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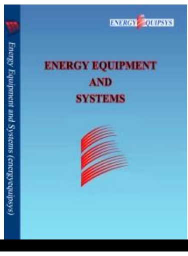


# Life span prediction for the last row blade of steam turbine

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**ABSTRACT**

*The steam turbine of Ramin Ahvaz power plant is a k300-240 model manufactured by the Russian power machine company, which has six units of 300 MW. The above turbine has a rotational speed of 3000 rpm; its final blade weight is 9.2 kg. Its centrifugal force causes one of the most important and effective stresses in the blade.*
*In this research, the first step, the forces acting on the blades such as centrifugal force are investigated, and then the stresses of these forces are calculated. By using these calculations and the properties of the blades, estimation of the blade life is made by applying stress-life correction coefficients. In the following, by using an engineering software named ABAQUS, a sample of the last row of fins is simulated. This simulated specimen after meshing is stress analyzed. By this method, the results of manual computations are compared by using different life criteria such as Goodman's and Gerber's life criteria results obtained by Abacus (finite element software). Finally, we evaluate the validity of the previous steps through the manufacturer's documentation.*

**Keywords:** Turbine Blade, Life Prediction, Tension, Centrifugal Force

## 1. Introduction

In the design and analysis of the industrial components, three important methods of fatigue life are used:

1- Tension-life method
2- Strain-life method
3- Linear fracture mechanics method

In these methods, the component life shown by the number of cycles (N), are number of cycles of a given load where the component can withstand until failure. If the component's lifespan is between N= 1 cycle and $10^3$cycles, it will be called the low-fatigue class, while the fatigue-laden ones will be used for N more than$10^3$.

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The tension-life method, which is based solely on stress levels, is the least accurate method. However, it is the most archaic method because it is the easiest method for a wide range of designs, which has many data and also, it is suitable for retirees.

Determination of material resistance in fatigue loads is performed by oscillating or variable loading with a specified amount and counting oscillations or reciprocating stresses until the sample is crushed. The Moore Speed Beam Testing Machine is the most commonly used type of fatigue test.

Computing the fatigue strength of any material requires many tests. In the rotary test, a constant bending load is applied, and the number of turns of the beam until failing is counted. The first test is performed with stress slightly lower than the final stress of the specimen, and it continues until a little below than stress of the first test. At the end of all tests, the plot of the results in the S-N curve is drawn. The width of the S-N graph is called the fatigue resistance $S_f$. Wherever this resistance is mentioned, it must always go along with stress N and its repetition number. The S-N diagrams can be obtained either for the test sample or for any actual mechanical fragment. For steels, it is considered as the knee-bending diagram, which has no breaks for the upper number. The resistance associated with this knee is called the endurance or fatigue limit of $S_e$. The diagrams for nonferrous alloys are never horizontal, so they are not durable.

Life Prediction of the Last Stage Blades of the Steam Turbine could result in the following conclusions:

1- Fatigue Life assessment and determining the replacement time of turbine blades even if the blade's appearance does not indicate a problem.
2- Frequency reduction and preventing the failure of turbine blades, which is a common problem of steam power plants.

This research is based on the practical results and has specified in dealing with reforms conducted in local power plant steam turbine blades.

## 1.2. Calculation of Stresses Applied on Turbine Blades

### 1.2.1. Tension Caused by Centrifugal Force

The stress calculated from the centrifugal force caused by the rotor rotation is one of the most important and effective stresses. To achieve this stress, the centrifugal force must be divided into the surface of the blade cross-section [1, 2]:

$$\sigma_{cf}(x) = \frac{F_{cf}\ (x)}{A(x)} \tag{1}$$

In the above equation, the centrifugal force is equal to:

$$dF_{cf} = dm \cdot \omega^2(Rr + z) \tag{2}$$

$$dm = \rho.A(z).dz \tag{3}$$

As a result, the centrifugal force of the blades is calculated via the following formula:

$$F_{cf}\ (x) = \int_x^{lb} \rho \cdot \omega^2 \cdot A(z)(Rr + z) \cdot dz \tag{4}$$

According to the blade cross-section, if the blade cross-section is the same, the calculation is easier but in the steam turbine blade, where we have a variable cross-section, we need to consider the mathematical relationship of the blade cross-section and the blade length change:

$$A(z) = A_r \cdot (\frac{A_t}{A_r})^{\frac{z}{lb}} \tag{5}$$

First, apply the equation of the variable cross-section (5) in Eq. (4) and then integrate it. The following equation is derived from calculating the centrifugal force of the blade:

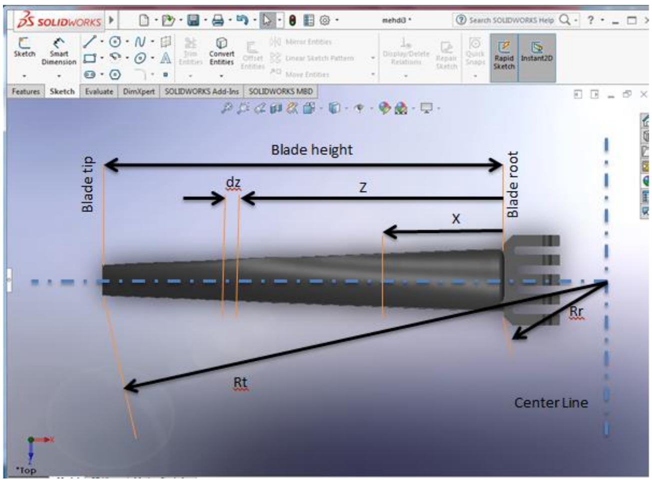


**Fig.1.** Geometrical profile of the turbine blades

$$F_{cf}(x) = \rho.\omega^2 \left[\left[\frac{A_r.(\frac{A_t}{A_r})^{\frac{z}{lb}}.Rr.Lb}{ln(\frac{A_t}{A_r})} + \frac{A_r.(\frac{A_t}{A_r})^{\frac{z}{lb}}.z.Lb}{ln(\frac{A_t}{A_r})} - \frac{A_r.(\frac{A_t}{A_r})^{\frac{z}{lb}}.Lb^2}{(ln\left(\frac{A_t}{A_r}\right))^2}\right]\right]_x^{Lb} \quad (6)$$

This equation indicates that, when x = Lb, the centrifugal force will be zero, and when x = zero, the force will be maximum and that would be the critical point of the blade intersection.

### 1. 2. 2. Bending Stresses Result from Fluid Impact

At this step, the bending stresses caused by the fluid impact are calculated.

The axial force caused by the encounter of the fluid on the blade is equal to [2]:

$$q_a = \rho_f . C_f (V_{w1} - V_{w2}) \quad (7)$$

In addition, the tangential force based on the impact of the fluid on the blade with neglecting the difference between the vertical entry velocity and the exit one in the blade due to the insignificance is equal to:

$$q_w = (p_1 - p_2).S \quad (8)$$

The result of the axial and tangential forces will be equal to:

$$q = \sqrt{{q_a}^2 + {q_w}^2} \quad (9)$$

To determine the bending of the principal axis by using the relationships of the second inertia moment of the surface, the angle between the forces in the direction of v and the resulting forces is:

$$Q = b + st \quad (10)$$

$$b = tan^{-1}\frac{q_a}{q_w} \quad (11)$$

$$st = tan^{-1}\left[\left(\frac{-I_{xy}}{\frac{I_x - I_y}{2}}\right).\frac{1}{2}\right] \quad (12)$$

Consequently, the values of q in the direction of v and u are calculated as follows:

$$q_u = qsin(Q) \quad (13)$$

$$q_v = qcos(Q) \quad (14)$$

The bending moments resulting from the encounter of the fluid along the blade are obtained in these two directions as follows:

$$M_v = \int_x^{lb} q_v(z).(z - x).dz \quad (15)$$

$$M_u = \int_x^{lb} q_u(z).(z - x).dz \quad (16)$$

Using the above equations, the bending stresses based on the impact of the fluid in the blade at the three points shown in Fig. 2 can be calculated as follows:

$$\sigma_{b1} = \frac{M_v \times vv_1}{I_u} + \frac{M_u \times uu_1}{I_v} \quad (17)$$

$$\sigma_{b2} = \frac{M_v \times vv_2}{I_u} - \frac{M_u \times uu_2}{I_v} \quad (18)$$

$$\sigma_{b3} = \frac{-M_v \times vv_3}{I_u} \quad (19)$$

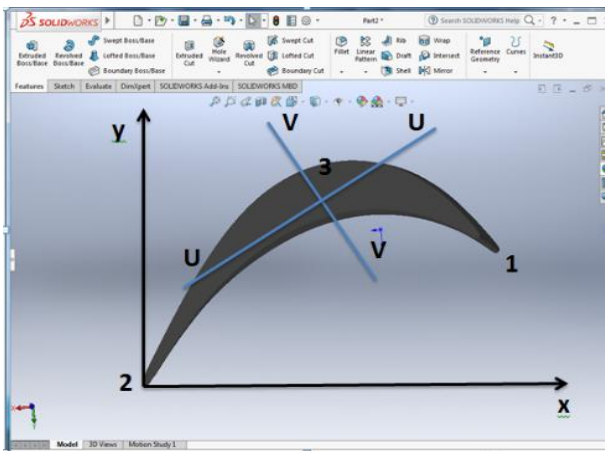


**Fig. 2.** The geometric location of the points in the transverse section of the turbine blade root

### 1.3. Specifications of Row 39th Turbine Blade

The blade rows 39th of k300-240, super-alloy AISI4340 turbines are mounted uniformly to the disc with a 3.75 degree angle. Disk row 39 has 96 moving blades with 48 fixed blades installed in front of them. The weight of each final blade is 9.2 kg, rotating with a rotational speed of 3000 rpm.
The surface area of the blade root is 1624 mm and the surface area of the blade tip is 400/7 mm. The distance from the center of the rotor to the blade root is 760 mm and the blade length is 755 mm.

## Nomenclature

| | |
|---|---|
| $N$ | Number of cycles |
| $F_{cf}$ | Centrifugal force |
| $V_{w1}$ | Inlet steam rotational speed(ms$^{-1}$) |
| $V_{w2}$ | Output steam rotational speed (ms$^{-1}$) |
| $C_f$ | Axial output steam speed (ms$^{-1}$) |
| $p_1$ | Steam Inlet Steam Inlet Pressure (Pa) |
| $p_2$ | Steam Exhaust Steam Pressure (Pa) |
| $S$ | Distance between two moving blades (mm) |
| $S_f$ | Fatigue strength (Mpa) |
| $S'_e$ | Shooter durability (Mpa) |
| $S_e$ | Blade durability (Mpa) |
| $S_{ut}$ | Ultimate strength(Mpa) |
| $k_a$ | Level polishing coefficient |
| $k_b$ | Size correction factor |
| $k_c$ | Load correction factor |
| $k_d$ | Temperature correction coefficient |
| $k_e$ | Reliability factor |
| $\tau$ | Shear Stress (Mpa) |
| $A_r$ | Cross section area of blade root |
| $A_t$ | Cross-sectional area of the tip of the blade |
| $lb$ | Blade length |
| $R_r$ | Radius of center of rotor to root of blade |
| $R_t$ | Rotor center radius to tip of blade |
| $q_a$ | The axial force of the steam enters the blade |
| $q_w$ | Steam tangential force applied to the blade |
| $q$ | The result of the axial and tangential forces of the steam applied to the blade) |
| $q_v$ | Vapor force in the direction v |
| $q_u$ | Steam force in the direction u |
| $M_v$ | Bending moment due to steam collision |
| $M_u$ | Bending moment due to the collision of steam in the direction u |
| $I_u$ | Moment of inertia in the direction u |
| $I_v$ | Moment of inertia in the direction of v |
| $vv1, vv2, vv3$ | The points 1, 2 and 3 of the axis v in Fig. 2 |
| $uu1, uu2, uu3$ | The points 1, 2 and 3 of the u-axis in Fig. 2 |
| $a$ | Level payout coefficient |
| $b$ | Surface payout coefficient view |
| $N)_{LMZ}$ | LMZ Proposed Blade Lifecycle |
| $h$ | LMZ Company's proposed blade operating hours |

## Greek signs

| | |
|---|---|
| $\rho$ | Density (kgm$^{-3)}$ |
| $\omega$ | Rotational speed (rpm) |
| $\sigma_{cf}$ | Centrifugal Tension Force (MPa) |
| $\rho_f$ | Vapor density (kgm -3) |
| $\sigma'_a$ | Von Mises Stress Range |
| $\sigma'_m$ | Moderate von Mises stress |
| $(\sigma_a)_b$ | Flexural Stress Range |
| $(\sigma_a)_{cf}$ | Centrifuge stress range |
| $(\sigma_m)_b$ | Moderate bending stress |
| $(\sigma_m)_{cf}$ | Medium strain centrifuge |
| $\tau_a$ | Shear stress amplitude |
| $\tau_m$ | Medium shear stress |

## 2. Calculating Von Mises Stresses in Row 39th Turbine Blade

It should be noted that the turbine blade is not under a simple bending or axial load, but it is quietly under combination loads. Therefore, the ultimate stresses used to estimate the life span, generated via using the Fumazes equation 1, are as follows [3]:

$$\sigma'_a = \left\{\left[(\sigma_a)_b + (\sigma_a)_{cf}\right]^2 + 3[\tau_a]^2\right\}^{1/2} \quad (20)$$

$$\sigma'_m = \left\{\left[(\sigma_m)_b + (\sigma_m)_{cf}\right]^2 + 3[\tau_m]^2\right\}^{1/2} \quad (21)$$

Any discontinuity or roughness in the piece changes the stress distribution near the discontinuity, consequently, the previous stress equations no longer indicate the actual stress state. Such discontinuity is called the incremental stress 2, and the discontinuity areas are known as stress concentration areas. In fact, the tensile stress coefficient () used to find the result of the greatest vertical stress is built on the fracture or defect of the part. By applying this coefficient to the Eqs. (20) and (21), these equations are modified as below:

$$\sigma'_a = \left\{\left[(k_f)_b(\sigma_a)_b + (k_f)_{cf}(\sigma_a)_{cf}\right]^2 + 3\left[\left(k_f\right)_t(\tau_a)_t\right]^2\right\}^{1/2} \quad (22)$$

$$\sigma'_m = \left\{\left[(k_f)_b(\sigma_m)_b + (k_f)_{cf}(\sigma_m)_{cf}\right]^2 + 3\left[\left(k_f\right)_t(\tau_m)_t\right]^2\right\}^{1/2} \quad (23)$$

### 2.1. Calculation of Stress Concentration Coefficient

The stress concentration factor is as like as below equation [4]:

$$q = \frac{k_f - 1}{k_t - 1} \quad \rightarrow \quad k_f = 1 + q(k_t - 1) \quad (24)$$

In Eq. (24), the value of $k_t$ is extracted on the basis of two parameters D/d and r/d, so according to on the size of the root, we will have:

$$for\ st39 \rightarrow \begin{cases} \frac{D}{d} = \frac{175}{120} = 1.46 \\ \frac{r}{d} = \frac{30}{120} = 0.25 \end{cases} \quad (25)$$

In consequence, it will be equal of $k_t$=1.4 for the tensile cases and $k_t$=1.5 for bending forces cases. Also, since the $S_{ut}$=1172 Mpa and the resizing radius is greater than 4 mm, q=0.9 would be obtained. As a result:

Tensile stress concentration factor

$(k_f)_{cf} = 1 + 0.9(1.6 - 1) = 1.54$

Bending stress concentration factor

$(k_f)_b = 1 + 0.9(1.45 - 1) = 1.4$

The shear stress concentration factor

$(k_f)_t = 1$

### 2.2. Calculation of Stress Components

Since the load applied to the turbine blades is composed of bending and tensile forces, it is necessary to extract the values of $\sigma_a$ and$\sigma_m$. First, the numerical components of the stresses related to the forces applied to the blade must be obtained. Rotational speed is defined as below:

$$\omega = \frac{3000 \times 2\pi}{60} = 100\pi = 314.2\ {}^{rad}/_{s} \quad (26)$$

In addition, for the super alloy blades we have p = 7860 so by replacing the aforementioned blade specifications in Eq. (6) we will have:

$$\mathrm{F_{cf}} = (7850).(314.2)^2[6.93 \times 10^{-4}] = 537.74\ \mathrm{kN} \quad (27)$$

The centrifugal force at the two points of zero degrees and 180 degrees will be minimum and maximum, respectively, because according to Fig.3, the force resulting from the weight of the blade will be once in the centrifugal force direction and once opposite side of the centrifugal force.

$$W = mg = 9.2 \times 9.82 = 90.344\ N \quad (28)$$

$$F_{max} = F_{ct} + W = 537734.8 + 90.344 = 537825.14\ N \quad (29)$$

$$F_{min} = F_{ct} - W = 537734.8 - 90.344 = 537644.06\ N \quad (30)$$

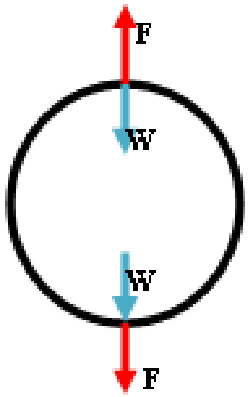


**Fig. 3**. direction of centrifugal force and weight force

Consequently, for the maximum stress based on the centrifugal force at the root section, we will have:

$$\sigma_{cf} = \frac{F_{cf}}{A} = 329.15\ Mpa \quad (31)$$

Using the Eqs. (17), (18) and (19) for the bending stresses, we will have:

$$(\sigma_m)_{be} = \frac{\sigma_{max})_{be} + \sigma_{min})_{be}}{2} = \frac{38 + (-33)}{2} = 2.5\ Mpa \quad (32)$$

$$(\sigma_a)_{be} = \left|\frac{\sigma_{max})_{be} - \sigma_{min})_{be}}{2}\right| = \left|\frac{38 - (-33)}{2}\right| = 35.5\ Mp \quad (33)$$

The shear stress of the blade will also be derived from the following equation [4]:

$$\tau = \frac{3q}{2A} = \frac{3 \times (2.19 \times 10^3)}{2 \times (1.634 \times 10^{-3})} = 2.01\ Mpa \quad (34)$$

2. 3. Numerical Calculations of Modified Von Mises Stresses of Blade

Now, the extracted parameters are applied in Eqs. (22) and (23):

$$\sigma_a' = \{[(1.54) \times (0.1) + (1.4) \times (35.5)]^2 + 3[2.01]^2\}^{1/2} = 49.975\ Mpa \quad (35)$$

$$\sigma_m' = \{[(1.54) \times (329.1) + (1.4) \times (2.5)]^2 + 3[2.01]^2\}^{1/2} = 510.326\ Mpa \quad (36)$$

## 3. Correction of the Blade Durability limit of Row 39th

Using Marin equation, we will have [5]:

$$S_e = k_a k_b k_c k_d k_e k_f S_e' \quad (37)$$

Therefore, considering the properties of the last row blades, we calculate the blade durability:

To calculate the life span of a steering wheel for steels, we have:

$$S_e' = \begin{cases} 0.504 S_{ut}\ Mpa & S_{ut} \le 1460\ Mpa \\ 740\ Mpa & S_{ut} > 1460\ Mpa \end{cases}$$

As we had the blade specification, $S_{ut}$ = 1172 Mpa, we will have:

$$S_e' = 0.504 \times 1172 = 590.688\ Mpa$$

Calculation of surface correction coefficient is calculated according to the following formula:

$$k_a = a S_{ut}^b \quad (38)$$

According to the values of a and b for the surface correction coefficient:

$$k_a = 1.58 \times (1172)^{-0.085} = 0.86$$

For the size correction coefficient for the bending and torsional forces, we have:

$$k_b = \begin{cases} 0.879 d^{-0.107} & 0.11 \le d \le 2\ in \\ 0.91 d^{-0.157} & 2 \le d \le 10\ in \\ 1.24 d^{-0.107} & 2.79 \le d \le 51\ mm \\ 1.5 d^{-0.157} & 51 \le d \le 257\ mm \end{cases}$$

In contrast, for axial force $K_b$ = 1.

The load correction coefficient kc is:

$$k_c = \begin{cases} 1 & for\ bending \\ 0.85 & for\ axial \\ 0.59 & for\ torsion \end{cases}$$

Therefore, in our calculations $k_c$ = 0.85.

For the temperature correction factor, we have:

$$k_d = \frac{S_T}{S_{RT}} \quad For\ \ T = 70°C \ \rightarrow \ k_d = 1$$

Finally, we have the reliability factor:

$$k_e = 1 - 0.08Z_a \quad For\ Reliability\ 99\% \rightarrow k_e = 0.814$$

According to the aforesaid information, we will finally have the corrected durability calculation as follows:

$$S_e = 0.86 \times 1 \times 0.85 \times 1 \times 0.814 \times 590.688 = 351.47\ Mpa$$

## 4. Estimating Life Span of Row 39th Blade

First, we compare the fatigue confidence factor with the yield factor. To calculate the fatigue confidence factor, we have:

$$n_f = \frac{1}{\frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}}} = \frac{1}{\frac{49.975}{351.486} + \frac{510.326}{1172}} = 1.73 \quad (39)$$

Moreover, to calculate the confidence factor, we will have:

$$n_y = \frac{S_y}{\sigma_a' + \sigma_m'} = \frac{1103}{49.975 + 510.326} = 1.97 \quad (40)$$

Since the fatigue confidence factor is lower than the yield factor, we conclude that the blade first becomes fatigued and then yields. [6, 4]

In this step, we calculate the fatigue life of the blades based on different life-cycle criteria. Therefore, to determine the number of cycles for a specific specimen works, we have:

$$S_f = aN^b \quad \leftrightarrow \quad N = \left[\frac{S_f}{a}\right]^{1/b} \quad (41)$$

So we have to compute the values of $S_f$ and the values of the constant of a and b as the following:

$$a = \frac{(fS_{ut})^2}{S_e} \quad (42)$$

$$b = -\frac{1}{3} log\left[\frac{fS_{ut}}{S_e}\right] \quad (43)$$

Considering the blade material $S_{ut}$ = 1172 Mpa and f = 0.79, we will have:

$$a = \frac{(fS_{ut})^2}{S_e} = \frac{(0.79 \times 1172)^2}{351.47} = 2438.97$$

$$b = -\frac{1}{3} log\left[\frac{fS_{ut}}{S_e}\right] = -\frac{1}{3} log\left[\frac{0.79\times1172}{351.47}\right] = -0.14$$

Next, in order to compute fatigue strength, we require determining which criterion we select to use to estimate blade life span.

### 4. 1. Estimation of Blade Life Span Based on Goodman's Criteria

Fatigue strength, according to Goodman's criteria, is as like as below:

$$\frac{S_a}{S_f} + \frac{S_m}{S_{ut}} = 1 \quad \rightarrow \quad S_f = \frac{S_a}{1 - \frac{S_m}{S_{ut}}} = \frac{\sigma_a'}{1 - \frac{\sigma_m'}{S_{ut}}} \quad (44)$$

$$S_f)_{godman} = \frac{\sigma_a}{1 - \frac{\sigma_m}{S_{ut}}} = \frac{49.975}{1 - \frac{510.326}{1172}} = 88.52 \quad (45)$$

Therefore, estimation of the blade life span based on Goodman's criteria, we will have:

$$N)_{goodman} = \left[\frac{S_f}{a}\right]^{1/b} = \left[\frac{88.52}{2438.97}\right]^{1/-0.14} = 1.937 \times 10^{10}\ cycle \quad (46)$$

### 4. 2. Estimation of Blade Life Span Based on Gerber's Criteria

According to Gerber's criteria, fatigue strength would be:

$$\frac{S_a}{S_f} + \left[\frac{S_m}{S_{ut}}\right]^2 = 1 \rightarrow S_f = \frac{S_a}{1 - \left[\frac{S_m}{S_{ut}}\right]^2} = \frac{\sigma_a'}{1 - \left[\frac{\sigma_m'}{S_{ut}}\right]^2} \quad (47)$$

$$S_f)_{gerber} = \frac{\sigma_a'}{1 - \left[\frac{\sigma_m'}{S_{ut}}\right]^2} = \frac{49.975}{1 - \left[\frac{510.326}{1172}\right]^2} = 61.667 \quad (48)$$

Therefore, estimation of the blade life span based on Gerber's criteria would be:

$$N)_{gerber,} = \left[\frac{S_f}{a}\right]^{1/b} \tag{49}$$
$$= \left[\frac{61.667}{2438.97}\right]^{1/-0.14}$$
$$= 2.56 \times 10^{11} \ cycle$$

4. 3. Estimation of Blade Life Span Based on ASME Criteria

In the ASME standard, fatigue strength is equal to:

$$S_f = \left\{\frac{{S_a}^2}{1-\left[\frac{S_m}{S_y}\right]^2}\right\}^{\frac{1}{2}} \tag{50}$$
$$= \left\{\frac{{\sigma'_a}^2}{1-\left[\frac{\sigma'_m}{S_y}\right]^2}\right\}^{\frac{1}{2}}$$

$$S_f)_{ASME} = \left\{\frac{{\sigma'_a}^2}{1-\left[\frac{\sigma'_m}{S_y}\right]^2}\right\}^{\frac{1}{2}} \tag{51}$$
$$= \left\{\frac{(49.975)^2}{1-\left[\frac{510.326}{1103}\right]^2}\right\}^{\frac{1}{2}} = 56.372$$

As a result, estimation of blade life span based on ASME, we will have:

$$N)_{ASME} = \left[\frac{S_f}{a}\right]^{1/b} \tag{52}$$
$$= \left[\frac{56.372}{2438.97}\right]^{1/-0.14}$$
$$= 4.863 \times 10^{11} \quad cycle$$

## 5. Stress Analysis of Turbine Blade Using Finite Element Software

Since the model studied in this research has a complex morphology, the finite element method is used for engineering analysis purposes. The mechanical analysis of this research was done via ABAQUS software which is one of the most powerful finite element software. The three-dimensional model is simulated in the Solidorex software and analyzed in the ABAQUS software.

5. 1. Modeling Row 39th Turbine Blade

The simulation of the blades of Ahwaz Ramin Power Plant was used to simulate the 39-row turbine blades, which is generally a model of Russian's Power Machine k300-240 turbine. The low-pressure steam turbine blade can be divided into two parts in terms of geometrical complexity. The first part is the root of the blade that has easier and measurable geometry. The second part pertained to the turbine blade stem, which is difficult to measure and simulate due to its geometrical complexity and curvature.

The blade and its root have 86 cm in length, and its fork-style root has five teeth, each of them is 2 cm with a gap of 1.8 cm. The blade root with 5 cm height consists of two rows of semi-cylindrical trenches with 1.8 cm diameters to lock the adjacent blades on the respective disc.

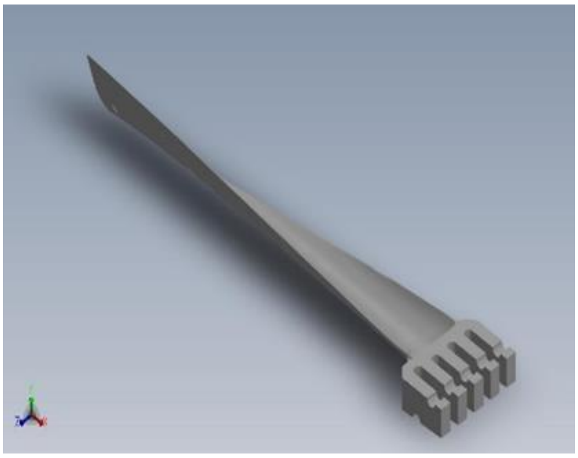

**Fig. 4.** Simulation of Row 39 in Software

5.2. Simulated Blade Analysis in ABAQUS Software

At this stage of the work, the simulated blades should be imported to the ABAQUS from SOLID WORKS software.

As the ABAQUS software is divided into several modules that each module defines one of the aspects of the modeling and analysis process, these modules need to be defined one by one in order to achieve the desired result.

For example, the simulated file importing was done by the geometrical definition module, and then we defined the blade material properties, the definition of the applied forces to the blade, the definition of boundary and blade conditions and blade tension and analysis step by step.

### 5. 3. Comparison of Software Outputs with Different Meshing

It is obvious that smaller dimension meshing will cause results that are more accurate. In this research, we have analyzed different meshing methods for achieving results that are more acceptable. They are presented in the table below. Later, we will discuss them later.

By examining the results presented in the table above, we find that in Figs. 4 and 5 we will have the highest convergence of results. To achieve a more favorable result, meshing number 4 can be obtained for our study. The turbine blade is divided into 167681 elements.

## 6. Estimation of the Life Span of Row 39th of Turbine Blade Based on Software Results

Based on the results from Table 1, we will have the minimum and maximum von Mises stresses related to the row 39th blade as like as below:

$$(\sigma_{max})_{mises} = 171.01\ Mpa$$
$$(\sigma_{min})_{mises} = 3.961 \times 10^{-18}\ Mpa$$

Consequently, the direction of the average stresses and the range of stress are found from the formula below:

$$(\sigma_m)_{mises} = \frac{171.01 \times 10^{6} + 3.961 \times 10^{-18}}{2} = 85.5\ Mpa \qquad (53)$$

$$(\sigma_a)_{mises} = \left|\frac{171.01 \times 10^{6} - 3.961 \times 10^{-18}}{2}\right| = 85.5\ Mpa \qquad (54)$$

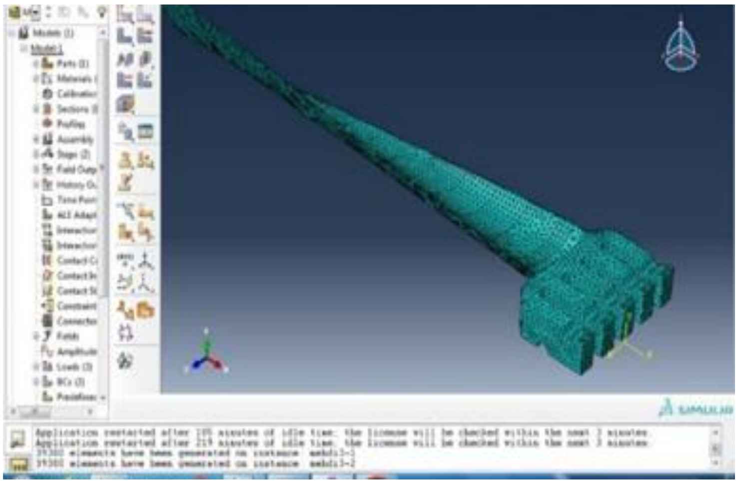

**Fig. 5.** Analysis of Simulated blade in ABAQUS software

**Table 1.** Comparison of blade stress analysis with different meshing in Abacus

| row | Size of meshing | Number of element | Maximum Von Mises stress | Minimum Von Mises stress |
|---|---|---|---|---|
| 1 | 0.7 mm | 38766 | 81.80 Mpa | $8.68\text{x}10^{-18}$ Mpa |
| 2 | 0.6 mm | 55914 | 781.31 Mpa | $10.898\text{x}10^{-18}$ Mpa |
| 3 | 0.5 mm | 89322 | 113.90 Mpa | $4.29\text{x}10^{-18}$ Mpa |
| 4 | 0.4 mm | 167681 | 171.01 Mpa | $3.961\text{x}10^{-18}$ Mpa |
| 5 | 0.3 mm | 359548 | 2186.4 Mpa | $2.519\text{x}10^{-18}$ Mpa |

In addition, by using Eqs. (42) and (43), we have:

a=2438.97 b=-0.14

### 6. 1. Estimation of blade life span based on Goodman's criteria:

Fatigue strength based on Goodman's criteria using Eq. (44) is equal to:

$$S_f)_{Goodman} = \frac{\sigma_a}{1-\frac{\sigma_m}{S_{ut}}} = \frac{85.5}{1-\frac{85.5}{1172}} = 92.2 \quad (55)$$

Therefore, to estimate the life of the blade based on Goodman's criterion we have:

$$N)_{Goodman} = \left[\frac{S_f}{a}\right]^{1/b} = \left[\frac{92.2}{2438.97}\right]^{1/-0.14} = 1.48 \times 10^{10} cycle \quad (56)$$

### 6.2. Estimation of blade life span through Gerber's criteria:

In Gerber's life criteria, fatigue strength according to Eq. (47) would be:

$$S_f)_{\text{Gerber}} = \frac{\sigma'_a}{1-\left[\frac{\sigma'_m}{S_{ut}}\right]^2} = \frac{85.5}{1-\left[\frac{85.5}{1172}\right]^2} = 85.96 \quad (57)$$

Therefore, to estimate the life span of the blade based on Gerber's criterion would be:

$$N)_{\text{Gerber}} = \left[\frac{S_f}{a}\right]^{1/b} = \left[\frac{85.96}{2438.97}\right]^{1/-0.14} = 2.39 \times 10^{10}\ cycle \quad (58)$$

### 6. 3. Estimation of Blade Life Span via ASME Criteria:

In ASME life cycle, the fatigue strength according to the transaction (50) is equal to:

$$S_f)_{ASME} = \left\{\frac{{\sigma'_a}^2}{1-\left[\frac{\sigma'_m}{S_y}\right]^2}\right\}^{\frac{1}{2}} = \left\{\frac{(85.5)^2}{1-\left[\frac{85.5}{1103}\right]^2}\right\}^{\frac{1}{2}} = 85.758 \quad (59)$$

As a result, to estimate blade life span based on ASME we have:

$$N)_{ASME} = \left[\frac{S_f}{a}\right]^{1/b} = \left[\frac{85.758}{2438.97}\right]^{1/-0.14} = 2.427 \times 10^{10} cycle \quad (60)$$

## 7. Comparison between Life Span Results Based on Manufacturer's Standard (LMZ):

As it was previously mentioned, our example is Ramin Ahwaz turbine power plant, which is generally Russian Power Machine plant and the turbine part was produced by LMZ Company, which is a subsidiary of the same turbine company. The working life span of the row 39th blade has been announced h= 100,000 hours by LMZ company [6]. According to the operating conditions of the turbine, which has a rotational speed of 314 rps or 3000 rpm, we have:

$$N)_{LMZ} = h \times 60 \times rpm \quad (61)$$

$$N)_{LMZ} = 100000 \times 60 \times 3000 = 1.8 \times 10^{10} \quad \text{Cycle}$$

In this section, we would like to analyze and compare the results obtained in the previous sections and compare the results generated from analytical and software analysis with each other as well as with the suggestion of the manufacturer.

**Table 2**. Comparison between results of turbine blade life span estimation from analytical solution with the software solution

| row | Life criterion | Life estimation | deflection |
|---|---|---|---|
| 1 | Goodman's criteria | $1.937 \times 10^{10}$ cycle | 0.076 |
| 2 | Goodman's criteria by software | $1.48 \times 10^{10}$ cycle | 0.177 |
| 3 | Gerber's criteria | $2.56 \times 10^{11}$ cycle | 13.22 |
| 4 | Gerber's criteria by software | $2.39 \times 10^{10}$ cycle | 0.327 |
| 5 | ASME's criteria | $4.863 \times 10^{11}$ cycle | 26.01 |
| 6 | ASME's criteria by software | $2.427 \times 10^{10}$ cycle | 0.348 |
| 7 | LMZ | $1.8 \times 10^{10}$ cycle | *** |

## 8. Conclusion

By studying the table above, we can conclude that the estimation of the life span of row 39th turbine blade in Goodman's life span criteria is the closest to the manufacturer's (LMZ) guidelines. In manual analysis calculations, we have the smallest standard deviation with 7% error and with the Goodman Life Criteria estimation, this error ranks second through ABAQUS finite element software calculations with 17% standard deviation.
Overall, considering the proximity of all three outputs from software computations, it should be announced that the finite element technique is the most appropriate method for stress and fatigue analysis in general. However, by comparing the results from each method, we will find that the Goodman, Gerber and ASME life span scales are more cautious in order. In addition, by comparing with the manufacturer's results, the perceived life span estimated for the blades increases. Consequently, in the set of calculations, using the Goodman Life Criterion, not only produces the most desirable result, it also predicts the most cautious output. Considering the importance and sensitivity of the turbine blade operating health, whether in system stability or damaging other equipment, taking into account the high turbine rotation times and the high costs of turbine blade repair and replacement, it is logical to use more cautious criteria in the repair and periodic maintenance of the turbine.

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