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Prediction of fatigue life of mistuned steam turbine blades subjected to variations in blade geometry

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***Abstract*— There is a large number of power stations suffering from fatigue failures of the steam turbine blades. The steam turbine blades are also subjected to steam flow bending, centrifugal loading, vibration response, and structural mistuning. These mentioned factors significantly contribute to the fatigue failure of the steam turbine blades. Low-Pressure (LP) steam turbines experience premature blade and disk failures due to the stress concentrations at the blade root area of its bladed disk. Driven by the problems encountered by the steam power plant electricity generating utilities with regards to steam turbine blades fatigue failure, this study of the mistuned steam turbine blades subjected to variation in blade geometry will be of great significance to the electricity generation industry. A simplified, scaled-down mistuned steam turbine bladed disk model was developed using ABAQUS finite element analysis (FEA) software. Acquisition of the vibration characteristics and steady-state stress response of the disk models was performed through FEA. Thereafter, numerical stress distributions were acquired, and the model was subsequently exported to Fe-Safe software for fatigue life calculations based on centrifugal and harmonic sinusoidal pressure loading. The vibration characteristics and the response of the variation steam turbine geometric blade was conducted. The FEA natural frequencies compared well with published literature of the real steam turbines indicating reliability of the developed FEA model. The study found that the fatigue life is most sensitive to changes in blade length, followed by the width, and then the thickness, in this order. The analytical life cycles and Fe-Safe software shows the percentage difference of less than 4.86%. This concludes that the developed numerical methodology can be used for real-life mistuned steam turbine blades subjected to variations in blade geometry.**

***Keywords*— Turbine Blades; Blade Geometry; Low-Pressure; Finite Element Analysis; Fe-Safe; Fatigue Life**

I. Introduction

Fatigue is a wide and vital domain in mechanical engineering, particularly in turbine components. The leading cause of the unavailability of large fossil fuel turbines worldwide has been recognized as the result of turbine blade failures and blade damage [1]-[2]. A single-blade failure of a steam turbine can lead to significant component damage and financial losses [3]. Fatigue damage in the engineering industry has cost several percentages in gross domestic product. Life prediction of steam turbine blades subjected to varying blade geometries attached to one common disk is a great benefit in fossil fuel power plants [2]. During turbine operation, the blades are subjected to high-pressure flow bending, centrifugal stress, and high vibration response induced by alternating stresses.

Uncertainties of steam turbine blade geometric deviations affect aerodynamic performance. Therefore, it affects the working conditions of the entire steam turbine component. The design and optimization methods of steam turbine blades are deterministic, therefore the approached method for life prediction is both probabilistic methodology development and deterministic.

A. Research Problem

Fatigue failures of turbomachines mostly occur on the mistuned blades because of the cyclic stress and mean stress, vibration resonance, and the harsh working environment to which they are exposed [4]. Commonly, fatigue cracks initiate at the root of the concave blade side and propagate later on [1]. Mistuning is any irregularity in the bladed disk system that makes the response of the system asymmetric. These irregularities can be initiated by structural and aerodynamic means. Structural irregularities arise as a result of the blade-to-blade geometric variations [4].

Variations in the boundary conditions and the fixity of the blades to the disk or shrouding might also complicate the variation in blade geometric dimensions [6]. Turbomachinery is generally designed to attain the perfect cyclic symmetry conditions, random deviations among the subdivisions caused by manufacturing tolerances, material defectiveness, non-uniform assembly and wear characterize the real blades [7].

Traditional research on the HCF of steam turbine blades is mostly based on the S-N curves and nominal stress approaches. Several methods for fatigue life prediction of steam turbine blades have been established along with comparative studies of the various methods. Blade life estimation has traditionally been performed through deterministic models, which often necessitate excessively conservative assumptions [5]. Incorporate probabilistic modelling, which could eliminate conservative assumptions and allow for uncertainty and variability in key variables to be accounted for [6].

B. *Important of the Study*

There is a large number of power stations worldwide suffering from fatigue failures of the steam turbine blades. Steam turbine blades operate at a high temperature, in a moist and corrosive environment. The steam turbine blades are also subjected to steam flow bending, centrifugal loading, vibration response, and structural mistuning. Steam turbine disk becomes asymmetric due to in-service wear caused by the deviation in blade geometry. These mentioned factors significantly contribute to the fatigue failure of the steam turbine blades. Driven by the problems encountered by the steam power plant electricity generating utilities with regards to steam turbine blades fatigue failure, it is hypothesised that the study of the mistuned steam turbine blades subjected to variation in blade geometry will be of great significance to the electricity generation industry.

Steam turbine-bladed disks are originally designed to be cyclically symmetric, and inherent factors such as manufacturing tolerances, material inhomogeneity, and in-service wear cause small dissimilarities to appear between each of their elementary sectors [8]. The phenomenon is known as steam turbine blade mistuning and is a significant concern in the steam electricity generation industry. Mistuning greatly affects the performance and life of steam turbine blades, therefore it needs to be dealt with well in advance. Mistuned blades subjected to variation in blade geometry factors indicate that HCF in steam turbine blades is a problem that needs to be further researched and developing solutions.

Consistency inspections and rapid replacement of damaged blades are required to manage the lifespan and maintenance of steam turbine blades. Replacing blades too early or too late is insufficient [9]. It is unknown when the fatigue will happen, which lead to a great loss to the power generation industry. The following reasons support the mentioned blade replacement factors.

i. Late replacement of damaged blades ultimately leads to the shutting down of the entire steam turbine with time and cost implications.
ii. If the damaged blade is not replaced immediately, it leads to more blade failures and greater losses.
iii. During the steam turbine outage and troubleshooting and inspection, as well as the replacement process and commissioning. Time and money are wasted without any electricity production.
iv. Significant time is also wasted while ordering the replacement blades from the Original Equipment Manufacturer.

Very little research has been reported on mistuned steam turbine blades resulting from variations in blade geometry. This study uniquely contributes to the concern of the prediction methodology of the HCF life of mistuned steam turbine blades including the effects of blade geometric variation.

The probabilistic technique approach to the prediction of the fatigue life of simplified, real values mistuned blade model has not been adequately explored. It is envisaged that a reliable methodology can be realistically developed upon integration of the above hypotheses. The approach needs to be undertaken with high accuracy since it has not been sufficiently investigated. Brown Miller algorithm in Fe-Safe software is then employed. The results findings will positively contribute to the power plant's safety, stability, efficiency, availability, and financial gain.

II. MATHEMATICAL MODELLING

The relevant mathematical theories used for HCF life calculation of the mistuned steam turbine blades are shown below. The correct formulation of the mathematical models which sufficiently represents both the physics and mistuned blade behaviour are imperatively applied.

A. *Vibration Equations for a Disk*

During energy production, steam turbine blades are in dynamic motion. Application of equation of motion relate are useful as they relate to the excitation forces acting on the entire mistuned-bladed steam turbine system. [10] studied the vibration dynamics of mistuned bladed disk. Dynamics are influenced by the excitation for the solution of the direct dynamic [10]. Therefore, the equation is most useful in the time function [11].

A typical structural model of the bladed disk of an aero-engine compressor is generalized in Fig. 1.

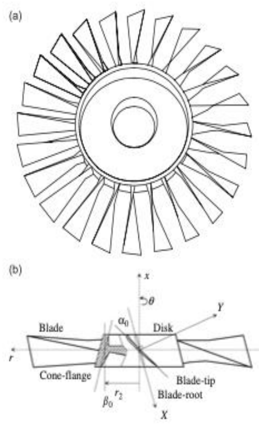

Fig. 1 Mechanical model of a typical bladed-disk [10]

The disk is simplified and divided into three parts as sub-structures: a variable-thickness disk with a fixed centre, a cone flange connected to the disk rim with thickness as a variable, and multiple blades twisted along the centred blade disk.

It is assumed that all the blades are fixed on the cone flange to the external rim of the disk. In this study, only the blade calculations are considered. Hypothetically, the longitudinal flexing of the cone flange is ignored due to the reinforced effect of multiple blade roots. The experimental mode analysis and mode correction have been carried out. With multiple vibration modes of the blade and the disk, the mode synthesis technique is applied. [10] finally provided the forced response vibration equation of motion for the mistuned bladed disk which is presented in Eq. 1.

$$[M][\ddot{q}] + [C][\dot{q}] + [K][q] = [F(t)] \quad 1$$

B. *Stress Due to Centrifugal Load*

Mistuned steam turbine blades centrifugal forces are used to determine if the yielding will occur or not.

Fig. 2 illustrates the centrifugal force action on the steam turbine blade.

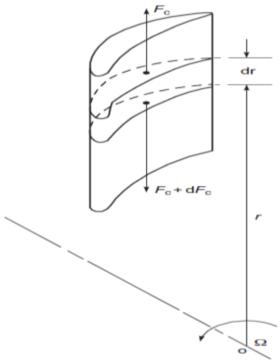

Fig. 2: Centrifugal forces acting on a steam turbine blade [12]

The centrifugal force and normal stress are estimated with an applied operational speed of 3000 *r/min* with a set of Eq. 2 [5]-[13],

$$F = m\omega^2 r \quad 2$$

where m is the mass of the bladed disk, ω is the rotational speed and r is the distance to the center of gravity of the blades.

$$F_C = (\frac{W}{g})(\frac{2\pi N}{60})^2(r) \quad 3$$

where W is the weight of the blade, g is the gravity constant, N is the rotational speed of the turbomachine and r is the radius of the blade to the stress concentration point.

The centrifugal stress at the blade root with Table I variables is calculated using Eq. 4.

$$\sigma_{CF} = \frac{F}{A} \quad 4$$

TABLE I
STEAM TURBINE BLADE VARIABLES [4]

Area (A)	$1.7 \times 10^{-4}\ m^2$
Gravity constant (g)	$9.81\ m/s^2$
Model Mass(m)	$2.798\ kg$
304 Stainless Steel Density (ρ)	$7\ 900\ kg/m^3$
Centrifugal Stress (σ_{CF})	$121.8\ MPa$

Dang Van proposed a method for assessing HCF for components subjected to multiple stresses. Therefore, the theory of Dang Van is useful for stress-based multiaxial creation for HCF life prediction. The method relates the variation of the stress state in material points to a critical parameter that ought not to be reached.

The Dang Van multiaxial mathematical method is set and applied with parameter equations that algorithm in Fe-Safe software for fatigue life prediction of mistuned steam turbine blades subjected to variation in blade geometry. [10] studied the method and uses a simplified formula in Eq. 5.

$$\tau_{max}(t) + \alpha_{DV}\sigma_H(t) \leq \tau_w \quad 5$$

where τ_w is the fatigue limit in reversed torsion, σ_w is the fatigue limit in pure bending, and $\sigma_H(t)$ is the instantaneous hydrostatic component of the stress tensor.

III. NUMERICAL MODELLING

A. *Geometric Model Development*

A scaled-down of the model of a complete LP steam turbine bladed disk assembly used in practice was developed. The ultimate purpose of the numerical model is to simulate the HCF life of mistuned steam turbine blades economically. Furthermore, the model comprises 8 simplified straight cantilever blades with dimensional varies as shown in Fig. 3. During the model development, the 5 *mm* fillets at the blade roots conforming to [6] experimental model were used to reduce the severe stress concentration.

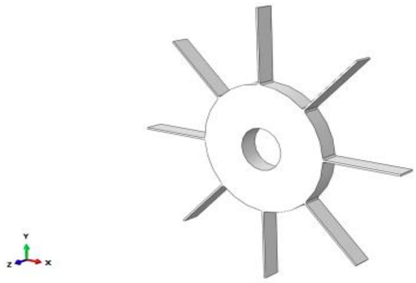

Fig. 3 FE model of a mistuned steam turbine-bladed disk

B. *Material Selection*

ASTM 304 stainless steel material is most widely used for steam turbine blades. The material has a relatively high ductility characteristic which is ideal for steam turbine blades that are subjected to high centrifugal and bending loads. This ductile allows root contact to undergo slight plastic deformation to better share the loads to eliminate the failure and crack [3]. Hence ASTM 304 was chosen material for this study.

The material and fatigue properties of ASTM 304 stainless steel were obtained from [14] and validated by [15]. The properties are shown in Table II.

TABLE II
304 STAINLESS STEEL MECHANICAL PROPERTIES [14]**Hata! Başvuru kaynağı bulunamadı.**

304 Stainless steel	Typical value
Tensile Strength *(MPa)*	600
Yield Strength, (Offset 0.2 %) (*MPa*)	310
Elongation (Percent in 50 *mm*)	60
Hardness (Brinell)	170
Fatigue Endurance Limit *(MPa)*	240

C. *Mesh Selection*

Generation of mesh is a critical aspect in simulation studies, it controls the FE model calculation time and the FEA results accuracy. More time may be needed if using a larger number of elements and if using a small number of elements, the accuracy of the solution can be compromised. Therefore, a mesh size of 1 *mm* was selected as the best mesh to use at the blade roots and default mesh size on the rest of the FE model.

D. *Mass Check*

The ASTM 304 stainless steel material was used to manufacture the 8 steam turbine bladed disk. The manufactured prototype disk model was subjected to variation in geometric dimensions, meaning the that the thickness, length, and width were altered to be different in a symmetrical pattern.

The FE model mass was found to be 2.798 *kg* and the experimental prototype manufactured using Computer Numerical Control (CNC) wire cutter model mass was measured to be 2.729 *kg* [6] as illustrated in Table III.

TABLE III
FE AND PHYSICAL MODEL MASS COMPARISON

FE model mass	2.798 *kg*
Physical model mass	2.729 *kg*
Percentage Error	2 %

Fig. 4 and Fig. 5 illustrates the geometry of the simplified, scaled-down mistuned blade disk model in 3-Dimensions and 2-Dimensions, respectively. The 2- Dimensional drawing units are in *mm*. When calculation the percentage error, Eq. 5 was used. Where *a* is the first value and *b* is the second value from the table.

$$\left[\frac{|a-b|}{\frac{a+b}{2}}\right] \times 100 \quad (5)$$

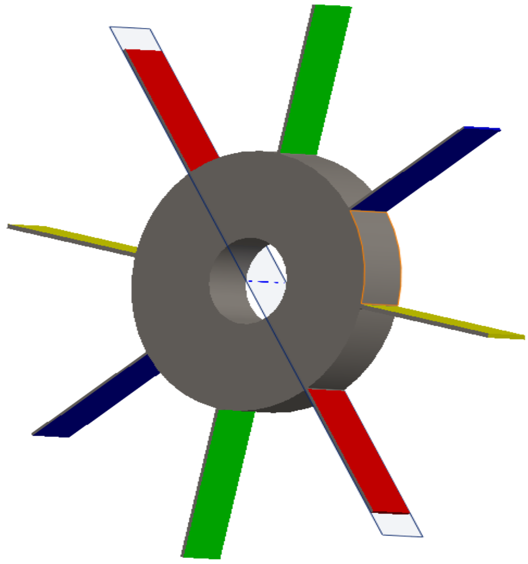

Fig. 4 Simplified scaled-down 3-Dimentional (CAD) steam turbine bladed disk model

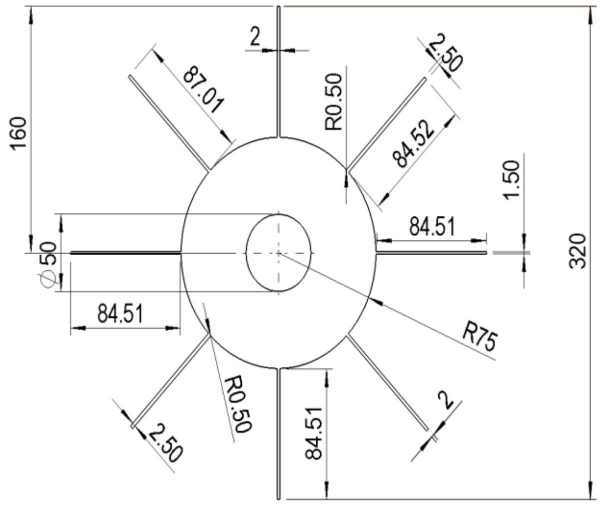

Fig. 5 Dimensional mistuned steam turbine bladed disk model

E. *Model Steady-State Stress Analysis at Critical Points*

The finite element model boundary condition was initially assigned by applying the rotational constraints to the centre hub surface to prevent axial movements within the simulation. Therefore, a rotational speed of 314.159 *rad/s* angular velocity was applied in an axial direction acting along the centre hub. The stress analysis distribution was acquired in this step. The von Mises stress is equivalent to the means stress [4].

The maximum FE simulation mean stress at blade roots is anticipated as 175.2 *MPa* as illustrated in Fig. 6. The mathematical centrifugal was calculated to be 121.8 *MPa.* Both stresses are compared to the material yield strength of 310 *MPa*, this can be concluded that no yielding takes place as both the analytical and finite element results are below the maximum yield strength of the material. The mean stress results, which are depicted in Fig. 6 also reveal that the peak stress location is in the blade root area, which is in agreement with the initial anticipation.

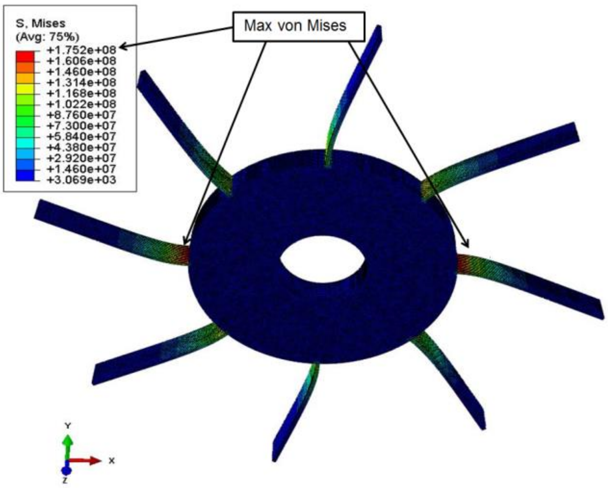

Fig. 6 Mean stress distribution

F. *Modal Analysis Results*

Mistuned steam turbine blades subjected to geometric variation blades were explored and simulated in 20 different FE model cases to study the behaviour of their modal parameter. 19 are different mistuned and 1 is a tuned reference case as illustrated in Table V, where t is the blade thickness, L is the length of the blade, and w is the width of the blade.

G. *Natural Frequencies*

Natural frequencies are critical for this study, hence eigen frequencies were extracted in FE modal analysis. Due to the asymmetrical nature caused by our 8 simplified cantilever blades of mistuned steam turbine, there are unrepeated natural frequencies [6]. Table IV summarises the first 8 cases of the mode's natural frequencies with corresponding operational rotation speed in FE analysis and validated with literature experimental frequencies of [6]-[9]. The highest natural frequency difference is noted on mode 7 and 8 due to the blade twisting and bending.

TABLE IV
SUMMARY OF MODES OF NATURAL FREQUENCIES

Mode	**Natural Frequencies *(Hz)***	**Experimental Natural Frequencies *(Hz)***	**Percentage difference (%)**
1	233	222	4.84
2	1430	1352	5.61
3	1878	1770	5.92
4	1960	1935	1.28
5	2263	2256	0.3
6	4002	3997	0.13
7	5780	5421	6.41
8	6163	6544	5.99

The amount of the stress distribution in the modal simulation used to identify the model hot spots. Fig. 6 with arrow representing area of initial crack. Moreover, Fig. 6 confirms that the cracks initiate at the blade roots with the literature of [6]-[9].

IV. LIFE PREDICTION

In this process the maximum dynamic stress distribution results acquired from the steady state dynamic stress analysis are later exported as a .obd (file) from ABAQUS software into Fe-Safe software.

Fe-Safe software was used to run all 8 cases of the fatigue life models for arbitrary geometric mistuned patterns. Where by the ideally tuned case model in Fig. 7 turned into a mistuned model in Fig. 5.

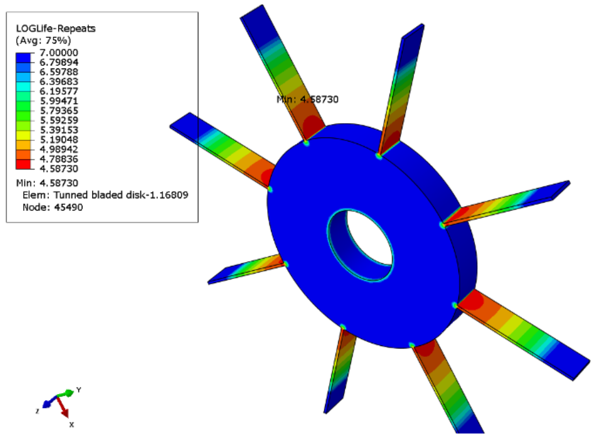

Fig. 7 Fe-Safe fatigue life of tuned model

The observation in LOGLife in Fig. 7 assures that the Fe-Safe expresses the actual life of the mistuned steam turbine-bladed disk. It can be concluded that the fatigue failure occurs at the stress-concentrated point on the blade root area depicted in Fig. 6 and Fig. 7 previously projected by the FEA results.

Moreover, Table V illustrates detailed fatigue life cycles due to of length, width, and thickness dimensions changed in 20 cases in descending order.

TABLE V
SENSITIVITY ANALYSIS

Case	**Model Description**	**Fatigue Life (Cycles)**
1	Reference/Tuned model	4.587×10^6
18	Decreased blade length of 2 adjacent (-1 mm, L)	4.574×10^6
19	Increased blade width of 2 adjacent (+1 mm, w)	4.188×10^6
20	Decreased blade width of 2 adjacent (-1 mm, w)	4.130×10^6
9	Mixed mistuned pattern (+/- L,t & w)	4.097×10^6
6	Increased blade width of 2 opposing (+1 mm, w)	4.093×10^6
7	Decreased blade width of 2 opposing (-1 mm, w)	4.090×10^6
10	Mistuned pattern (rubbing/rotor misalignment) (-L)	4.090×10^6

8	Mixed mistuned pattern (+/- L,t & w)	4.043×10^6
12	Mixed mistuned pattern (+/- L& t)	4.020×10^6
5	Decreased blade length of 2 opposing (-1 mm, L)	4.016×10^6
13	Mixed mistuned pattern (+/- w & t)	4.012×10^6
2	Increased blade thickness of 2 opposing (+1 mm, t)	3.997×10^6
11	Mixed mistuned pattern (+/- L& w)	3.980×10^6
4	Increased blade length of 2 opposing (+1 mm, L)	3.960×10^6
17	Increased blade length of 2 adjacent (+1 mm, L)	3.955×10^6
15	Increased blade thickness of 2 adjacent (+1 mm, t)	3.911×10^6
16	Decreased blade thickness of 2 adjacent (-1 mm, t)	3.855×10^6
3	Decreased blade thickness of 2 opposing (-1 mm, t)	3.835×10^6
14	Mixed mistuned pattern (+/- L,t & w)	3.830×10^6

TABLE VI
ANALYTICAL AND SIMULATED FATIGUE LIFE DIFFERENCE

Case	Analytical Fatigue Life (cycle)	Simulated Fatigue Life (cycle)	Percentage Error (%)
1	4.435×10^6	4.587×10^6	3.37
18	4.357×10^6	4.574×10^6	4.86

Table VI illustrates the comparison and validation between analytical and simulated life cycles results of randomly chosen cases. Brown Miller strain-life equation gives realistic life estimation and the life $2N_f$ are calculated by solving Eq. 6 for analytical fatigue life cycles. Moreover, Eq. 5 was applied to calculate percentage error.

$$\frac{\Delta\gamma_{max}}{2} + \frac{\Delta\varepsilon_n}{2} = 1.65\frac{\sigma'_f}{E}(2N_f)^b + 1.75\varepsilon'_f(2N_f)^c \qquad 6$$

V. CONCLUSIONS

Engineering approximations regarding numerical data has been made whenever essential due to insufficient information on the mistuned blades, turbine performance and material fatigue data. The analytical life cycles and Fe-Safe software shows the percentage difference of 3.37% to 4.86%. The procedure was to individually manipulate the blade geometries based on these geometric configurations and their respective fatigue cycles to failure.

The maximum simulated fatigue life cycle shows to be 4.587×10^6 on tuned bladed disk and then 4.574×10^6 on mistuned bladed disk on length adjustment. The fatigue initiate at the blade root. Fatigue life is most sensitive to changes in blade length, followed by the width, and then the thickness, in this order.

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