Three Dimensional Analysis on Stress Concentration in a Gas Turbine Blade

by

Nur Hazwana binti Kabol

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the requirements for the
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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
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(MECHANICAL ENGINEERING)

Approved by,

____

(Dr Khairul Fuad Muhammad Rasyid)

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May 2010

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources of persons

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NUR HAZWANA BINTI KABOL

ABSTRACT

Gas turbine works under high stress and high temperature. The stress is distributed over the whole turbine blade when it is operating in the jet engine. The presence of fillets and sharp edges in the profile of gas turbine blade localizes some of the stresses which is called as stress concentration. If the material unable to withstand the stresses subjected on it which are centrifugal force of 16.58 kN and fluid pressure difference of 1596.4 kN, fracture is highly happened. This project studies about the stress being concentrated within the turbine blade. Finite element analysis will be used in this study to predict the spot that most likely has higher stress. All components acted upon the turbine blade will be investigated to produce a set of data for the simulation of the stress concentration. A three-dimensional model will be developed using CATIA before it will be analyzed using ANSYS. The results show that stress is most likely to be concentrated at the fillets at turbine blade leading and trailing edge close to the root. Providentially, the maximum stress concentrated of 377.2 Pa is too low to cause any failure to the turbine blade since its maximum allowable limit is 1036 MPa. However, under continuous cycle of stress with high temperature and pressure, this stress cannot be neglected. Thus, recommendations of the design and maintenance are proposed to prolong turbine blade life span. The outcome of this project is as expected in the objective which is to investigate its stress distribution to acknowledge parts of stress concentrated.

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CHAPTER 1
INTRODUCTION

1.1 BACKGROUND STUDY

This study is about analyzing the stress concentration of gas turbine blade. The research of gas turbine blade has been done by a lot of researchers especially in finding ways to improve its efficiency. Gas turbine blade is a very unique part of gas engine as it requires aerodynamic shape to allow air flows into the engine to produce power. Design of high-efficiency airfoils and blades are essential for optimum aerodynamic, thermoeconomic, and overall performance of turbomachinery-based powerplants [1].

Turbine blade is one of the most important components in a gas turbine. This is the component across which flow of high pressures gases takes place to produce work. It can be defined as the medium of transfer of energy from the gases to the turbine rotor [2]. Turbine blade is usually a cantilever beam or plate tapered and twisted with an airfoil cross-section [3]. The turbine blade that is going to be used in this study is a Low Pressure Turbine (LPT) taken from the second stage of CFM56-5C engine as shown in **Figure 1-1**.

Figure 1-1 CFM56-5C LPT Blade Stage 2

The CFM56-5C (see **Figure 1-2**) is the sole cost-effective propulsion system perfectly customized for the long-range Airbus A340-200 and A340-300 aircraft. It is the most powerful engine in the CFM56 family [4]. CFM56-5C offers 1.1 % reduction in fuel consumption, a 15 °C lower exhaust temperature and by this, up to 2,000 additional cycles [5]. Adaption from its previous version of CFM56 series, this engine produces the range of 139 kN to 151 kN of thrust to the aircraft.

Figure 1-2 CFM56-5C Engine [36]

Gas turbine usually operates on an open cycle which after the gas has passed through the turbine; it is not used again, but transferred to the environment for cooling. Therefore it is also particularly suitable for airplanes, which don't have water for cooling in contrast to ships [6]. Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. The high-pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure.

The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power. The exhaust gases leaving the turbine are thrown out (not re-circulated), causing the cycle to be classified as an open cycle [7]. However, by utilizing the air-standard assumptions, the closed cycle can be modeled for the gas turbine cycle. The combustion process is replaced by a constant-pressure heat-addition process from an external source and the exhaust process is replaced by a constant-pressure heat rejection process to the ambient air.

For this case, the cycle is thermodynamically assumed as close loop for the ease for the analysis. The cycle that the engine undergoes is the Brayton cycle as shown in **Figure 1-3**.

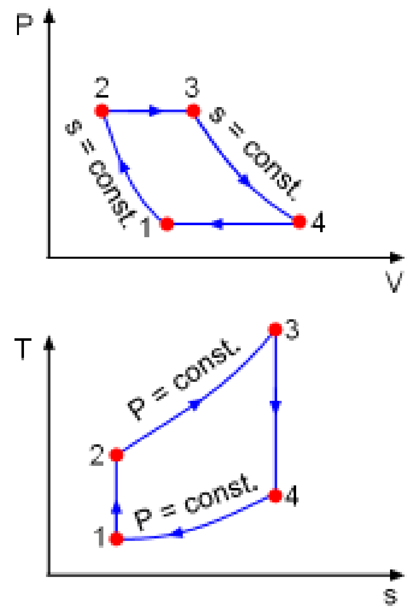

Figure 1-3 P-v and T-s Diagrams of Brayton Cycle [38]

1 – 2: Isentropic compression (in a compressor)

2 – 3: Constant-pressure heat addition

3 – 4: Isentropic expansion (in a turbine)

4 – 1: Constant-pressure heat rejection

Gas turbine operates under high temperature and high pressure. Due to this extreme condition, turbine blades are subjected with high stress. Enormous load from compressed air from compressor enters turbine at a very high temperature and pressure. To withstand the exerted stress and load, turbine blades need to be designed with certain characteristics such as advanced material used with high strength properties. While stress is being distributed within the turbine blade, some parts of the turbine blade experienced concentrated stress due to their profiles.

Stress concentration can be defined as the modification of simple stress distribution due to presence of shoulders, grooves, holes, keyways, threads, and so on, which results in localized high stresses [8]. It is measured by the stress concentration factor

K_t which is defined as the ratio of the maximum stress, σ_{max} to the nominal stress, σ_{nom} [9]. That is,

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}} \quad \text{for normal stress (tension or bending)} \qquad \text{(E 1.1)}$$

$$K_t = \frac{\tau_{max}}{\tau_{nom}} \quad \text{for shear stress (torsion)} \qquad \text{(E 1.2)}$$

where σ_{max}, τ_{max} represent the maximum stresses to be expected in the member under the actual loads and the nominal stresses σ_{nom}, τ_{nom} are reference normal and shear stresses. The subscription t indicates that the stress concentration factor is a theoretical factor. That means, the peak stress in the body is based on the theory of elasticity, or it is derived from a laboratory stress analysis experiment [8].

1.2 PROBLEM STATEMENT

Airplane has becomes one of the most important and popular transportation nowadays. This has increase the usage of the airplane service tremendously. As the result, the engine also being used rapidly and this will cause the turbine blade being subjected to more stresses than usual. Turbine blades need to endure the stresses and for that, the turbine blades are made from advanced material and coatings. In this case, the material used for the CFM56-5C LPT Stage 2 is nickel-based alloy (Inconel 718).

The design of the turbine blade itself is important. Operated under extreme conditions of high temperature and high pressure, stress that being distributed within turbine blades might concentrates at some parts due to the profile of the turbine blade. The presence of sharp edges and fillets cause the stress to localize at these parts. If the stress being concentrated on these parts exceeds the allowable limit, it can cause breakage and failure to the turbine blade. This chain of failure needs to be prevented to avoid any loss especially lives of the people on board.

Therefore this study is important as to put in the picture on how stress concentration affects the turbine blade life. With this study, manufactures can take it as additional information for the future design of their turbine blades. Assistance from this study also can improve the current design of turbine blades especially for CFM56-5C LPT Stage 2.

1.3 OBJECTIVES

The objective of this study is to investigate the stress concentration of the gas turbine blade. For that, the load applied and stress distribution within the gas turbine blade are to be analyzed.

1.4 SCOPE OF STUDY

This study will only concern the CFM56-5C LPT Stage 2 as we can assume that the stresses distributed on all the turbine blades in the low pressure turbine are equal. This is because the dimensions for the turbine blades that come from the same stage are the same with each other. The dimensions of the chosen blade (CFM56-5C LPT Stage 2) are taken using vernier calliper for future use in modelling the turbine blade using CATIA and ANSYS software.

The theory of gas turbine cycle (Brayton cycle) is studied to get familiar with the operation (see **Figure 1-3**). With knowing the basic of the cycle and dimensions of its turbine blade, some assumptions such as its nozzle angle can be made on the process to gain required data such the forces acted upon the turbine. The data is then used to analyze the stress distribution within the turbine blade. Thus, using Finite Element Analysis (FEA) by ANSYS, parts that being concentrated by stress can be known.

CHAPTER 2
THEORY/LITERATURE REVIEW

2.1 THEORY

2.1.1 CFM56-5C Engine

The specifications of CFM56-5C engine (see **Table 2-1**) has been studied as it is the type of engine which use the turbine blade of this study. It has lowest specific fuel consumption (SFC) of the CFM56 family and the quietest engine in its thrust class. The excellence takeoff performance for high-altitude and hot airfields are because of its high thrust-to-weight ratio. This second-generation of Full Authority Digital Engine/Electronics Control (FADEC) produces 36,000 pounds of thrust demonstrated during ground testing and its long-duct, mixed-flow nacelle developed by CFM to provide significant noise attenuation, reduced fuel burn, and increased climb thrust [24].

Table 2-1 Specifications of CFM56-5C Engine [24]

Take Off Conditions	Value
Thrust	139 – 151 kN
Bypass Ratio (BPR)	6.6
Total Pressure Ratio	31.5
Flow Rate of Air	466 kg/s
Maximum Flight Mach Number	0.80
SFC	17.08 g/kN.s

2.1.2 Gas Turbine

Gas turbine can be illustrated as **Figure 2-1** which reflects the Brayton cycle. Based on the figure, it explains the process flow of the cycle. Starting with air being drawn to the compressor and compressed to the desired temperature and pressure before injected with fuel and burned in the combustion until it expands in the turbine and produced thrust for the engine.

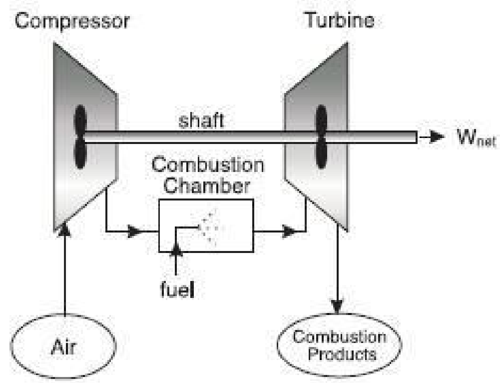

Figure 2-1 Open Cycle Gas Turbine [37]

The focus of the study is about turbine blade, which means only the parts in the turbine are the focal point. Turbine is the device that drives the electric generator. As the gas passes through the turbine, work is done against the blades, which are attached to the shaft. As the result, the shaft rotates, and the turbine produces work [7].

2.1.3 Airfoil Theory

Airfoil is defined as an object with a special shape that is designed to produce lift efficiently when the object is moved through the air. It is a streamlined body bounded principally by two flattened curves and whose length and width are very large in comparison with thickness [2]. It is designed to produce good lift force and low drag force.

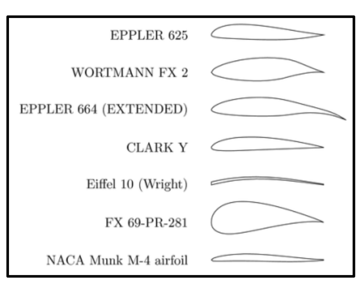

Figure 2-2 Examples of Airfoil Profiles [39]

2.1.4 Blade Stresses

One of the critical factors in designing turbine blade is the stresses acted upon it. Bending stresses are enforced by centrifugal force, fluid pressure differences and vibration [38]. But in this case study, vibration is neglected for the ease of analysis. Among these three, centrifugal force plays the main role as the turbine blade is rotated at high velocity.

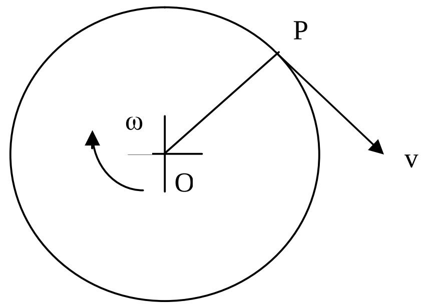

Figure 2-3 Centrifugal Force on Rotating Bodies [25]

2.1.4.1 Centrifugal Force

Centrifugal force occurs at all rotating bodies. Consider point P is rotating about at centre O with constant angular velocity ω (see **Figure 2-3**). Centrifugal stress is a function of the mass of material in the blade (inconel), blade length and its speed. The component of centrifugal force that acting radially outward causes tensile stress at the root. For the blade to be able to withstand this stress, sufficient cross-sectional area at the blade root and suitable material should be provided.

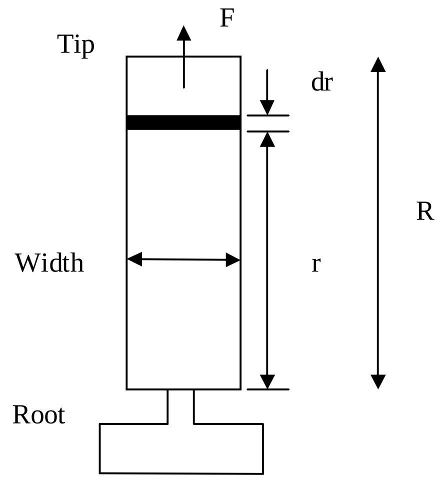

Figure 2-4 Centrifugal Force on Turbine Blade [38]

Since the blade (see **Figure 2-4**) is attached to a 0.5 m radius of rim, the total radius from the rim centre to the tip is including the blade length. Consider a small length dr at radius r and the mass of this element is

$$dm = \rho A dr \tag{E 2.1}$$

The centrifugal force produced by this mass is dF.

$$dF = dm\omega^2 r$$

$$= \rho A dr \omega^2 r$$

In this limit as dr → dr this becomes dF = $\rho A dr \omega^2 r$

The centrifugal force acting on this section due to the mass is then found by integration

$$F = \rho A \omega^2 \int_r^R r dr$$

$$= \rho A \omega^2 \frac{[R^2 - r^2]}{2} \tag{E 2.2}$$

where A is the area that attach the root with turbine blade.

Since $\sigma = \frac{F}{A}$,

$$\sigma = \rho \omega^2 \frac{[R^2 - r^2]}{2} \tag{E 2.3}$$

2.1.4.2 Fluid Pressure Difference

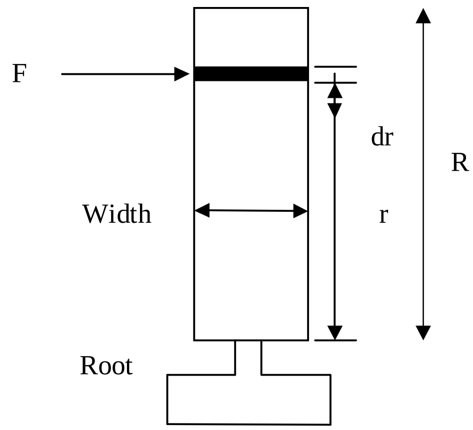

Figure 2-5 Force on Turbine Blade

Consider there is a force exerted on the element dr and a relative velocity V_r acting on the blade based in its rotational velocity V_b and fluid velocity V.

$$\begin{aligned} \Delta F &= \dot{m}V_r \\ &= \rho V_b a V_r \end{aligned} \tag{E 2.4}$$

2.2 LITERATURE REVIEW

2.2.1 Stress Concentration

Stress is at maximum at the crack tip and decreased to the nominal applied stress with increasing distance away from the crack. The stress that is concentrated around the crack tip or flaw develops the concept of stress concentration. The flaw amplifies the stress surrounding it; however, stress amplification not only occurs on a microscopic level (e.g. small flaws or cracks) but can also occur on the microscopic level in the case of sharp corners, holes, fillets and notches [10]. Gas turbine blade has a lot of sharp corners and fillets to achieve certain characteristics for the purpose of air flowing through the turbine itself and expands it. The presence of these shapes results in modifications of the simple stress distribution of the part, thus localized high stresses occur at that particular shape and measured by the stress concentration factor K [8].

E.A. de Carvalho (2005) evaluated the stress concentration factors for an internally pressurized cylinder containing a radial U-notch along its length [11]. Finite element analysis [12] is used to predict the location of hot-spot stresses. The stress distribution analysis in the vicinity of the tubular T-joints exhibiting such defects proves very important in locating hot-spot-stresses [12]. A. The study's outcome showed that distribution of calculated stresses in the notch bottom indicates that stress concentration factors rise as width of the notch decreases (N'Diaye, S. Hariri, G. Pluvinage & Z. Azari, 2009). In 1988 H. C. Hsiung, A. J. Dunn, D. R. Woodling and D. L. Loh studied the stress analysis and design procedures to achieve a longer life turbine blades [28].

J. Li and X.B. Zhang (2006) studied on the important of effect on the stress concentration. This study was done by simply experimenting plates with a central hole under uniaxial tensile loading. The outcome of it was that most of these criteria, including criteria with a single material parameter and those with two material parameters, are not suitable for fracture prediction of materials under non-singular stress concentrations [30]. Geometrical discontinuities also affects the stress distribution within a structural member thus influences the stress concentration. For that, a study was done on how axial struts with geometrical discontinuities subjected to dynamic loading conditions [31]. Experiment on butt weld of pipelines also being studied as stress concentrations are presented in the design. An analytical expression for the bending stress in the pipe wall due to this out-of-roundness is the outcome. The derived stress concentration factors can be used together with a hot spot stress S–N curve for calculation of fatigue damage [32].

A round bar with a circular-arc or V-shaped notch were investigated with its stress concentration factors under torsion, tension and bending. After thorough studies, a set of convenient formulas useful for any shape of notch in a round test specimen is proposed as the result. It is concluded that the stress concentration of a circumferential groove in a round bar is important for test specimens used to investigate fatigue strength of engineering materials under torsion, tension and bending [33]. Zheng Yang, Chang-Boo Kim, Chongdu Cho and Hyeon Gyu Beom (2008) studied the elastic stress and strain fields of finite thickness large plate containing a hole. In this analysis, the values decrease rapidly and have a tendency to each constant related to Poisson’s ratio with plate thickness increasing. The difference is larger when the plate is thicker or the Poisson’s ration is larger [34].

The values of existing theoretical stress concentration factors for rectangular uniform thickness plates, with opposite U-shaped notches, subjected to in-plane bending do not include the effect of length as a significant parameter, based on N. Troyani, S.I. Herna´ndez, G. Villarroel, Y. Pollonais and C. Gomes study (2004). Thus, they demonstrated that below a threshold value, defined as transition length, these stress concentration factors cease to be valid and, particularly, also demonstrated that below this threshold the magnitude of the stated factors may be significantly larger than existing values; a fact that may have important consequences for the accurate estimates of fatigue life. The result of this study is that existing values of the theoretical stress concentration factors for the geometry and loading treated here are applicable in the long length regime only. The outcome showed that the theoretical stress concentration factors only valid for long members, and are independent of the way the bending moment is applied [35].

2.2.2 Finite Element Analysis by Ansys

Surface damage produces blades dimensional changes, which result in operational stress increase and turbine efficiency deterioration. An analytical calculation parallel to the finite element method was utilized to determine the static stresses due to huge centrifugal force. The dynamic characteristics of the turbine blade were evaluated by the finite element modal and harmonic analyses [13]. According to Maoqiu Wang, Eiji Akiyama, & Kaneaki Tsuzaki (2005), the stress and strain distributions in the notched specimens were calculated by means of FEA (Finite Element Analysis) using the commercial finite element modeling software ANSYS. The result was the local fracture stress decreases with increasing local hydrogen concentration as the diffusible hydrogen content or stress concentration factor increases, thus resulting in the decrease in the notch tensile strength [14]. Avinash V. Sarlashkar, Girish A. Modgil, & Mark L. Redding showed examples of how the program makes effective use of the ANSYS preprocessor to mesh complex turbine blade geometries and apply boundary conditions are presented using specific examples [15]. It is concluded that the presence of the stress concentrations at the notch affects, as expected, the stress distributions not only locally but all over the section [11].

2.2.3 Gas Turbine Design

The aerodynamic design of a gas turbine stage comprises two major concepts. The first one is it is necessary to select the vector diagram which will offer the optimum efficiency features of such constraints like blade speed, blade temperature, turbine size and exhaust swirl as may be relevant [16]. The shape of the airfoil is important to determine the efficiency of the turbine. The used turbine blade is needed to be reshape to its original profile to gain back the perfect airflow and aerodynamic. A research is developed to re-contour the airfoil profile automatically based on the algorithm which adopted the neutral line concept [17] and the interpretation vector method by finding the relationship between the original profile of the unused blade and the used blade under each section layers in two-dimensional plan (W. J. Lin, B. T. J. Ng, X. Q. Chen and Z. M. Gong). Pezhman Akbari and Norbert Müller (2003) said that wave-rotor compression ratio is one of the most important topping parameters in gas turbine design.

Without a doubt, the biggest challenge in turbine design is associated with the temperature environment especially at the inlet. As much as 30% of the core flow is used for turbine cooling to ensure durability [18]. Film cooling has been incorporated into blade designs. In film cooling, cool air is bled from the compressor stage, ducted to the internal chambers of the turbine blades, and discharged through small holes in the blade walls. This air provides a thin, cool, insulating blanket along the external surface of the turbine blade [19]. The material used in the turbine blade is known as Inconel 718 Alloy [20] which is a high-strength, corrosion-resistant nickel chromium material used at -423° to 1300°F. It can be fabricated, combined with good tensile, fatigue, creep, and rupture strength, have resulted in its use in a wide range of applications [21]. Lisa O’Donoghue said, to prevent from melting due to excessive high temperature during operation, turbine blade is covered with coating that can put off melting problem and also preventing the turbine blade from corrosion at high temperature. Thus, the overall design of the turbine blade is important including the parts that being concentrated with stress.

2.2.4 Centrifugal Force

When an object moves in circle, it will behave as if it is experiencing an outward force and this force is called as centrifugal force. This force depends on the mass of the object, the speed of rotation and the distance from the centre [23]. N.N. Alder, W.T. Pockman, J.S. Sperry and S. Nuismer in 1997 used the centrifugal force application to measure the occurrence of cavitations as a function of negative pressures in xylem. This centrifugal force allows the results to be more precise as any desired pressure can be imposed on this study [26]. According to M. A. J. Bossak and O. C. ZienkieWicz (1973) in their study, the stress especially caused by centrifugal force can affect the natural frequencies of rotating machineries. The study shows the analysis based on the solid three-dimensional elements as the blades of turbines which often have the shapes that cannot be approximated by only plate assumptions [36].

CHAPTER 3
METHODOLOGY

3.1 RESEARCH

Study on gas turbine blade has a very wide area. The investigations done by researchers help a lot in finding the data needed in this study. The research done is mainly focusing on gas turbine blade operation and stress concentration. Knowing the basic principles of gas turbine cycle is important as this study is fundamentally about gas turbine blade. Meanwhile for stress concentration is because it is going to be analyzed on the gas turbine blade.

The research being done is based on journals, books and reliable websites. For journal, there is no analysis that is exactly like this study. However, some of the information from the journals are related to the purpose of this study. Most of the journals can be found by online from reliable sources. These journals are uploaded by the schools, universities and also international specialization organizations such as *International Journal of Fatigue 29* [22].

Books that are used as reference are taken based on the relevancy of the books with this study. For example *Peterson's Stress Concentration Factor* [8] explained a lot about the stress concentration within many types of structure such as bolt head, T-joint and shoulder fillets. Data gained from the websites are only taken if the sources are reliable. The manufacturer's website [4] is one of the trusted sources that is used for the research of this study.

3.2 DATA INTERPRETATION

With above methods of research, data can be gained for the purpose of this study. Some assumptions are made due to the confidentiality of the engine such as the flow speed and turbine blade specifications. These assumptions are made based on the available information that can be found and must be reasonable. With data gathered by the research done and rational assumptions, the result can be computed using mathematical analysis such as centrifugal force and bending stress acted on the turbine blade.

3.3 MODELING

As this study requires a three-dimensional (3D) analysis, a 3D model of the turbine blade is to be constructed using the design collaborative software, CATIA [23]. The model will be based on the dimensions and specifications of the chosen turbine blade which is CFM56-5C LPT Stage 2 (see **Figure 1-1**). Vernier caliper is to be used to measure the dimensions of the turbine blade. The expected 3D model is almost similar like as shown in **Figure 3-1**.

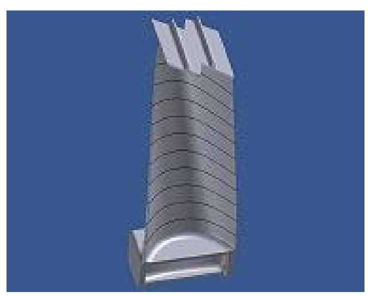

Figure 3-1 3D Model of a Turbine Blade [43]

3.4 SIMULATION

Once the 3D model is ready, it will be converted into ANSYS, the engineering simulation software. The data input from the analysis of stress distribution within the turbine blade and load subjected on the turbine blade will produce a simulation of which part of the turbine blade being concentrated with stress. This will be the expected outcome of this study, a 3D analysis on stress concentration in a gas turbine blade.

ANSYS is a general-purpose finite element modeling package for numerically solving a wide variety of mechanical problems including this case study [41]. In general, the process of simulating can be broken down into following three stages:

i. *Preprocessing: defining the problem*

   The turbine blade lines, areas, volumes, type and material properties (Elastic Modulus of 2.11 GPa and poisson ratio of 0.275) are defined.

ii. *Solution: assigning loads, constraints and solving*

   The loads which are the forces calculated are applied on the turbine blade after setting its constraint which is its root.

iii. *Postprocessing: further processing and viewing the results*

   The stress contour diagrams are produced and viewed for further analysis.

3.5 PROJECT ACTIVITIES

The execution of this study can be referred in Gantt Charts (see Appendix A and B)

3.6 TOOLS REQUIRED

As there will be no experiment being done in this study due to constraints of cost and budget, only software such as CATIA and ANSYS are needed to do the analysis and vernier caliper to measure the dimensions of turbine blade.

CHAPTER 4
RESULTS AND DISCUSSIONS

4.1 GAS TURBINE

To be able to calculate the work done and forces acting upon the turbine blade, the specifications on the turbine need to be known. Due to unavailability on the accurate data by the manufacturer because of the private and confidential reason, the raw data were taken based on the studies done by researchers. However, the data cannot be simply taken from anywhere as each turbine has different specifications. Thus, the data were taken from the most comparable case with this study.

Table 4-1 Data of Low Pressure Turbine [27, 29]

Elements	Value
Rotation Direction	Counter clockwise
Rotational Speed (ω)	4983 rpm
Mass Flowrate ($\dot{m}$)	91.91575 kg/s
Volume Flowrate at Outlet ($\dot{V}$)	129.57 m³/s
Total Pressure Ratio	3.015
Total Inlet Pressure (P_3)	533.3 kPa
Total Inlet Enthalpy (h_3)	1203.44 kJ/kg
Total Inlet Temperature (T_3)	1103 K

4.2 STRESSES

The stresses that are going to be considered in this case study are centrifugal stress and the blade fluid pressure difference. These stresses need to be considered to ensure that the blade can withstand them to prevent any failure in its design. **Table 4-2** shows the properties of the material of turbine blade which is Inconel 718 at room temperature.

Table 4-2 Properties of Inconel 718 [40]

Properties	Values
Ultimate Tensile Strength	1240 MPa
Yield Strength (0.2 % offset)	1036 MPa
Elongation in 50mm	12 %
Elastic Modulus (Tension)	211 GPa

To be able to calculate the stresses, dimensions of the turbine blade are measured and simplified as in **Figure 4-1** and summarized in **Table 4-2**.

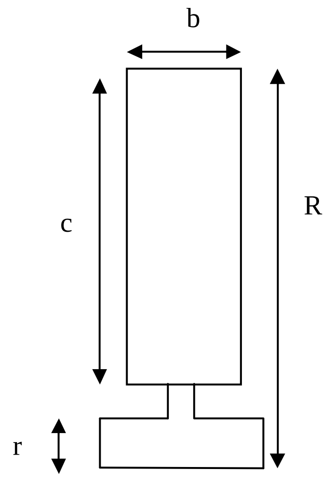

Figure 4-1 Turbine Blade

Table 4-3 Dimensions of turbine blade

Parts	Dimensions
Rim radius	0.5 m
Tip radius (R) + Rim radius	0.137 m + 0.5 m = 0.637 m
Root radius (r) + Rim radius	0.007 m + 0.5 m = 0.507 m
Cross-section area a = (b x c)	0.022 m x 0.105 m = 2.31 x 10^{-3} m^2
Root cross-section area A	0.02 m x 0.005 m = 1 x 10^{-4} m^2

With E 2.1, the centrifugal force acting on the turbine blade is

$$F_c = \rho A \omega^2 \frac{[R^2 - r^2]}{2}$$

$$F_c = [(8190\, kgm^{-3})(1 \text{ x } 10^{-4}\text{m}^2)\left(4983 \text{ rpm } \times \frac{2\pi}{60\text{s}}\right)^2] \div 2 \times (0.637^2 - 0.507^2)$$

$$F_c = 16\,582.97\ N$$

$$\boldsymbol{F_c = 16.58\ kN}$$

Thus,

$$\sigma_c = \rho\omega^2 \frac{[R^2 - r^2]}{2}$$

$$\boldsymbol{\sigma_c = 165.83\ MPa}$$

Each turbine blade experienced 16.58 kN of centrifugal force while being rotated at the speed of 4983 revolutions per minutes with 0.5 m radius of the rim. Even though it is a big number for a single turbine blade to handle, the ability of the design and material used in the turbine blade make it possible to withstand the force.

For fluid pressure difference part, velocity diagram is used to determine its relative velocity V_r by using its rotational velocity of the blade V_b and flow velocity V. It is known that

$$V_b = \omega R = \left(4983\, rpm \times \frac{2\pi}{60s}\right)(0.637m) = 332.4\, ms^{-1}$$

and by assuming that $V = 500\, ms^{-1}$, nozzle angle α as 10° and by simplifying the inlet blade angle β as 20°,

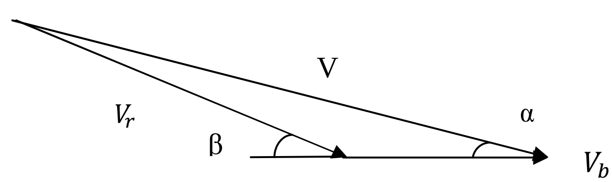

Figure 4-2 Velocity Diagram

$$\frac{V_r}{sin\ \alpha} = \frac{V}{sin\ 180 - \beta}$$

$$V_r = 253.86\, ms^{-1}$$

Thus,

$$\Delta F = \dot{m}V_r$$

$$\Delta F = \rho V_b a V_r$$

$\Delta F = (8190\ kgm^{-3})(332.4\ ms^{-1})(2.31 \times 10^{-3} m^2)(253.86\ ms^{-1})$

$\Delta F = 1\ 596\ 434.8\ N$

$\boldsymbol{\Delta F = 1\ 596.4\ kN}$

and

$\boldsymbol{\Delta \sigma = 691.1}$ **MPa**

1596.4 kN of force is exerted on the turbine blade planform area. This force is bigger than centrifugal force due to bigger load from fluid velocity impacting the planform area compared to velocity of the blade. Both stresses are proven to be not exceeding its tensile stress. This shows that the turbine blade can withstand the stresses acted upon it and being operated under high stress, pressure and temperature.

4.3 Three-Dimensional Model of Turbine Blade

Using CATIA software, a 3D model of CFM56-5C low pressure turbine of stage 2 (see **Figure 4-3**) is produced. Difficulties have been faced throughout the process of designing the blade such as its airfoil shape and twisting the blade (see **Figure 4-4**).

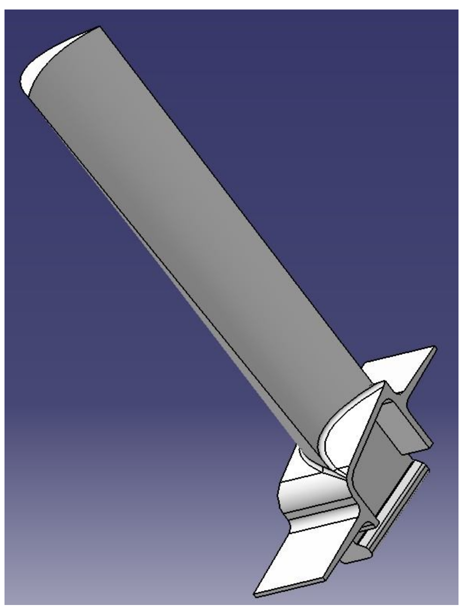

Figure 4-3 3D Model of Turbine Blade

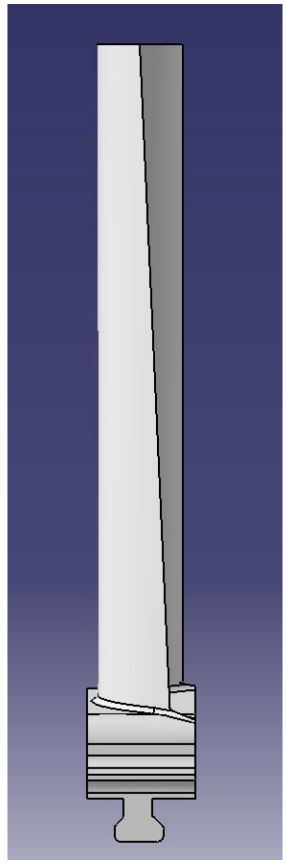

Figure 4-4 Twisted Turbine Blade from Side View

This 3D model is then exported into ANSYS software for its analysis. Based on the data (see **Table 4-4**) and using automesh, the meshing of the turbine blade is produced as shown in **Figure 4-5.**

Table 4-4 Data for ANSYS

Properties	Value
Elastic Modulus	211 GPa
Yield Strength	1036 MPa
Ultimate Tensile Strength	1240 MPa
Poisson Ratio	0.275
Element size	5 mm
Type of Analysis	Linear Static
Number of Elements	1509
Number of Nodes	590
Element Type	First Order Tetra

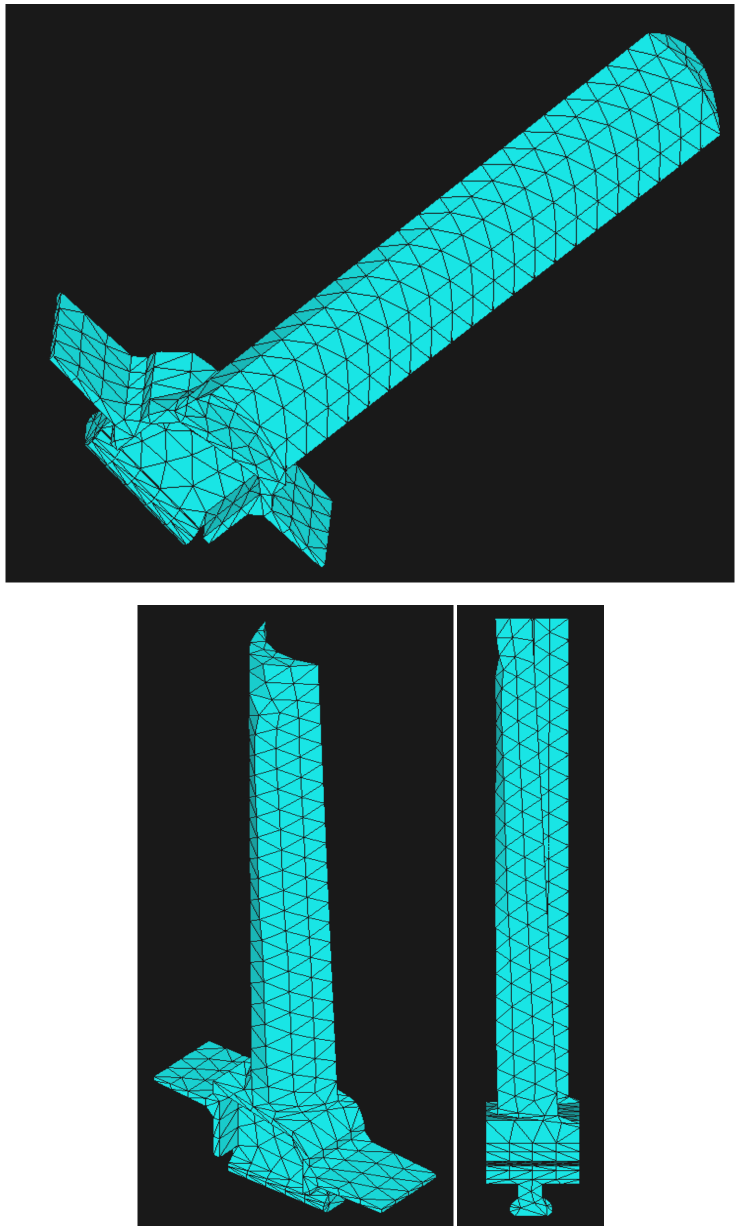

Figure 4-5 Meshing of the Turbine Blade

As it can be seen in the **Figure 4-5**, the mesh is done based on first order tetra. The reason for using the first order instead of higher order such as second order is to speed up the calculation since higher order will consist of more nodes compared to the first order.

Next step is to set the turbine blade constraint which is at its root as shown in **Figure 4-6** below. The blue force is denoted as F_c and red force as ΔF. The load of 16.58 kN (blue arrow) is applied at the tip of the turbine blade representing centrifugal force acting on that part. Meanwhile, 1596.4 kN (red arrow) of load is applied at the middle of the turbine blade for simplification. Both coordinates are chosen to represent the mean or average spots that will be impacted by the forces. The red triangles are the constraint for the turbine blade.

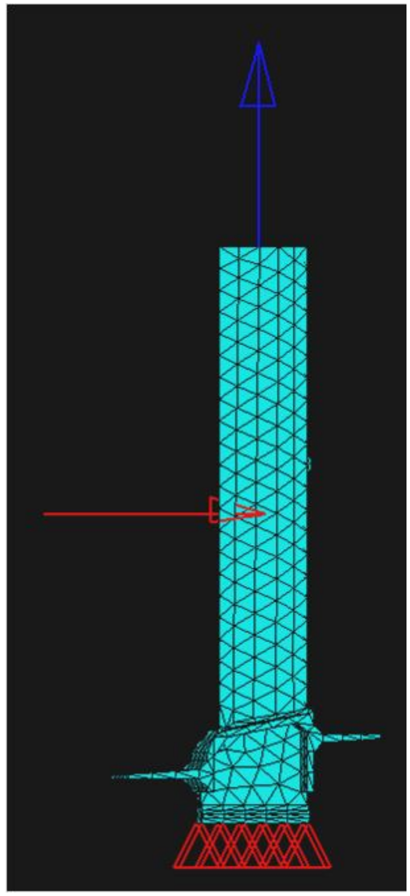

Figure 4-6 Constraint and Loads

The analysis is done with three cases so that the analysis covers the entire required field in determining which part of the turbine blade is mostly concentrated with stress. The case studies are:

i. Centrifugal force only F_c

   The stress is thought to be more on the root as centrifugal force is acting radially outward and causes the root to experience more stress. However, after the analysis is done, the stress is being concentrated at the trailing edge of the turbine blade (see **Figure 4-7**). The most concentrated part is at the trailing edge close to its root which somehow proving that the root experiences most of the stress. Trailing edge has the smallest thickness part of the turbine blade and this might cause the stress to localize. It shows that the maximum stress on that part is 54.43 Pa which is way below the maximum allowable limit 1036 MPa (yield strength).

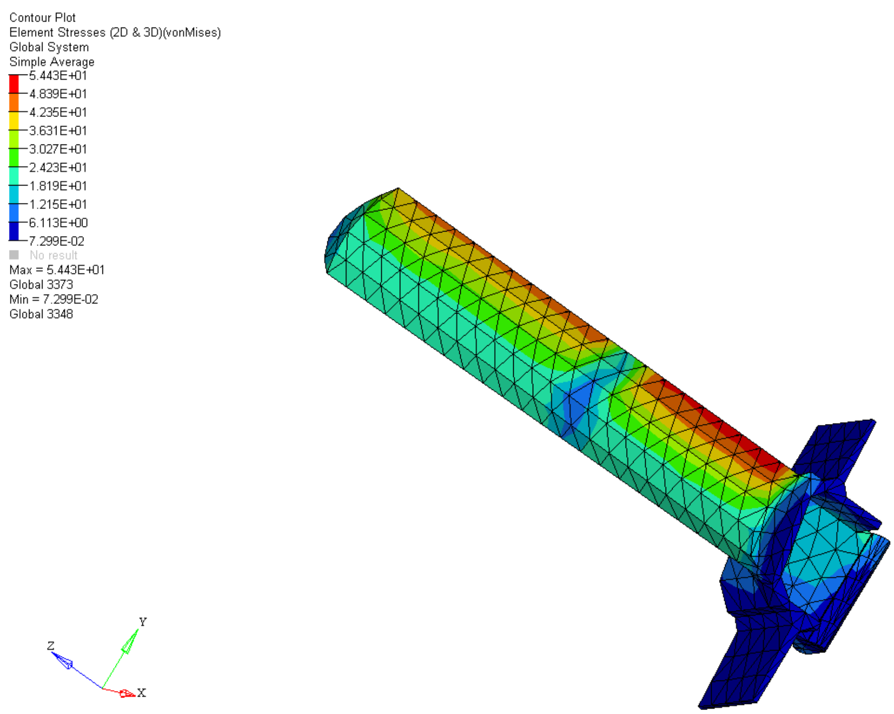

Figure 4-7 Analysis on Centrifugal Force

ii. Fluid pressure difference only ΔF

The load of 1596.4 kN is applied at the middle of the turbine blade as shown in **Figure 4-6** in blue arrow. The result is as expected where the stress is concentrated at the leading edge near its root. During the contact of load on the turbine blade, the leading edge is impacted with high temperature air with the velocity of 4983 rpm. Since there is a presence of fillets at the root, it is likely to be localized with stress as shown in **Figure 4-8** below.

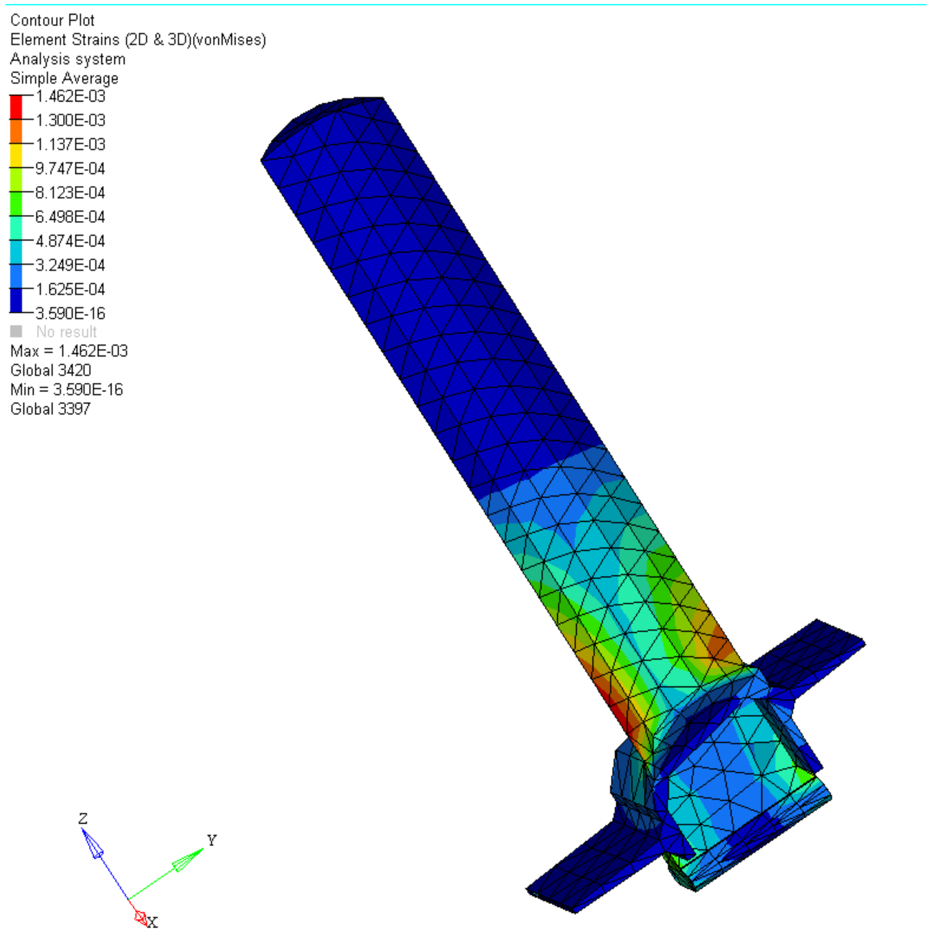

Figure 4-8 Analysis on Fluid Pressure Difference Force

iii. Both forces

Lastly, both forces are combined and applied on the turbine blade to see its whole stress distribution. The result shown in **Figure 4-9** indicates that most stress is concentrated at leading edge near its root. It is obvious that the centrifugal does not contribute that much on the stress acted upon the turbine blade compared to the fluid pressure difference. The maximum stress on the turbine blade is 377.2 Pa, which is also below its allowable limit. The reason for location of stress at the leading edge near root is due to presence of fillet at that part. Since leading edge is impacted by the force more than the trailing edge, the stress is most likely to localize at that part more.

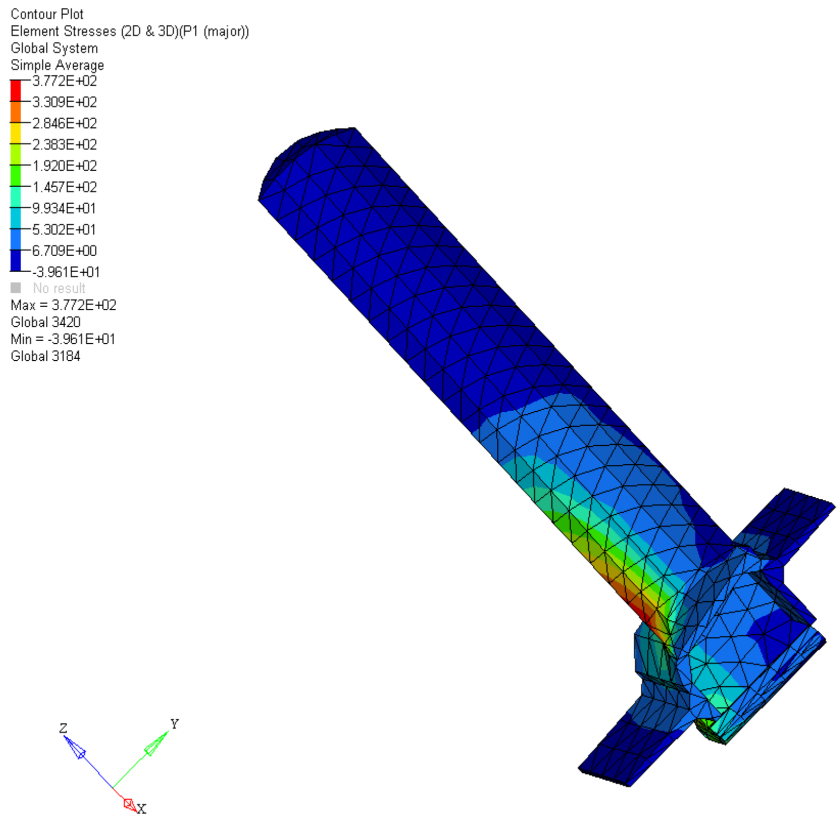

Figure 4-9 Stress Distribution

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

It is almost impossible to design without having some parts being concentrated with stress. Stress concentration has been one of the important parts in designing engineering structure. Knowing how much stress concentration will affect the structure will help in a better design and prevents unwanted accidents due to design failure. The analysis of stress concentration of gas turbine blade is crucial as it is being used intensively under high temperature and improper investigation and inspection can lead to unwanted incidents.

Gas turbine is being used in high cost applications such as in jet engines and plants and involves many lives. Thus if there is a possibility that the design of the blade can caused uncalled failure, then necessary action shall be done to prevent it. The analysis can be done using finite element analysis (ANSYS) instead of experimenting it with high cost. By doing this analysis, it will give awareness to people about stress concentration factors in designing and with the assistance of this study, a better turbine blade can be designed in the future.

The objective of this project is to analyze the stress concentration of the gas turbine blade by studying the stress distribution within the turbine blade. The basic theory and operation of the turbine is studied so that the data collected can be used in the calculation of forces acted upon the turbine blade. In the simulation of ANSYS, the stress is assumed to be distributed evenly with applied load at the tip and the middle of turbine blade.

It is known the Inconel 718 is the material for the turbine blade and has yield strength of 1036 MPa and this is the maximum allowable limit for any stress being imposed upon the turbine blade. The result of maximum stress of 377.2 Pa acted on the turbine blade shows that the material is strong enough to withstand the forces impacted on it. It is proven too that the stress will localize at the fillets of the turbine blade at its root. In conclusion, the analysis of the turbine blade on its stress distribution leads to the knowing of the parts being concentrated with stress.

Even though it is proven that the strength of Inconel 718 is sufficient enough to endure the stress, gas turbine is operated under high pressure and temperature and thus will undergo high stress cycle. On design side, the radius of fillet can be increase to reduce the amount of stress concentrated on it. The turbine blade is a twisted type which has the purpose to make its velocity diagram uniform from root to tip. Its airfoil shape also can be improved for a better lift-drag effect so that the air can flows through the turbine blade without harming it.

A long lifespan of turbine blade does not only rely on the design and material but also how it undergoes the repair and maintenance. A good maintenance process can ensure a longer life expectancy of the turbine blade such as surface enhancement. Surface enhancement is introduction of a surface layer of compressive residual stress to minimize sensitivity to fatigue or stress corrosion failure mechanisms to improve the performance and life of turbine blade. Example of surface enhancement is shot peening.

This project has achieved its objective. However, it can be further investigated for more details and accurate results.

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APPENDICES

Appendix A

Gantt Chart for Final Year Project (FYP) I

No	Project Flow	Week 1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Final Yeat Project 1 Briefing														
2	Selection of Project Topic (Submission of Form 01 and Form 02)														
3	**Preliminary Research Work**														
	Research on books and journals related to the project topic														
4	Submission of Preliminary Report					●									
5	**Project Work**														
	Familiarization with gas turbine blade (dimensions and specifications)														
	Analyzing the components acted upon gas turbine blade														
	Compiling the data gathered														
6	Submission of Progress Report								●						
7	Seminar (compulsory)								●						
8	**Project Work (continues)**														
	Analyzing the data														
	Computing the data														
9	Submission of Interim Report Final Draft														●
10	Oral Presentation										During Study Week				

Appendix B

Gantt Chart for Final Year Project (FYP) II

No	Project Flow	Week													
		1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	**Project Work (continues)**														
	3-D model of the gas turbine blade (CATIA)														
2	Submission of Progress Report 1				●										
3	**Project Work (continues)**														
	Meshing on the 3D Model														
4	Submission of Progress Report 2								●						
5	Seminar (compulsory)								●						
6	**Project Work (continues)**														
	Stress Analysis														
	Data interpretation														
7	Poster Exhibition										●				
8	Submission of Dissertation Final Draft														●
9	Oral Presentation										During study week				
10	Submission of Dissertation (hard bound)										7 days after oral presentation				