Eng.Tech.Vol.26,No.2,2008

Stress Evaluation of Low Pressure Steam Turbine Rotor bBlade and Design of Reduced Stress Blade

Dr. Arkan K. Husain Al-Taie

Received on: 18/12/2005
Accepted on: 3/5/2007

Abstract:
The low pressure steam turbine blade rows have a history of stress failure . They suffer from tensile and bending stresses partly due to the centrifugal force as a result of high rotational speeds and partly due to high pressure, temperature and speed steam loading. The centrifugal force is one of the problems that face the designers of turbine blades especially the long ones. The designer always aims at reducing these stresses. One way to do so is by the reduction of blade mass. That is to make the blade of variable cross section stead in of straight. This paper presents the method of reducing cross section. Analysis of such blade is also done as applied to the (P 23-14A) steam blade.

Key words: Steam Turbines, Design, Stress.

الخلاصة:
ان لريش التوربين البخاري الطويلة تأريخ من الفشل الاجهادي. فهي تعاني من اجهادات شدية و التوائية جزئيا" بسبب قوة الطرد المركزي الناتج من سرعة الدوران العالية وجزئيا" بسبب تحميل البخار العالي الضغط، درجة الحرارة و السرعة. ان قوة الطرد المركزي هي أحدى المشاكل التي يواجهها المصمم لريش التوربابن و خصوصا" الطويلة منها. ان المصمم دائما" يهدف الى تقليل تلك الاجهادات. أحدى الطرق لتحقيق ذلك هو بتقليل كتلة الريشـــة. و ذلك بصـــناعة ريشـــة ذات مقطع متغير بدلا" من الريـــش المســــتقيمة . هذا البحث يســتعــرض طريقة لتقليل مســـاحة مقطع الريشــــة. وكذلــك تم اســـــتعراض تحليل لــهــكــذا ريشـــة .

Introduction:

A steam turbine may be defined as a form of heat engine in which the energy of the steam is transformed into kinetic energy by expansion through nozzles , and the kinetic energy of resulting jet is in turn converted into force doing work on rings of blades mounted on a rotating disc.

The majority of steam turbines have, therefore two important elements:

Mechanical Engineering Dept/ Univ. of Tech.

https://doi.org/10.30684/ etj.26.2.6
University of Technology-Iraq, Baghdad, Iraq/2412-0758

i) The Nozzle (stator), in which the steam expands from a high pressure and a state of comparative rest to a lower pressure and a state of comparatively rapid motion.
ii) The Blade (rotor), in which the stream of steam particles has its direction and hence its momentum changed. A blade force

results from the difference between the momentum entering and the momentum exiting the rotor blade row.
The blades are attached to the rotating element of the machine or rotor shaft. The resultant of blade forces is then converted into shaft power to drive the load. On the ather hand, the nozzles are attached to the stationary part of the turbine (casing), as shown in Fig (1-1).
The rotor blades facing the steam will carry most of the flow loading . Hence to design these blades , the flow has to be studied first , flow forces to be estimated and blade stresses to be evaluated.

Nomenclature:

Ar = cross section area of blade root.
Am = cross section area of blade middle.
At = cross section area of blade tip.
Fcf = centrifugal force.
Rr = radius from shaft center at blade root.
Rm = radius from shaft center at blade middle.
Rt = radius from shaft center at blade tip.
m = mass of blade.
Lb = blade height.
Mv = bending moment about the axes v.
Mu = bending moment about the axes u.
ω = speed in radians per second.
ρ = density of blade material.
cf2 =axial velocity at inlet.
cf3 =axial velocity at outlet.
Vw2 = whirl velocity at inlet.
Vw3 = whirl velocity at outlet.
p2 =pressure at inlet to the moving blade.
p3=pressure at outlet to the moving blade.
s =spacing between two moving blades.
ρf = density of steam.
σcf = centrifugal stress.
σt =total stresses (tension and bending).
$vv1, vv2, vv3$ =distances from the axis u to points 1,2, and 3, respectively.
$uu1, uu2, uu3$ =distances from the axis v to points 1,2, and 3, respectively.
Iv =moments of inertia of axis vv.
Iu =moments of inertia of axis uu.

Forces on Rotor Blades:

Turbine rotor blades are positioned in a very harsh environment .They the incoming , hot , fast and high pressure streams exiting from the nozzles. The loading on a rotating turbine blade is composed of:
i) centrifugal force due to rotation.
ii) bending force due to the fluid pressure and
change of momentum.
iii) bending force due to centrifugal action, resulting from that the centroids of all section do not lie along a single radial line.

Stresses:

The steady state stress at any section of a parallel sided blade is a combination of direct tensile load due to centrifugal forceb and the bending load due to steam force, Both of which are acting on that portion of blade between the section under consideration and the tip section. The direct tensile stress is maximum at the blade root. It is decreasing towards the blade tip. The tensile stress (centrifugal stress) depends on the mass of material in the blade , blade length , rotational speed , and the cross sectional area of blade. Impulse blades are subject to bending stresses from centrifugal force if the centroids of all sections do not lie a long a single radial line

Mechanical Engineering Dept/ Univ. of Tech.

and the tangential force exerted by fluid . While Reaction blades have an additional bending stress that is due to the large axial thrust resulting from the pressure drop which occurs in the blade passages. The bending stress is maximum at the blade root. The combined tension and bending stress is also maximum at blade roots and diminishes with radius.

If the blade is tapered, the direct tensile stress diminishes rapidly towards the blade 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. The blade material is much more effectively utilized there. Provision must be made in the blade design to get a blade that withstands all these stresses encountered in operation at an acceptable long service life.

Calculation of Centrifugal Stresses:

The centrifugal stressesare more important than other stresses exerted by flow forces that is its value is dominant. It is not reduced by increase of section area , since increased section simply means a greater load carried on a proportionately larger area , resulting in the same stress.

Obviously sufficient cross- section area must be provided in the blade root to withstand the maximum force there. On the ather hand, the material used must be capable of withstanding these stresses without fatigue failure. To calculate the centrifugal stresses on a moving blade , assume a single blade fixed to the disc similar to the one shown in Figure (1-2) below:

The root section of the blade has radius Rr and peripheral section radius Rt . The area of the blade profile in section (x) may be denoted as A(x) . Let the x axis be chosen to be perpendicular to the axis of rotation.

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 that section divided by the area of the blade section, and can be written as :-

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

If an infinitesimal element (dz) is separated in section (Z) shown in Fig (1-2) ,the force developed by this element during disc rotation is :-

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

Where,

$$dm = \rho.\ A(z).\ dz \qquad (3)$$

Substituting Eq. (3) into Eq. (2) gives:-

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

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

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

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

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

The cases of interest are taken in this study given the code names:

Case A

In this case the blade is assumed to be of constant cross-section area. The centrifugal force given by Eq.(5) becomes:-

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

Integrating gives :-

Mechanical Engineering Dept/ Univ. of Tech.

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

It is clear that at the tip of the blade, that is when (x = Lb) , the force F_{cf} (Lb) = 0 . On the ather hand the maximum centrifugal force is at the blade root, that is when (x = 0),
the Centrifugal stress is determined by using Eq. (6) , and Eq. (1) becomes:-

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

where (Z) = A (X) = constant
It is clear that at the tip of the blade i.e at (x = Lb) , the stress σ_{cf} (Lb) = 0 . Whereas the maximum centrifugal force is at the blade root (x = 0).The maximal centrifugal stresses for a constant cross-section area blade are.

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

or

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

Case B

In this case the blade is assumed of variable cross–sectional area . To find the centrifugal force acting on the rotor blade, it is necessary to find an equation describing the change of cross-section with blade height . A relation of the form [5] is used:

$$A(z) = Ar \,.\left(\frac{At}{Ar}\right)^{\frac{z}{Lb}} \qquad (12)$$

For the determination of the centrifugal force use Eq. (12) and Eq. (5) , then:-

$$F_{cf}\ (x) = \rho\, \omega^2 \int_x^{Lb} [\, Ar \,.\left(\frac{At}{Ar}\right)^{\frac{z}{Lb}}].\ (Rr + Z)\ dz \qquad (13)$$

After Integration it becomes:-

$$F_{cf}\ (x) = \rho.\omega^2 \left[\frac{Ar.\left(\frac{At}{Ar}\right)^{\frac{z}{Lb}}.Rr.Lb}{\ln\left(\frac{At}{Ar}\right)} + \frac{Ar.\left(\frac{At}{Ar}\right)^{\frac{z}{Lb}}.z.Lb}{\ln\left(\frac{At}{Ar}\right)} - \frac{Ar.\left(\frac{At}{Ar}\right)^{\frac{z}{Lb}}..Lb^2}{\left(\ln\left(\frac{At}{Ar}\right)\right)^2} \right]_x^{Lb} \qquad (14)$$

When (x = Lb) at blade tip the centrifugal force is zero, and at blade root (x = 0) the centrifugal force is maximum .
The Centrifugal stresses are determined by dividing Eq. (14) by Eq. (12) as follows:

$$\sigma_{cf}\ (x) = \frac{Fcf\ (x)}{A(x)} \qquad (15)$$

Case C

If the variable cross–sectional area of the moving blade follows the relation [6]:

$$A(z) = Ar- (Ar - At) \left(\frac{z}{Lb}\right)^{\Re} \qquad (16)$$

where,

$$\Re = \frac{\ln\left(\frac{Ar - At}{Ar - Am}\right)}{\ln 2} \qquad (17)$$

To determine the centrifugal force acting on the blade, Eq. (16) and Eq. (5) become:-

$$F_{cf}\ (x) = \rho\, \omega^2 \int_x^{Lb} [\, Ar- (Ar - At) \left(\frac{z}{Lb}\right)^{\Re}]\ (Rr + z)\, dz \qquad (18)$$

$$F_{cf}\ (x)=\rho.\omega^2 \left[Ar(Rr.z+\frac{z^2}{2}) - \frac{(Ar - At)}{Lb^{\Re}} \left(\frac{Rr.z^{\Re+1}}{\Re+1} + \frac{z^{\Re+2}}{\Re+2}\right)\right]_x^{Lb} \qquad (19)$$

When (x = Lb) at blade tip the centrifugal force is zero, while at the root section of the blade (x = 0), the centrifugal force is maximum and Eq. (19) becomes :-

$$F_{cf}\ (0) = \rho.\omega^2 \left[Ar.Lb\ (Rr + \frac{Lb}{2}\ ) - (Ar\text{-}At)\ (\frac{Rr.Lb}{\Re+1} + \frac{Lb^2}{\Re+2})\right] \quad (20)$$

The Centrifugal stresses are determined by dividing Eq. (19) by Eq (16). The maximum centrifugal stress at blade root becomes:-

$$\sigma\ cf\ (0) = \rho.\omega^2 \left[Lb\ (Rr + \frac{Lb}{2}\ ) - (1\text{-}\frac{At}{Ar})\ (\frac{Rr.Lb}{\Re+1} + \frac{Lb^2}{\Re+2})\right] \quad (21)$$

Bending Stresses:

The moving blade suffers from the gust forces in addition to the centrifugal force. The gust forces are exerted by the working fluid (the hot, fast and high pressure steam flow), on the moving blade surface. This is essentially a distributed load in which the general case varies along the blade length . The impulse blade is subjected to bending from the tangential force exerted by the fluid . While the reaction blades have an additional force resulting from 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)).

The moment of this force and the stress due to it are both maximum at the blade root . The bending stress diminishes with increase in the radius if the blade is tapered. The bending stress can be made to increase at greater radii. Consider the blade as a cantilever beam fixed by root section . The bending moment effect, on the level of the main axes of inertia (u,v) shown Fig (1-3).

The axial force:

$$qa = \rho f.cf3\ (Vw_2 - Vw_3\ ) \quad (22)$$

The tangential force:

$$qw = \rho_f\ .cf3.(cf2\text{-}cf3) + (p2\text{-}p3).s \quad (23)$$

Or when constant axial velocity (cf2=cf3) is assumed, then the tangential force is:

$$qw = (p2\text{-}p3).s \quad (24)$$

The resultant force is:-

$$q = \sqrt{qw^2 + qa^2} \quad (25)$$

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

The force for the axis (U):

$$qu = q.\sin\ (Q) \quad (26)$$

The force for the axis (V):

$$qv = q.\cos\ (Q) \quad (27)$$

But Q is the angle between the force on the axis (v) and the result force, hence is:

$$Q=b+st \quad (28)$$

$$b=\tan^{-1}\ (\frac{qa}{qw}) \quad (29)$$

$$st = [\tan^{-1}[\frac{-Ixy}{\frac{Ix - Iy}{2}}]/\ 2] \quad (30)$$

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

$$Mv = \int_x^{Lb} qv\ (z).(z - x).dz \quad (31)$$

$$Mu = \int_x^{Lb} qu\ (z).(z - x).dz \quad (32)$$

The bending stress will be calculated for the following points in which the stress is maximum [3] , [4] shown in Fig (1-3) .

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

$$\sigma_{b1} = \frac{Mv * vv1}{Iu} + \frac{Mu * uu1}{Iv} \quad (33)$$

While for the exit or trailing edge, it is:

$$\sigma_{b2} = \frac{Mv * vv2}{Iu} - \frac{Mu * uu2}{Iv}$$

(34)

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:

$$\sigma_{b3} = -\frac{Mv * vv3}{Iu}$$

(35)

Where the negative sign stands for compression:

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 .

It is clear that since the leading edge of the blade is in tension from the bending force and the centrifugal stress, the maximum stress will be found at the leading edge as follows:

The total stress in leading edge point (1) is:-

$$\sigma t1 = \sigma_{cf} + \sigma_{b1} \quad (36)$$

The total stress in trailing edge point (2) is:-

$$\sigma t2 = \sigma_{cf} + \sigma_{b2} \quad (37)$$

And the total stress in point (3) is:-

$$\sigma t3 = \sigma_{cf} + \sigma_{b3} \quad (38)$$

Discussion:

Fig (1-4) shows the variation in cross-sectional area with blade height for cases A, B, and C. It is noticed that the blade area of case C is less than that for case B is less than that for case A. One can imagine the amount of area removed by comparing cases A and C.

Fig. (1-5) shows the variation in centrifugal force with blade height for the three cases. It shows that for all cases the centrifugal force reduces with blade height. However,in case A a blade of (constant cross-sectional area) produced larger centrifugal force. while case C produce the minimum centrifugal force. The variation is linear. This is due to the fact that the centrifugal force is a function of blade height. With blade height increase, the remaining mass of blade trying to pull away the blade from its root is reduced , hence producing less centrifugal force. Comparing case A with case C, it is noticed that the centrifugal force in case C reduces to less than one half of the centrifugal force for case A, this is because the area of rotor blade is reduced.

Fig (1-6) shows the variation in centrifugal stress with blade height for the three cases. This figure shows a similar trend to that for the centrifugal force shown in Fig. (1-5). The centrifugal stress is function of centrifugal force and cross-sectional area. It again shows that case C produces less than one half the centrifugal stress of case A as when calculated at blade root (section of maximum stress). This is mainly due to the reduction in centrifugal force.

Fig (1-7) shows the variation in bending stress with blade height for the three cases as calculated for point no. (1). It shows that the bending moment is maximum at blade root, and reduces with blade height for the three cases. It also shows that the case A is (26) producing less bending stress as when compared to the other two cases. This can be understood referring to the equation used to calculate the bending stress and the misalignment of blade axis.

Fig (1-8) shows the variation in the bending stress of the three cases for point no. (2). It shows that the stress increases with radius to about the middle section and them starts to drop back to zero at the blade tip. It can be seen that case A gave the least stress all along. Again this can be understood referring to the mathematical expression used and the misalignment of axis of the blade.

Fig (1-9) shows the variation in bending stress with blade height of the three cases for point 3. It shows that case A gaves the minimum stress all along the blade height as when compared to the other two cases.

Fig (1-10) shows the variation in the total stress of the three cases for point no. (1). It shows that case A gives the highest stress at blade root, and all cases give a diminishing total stress with blade height. This is due to the fact that the centrifugal stress is more dominant than the bending stress. And that the blade of case A has the greatest mass than the other two. And that as the radius increases this mass reduces.

Fig (1-11) shows the variation in the total stress of the three cases for point no. (2). Again the same discussion presented for Fig. (1-10) can be presented here.

Fig (1-12) shows the variation in total stress along the blade for point 3. It is noticed that the effect of the bending stress on total stress can not be seen. This is due to the difference in the order of magnate of the two stresses.

Conclusions:

The following concluding remarks can be made:

1. In the mechanical design of rotor blades , the centrifugal force, and hence, the centrifugal stress are dominant as when compared with bending.
2. The centrifugal force is maximum at the blade root. To carry this load the blade root cross-sectional area is made larger as when compared to other sections.
3. The use of variable cross sectional area can reduce the centrifugal force and the centrifugal stress.
4. The bending stress is greater for variable cross sectional area blades when compared with straight blades.
5. The total stress depends mainly on centrifugal stress and takes its trends.

References:

1- Kostyok A. and Frolov V. ,"**Steam and Gas Turbine** ", English translation, Mir publisher, Moscow, 1988.

2- Church Edwin F. ., "**Steam Turbines**", revised edition, McGraw-Hill book company Inc., London, 1987.

3-.Lee John F., "**Theory and design of steam and gas turbines**", McGraw-Hill book company Inc., London, 1999.

4-Mukherjee D. K., "**Stresses in turbine blades Due to temperature and load variation**", ASME paper no.78-GT,1988,pp.158.

5- ШУбенко-ШУбИН Л. А., "**Прочност паровых турбин**",М.,Машиностроение, Москва, 1993.

6- Жирицкий Г.С., Стрункин В. А. "**Конструкции и расчёт на прочность Деталей паровых и ґазовых и турбин**" м. машґиз, 1988.

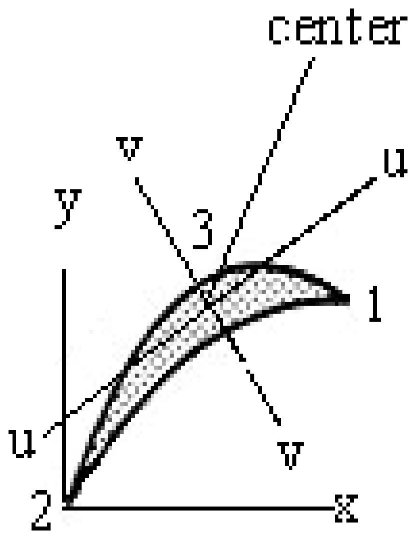


Fig (1-3) Cross-section area of moving blade

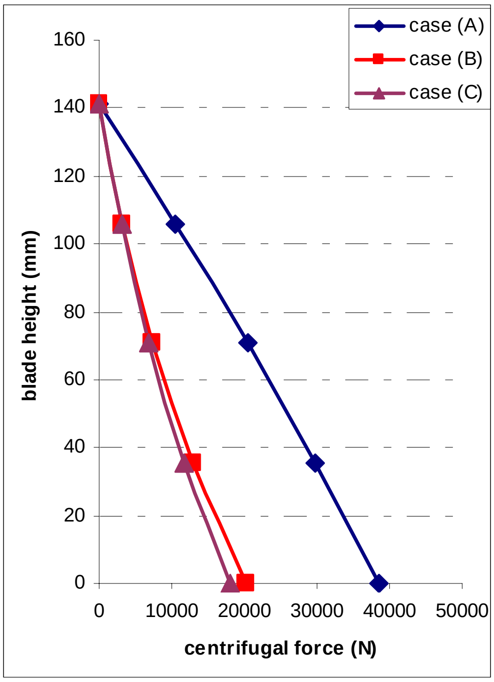


Fig (1-5) Variation in centrifugal force with the blade height

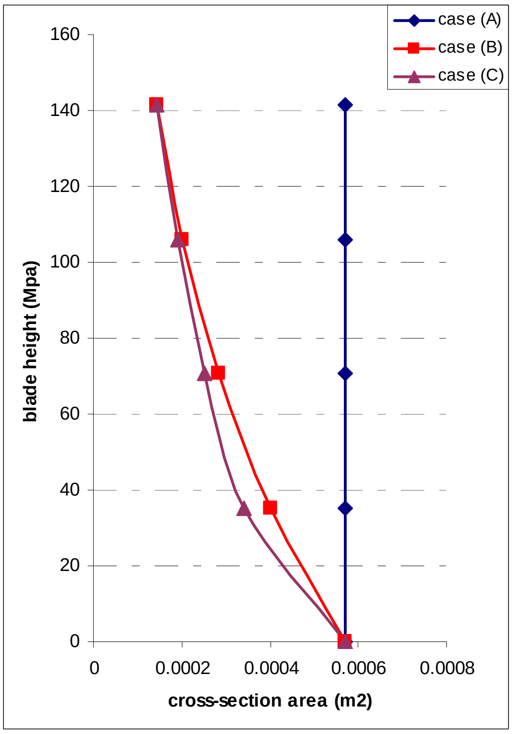


Fig (1-4) Variation in cross-section area with the blade height

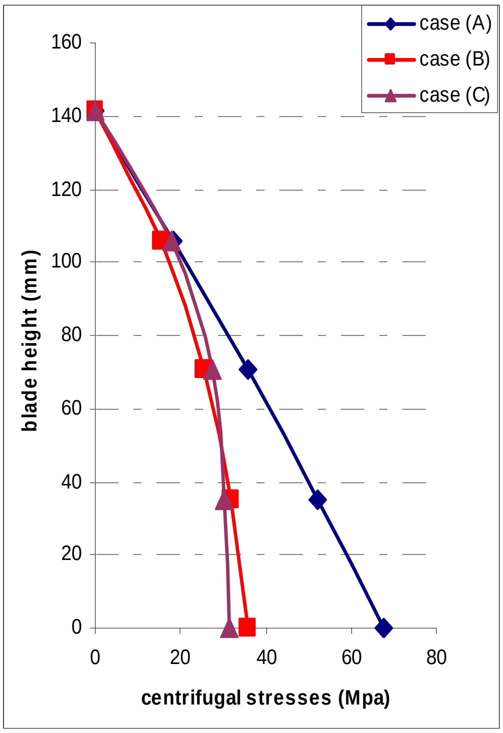


Fig (1-6) Variation in centrifugal stress with the blade height

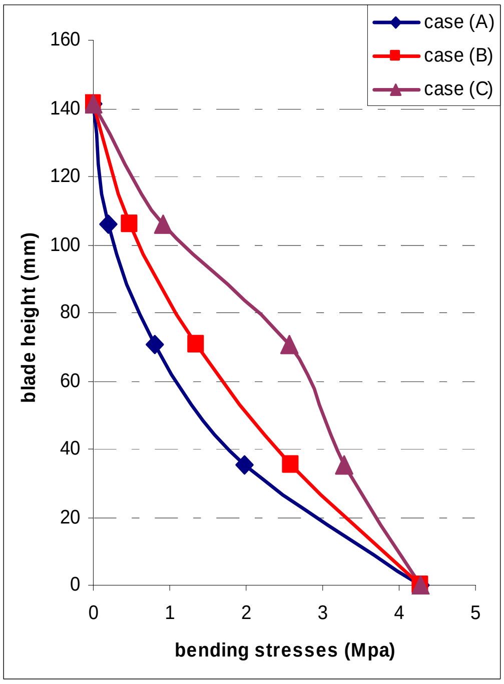


Fig (1-7) Variation in bending stress with the blade height point (1)

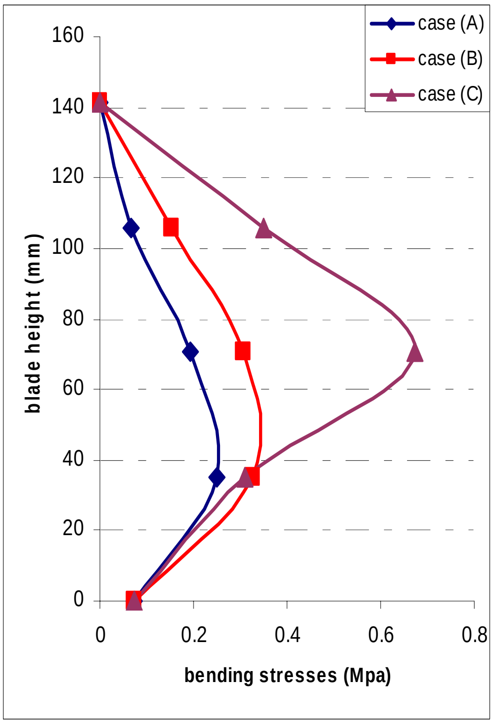


Fig (1-8) variation of bending stress with the blade height point (2)

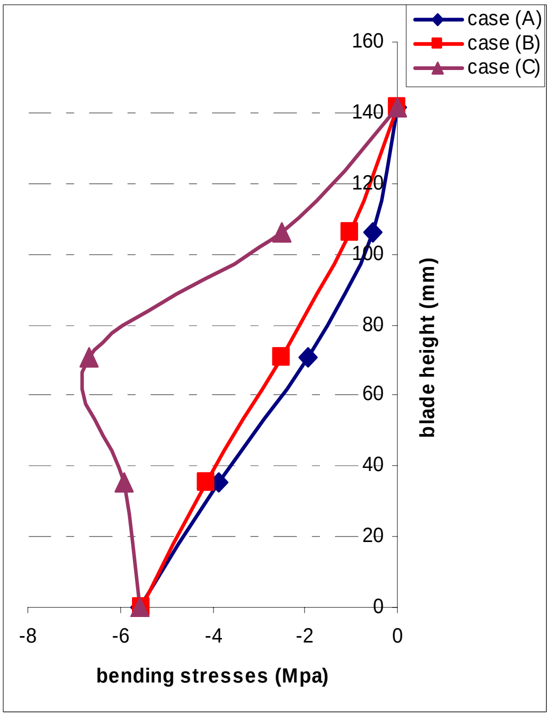


Fig (1-9) Variation in bending stress with the blade height point (3)

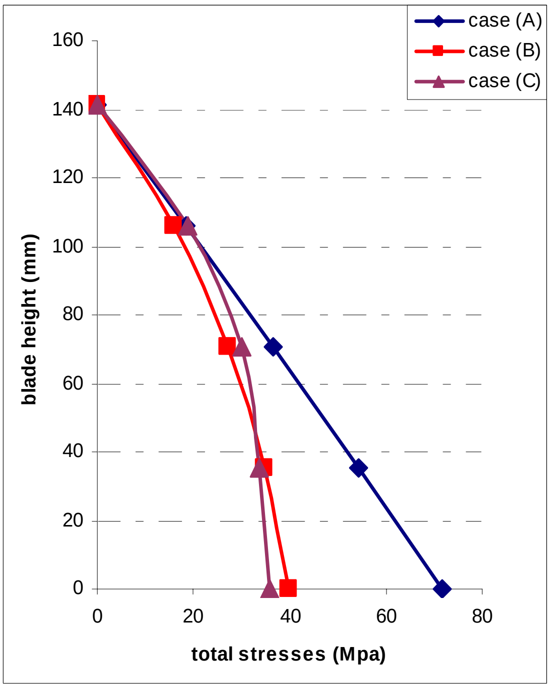


Fig (1-10) variation of total stress with the blade height point (1).

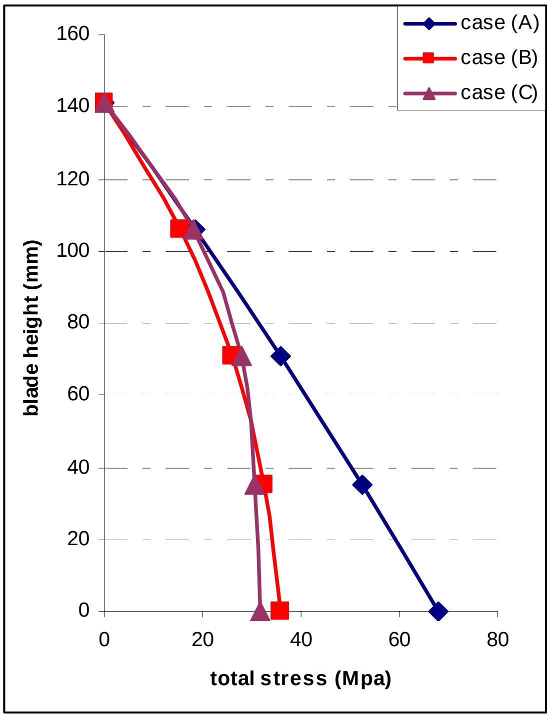


Fig (1-11) Variation in total stress with the blade height point (2).

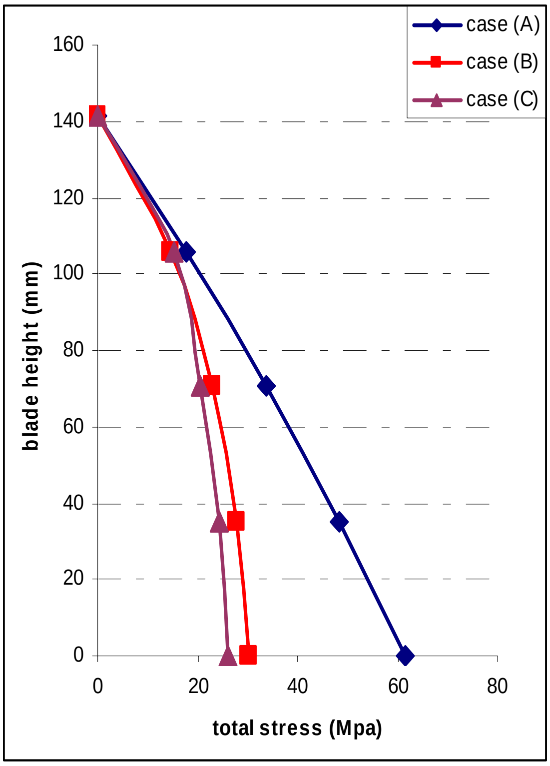
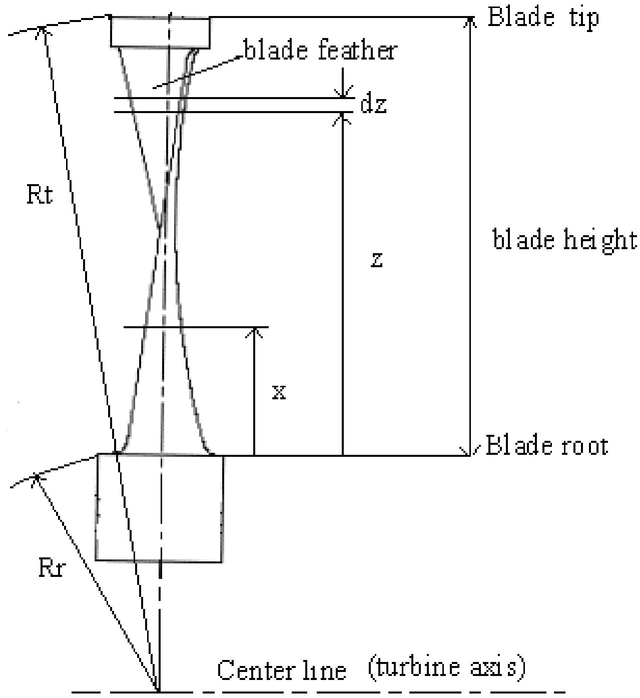


Fig (1-12) Variation in total stress with the blade height point (3)



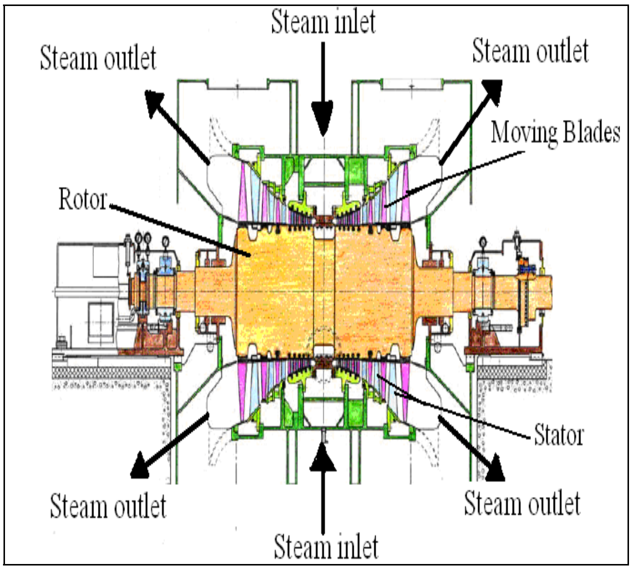


Fig (1-1)cross section of double entry steam Turbine.