AN INVESTIGATION INTO EFFECT OF NITRIDING ON FATIGUE LIFE OF PISTON

HILMEE BIN MAT JUSOH

Report submitted in partial fulfillment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering with Automotive Engineering

Faculty of Mechanical Engineering
Universiti Malaysia Pahang

NOVEMBER 2008

SUPERVISOR'S DECLARATION

I hereby declare that i have checked this project and in my opinion this project is satisfactory in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Manufacturing.

Signature:
Name of Supervisor: DR. MD. MUSTAFIZUR RAHMAN
Position: SENIOR LECTURER
Date: 14 NOVEMBER 2008

PANEL'S DECLARATION

I hereby declare that i have checked this project and in my opinion this project is satisfactory in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering

Signature	: ……………………………………..
Name of Panel	: MR LEE GIOK CHUI
Position	: LECTURER
Date	: 14 NOVEMBER 2008

STUDENT'S DECLARATION

I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

Signature:

Name: HILMEE BIN MAT JUSOH

ID Number: MH05004

Date: 14 NOVEMBER 2008

Dedicated to my beloved

Mother and Father

For their endless support in term of motivation,

Supportive and caring as well throughout the whole project

ACKNOWLEDGEMENTS

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Not forgetting my lovely mother and father in giving me lots of supports in the aspects of moral, social and financial during my degree. This project definitely not exists without full encouragement from them.

ABSTRACT

The piston is the crucial part of the internal combustion engine. Prediction of fatigue life on piston for four stroke engine using variable amplitude loading is presented. The objectives of this project are to predict fatigue life of piston for four stroke engine using stress-life method, to identify the critical locations, to investigate the effect of mean stress and to optimize the component material. The structural and finite element modelling has been performed using a computer aided design and finite element analysis software package. The finite element model of component then analyzed using the *S-N* approach. Finally, the stress-strain state of component obtained previously will be use as input for the fatigue life. The effected mean stress and materials optimize was also investigated. The failure of piston can result in devastating damage to the engine including all the components from a tiny screw till a huge belting system. Life of piston needs to be improved to prevent from any unpleasant problems. The result of the analysis show that there are no serious failure occurs at the part of the piston. However it is observed that the minimum predicted life at the critical location is $10^{2.38e-7}$ under variable amplitude loading for stress life approach. The optimization result show that AISI 4042 is the most superior material among the others.

ABSTRAK

Piston ialah bahagian yang paling penting sekali di dalam sesebuah enjin. Ramalan jangka hayat bagi piston enjin empat lejang dipersembahkan. Objektif untuk projek ini ialah untuk meramal jangka hayat bagi piston untuk enjin empat lejang, untuk mengenal pasti lokasi kritikal yang terdapat pada piston, untuk menyiasat hubungan dan kesan keterikan purata daripada jangka hayat keterikan dan untuk mengoptimumkan pemilihan bahan. Model struktur dan elemen finiti telah dibuat menggunakan perisisan lukisan secara berkomputer dan analisis elemen finiti. Model bagi elemen finiti tersebut kemudian dianalisa menggunakan pendekatan *S-N* . Akhir sekali, keterikan dan ketegasan bagi komponen tersebut yang telah dicapai akan digunakan sebagai input untuk jangka hayat lesu. Kesan keterikan purata dan bahan yang optimum akan diperiksa. Kegagalan piston untuk berfungsi dengan baik akan menyebabkan kerosakan teruk kepada enjin termasuklah semua komponen samada dari sekecil bahagian seperti skru hinggalah kepada system enjin. Jangka hayat piston perlulah diperbaiki untuk mengelakkan daripada berlakunya masalah-masalah yang tidak diingini. Keputusan analisa menunjukkan tiada kegagalan yang serius berlaku pada mana-mana bahagian piston. Walau bagaimana pun daripada pemerhatian menunjukkan jangka hayat paling minima di lokasi kritikal ialah $10^{2.38e-7}$ dibawah tindakan amplitud yang berubah untuk hayat keterikan. Akhir sekali, bahan yang terbaik hasil daripada analisa ialah AISI 4042 berbanding dengan baha-bahan lain.

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LIST OF SYMBOLS

C_p	Specific heat of constant pressure
C_v	Specific heat of constant volume
N	Engine speed
P	Pressure
P	Power
Q_{HHV}	Higher heating value
Q_{LHV}	Lower heating value
R	Gas constant
T	Temperature
T_o	Standard temperature
V	Cylinder volume
V_{BDC}	Cylinder volume at bottom dead center
V_d	Displacement volume
V_{TDC}	Cylinder volume at top dead center
W	Work

k	Ratio of specific heats
m	Mass
m_a	Mass of air
m_f	Mass of fuel
m_m	Mass of gas mixture
n_r	Number of crank revolutions
q	Heat transfer per unit mass
r_c	Compression ratio
v	Specific volume
v_{BDC}	Specific volume at bottom dead center
v_{TDC}	Specific volume at top dead center
w	Specific work
η_t	Thermal efficiency
ρ	Density

CHAPTER 1

INTRODUCTION

1.1 Background

Every engineering product is not just made or is made, but it must follow the sequence step to achieve good product. The steps are design, manufacture and lastly perform specific function to human needs. Design engineering is actually a decision-making process to developed or improved product at a reasonable cost. Parameters included in such an analysis are:

- Geometry and dimensions of different parts,
- Types of materials used in manufacturing and their specifications,
- Fabrication and assembling techniques
- Service conditions.

In real life to developed a product, high cost is need because the product must follow a few testing using a few materials and a few design to get a good product. Because of that, engineers nowadays use a modern computational approach based on finite element analysis to reduce the testing cost. This paper describes the finite element analysis techniques to predict the fatigue life and identify the critical locations of the piston that have been treatment using nitriding process. The finite element modeling and analysis has been performed using a computer-aided design and a finite element analysis software package, and the fatigue life prediction was carried out using finite element based fatigue life prediction codes. The results showed the contour plots of the fatigue life histogram and damage histogram at the most critical location.

1.2 Problem Statement

During the power stroke, up to 18000 N of force is suddenly applied to the piston head. This happens 30 or 40 times a second to each piston at highway speed. The fatigue life of piston must be taken seriously. Thus method for predicting fatigue life of a piston using stress-life method are proposed

1.3 Objectives of Project

The objectives of this project are as follows;

i. To predict fatigue life of piston for four stroke engine using stress-life method and to identify critical locations
ii. To investigate the effect of mean stress
iii. To optimize the material of the component

1.4 Scopes of Project

The scopes are dividing into 4:

i. Structural modeling of piston using SolidWork software
ii. Finite element modeling (FEM)
iii. Finite element analysis
iv. Fatigue analysis using stress life approach
v. Optimization

1.5 Outline of Report

Chapter 1 is explanation about introduction of the project. In this chapter also include the objectives and scope of the project. Chapter 2 discusses on the literature review of piston, nitriding treatment and S-N method. Chapter 3 provides the project methodology for analysis the fatigue life of the piston. The methodology consists of piston model using SolidWork software, and all the analysis included finite element analysis, fatigue analysis and optimization using MSC.Patran and MSC.Nastran. Chapter 4 discuss about the fatigue life of piston and optimization of the results. Finally, Chapter 5 concludes the result of analysis. The recommendation is provided for future research.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

This chapter provided the review of the past research related to the fatigue analysis, finite element analysis and nitrided treatment on the fatigue life. The past research effort can be properly guided to justify the scope and direction of the present effort.

2.2 Case Study

Firstly is to choose the type of the piston four stroke engines as a case study. GN5 100cc four stroke engine types or EX5 dream will be selected to running this analysis. For this engine type it used air cooled system and, 4 cycle engine overhead camshaft (OHC). Other wise this engine type is has compress ratio about 9: 1 and the maximum torque is about 0.93 kg.m / 6000 rpm. The typical example of the piston as shown on Figure 2.1.

Figure 2.1: Typical piston for four stroke engine

2.3 Otto Cycle

The cycle of a four-stroke, spark ignition (SI) naturally aspirated engine at wide open throttle (WOT) is shown Figure 2.2. This is the cycle of most automotive engine and other four-stroke SI engines.

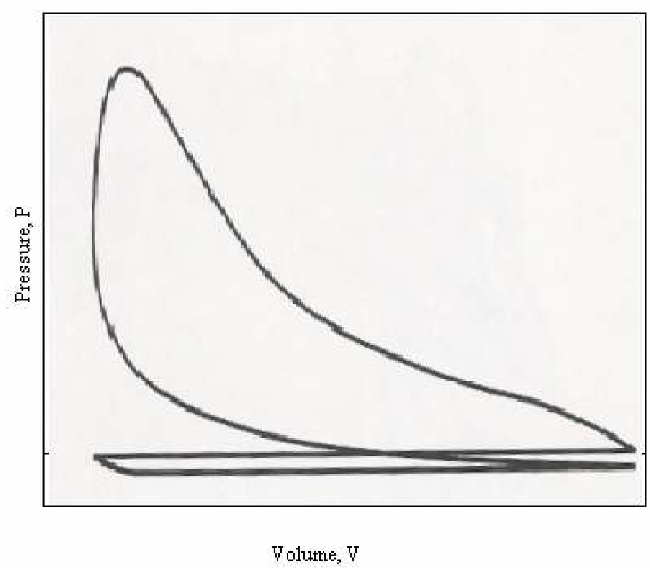


Figure 2.2: Indicator diagram for a typical four stroke cycle SI engine

The intake stroke of the Otto cycle starts with the piston at top dead center (TDC) and is a constant-pressure process at an inlet pressure of one atmosphere. This is good approximation to the inlet process of a real engine at WOT, which will actually be at a pressure slightly less than atmospheric due to pressure losses in the inlet air flow. The temperature of the air during the inlet stroke is increased as the air passes through the hot intake manifold. The temperature at point 1 will generally be on the order of 25º to 35ºc hotter than the surrounding air temperature. The process will show in Figure 2.3.

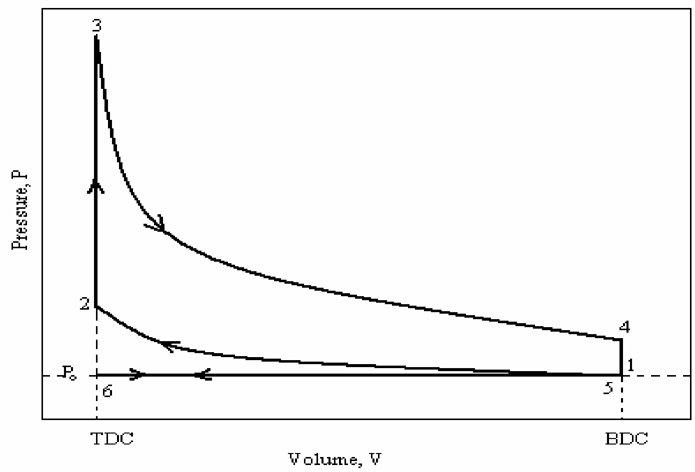


Figure 2.3: Ideal air-standard Otto cycle

The second stroke of the cycle is the compression stroke, which in the Otto Cycle is an isentropic compression from bottom dead center (BDC) to TDC (process 1-2). In this process, the air/fuel mixture will compress. The end of compression is affected by the firing of the spark plug before TDC.

The compression stroke is followed by a constant-volume heat input process 2-3 at TDC. This replaces the combustion process of the real engine cycle, which occurs at close to constant- volume conditions. During combustion or heat input, a large amount of energy is added to the air within the cylinder. This energy raises the temperature of the air to very high values, giving peak cycle temperature at point 3.

This increase in temperature during a closed constant- volume process result in a large pressure rise also.

The very high pressure and enthalpy values within the system at TDC generate the power stroke which follows combustion. High pressure on the piston face forces the piston back towards BDC and produces the work and power output of the engine.

The last stroke of the four- stroke cycle now occurs as the piston travels from BDC to TDC. Process 5-6 is the exhaust stroke that occurs at a constant pressure of one atmosphere due to the open exhaust stroke, which occurs at a pressure slightly higher than the surrounding pressure due to the small pressure drop across the exhaust valve and in the exhaust system.

2.3.1 Temperature Analysis

In air-standard cycles, air is considered an ideal gas such that the following ideal gas relationships can be used:

$$Pv = \mathrm{RT} \tag{2.1}$$

$$PV = \mathrm{RT} \tag{2.2}$$

$$P = \rho \mathrm{RT} \tag{2.3}$$

Process 6-1 constant-pressure intake of air at P_o

Intake valve open and exhaust valve closed:

$$P_1 = P_6 = P_o \tag{2.4}$$

$$w_{6-1} = P_o\ (v_1 - v_2) \tag{2.5}$$

Process 1-2 compression stroke

All valve closed:

$$T_2 = T_1 \ (v_1/v_2)^{k-1} = T_1 \ (V_1/V_2)^{k-1} = T_1 \ (r_c)^{k-1} \qquad (2.6)$$

$$q_{1-2} = 0$$

k is specific heat ratio C_p/C_v

k = 1.4

r_c is compression ratio $V_1/V_2 = v_1 - v_2$

$$w_{1-2} = (P_2 v_2 - P_1 v_1) \ / \ (1-k) = R \ (T_2 - T_1) / \ (1-k) \qquad (2.7)$$

Process 2-3 constant-volume heat input (combustion).

All valves closed:

$$v_3 = v_2 = v_{TDC}$$

$$w_{2-3} = 0$$

$$Q_{2-3} = Q_{in} = m_f Q_{hv} \eta_c = m_m C_v (T_3 - T_2) \qquad (2.8)$$

$$= (m_a + m_f) C_v (T_3 - T_2)$$

$$Q_{hv} \eta_c = (AF + 1) \ C_v (T_3 - T_2) \qquad (2.9)$$

$$q_{2-3} = q_{in} = C_v (T_3 - T_2)$$

$$T_3 = T_{max}$$

$$P_3 = P_{max}$$

Process 3-4 power or expansion stroke.

All valves closed:

$$q_{2-3} = 0$$

$$T_4 = T_3 (v_3/v_4)^{k-1} = T_3 (V_3/V_4)^{k} = T_3 (1/r_c)^{k-1}$$

$$P_4 = P_3 (v_3/v_4)^{k} = P_3 (V_3/V_4)^{k} = P_3 (1/r_c)^{k-1}$$

$$w_{3-4} = (P_4 v_4 - P_3 v_3)/(1 - k) = R(T_4 - T_3)/(1 - k)$$

$$= C_v (T_3 - T_4) \qquad (2.10)$$

Process 4-5 constant-volume heat rejection (exhaust blowdown).

Exhaust valve open and intake valve closed:

$$v_5 = v_4 = v_{BDC}$$

$$w_{4-5} = 0$$

$$Q_{4-5} = Q_{out} = m_m C_v(T_5 - T_4) = m_m C_v(T_1 - T_4)$$

$$q_{4-5} = q_{out} = m_m C_v(T_5 - T_4) = C_v(T_1 - T_4) \qquad (2.11)$$

Process 5-6 constant-pressure exhaust stroke at P_o

Exhaust valve open and intake valve closed:

$$P_5 = P_6 = P_o$$

$$w_{5-6} = P_o (v_6 - v_5) = P_o (v_6 - v_1) \qquad (2.12)$$

Thermal efficiency of Otto cycle:

$$(\eta_t)_{\text{OTTO}} = |w_{net}|/|q_{in}| = 1 - (|q_{out}|/|q_{in}|)$$

$$= 1 - [C_v(T_1 - T_4)/\ C_v(T_3 - T_2)]$$

$$= 1 - [(T_4 - T_1)/\ (T_3 - T_2)] \qquad (2.13)$$

Only cycle temperature need to be known to determine thermal efficiency. This can be simplified further by applying ideal gas relationships for the compression and expansion strokes and recognizing that $v_1 = v_4$ and $v_2 = v_3$:

$$(T_2/T_1) = (v_1/v_2)^{k-1} = (v_4/v_3)^{k-1} = (T_3/T_4) \qquad (2.14)$$

Rearranging the temperature terms gives

$$T_4/T_1 = T_3/T_2$$

$$(\eta_t)_{\text{OTTO}} = 1 - (T_2/T_1)\ \{[(T_4/T_1) - 1]/\ [(T3/T_2) - 1]\}$$

$$(\eta_t)_{\text{OTTO}} = 1 - (T_2/T_1) = 1 - [1/(v_1/v_2)^{k-1}]$$

$$(\eta_t)_{\text{OTTO}} = 1 - (1/r_c)^{k-1}$$

$$(\eta_t)_{\text{OTTO}} = W_{net} \qquad (2.15)$$

Only the compression ratio is needed to determine the thermal efficiency.

2.4 Fatigue Damage Mechanism

Fatigue is a localized damage process of a component produced by cyclic loading. It is the result of the cumulative process consisting of crack initiation, propagation and final fracture of a component. During cyclic loading, localized plastic deformation may occur at the highest stress site. This plastic deformation induces permanent damage to the component and a crack develops. As the component experiences an increasing number of loading cycles, the length of the crack increase. After a certain number of cycles, the crack will cause the component to fail.

2.5 Stress-life (S-N) and Fatigue Limit Testing

Between 1852 and 1857, August Wohler has contribute his study about *S-N* fatigue test in which constant-amplitude stress cycle with a specific mean stress level are applied to test specimens are sometimes called classical Wohler Test. These test are the most commons type of fatigue testing. From these test, it is possible to develop *S-N* curves that represent the fatigue life behavior of a component or of a material test specimen.

There is a need for statistical *S-N* testing to predict fatigue life at various stress amplitude and mean stress combination. The *S-N* methods presented by The Japan Society Of Mechanical Engineers (1981) are widely used by researchers for *S-N* testing and fatigue life prediction.

The median *S-N* test method with a small sample size can be used as a guideline to determine an *S-N* curve with a reliability of 50% and a minimum sample size. This method requires 14 specimens. Eight specimens are used to determine the finite fatigue life region, and six specimens are used to find the fatigue limit. The curve for the finite life region is determined by testing two samples at each of four different levels of stress amplitude and the fatigue limit is tested by the staircase method with six specimens. The recommended test sequence is shown is Figure 2.4, in which the number next to the data point represents the order in which the specimen is to be tested. The finite life region data is assumed linear in the log-log coordinates, and the data are analyzed by the least-squares method. The fatigue limits is determined by taking the average of the stress levels in a staircase test.

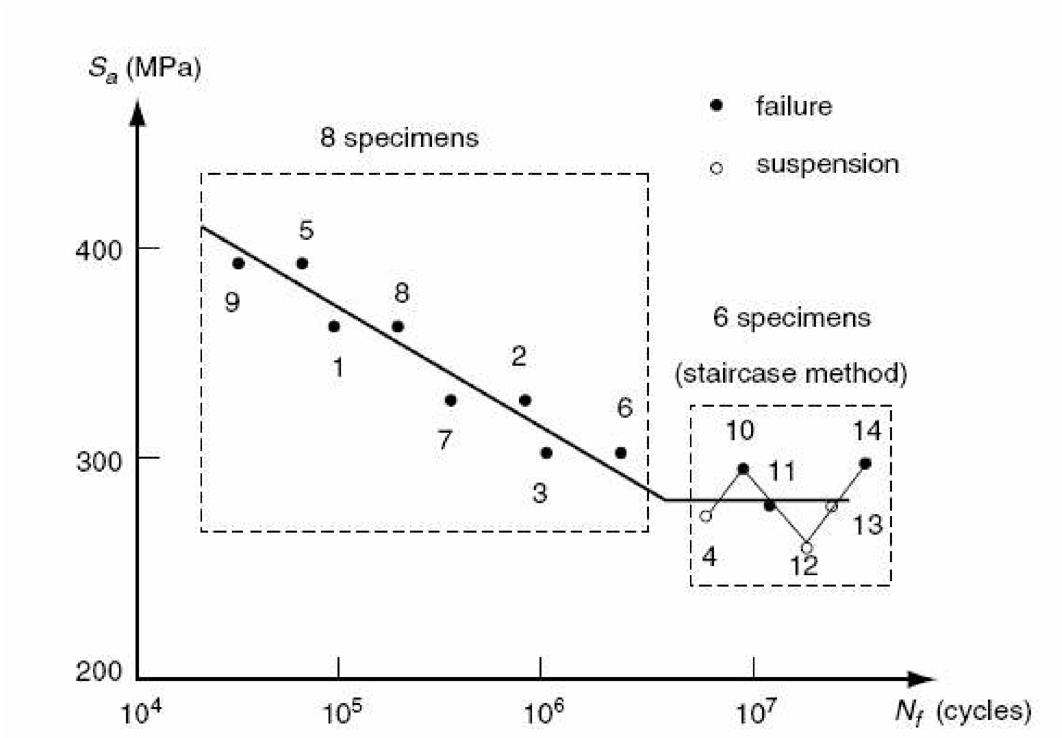


Figure 2.4: S-N testing with a small sample size

The guidelines for the generation of statistical S-N curves are documented elsewhere (Wirshing, 1983; Shen, 1994). It is recommended that more than one specimen be tested at each stress level. Test with more than one test sample at a given stress amplitude level are called test with replicate data. Replicate test are required to estimate the variability and the statistical distribution of fatigue life. Depending on the intended purpose of the *S-N* curve, the recommended number of samples and number of replicated test vary. The recommended sample size for the number of test samples used to generate the *S-N* curve is:

- 6 – 12 for preliminary and research and development tests
- 12-24 for design allowable and reliability tests

The percent replication (PR), based on the numbers of stress levels (L) and a sample size (n_s) is defined as follows:

$$PR = 100(1-L/n_s) \tag{2.16}$$

The percent replication value indicates the portion of the total number of specimens tested that may be used for determining an estimated of the variability of replicate test. These guidelines recommend that the percent replication for various test be

- 17-33 for preliminary and exploratory tests
- 33-50 for research and development tests
- 50-75 for design allowable data tests
- 75-88 for reliability tests

2.5.1 Analysis Of Fatigue Data In Finite Life Region

Once fatigue life data in the finite life region has been collected from *S-N* test, the least-squares method for generating a line of best fit from the data is recommended. For the statistical analysis of the fatigue data, this method of generating a line of best fit is feasible because the data can be represented as a straight line on a log-log plot of stress amplitude versus reversals to failure. It is assumed that the fatigue life resulting at a given stress amplitude level follows the lognormal distribution and that the variance of log life is constant over the tested range. The assumption of constant variance for all stress levels is referred to in statistic as the assumption of homoscedasticity. The least-squares regression model is

$$Y = A + BX + \varepsilon \tag{2.17}$$

where ε is a random variable for error. The regression line is

$$\hat{Y} = \hat{A} + \hat{B}X \tag{2.18}$$

Where the estimated values of Â and B are obtained by minimizing the sum of the square of the deviations of the observed values of Y from those predicted

$$\Delta^2 = \sum_{i=1}^{n_s} (Y_i - \hat{Y}_i)^2 = \sum_{i=1}^{n_s} (Y_i - \hat{A} - \hat{B}X_i)^2 \tag{2.19}$$

Where the number of test sample is represented by the variable n_s. This leads to the following :

$$\begin{aligned} \frac{\partial \Delta^2}{\partial \hat{A}} &= \sum_{i=1}^{n_s} 2(Y_i - \hat{A} - \hat{B}X_i)(-1) = 0 \\ \frac{\partial \Delta^2}{\partial \hat{B}} &= \sum_{i=1}^{n_s} 2(Y_i - \hat{A} - \hat{B}X_i)(-X_i) = 0 \end{aligned} \tag{2.20}$$

From these, the least-squares method estimates B and Â as follows

$$\hat{B} = \frac{\sum_{i=1}^{n_s} (X_i - \overline{X})(Y_i - \overline{Y})}{\sum_{i=1}^{n} (X_i - \overline{X})^2}$$

$$\hat{A} = \overline{Y} - \hat{B}\overline{X} \tag{2.21}$$

Figure 2.5 shows the comparison between the predicted median *S-N* curve and the experimental fatigue data in a semilog scale. The limitations of the predicted *S-N* equation are as follows:

1. Only valid for the test data presented (i.e., effective life range from 9.8×10^3 to 9.0×10^7 reversals).
2. Only valid for the same failure mode due to rotating bending testing.
3. Representative of the median fatigue data.
4. Lognormal distribution of N_f is assumed.
5. Variance of N_f is considered constant for all N_f.

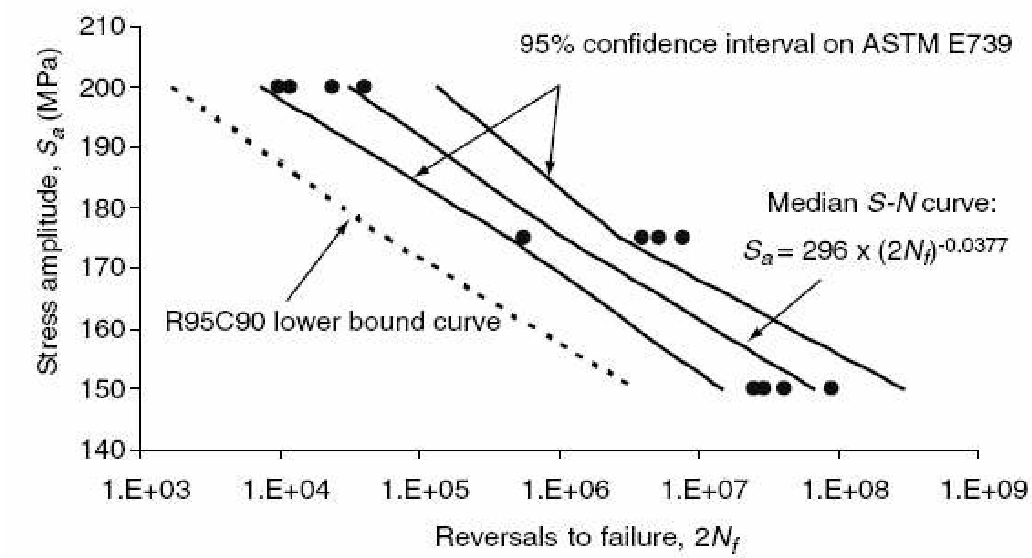


Figure 2.5: Predicted and Experimental *S-N* data

2.5.2 Design S-N Curves In Finite Life Region

This section presents a practical method for constructing a design S-N curve that characterizes the minimum fatigue life at a given fatigue strength level so that majority of the fatigue data fall above the minimum or the lower bound value. A schematic representation of the design *S-N* curve is given in Figure 2.6. The choice of the lower-bound *S-N* curve is fairly arbitrary and dependent on material cost, safety policy and industry standards. For example if a value of R95C90 is used for component designs, this particular values ensures that there is a 95% possibility of survival (reliability) with a 90% of confidence level for a fatigue life at a specified stress level. Because a sample size is usually limited, the 90% confidence level is introduced to ensure that there is a 90% of possibility that the actual 95% of reliability value may be expected to fall above this lower bound.

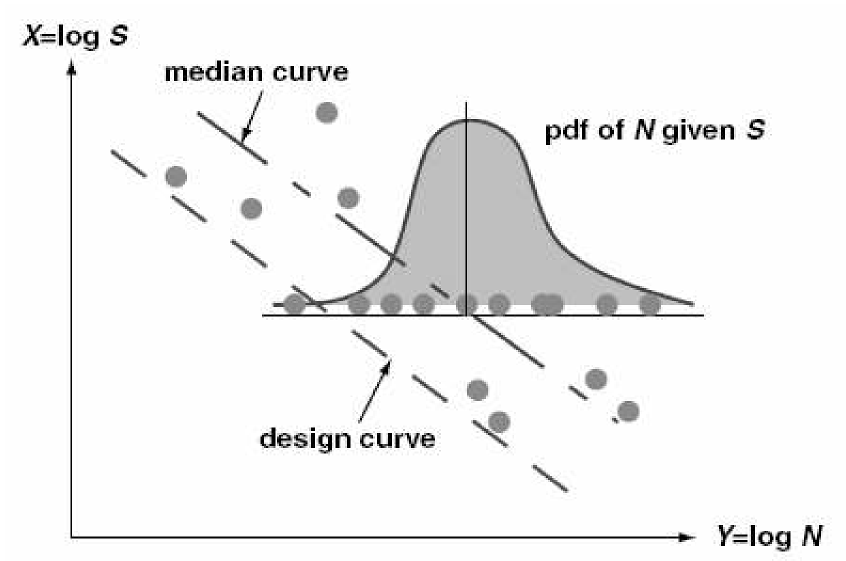


Figure 2.6: Concept of a design S-N curve

2.6 Nitriding

Nitriding is a surface-hardening heat treatment that introduces nitrogen into the surface of steel at a temperature range (500°C to 550°C), while it is in the ferrite condition. Thus, nitriding is similar to carburizing in that surface composition is altered, but different in that nitrogen is added into ferrite instead of austenite. Because nitriding does not involve heating into the austenite phase field and a subsequent quench to form martensite, nitriding can be accomplished with a minimum of distortion and with excellent dimensional control.

The mechanism of nitriding is generally known, but the specific reactions that occur in different steels and with different nitriding media are not always known. Nitrogen has partial solubility in iron. It can form a solid solution with ferrite at nitrogen contents up to about 6%. At about 6% N, a compound called gamma prime (γ'), with a composition of Fe_4N is formed.

At nitrogen contents greater than 8%, the equilibrium reaction product is ε compound, Fe_3N. Nitrided cases are stratified. The outermost surface can be all γ' and if this is the case, it is referred to as the white layer. Such a surface layer is undesirable: it is very hard profiles but is so brittle that it may spall in use. Usually it is removed; special nitriding processes are used to reduce this layer or make it less brittle. The ε zone of the case is hardened by the formation of the Fe_3N compound, and below this layer there is some solid solution strengthening from the nitrogen in solid solution.

Alloying elements commonly used in commercial steels, aluminum, chromium, vanadium, tungsten, and molybdenum are beneficial in nitriding because they form nitrides that are stable at nitriding temperatures. Molybdenum, in addition to its contribution as a nitride former, also reduces the risk of embrittlement at nitriding temperatures. Other alloying elements, such as nickel, copper, silicon, and manganese, have little, if any, effect on minding characteristics.

Although at suitable temperatures all steels are capable of forming iron nitrides in the presence of nascent nitrogen, the nitriding results are more favorable in those steels that contain one or more of the major nitride-forming alloying elements. Because aluminum is the strongest nitride former of the common alloying elements, aluminum-containing steels (0.85 to 1.50% **Al**) yield the best nitriding results in terms of total alloy content. Chromium-containing steels can approximate these results if their chromium content is high enough. Unalloyed carbon steels are not well suited to gas nitriding because they form an extremely brittle case that spalls readily, and the hardness increase in the diffusion zone is small.

Aluminum-containing steels produce a nitrided case of very high hardness and excellent wear resistance. However, the nitrided case also has low ductility, and this limitation should be carefully considered in the selection of aluminum-containing steels. In contrast, low-alloy chromium-containing steels provide a nitrided case with considerably more ductility but with lower hardness. Tool steels, such as H11 and D2, yield consistently high case hardness with exceptionally high core strength

All hardenable steels must be hardened and tempered before being nitrided. The tempering temperature must be high enough to guarantee structural stability at the nitriding temperature: the minimum tempering temperature is usually at least 30°C higher than the maximum temperature to be used in nitriding.

In certain alloys, such as series 4100 and 4300 steels, hardness of the nitrided case is modified appreciable by core hardness: that is, a decrease in core hardness results in a decrease in case hardness. Consequently, in order to obtain maximum case hardness, these steels are usually provided with maximum core hardness by being tempered at the minimum allowable tempering temperature.

Either a single- or a double-stage process may be employed when nitriding with anhydrous ammonia. In the single-stage process, a temperature in the range of about 495 to 525°C is used, and the dissociation rate ranges from 15 to 30%. This process produces a brittle, nitrogen-rich layer known as the white nitride layer at the surface of the nitrided case.

The first stage of the double-stage process is, except for time, a duplication of the single-stage process. The second stage may proceed at the nitriding temperature employed for the first; stage, or the temperature may be increased to from 550 to 565°C (1025 to 1050°F): however, at either temperature, the rate of dissociation in the second stage is increased to 65 to 80%. Generally, an external ammonia dissociator is necessary for obtaining the required higher second-stage dissociation.

CHAPTER 3

METHODOLOGY

3.1 Project Methodology

Project methodology play important role in order to analysis fatigue life of piston. It is because this project methodology becomes a guideline for the overall project progress from the beginning until finish. Before start the fatigue life analysis, the function and characteristic of the piston should be known. So, at the beginning of the project, some research and literature review about piston is important to enhance the knowledge.

Generally, this project involves of an investigation into the effect of nitriding on fatigue life of piston using high strength steel. The piston model is designed by using Solid Work software.

Then the model was analyzed using MSC.PATRAN and MSC.NASTRAN to obtain the stress and strain results. This finites element analysis result were important to determine the stress state of piston for a given load condition. For this project, the given load is 7 MPa. After that, using the FEA result, this project was continuing on fatigue analysis and lastly optimization. Figure 3.1 shows the flow chart of project methodology.

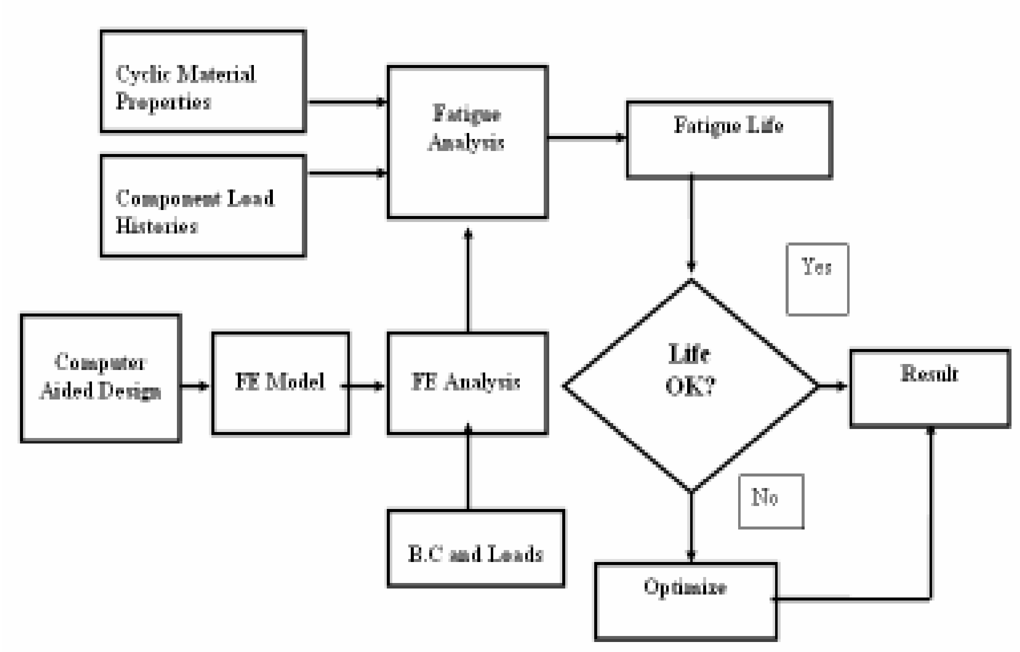


Figure 3.1: Flow chart of project methodology

3.2 Structural Component

First step was to get the dimension of the piston to draw the model. The piston height was about 45mm and diameter about 50mm also for the weight 84 gram. This piston was developed using AISI 4042. Table 3.1 below show the monotonic and cyclic properties of AISI 4042.

Table 3.1: Monotonic and cyclic properties of AISI 4042

Materials properties	Value
Young's modulus, E, GPa	205
Ultimate tensile strength, S_u, MPa	1800
Tensile Yield Strength	1662
Density	7.85
Poisson ratio	0.29
Cyclic and Fatigue Properties	
Fatigue strength coefficient, σ'_f, MPa	1700
Fatigue strength exponent, *b*	-0.087
Fatigue ductility coefficient , ε'_f	0.1637
Fatigue ductility exponent, *c*	-0.58
Cyclic strength coefficient, K', MPa	2970

3.3 Structural Modeling

From the dimension, the drawing is transfer using Solid Work software to developed the 3D model. Figure 3.2 shows the model of the piston from the Solid Work.

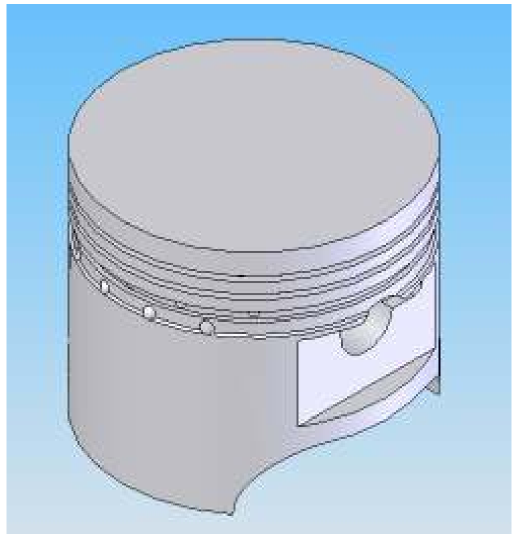


Figure 3.2: Structural model of piston

3.4 Finite Element Modeling

After design the piston, the piston model was transferred to MSC.PATRAN. For the finite element modeling, the piston was mashed using topology Tetrahedral 10 and 0.6 as the global edge length.

3.5 Finite Element Analysis

This step is to determine stress of the piston for a given load using MSC.NASTRAN. The load on top of piston is 7MPa (Rahman et al,2007) and the constraint was fixed at the piston pin hole. Figure 3.3 shows the loading and the boundary condition. The chosen material for the analysis is AISI 4042. Figure 3.3 shows the loading and boundary condition to the piston.

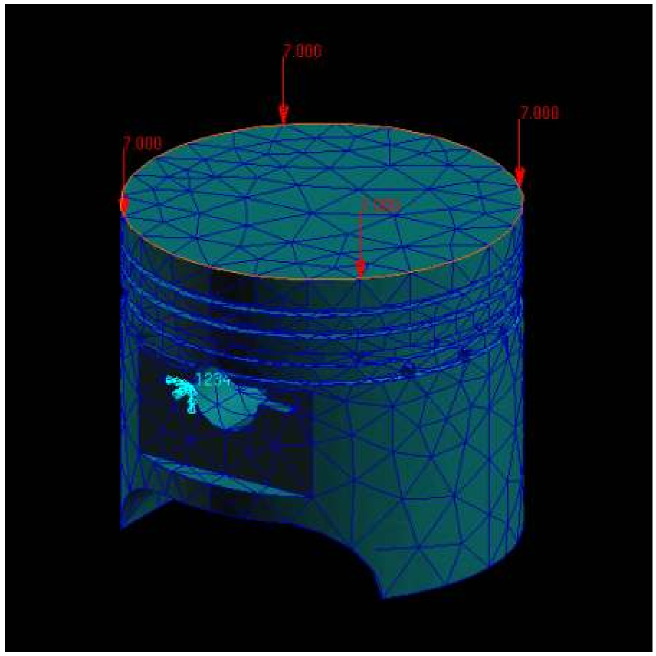


Figure 3.3: Loading and boundary condition

3.6 Fatigue Analysis

The fatigue analysis is used to compute the fatigue life of piston and identify the critical locations. The required inputs for the fatigue analysis process are shown in Figure 3.4. The three input information are descriptions of the material properties, loading histories and geometry. The details of these inputs are as follows;

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

(ii) Load histories information – measured or simulated load histories applied to a component. The term "loads" is used to represent forces, displacements, accelerations, etc

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

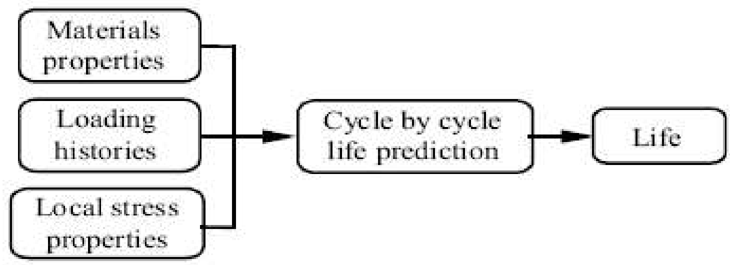


Figure 3.4: Schematic diagram of fatigue life estimation

Figure 3.5 shows that variable amplitude loading histories was applied to the piston. This because when the combustion happened, the pressure that hit the top of the piston is variable follow the movement of the piston. From the figure, maximum load is 6.993 MPa at 0 sec and minimum load is -3.465MPa at 1743.333 sec.

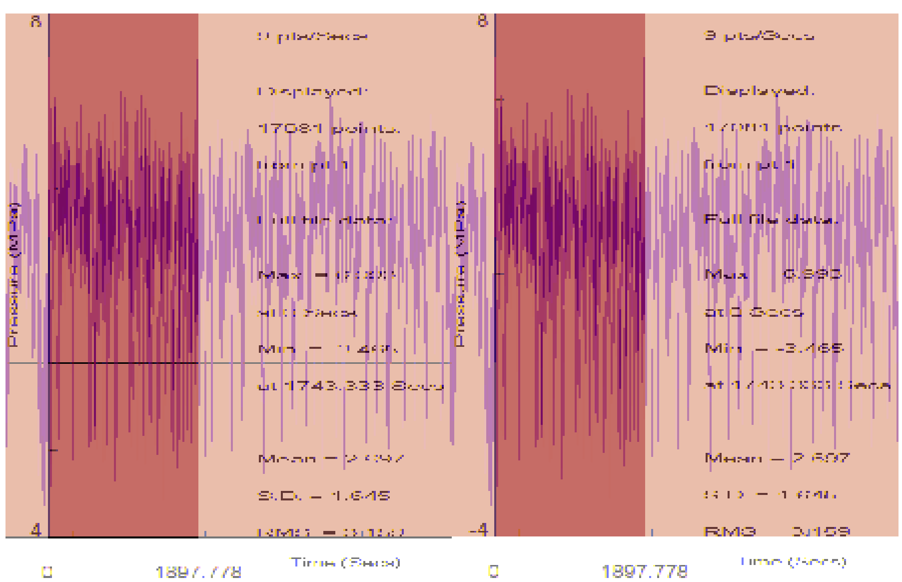


Figure 3.5: Variable amplitude loading histories

3.7 Conclusion

After all the analysis finish, the fatigue life results were optimized using other material, surface treatment and mean stress correction method.

CHAPTER 4

RESULTS AND DISCUSSION

4.0 Introduction

In this chapter will discuss about the finite element modeling result, finite element analysis result, fatigue analysis result and lastly optimization result. The finite element modeling result consist of choosing either tetrahedral 10 or tetrahedral 4 as a topology, and choosing the best global edge length to get the highest stress result. Then for the finite element analysis result consist of choosing either max principal or von misses for the highest stress result. The next analysis is fatigue analysis result. The result consists of to determine critical locations on the piston. Lastly optimization result. This result included stress life with Mean Stress Correction, stress life with material comparison and lastly stress life with comparison surface treatment.

4.1 Finite Element Modeling

Table 4.1 shows result of maximum stress between TET4, and TET10. From the result for von mises, obvious that TET 10 got the highest stress result among the TET types. The TET10 von Mises result was 519MPa and TET4 for 214MPa. For the Maximum Principal, its also show that TET10 got higher result than TET4. For the TET10 was 433 MPa and for TET4 was 102 MPa. Figure 4.1 shows graph maximum stress versus type of topology using von Mises and Maximum Principal. It's obvious that result TET10 for von Mises was higher than result TET10 for

Maximum Principal. Because of that, TET10 using von Mises was chosen for meshing.

Table 4.1: Maximum stress between TET types

Tet Type	Von Mises(MPa)	Maximum Principal(MPa)
4	2.14E+02	4.33E+01
10	5.19E+02	1.02E+02

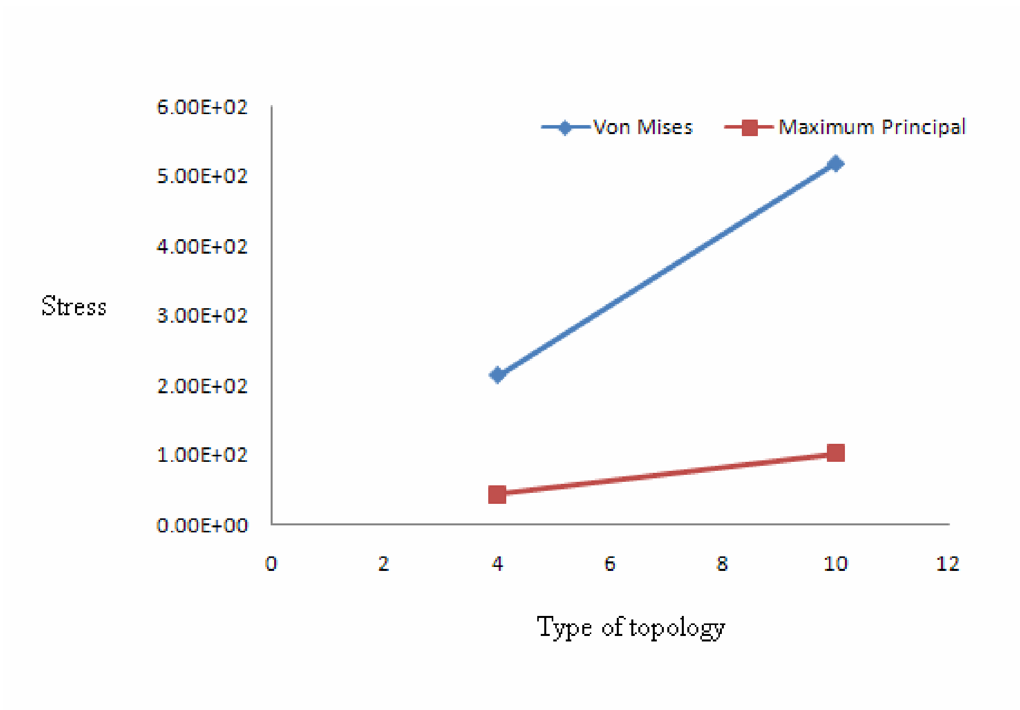


Figure 4.1: Stress versus type of topology

4.1.1 Comparison Maximum of Stress

Figure 4.2 shows the maximum stress versus global edge that is plot from information in Table 4.3. The figure shows that at 0.3 global edge length perform the highest stress either for von Mises or Max Principal. Between von Mises and

Maximum Principal, von Mises got the highest pressure about 590MPa and Maximum Principal just about 260MPa. From that, 0.3 global edges were chosen and von Mises is used to get the highest pressure.

Table 4.2: Maximum of stress between global edge lengths for TET10

Global edge length	Von mises(Mpa)	Max.principle(Mpa)
0.3	5.90E+02	2.60E+02
0.4	5.79E+02	2.46E+02
0.5	5.06E+02	2.02E+02
0.6	4.48E+02	1.60E+02
0.622272	4.06E+02	1.18E+02

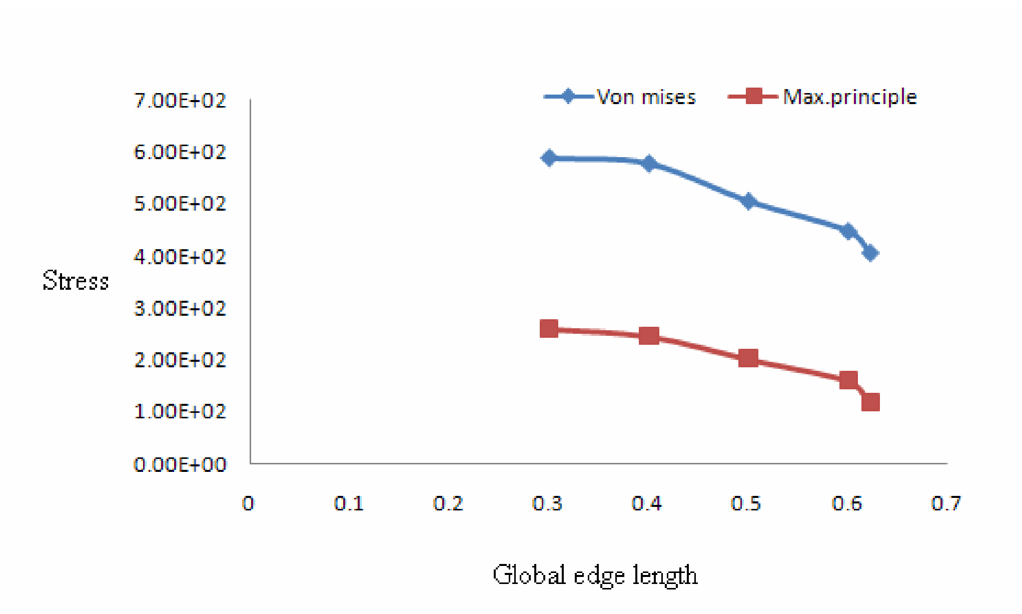
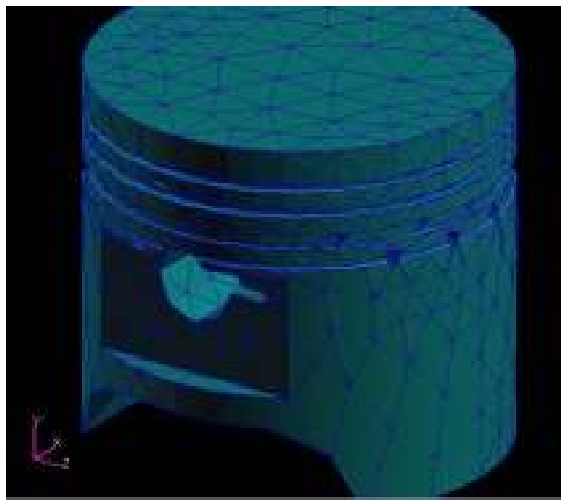


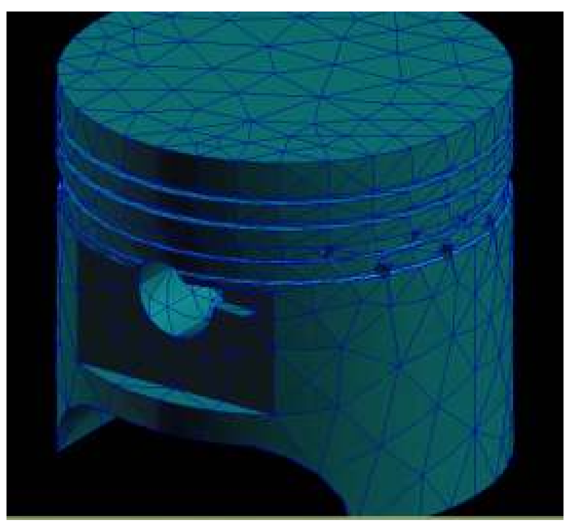
Figure 4.2: Maximum stress versus global edge length

4.1.2 Result for Finite Element Modeling and Analysis

Figure 4.3 shows finite element meshing for TET4 and TET10 were using same global mesh length. From the comparison between TET4 and TET10 shows that TET10 mesh predicted higher von Mises stresses than TET4 mesh.



(a) TET4, 5494 elements



(b) TET10, 5694 elements

Figure 4.3: Finite element meshing for (a) TET4 (b) TET10 using same global mesh length

Figure 4.4 presents the loading and boundary condition. The pressure of 7.0 MPa was applied on the surface of the piston to generate compressive load. For constraints, it was applied at the pin piston hole.

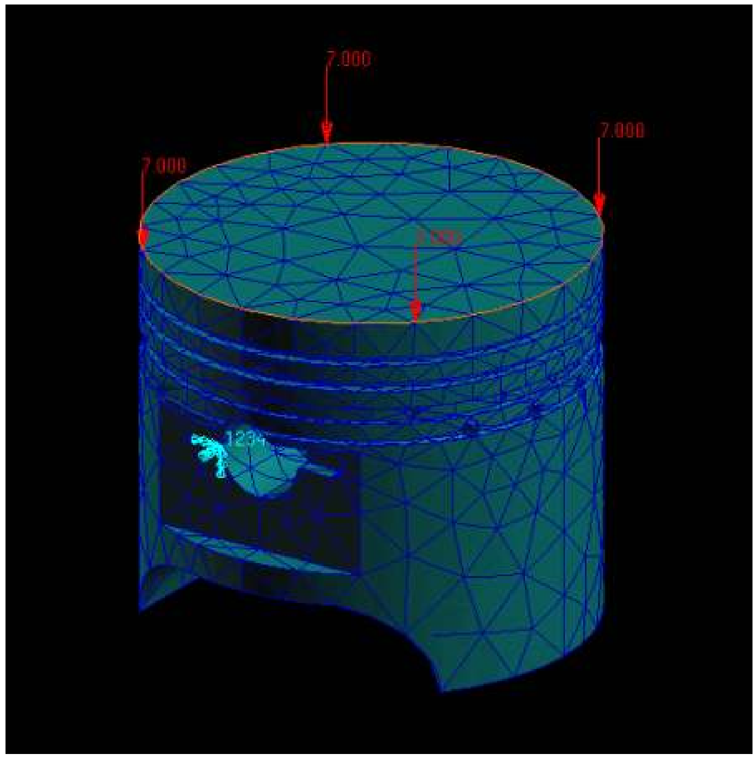
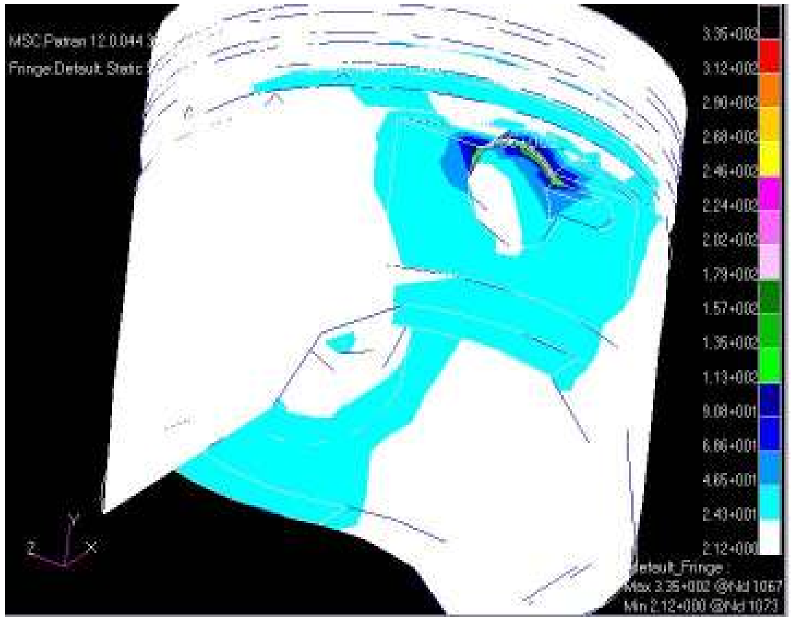
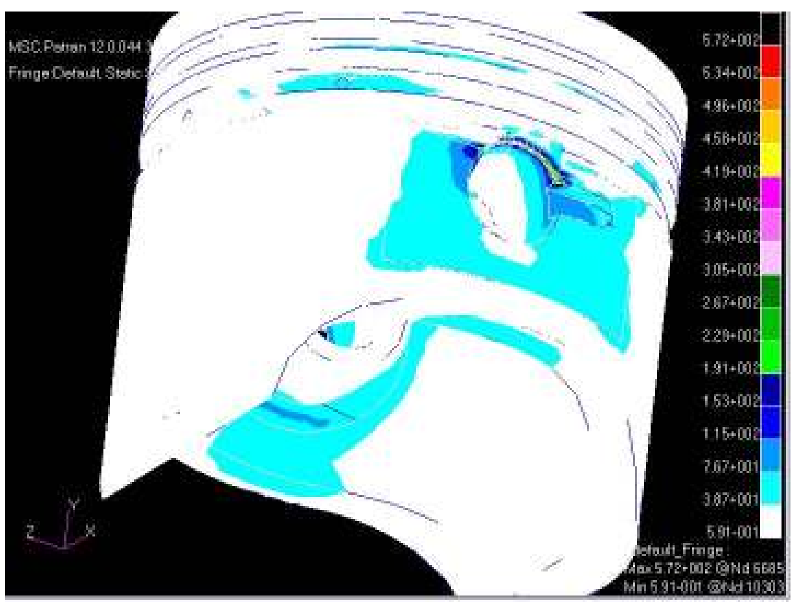


Figure 4.4: Loading and constraints

Figure 4.5 shows von Mises stresses contours (a) TET4 and (b) TET10 at high load level. From the result shows both of the TET4 and TET10 can exactly show the stress state of piston for a 7 MPa load condition. The result for using TET4 is 335 MPa while for TET10 is 572 MPa. Even though TET4 mesh is still capable of identifying critical areas, among these result obvious that TET10 mesh predict higher von Mises stresses than the TET4. So TET10 mesh was chosen for the fatigue analysis.



(a)



(b)

Figure 4.5: von Mises stresses contours (a) TET4 and (b) TET10 at high load level

4.2 Fatigue Analysis

Figure 4.6 shows the predicted fatigue contours using S-N method. It's obvious that pin piston hole found to experience the critical locations. It is observed that the minimum predicted life at the critical location is $10^{2.38e-7}$ under variable amplitude loading for *S-N* approach

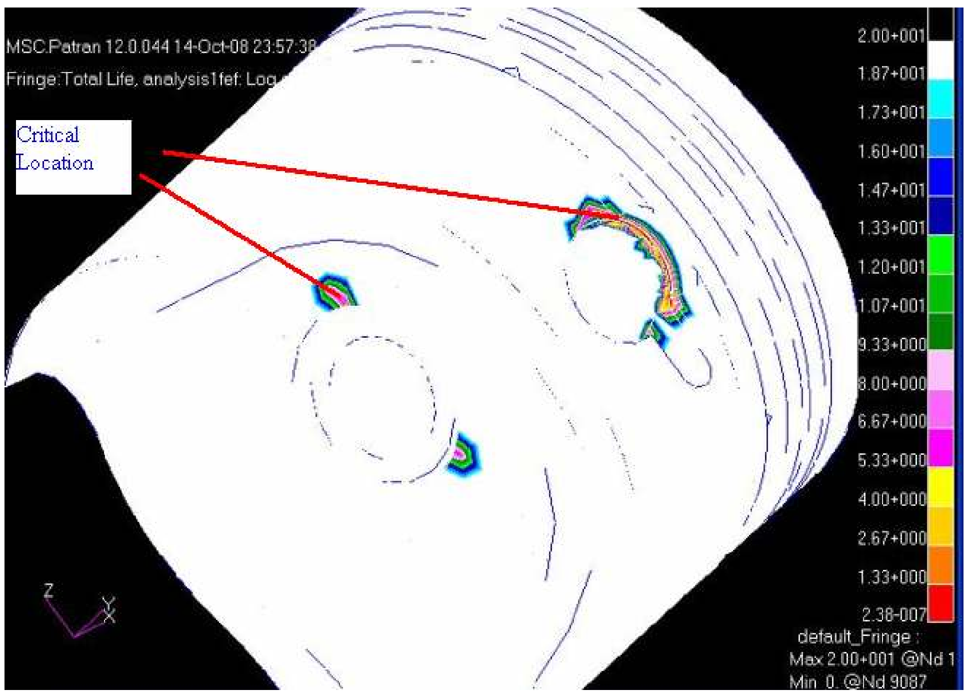


Figure 4.6: Predicted fatigue life contours plotted for TET10 using S-N method

4.2.1 Effect of Mean Stress Correction

There are two types of mean stress correction in stress life approach. There are Goodman mean stress correction and the second is Gerber mean stress correction method. Figure 4.7 shows the life (repeats) versus pressure using S-N, Gerber and Goodman method. From that graph show that Goodman method is the best among S-N and Gerber method. The graph was plot from data in table 4.4. From the table show S-N method had broken at 5.95 MPa and Gerber at 6.65 MPa. But it's different with Goodman where the piston broke at 8.4 Mpa.

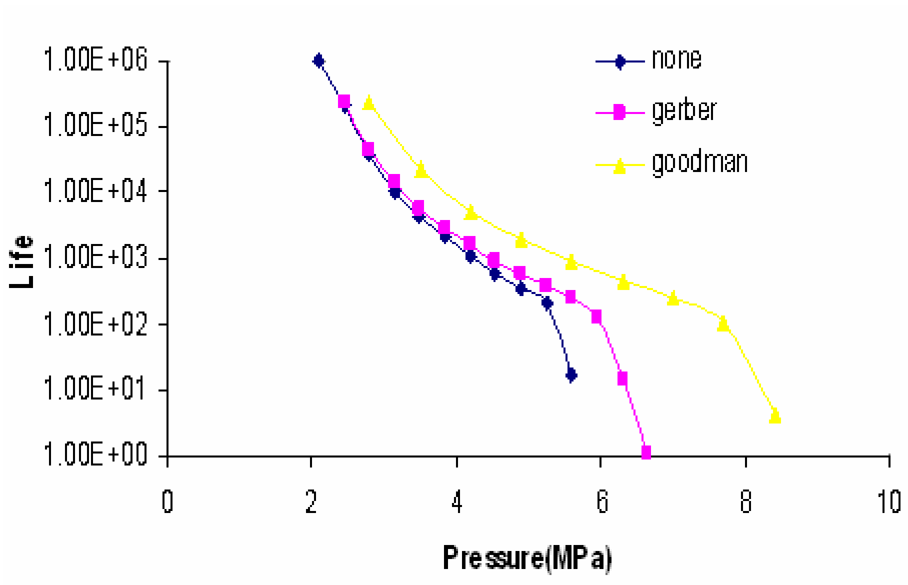


Figure 4.7: Life (repeats) versus pressure.

4.2.2 Material Comparison

This section is about to discuss about optimization with material comparison. Material comparison is consisting six materials. The materials are AISI 4042, AISI 4037, AISI 4130, AISI 4140, AISI 4150 and AISI 4340. Figure 4.8 below shows the fatigue life results for every material. Figure 4.8 shows life (repeats) versus pressure of six materials. The graph shows that AISI 4042 was the best among the six steel. It can be say that AISI 4042 suitable for produce piston to longer its fatigue life.

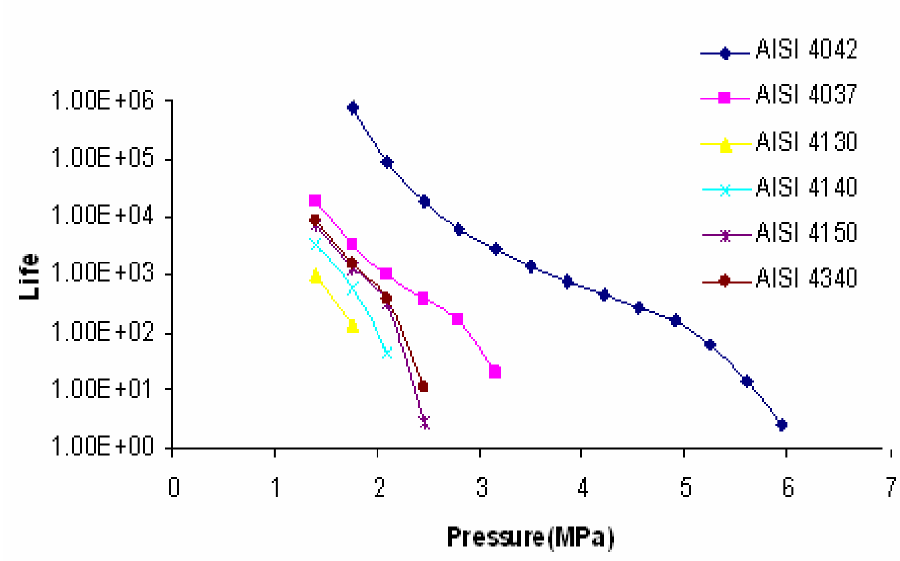


Figure 4.8: Life (repeats) versus pressure of materials

4.2.3 Surface Treatment Comparison

Figure 4.9 shows that life (repeats) versus pressure of surface treatments. From the graph, its obvious that nitriding was the best surface treatment than cold rolling and shot peening.

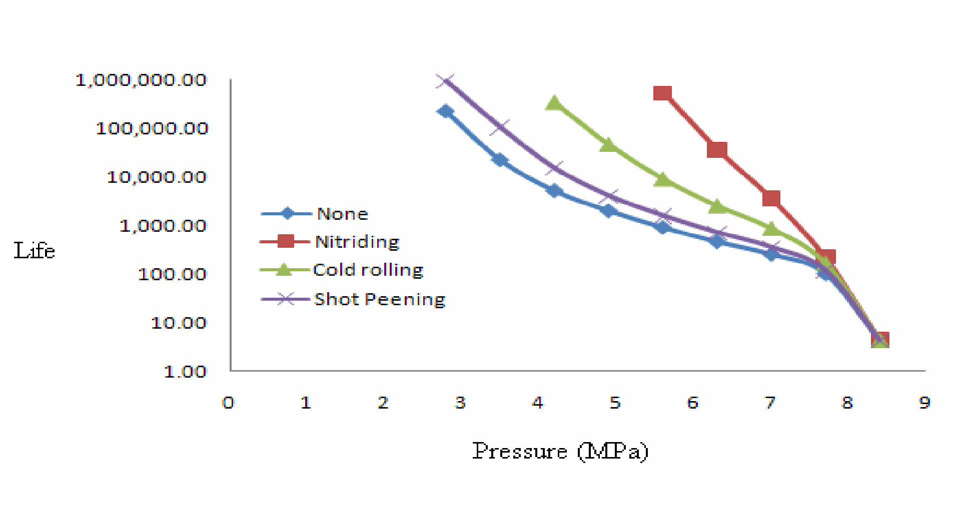


Figure 4.9: Life (repeats) versus pressure of surface treatments

CHAPTER 5

CONCLUSION

5.1 Conclusion

As a conclusion from the result, fatigue life of the piston can be predicted using the S-N method. From the result it shows that this piston will break at 5.6 MPa without using any surface treatment. While when using nitriding as a surface treatment, the the fatigue life of piston can be longer than before. When using nitriding, the piston will broken at 8.4 MPa. This also shows that nitriding treatment exponentially improves the fatigue life of the piston.

For material comparison, AISI 4042 is the best steel that can be use for improve life of piston. The material will broken at 5.95 MPa while the others material generally will broke at 3.15 MPa below.

5.2 Recommendation

i. The result value should be compared with the experimental results or the past literature for a validation.

ii. The result of the analysis need to be compared with others engineering software that relevant for a validation data.

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