling sesuai untuk “lower suspension arm”.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

ε_e	Elastic component of the cyclic strain amplitude
c	Fatigue ductility exponent
$\frac{\Delta\varepsilon}{2}$	Strain amplitude
N_f	No of cycle to failure
ε'_f	Fatigue ductility coefficient
b	Fatigue strength exponent
E	Modulus of elasticity
σ_o	Local mean stress
σ_{max}	Local maximum stress
σ'_f	Fatigue strength coefficient

LIST OF ABBREVIATIONS

Al	Aluminum
CAD	Computer-aided design
CAE	Computer-aided engineering
SWT	Smith-Watson-Topper
FE	Finite element
FFM	Finite element modeling
SAETRN	Postive mean loading
SAESUS	Negative mean loading
SAEBKT	Bracket mean loading
MBD	Multibody dynamics
LFC	Low fatigue cycle
TET	Tetrahedral
SAE	Society of Automotive Engineers

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Suspension is the system of linkages and springs or shocks that allows the wheels to move up and down independent of the body. This is important for absorbing bumps in rough terrain, gracefully landing jumps, and getting the right amount of body lean and weight transfer in turns. Both end of this component are fixed to the wheel and the chassis. Suspension components, along with wheel rims and brake components are un-sprung masses, which make weight reduction important for ride quality and response as well as for reducing the total vehicle weight. Every automotive suspension has two goals, passenger comfort and vehicle control. Comfort is provided by isolating the vehicle's passengers from road disturbances like bumps or potholes. Control is achieved by keeping the car body from rolling and pitching excessively, and maintaining good contact between the tire and the road.

The safety of the lower suspension arm can be analyzed by the finite element analysis. The safety aspect of the component is the important factor to develop the automotive industry. The most of the failure observed in the real structure and mechanical component are due to the fatigue. In the design of the real system subjected to the environment loadings, both the fatigue strength and dynamic properties of the external loads are important. In automotive industry, aluminium (Al) alloy has limited usage due to their higher cost and less developed manufacturing process compared to steels. However, Al alloy has the advantage of lower weight and therefore has been used increasingly in car industry for the last 30 years, mainly as engine block, engine parts, brake components, steering components and suspension arms where significant weight can be achieved Kyrre (2006). The increasing use of Al is due to the safety, environmental and performance benefits that aluminum offers, as well as the improved fuel consumption because of light weight.

1.2 PROBLEM STATEMENT

One of the important structural limitations of an aluminium alloy is its fatigue properties. This study is aimed at the automotive industry, more specifically a wrought aluminium suspension system, where safety is of great concern Kyrre (2006). Most of the time to failure consists of crack initiation and a conservative approach is to denote the component as failed when a crack has initiated. This simplification allows designers to use linear elastic stress results obtained from multibody dynamic FE (finite element) simulations for fatigue life analysis. The lower suspension arm is facing the vibration from the variation of road surface. Therefore it is subjected to cyclic loading and it is consequently prone to fatigue damage. The stress from the wheel unit and shock absorber are acting on the lower suspension. The best design of the lower suspension arm is considered for benefit of the cost management of the production. The best material that will be used for the manufacturing process of the component is important to predict the life.

1.3 SCOPE OF STUDY

This study is concentrates on the stress-life and strain-life approach under variable amplitude loading. The scopes of study are structural modeling, finite element modeling (FEM), finite element analysis (FEA), fatigue analysis, and material optimization.

1.4 OBJECTIVES OF THE PROJECT

The objectives of the project are as follow:

i. To predict the fatigue life of suspension arm using stress-life and strain-life method and identify the critical location.

ii. To investigate the influence of the mean stress.
iii. To optimize the material for the suspension arm.

1.5 OVERVIEW OF THE REPORT

Chapter 1 gives the brief the content and background of the project. The problem statement, scope of study and objectives are also included in this chapter. Chapter 2 discusses about variable amplitude loading, stress-life method and strain-life method. Chapter 3 presents the development of methodology, finite element modeling and analysis, fatigue life prediction technique and linear elastic analysis. Chapter 4 discusses the result of the project. Chapter 5 presents the conclusions of the project, and recommendations for the future work.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

The purpose of this chapter is to provide a review of the past research related to the fatigue life method, variable amplitude loading, and stress-life and strain-life method. The review is organized chronologically so as to offer approaching to how research hard works have laid the base for subsequent studies, including the present research effort. The review is fairly detailed so that the present research effort can be properly modified to add to the present body of literature as well as to justify the scope and direction of present research effort.

2.2 FATIGUE LIFE PREDICTION METHOD

Fatigue analysis can be used to determine how long the component can maintain in a given service condition. In general, fatigue life refers to the ability of a component to function in the presence of defect for a given loading. In practice, the predominant failure mode is fatigue and hence, the term fatigue life analysis was used to describe the analysis of the fatigue performance. Takahashi et al. (2008) were studied on creep-fatigue life prediction methods for low-carbon nitrogen-controlled 316 stainless steel (316FR). The authors were conducted long-term creep and creep-fatigue tests for several products of this steel. Superiority of the ductility exhaustion approach against time fraction approach was made clear. Afterwards, additional tests at lower strain range or longer hold time were started to evaluate the applicability to longer-term region. Some new data have been obtained from these tests and the observations obtained in the early stage were evaluated again. In order to address the concerns about applicability of the life prediction method to multiaxial stress states, biaxial fatigue and creep-fatigue tests using cruciform specimens were additionally performed during this phase of the program.

Kyrre (2006) was investigated the fatigue assessment of aluminium suspension arm. Fatigue life prediction from finite element analysis has been discussed. Although the methods can be used for all structural alloys, author focuses on aluminium alloys in automotive structures. The software package nSoft was used for fatigue life prediction and Fedem is used for the dynamic simulations. The author concluded that the dynamic finite element analysis was very computationally intensive. The model must therefore be simple, possibly confined to separate sections of the vehicle. Then the accuracy what was required for static analysis required, since small inaccuracies in peak stresses affect the life prediction can be determined significantly. This was shown for a mesh typically used in static strength evaluations. The mesh was converted using higher order elements and compared to the initial mesh. The new mesh proved to be much more conservative in fatigue life predictions. He applied the Smith-Watson-Topper (SWT) parameter and Morrow mean stress correction and found that stress-life was better correlation at high fatigue life, but the strain-life method must be used if plastic overloads are observed.

Ås et al. (2008) were studied surface roughness characterization for fatigue life predictions using finite element analysis. The authors were established a method to improve the fatigue life prediction of components with rough surfaces. A new method was proposed, in which microscopic surface measurements are used to create finite element models of surface topography. The influence of surface roughness on fatigue life can then be based on stress solutions instead of empirically derived reduction factors. Conle and Mousseau (1991) used the vehicle simulation and finite element result to generate the fatigue life contours for the chassis component using automotive proving ground load history result combine with the computational techniques. They concluded that the combination of the dynamics modeling, finite element analysis is the practical techniques for the fatigue design of the automotive component.

Kyrre et al. (2005) were conducted the fatigue life prediction of suspension arm using finite element analysis of surface topography. They concluded that fatigue strength of the structure is highly depending on the surface quality. Current methods to predict fatigue life rely on empirical relations between geometric surface parameters and observed endurance lives. The uncertainty associated with these methods is typically high, since parameters based on geometrical averages can fail to describe important characteristics of surface topography. Then they proposed a new approach where detailed finite element analysis of surface topography is used as a foundation for fatigue life prediction.

Kim et al. (2002) were studied a method for simulating the vehicles dynamic loads but they add durability assessment. For their multibody dynamic analysis, they used DADS and a flexible body model. The model was for a transit bus. For their dynamic stress analysis, MSC.NASTRAN was used. The fatigue life was then calculated using a local strain approach. From the fatigue life, it was found that the majority of the fatigue damage occurred over a frequency range that depend on terrain traveled (service or accelerated test course). This showed that the actual service environment could be simulated instead of using an accelerated testing environment.

Nadot and Denier (2003) were studied fatigue phenomena for nodular cast iron automotive suspension arms. They found that the major parameter influencing fatigue failure of casting components are casting defects. The high cycle fatigue behaviour is controlled mainly by surface defects such as dross defects and oxides while the low cycle fatigue is governed by multiple cracks initiated independently from casting defects.

Shim and Kim (2008) was studied the cause of failure and optimization of a V-belt pulley considering fatigue life uncertainty in automotive applications. Authors also analyzed a critical part by using plastic processing methods and investigated the cause of failure. The applied stress distribution of the pulley under high-tension and torque was obtained by using FEA. Based on these results, the fatigue life of the pulley considering the variation in the fatigue strength was estimated with a durability analysis simulator. A study on the shape of the optimal design was performed to increase the fatigue life of the pulley, while minimizing the weight of the V-belt pulley in the compressor system of a vehicle.

Rahman et al. (2008) were studied finite element based fatigue life prediction of cylinder head for two-stroke linear engine using stress-life approach. Fatigue stress-life approach was used and sensitivity analysis on fatigue life is discussed. Stresses obtained previously are employed as input for the fatigue life. From the result, it was shown that the Goodman mean stress correction method is predicted more conservative (minimum life) results. It was found to differ considerably the compressive and tensile mean stresses to give noticeable advantages and fond to be design criteria.

The fatigue strain-life approach involves the techniques for converting the loading history, geometry, and materials properties (monotonic and cyclic) input into a fatigue life prediction. The operations involved in the prediction process must be performed sequentially. First, the stress and strain at the critical region are estimated, and the rainflow cycle counting method Matsuishi and Endo (1968) is then used to reduce the load-time history based on the peak-valley sequential. The next step is to use the finite element method to convert the reduced load-time history into a strain-time history and also to calculate the stress and strain in the highly stressed area. Then, the crack initiation methods are employed to predict the fatigue life. The simple linear hypothesis proposed by Palmgren, (1924) and Miner (1945) is used to accumulate fatigue damage. Finally, the damage values for all cycles are summed until a critical damage sum (failure criteria) is reached.

In order to perform fatigue life analysis and to apply the stress-strain approach in complex structures, Conle and Chu (1977) used the strain-life result which is simulated using three-dimensional models to evaluate the fatigue damage. After the complex load history was reduced to an elastic stress history for each critical element, a neuber plasticity correction method was used to correct the plastic behavior. Elastic unit load analysis using strength of materials and an elastic finite element analysis model combined with a superposition procedure of each load point’s service history was proposed. Savaidis (2001) verified that the local strain approach is suitable for a fatigue life evaluation. In this study, it is consider that the local strain approach using the Smith-Watson-Toper (SWT) strain-life model is able to represent and to estimate the parameters. These include mean stress effects, load sequence effects above and below the endurance limit, and manufacturing process effects such as surface treatment and residual stresses, and also stated by Juvinall and Marshek (2000).

Rahman et al. (2009) were conducted fatigue life prediction of lower suspension arm using strain-life approach. From the fatigue analysis, Smith-Watson-Topper mean stress correction was conservative method when subjected to SAETRN loading, while Coffin-Manson model is applicable when subjected to SAESUS and SAEBRKT loading. From the material optimization, 7075-T6 aluminum alloy is suitable material of the suspension arm.

2.3 VARIABLE AMPLITUDE LOADING

When components are subjected to variable amplitude service loads, additional uncertainties arise, whether the loading in laboratory tests related to the loads that could be expected to appear. Traditionally this problem is solved by using the simplifying assumption of damage accumulation, and constant amplitude tests in laboratory are transformed to variable amplitude severity by the Palmgren-Miner rule which says that a load cycle with amplitude *Si* adds to the cumulative damage *D*, a quantity (*1*/Ni). Here, Ni denotes the fatigue life under constant amplitude loading with amplitude *Si* and n_i is the number of load cycles at this amplitude.

$$\mathrm{D} = \sum_{i=1}^{m} \frac{n_i}{N_i} \tag{2.1}$$

The lack of validity of this accumulation rule has been demonstrated in many applications and in consequence its usage will introduce uncertainties which must be compensated for by safety factors. One possible way to diminish the deviations from the damage accumulation rule is to perform the laboratory experiments closer to the service behaviour with respect to the loads. A method for establishing a Wohler curve based on variable amplitude loads has recently been developed and is presented in a parallel paper Johannesson et al. (2005). The use of this method should be customized to each specific application by performing laboratory tests with load spectra covering different service requirements. One idea is that service measurements are used to establish a few reference load spectra for use in laboratory tests. Based on the resulting variable amplitude Wohler curve, fatigue life can be predicted for load spectra similar to the reference types.

Svensson et al. (2005) was conducted the fatigue life prediction based on variable amplitude tests-specific applications. Three engineering components have been tested with both constant amplitude loading and different load spectra and the results are analysed by means of a new evaluation method. The method relies on the Palmgren-Miner hypothesis, but offers the opportunity to approve the hypothesis validity by narrowing the domain of its application in accordance with a specific situation. In the first case automotive spot weld components are tested with two different synthetic spectra and the result is extrapolated to new service spectra. In the second case, the fatigue properties of a rock drill component are analyzed both by constant amplitude tests and by spectrum tests and the two reference test sets are compared. In the third case, butt welded mild steel is analyzed with respect to different load level crossing properties and different irregularity factors.

Molent et al. (2007) was evaluated the spectrum fatigue crack growth using variable amplitude data. This paper summarizes a recent semi-empirical model that appears to be capable of producing more accurate fatigue life predictions using flight load spectra based on realistic in-service usage. The new model described here provides an alternative means for the interpretation of full-scale and coupon fatigue test data, and can also be used to make reliable life predictions for a range of situations. This is a very important capability, particularly where only a single full-scale fatigue test can be afforded and should lead to more economical utilization of airframes.

Rahman et al. (2007a) were studied about finite element based durability assessment in a two- stroke free piston linear engine component using variable amplitude loading. Authors discussed the finite element analysis to predict the fatigue life and identify the critical locations of the component. The effect of mean stress on the fatigue life also investigated. The linear static finite element analysis was performed using MSC. NASTRAN. The result was capable of showing the contour plots of the fatigue life histogram and damage histogram at the most critical location.

Nolting et al. (2008) was investigated the effect of variable amplitude loading on the fatigue life and failure mode of adhesively bonded double strap (DS) joints made from clad and bare 2024-T3 aluminum. They concluded that the fatigue life of a variable amplitude loading spectra can be calculated with reasonable accuracy using an effective stress range vs. life fatigue curve. The effective stress range vs. failure life curve is dependent on the bond geometry and therefore this curve must be developed for each component geometry of interest. The effective stress range versus life fatigue curve should be used to predict the fatigue life of clad specimens if the failure mode of the clad specimens is expected to be adhesive failure.

2.4 CONCLUSION

This chapter is about the summary of previous works that related to this project. The works were discussed about fatigue life prediction method, variable amplitude loading, strain-life method and stress-life method. The next chapter is about the methodology of the project.

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter presents the overall methodology of the finite element based fatigue analysis. One of the essential goals in the fatigue process study is to predict the fatigue life of a structure or machine component subjected to a given stress–time history. To allow this prediction, complete information about the response and behaviour of the material subjected to cyclic loading is necessary. In addition to the characterization of the cyclic stress–strain response, quantitative information on resistance to crack is primary importance.

3.2 PROJECT FLOWCHART

The flowchart of the finite element based fatigue analysis is shown in Figure 3.1.

3.3 FINITE ELEMENT BASED FATIGUE LIFE ANALYSIS

Fatigue analysis has traditionally been performed at a later stage of the design cycle. This is due to the fact that the loading information could only be derived from the direct measurement, which requires a prototype (Bannantine et al., 1990; Stephens et al., 2000). Multibody dynamics (MBD) (Kim et al., 2002) is capable of predicting the component loads which enable designer to undertake a fatigue assessment even before the prototype is fabricated. The purpose of analyzing a structure early in the design cycle is to reduce the development time and cost. This is achieved by determining the critical region of the structure and improving the design before prototype are built and tested.

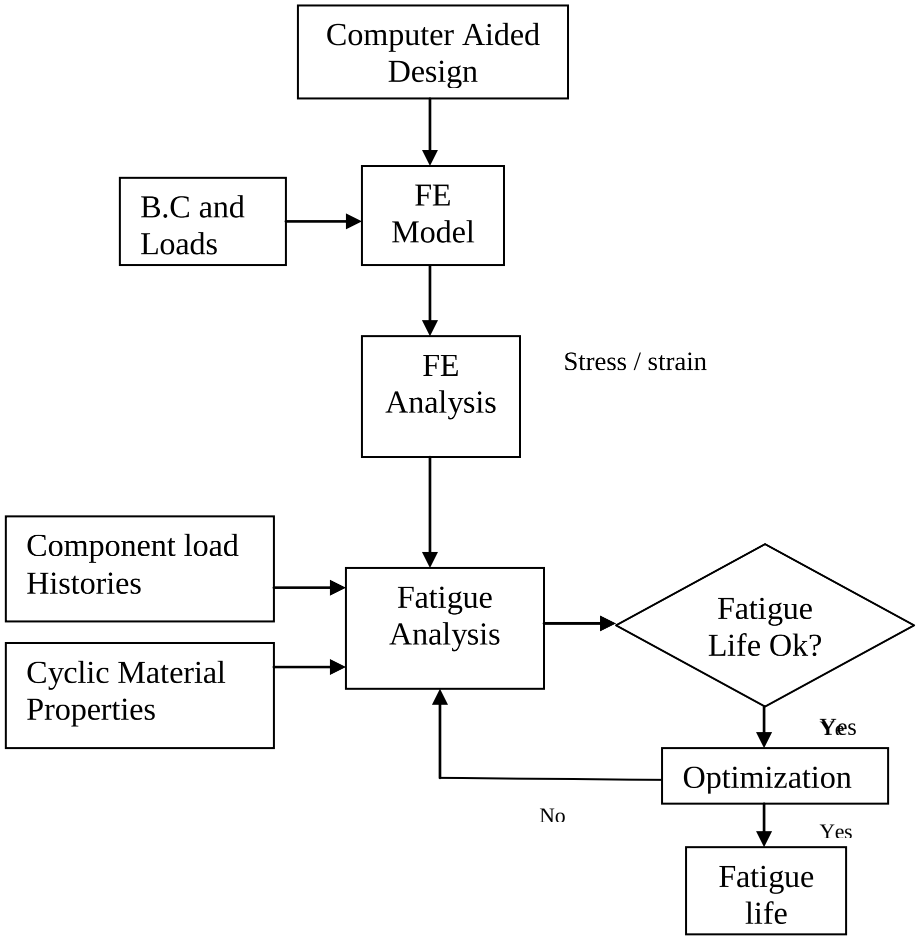

Figure 3.1: The finite element based fatigue analysis

Two computational processes are utilized to perform the finite element based fatigue analysis processes are as follows:

(i) Finite Element (FE) – to determine the stress/strain state of a component for a given load condition;

(ii) Fatigue analysis – to calculate the fatigue life for the component of interest and identify the critical locations.

The required inputs for the fatigue analysis are shown in Figure 3.2. The three input information are description of the material properties, loading histories and geometry. The details of these inputs are as follows:

(i) Material information – cyclic or repeated material based on a constant amplitude testing.

(ii) Loads histories information – measure or simulated load histories applied to a component. The term “loads” is used to represent forces, displacement, accelerations, etc.

(iii) Geometry information – relates the applied loads histories to the local stresses and strains at the location of interest. The local stresses and strains information are usually derived from the FE result.

The FE based fatigue analysis can be considered as a complete engineering analysis. The fatigue life can be estimated for every element in the finite element model, and the contour plots of life damage can be obtained. The geometry information is provided by the FE result for each load applied independently. Appropriate material properties are also provided for the desired fatigue analysis method. An integrated approach to fatigue life analysis combines the finite element analysis, and the fatigue analysis into a consistent entity for the prediction of the fatigue life of a component (Rahman et al. 2007b).

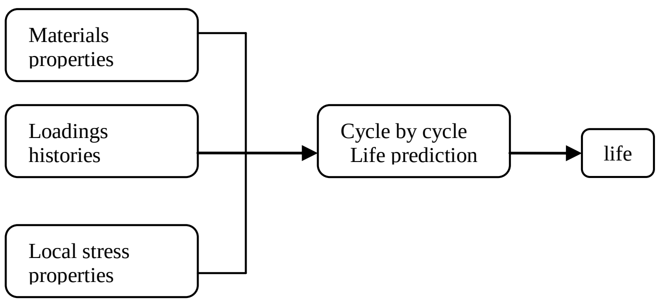

Figure 3.2: Schematic diagram of fatigue life estimation

Source: (Rahman et al., 2007b)

3.4 STRESS- LIFE METHOD

The stress life *(S-N)* method was first applied over hundred years ago (Wöhler, 1867) and consider nominal elastic stresses and how they related to life. This approach to the fatigue analysis of components works well for situations in which only elastic stresses and strains are present. However, most components may appear to have nominally cyclic elastic stresses but stress concentration are present in the component may result in load cyclic plastic deformation (Rahman et al., 2007b).

The *S-N*, method was first approach used in an attempt to understand and quantify metal fatigue. It was the standard fatigue design method almost 100 years. The *S-N* approach is still widely used in design applications where the applied stress is primary within the elastic range of the material and the resultant lives (cycles to failure) are long, such as power transmission shaft. The stress-life method does not work well in low-cycle applications, where the applied strain have significant plastic component. The dividing line between low and high cycle fatigue depends on the material being considered, but usually falls between 10 and 10^5 cycles (Banantine et al., 1990).

This method is also distinguished from the other fatigue analysis and design technique by several features: (Lee et al., 2005)

(i) Cyclic stresses are the governing parameters for the fatigue failure.
(ii) High-cyclic fatigue conditions are present (high number of cycles to failure and little plastic deformation due to cyclic loading

During fatigue testing, the test specimen is subjected to alternating loads until failure. The loads applied to the specimen are defined by either a constant stress range (S_r) or constant stress amplitude (S_a). The stress range and stress amplitude are defined as Equation (3.1) and (3.2), respectively.

$$S_r = S_{\max} - S_{\min} \tag{3.1}$$

$$S_a = \frac{S_r}{2} = \frac{(S_{\max} - S_{\min})}{2} \tag{3.2}$$

Typically, for fatigue analysis, it is a convection to consider tensile stresses positive and compressive stresses negative. The magnitude of the stress range or amplitude is the controlled (independent) variable and the number of cycles to failure is response (dependent) variable. The number of cycles to failure is the fatigue life (N_f), and each cycle is equal to two reversals (2 N_f). The symbols of stresses and cycles mentioned previously are illustrated in Figure 3.3. Most of the time, *S-N* fatigue testing is conducted using fully reversed loading. Fully reverse indicates that loading is alternating about zero mean stress. The mean stress (S_m) is defined as

$$S_m = \frac{(S_{\max} + S_{\min})}{2} \tag{3.3}$$

Exceptions exist when stress life testing is performed for specimens for which this type of lading is physically nit possible or is unlikely. One example is the fatigue testing of spot welded specimens. Cyclic loading varying from zero to tension is used in fatigue testing on specimens with the single spot weld, because compression may cause local buckling of the thin sheet metal.

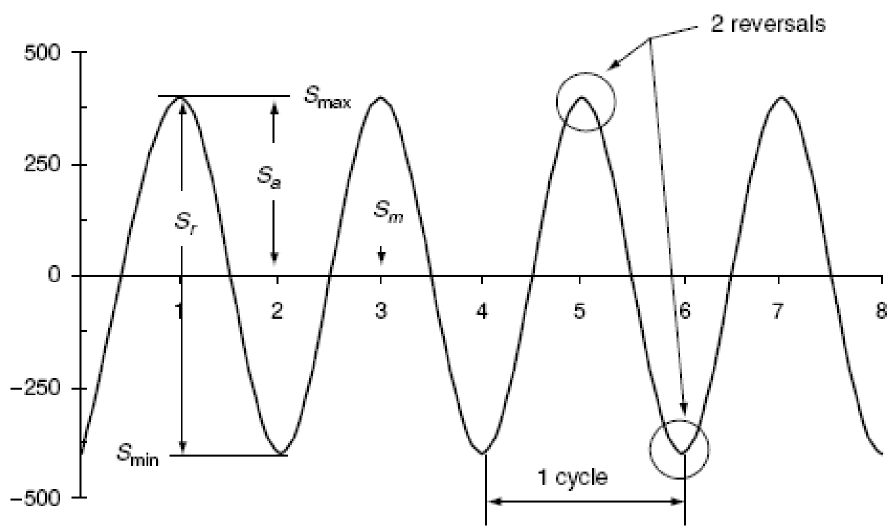

Figure 3.3: Symbols used with cyclic stresses and cycles

Actual structural components are usually subjected to the alternating loads with a mean stress. Two parameters, the stress ratio *(R)* and the amplitude ratio *(A)*, are often used as representations of the mean stress applied to an object. The stress ratio is defined as the ratio of minimum stress to maximum stress:

$$R = \frac{S_{\min}}{S_{\max}} \tag{3.4}$$

The amplitude ratio is the ratio of the stress amplitude to mean stress:

$$A = \frac{S_a}{S_m} = \frac{1-R}{1+R} \tag{3.5}$$

The stress-life approach was the first well-developed approach to the fatigue analysis. It is suitable to predict high cycle fatigue and has been extensively used in automotive industry. Fatigue life depends primarily on loads, materials, geometry and environmental effects and it's usually described by *S-N* curve. The stress-based approach considers the controlling parameters for fatigue life to nominal stress. The relationship between the nominal stress amplitude and fatigue life is often represented as *S-N* curve, which can expressed in Equation (3.6)

$$\sigma_a = \sigma'_f (2N_f)^b \tag{3.6}$$

where σ_a is stress amplitude, σ'_f is a fatigue coefficient, $2N_f$ is the reveals to failure and b is the fatigue strength exponent (Rahman et al., 2007b). The *S-N* curves shows in Figure 3.4.

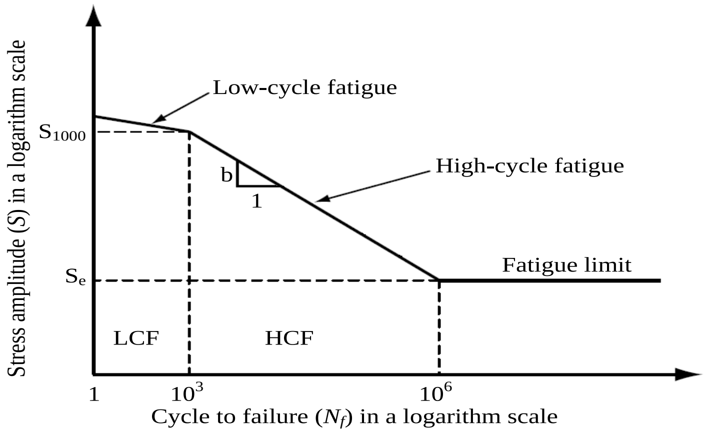

Figure 3.4: *S-N* curve

3.5 STRAIN-LIFE METHOD

The strain-life method is based on the observation that in many components the response of the material in critical location (notches) is strain or deformation dependent. When loads are low, stress and strain are linearly related. Consequently, in this range, load-controlled and strain-controlled test results are equivalent. The stress –life data are generated from load-controlled test. At high load levels, in the low cycle fatigue (LFC) regime, the cyclic stress-strain response and the material behaviour are best model under strain-controlled conditions.

The local strain-life approach is preferred if the loading history is irregular and where the mean stress and the loads sequence effect are thought to be important. The operations involved in the prediction process must be performed sequentially. First, the stress and the strain region are estimated and then the rainflow cycle counting method Matsuishi and Endo (1968) is used to the load-time history into a strain-time history. The next step is to use the finite element method to convert the reduced load-time history and also to calculate the stress and strain in the highly stressed area. Then, the crack initiations method is employed to predict the fatigue life. The simple linear hypothesis proposed by Palmgren (1924) and Miner (1945) is used to accumulate the fatigue damage. Finally, the damage values for all cycles are summed until a critical damage sum (failure criteria) is reached. Coffin and Manson prposed the strain-life relationship in Equation (3.7).

$$\varepsilon_t = \frac{\sigma'_f}{E}(2N_f)^b + \varepsilon'_f(2N_f)^c \qquad (3.7)$$

where, ε_e is the elastic component of the cyclic strain amplitude

c is the fatigue ductility exponent

σ'_f fatigue strength coefficient

N_f is the number of cycles to failure

b is fatigue strength exponent

ε'_f is fatigue ductility coefficient

E is modulus of elasticity

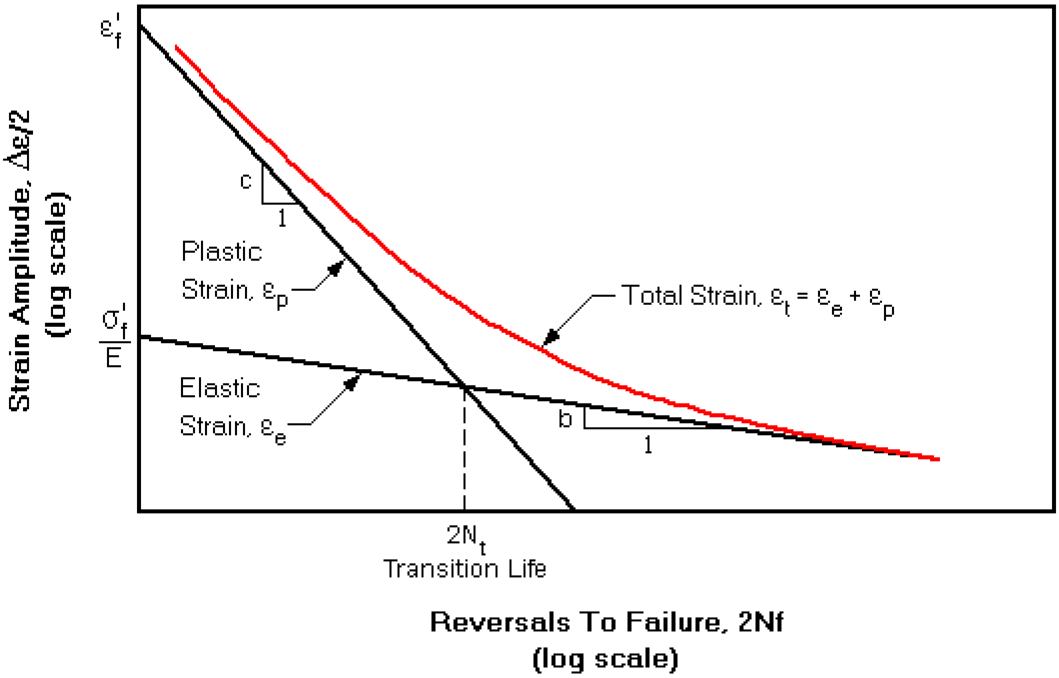

Figure 3.5: Typical Strain-Life Curve

The strain-life curve can be formed by summing the elastic and plastic components. The influence of the elastic and plastic components on the strain-life curve is shown in Figure 3.5. The transition life ($2N_t$) represents the life at which the elastic and plastic strain ranges are equivalent. As shown in Figure 3.5, elastic strains have a greater influence on fatigue lives above the transition life. Plastic strains have a greater influence below the transition life. Thus the transition life provides a convenient delineation between low-cycle and high-cycle fatigue regimes. Note that at long fatigue lives the fatigue strength (σ_f'/E) controls the fatigue performance and the Strain-Life and Stress-Life approaches give essentially the same results. For short fatigue lives, plastic strain is dominant and fatigue ductility (ε_f') controls the fatigue performance. The optimum material is therefore one that has both high ductility and high strength. Unfortunately, there is usually a trade-off between these two properties and a compromise must be made for the expected load or strain conditions being considered.

3.6 MEAN STRESS CORRECTION METHOD

3.6.1 Stress-Life Method

From the perspective of applied cyclic stresses, fatigue damage of component correlates strongly with the applied stress amplitude or applied stress range and is also influenced bye the mean stress (a secondary factor). In the high- cycle fatigue region, normal mean stresses have a significant effect on fatigue behaviour components. Normal mean stresses are responsible for the opening and closing state microcracks. Because the opening of microcracks accelerates the rate of crack propagation and the closing of microcraks retards the growth of cracks, tensile normal mean stress are detrimental and compressive normal mean stresses are beneficial in terms of fatigue strength. The shear mean stress does not influence the opening and closing state of microcraks, and not surprisingly, has little effect on crack propagation. There is very little or no effect of mean stress on fatigue strength in the low- cycle fatigue region in which the large amount of plastic deformation erase any beneficial or detrimental effect of a mean stress (Lee et al., 2005). The modified Goodman and Gerber equations are given by Equation (3.8) and (3.9) respectively:

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_u} = 1 \tag{3.8}$$

$$\frac{\sigma_a}{S_e} + \left(\frac{\sigma_m}{S_u}\right)^2 = 1 \tag{3.9}$$

where σ_a is altering stress in the presence of mean stress, S_e is alternating stress for equivalent completely reversed loading, σ_m is mean stress, and S_u is ultimate tensile strength. The typical representation of these mean stress equations is shown in Figure 3.6. Mean stresses are detrimental when they reduce the fatigue resistance, and beneficial when they improve the fatigue resistance of the material (Rahman et al., 2006).

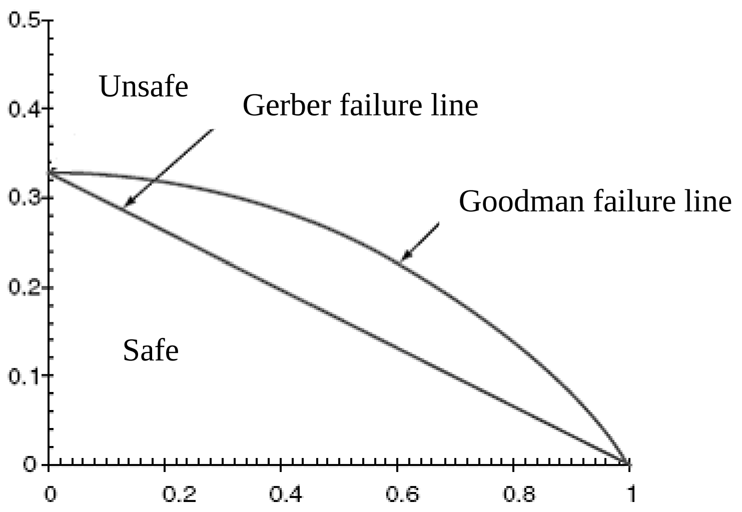

Figure 3.6: Comparison of Mean Stress equation

3.6.2 Strain-Life Method

Most basic *S-N* fatigue data collected in the laboratory is generated using a fully-reversed stress cycle. However, actual loading applications usually involve a mean stress on which the oscillatory stress is superimposed. The effect of mean stress on the strain-life curve is shown schematically in Figure 3.7. Mean stress primarily affects component life in the high-cycle regime, with compressive means extending life and tensile means reducing it. In the plastic regime, large cyclic plastic strains cause mean stress relaxation, and any mean stress tends towards zero. Morrow (1968) was the first to propose a modification to the baseline strain-life curve to account for the effect of mean stress. In terms of the strain-life relationship, the Morrow Mean Stress Correction can be expressed in Equation (3.10).

$$\varepsilon_t = \frac{\Delta\varepsilon}{2} = \frac{\sigma'_f - \sigma_o}{E}\,(2N_f)^b + \varepsilon'_f\,(2N_f)^c \qquad (3.10)$$

where the mean stress σ_0 is positive for tensile stress and negative for compressive stress.

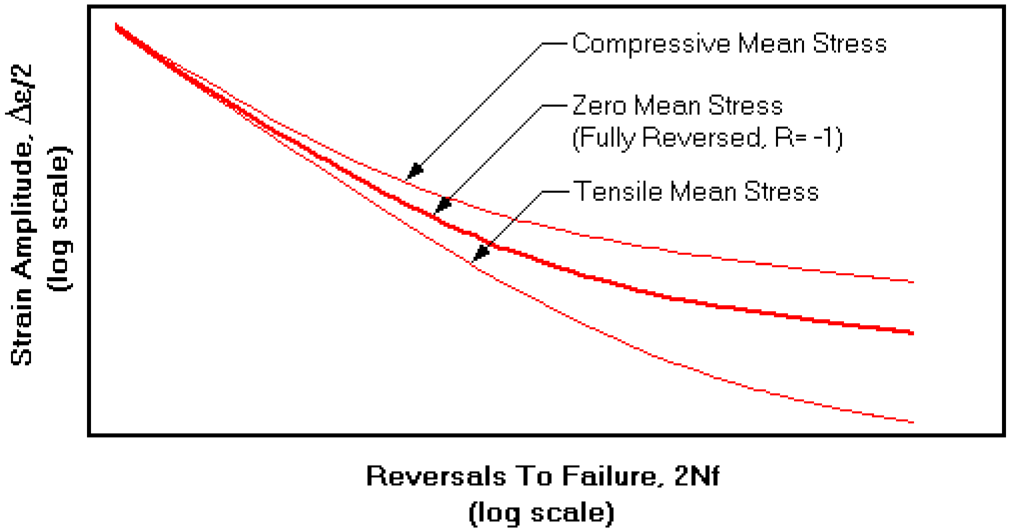

Figure 3.7: Effect of mean stress on Strain-Life curve

Figure 3.8 illustrates the effect of a tensile mean stress in modifying the strain-life curve using the Morrow equation. A different method for modifying the strain-life curve to account for mean stress was proposed by Smith, Watson, and Topper (Smith et al., 1970) which is expressed in Equation (3.11)

$$\sigma_{max}\frac{\Delta\varepsilon}{2} = \frac{\sigma'_f}{E}(2N_f)^{2b} + \varepsilon'_f(2N_f)^{b+c} \qquad (3.11)$$

The SWT equation predicts that no fatigue damage occurs when the maximum stress is zero or negative (i.e., compressive), which is not always true. Therefore the Morrow correction should be used for loading sequences that are predominantly compressive. In cases of predominantly tensile loading, the SWT approach is more conservative than the Morrow approach and is thus recommended.

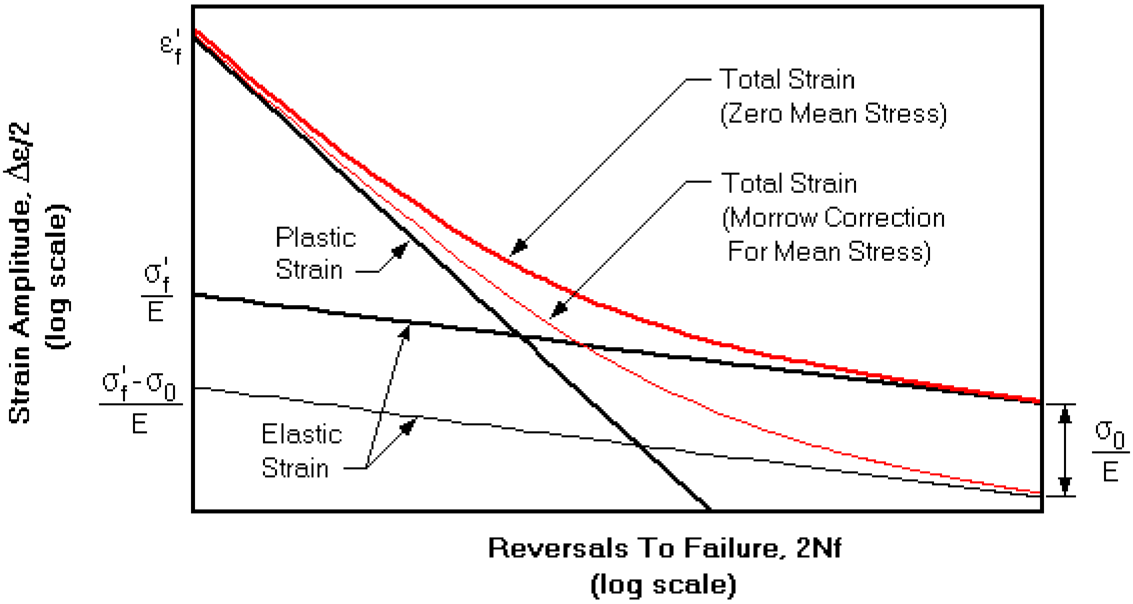

Figure 3.8: Effect of mean stress on strain-life curve (Morrow Correction)

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3.7 MATERIAL INFORMATION

Material model and material properties play an important role in the result of the FE method. The material properties are one of the major inputs which is the definition of how a material behaves under the cyclic loading conditions. The cyclic material properties are used to calculate the elastic-plastic stress-strain response and the rate at which fatigue damage accumulate due to each fatigue cycle. The materials parameters required depend on the analysis methodology being used. The mechanical properties of 5083-87 aluminum alloy are shown in Table 3.1.

Table 3.1: Mechanical properties of 5083-87 alumnium alloy

Properties	**Aluminum alloy 5083-87**
Monotonic Properties	
Tensile strength, σ_{UTS} (MPa)	385
Yield strength, σ_{YS} (MPa)	285
Young's modulus, E (GPa)	69
Density (kg/m^3)	2790
Cyclic and fatigue properties	
Fatigue strength exponent, b	-0.094
Fatigue strength coefficient, σ'_f (MPa)	650
Fatigue ductility exponent, c	-1.01
Fatigue ductility coefficient, ε'_f	2.26

3.8 LOADING INFORMATION

Loading is another major input to the finite element based fatigue analysis. Several types of variable amplitude loading history were selected from the Society of Automotive Engineers (SAE) profiles. The component was loaded with three random time histories, corresponding to typical histories for transmission, suspension and bracket components at different load levels. These histories were scaled to two peak strain levels and used as full-length histories. In addition, a random history including many spikes was selected for the simulation of spike removal. The variable amplitude load-time histories are shown in Figure 3.9. The terms of SAETRN, SAESUS, and SAEBRKT represent the load-time history for the positive, compressive, and zero mean loading respectively. The considered load-time histories are based on the SAE's profile. The realistic loading time histories are often difficult to obtain. A measurement of these is required during typical and extreme operating conditions. This is normally carried out by a dedicated measurement device such as transducers and data acquisition equipment. The loading information usually requires a measurement to instrument and collect the required load time histories under realistic condition. The finite element based on fatigue analysis help to eliminate unnecessary tests by allowing the engineer to check the fatigue performance analytically and optimize the selection of the material, manufacturing process and geometry within the constraints of the total cost and loading environment.

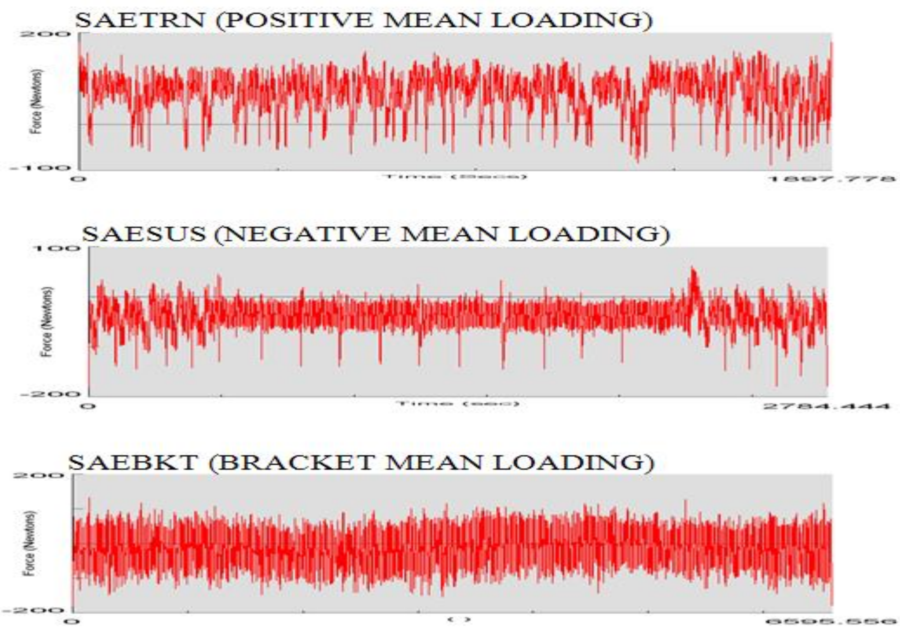


Figure 3.9: The variable amplitude load-time histories

3.9 CONCLUSION

This chapter is about the fatigue based finite element analysis and the implementation of strain-life and stress-life approach. The variable amplitude loading also discussed it is an importance major input in the prediction of fatigue life. The next chapter will be described about structural modeling, finite element modeling (FEM) and the fatigue analysis. The result and discussion about fatigue life also will be discussed in the next chapter.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 INTRODUCTION

This chapter is represents about the results of the FE analysis, fatigue life analysis and material optimization of the suspension arm.

4.2 FINITE ELEMENT MODELING

The suspension arm is one of the important components in the automotive suspension component. A simple three-dimensional model of suspension arm was developed using SolidWorks software as shown in Figure 4.1**.** Overall dimensios of the suspension arm is shown in Figure 4.2. A 10 node tetrahedral element (TET10) was used for the solid mesh. Sensitivity analysis was performed to determine the optimum element size. These analyses were preformed iteratively at different mesh global length until the appropriate accuracy obtained. Convergence of the stresses was recorded as the mesh global length was refined. The mesh global length of 0.3mm was considered and the force 180 N was applied one end of the bushing that connected to the tyre. The other two bushing that connected to the body of the car are constraint. These preload is based on Nadot and Denier, (2003). The three-dimensional FE model, loading and constraints of suspension arm are shown in Figure 4.3.

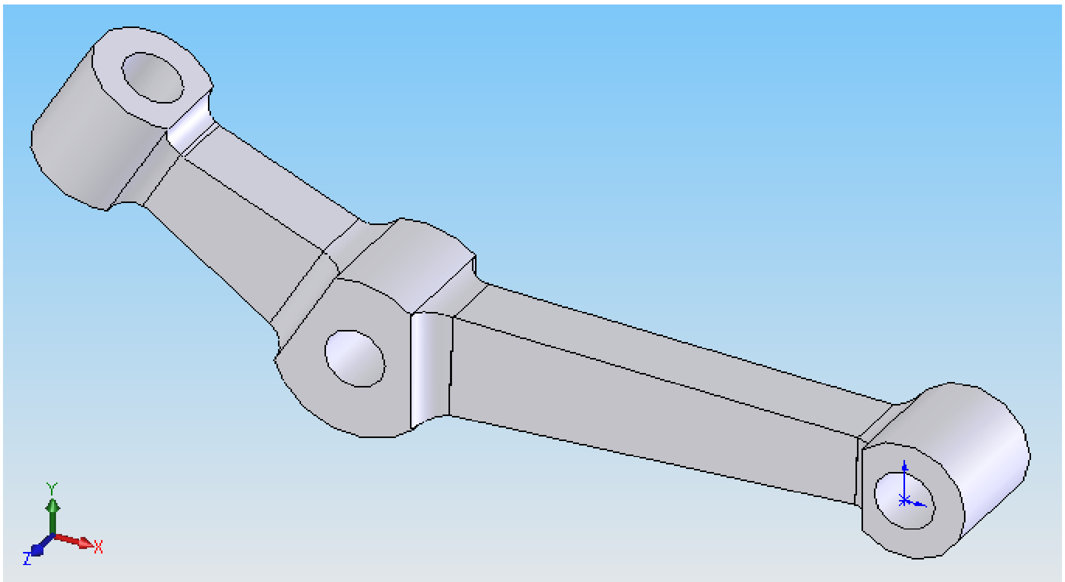


Figure 4.1: Structural model of suspension arm

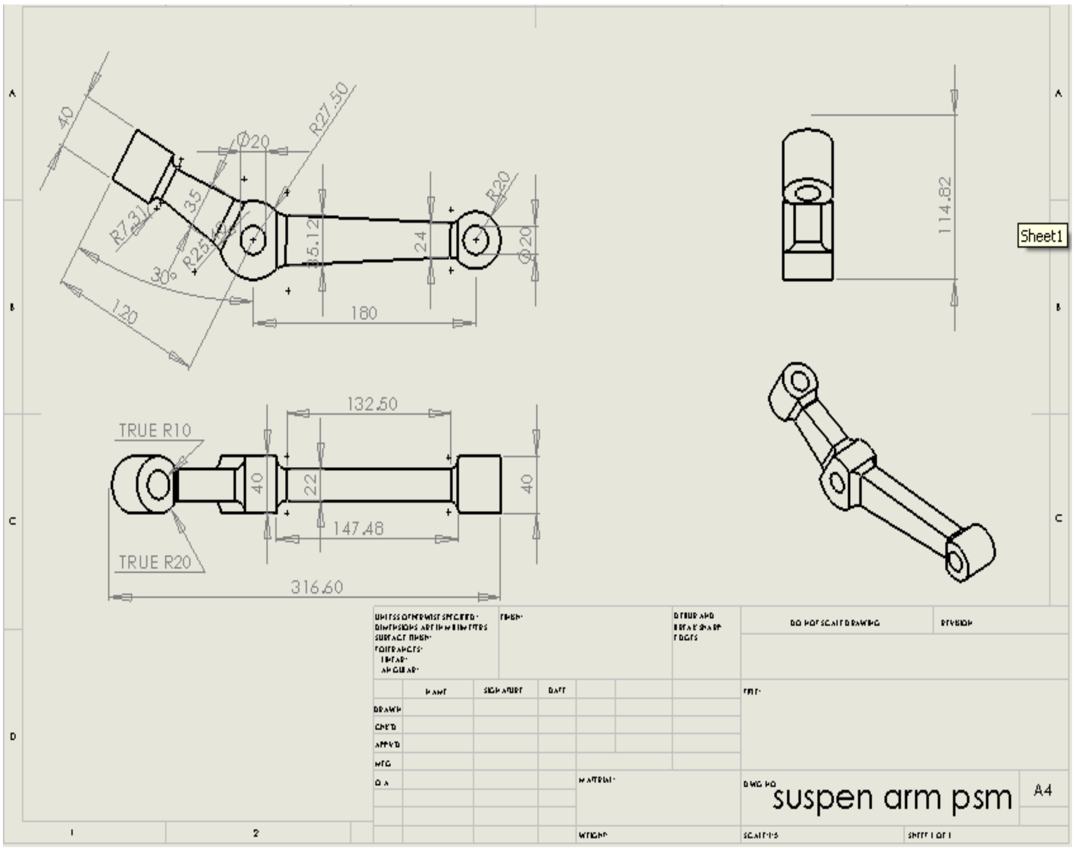


Figure 4.2: Overall dimensions of suspension arm

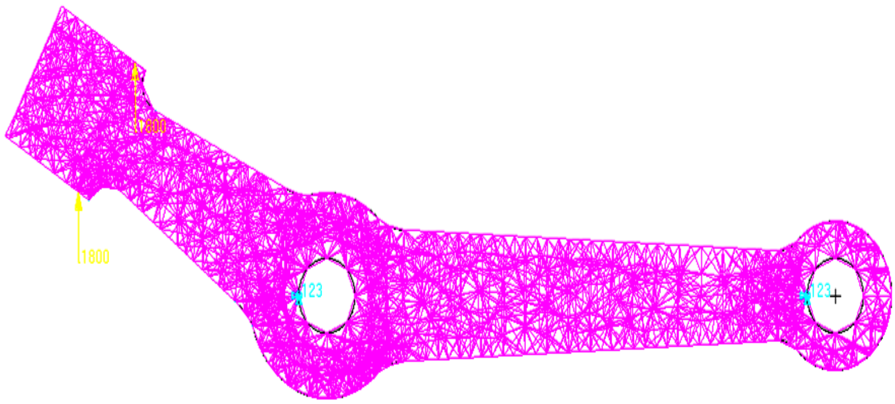
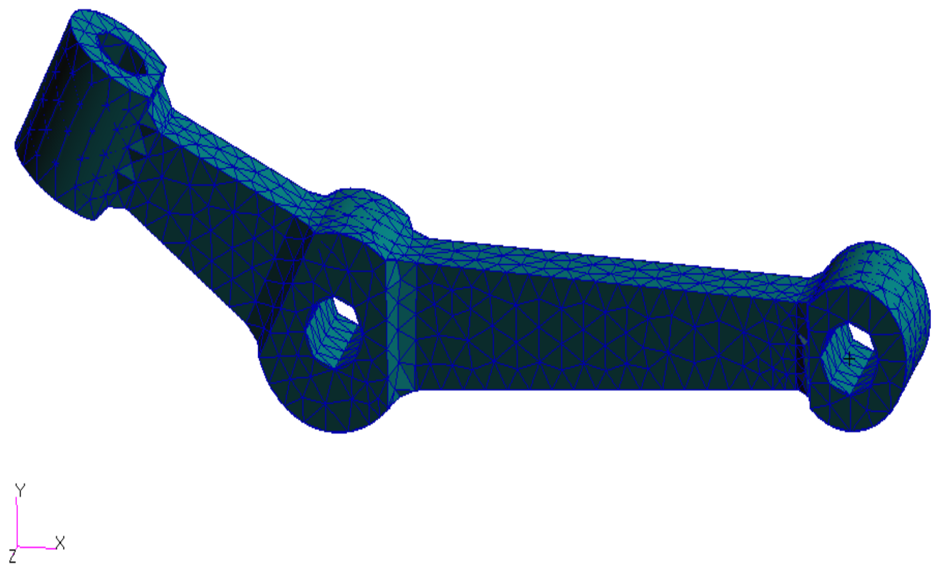


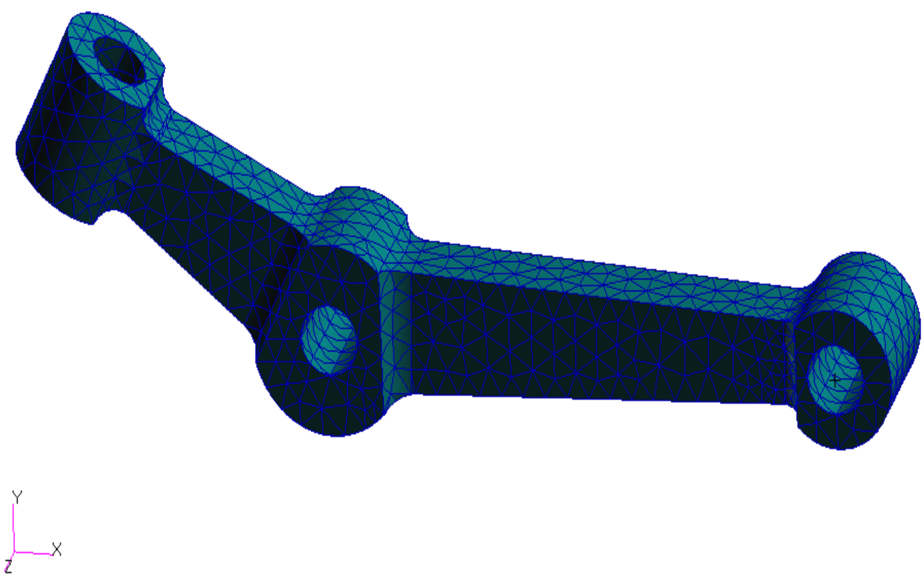
Figure 4.3: Three-dimensional FE model, loading and constraints

4.3 SELECTION OF THE MESH TYPE

Mesh study is performed on the FE model to ensure sufficiently fines sizes are employed for accuracy of calculated results but at competitive Computer Processing Unit (CPU) time. In the process, specified field variable is selected and its convergence is monitored and evaluated. Selecting the right techniques of meshing are based on the geometry, model topology, analysis objectives. Tetrahedral meshing produces high quality meshing for boundary representation solids model imported from the most CAD system. The tetrahedral elements (TET10) and tetrahedral elements (TET4) are used for the initial analysis are shown in Figure 4.4. It can be seen that TET10 mesh predicted higher von Mises stress than TET4 mesh as shown in Figure 4.5. The maximum von Mises stresses are obtained 81.8 MPa and 436 MPa for TET4 and TET10 respectively. Then, the comparison was made between these two elements based on different stresses techniques such as von Mises, Tresca, maximum principal and as well as displacement which are shown in Figures 4.6 and 4.7 respectively. The results show that TET10 has capture higher stresses concentration compared to TET4 for the same mesh global length. It can be seen that the TET10 capture the highest stresses for different techniques throughout the range than TET4.

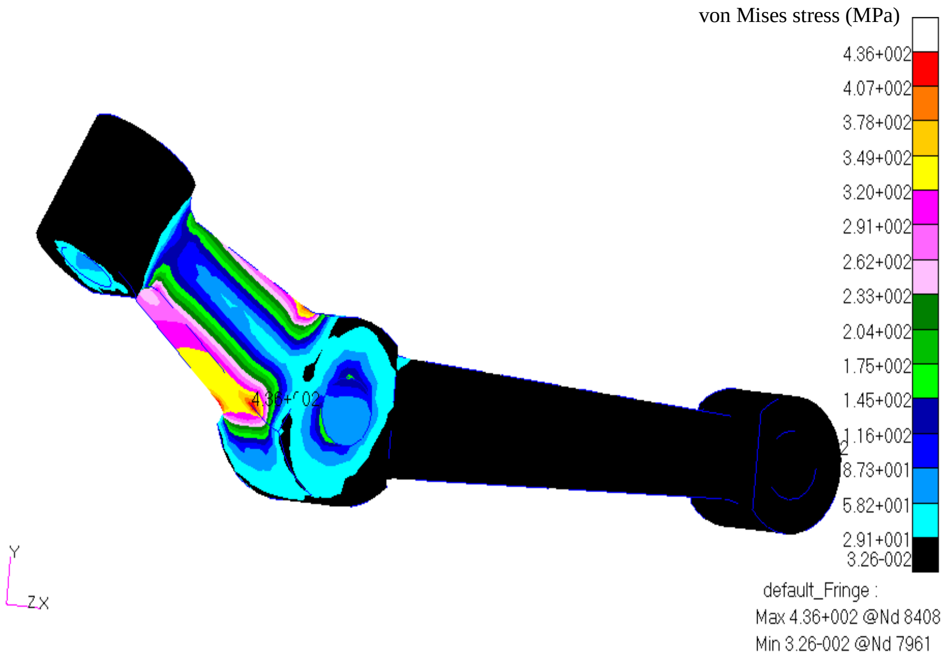


(a) TET4

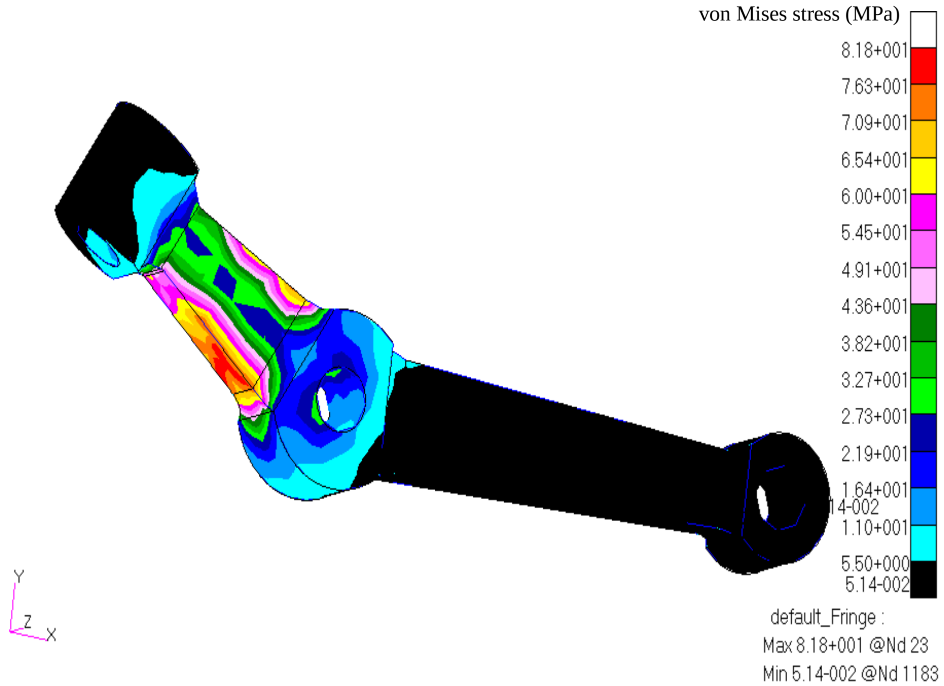


(b) TET10

Figure 4.4: Finite Element Modeling with different meshing



(a) for TET10



(b) for TET4

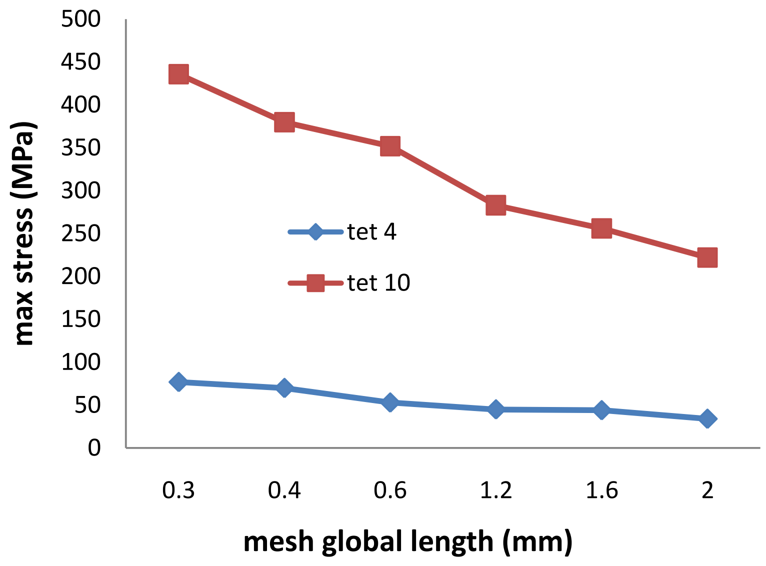
Figure 4.5 : Von Mises stress for different meshing techniques

Table 4.1: Variation of stresses concentration at the critical location of the suspension arm for TET10 mesh

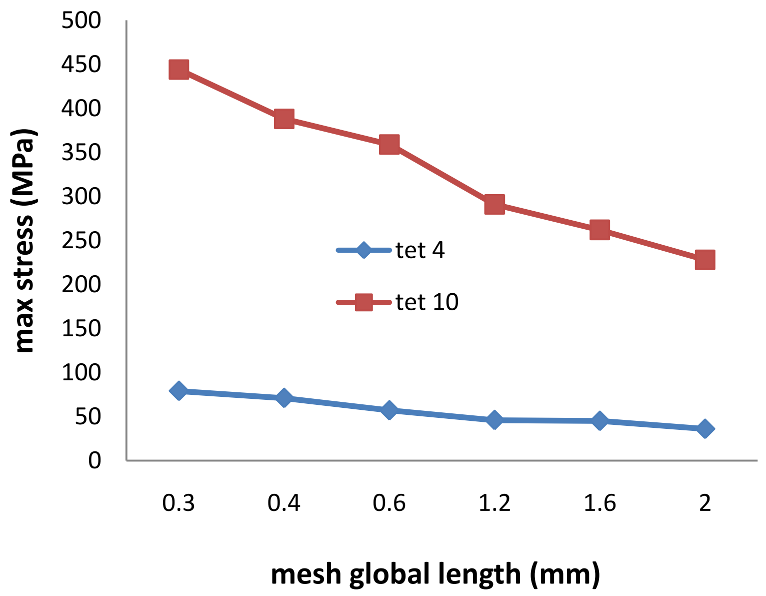
Mesh Size (mm)	Number of Node	Number of Element	von Mises (MPa)	Tresca (MPa)	max principal stress (MPa)	Displacement (mm)
0.3	9466	5585	436	444	449	0.065
0.4	7675	4418	380	388	396	0.056
0.6	6659	3836	352	359	363	0.049
1.2	5298	3067	283	291	313	0.042
1.6	3925	2173	256	262	286	0.041
2.0	2414	1276	222	228	269	0.038

Table 4.2: Variation of stresses concentration at the critical location of the suspension arm for TET4 mesh

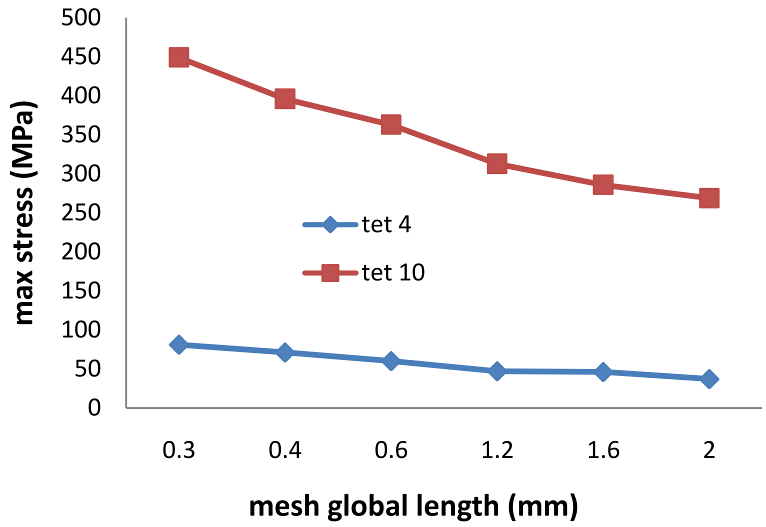
Mesh Size (mm)	Number of Node	Number of Element	von Mises (MPa)	Tresca (MPa)	max principal stress (MPa)	Displacement (mm)
0.3	1467	5634	77	79	81	0.016
0.4	1204	4404	70	71	71	0.013
0.6	1044	3827	53	57	60	0.011
1.2	820	3019	45	46	47	0.010
1.6	622	2132	44	45	46	0.009
2.0	396	1248	34	36	37	0.007



(a) Von Mises stress



(b) Tresca stress



(c) Maximum stress

Figure 4.6: Variation of stress against global mesh edge length

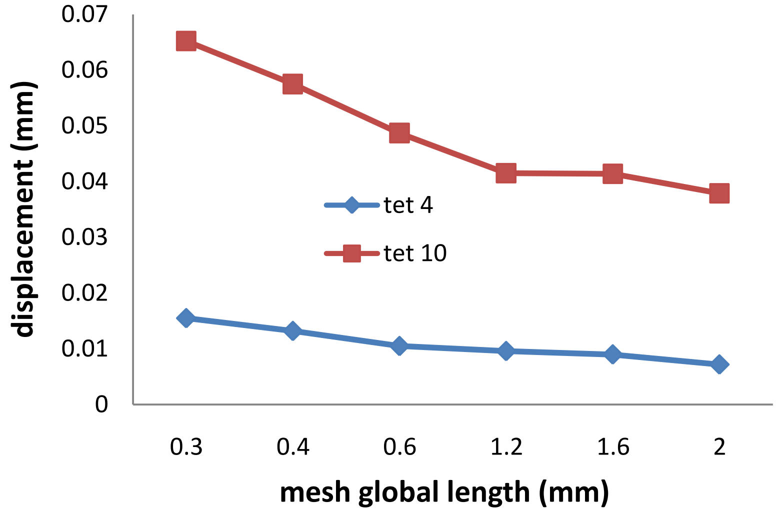


Figure 4.7: Variation of displacement with global edge length of mesh

4.4 IDENTIFICATION OF MESH CONVERGENCE

The convergence of the stress was considered as the main criteria to select the mesh type. The finite element mesh was generated using the TET10 for various mesh global length as shown in Table 4.3. Figure 4.8 shows the predicted result of stresses at the critical location of the suspension arm. It can be seen that the smaller the mesh size capture the higher predicted stresses. It is seen from Figure 4.8 that the maximum principal stresses is suitable for fatigue analysis. It can be seen that mesh size of 0.3 mm (5585 elements) has obtained the maximum stresses. The smaller mesh size than 0.3mm is not implemented due the limitation of computational time (CPU time) and storage capacity of the computer. Therefore, the maximum principal stress based on TET10 at 0.3 mm mesh size is used in the fatigue life analysis cause of the stress is higher compared to Von Mises and Tresca principal stress.

Table 4.3: Variation of mesh size related to number of element and node for TET10

Mesh size (mm)	Number of node	Number element
0.3	9466	5585
0.4	7675	4418
0.6	6659	3836
1.2	5298	3067
1.6	3925	2173
2.0	2414	1276

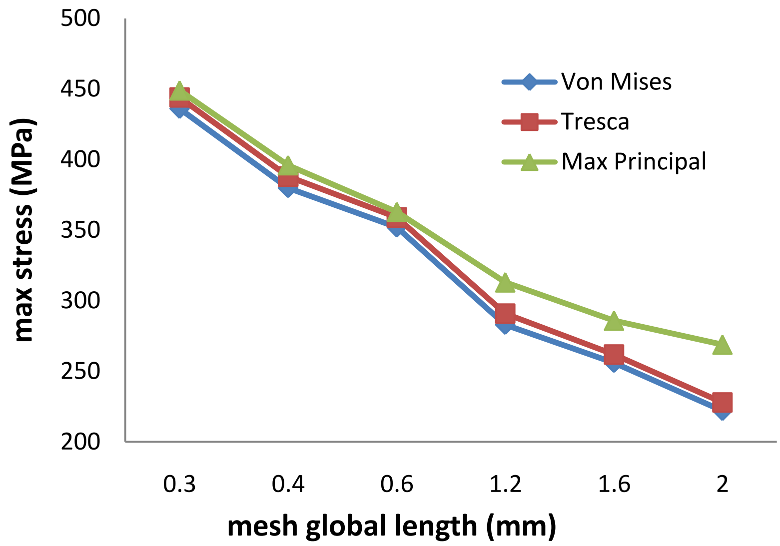


Figure 4.8: Stresses concentration versus mesh size at critical location of suspension arm for TET10 to check mesh convergence

4.5 LINEAR STATIC STRESS ANALYSIS

The linear static analysis was performed using MSC. NASTRAN software to determine the stress and strain results from the finite element model. The material utilized in this work consists of a linear elastic and isotropic material. Model loading consists of the applied mechanical load. From the analysis, the fillet of the bushing is found to experience the largest stresses. Hence, the result of the maximum principal stresses is used for the fatigue life analysis. The maximum principal stresses of the suspension arm for the linear static stress analysis is shown in Figure 4.9 for 5083-87 Aluminum alloy. From the result, the maximum principal stresses of 449 MPa occurred at node 8408.

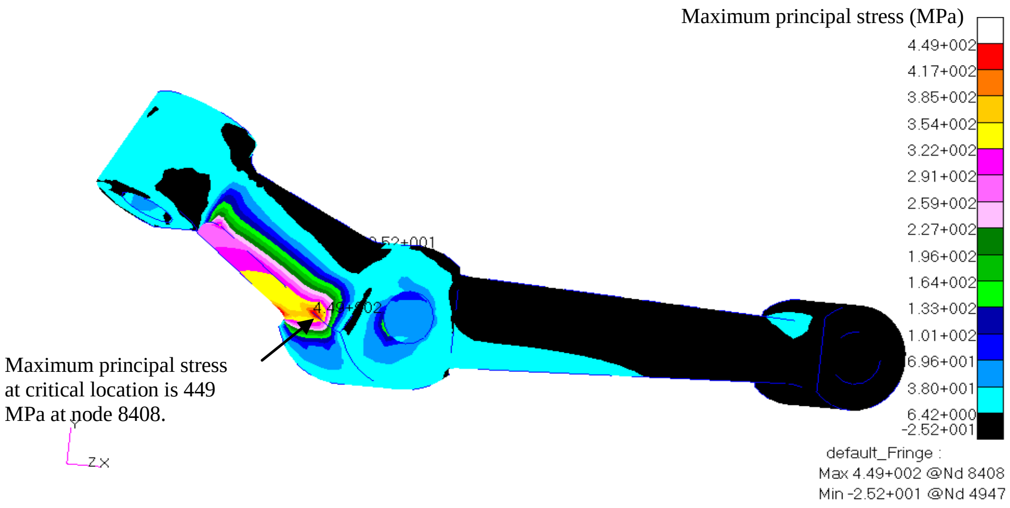


Figure 4.9: Maximum principal stresses contour plotted for 5083-87 aluminum alloy with SAETRN loading

4.6 FATIGUE ANALYSIS

The fatigue life of the suspension arm is initially predicted using 5083-87 aluminum alloy with SAETRN loading using the stress-life and strain-life method. This analysis is focused on the critical location at node of 8408. The fatigue life is expressed in cycle for the variable amplitude loading. It is observed that the fillet of middle bushing that connected to absorber is the most critical location of the suspension arm. This analysis is done to determine the fatigue life using the stress-life and strain-life method based on various variable amplitude loading time histories such as SAERTN (positive mean loading), SAESUS (compressive mean loading) and SAEBRKT (zero mean loading) as shown in Table 4.4 and Table 4.5 respectively. It can be observed from Table 4.4 and 4.5 that, the SAESUS loading histories gives the higher life compared to SAETRN and SAEBKT loading histories for different mean stress correction method. Goodman method is more conservative method compared to Gerber and no mean stress correction method for stress-life method while Smith-Watson Topper (SWT) mean stress correction method is more conservative method compared to Coffin-Manson and Morrow method for strain life method. Thus, Goodman and Smith-Watson Topper correction method are used for the material optimization. The distribution of fatigue life in term of log of life (cycle) contour plotted for 5083-87 aluminum alloy using stress-life and strain-life with no mean stress correction method based on SAETRN (positive mean loading) are showed in Figure 4.10 and Figure 4.11

respectively. It can be seen that the minimum predicted fatigue life for the suspension arm is $10^{3.36}$ cycles using stress life method while $10^{2.88}$ cycles using strain life method.

Table 4.4: Fatigue life at critical location of node (8408) for various loading histories for 5083-87 by using stress-life method

Mean Stress Correction Method	Prediction fatigue life (cycles)		
	SAETRN	**SAESUS**	**SAEBKT**
No correction	2290	11481	389
Goodman	251	588	195
Gerber	1288	10715	426

Table 4.5: Fatigue life at critical location of node (8408) for various loading histories for 5083-87 by using strain-life method

Mean Stress Correction Method	Prediction fatigue life (cycles)		
	SAETRN	**SAESUS**	**SAEBKT**
Coffin-Manson	759	3388	126
Morrow	447	2238	129
SWT	323	1514	120

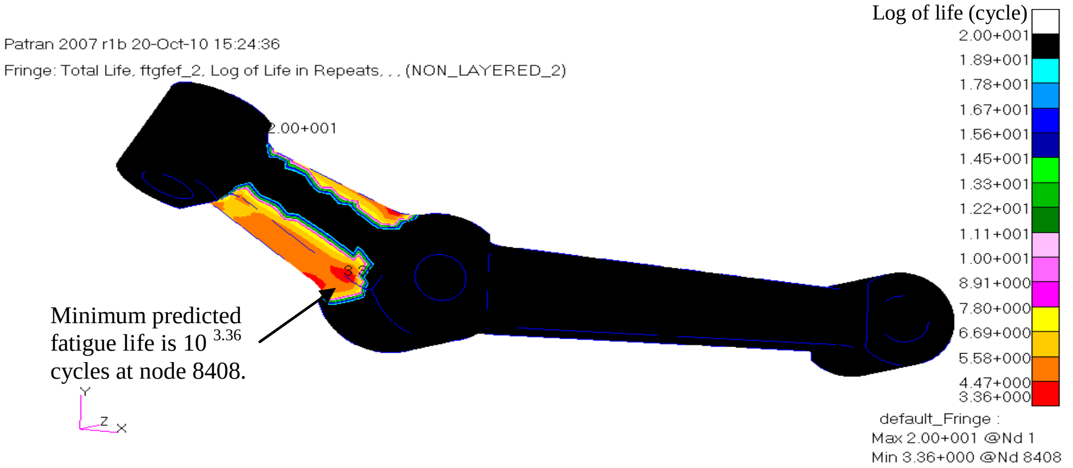


Figure 4.10: Predicted life contours plotted in term of log of life for 5083-87 aluminum alloy with SAETRN loading by using stress-life method

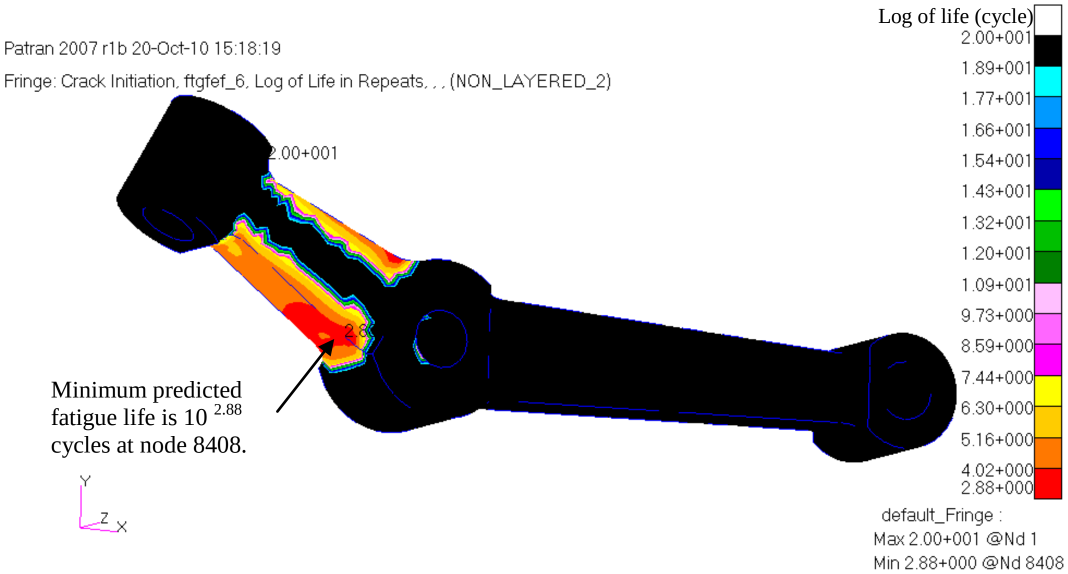


Figure 4.11: Predicted life contours plotted in term of log of life for 5083-87 aluminum alloy with SAETRN loading by using strain-life method

4.7 MATERIAL OPTIMIZATION

The material optimization was done to determine the suitable material for the suspension arm. The comparison was made in a series of aluminum alloy such as 2014-T6, 3004-H36, 5052-H32, 5083-87, 6061-T6 and 7175-T73. The effect of mean stress correction method such as Goodman mean stress correction and Smith-Watson Topper (SWT) mean stress correction are considered to improve the fatigue life. The implementation of these optimizations is to find out which is the better method to improve the fatigue life of the suspension especially at the critical location. The results of the material optimization based on various loading histories are shown in Table 4.6.

From Table 4.6, 7175-T73 aluminum alloy has the higher life compared to other materials based on SAETRN loading histories at the critical location of node (8408). Goodman method is more conservative method compared to SWT method for SAETRN and SAESUS loading condition while SWT method is more conservative in SAEBKT loading histories. Then less life is predicted using the variable amplitude loading of SAEBKT compared to SAETRN and SAESUS time histories. Thus it can be conclude that the minimum predicted fatigue life at the critical location at node (8408) of the suspension is strongly related to variable amplitude loading. The acquired results show that, 7175-T6 gives higher fatigue life for the suspension arm.

Table 4.6: Comparison between the different materials for various loading time histories using stress-life and strain-life method

Loading Condition	Prediction fatigue life at critical location (cycles)					
	SAETRN		SAESUS		SAEBKT	
Materials (Al Alloys)	Goodman	SWT	Goodman	SWT	Goodman	SWT
2014-T6	1047	537	5370	2238	389	316
3004-H36	199	472	482	873	120	94
5052-H36	Broken	128	Broken	794	Broken	28
5083-87	251	323	588	1514	194	120
6061-T6	Broken	457	9.55	2089	7.74	141
7175-T73	8511	2630	61659	11748	10471	2041

4.8 CONCLUSION

The material information used on monotonic and cyclic behaviors is well presented in this chapter. The finite element modeling and analysis of suspension arm has also been presented. The stress-life and strain-life method has been used to account for the mean stress effect include the effect of mean stress has been investigated. Detailed fatigue element models were discussed and modal analysis was performed to determine the dynamic characteristic. The summary of the finding will be present in the next chapter and also recommendations.

CHAPTER 5

CONCLUSION

5.1 INTRODUCTION

This chapter summarized the conclusion and recommendations for the overall findings of the project based on finite element analysis.

5.2 CONCLUSIONS

Stress-life and strain-life approach based on finite element of the fatigue life prediction of the suspension arm are presented. From the analysis conducted, several conclusions can be drawn as follows.

(i) Prediction of the fatigue life is focused on critical location of node 8408.

(ii) Goodman method is conservative method when subjected to SAETRN and SAESUS loading histories while SWT method is conservative in SAEBKT loading histories.

(iii) Stress-life method is capable to give higher fatigue life when subjected to bracket mean loading (SAEBKT) while strain-life method is capable to give higher fatigue life when subjected to positive mean loading (SAETRN) and negative mean loading (SAESUS).

(iv) No design modification is made on structural model of the suspension arm.

(v) 7175-T73 is suitable material compared to others material in the optimization.

5.3 RECCOMMENDATIONS

For further research, the experimental works under controlled laboratory conditions should be done to determine the validation of the result from the software analysis. Besides, the dimension of the structural model of the suspension arm should be modified to get the significant result during the experiment. 7175-T73 aluminium alloy should be considered as the suitable material for the fabrication of the suspension arm.

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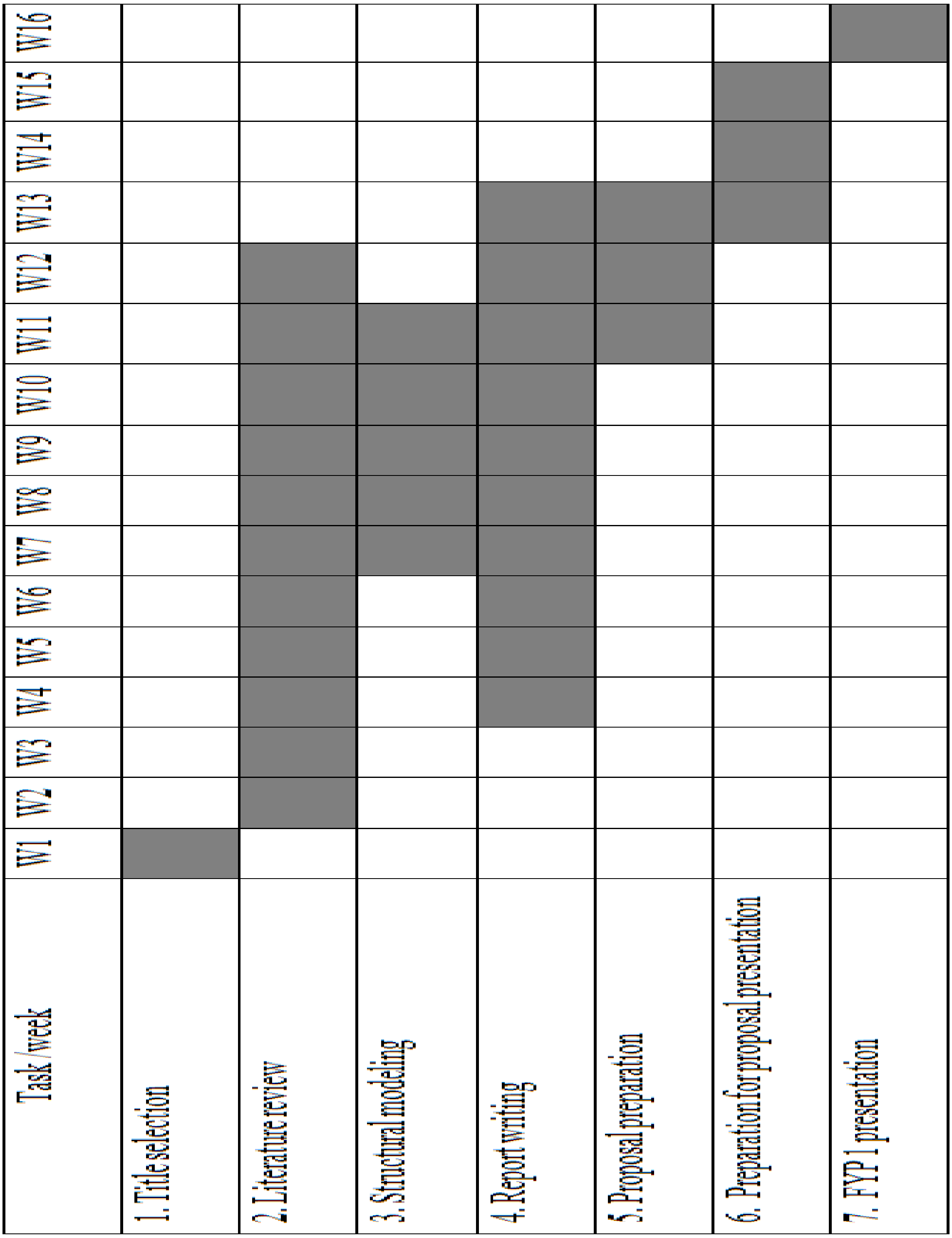
APPENDIX A

GANTT CHART FOR FINAL YEAR PROJECT 1



APPENDIX B

GANTT CHART FOR FINAL YEAR PROJECT 2