INVESTIGATE BLADE FAILURE
IN AXIAL COMPRESSOR

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DECLARATION

I hereby declare that this project report entitled
INVESTIGATE BLADE FAILURE IN AXIAL COMPRESSOR
is written by me and is my own effort except the ideas and summaries which I have clarified their sources.

Signature	:…………………………………
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VERIFICATION

"I hereby declare that I have read this thesis and in my opinion this thesis is sufficient in terms of scope and quality for the award of degree of Bachelor Mechanical Engineering (Structure & Material)"

Signature	:…………………………………
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ABSTRACT

This project presents the analysis blade of the axial compressor by using the Finite Element Method (FEM). The axial compressor is one of the important devices that always use in industry. The maintenance for the compressor and its parts is necessary in order to ensure that the compressor perform well. This need to be done for certain period that had been set. Sometimes, the compressor will have a problem or fail to perform well. There are many factors that cause the failure of this compressor. The one of the factor that may cause the failure is the damage of the component of the compressor. The problem for compressor to perform well will disturb the flow of the production in industry, thus will make the cost increase. To avoid this situation, the compressor and its components need to be service for every period that has been set. Since blade is the one of the main parts to generate the compressor this project will deal with analysis for the blade of axial compressor. Hope that this study will help to reduce failure of the blade which is one of the components of the axial compressor. The blade will be designed base on the actual dimension by using the Solidwork software, and then it will be exported in the MSC Nastran Patran software. The blade had been simulated in different of mesh or global length. The best result which has the lowest percentages of error was selected to undergo for further analysis. After undergo the simulation analysis with MSC Nastran Patran, the maximum stress was found to be 66MPa while the maximum displacement is $2.91\times10^{-4}m$. The approximation theoretical value for maximum stress which is 78.66MPa shows the 16% of percentage of error for the simulation with the Nastran Patran software. Lastly the factor of the safety for the blade was calculated and the value is 3.65.

ABSTRAK

Projek ini mempersembahkan analisis bilah pemampat paksi dengan menggunakan Kaedah Unsur Terhingga . Pemampat paksi adalah salah satu peranti penting yang selalu digunakan dalam industri. Penyelenggaraan untuk pemampat paksi dan komponen-komponenya adalah perlu untuk memastikan ianya sentiasa berfungsi dengan baik. Ini hendaklah dilakukan untuk jangka masa tertentu yang telah ditetapkan. Kadang kala, pemampat akan menghadapi masalah atau gagal berfungsi dengan baik. Terdapat banyak faktor yang menyebabkan kegagalan pemampat ini. Salah satu faktor yang menyebabkan kegagalan ini adalah kerosakan pada komponen pemampat. Masalah kepada pemampat untuk berfungsi dengan baik akan mengganggu aliran pengeluaran dalam industri, seterusnya akan membuat kos meningkat. Untuk mengelakkan situasi ini, pemampat dan komponennya perlu diservis untuk setiap jangka masa yang telah ditetapkan. Memandangkan bilah adalah salah satu bahagian utama untuk menggerakkan pemampat, projek ini akan berurusan dengan analisis untuk bilah pemampat paksi. Diharap agar kajian ini akan membantu untuk mengurangkan kegagalan pada bilah dimana ia adalah salah satu komponen pemampat paksi. Bilah ini telah disimulasikan dalam berbeza sirat atau panjang global. Keputusan terbaik yang telah mendapat peratusan terendah ralat telah dipilih untuk menjalani analisis seterusnya. Selepas menjalani analisis simulasi dengan MSC Nastran Patran, tekanan maksimum yang diperolehi adalah 66MPa manakala sesaran maksimum adalah $2.91 \times 10^{-4} m$. Penghampiran nilai teori untuk tekanan maksimum yang mana adalah 78.66 MPa menunjukkan 16% peratusan ralat untuk simulasi dengan menggunakan perisian Nastran Patran. Akhir sekali, faktor keselamatan untuk bilah telah dikira dan nilainya adalah 3.65.

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M	-	mass
b	-	width
k	-	stiffness
d	-	distance
F	-	force
Y	-	distance from the neutral axis
I	-	moment inersia
σ_m	-	maximum stress
$\{u\}e$	-	the displacement vector
$\{f\}e$	-	forces acting on the element.
$[k]$	-	stiffness matrix
q_{tot}	-	theoretical total head
q_{df}	-	loss from disc friction
q_{th}	-	head available
ηimp	-	efficiency of impeller
$\eta stage$	-	stage efficiency
q_{osf}	-	frictional forces encountered in the stator
q_{oa}	-	the overall actual adiabatic head

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CHAPTER 1

INTRODUCTION

1.1 Background

The axial-flow compressor often use in turbine gas especially over 5MW units. An axial-flow compressor is one in which the flow enters the compressor in an axial direction (parallel with the axis of rotation), and exits from the gas turbine also in an axial direction.The axial-flow compressor compresses its working fluid by first accelerating the fluid and then diffusing it to obtain a pressure increase. The fluid is accelerated by a row of rotating airfoils (blades) called the rotor, and then diffused in a row of stationary blades (the stator). All angles that describe the blade and its orientation are measured with respect to the shaft (z axis) of the compressor.

Blades in axial compressor were designed by arranging thickness distribution around a camber line. Recent research in the axial flow compressor blade, based on understanding the flow behaviors in the blades, where stacking lines of the airfoils are curvilinear, has generated favorable results in terms of loss reduction and thereby efficiency. The blade also can have some defect or fail to operate successful after a few year operations. The damage could happen due to many factors. High temperature and vibration may cause the power loss and damage to the blade in the axial compressor. This project will investigate and analysis the failure using the Nasran Patran software. The model of the blade will be designed according to the real dimension using the Solidwork. The complete model from Solidwork will be eksported to Nasran Patran software to analysis and calculate the stress on the blade.

1.2 Objectives

This project carried out the study and analyze problem those related to the failure of blade in axial compressor. The other objectives that contain in this project are:

1. To model the blade of the axial compressor based on the actual dimensions by using Solidwork software.
2. To simulate the blade and investigate the failure of axial compressor blade using MSC Nasran/Patran software.

1.3 Scopes

The scopes of this study are:

1. Modeling the blade in axial compressor based on the actual dimension with Solidwork software.
2. Simulate the blade with different mesh using Nasran Patran and comparison with theoretical value need to be done.

1.4 Problem Statement

After a long time usage of a blade in axial compressor, it may have some defect or failed to give a better performance. If this things occur in industry, it will disturb the flow of production thus will make the cost increase. This thing should not be happen and must be avoided to ensure that the low cost of the production in order to make a profit for an industry. So, this study will deal with an analysis using the MSC Nastran Patran software to simulate the blade.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

The literature review consists of various aspects regarding the investigate blade failure, performance loses in axial compressor, and analysis using the finite element method.

2.2 Investigate blade failure

Xiaolei Xu and Zhiwei Yu (2006) have conducted a failure investigation on the turbine blades used in a locomotive turbochanger, which are made from K418 Ni-base superalloy. Fractography investigation on the troubled blade indicates that cracks initiated from the surface of the concave side close to the trailing edge and propagated towards to the leading edge. The multi-origin fatigue fracture is the dominant failure mechanism of the blade. Metallographic morphology typical of over-heat damage features, such as re-dissolution of the eutectic $\gamma + \gamma'$ melting of the local region of the grain boundary appears in the microstructure of the airfoil part of the failed blades. Appearance of over-heat damage structure in the serviced blades makes the strength of the blade material decrease intensely to initiate fatigue cracks and make one of the blades fracture first. Fragments from the blade fractured first would crash the other blades to make the blades break or bending deformation.

From the investigations that have been done, the blades used in a locomotive turbocharger were damaged to different extent in servicing. The accumulated service time of the blades is about 6 years. It was reported that a mayor repair had been conducted on the turbine once. The blade material is specified as K418 superalloy (C: 0.08–0.16; Al: 5.50–6.40; Si 6 0.50; Nb: 1.80–2.50; Mo: 3.80–4.80; Ti: 0.50–1.00; Cr: 11.50–13.50; balance: Ni). The observation and analysis of failure of a few representative failed blades to assess the possible failure reasons.

The method of investigation that uses to determine the chemical composition of the blade material is spectroscopy chemical analysis method. The micro-composition in various zones of the fracture surfaces was determined by energy dispersive X-ray spectrometer (EDX). The microstructure of the sectional specimens near the fracture was observed by scanning electron microscopy (SEM) on a Philips XL-30 scanning electron microscope. The fracture surfaces were analyzed by visual and SEM observation to study the failure mechanism.

N. Vardar and A. Ekerim (2006) had done a case study of failure analysis of a 40 MW gas turbine blade made of Udimet 500. The cause of failure is found to be intergranular cracks which started during exposure to high temperature. The cracks initiated from the grain boundaries and propagated to the critical length to result in catastrophic fracture. In many locations M_6C type secondary carbides were found agglomerated on grain boundaries. Also micro-cavities were found on fracture surfaces which served as an origin of creeping failure mechanism. The investigation of the failure was carried out by using several experimental tests, including optical microscopy, scanning electron microscopy (SEM), energy dispersive spectroscopy (EDS), X-Ray diffraction (XRD) and X-Ray fluorescence (XRF). The microstructural investigation of the blade airfoil revealed the presence of continuous film of carbides in grain boundaries of the base material as a result of transformation of carbides of MC type to carbides of M_6C type due to high temperature operation of the blades.

2.3 Performance losses in axial compressor

In axial-flow compressor journal written by Meherwan P. Boyce (2006) stated that the calculation of the performance of an axial-flow compressor at both design and off-design conditions requires the knowledge of the various types of losses encountered in an axial-flow compressor. The accurate calculation and proper evaluation of the losses within the axial-flow compressor are as important as the calculation of the blade-loading parameter, since unless the proper parameters are controlled, the efficiency drops. The evaluation of the various losses is a combination of experimental results and theory. The losses are divided into two groups: (1) losses encountered in the rotor, and (2) losses encountered in the stator. The losses usually are expressed as a loss of heat and enthalpy. A convenient way to express the losses is in a no dimensional manner with reference to the blade speed. The theoretical total head available (q_{tot}) is equal to the head available from the energy equation ($q_{th} - q_{tot}$) plus the head, which is loss from disc friction.

$$q_{tot} = q_{tot} + q_{df} \tag{1}$$

The adiabatic head that is actually available at the rotor discharge is equal to the theoretical head minus the heat losses from the shock in the rotor, the incidence loss, the blade loadings and profile losses, the clearance between the rotor and the shroud, and the secondary losses encountered in the flow passage

$$q_{ia} = q_{th} - q_{in} - q_{sh} - q_{bl} - q_{c} - q_{sf} \tag{2}$$

Therefore, the adiabatic efficiency in the impeller is

$$\eta imp = \frac{q_{ia}}{q_{tot}} \tag{3}$$

The calculation of the overall stage efficiency must also include the losses encountered in the stator. Thus, the overall actual adiabatic head attained would be the actual adiabatic head of the impeller minus the head losses encountered in the stator from wake caused by the impeller blade, the loss of part of the kinetic head at the exit of the stator, and the loss of head from the frictional forces encountered in the stator

$$q_{oa} = q_{ia} - q_w - q_{ex} - q_{osf} \tag{4}$$

Therefore, the adiabatic efficiency in the stage

$$\eta stage = \frac{q_{oa}}{q_{tot}} \tag{5}$$

There are many losses that may occur in the axial compressor system. For the example is disc friction loss. This loss is from skin friction on the discs that house the blades of the compressors. This loss varies with different types of discs. The other loss is incidence loss. This loss is caused by the angle of the air and the blade angle not being coincident. The loss is at a minimum to about an angle of $\pm 4^\circ$, after which the loss increases rapidly. Blade loading and profile loss. This loss is due to the negative velocity gradients in the boundary layer, which gives rise to flow separation. The loss is from skin friction on the blade surfaces and on the annular walls and this is called a skin friction loss. Clearance loss. This loss is due to the clearance between the blade tips and the casing. Wake loss. This loss is from the wake produced at the exit of the rotary. Stator profile and skin friction loss. This loss is from skin friction and the attack angle of the flow entering the stator. For the exit loss the loss is due to the kinetic energy head leaving the stator.

2.4 Analysis using Finite Element Method (FEM)

The finite element method (FEM) is a numerical technique to find approximate solutions of partial differential equations (PDE) as well as of integral equations. This method also called finite element analysis. The solution approach is based either on eliminating the differential equation completely (steady state problems), or rendering the PDE into an approximating system of ordinary differential equations, which are then solved using standard techniques such as Euler's method, Runge-Kutta, etc.

There are many study cases and analysis had been carried out by using this method. For the example, Wang Wei, Li Junfeng and Wang Tianshu (2008) had done a modal analysis of liquid sloshing with different contact line boundary conditions using finite element method. A finite element method (FEM) for liquid sloshing modal analysis is established. Surface tension and three kinds of contact line boundary conditions, namely, free-end, pin-end and wetting boundary conditions, are taken into account.

Sloshing damping caused by energy dissipation at the wall, in the interior fluid and at the contact line is calculated. Numerical results are compared with the analytical values and measurements. For the pin-end and free-end boundary conditions the differences between numerical value and analytical value are small, and for the wetting boundary condition,because approximation is used, the differences are more significant.

This finite element also has software to carry out the certain analysis. For the example is MSC Nastran Patran. M. E. Stavroulaki, G. E. Stavroulakis and B. Leftheris (1995) had conducted a structural analysis and optimization techniques within the MSC/NASTRAN software for the analysis of optimal prestress restoration of buildings. This analysis is done to alleviate admissible stress violations a minimum prestress reinforcement is sought along with the position of the prestressed elements. Prestressing is modelled by fictitious thermal loading on the linear (rod) elements which model the prestressing cables (tendons). A quadratic cost function for the prestressing cost with a penalty term that counts for stress violations is assumed for the optimal prestressing problem. Certain aspects of the computer implementation, including the use of mathematical programming and structural optimization tools for large-scale structures, are included. The theory is illustrated by numerical examples concerning the prestress restoration of a masonry wall subjected to static loading.