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Analysis of Thermal Stresses Induced in Radiant Tubes for Heat Exchanger Applications Using Finite Element Method

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ABSTRACT

In this research, the thermal stresses in the radiant tubes were analyzed using finite element method successfully. Thermal stresses are critical factor influencing on the strength of heat exchanger tubes. Radiant tube of Steel Alloy, Super 22H was analyzed by using finite element analysis. It is investigated that thermal stresses in radiant tubes faces different temperature gradients in axial, circumferential and radial directions. There are no significant thermal stresses in axial direction when there is linear temperature gradient. In addition, it is observed that considerable thermal stresses are induced with nonlinear temperature gradient in axial direction. In order to calculate thermal stresses in circumferential direction, the temperature variations in angular position is also applied. Radial thermal stresses are calculated by applying specific temperature difference between inner and outer radii. Localized heating & hot spot in inner or outer radii and bend in tubes are also major causes of thermal stresses.

1. Introduction

Radiant tubes are widely used in modern industrial heat exchangers to enhance heat transfer. It is commonly used in many applications where isolation of the workload from the combustion environment is required. Heat exchanger has widely used applications in different industries. Analytical and finite element analysis of thermal stresses produced in radiant tubes show that radial and circumferential temperature distributions are main source of thermal stresses [1]. Thick wall tubes in transient state are also suffering thermal stresses at operating condition and its value is reduced with operation condition [2]. Numerical analysis of bayonet tube shows that large thermal stresses are produced at joint of inner fin and outer tube. To avoid these stresses there should be gap between fin and tube [3]. Liuet al [4] discussed the numerical analysis of flexible tube sheet. Rahimi et al [5] finite element analysis reveals that thermal stresses increase during cooling process and maximum stresses are produced on striking point of soot blower jet. Flores et al [6] use numerical method to find thermal stresses in tube of central receiver facing different operating temperature and weather condition. He uses three different configurations of tubes and calculates thermal stress in each case and concluded that tube with smaller diameter has lowest thermal stresses.

Ozceyhan and Altuntop [7] discusses thermal stresses in grooved tubes and effect of grooves on thermal stresses using finite element method. Maximum thermal stress is occurring near the groove part of tubes. Also thermal stresses increase with in volume flow rate. They also determine that maximum value of groove tube is twice than smooth tube for all inlet velocities. Yapici et al [8] discuss conjugate heat transfer and effective thermal stresses in SiC/SiC composite pipe with sodium/Lithium as fluid. They found that thermal stresses are maximum at inner radius of pipe, then decrease up to some distance and then increases. Asemi et al [9] used finite element method to find thermal stresses in truncated cone of functionally graded material. Analysis show that radial stress component is small near base of cone also nature of axial and tangential stresses is changed by increasing semi vertex angle of cone. The effect of thermal conductivity on thermal stresses is studied by Stasynk et al [10]. He found that effect of thermal conductivity on thermal stresses is very small for small value of internal flow. However, effect is increased if internal heat flow is increased. Wang et al [11] used CFD analysis to compute thermal stresses of tube receiver by using three different materials which are stainless steel, copper, and aluminum.

Analysis reveals that stainless steel has maximum and copper has minimum stresses. So, copper tube for tube receiver of solar irradiation is preferred. Alzaharnah et al [12] compute thermal stresses in pipe of fully developed flow with external heating. He calculated thermal stresses due to temperature field and its effect on wall thickness and pipe diameter. He concluded that stress ratio (ratio of effective stress and yield stress) increases with wall thickness. Jabbari et al [13] used analytical method to determine thermal and mechanical stresses in hollow cylinder of FGM with temperature remains constant at

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inner and outer radii but material properties change along the radius of tube. He concluded that thermal stress component is highly dependent on material properties in radial direction. Bayat et al [14] gave analytical model of residual stresses in rotating disk of different shapes and compared it with disk of uniform thickness. It was observed that residual stresses are greater in the case of parabolic and hyperbolic shape. Equivalent stresses due to thermal and centrifugal loading is also lesser than disk of uniform thickness. Gromoykand Ivanik [15] analyzed thermal stresses and displacement in heat sensitive ceramic tube in the case of convective heat transfer.

Thermal stress that depends on ratio of radii is changed considerably in case of heat sensitive material. Yapiciand Albayrak [16] showed the behavior of thermal stresses and heat conduction under different condition. He described temperature and stress distribution within cylinder under different mean velocities with uniform and non-uniform heat fluxes. He concluded that both of these parameter velocities and heat flux effect thermal stresses and temperature distribution within the pipe. So, both these parameters should be considered in design of heat exchanger. Chang et al [17] described temperature and thermal stress distribution in solar thermal absorber tubes. They concluded the heat transfer distribution in uneven in axial, radial and circumferential direction experimentally as well as numerically. It generates significant temperature and thermal stress distribution in tubes [18].

In this paper, thermal stresses due to temperature difference are studied and investigated through finite element analysis (FEM). These thermal stresses deformed the tubes which may cause to expansion and contraction. There are no significant thermal stresses in axial direction when there is linear temperature gradient. In addition, it is observed that the considerable thermal stresses are induced with nonlinear temperature gradient in axial direction. In order to calculate thermal stresses in circumferential direction, the temperature variations in angular position is also applied. Radial thermal stresses are calculated by applying specific temperature difference between inner and outer radii. Localized heating & hot spot in inner or outer radii and bend in tubes are also major causes of thermal stresses.

2. Materials and Methods

An Alloy of Steel, Super 22H is used for manufacturing of radiant tubes because of its high resistance against creep at elevated temperatures. Mechanical properties of material used in finite element simulation are explained in Table 1. Finite element analysis of radiant tube was conducted under different temperature loads, which are the primary loads experienced by tubes. The results from mathematical models were verified from finite element analysis. Valuable results are obtained from finite element analysis of thermal stresses in tubes. It will help the designer to prevent radiant tube failures and predicts life. Thermal stresses obtained due to axial temperature gradients, circumferential temperature gradients, radial temperature gradients, localized heating & hot spot, and effects of bend in the tube geometry are vital for radial tube life and failure analyses.

Following sections explain each case in detail by using FEA. Each section explains finite element model, element type, material properties, meshing, loads, boundary conditions, solver, and post processor for result viewer. A 195 mm diameter radiant tube, 8.5mm thick, has 800 mm length (1200 mm length is used to calculate thermal stresses in radial and localize hot spot case). Quarter of radiant tube was modelled with thermal element, SOLID186. This element type was used because of its applicability to problem in which thermal loadings can be applied to determine thermal stresses.

Table 1: Mechanical properties of material used in finite element simulation.

Mechanical properties	Values
Young’s Modulus, [GPa]	200
Poisson Ratio	0.3
Coefficient of thermal expansion, [1/°C]	1.5E-5
Tensile strength, [MPa]	250

3. Results and Discussion

3.1 Thermal Stresses due to Axial Temperature Distribution

Irfanand Chapman [1] developed formulae for thick cylinder of elastic material of radius c and length L with temperature vary over cross section in axial direction.

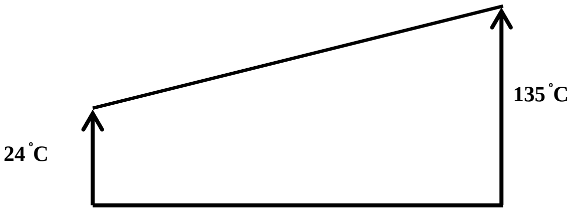
$$\partial X = \frac{E\alpha}{1-\mu^2}\eta\left[\left(\frac{C}{L^2}\right)(\lambda - 1) - \frac{3D}{\beta L^2}(\beta X + \psi)\right] \qquad (1)$$

Where

$$\sqrt[4]{\frac{3(1-\mu^2)}{h^2C^2}} \qquad (2)$$

SOLID186 is a higher order 3-D 20-node solid element that exhibits quadratic displacement behavior. The element is defined by 20 nodes having three degrees of freedom per node: translations in the nodal x, y, and z directions.

The element supports plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elastic plastic materials, and fully incompressible hyper elastic materials. Two types of loadings are considered applied in axial directions. One is linear temperature gradients with temperature varied from 24 °C to 135 °C. Second loading type is non-linear temperature distribution, such as sudden rise in temperature 24 °C higher than its surrounding sections.



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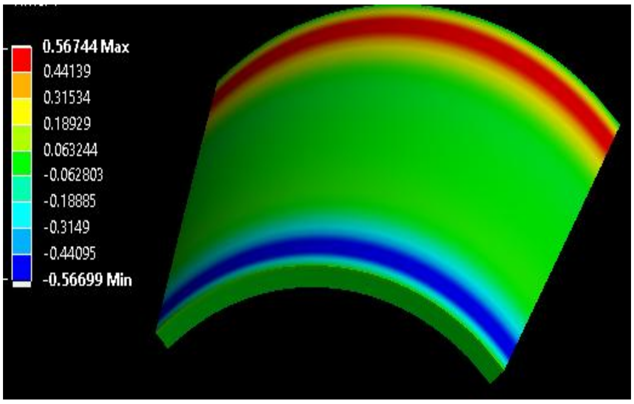

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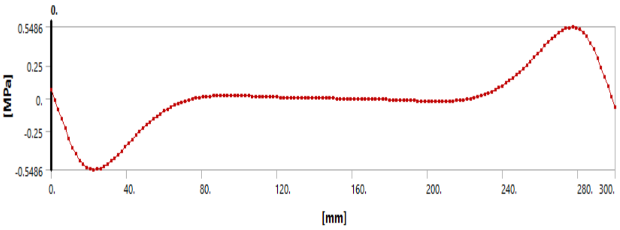


(c)

Fig. 1: (a) linear temperature in axial direction. (b) and (c) thermal stresses (MPa) generate in tube due to linear temperature distribution.

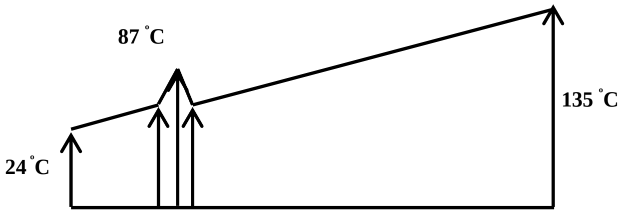
Fig. 1 describes that temperature increase across the length of tube. At start of tube, there is minimum temperature and corresponding thermal stresses are also minimum and at the other end, temperature is maximum, corresponding thermal stresses are also maximum. Fig. 2 describes that there is maximum temperature at one section of the tube and corresponding thermal stresses are also maximum.

3.2 Thermal Stresses due to Circumferential Temperature Distribution

When there is uniform heat flux across circumference, there will be no thermal stressesbut when the temperature varies along the circumference with angular position θ with function of cosine :

$$T_1(\theta) = A_0 + A_1 \cos\theta + A_2 \cos 2\theta + \quad (3)$$

$$T_2(\theta) = A_0 + A_1 \cos\theta + A_2 \cos 2\theta + \quad (4)$$



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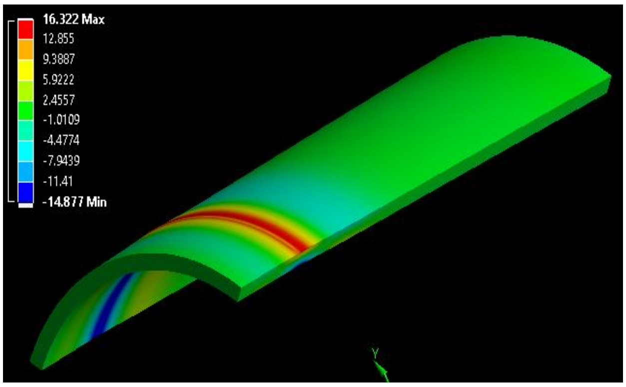

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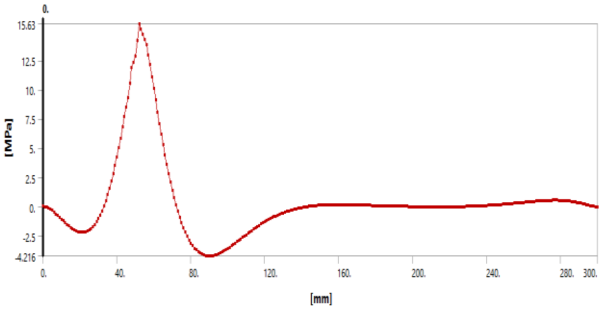

(c)

Fig. 2: (a) Nonlinear temperature distribution, (b) and (c) show thermal stresses (MPa) produce due to nonlinear temperature in axial direction of tube.

Above temperature field generates thermal stresses. Goodier [19] derive the hoop stresses generated by above field is given by

$$\sigma_\theta = Kr\cos\theta\,(\frac{a^2b^2}{r^4} + \frac{a^2+b^2}{r^2} - 3) \quad (5)$$

Where

$$K = \frac{E\alpha}{2(1-v)}(\frac{A_1}{a} + \frac{A_1}{b})(\frac{a^2b^2}{b^4 - a^4})$$

Thermal stresses generate due to above temperature field is verified using finite element analysis. In this case circumferential temperature distribution was applied with temperature difference of 15 ˚C on outer surface of radiant tube. Fig. 3 describes that when temperature difference along circumference of tube is applied, thermal stresses are generated along circumference of tube and negligible thermal stresses are generated along axial direction. Also there is maximum thermal stresses are produced on hottest side of tube.

3.3 Thermal Stresses due to Circumferential Temperature Distribution

Roark formula of thermal stress due radial temperature difference is

$$\sigma_{axial} = \Delta T \frac{E\alpha}{2(1-\nu)} \ln\left(\frac{r_o}{r_i}\right)\left(1 - \frac{2r_i^2}{r_0^2-r_i^2}\ln\left(\frac{r_o}{r_i}\right)\right) \quad (6)$$

Roark formula for thermal stress in axial radial and circumferential direction due to radial temperature difference is given by

$$\sigma_\theta = \frac{E\alpha}{(1-\nu)r^2}\left[\frac{r^2+r_i^2}{r_0^2-r_i^2}\int_{r_i}^{r_0} Trdr + \int_{r_i}^{r} Trdr - Tr^2\right] \quad (7)$$

$$\sigma_r = \frac{E\alpha}{(1-\nu)r^2}\left[\frac{r^2+r_i^2}{r_0^2-r_i^2}\int_{r_i}^{r_0} Trdr - \int_{r_i}^{r} Trdr\right] \quad (8)$$

$$\sigma_x = \frac{E\alpha}{(1-\nu)}\left[\frac{2}{r_0^2-r_i^2}\int_{r_i}^{r_0} Trdr - T\right] \quad (9)$$

ΔT= Temperature difference between inner and outerradii = 9 °C

E = Modulus of elasticity= 200GPa

α =cofficientofthermalexpansion=1.2x10^{-5}

Inner radius=195 mm

Outer radius=203.5 mm

It is found that longitudinal and hoop stresses are generated due to temperature difference of 9 °C. These stresses are verified through finite element analysis.Fig.4(a, b, c, d) describes the thermal stress variation from inner radii to ouer radii as well as along length of tubes.

3.4 Thermal Stresses due to Localized Heating and Hotspot

Kent [20] developed formulae for thermal stresses with outer radius of tube is heated to length λ at temperature T_2 where rest of tube outer portion and inner portion is maintained at temperature T_1. Maximum circumferential stress can be calculated as under;

$$\sigma_\theta = \frac{3}{4}(T_2 - T_1)/(1-\upsilon)\ \frac{\lambda}{R} \quad (10)$$

For Inside hot spot hoop and axial stress is;

$$\sigma_r = \sigma_\theta = -1/2E\alpha(T_1 - T_0) \quad (11)$$

For Outside Hot spot hoop and axial stress is;

$$\sigma_\theta = 1/2E\alpha(T_1 - T_0)ar^2/r^2 \quad (12)$$

$$\sigma_r = -1/2E\alpha(T_1 - T_0)ar^2/r^2 \quad (13)$$

In this case radial temperature distribution is applied with temperature difference of 9°C on outer face. Hot Spot has temperature of 34°C and rest of tube has uniform temperature of 24°C.

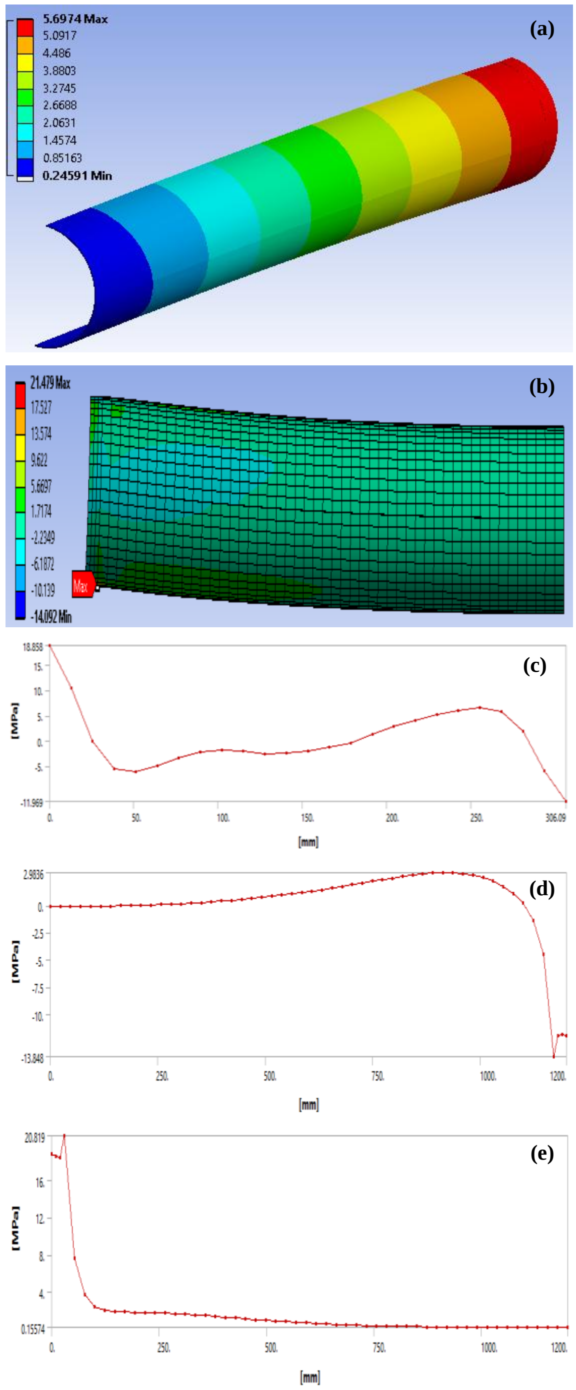

Fig. 3. (a) Deformation of pipe under circumferential temperature distribution in mm, (b) and (c) Hoop Stress (MPa) in pipe due to circumferential temperature distribution, (d) Hoop stresses on coolest side of radiant tube and (e) Hoop Stresses on hottest side of radiant tube.

Fig. 5 (a) describes the hoop stresses in radiant tube with hot spot temperature that is 10 °C higher than surrounding whereas Fig. 5 (b) decribes the varition of hoop stresses with the length of radiant tube in term of Mpa. This inrease in the hop spot region promises the higher exchnage in heat flow through the tubes, reulting increase in heat ransfer coefficient.

3.5 Effect of Bends in Radiant Tube under Linear Temperature Distribution

Thermal stresses due to bends is discussed by using finite element analysis. Linear temperature distribution along W-tube length, Symmetric boundary conditions were applied as half of radiant tube was modelled.FEA model was solved for hoop stresses. The results of this analysis are shown in Figs. 6 (a) and (b).

4. Conclusions

In the present investigation, the thermal stresses in the radiant tubes have been analysed successfully. The research has been concluded as there are no significant thermal stresses in axial direction when there is linear temperature gradient. In addition, it is observed that the considerable thermal stresses induced with nonlinear temperature gradient in axial direction. In order to calculate thermal stresses in circumferential direction, the temperature variations in angular position is also applied. Radial thermal stresses are calculated by applying specific temperature difference between inner and outer radii. Localized heating & hot spot in inner or outer radii and bend in tubes are also major causes of thermal stresses.

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