Fatigue Analysis of Low-Pressure Steam Turbine Blades during Start-up using Probabilistic Concepts

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Abstract
The deterministic approach in engineering design and analysis compensates for the uncertainty in specific system parameters through an empirical factor of safety. The probabilistic approach incorporates this randomness or uncertainty and ensures optimal use of resources without compromising safety. This paper presents a probabilistic model to determine the fatigue life of a low-pressure steam turbine blade under the influence of transient loading conditions characteristic of a start-up. The methodology consists of simulating the randomness in the values of structural damping, rotational speed, and the blade material's elastic modulus through Finite Element Analysis (FEA). Monte Carlo simulation is utilized to generate random stress and fatigue cycle values obtained from the results of Finite Element Analysis. The Palmgren-Miner rule is then used to determine the probabilistic fatigue life. Results show that the deterministic approach underestimates fatigue life. Also, it fails to demonstrate the relative impact of these random variables on fatigue life. The probabilistic model developed for this study clearly establishes that randomness in the values of turbine rotational velocity has the most significant effect on the probabilistic fatigue life of a low-pressure steam turbine blade.

Keywords
Probabilistic Approach, Fatigue, Finite Element Analysis, Monte Carlo Simulation, Palmgren-Miner Rule.

Introduction
Steam turbine plants account for more than half of the total energy production in most developed as well as developing countries. An industrial steam power plant typically consists of three types of turbines: the High Pressure (HP), Intermediate Pressure (IP) and Low Pressure (LP) steam turbines. Despite rapid developments in the renewable energy sector, coal and gas-fired steam turbines are expected to continue fulfilling the rigorously increasing energy demands till at least 2035 [1]. The global energy situation not only demands new designs for more efficient steam turbines, it also warrants maintenance and refurbishment of current power plants in operation [1].
Reliability assessment plays a significant role in the aforementioned maintenance and refurbishment of currently operational steam turbines. To ensure an uninterrupted supply of energy, it is necessary to understand the causes of fatigue failure and the methods that can be employed to prevent it. The rotor blades of a steam turbine experience significant mechanical stresses due to high rotational velocities and pressure of the incoming steam. In addition, escalated temperature in the blades due to high temperatures of the interacting steam result in thermal stresses. If the turbine is subjected to an extensive number of start-ups, the cyclic

nature of the aforementioned loadings may prove sufficient to initiate and propagate a fatigue crack. Booysen et al [2] studied fatigue analysis of a last stage low-pressure steam turbine. They concluded that resonant stresses as the turbine passes through the critical speed result in the greatest fatigue damage. In the case of this study, the resonant stresses were significantly lower in magnitude and did not contribute nearly as significantly to the fatigue damage accumulated during a start-up. Thermal stresses also did not contribute as significantly to the overall fatigue damage. The greatest accumulation of fatigue damage occurred when the turbine reached the maximum rotational velocity at the end of a start-up resulting in the greatest magnitude of centrifugal stresses on the blade. Reliability assessment in the form of a fatigue life analysis may prove invaluable for the timely replacement of steam turbine blades nearing the end of their fatigue lives.

Relevant literature lists two methods that can be employed to compute the remaining life of low-pressure steam turbine blades. The deterministic approach is the conventional approach in engineering design and analysis. It makes use of an empirical factor of safety. Uncertainty in the values of certain system parameters, such as rotational speed and material properties in the case of steam turbines, is compensated through overly-conservative values of these factors of safety. This renders the deterministic approach consistently prone to over-design. The probabilistic technique incorporates the uncertainty and variation in the values of certain system parameters. It also elaborates the relative impact of the variation in each of these uncertain parameters on the primary outcome under study.

Details of the Model

The last stage long blades of the low-pressure steam turbine are most susceptible to fatigue failure [2]. Figure 1 shows the prominent characteristics of the blade that was selected for the fatigue analysis. The root of the blade has a curved fir-tree profile which is very common among modern last stage steam turbine blades [3].

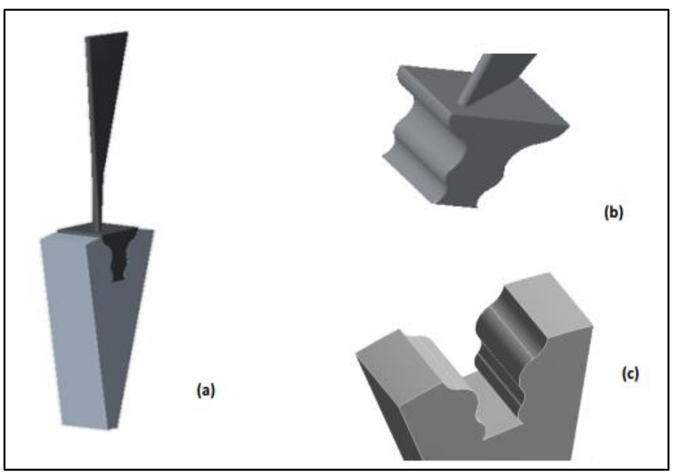


Figure 1: a) Blade along with rotor sector b) Blade root profile c) Rotor sector profile

The selected low-pressure blade was free-standing. For the purpose of simplicity, a 3D model of a rotor sector was generated instead of the complete rotor. Cyclic symmetry was then specified as a boundary condition. The contact between the blade and the rotor sector is non-linear in reality. Non-linear contact was simulated through the selection of frictionless contact in ANSYS.

The material selected for the aforementioned turbine blade model was 12% chromium stainless steel (X22CrMoV12-1). Booysen et al [2] say that this material is widely used for low-pressure turbines in Eskom power stations in South Africa due to its high strength and corrosion resistance properties. Fatigue analysis requires both the mechanical properties such as the yield strength, ultimate tensile strength and Young's modulus as well as the material fatigue properties. Booysen et al [2] obtained both the mechanical and fatigue properties experimentally. The same properties have been used in the present study.

Booysen et al [2] conducted tensile tests on nine different specimens created from samples obtained from the root of a failed low-pressure turbine blade. Variability in the material properties from sample to sample was observed, which substantiates the need for multiple samples. Table 1 summarizes the results of the testing.

Table 1: Summary of Tensile Testing Results (Booysen, 2014)

	Young's Modulus (GPa)	Yield Strength (MPa)	Ultimate Tensile Strength (MPa)
Mean	215	723	893
Standard Deviation	8.42	18.8	17.5

The fatigue strength co-efficient $\sigma_f{}'$ and fatigue strength exponent b were determined from uniaxial fatigue testing [2]. Subsequently SN curves for completely reversed loading with a stress ratio of -1 and tension-tension loading with a stress ratio of 0.1. Figure 2 illustrates the SN curves for the two stress ratios. The experimental values of $\sigma_f{}'$ and b were 1194 MPa and -0.077 for the first case and 970 MPa and -0.087 for the second case.

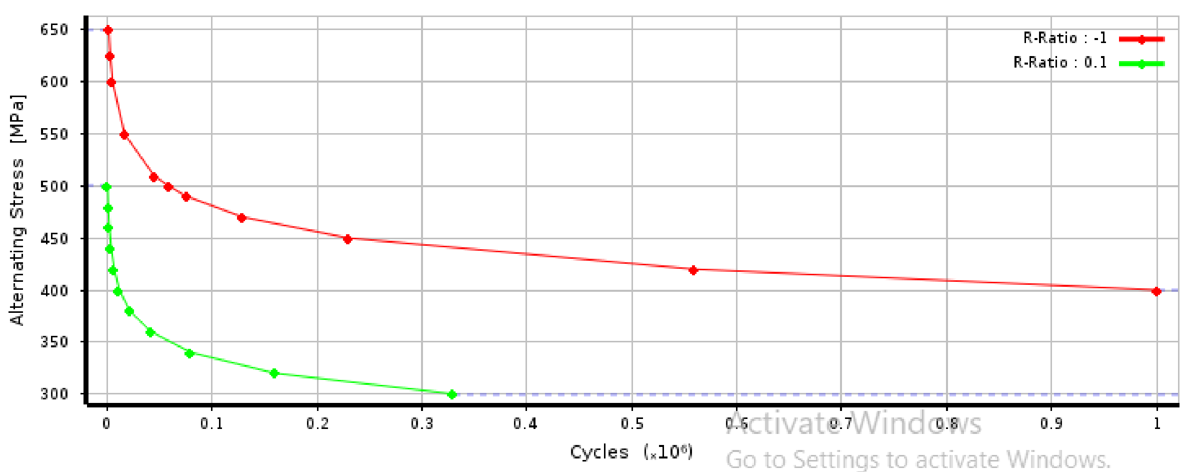


Figure 2: SN Curve for R-ratio -1 in red and R-ratio 0.1 in green (Booysen, 2014)

Benchmarking of the Model

In order to ensure that the results of this study are reliable, an important part of the research involved application of the probabilistic model to the turbine blade and operating conditions considered by Booysen et al [2] for their analysis. These operating conditions consisted of the turbine passing through resonant conditions encountered at a critical speed of 2000 RPM. The normal steam pressure on the blade was computed to be 694 Pa by Booysen et al [2]. Frictionless contact was assumed between the rotor sector and the blade root. Thermal stresses due to the temperature of the steam were not considered. Table 5 shows the comparison between the probabilistic fatigue life calculated in the earlier study [2] and the fatigue life calculated using the model presented in this study. The table clearly demonstrates that the model very closely reproduces the life determined by the benchmark model.

Table 2: Comparison between life calculated by [2] and calculated life using the current model

Benchmark Life	Calculated Life
7892 start-ups	7931 start-ups
% Difference	0.4%

Finite Element Analysis

Finite Element Analysis using ANSYS software package was employed to simulate the transient loading experienced by a last stage low-pressure turbine blade. The variation or randomness in the values of the selected random variables was incorporated into the FEA. These random variables were selected to be the turbine rotational speed, blade damping ratio and the material's Young's modulus. The results of the FEA were then utilized as inputs for the probabilistic model to compute the probabilistic fatigue life. Figure 3 shows a flow chart illustrating the relationship between the FEA and its subsequent analyses and the probabilistic model.

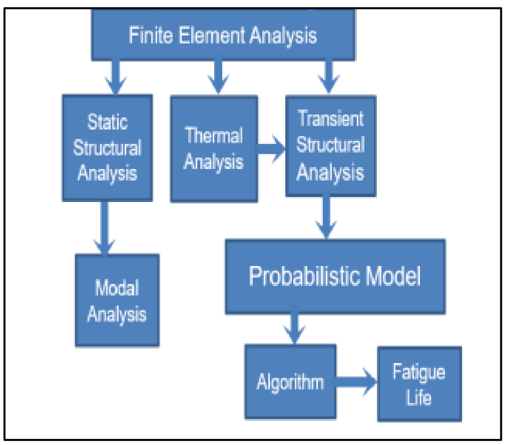


Figure 3: Flowchart for FEA

The first step in the methodology employed for this research is the static structural analysis. The analysis served two purposes. The first was confirmation of whether or not the blade was yielding statically under the considered operating conditions. The second purpose was the computation of the maximum stress at the critical location of the turbine blade. This maximum stress value is utilized in the probabilistic model to determine the fatigue life. Figure 4 shows the boundary conditions that were applied to determine the static stresses.

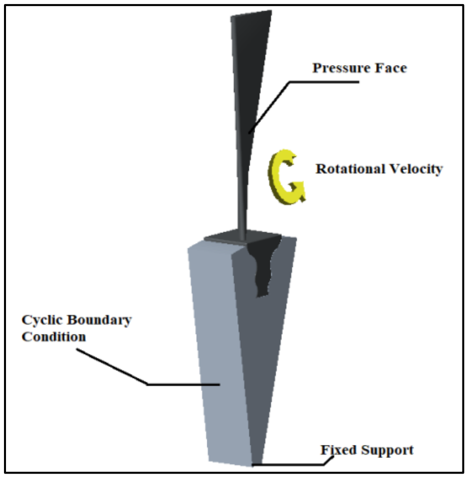


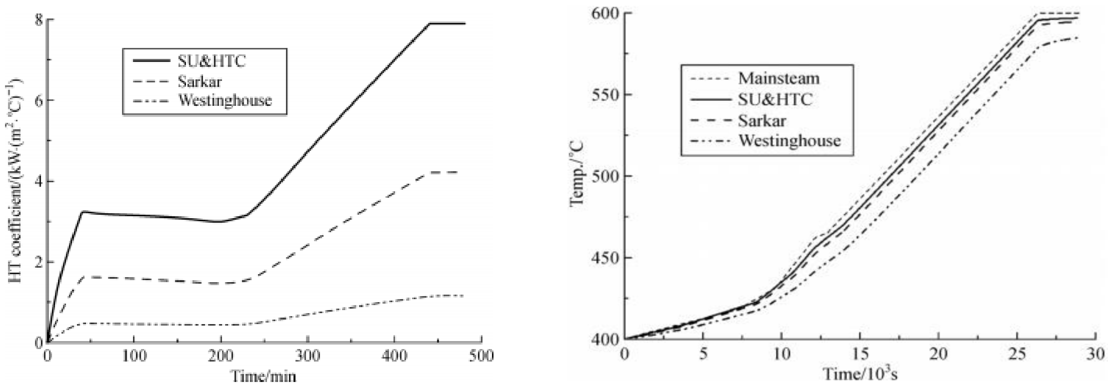
Figure 4: Boundary Conditions

A rotational speed of 3000 RPM was applied to the blade and rotor sector. The rotor sector was fixed at the bottom face as shown in the image. Booysen et al [2] computed the pressure magnitude of 680 Pa acting normal to the pressure face shown in the figure. The pressure magnitude is very low and characteristic of a low-pressure steam turbine. Because a rotor sector was considered instead of the complete rotor, cyclic boundary condition was applied to the model. The maximum stress magnitude at the critical location was 530 MPa. This value is much less than the yield strength of the material (723 MPa), and it was concluded that the blade did not yield statically.

Next, thermal analysis was performed. The purpose of the thermal analysis was to ensure that the maximum temperatures experienced by the blade were below the melting temperature of the material (X22CrMoV12-1) considered. The model that was employed to simulate the thermal effects in a low-pressure steam turbine was based on the algorithm suggested by Moroz et al [4]. They explained that the primary mode of heat transfer between the steam and the blade is convection. The important parameters in this model include the initial temperature of the blade and the inclusion or exclusion of steam condensation.

The initial temperature of the blade plays a significant role in the development of thermal stresses on the blade. The selection of the initial blade temperature depends on the type of start-up under consideration. A start-up can be cold, warm or hot depending upon the relative values of the initial blade temperature. For the present study, an initial temperature of 30^{o}C, which is representative of a cold start-up, was considered. Moroz et al [4] explain that if the saturation temperature of the steam drops below the blade temperature, condensation occurs in the low-pressure turbine. In such a case, the saturation temperature of the steam is utilized as the ambient temperature in locations where condensations has occurred along with specific values of the convection heat transfer co-efficient. Condensation effects were, however, assumed negligible for the present study.

Since convection was designated as the mode of heat transfer, a convective boundary condition was applied to the outer protruding part of the blade, excluding the root or base attached to the rotor sector. Figure 5 shows the transient nature of the heat transfer co-efficient with regards to the duration of a complete start-up. These values, along with the time-dependent steam temperature, for a steam turbine during a cold start-up were obtained from the work of Guo et al [5].



Figures 5: Variation of steam heat transfer co-efficient during a start-up (left) and steam temperature (right)

The three lines are differentiated by the empirical formula that was employed to determine the co-efficient values. For the present study, the values derived using the SU&HTC formula were considered. A transient thermal analysis on ANSYS produced the resultant temperature changes shown in Figure 6. The results of the thermal analysis were imported into the subsequent transient structural analysis to determine the thermal stresses on the blade.

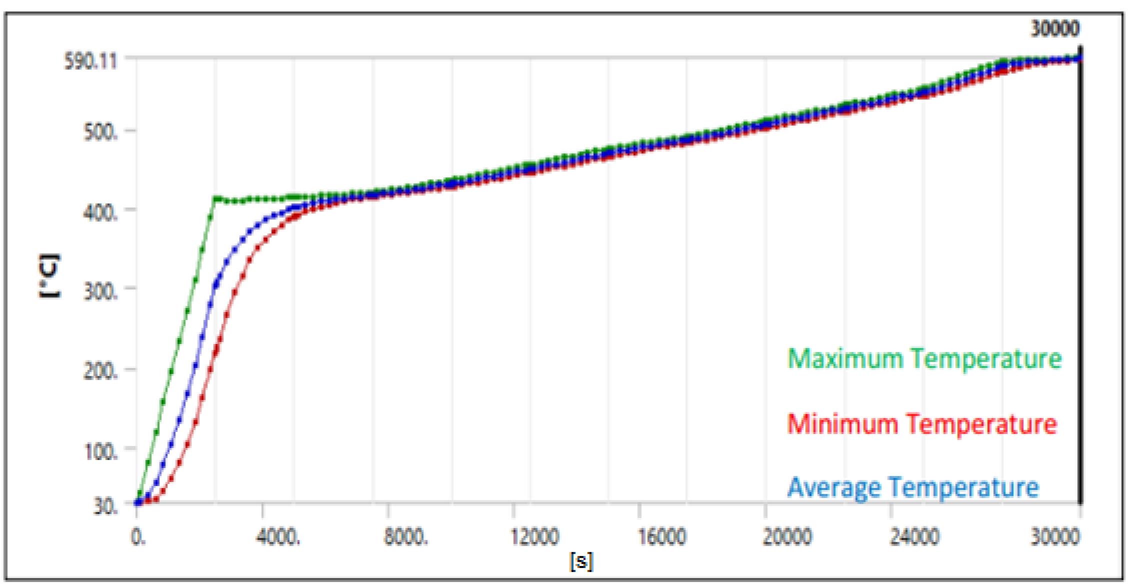


Figure 6: Temperature rise of the blade during start-up

In order to determine the time-variant mechanical and thermal stresses experienced by the blade during one start-up, a transient structural analysis was performed. The transient temperature of the steam and the convection heat transfer co-efficient values by Guo et al [5] were used as the boundary conditions. The temperature changes during the start-up as shown

in Figure 6, with time in seconds on the x-axis, were imported into the structural analysis to determine the thermal stresses. The duration of a typical steam turbine start-up is approximately six to eight hours. As can be seen from the x-axis in Figure 4, the turbine blade reaches a steady temperature of 590.11°C at the end of 30,000 seconds which is a lot less than the melting temperature of the material. The melting temperature for 12% chromium stainless steel (X22CrMoV12-1) falls in the range between 1370°C to 1530°C.
Booysen et al [2] observe that the simulation of a complete start-up is impractical and extremely computationally expensive. However, the point where the blade undergoes maximum mechanical stresses can be considered to account for more than half of the accumulated fatigue damage. Thus, a structural analysis of the turbine blade during the short duration that the turbine passes through this point of maximum stress can be considered a good alternative to simulating a complete start-up. Booysen et al [2] concluded that the maximum stress occurred as the turbine encountered resonance while passing through a critical speed at 2000 RPM. The maximum resonant stress was in excess of 200 MPa. However, a modal analysis followed by a harmonic analysis in the case of this study showed that the resonant stresses were insignificant as can be seen in figure 7. The point of maximum mechanical stress selected for the present study was when the turbine reached its maximum rotational velocity of 3000 RPM. This was due to the fact that centrifugal stresses, which have the largest contribution to overall mechanical stresses, encountered during a start-up were significant at higher rotational velocities.

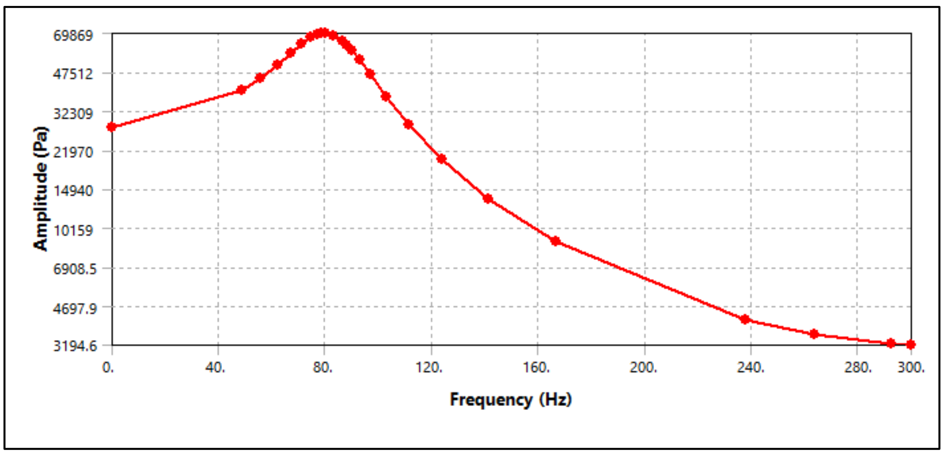


Figure 7: Result of the harmonic analysis

Random Variables

The computation of the probabilistic fatigue life requires the consideration of the uncertainty or randomness in the values of certain system parameters. The effect of this variation in values on the transient loading experienced by the blade was illustrated during this step. The three random variables that were selected for the present study are turbine rotational velocity ω, blade damping ratio ζ and the material's Young's modulus E. Damping ratio of the blade was considered as a random variable by Booysen et al [2] in their probabilistic study of steam turbines. The normal distribution of the variable presented in the study was utilized for the present research. Zhang et al [6] referred to turbine rotational velocity as well as material properties such as yield strength, Young's modulus and even fatigue properties as reasonably good random variables. The normal distribution of the material's Young's modulus as well as rotational speed was obtained from the work of Duan and Wang [7]. Table 2 enlists all the three random variables along with their statistical information.

Table 3: Random variables and their statistical parameters

	Rotational Velocity ω (RPM)	Damping Ratio ζ	Young's modulus E
Mean	3000	0.3	217
Standard Deviation	30	0.08	10.85
Distribution	Normal	Normal	Normal

The variations in the aforementioned random variables affect the transient mechanical stresses influencing the blade during a start-up. The Taguchi method was used to generate twenty-five different combinations of the three random variables. The twenty-five combinations were then used as inputs for the transient structural analysis. The boundary conditions were kept the same as in Figure 4 for static structural analysis. There was a slight variation in the magnitude of the excitation pressure. The pressure was kept harmonic or sinusoidal with an amplitude of 680 Pa, reflective of the transient nature of applied loadings during a start-up. The exact frequency of the sinusoid was derived from the work of Booysen at al [2]. Figures 8 and 9 show two stress history plots along with the combinations of the three random variables that were used as inputs for their generation.

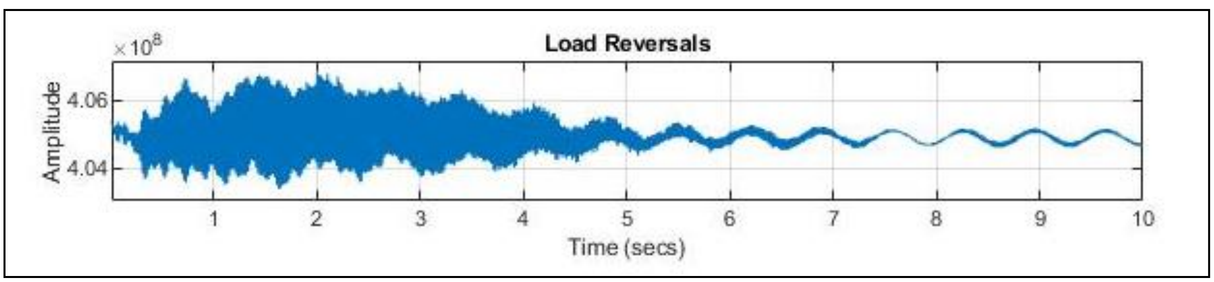


Figure 8: Stress history plot for 3033 RPM, 206 GPa Young's Modulus and 0.199 Damping Ratio

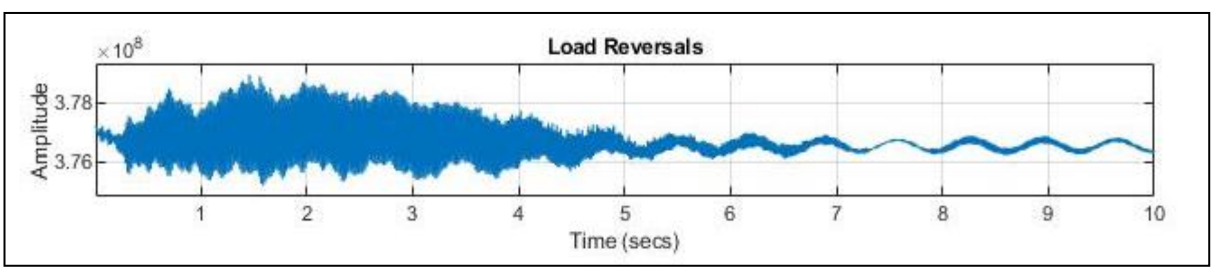


Figure 9: Stress history plot for 3016 RPM, 231 GPa Young's modulus and 0.199 Damping Ratio

Monte Carlo Simulation

The Monte Carlo technique is commonly used in the computation of probabilistic fatigue life [8]. The first step is the selection of the random variables followed by illustrating their variation through probability density functions. The next step involves utilizing these density functions or distributions to generate random values. These random values are then inserted into a deterministic relation for the calculation of the probabilistic outcome. The random variables considered for this study are the turbine's rotational velocity; the blade's damping ratio and the blade material's Young's modulus. Finite Element Analysis was performed to represent the effect of the uncertainty in the values of these variables on the time-variant stresses acting on the blade during a start-up. Twenty-five stress-history plots were generated for each of the twenty-five combinations generated by the Taguchi method.

Rainflow cycle diagrams for each of the twenty-five stress-history plots were generated using ANSYS to count the number of fatigue cycles and their associated alternating and mean stresses. The maximum alternating stress amplitudes and their associated fatigue cycles from each rainflow matrix were plotted against one another as shown in Figure 11. The rainflow cycle diagram for each stress-history plot represented the transient loading in the form of thirty two blocks or bins. Each bin represented an alternating stress magnitude, a mean stress magnitude and the associated fatigue cycles.

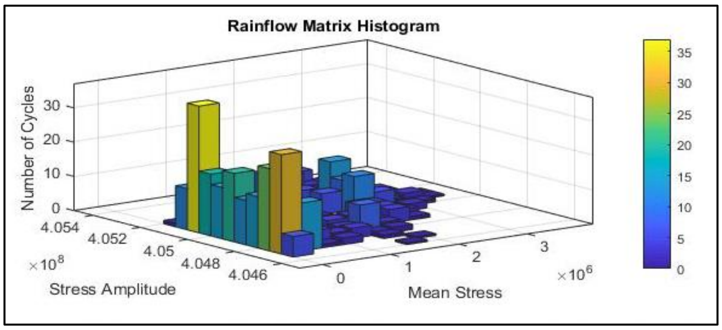


Figure 10: Rainflow matrix for stress-history plot in figure 7

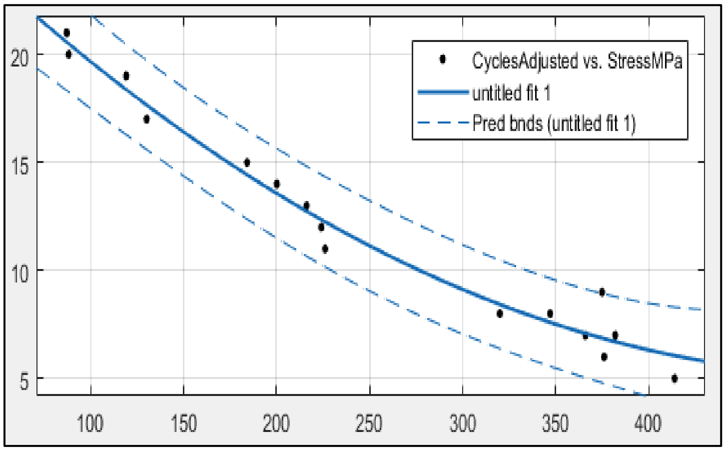


Figure 11: Stress-cycles plot

Random values of alternating stresses and fatigue cycles from the stress-cycles plot in Figure 11 were inserted into the Palmgren-Miner rule for computing the fatigue damage.

$$D_i = \frac{n_i}{N_{fi}} \tag{1}$$

n_i in the numerator represents one random value of the fatigue cycles generated from the stress-cycles plot. N_{f_i} in the denominator represents one random value of the fatigue cycles to failure. [2] utilized the following relation derived from Morrow's stress-life equation to determine to generate the random values of N_{f_i} from the values of alternating stresses and the static mean stress.

$$N_{f_i} = \frac{1}{2}\left(\frac{\sigma_{ar_i}}{\sigma_f - \sigma_m}\right)^{\frac{1}{b}} \tag{2}$$

σ_{ar_i} is the alternating stress magnitude from the rainflow matrices, and σ_m is the static mean stress obtained from the static structural analysis at the critical location. The probabilistic fatigue life was calculated using the following relation.

$$B_f D_i = 1 \tag{3}$$

D_i represents the damage accumulated for one start-up. Fatigue life B_f is given in terms of the number of start-ups before the turbine blade fails.

Results and Discussion

The probabilistic method, as described in the previous chapter, employed the rainflow matrices for the twenty-five different combinations of the random variables. Monte Carlo simulation through MATLAB was used to generate 10^8 random values of fatigue cycles and alternating stress magnitudes. These values were then used as inputs into the Palmgren Miner rule to compute the probabilistic damage and life. The fatigue life was computed using the stress-life approach. The Goodman mean stress-correction theory was used to cater for the mean stresses involved.

Table 4: Comparison between deterministic and probabilistic approach

Model	Cycles (Start-ups to failure)
Deterministic	7856
Probabilistic	8546
% Difference	8.8%

Table 3 draws a comparison between the fatigue life values of both the deterministic as well as the probabilistic approach. The term 'cycle' here refers to one complete stress-history plot or one complete start-up. The fatigue life with a certain value of cycles to failure will thus also indicate how many additional start-ups the turbine blade can withstand before fatigue failure occurs. As mentioned earlier, the fatigue damage was accumulated, however, not over the complete start-up, but over a brief period of time during the start-up when maximum stresses are experienced by the blade. This point occurs when the turbine reaches or is about to reach the maximum rotational velocity which in this case is 3000 revolutions per minute (RPM).

The deterministic life is 7856 start-ups before failure occurs. This value is very similar to the fatigue lives computed in literature for low-pressure steam turbine blades although differences in shape of the blade root, material properties and whether or not the blade is free-standing or uses a shroud induces minor changes. The probabilistic life is 8546 start-ups with an 8.8% increment. This substantiates the initial hypothesis that the deterministic is overly-conservative at times and underestimates fatigue life. The deterministic approach does not take into consideration the randomness of some of the variables involved in steam turbine blade analysis. It considers mean values of these variables determined through statistical techniques and utilizes a factor of safety to compensate for the removal of this uncertainty or randomness in values. As a result, occurrences of low probability such as a steep variation in the value of one variable and its impact on fatigue life are ignored. The probabilistic approach, however, considers all possible occurrences, whether highly probable or not, and considers their impact on fatigue life. As a result, the probabilistic life is much closer to the actual fatigue life value. This is in perfect agreement with the results of Welling and Lynch [9] who also established the overly conservative nature of the factor of safety approach.

The probabilistic approach not only provides accurate values of fatigue life, but it also quantifies the impact of each random variable on the maximum alternating stress encountered by the blade and thus the fatigue life [8]. The former statement has already been verified in detail in the preceding paragraphs. The probabilistic model developed for the present research also verifies the latter observation.

Table 5: Effect on maximum stress on the blade due to 1% change in random variable

Random Variable	% Change in Maximum Stress
Rotational Velocity	4.727
Damping Ratio	0.225
Young’s Modulus	0.61

Table 4 shows the impact of introducing a 1% change in the values of each of the random variables on the maximum alternating stress encountered by the blade during a start-up. As can be seen, the most significant change on the stress is introduced by varying the value of the rotational velocity. This clearly depicts that the maximum stress amplitude on the rainflow matrix is relatively more sensitive to changes in the values of rotational velocity as compared to damping ratio of the blade and the material’s Young’s modulus. Rotational velocity also has the greatest impact on the fatigue life since the maximum stress amplitude on the rainflow matrix accounts for more than 50% of the total fatigue damage [2].

Conclusion

The deterministic approach in engineering design and analysis uses a factor of safety to compensate for the uncertainty or randomness in certain system parameters. The deterministic approach considers average values of system parameters obtained through statistical techniques and does not account for highly improbable occurrences. It also does not quantify the relative impact of these uncertain parameters on the primary outcome. As a result, the deterministic approach in engineering is often overly-conservative and does not permit optimal use of resources without risking safety.

The probabilistic approach considers the randomness of system parameters. The present study determined the fatigue life of low-pressure steam turbine blades using the probabilistic approach. The turbine rotational speed, material's Young's modulus and the blade's damping ratio were considered as random variables. Results indicate that the probabilistic approach more accurately predicts remaining fatigue life of the blade. It also illustrates the relative impact of each of the three random variables on accumulated fatigue damage and remaining life.

Nomenclature

$\sigma_f{}'$	fatigue strength co-efficient [kg m s^{-2}]
b	fatigue strength exponent [dimensionless]
D_i	fatigue damage [dimensionless]
n_i	fatigue cycles [dimensionless]
N_{f_i}	fatigue cycles to failure [dimensionless]
σ_{ar_i}	rainflow alternating stress [kg m s^{-2}]
σ_m	static mean stress [kg m s^{-2}]
ω	rotational velocity [s^{-1}]
ζ	damping ratio [dimensionless]
E	young's modulus [kg m s^{-2}]

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