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Design of a Constant Stress Steam Turbine Rotor Blade

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Abstract

This paper presents a design procedure for a constant stress low pressure steam turbine blade. Such blades suffer from different types of loadings leading to different types of stresses, like tensile and bending stresses due to centrifugal force and steam flow loading. The centrifugal force is one of the problems that face the designers of blades especially the long ones. One way to tackle this is by using twisted blades and/or using variable cross-section area blades.

Six types of blades were studied in this paper. One for constant cross sectional area blade as a reference. The other five are for reducing cross sectional area blades of varying curvature. Reducing the cross sectional area reduces the mass of blade and hence reduces the centrifugal stresses. The performance of the cases was then checked and found to be within satisfactory ranges.

The results obtained are then utilized in the design of the constant stress blade. Which was checked with references for three locations (stages) and was found satisfactory?

الخلاصــــة

يعرض هذا البحث طريقة تصميم ريش توربين بخاري ذي الضغط المنخفض. هذه الريش تتعرض إلى أنواع مختلفة من الأحمال و التي تؤدي إلى أنواع مختلفة من الاجهادات، مثل إجهاد الشد و الانحناء التي تنتج عن أحمال قوى الطرد المركزي و جريان البخار على التوالي. إن قوة الطرد المركزي تمثل احد المشاكل التي تواجه مصممو الريش و خاصة الريش الطويلة. احد الطرق للتعامل مع هكذا مشكلة هو باستخدام ريش ذات مساحة مقطع متغيره أو ريش ملوية ذات مساحة مقطع ثابتة أو متغيرة.

تم دراسة ست حالات في هذا البحث أحداها ريشة ذات مساحة مقطع ثابتة تستخدم للمقارنة والخمس الأخرى ذات مساحة مقطع متغيرة و ذي انحناءات مختلفة. أن نقصان مساحة المقطع يؤدي إلى نقصان كتلة الريشة مما يؤدي إلى تقليل الإجهادات المركزية.

وقد تم التأكد من أداء هذه الحالات و وجد ضمن المدى المسموح. تم بعدها استخدام النتائج لتصميم ريشة ثابتة الإجهاد. و التي تم مقارنتها مع المصادر و لثلاث مواقع (مراحل) من التوربين و وجد الأداء مقنعا.

1. Introduction

Steam turbines are the core elements of the steam turbine power plants (Heat power plants). They form the harvesting component of the plant in which the energy of the hot, fast, and high pressure stream of steam is first transformed into kinetic energy by expansion through nozzles, and then the kinetic energy of the resulting jet is converted into force doing work on rings of balding mounted on a rotating disc.

Steam turbines have two important elements [1,2]:

1. The stator (nozzle), in which the energy of steam is converted into kinetic energy, hence into flow of high momentum.
2. The rotor blades, in which the high momentum stream of steam is forced to change direction, and so reducing its momentum. The difference in momentum is a force applied to the rotor blades forcing them to rotate. These blades are attached to the rotating shaft (wheel), hence transferring this force to torque and then to power acting on the shaft. **Figure (1)** shows the cross-section of a double entry steam turbine.

The rotor blades are the critical elements of the turbine. They carry the entire flow loading. To design such a blade, the designer needs to calculate the flow properties at first, estimate the flow forces, and evaluate the stresses on the blades.

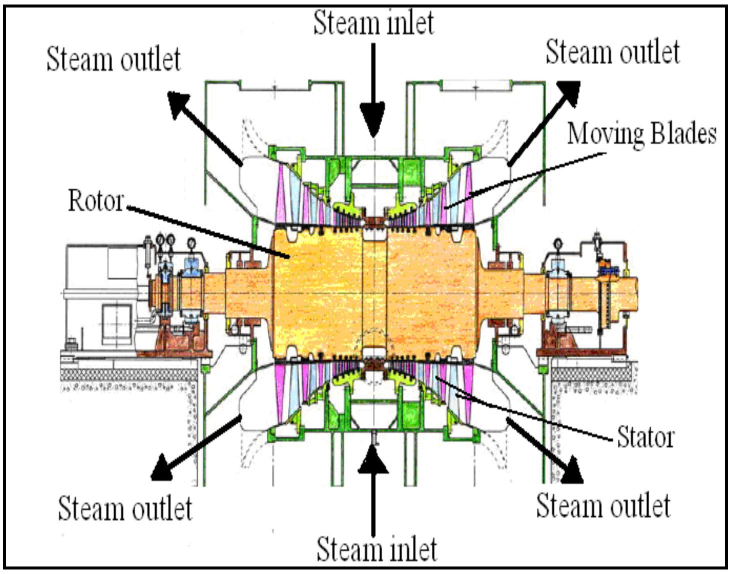

Figure (1) Cross section of a double entry steam turbine [8]

2. Blade Loading

The loading on a rotating turbine blade is composed of:

1. Centrifugal forces due to rotation.
2. Bending forces due to the fluid pressure and change of momentum.
3. Bending forces due to centrifugal action if the centroids of all sections do not lie along one radial line.

The steady stress at any section of a parallel sided blade is a combination of direct tension due to centrifugal force and bending due to steam force. Both of which are acting on that portion of blade between the section under consideration and the tip.

The direct tensile stress is maximum at the blade root and is decreasing towards the tip [7]. The centrifugal stress depends on the mass of material in the blade, blade length, rotational speed, and the cross sectional area of the blade.

However, the impulse turbine blades are subjected to bending stresses as a result of the centrifugal forces, the tangential force exerted by the fluid on the blade and if the centroids of all blade sections do not lie along the same radial line, an additional load is created.

While the reaction blades have an additional bending stresses resulting from the large axial thrust as a result of the pressure drop occurring in the blades passages.

All turbine blades are subjected to vibration loads and hence, to stresses due to vibration. It is worth mentioning that the bending stresses are also maximum at the blade roots. The combined tensile and bending stress is, hence maximum at the root and diminishes with radius.

For tapered blades the direct tensile stress diminishes less rapidly outwards the tips, while the bending stress can be made to increase at greater radii. It is therefore possible to design the blade as a cantilever, with constant tensile stress (centrifugal stress) and bending stress and by doing so the blade material is much more effectively utilized.

The centrifugal stresses are more significant than other stresses as they have the greatest effect on total stress. Centrifugal stresses are a function of the mass of the material in the blade, blade length, cross-sectional area of blade (profile area) and rotational speeds. To calculated the centrifugal stresses on a moving blade, assuming a single blade fixed to the disc as shown in **Fig.(2)**.

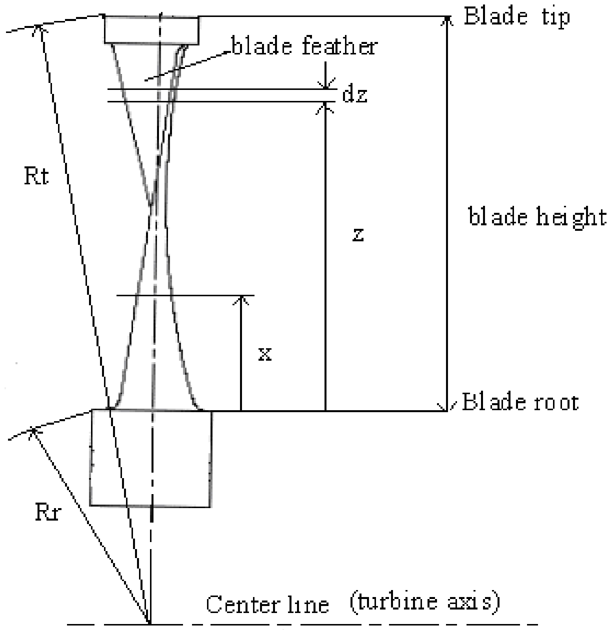

Figure (2) Moving blade nomenclature

Let Rr be the radius of blade at root section, and Rt be the radius at the tip. The area of the blade profile in section (x) may be denoted as A(x). Let x be an axis chosen along the blade so as to pass through the axis of rotation of the rotor. It is assumed that the centers of gravity of all blade sections lie on the axis (x).

The centrifugal stresses at the blade section are the centrifugal force in this section divided by the area of the blade section, and can be found by formula:

$$\sigma_{cf}(x) = \frac{F_{cf}(x)}{A(x)} \quad \text{.. (1)}$$

If an infinitesimal element (dz) be separated in section (Z) shown in **Fig.(2)**, The force developed by this element during disc rotation is therefore:

$$dF_{cf} = dm.\ \omega^2\ (Rr+z) \quad \text{.. (2)}$$

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

Substituting Eq.(3) into Eq.(2) gives the centrifugal force in this element [9]:

$$dF_{cf} = \rho.\ \omega^2.\ A(z).\ (Rr+z)\ .dz \quad \text{... (4)}$$

By integrating Eq.(4), the centrifugal force becomes:

$$F_{cf}(x) = \int_x^{Lb} \rho.\ \omega^2\ A(z)\ (Rr+z).dz \quad \text{.. (5)}$$

where:

Lb=Rt-Rr

The stresses from the centrifugal forces in a blade arbitrary profile can be determined by the relationship:

$$\sigma_{cf}(x) = \frac{\omega^2}{A(x)}\left[\rho.\int_x^{Lb} A(z).(Rr+z).dz\right] \quad \text{... (6)}$$

Two types of blades were taken under consideration:

I) Constant Cross-Section Area Blade

For a blade of constant cross-sectional area, the centrifugal force given by Eq.(5) then becomes:

$$F_{cf}(x) = \rho.\ \omega^2\ A(z) \int_x^{Lb} (Rr+z).dz \quad \text{... (7)}$$

Integrating Eq.(7) gives:

$$\mathbf{F_{cf}(x)=\rho\omega^2\,[Rr(Lb - x) + \frac{1}{2}(Lb^2 - x^2)]} \quad \text{.. (8)}$$

It is clear that at the tip of the blade, that is when (x = Lb), the force F_{cf} (Lb)=0. Whereas the maximum centrifugal force is at the blade root section when (x = 0).

To estimate the centrifugal stresses, Eq.(5), and Eq.(1) are used to give:

$$\mathbf{\sigma_{cf}(x) = \rho\,.\,\omega^2.\int_x^{Lb}(Rr + z)dz} \quad \text{.. (9)}$$

where:

A(Z) =A (X) = constant

It is clear that at the tip of the blade that is when x = Lb, the stress σ_{cf} (Lb) = 0. Whereas the maximum centrifugal force is at the blade root (x = 0). The maximum centrifugal stresses for a constant cross-sectional area blade are.

$$\mathbf{\sigma_{cf}(0)= \rho\,.\,\omega^2.Lb.\,(Rr+\frac{Lb}{2})} \quad \text{... (10)}$$

where:

$$Rm = Rr + \frac{Lb}{2}$$

$$\mathbf{\sigma_{cf}(0)= \rho\,.\,\omega^2.Lb.Rm} \quad \text{... (11)}$$

II) Variable Cross-Sectional Area Blade

If the blade is of variable cross-sectional area, and to find the centrifugal force, it is necessary to formulate the change of the cross-sectional area as a function of blade height. Using the relation [9].

$$\mathbf{A(z) =Ar- (Ar - At)\,(\frac{z}{Lb})^{\Re}} \quad \text{... (12)}$$

where:

$$\Re = \frac{\ln(\frac{Ar - At}{Ar - Am})}{\ln 2} \quad \text{.. (13)}$$

The centrifugal force on the blade is determined by using Eq.(12) and Eq.(5) to give:

$$\mathbf{F_{cf}\ (x) = \rho\ \omega^2 \int_{x}^{Lb} [\,Ar- (Ar - At\,)\,(\frac{z}{Lb})^{\Re}\,]\,(Rr + z)\,dz} \ \ldots\ldots\ldots\ldots\ldots\ldots.. (14)$$

and,

$$\mathbf{F_{cf}\ (x)=\rho.\omega^2\ [Ar(Rr.z+\frac{z^2}{2}) - \frac{(Ar-At)}{Lb^{\Re}}\ (\frac{Rr.z^{\Re+1}}{\Re+1}+\frac{z^{\Re+2}}{\Re+2})]\,\Big|_{x}^{Lb}} \ \ldots\ldots. (15)$$

When x = Lb, at the tip of the blade, the centrifugal force is zero. While at the root section of the blade, where x = 0 the centrifugal force is maximum and Eq.(15) becomes:

$$\mathbf{F_{cf}\ (0) = \rho.\omega^2\ [Ar.Lb\ (Rr + \frac{Lb}{2}) - (Ar\text{-}At)\ (\frac{Rr.Lb}{\Re+1}+\frac{Lb^2}{\Re+2})]} \ \ldots\ldots\ldots\ldots. (16)$$

The Centrifugal stresses are determined by using Eq.(15) to determine the force divided by the area given by Eq.(12), and the maximum centrifugal stress at blade root becomes:

$$\mathbf{\sigma_{cf}\ (0) = \rho.\omega^2\ [Lb\ (Rr + \frac{Lb}{2}) - (1\text{-}\frac{At}{Ar})\ (\frac{Rr.Lb}{\Re+1}+\frac{Lb^2}{\Re+2})]} \ \ldots\ldots\ldots\ldots\ldots\ldots (17)$$

If the cross-sectional area varies linearly from root to tip then ($\Re$ =1) and Eq.(12) becomes:

$$\mathbf{A(z) = Ar- (Ar - At\,)\,(\frac{z}{Lb})} \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (18)$$

The centrifugal force as given by Eq.(5) and Eq.(18) becomes:

$$\mathbf{F\ cf\ (x) = \rho.\omega^2.\int_{x}^{Lb} [\,Ar- (Ar - At)\,(\frac{z}{Lb})].(Rr+z).dz} \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots... (19)$$

Integrating Eq.(19) gives:

$$\mathbf{F\ cf\ (x)=\rho.\omega^2\ [Ar(Rr.z+\frac{z^2}{2}) - \frac{(Ar-At)}{Lb}.(\frac{Rr.z^2}{2}+\frac{z^3}{3})]\,\Big|_{x}^{Lb}} \ \ldots\ldots\ldots\ldots\ldots (20)$$

When (x=0) at the blade root, the centrifugal force is maximum and Eq.(20) becomes:

$$\mathbf{F\ cf\ (0)=\rho.\omega^2\ [Ar\ (Rr.\ Lb+\frac{Lb^2}{2}) - (Ar\text{-}At)\ .\ (\frac{Rr.Lb}{2}+\frac{Lb^2}{3})]} \ \ldots\ldots\ldots\ldots (21)$$

And the maximum centrifugal stress is:

$$\mathbf{\sigma\ cf\ (0)= \rho.\omega^2\ [(Rr.Lb+\frac{Lb^2}{2}) - (1\text{-}\frac{At}{Ar})\ .\ (\frac{Rr.Lb}{2}+\frac{Lb^2}{3})]} \ \ldots\ldots\ldots\ldots\ldots.. (22)$$

3. Bending Stresses

The moving blade suffers from the bending force in addition to the centrifugal force. The bending force exerted by the working fluid on the moving blade is essentially a distributed load, which in the general case varies along the blade height. The impulse blade is subjected to bending from the tangential force exerted by the fluid [3]. Reaction blades have an additional bending force due to the large axial thrust because of the pressure drop which occurs in the blade passages. The blade without banding or lacing wire can be regarded as a cantilever beam of a variable cross-section area which is stressed by a distributed load (q(x)) as shown in **Fig.(3)** The components of aerodynamic force (q), is the axial force (qa) and the tangential force (qw) shown in **Fig.(3)** this force produces bending load on the blade.

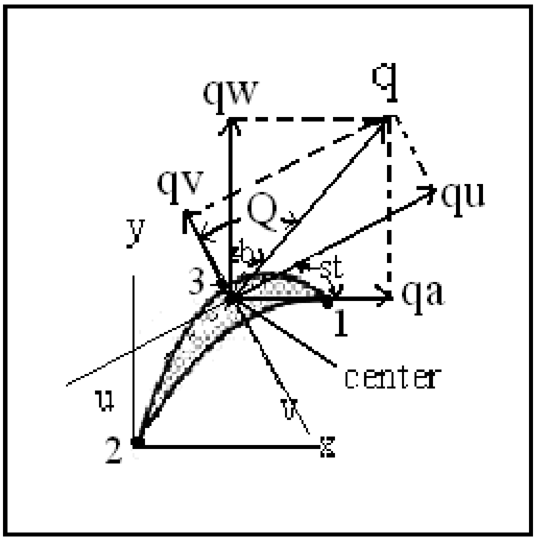

Figure (3) Cross-section area of moving blade [6]

The moment of this force and the stress due to it are both maximum at the blade root. The bending stress diminishes with increase of the radius. If the blade is tapered, the bending stress can be made to increase at greater radii. Consider the blade as cantilever beam fixed at the root section. The bending moment effect is on the level of the main axes of inertia (u,v) shown **Fig.(3)**. The axial force [9]:

$$\mathbf{qa = \rho f.cf3\ (Vw_2 - Vw_3)} \quad \ldots\ldots \mathbf{(23)}$$

The tangential force [9]:

$$\mathbf{qw = \rho_f\ .cf3.(cf2-cf3) + (p2-p3).s} \quad \ldots\ldots \mathbf{(24)}$$

Or**,** when the axial velocity is constant cf2 = cf3, then:

The tangential force:

$$qw = (p2-p3).s \quad \text{……} (25)$$

The resultant force is:

$$q = \sqrt{qw^2 + qa^2} \quad \text{……} (26)$$

where, the component force on the main axes (U,V):

The force for the axis (U):

$$qu = q.\sin(Q) \quad \text{……} (27)$$

The force for the axis (V):

$$qv = q.\cos(Q) \quad \text{……} (28)$$

But Q is the angle between the force on the axis (v) and the result force shown **Fig.(3)**, then:

$$Q = b + st \quad \text{……} (29)$$

$$b = \tan^{-1}\left(\frac{qa}{qw}\right) \quad \text{……} (30)$$

$$st = \left[\tan^{-1}\left[\frac{-Ixy}{\frac{Ix-Iy}{2}}\right]/2\right] \quad \text{……} (31)$$

The bending moment about axes (V, U) at any section of the blade height is:

$$Mv = \int_{x}^{Lb} qv(z).(z-x).dz \quad \text{……} (32)$$

$$Mu = \int_{x}^{Lb} qu(z).(z-x).dz \quad \text{……} (33)$$

The bending stress will be calculated for the following points in which the stress is maximum [3, 4] as shown in **Fig.(3)**.

For the entrance or leading edge of the blade, the stress is:

$$\sigma_{b1} = \frac{Mv * vv1}{Iu} + \frac{Mu * uu1}{Iv} \quad \text{.......................................} \quad (34)$$

While for the exit or trailing edge:

$$\sigma_{b2} = \frac{Mv * vv2}{Iu} - \frac{Mu * uu2}{Iv} \quad \text{.......................................} \quad (35)$$

And for a point located at the intersection of the v axis and the back of the blade which of course is subjected to compression stresses is:

$$\sigma_{b3} = -\frac{Mv * vv3}{Iu} \quad \text{.......................................} \quad (36)$$

where, the negative sign is for compression.

4. Total Stresses

The total stress at a given point on a turbine blade may be found by adding the centrifugal stress at that point to the bending stress.

The total stresses in leading edge point (1) of **Fig.(3)** are:

$$\sigma_{t1} = \sigma_{cf} + \sigma_{b1} \quad \text{.......................................} \quad (37)$$

The total stresses in trailing edge point (2) are:

$$\sigma_{t2} = \sigma_{cf} + \sigma_{b2} \quad \text{.......................................} \quad (38)$$

The total stresses in point (3) are:

$$\sigma_{t3} = \sigma_{cf} + \sigma_{b3} \quad \text{.......................................} \quad (39)$$

5. Blade Terminology

It is necessary to define the parameters used in describing blade shapes and configurations of blade. Blade profiles are usually of airfoil shape for optimum performance, although cost is more important than the ultimate in efficiency, simple geometrical shapes composed of circular areas and straight line are of ten used.

The spacing or pitch of the blade is the distance between corresponding points of adjacent blade and is expressed either by the pitch-chord ratio or alternatively the solidity. When the blades are evenly spaced around a rotor, the pitch is the circumference at any radius divided by the number of blades.

The Lift Coefficient (CL) is found by using velocity diagram and the solidity [3].

$$CL = \frac{2.\Delta v_w .s}{v_r .c} \qquad \text{.......... (40)}$$

or,

$$CL = \frac{2.\Delta v_w}{c_r .SO} \qquad \text{.......... (41)}$$

$$\text{Solidity (SO)} = \frac{c}{s} \qquad \text{.......... (42)}$$

$$\text{Spacing(c)} = \frac{2.\pi.R}{N_b} \qquad \text{.......... (43)}$$

The solidity factor (SO) depends on the chord and pitch of the blades. The pitch in turn depends on the number of blades used.

6. The Design of a constant stress blade

To minimize stresses on long blades, they are designed with variable cross sectional area along the blade height. The cross sectional area of tapered blades may be varied in such a way that the centrifugal stresses are uniform over most of the blade length. Therefore, where the peripheral speed of the blade is maximum the mass per unit length of blade is minimum, and, where the resultant centrifugal force is maximum, the blade section is maximum. This gives a more uniform stress blade than would be obtained in a blade of constant cross section. In order to obtain an even more uniform centrifugal stress, the blade is often tapered in width [1,6]. To calculated the area along the blade as shown in **Fig.(4)**, let (A) be the cross-sectional area of the blade at radius (R), and (A+dA) the cross-sectional area at radius (R+dR). And suppose σ_{cf} is the uniform centrifugal stress. Then, F_{cf} is the internal force in blade at radius R.F_{cf} + dFcf is the internal force in blade at radius R+dR, and A. dR is the Volume of the blade element. Centrifugal force on blade element = $\rho \, . \, A \, . \, \omega^2 \, . \, R \, . \, dR$

Equilibrium equation of element:

$$F_{cf} + d\,F_{cf} + (\rho \, . \, A \, . \, \omega^2 \, . \, R \, . \, dR) = F_{cf}$$

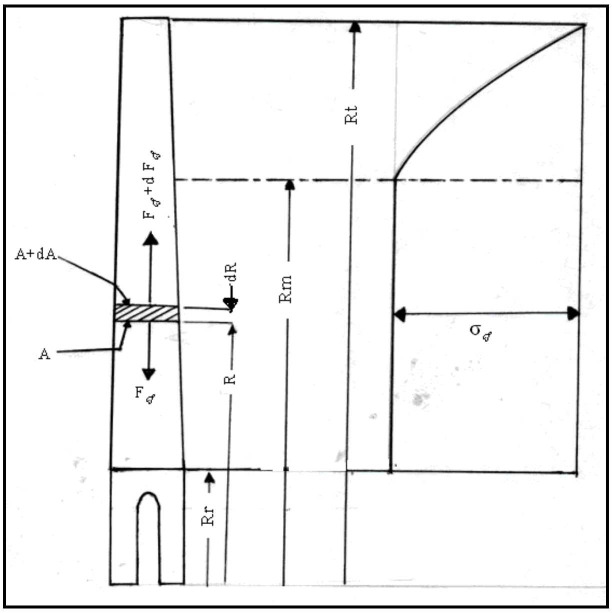

Figure (4) Design of a constant stress blade [8]

The internal force on the element is:

$$dF_{cf} = -\rho.A.\omega^2.R.dR \quad \text{.......... (44)}$$

Then since $F_{cf} = \sigma_{cf}$. A

and, $F_{cf} + d\,F_{cf} = \sigma_{cf}$. (A+dA), Therefore $d\,F_{cf} = \sigma_{cf}$. dA

Substituting into Eq.(39), the area element becomes:

$$dA = -\frac{\rho.\omega^2.A.R}{\sigma_{cf}}dR \quad \text{.......... (45)}$$

or,

$$\frac{dA}{A} = -\frac{\omega^2.\rho.R}{\sigma_{cf}}dR \quad \text{.......... (46)}$$

After integration, Eq.(42) becomes:

$$\ln\frac{Ar}{A} = \frac{\rho.\omega^2}{2.\sigma_{cf}}(R^2 - Rr^2) \quad \text{.......... (47)}$$

where, in this design the centroids of the various cross-sections area should, as far as possible, lie on one radial line so as to eliminate bending due to inertia forces. By setting the line of centroids a little forward of the radial line, the bending moment due to centrifugal force may be utilized partially to counteract the moment due to impulse.

7. Results and Discussion

Table (1), list the six cases taken in this study. **Figure (5)** shows the variation of cross-sectional area with blade height. It shows that the greatest area removed is between case A and case E.

Table (1) Type of cases

Cases	Blade edge	Blade cross-sectional area	$\Re$
A	straight	constant	1
B	straight	variable	1
C	straight	variable	0.577
D	tapered	variable	1
E	tapered	variable	0.431
G	Constant stress		-

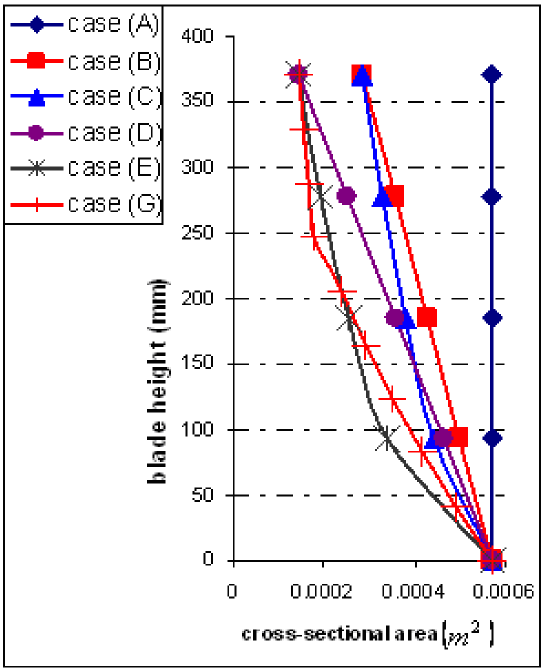

Figure (5) Variation of cross-section area with the blade height

Figure (6) shows the variation of centrifugal force with blade height at point (1). It shows that the centrifugal force reduces with blade height for all cases. However, case A (constant cross-sectional area) produced larger force while case E produced the minimum force. The variation is seen to be linear. This is due to the fact that the centrifugal force is a function of blade mass, with blade height increase; the remaining mass of blade reduces, hence producing less centrifugal force. Comparing case A and case E, it is noticed that the centrifugal force in case E reduces to less than one half of the centrifugal force in case A, this is because the area of the blade is reduced as was show in **Fig.(5)**.

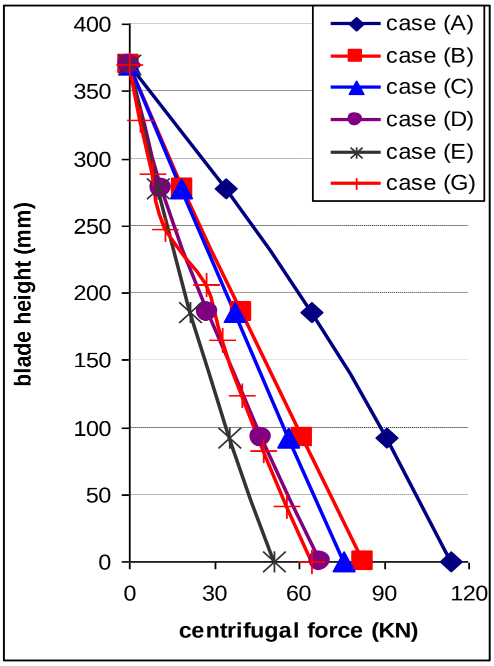

Figure (6) Variation of centrifugal force with the blade height point (1)

Figure (7) shows the variation of centrifugal stress with blade height at point (2). The centrifugal stress in case E is one half of the centrifugal stress in case A at blade root (maximum stress), because of reducing the centrifugal force. Variable cross sectional area with constant stress, case (G) gives acceptable stress. This is clear when comparing case (A) with case (G), in case A the assumption is a blade of constant cross sectional area. While case (G) the blade is tapered. This taper reduces blade sectional mass, hence reducing the stress to less than one half that of case A.

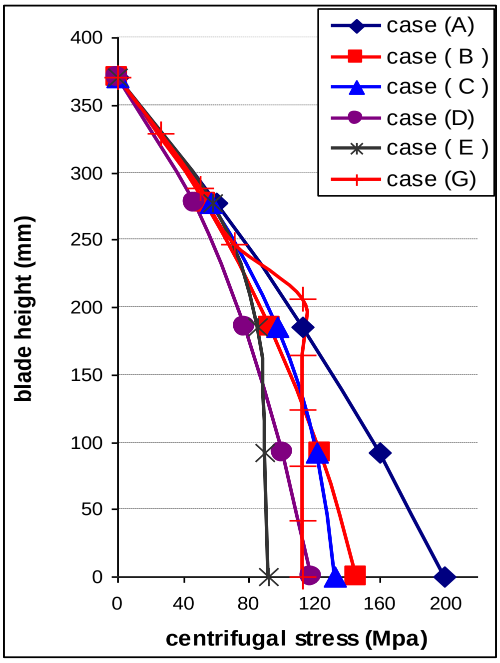

Figure (7) Variation of centrifugal stress with the blade height point (2)

Figures (8), **(9)**, and **(10)** show the variation of bending stress with blade height for points 1, 2 and 3, respectively. It is noticed that the bending stress in case E is larger than the bending stress in case A, because of the reduction cross sectional area.

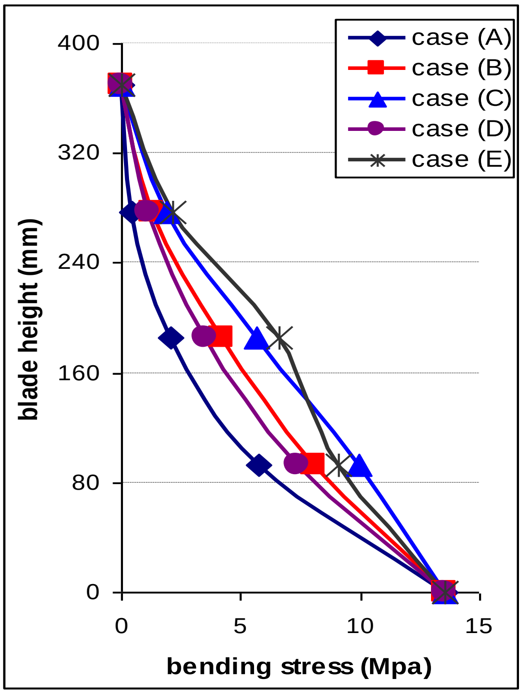

Figure (8) Variation of bending stress with the blade height point (1)

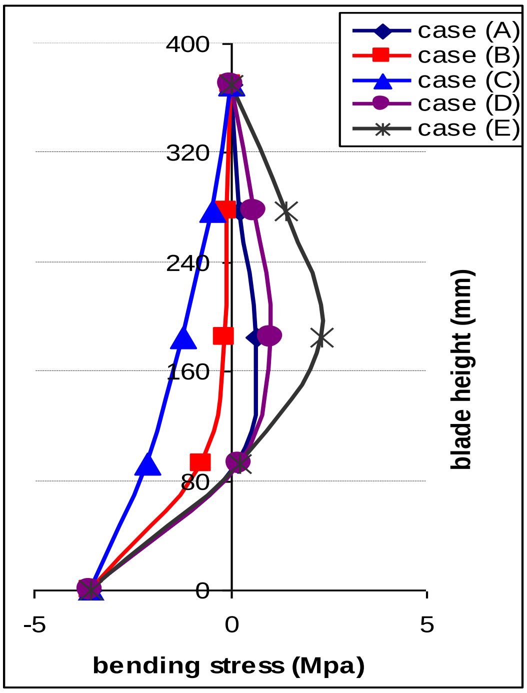

Figure (9) Variation of bending stress with the blade height point (2)

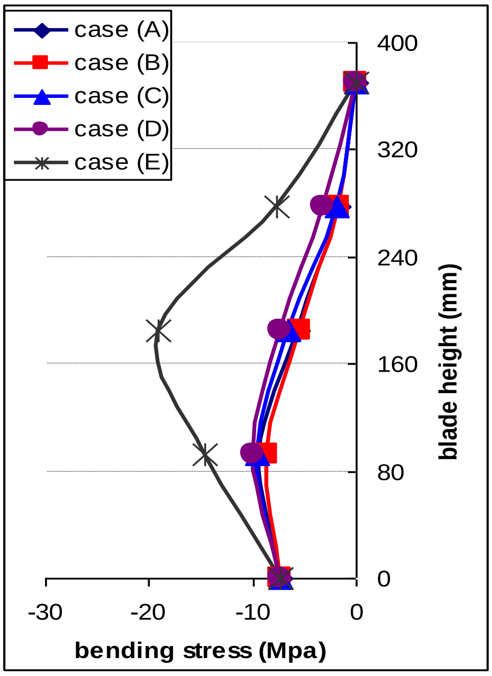

Figure (10) Variation of bending stress with the blade height point (3)

Figures (11), **(12)**, **(13)** show the variation of total stress along the blade for points 1, 2 and 3, respectively. It is noticed that the value of bending stress didn't affect the value of total stress because the bending stress is very little as compared with the centrifugal stress.

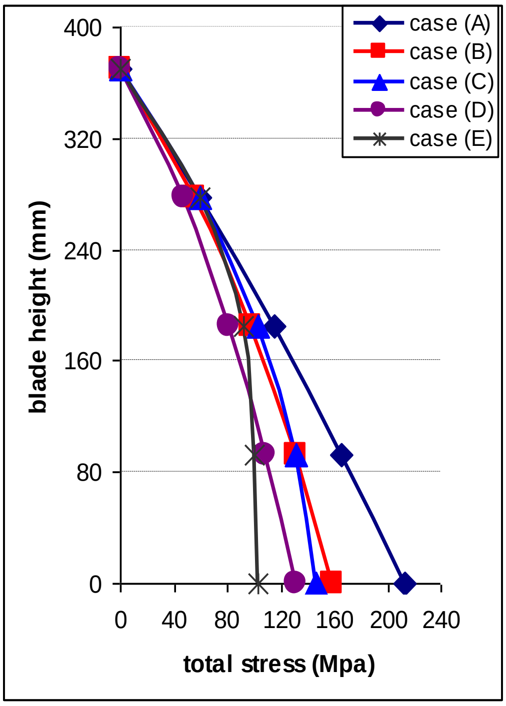

Figure (11) Variation of total stress with the blade height point (1)

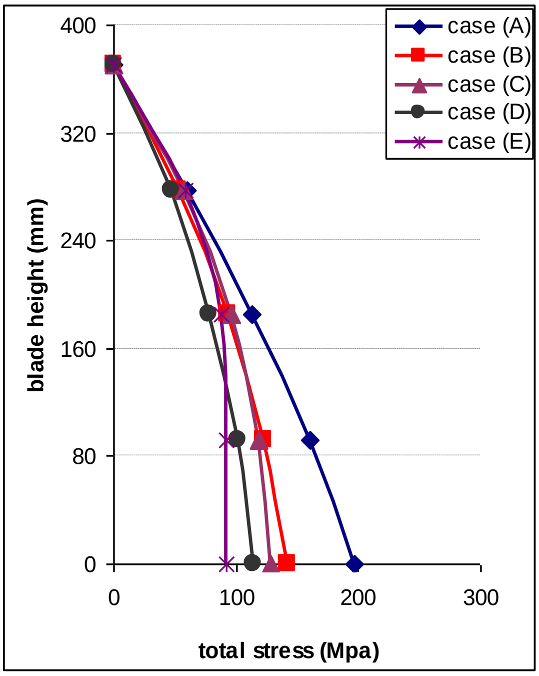

Figure (12) variation of total stress with the blade height point (2)

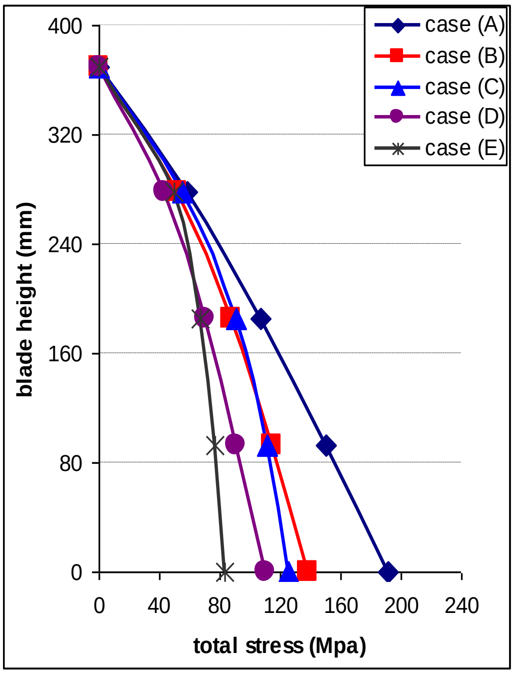

Figure (13) Variation of total stress with the blade height point (3)

Figure (14) shows a comparison centrifugal stresses between the constant area blade case (A) and the constant stress blade case (G). It shows that case (G) is better and hence safer at the root section, the critical and maximum stress section. While at the upper half, both have similar stresses. This figure shows that Case (G) is favored up on case (A).

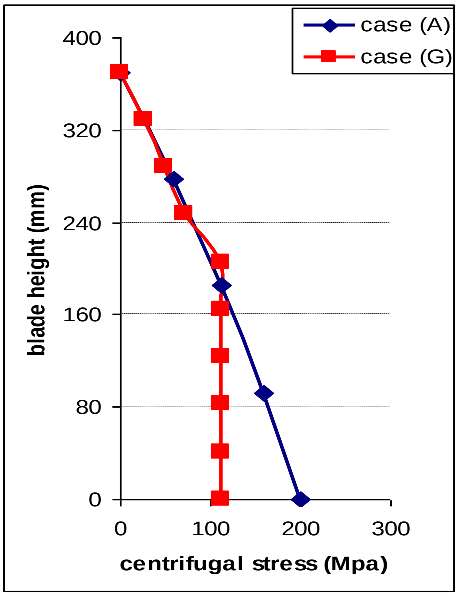

Figure (14) Comparison between constant cross sectional area blade case (A) with constant stress blade case (G)

Figure (15) shows the performance of the blades. It shows that all the cases drawn fall within the acceptable range of lift, [4].

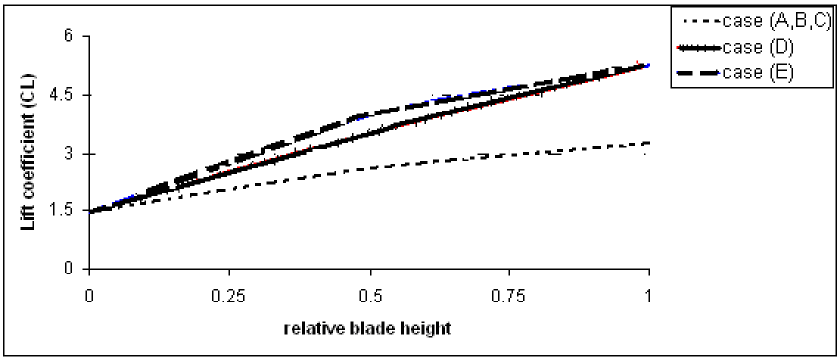

Figure (15) Performance of blades

Figure (16) shows a comparison of the case (G) to stages (11, 13, 15) taken with that of Ref. [7]. It shows an acceptable loading trend.

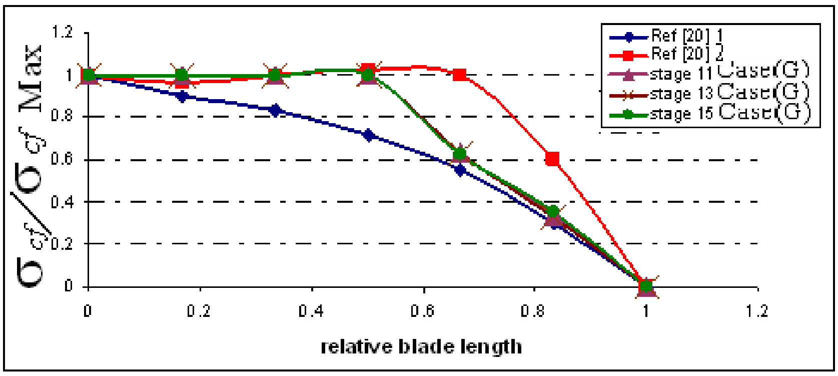

Figure (16) Comparison of case (G) with Ref.[7]

8. Conclusions

1. In design of rotor blades, the centrifugal stress must be calculated because it is considered the main source of stresses applied to the turbine rotor blade.
2. The maximum stress is at the blade root.
3. To minimize stresses, it is a normal practice to use variable cross sectional area blades.
4. Design of a constant stress blade gives a blade with less stress at the lower half and a similar stress at the upper half.
5. Reducing the blade material by reducing the blade cross section did not affect blade performance.
6. Comparison of calculated stresses with literature was good.

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List of Symbols

Ar =	Cross section area of blade root.
Am =	Cross section area of blade middle.
At =	Cross section area of blade tip.
A =	Is the area of blade at radius R.
Fcf =	Centrifugal force.
R =	Raduis.
Rr =	Radius from shaft center at root blade.
Rm =	Radius from shaft center at middle blade.
Rt =	Radius from shaft center at tip blade.
m =	Mass of blade.
Lb =	Blade height.
Mv, Mu =	Bending moment about the axes v,u.
ω =	Speed in radians.
ρ =	Density of blade material.
cf2.cf3 =	Axial velocity of inlet and outlet.
Vw2, Vw3 =	Whirl velocity of inlet and outlet.
C_r =	Relative velocity.
Nb =	Number of blades.
p2, p3 =	Pressure inlet and outlet of moving blade.
s =	Spacing or pitch from two moving blade.
ρ_f =	Density of steam.
σcf =	Centrifugal stress.
σt =	Total stresses (tension and bending).
σ_{cfm} =	Is the centrifugal stress at mid height for constant cross-sectional area.
$\sigma_{b1}, \sigma_{b2}, \sigma_{b3}$ =	Bending stresses in point 1, 2, 3.
$vv1, vv2, vv3 =$	Destines from the axis u and the point 1, 2, 3.
$uu1, uu2, uu3 =$	Destines from the axis v and the point 1, 2, 3.
Iv, Iu =	Moments of inertia of axis vv and uu.