



Effect of the Mechanical and Thermal Stresses of Rotating Blades

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Abstract:

Rotating blades are the important parts in gas turbines. Hence, an accurate mathematical estimation (F.E.M) of the stresses and deformations characteristics was required in the design applications to avoid failure. In recent year's there are researchers interest in the effect of temperature on solid bodies has greatly increased, The main of this study investigated the thermal and rotational effects. So, the thermal stresses due to high pressure and temperature are studies, also determine the steady state stresses and deformations of rotating blades due to mechanical effect. Many parameters such as thickness and centre of rotating are investigated in this paper. The study results can ensure good recommendation for the effect of the mechanical and thermal stresses of rotary blades.

Keywords: Thermal Stresses, Finite Element Method, Mechanical Stresses , Blade Material , Design .

Introduction

Rotating blades was one of the common structural elements used in several practical machines such as turbomachinery system, turbojet engines, turbofan, and helicopter rotors. Because turbomachinery blades are assumed as a shell, the numerical technique should be capable of representing this complicated structure based upon the shell theory. In this paper the type of rotating blades were used is rectangular blade, the geometry for this type was show in figure (1). Ramamurti and Sreenivasamurthy [1] studied the dynamic stress analysis of rotating twisted and tapered blades. The

finite element method was used to determine the stresses and deformations. Three-dimensional, twenty-node isoparametric elements have been used for the analysis. Extensive analysis has been done for various pre-twist angles, skew angles, breadth to length ratios, and breadth to thickness ratios of the blades. Experiments were carried out to determine the stresses for the verification of the numerical results. Lee [2] determined the frequencies and mode shapes of turbomachinery blade having both camber and twist. The body forces were considered to be centrifugal forces generated within the shell due to rotation of the blade. Omprakash and Ramamurti

[3] carried out the steady state, dynamic stress and deformation analysis of high pressure stage turbomachinery bladed discs taking into account all the geometric complexities involved and included the contributions due to initial stress and membrane behavior, which used a triangular shell element with six degrees of freedom per node. Yoo [4] studied the vibration analysis of rotating pre-twisted blade with a concentrated mass. The blade has an arbitrary orientation with respect to the rigid hub to which it is fixed. The equations of motion are derived based on a modeling method that employs hybrid deformation variables. The resulting equation for the vibration analysis was transformed in the dimensionless parameters on the model characteristics of the rotating blades were investigated through numerical analysis

Blade Materials

The elimination of blade corrosion and designing against creep and fatigue, to a large extent, is a problem of finding suitable materials. Another factor, especially in high-pressure work, is the working temperature of the blades. Under high temperatures, some materials tend to disintegrate, and certain mechanical properties, e.g. hardness (in the sense of resistance to abrasion), are affected adversely.

The properties, which are desirable for example in turbine blading, vary according to the conditions of operation and according to the location of the blading in the turbine [5]. At high-pressure end, the temperature is high, and the blades are sometimes subjected to erosive action. Great strength is not required as the blades are usually short. The material should be hard at high temperatures. Suitable materials are 5% Nickel Steel, Stainless Steel, and Monel

metal. But for high temperature gas turbines, "Nimonic" alloy series are recommended. They are basically Nickel-Chromium alloys, which withstand oxidation, mechanical stresses, and creep at working temperatures in excess of 1000° C. in the middle part of the turbine, the temperature and stress conditions are moderate, but corrosion is likely. Brass is probably the most suitable material.

In the final stages, and especially in large machines, the blades are long and heavily stressed. Fatigue action due to vibration may exist. Further than this, anti corrosion properties are also desirable. Brass is a suitable material where the stress conditions allow of its use, but when the stresses are high, 5% Nickel Steel, Stainless Steel, Monel metal or Phosphor Bronze may be used.

Turbine Blade Design

Turbine blades operate at 1400 °C and are required to have a service life of 10000- 20000 hours. The techniques used to meet these requirements are:

- 1) The use of nickel based superalloys (containing many precipitates, solid solution atoms and having high oxidation resistance)
- 2) The blades are cast a single crystal. This means that there are no grain boundaries within the structure and thus minimizes creep see Fig. (2)
- 3) The blades are internally cooled to allow increased operating temperatures. Typical cooling design features are shown in Fig. (3)
- 4) The blades are coated to increase the oxidation resistance as shown in Fig. (4). A timeline of techniques to raise engine-operating temperatures is shown in Fig. (5 &6)

Finite Element technique

The difficulties in evaluating the stress and deformations of rotating blades comes from the complex geometry of blades due to asymmetry of cross-section, pre-twist, and mounting the blades at a skew angle to the rotating disk. The pre-twist and skew of the blade cause coupling in both bending directions, and the asymmetry of the cross-section causes coupling with the tensional motion of the blade. For the above complexities, all previous investigations in this field had declared that plate or shell analysis for blades is suitable and reliable. Therefore, the shell theories with finite element provide a useful tool to analyze such structures. A superparametric shell element will be used to investigate the thermal and mechanical stresses of rotating blades. This element consists of four corner and four midside nodes as shown in Fig. (7). The degrees of freedom considered at The s are the three translations u, v, w of the midsurface and two rotations α and β of the normal to the midsurface. The Cartesian coordinates of any point of the shell and the curvilinear coordinates can be written in the form:

$$\begin{bmatrix} x \\ y \\ z \end{bmatrix} = \sum f_i \begin{bmatrix} x_i \\ y_i \\ z_i \end{bmatrix} + \sum f_i \frac{\zeta}{2} \begin{bmatrix} l_{3i} \\ m_{3i} \\ n_{3i} \end{bmatrix} h \dots\dots(1)$$

where l_{3i} , m_{3i} and n_{3i} are the direction cosines of the normal to the middle surface. Here f_i is a function taking a value of unity at the node i and zero at all other nodes, it is called as “shape function “⁽⁴⁾.as shown in table.(1).

In the kinematics formulation two assumptions are imposed:

- 1- Nodal fiber is inextensible.
- 2- Only small rotations are considered.

The displacements at any point (ξ, η, ζ) can be expressed in terms of the

nodal displacements as

$$\begin{bmatrix} u \\ v \\ w \end{bmatrix} = \sum_{i=1}^8 f_i \begin{bmatrix} u_i \\ v_i \\ w_i \end{bmatrix} + \sum_{i=1}^8 N_i \zeta \frac{h_i}{2} f_i \begin{bmatrix} \alpha_i \\ \beta_i \end{bmatrix} \dots\dots(2)$$

In this formula the symbol μ_i denote the following matrix:

$$\mu_i = \begin{bmatrix} -l_{2i} & l_{1i} \\ -m_{2i} & m_{1i} \\ -n_{2i} & n_{1i} \end{bmatrix}$$

Column 1 in this array contains negative values of the direction cosines of the second tangential vector V_{2i} , and column 2 has the direction cosines for the first tangential vector V_{1i} . These vector are orthogonal to the vector V_{3i} and to each other.

The strain in the direction the normal to the mid-surface is assumed to be negligible ($\mathcal{E}_{z'}$)

The normal to the mid-surface of the shell element will remain normal to the mid-surface after deformation.

The displacement shape functions may be cast into the matrix form:

$$[f_i]=[f_{Ai}]+\zeta [f_{Bi}] \quad (i=1,2\dots8) \dots\dots(3)$$

where

$$[f_{Ai}] = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \end{bmatrix}, \quad f_i \ \& \quad [f_{Bi}] = \begin{bmatrix} 0 & 0 & 0 & -l_{2i} & l_{1i} \\ 0 & 0 & 0 & -m_{2i} & m_{1i} \\ 0 & 0 & 0 & -n_{2i} & n_{1i} \end{bmatrix} \frac{h_i}{2} f_i \quad \dots\dots(4)$$

The 3 X 3 Jacobian matrix required in this formulation is:

$$[J]= \begin{bmatrix} x_{,\xi} & y_{,\xi} & z_{,\xi} \\ x_{,\eta} & y_{,\eta} & z_{,\eta} \\ x_{,\zeta} & y_{,\zeta} & z_{,\zeta} \end{bmatrix} \dots\dots\dots(5)$$

where “,” indicate differentiation with respected to the symbol following the coma.

The derivatives in matrix [J] can be found from equation (1)

$$x_{,\xi} = \sum_{i=1}^8 f_{i,\xi} x_i + \sum_{i=1}^8 f_{i,\xi} \zeta \frac{h_i}{2} l_{3i}$$

$$x_{,\eta} = \sum_{i=1}^8 f_{i,\eta} x_i + \sum_{i=1}^8 f_{i,\eta} \zeta \frac{h_i}{2} l_{3i} \quad \text{And}$$

$$x_{,\zeta} = \sum_{i=1}^8 f_i \frac{h_i}{2} l_{3i}$$

so on

For this element, six types of non-zero strains exit, as follows:

$$\boldsymbol{\varepsilon} = \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ \gamma_{xy} \\ \gamma_{yz} \\ \gamma_{zx} \end{bmatrix} = \begin{bmatrix} u_{,x} \\ v_{,y} \\ w_{,z} \\ u_{,y} + v_{,x} \\ v_{,z} + w_{,y} \\ w_{,x} + u_{,z} \end{bmatrix} \dots\dots\dots(6)$$

The stress-resultant vector in the local coordinate system is,

$$\{f'\} = \{f_x f_y f_{xy} Q_y Q_x M_x M_y M_{xy}\}^T \dots\dots\dots(7)$$

Where Q_x was the shear stress per unit length in the x and y direction. The relationship between the stress resultants and the generalized strains can be stated as follows,

$$\{f'\} = [D'] \{\varepsilon'\} \dots\dots\dots(8)$$

where [D'] is the rigidity matrix. A typical rigidity matrix for flat plate shell is given

$$[D'] = \frac{Eh}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 & 0 & 0 & 0 & 0 & 0 \\ \nu & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{1-\nu}{2} & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1-\nu}{2k} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1-\nu}{2k} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{h^2}{12} & \frac{h^2\nu}{12} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{h^2\nu}{12} & \frac{h^2}{12} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{h^2(1-\nu)}{24} \end{bmatrix} \dots\dots\dots(9)$$

k =shear shape factor (assumed k=1.2) (5).

Numerical Solution

Two cases were studied, steady state and thermal stresses as follows:

Steady -State Analysis

In this section the steady state stresses and deformations were calculated, The blades considered were made of steel; and the material properties are: modules of elasticity = $207 * 10^3$ Mpa, Poisson's ratio = 0.3, Density = 7850 kg/m^3 . Other information are: Speed of rotation = 5000 r.p.m, Disk radius/width = 4, width = 0.05m. In this respect, a numerical study for a rectangular cross-section blade is given in details because it is a common model.

Comparison of the results

A comparison with experimental published results is achieved between the present work and the experimental work [1].

In this comparison, a rectangular steel blade is considered, having the following dimensions and material properties: Length = 100 mm, Width = 25mm, Thickness =3mm, Skew angle = 90° , Radius to root = 125mm, and maximum speed of rotation = 2500 r.p.m, Modulus of elasticity = $207 * 10^3$ Mpa, Poisson's ratio =0.3, Density = 7850 kg/m^3 . Strain gauges were used as transducers and the results are shown in Table 2

Effect of Rotational speed

The variation of stresses and deformations with eight speeds of rotation (250, 500, 750, 1000, 1250, 1750, and 2000 r.p.m) for three aspect

ratios (3, 4, and 5) are investigated and shown in fig. (9,10, and 11) show the variations of v-deflection, xx- stresses, yy-stresses respectively with speed of rotation. It was observed that stresses and deformations increase that when the speed of rotation increased too.

Effect of thickness

In order to investigate the thickness of blade, eight thickness (1.5, 2, 2.5, 3, 3.5, 4, 4.5 and 5 mm) and three aspect ratios (3, 4 and 5) were selected for this investigated. Figs (12, 13, and 14) show the variations of v-deflection, xx-stresses, yy-stresses respectively with thickness. It can be noted that increases of the blade thickness leads to decreases in the stresses and deformations (i.e. thin blades give greater stresses and deformations than the thick blades) and that due the higher reduction in the structural stiffness. Also, it can be seen that when the aspect ratio increases the reduction in stresses and deformations increases.

Blade Deflection due to Thermal Stresses

The blade is relatively thin, and thus the thermal gradient through the thickness of the blade is assumed negligible. Therefore, the blade is assumed to have a uniform temperature equal to the surface temperature. This simplifies the thermal stress analysis and results in a simple linear relation:[12]

$$u=\alpha TL \dots\dots\dots(10)$$

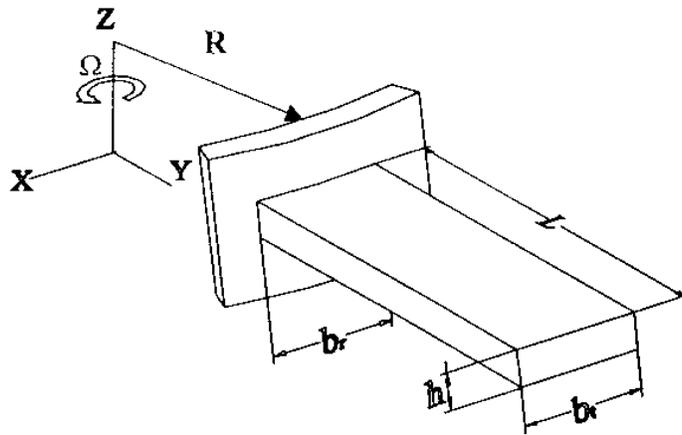
Where L= blade length (m), α = thermal expansion coefficient (1/C°) and T=Temperature (C°). Figs (15, 16, 17, and 18) show the variations of temperature u-deflection, xx-stresses, yy-stresses respectively with x-axis and Figs (19, 20, 21, and 22) show the contour of temperature, u-deflection, xx-stresses, yy-stresses respectively. It can be noted that increases

of the blade temperature leads to increase in the stresses and deformations

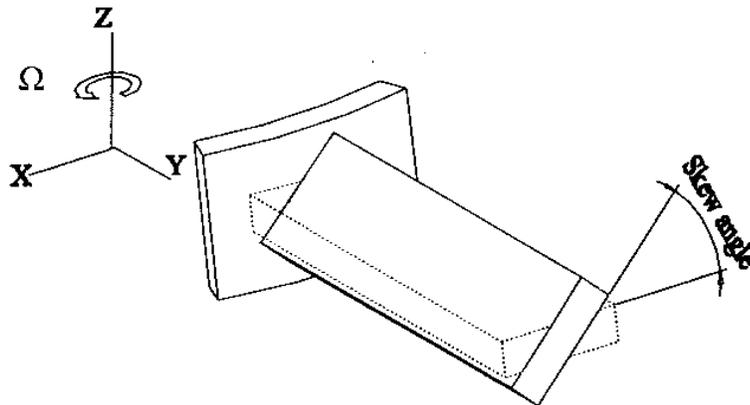
Conclusions:

The conclusions obtain from the present work can be summarized as when the increases of the blade thickness leads to decrease in the stresses and deformation, (i.e. thin blades give greater stresses and deformations than the thick blades) and

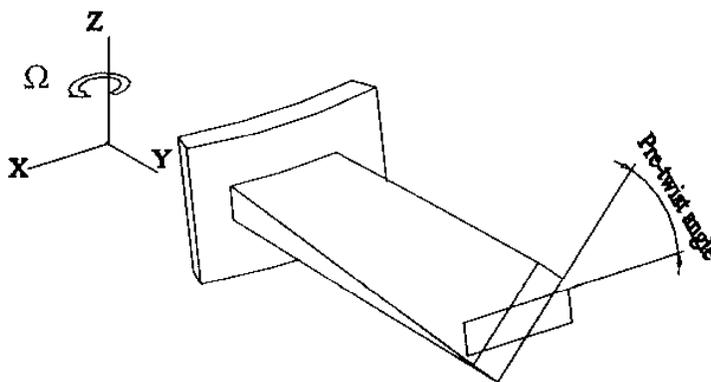
that is duo to the higher reduction in structural stiffness, the generated stresses will be increases for the blades having large aspect ratios, as well as the deformations become large, since the total mass of the structure is proportional to the aspect ratio. The increases of the blade temperature leads to increase in the stresses and deformations so, shaped film-cooling and full coverage film cooling are the most useful cooling schemes for reducing the amount of cooling air and decreasing thermal stresses.



a- General rectangular fan blade.



b- Rectangular fan blade with skew angle.



c- Rectangular fan blade with pre-twist angle.

Fig. (1) The geometry of rectangular fan blade.

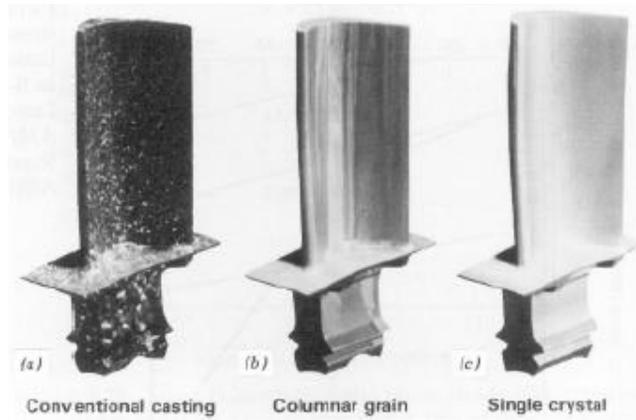


Fig. (2) Turbine blades cast to promote different grain structures [9]

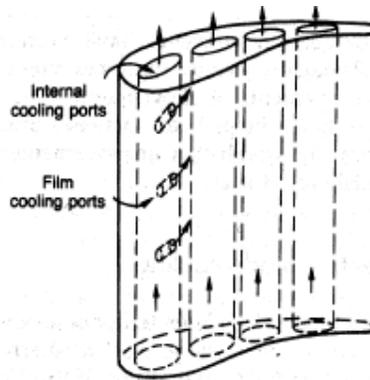


Fig. (3) Blade cooling techniques [10]

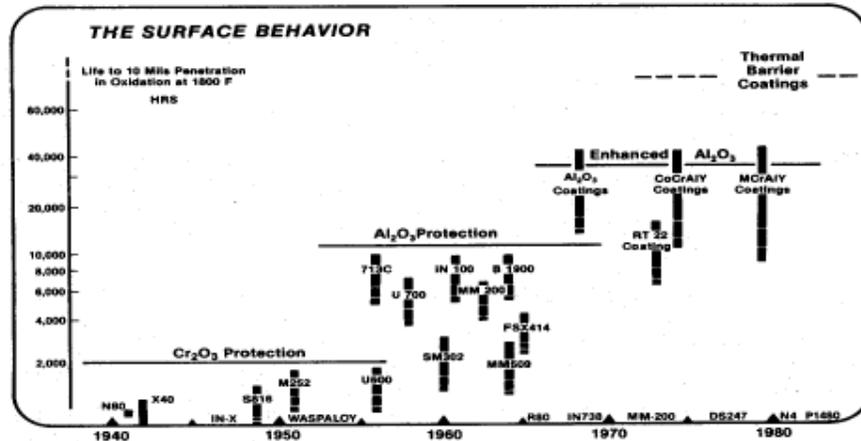


Fig. (4) History of turbine blade coating systems [11]

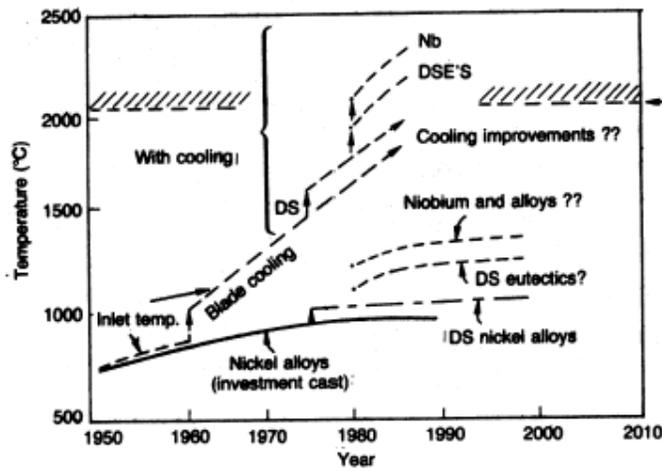


Fig. (5) Developments to increase engine-operating temperatures [10]

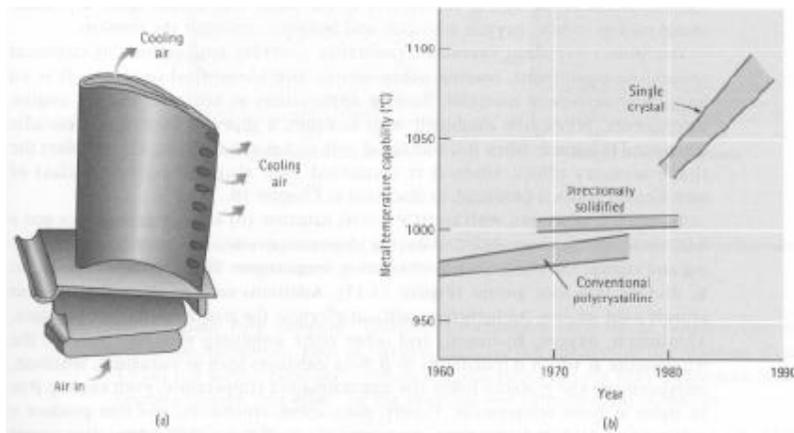


Fig. (6) Developments to increase turbine metal capability [8]

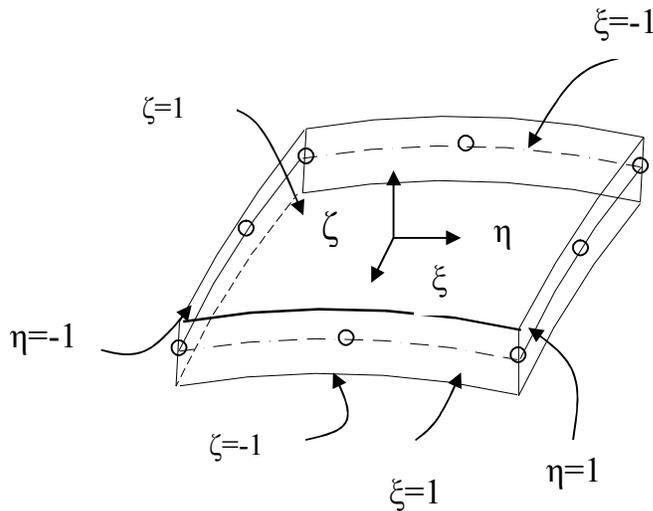


Fig. (7) Eight noded shell element

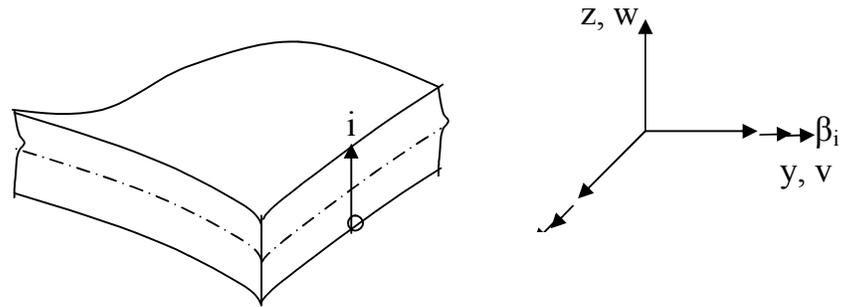


Fig. (8) Degrees of freedom at a node

Table (1) Shape Function for Midsurface Interpolation of Shell Elements

Serendipity 8-node element:
Corner nodes: $f_i = \frac{1}{4}(1 + \xi\xi_i)(1 + \eta\eta_i)(\xi\xi_i + \eta\eta_i - 1)$
Midside nodes: $f_i = \frac{1}{2}\xi_i^2(1 + \xi\xi_i)(1 - \eta^2) + \frac{1}{2}\eta_i^2(1 + \eta\eta_i)(1 - \xi^2)$

Table 2 Comparison with experimental results. (radial stress)

α°	Distance	Exp. (Mpa)	Present work (Mpa)	Ω (r.p.m)
0	0.1 L	9.1154	9.0086	2500
15	0.25 L	5.7115	5.5613	2000

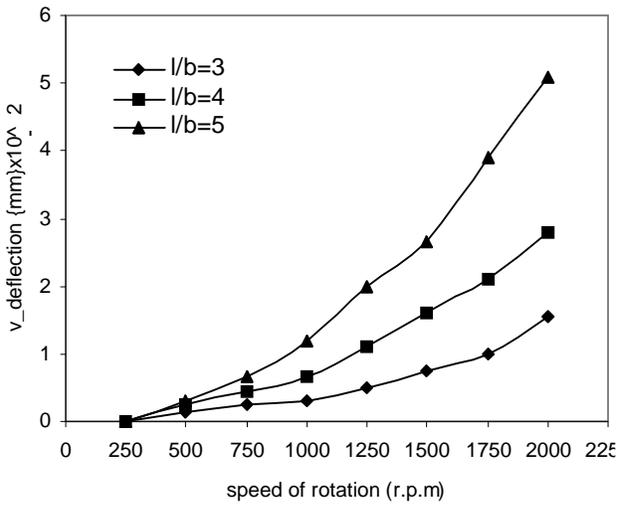


Fig. (9) Variation of v-deflection with speed of rotation

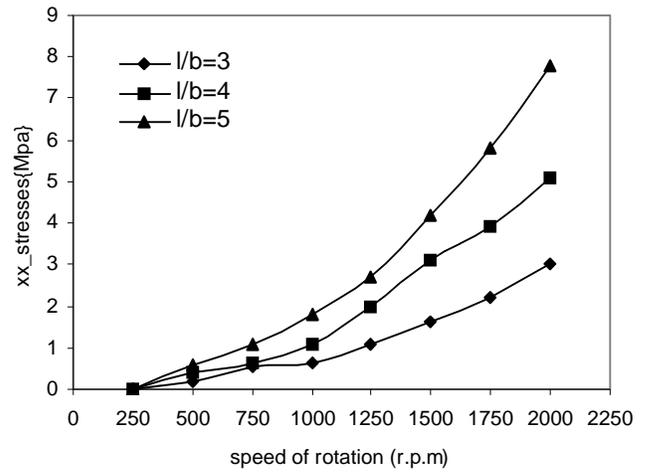


Fig. (10) Variation of xx-stresses with speed of rotation

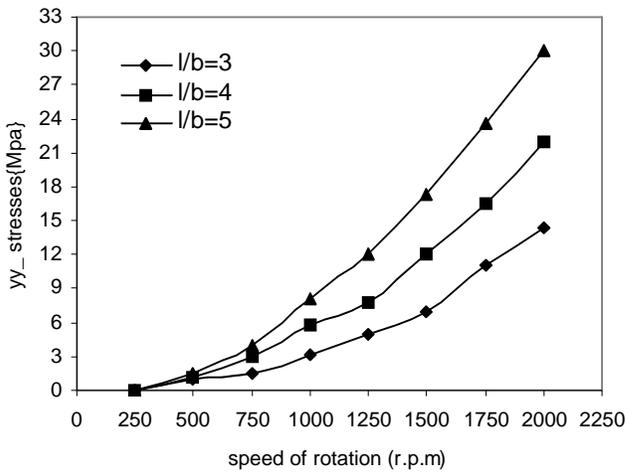


Fig. (11) Variation of yy-stresses with speed of rotation

R=0.225 m, h=0.0018 m
 Skew angle =20°
 Pre-twist angle = 12°
 Al-alloy (5052)

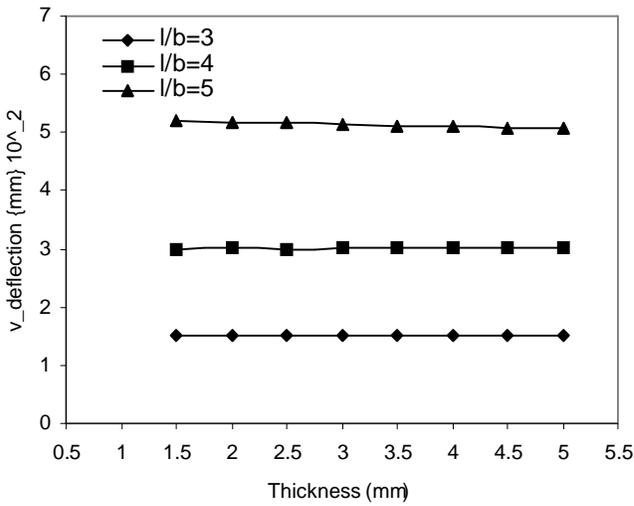


Fig. (12) Variation of v-deflection with thickness

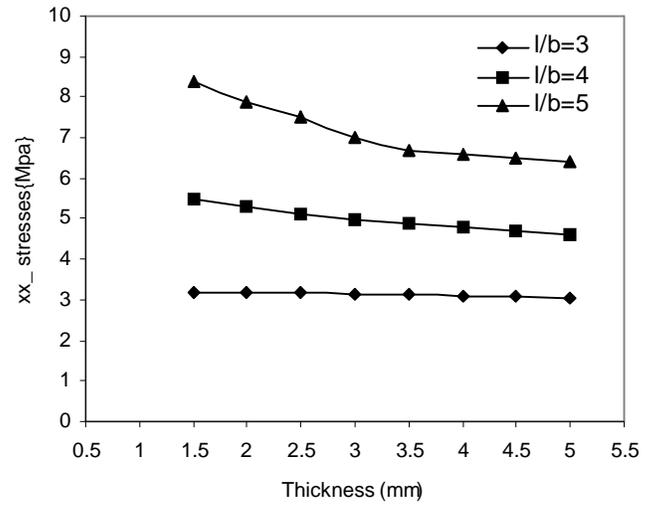


Fig. (13) Variation of xx-stresses with thickness

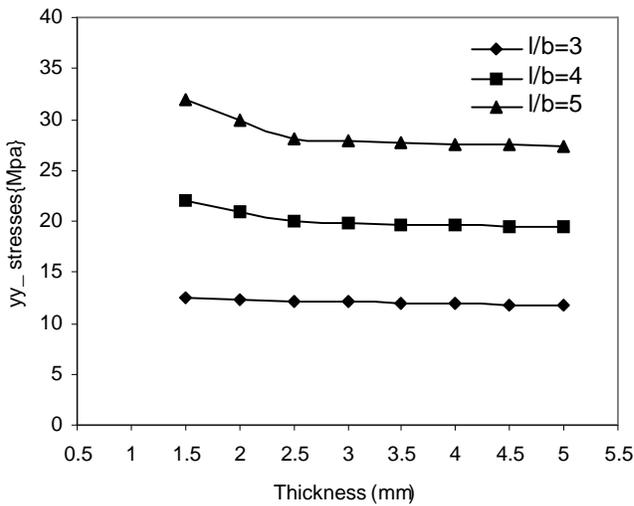


Fig. (14) Variation of yy-stresses with thickness

R=0.225 m, $\Omega=2000$
 r.p.m
 Skew angle = 20°
 Pre-twist angle = 12°
 Al-alloy (5052)

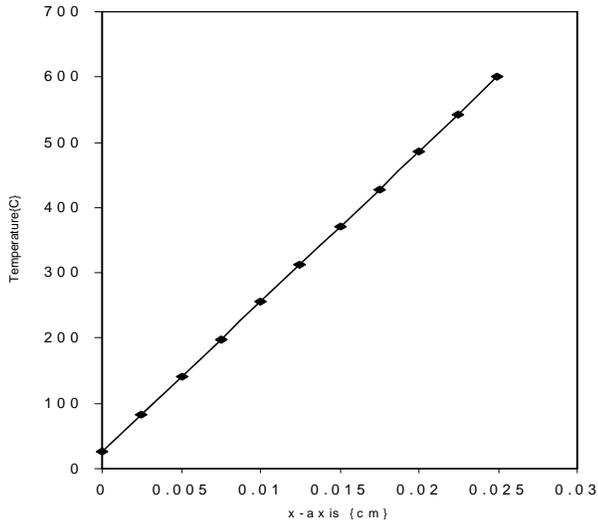


Fig. (15) Variation of temperature with x-axis

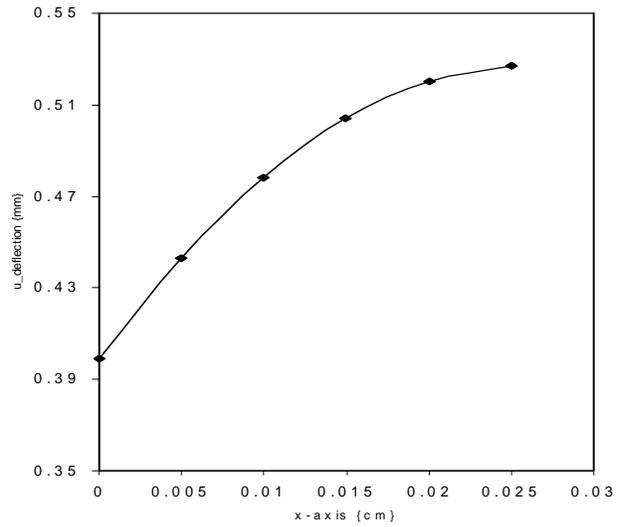


Fig. (16) Variation of u-deflection with x-axis

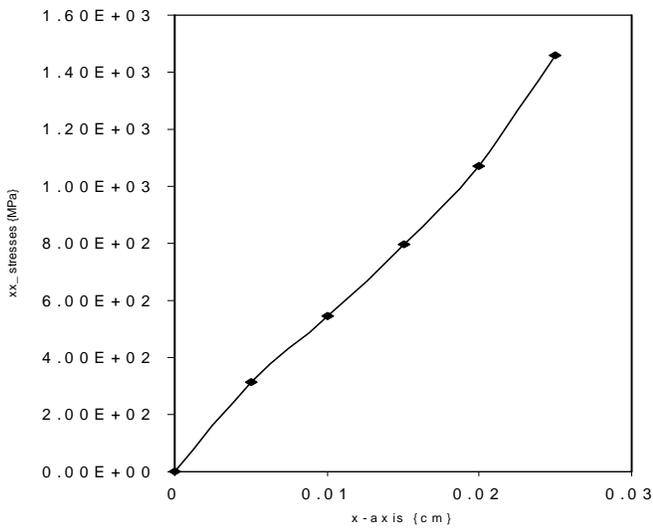


Fig. (17) Variation of xx-stresses with x-axis

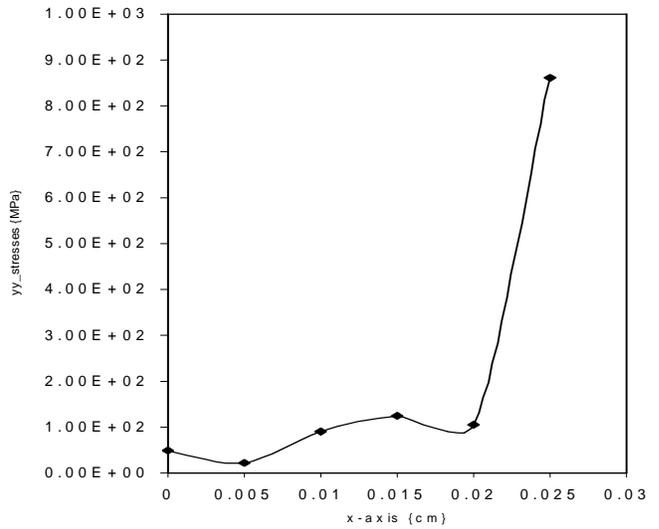


Fig. (18) Variation of yy-stresses with x-axis

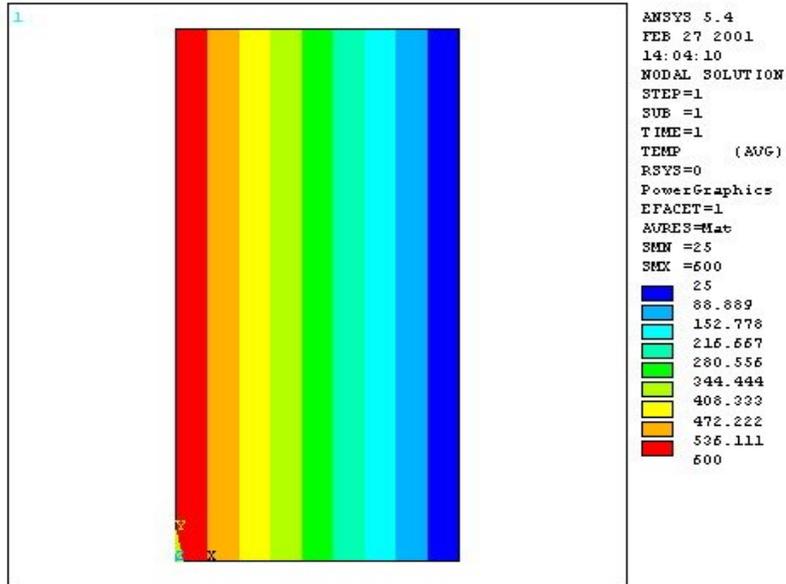


Fig. (19) Contour of temperature

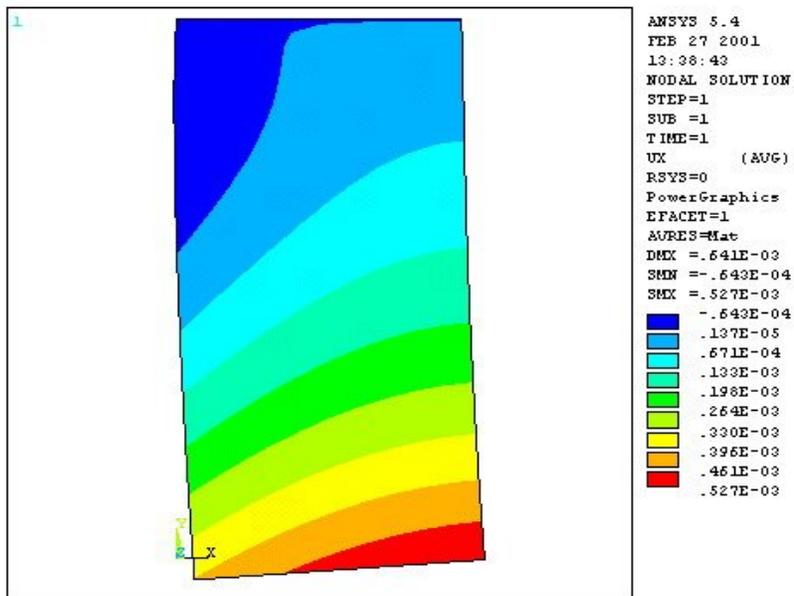


Fig. (20) Contour of u-deflection

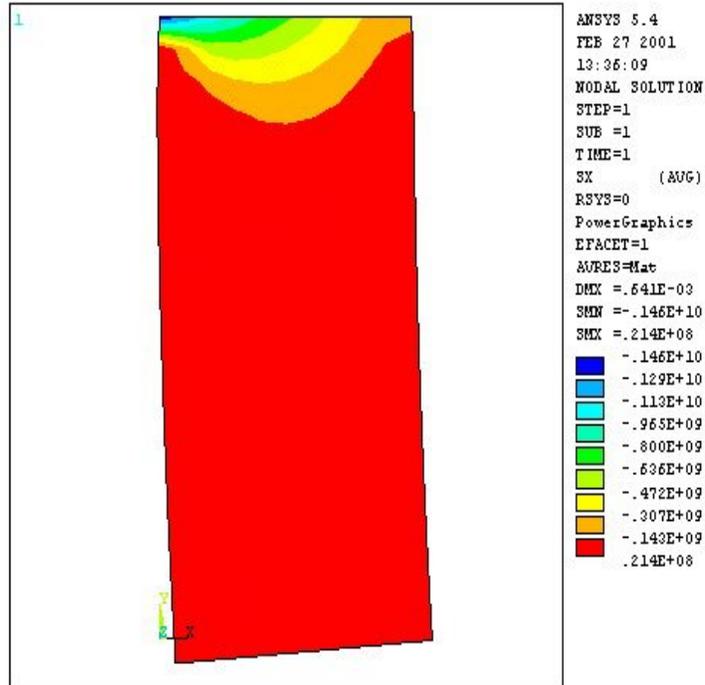


Fig. (21) Contour of xx-stresses

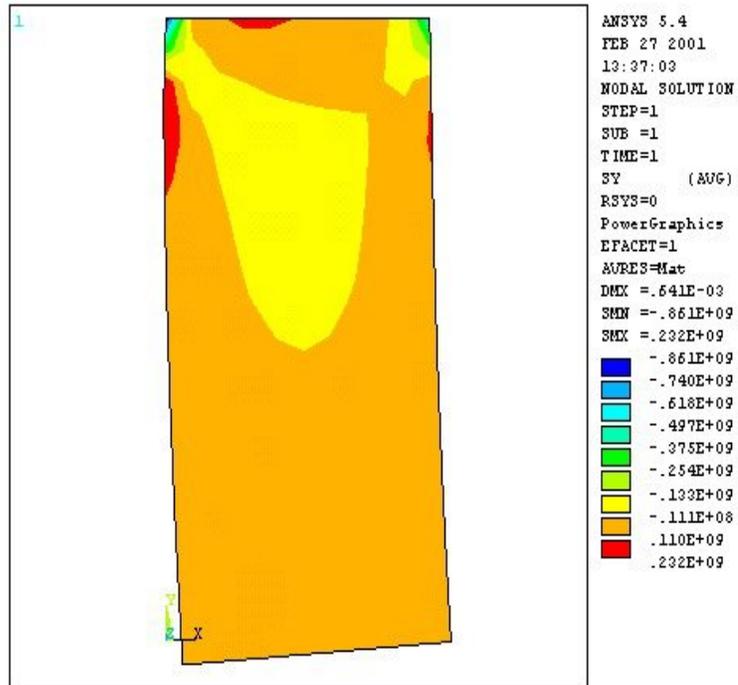


Fig. (22) Contour of yy-stresses

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<http://www.sti-tech.com/>

تأثيرات الاجهادات الميكانيكية والحرارية للمراوح الدوارة

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الخلاصة:

ألمراوح الدوارة هي من الأجزاء المهمة في التوربينان الغازية، لذلك أن التمثيل الرياضي (طريقة العناصر المحدودة) للأجهادات والتشوهات ضروري من اجل التطبيقات التصميمية لتجنب الفشل. في السنوات الأخيرة زاد أهتمام الباحثين بتأثير درجات الحرارة على الأجسام الصلبة. الهدف الرئيسي من هذه الدراسة ، دراسة الأجهادات الحرارية الناتجة من الضغط ودرجة الحرارة العاليين. وكذلك تم أحتساب الأجهادات المستقرة والتشوهات للمراوح الدوارة الناتجة من التأثيرات الميكانيكية. العديد من العوامل تم التقصي عنها في هذا البحث مثل السمك ومركز الدوران ألخ نتائج الدراسة تؤمن توصيات جيدة لتأثيرات الأجهادات الميكانيكية والحرارية للمراوح الدوارة .