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Models for Stress Determination in the Material Selection for Palm Kernel Shell-fired Steam Turbine

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ABSTRACT
Steam turbine is used for electricity generation in most power plants throughout the world. The parameters during operations are very important since the safety and economy of the system depends on its efficiency which is a function of the material selection of the critical parts. Steam turbines are majorly powered with steam from steam generator fueled with coal. There is need to research into other energy source like the renewable energy in fuelling the steam generator as well as considering the material selection for safety and economy of the system. This research utilizes the material consideration for the steam turbine blade and rotor. Models were developed from existing models for stress determination in order to select appropriate material during design stage. With design equations and analysis; the centrifugal force, centrifugal stress, the thermal stress and area thermal strain on the blade as well as the torque and power on the rotor for both empirical data and real life experimental data were analyzed. The results from empirical data for the centrifugal force and stress are 683 N and 0.4563 MPa for a speed of 3000 rpm using the blade dimensions of the developed steam turbine while the torque and power for the rotor are 0.8542kN-m and 758 kW (using 3000 rpm) for 20 mm rotor shaft diameter. Also, for real-life experimental data, using the average speed of 3290 rpm for the last four trials, the torque and power are 2.413 kN-m and 831.85 kW respectively. The blade bending stress was found to be 193.413 MPa for fluctuation and 228.529 MPa for vibration which is lower than the shear stress of the material used. The outcome of the bending stress under vibration S_{abp} (228.59 MPa) is lesser than the ultimate shear stress (375 MPa) of stainless steel 304 grade, therefore the material can be considered for steam turbine blade and rotor. The models can be used for possible stress determination during design stage for steam turbine.

Keywords: Models, Ultimate shear stress, temperature, pressure, steam turbine, blade, rotor, bending stress, fluctuations, vibration.

I. INTRODUCTION

Power generation and energy is the backbone for every nation's survival in this world. Presently, the world needs abundant supply of clean and affordable energy to support economic, social progress and also build a better quality of life particularly in developing countries [1]. Electricity has become an essential facet of life in all spheres of modern society and the industrialization of a nation can only strive better with the availability of electricity [2][3]. Steam Turbine accounted for about 42% percent of U.S. electricity generation in 2022 [4]. According to [5], more than 80% of global electricity production is contributed by thermal power plants (TPP). With such a substantial amount of electrical energy being produced by steam turbine generators, it is in the best interest of society to make these generators as efficient and as sustainable as possible. Steam turbines are used in various types of power generation including nuclear power generation, coal-fired power generation, gas turbine combined-cycle power generation and other power generation systems. Along with the extended use of renewable

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energy, providing electric power generated by steam turbines is about to increase [6]. In the thermal power station, the steam turbine is the prime mover which generates enough torque to produce power from generator. Steam turbine is the mainstay of electricity generation worldwide and obtains its power by the adiabatic expansion of steam flow through the blades and it operates on the basis of the Rankine Cycle [7].

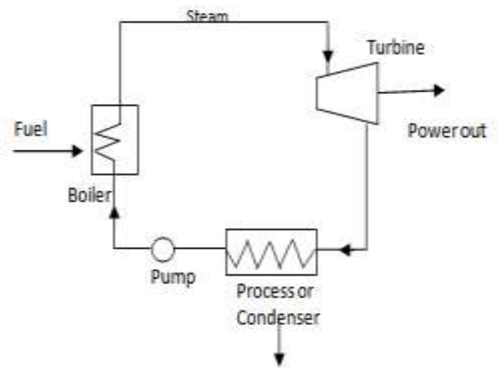

Figure 1. A schematic representation of a steam turbine power system [8][9]

Steam turbine blades are the heart of the turbine which experiences the most intense static and dynamic conditions throughout its life span. It participates in the conversion of kinetic energy to mechanical energy. Therefore, blades' analysis is compulsory to avoid any failure [10]. Steam turbines blades are subjected to high pressure and thermal stresses during load changes as well as start-up and shut down operations [11]. Thermal Stress plays important role during turbine cold start up. When it occurs, the most significant differences of temperature through the rotor cross section can lead to fatigue failure which eventually leads to the failure of the entire plant [12]. An important characteristic of a steam turbine power plant is its ability to maintain reliability and safety of plant against frequent start-ups and load changes [13].

II MATERIALS AND METHODS

In this work, models were modified from existing models to estimate some selected parameters on two critical steam turbine parts; the stresses on the blade (centrifugal stress, bending stress, centrifugal bending stress and the dynamic stress that may result during the operation of the steam turbine) and, torque and power transmitted by the rotor. These models were tested using empirical data from literature as well as data collected from a steam turbine developed and test run using superheated steam from a palm kernel shell-fueled steam generator.

III. MODEL FORMULATION

1. System Model For Turbining Parameters Estimation

A micro steam turbine was locally developed based on the steam properties of the steam generator to be used and the parameters from it during trials were used to test the models developed. Super heated steam from steam generator fueled with palm kernel shell was used to drive the developed turbine. Plate 1 is the picture of the developed steam turbine while figure 2 is the full assembly drawing of developed micro steam turbine. Data collected during the operation of the steam turbine was used to estimate some parameters that determine easy selection of material for design consideration and sustainability of steam turbine in service.

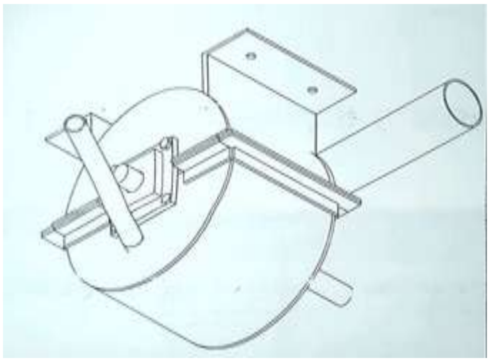

Figure 2 Sketch drawing of developed micro turbine

Plate 1. Picture of the turbine developed

Steam Turbine Blades

Steam turbine blade failure is related directly to some forces at a given corrosion environment as:

$$b_s = kF_b \quad (1)$$

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where, k , b_s and F_b are constant of proportionality, failure rate, and a force on the blade at a given corrosion environment, that is steam (from water) in this case respectively.
Total forces acting on the blade can therefore be modeled as Equation 2

$$F_b = \sum_{i=1}^{n} \sum_{j=1}^{m} F_{cij}, \qquad (2)$$

where, i, is the counter for different forces acting on blade, j, is the counter for the corrosion environment and F_{cij}, is the centrifugal force i at a given environment tj

Evaluation of the Centrifugal force

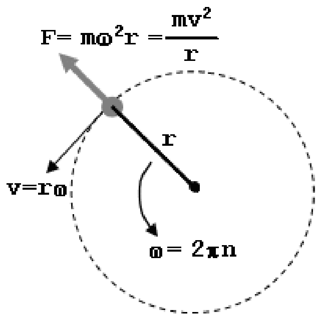

Figure 3. Schematic representation of centrifugal force [14].

The centrifugal force on an element dr at a radius, r

The forces stated in Equation 2 can also be modeled in static and dynamic conditions, depending on the flow rate/speed and weight of the blade. High turbine speed leads to enormous centrifugal forces on turbine blade during rotor rotation, hence high vibration (failure rate) results, due to dynamic forces condition. Constant k is a useful parameter for the conversion of static forces to dynamic ones, and vice versa.
Then, centrifugal force F_{cij} exerted at the blade root as adapted from Figure 3, Equation 2 is now;

$$F_{cij} = \int_{r_r}^{r_t} \gamma a dr\, w^2\, r = \frac{\gamma a w^2}{2}(r_t^2 - r_r^2)_{ij} \qquad (3)$$

where, γ, a, w, r_t and r_r are specific weight of the blade material (kg/m^3), blade cross-sectional area (m^2), angular velocity (rad/sec), tip radius (m) and root radius (m).
By substituting $\frac{A}{\pi}$ for $(r_t^2 - r_r^2)$ in equation (3), we have,

$$F_{cij} = (\frac{\gamma a w^2}{2\pi} A)_{ij} \qquad (4)$$

By substituting converted angular velocity $w\,(rpm)$ to $\frac{2\pi N}{60} rad/sec.$, the relationship becomes;

$$F_{cij} = (\frac{\gamma a}{2\pi}\left(\frac{2\pi N}{60}\right)^2 A)_{ij} \qquad (5)$$

where;
A, is annular area, which is equals to $\pi(r_t^2 - r_r^2)$ as can be proved from equations (2 and 3).
These loading phenomena from steam flow resulted in different stresses on the blade. Among the stresses experienced by the blade are: centrifugal stress, thermal stress and strain, centrifugal bending stress, steam flow stretching/compressive stress and dynamic stress.
The total stress acting on the blade, S_{Tb} at a given environment was formulated as;

$$S_{Tb} = \sum_{i=1}^{n} \sum_{j=1}^{m} S_{bij} \qquad (6)$$

where, i and j are counter for stress type S_{bij}, and the blade environment (that is, steam), respectively. The stated stresses were respectively evaluated as follows:

For centrifugal stress on blade, S_c;

$$S_c = \frac{F_b}{a_r} \qquad (7)$$

where S_c, is the centrifugal, tensile or compressive stress, F_b is the centrifugal force and a_r is the root area of the blade.

Thermal stress S_{th} due to variation in steam temperature on blade root was evaluated from;

$$S_{th} = E\alpha_a \partial t \qquad (8)$$

where, E is the blade material young modulus, α_a, is the coefficient of area thermal expansion, and ∂t, is the change in blade material temperature.
The coefficient of area thermal expansion was obtained from;

$$\alpha_a = \frac{a_r - a_{r0}}{a_{r0} t} \qquad (9)$$

Hence,

$$a_r - a_{r0} = \alpha_a a_{r0} t \qquad (10)$$

From which, $a_r = \alpha_a a_{r0} t + a_{r0} = a_{r0}(\alpha_a t + 1)$
Hence, change in area due to increase in temperature was expressed from equation 10 as

$$\partial a_r = \alpha_a a_{r0} \partial t \qquad (11)$$

The corresponding thermal strain in the blade can be computed from

$$\varepsilon_{ath} = \alpha_{a_r} \partial t \qquad (12)$$

where; $\varepsilon_{ath}, \alpha_{a_r}$ and ∂t are strain in a specific direction, area thermal coefficient and temperature

International Journal of Advances in Engineering and Management (IJAEM)
Volume 7, Issue 01 Jan. 2025, pp: 604-611 www.ijaem.net ISSN: 2395-5252

change in the blade material, respectively. For a known material property, area expansion was determined on the basis of Hooke's law as;

$S_{th} = \varepsilon_{ath} a_r$ (13)

Centrifugal bending stress was considered only if the center of gravity of the blade section and that of the blade root are not in a straight line at different height; otherwise it was not necessary to considered it since direct loading took a lion share as shown in Equations 2 and 3. On necessity, centrifugal bending stress was expressed as;

$\sigma_b = My/I$ (14)

And otherwise, as
$S_b = F_b/a_r$ (15)

Or both as expressed in Equation (16);
$S_{\sigma b} = My/I + F_b/a_r$ (16)

where,
$M, y, and\ I$ are moment of blade material about centre of gravity, load distance from centre of gravity and moment of inertia, respectively; F_b and a_r are as defined before.

The first and second terms of Equation (16), respectively, are representatives of the stresses due to bending moment and direct (tensile or compressive) loading of blade root.

Alternatively, Centrifugal bending stress was also estimated based on steam flow loading which caused blade to bend longitudinally due to stretching and compressive stress. On this basis, σ_b, was empirically estimated as

$\sigma_b \geq 0.1\, S_b$ (17)

The constant numerical value y, where y varies between $0.1\ to\ 1.0$ based on the degree of the steam fluctuation, that is, low pressure level LP, Intermediate Pressure level IP and high pressure steam HP, at which the pressure difference between the blades is the greatest.
On this basis (Equation 16 and 17) can be put together, then,

$S_{\sigma b} = S_b + yS_b$ (18a)

However, steam flow rate fluctuation needed to be maintained at minimum possible level, and hence,

$S_{\sigma b} = S_b + 0.1S_b$ (18b)

Also, turbine operated at high speed/flow rate is susceptible to blade vibration due to the existence of non-uniform flow of steam. Two cases were considered: first, low vibration, in which the amplitude of the stress fluctuation is very small, but not beyond the level that the blade can bear; and, second, high vibration, in which attendant high amplitude of the stress on the blade can lead to sudden failure if exceeded what material can bear. On this basis, Equation 18 was modified to reflect dynamic stress on the blade at low (0.1) and high allowable vibration level (with constant y, greater than 0.1, say 0.2, for example) as
$S_{\sigma bp} = S_b + (0.1)S_b + (0.2)S_b$ (19a)
That is,
$S_{\sigma bp} = 1.3S_b$ (19b)

The two cases in Equation (19) can happen simultaneous due to continuous instability in blade vibration caused by continuous irregularity in the steam flow process.

The suitability of the chosen blade materials for the new and existing blade designs were determined based on the outcome of Equation (19b), which should not exceed the ultimate (maximum allowable) stress S_u that the chosen material can bear. That is,
$S_{\sigma bp} < S_u$ (19c)

Steam Turbine Rotor

Attached to the turbine rotor are the blades. The rotor converted the thermal energy from the steam to the mechanical energy of rotation. The torque T transmitted during rotation was computed from

$T = \frac{\pi\tau D^3}{16}$ (20)

where, τ and D are the shear stress and the rotor diameter, respectively; π, is as defined before. Hence, the power, P transmitted by the rotor was evaluated from,

$P = \frac{2\pi\tau NT}{60}$ (21)

where, N is the rotor speed in revolution per minute (rpm).

The suitability of the chosen rotor material was determined on the basis of shear stress τ at the given torque and power intended to transmit. The shear stress of chosen rotor material τ_u should be able to cope with the shear stress τ on the turbine rotor.

That is,
$\tau < \tau_u$ (22)

2. Model Analysis

The formulated models were analyzed in two ways: first, by using empirical turbine design data obtained from the literature, and second, by using real-life data obtained from an experimented 5 kW Steam turbine developed.

Analysis using Empirical Turbine Design Data

This was done using the parameters of the properties from stainless steel 403 grade as possible blade material

Table 2: Material Property of Possible Blade Material

Steel grade	Ultimate tensile strength MPa	Elastic Modulus GPa	Linear Thermal Expansion coefficient (x10^{-6})/m^{o}C	Specific weight of the blade material, kg/m^3
Stainless steel 403 grade	485	190 – 200	9.9	7800
Stainless steel 410 grade	500	200	5.5	7740
Stainless steel 422 grade	965	190 - 200	11.2	7800
Ti – 6Al -4V (Titianum)	≥ 895	114	8.5	4410

(**Source:** Azom. 2021)

Using the parameters of stainless steel 403 grade and with the area parameter of the blade developed, the centrifugal force and centrifugal stress were calculated using 3000 rpm.

$$F_b = \frac{7800 \times 193.873 \times 10^{-6} \times 28788.103 \times 10^{-6}}{2\pi}\left(\frac{2 \times \pi \times 3000}{60}\right)^2 = 683\ N$$

$$S_c = \frac{F_b}{a_r} = \frac{683 \times 10^6}{1498.69} = 456{,}308.435\ Pa = 0.4563\ MPa$$

The torque and power were also calculated using Equation 20 and 21 respectively with a rotor shaft diameter of 20 mm and a speed of 3000 rpm for ASTM A 470 grade D class 8 (Ultimate Tensile Strength, 725 MPa) as the possible rotor material, the shear stress was found out to be 543.75 MPa using the relation;

$Shear\ stress = 0.75\ x\ Ultimate\ tensile\ stress$
(for stainless steel) (23)

$$Torque = \frac{\pi \times 543.73 \times 20^3}{16}$$
$$= 0.8542 \text{ kN-m}$$

$$Power = \frac{2\pi NT}{60} = \frac{2 \times \pi \times 3000 \times 2.413}{60} = 758\ kW$$

Analysis using Real-life Experimental Data

The developed steam turbine was tested with super heated steam from steam generator of 5 kW fuelled with palm kernel shell and the results from five trials were tabulated in table 3.

Table 3: Experimental Data Result

Trials	Inlet Steam Properties		Outlet Steam Properties		Temperature Difference ΔT (^{o}C)	Pressure Difference (bar)	Speed (N) rpm
	Temperature, T_1 (^{o}C)	Pressure, P_1 (bar)	Temperature, T_2 (^{o}C)	Pressure, P_2 (bar)			
First	250	5	116.11	4	133.89	1	8675

International Journal of Advances in Engineering and Management (IJAEM)
Volume 7, Issue 01 Jan. 2025, pp: 604-611 www.ijaem.net ISSN: 2395-5252

Trial							
Second Trial	168.33	4.0	109.44	3	58.89	1	4850
Third Trial	137.77	2.0	120.55	0.75	17.21	1.25	3292
Fourth Trial	130.1	1.0	108.67	0.45	21.43	0.55	2609
Fifth Trial	123.33	0.5	88.89	0.23	34.44	0.27	2406

Blade Material – AISI 304 Stainless steel, Specific weight of blade material 7930 kg/m^3
Blade cross sectional area = 193.873 mm^2 = 193.873 × 10^{-6} m^2
Blade Annular Area = 28,788.103 mm^2 = 22788.103 × 10^6 m^2 ,
Blade Root Area = 1498.69 mm^2 = 1498.69 × 10^{-6} m^2

γ = specific weight of blade material, kg/m^3 = 7980 kg/m^3
For the Steam Turbine blade, the data obtained for the first experimental trial on the speed was used in Equations 5 and 7 to calculate the centrifugal force and the centrifugal stress respectively.

$$F_b = \frac{7980 \times 193.873 \times 10^{-6} \times 28788.103 \times 10^{-6}}{2\pi}\left(\frac{2 \times \pi \times 8675}{60}\right)^2 = 5{,}849.89\ N$$

$$Centrifugal\ stress = \frac{5849.89}{1498.69 \times 10^{-6}} = 3{,}903{,}335.57\ Pa$$
$$= 3.903335\ MPa$$

Table 4: Speed (N) and corresponding Centrifugal Force and Stress

Trials	**Speed (N) rpm**	**Deviation of N**	$\boldsymbol{F_c}$ **(N)**	$\boldsymbol{\sigma_c = MPa}$
First Trial	8675		5,849.89	3903.33
Second Trial	4850	3820	1,828.49	1220.059
Third Trial	3292	1558	842.42	562.104
Fourth Trial	2609	683	529.123	353.057
Fifth Trial	2406	203	449.98	300.253

Average for the five trial 1897.19 1267.76
Average without the first trial 912.50 608.87
The thermal stress and area thermal strain was calculated using Equations 8 and 9
E for the blade material = 191 GPa, α_a = 17.3 x 10^{-6}/^{o}C

For the first trial, $S_{th} = 191 \times 10^3 \times 17.3 \times 10^{-10} \times 133.89 = 442\ MPa$
The corresponding $\alpha_a = 17.3 \times 10^{-6} \times 133.89 = 0.00232$
The result for the other four trials are tabulated in table 5

Table 5: Thermal Stresses and Area thermal Strain

Trials	**Temperature Difference (^{o}C)**	**Thermal Stress (MPa)**	**Area Thermal Strain**
First trial	133.89	442	0.00232
Second trial	58.89	194.59	0.00102
Third trial	17.16	56.70	0.000297
Fourth trial	21.45	70.877	0.000371
Fifth trial	34.41	113.7	0.000595
Average		175.5734	0.0009206

The average strain was used in calculating the blade bending stress

International Journal of Advances in Engineering and Management (IJAEM)
Volume 7, Issue 01 Jan. 2025, pp: 604-611 www.ijaem.net ISSN: 2395-5252

$$\alpha_a = \frac{0.00232 + 0.00102 + 0.000297 + 0.000371 + 0.000595}{5}$$
$$Average\ strain = 0.0009206$$

The blade bending stress was calculated using;

$$Blade\ bending\ stress = strain \times modulus\ of\ \text{Elasticity of the material}$$

The blade bending stress $= 0.0009206 \times 191 \times 10^3 = 175.83$ MPa

For fluctuation, $S_{ab} = 193.413$ MPa
For Vibration $S_{\sigma bp} = 1.3\ S_b = 228.579$ MPa

The first trial speed was instantaneous and too high compared to the other four trials hence not considered. The average speed for the last four experimental data was used in the calculation

$$\text{i.e.} \frac{4850 + 3292 + 2609 + 2404}{4} = 3290 \text{ rpm}$$

Equation 23 was used to determine the shear stress of the material.
0.75×500 MPa $= 375$ MPa

Torque $= \frac{\pi \times 375 \times 32^3}{16} = 2{,}412{,}743.012$ Nmm $= 2.413$ kN − m
For the speed of 3290 rpm

$$\text{Power becomes } \frac{2 \times \pi \times 3290 \times 2.413}{60} = 831.85 \text{ kW}$$

IV. RESULT AND DISCUSSION

For the empirical data, some possible blade materials were selected from which the stainless-steel grade 403 material property parameter was used in calculating the possible centrifugal force and centrifugal stress that can act on the blade material using a speed of 3000 rpm. The centrifugal force and stress calculated were 683 N and 0.4563 MPa respectively. The torque and power of the possible steam turbine rotor materials (ASTM A 470 grade D class) were computed and the results obtained were 0.8542 kN-m and 35.7853 kW respectively for a rotor shaft diameter of 20 mm.

For the real-life experimental data, the outcome of the first experimental trial was used in calculating the centrifugal force and stress and, the result were found to be 5,849.89 N and 3.903335 MPa respectively. The result of the speed obtain for the first real-life experiment was instantaneous and as such very high compare to the other four experimental trials results hence, the average of the last four experimental data for the speed was used in other computations. For the real-life data, the torque was computed to be 2.413 kN-m while the power was 831.85 kW. Other parameters computed were the average thermal stress and the average area thermal strain, the bending stress for fluctuation in temperature and vibration. The results were found to be 175.57 MPa for the blade bending stress, 0.0009206 for average area thermal strain, 193.415 MPa for bending stress during fluctuation in temperature and, 228.59 MPa for bending stress due to vibration.

V. CONCLUSION

From the discussion in the previous sections, the following conclusions are drawn:

(i) Easy determination of the possible stresses that can act on a steam turbine blade is very essential. This will make the designer make good decision on the right material choice depending on the expected temperature, pressure and flow rate from the steam generator to be used.

(ii) The models can also be used to determine possible stresses both for fluctuation and vibration that can act on possible steam turbine blade material depending on the temperature and pressure of super heated steam expected from the steam generator that will supply super heated steam to the turbine.

(iii) The outcome of the computed bending stress under vibration S_{abp} (228.59 MPa) is lesser than the ultimate shear stress (375 MPa) of stainless steel 304 grade for the developed steam turbine therefore, the material can be considered for steam turbine blade considering the steam properties of the steam generator. The stresses must not be more than the shear stress of the material used for the steam turbine blade and rotor so as to avoid failure of the material.

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