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Research Article

Heat transfer and Thermal Stress Analysis in Internal Grooved tube in comparison with bare tube by Finite element method

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Abstract

The heat exchanger is a device used for the process of heat exchange between two the fluids at different temperatures. The heat exchange process in heat exchangers can be described by the principles of conduction, convection, radiation and evaporation or condensation. Heat exchangers are useful in many engineering processes like those in refrigerating and air conditioning systems, power systems, food processing systems, and chemical reactors. Heat transfer and thermal stresses, induced by temperature differences in the internally grooved tubes of heat transfer equipment, have been analyzed numerically. The analysis has been conducted for four different kinds of tube material like aluminum (Al), cast iron (C.I), brass (Br), copper (Cu) and steel (St). Constant temperature was applied from the external surface of the tube and inlet outlet water temperatures and constant fluid to wall heat transfer coefficient was applied for the estimation of thermal profiles. These thermal results are incorporated for the estimation of Vonmises stress distribution. Energy and governing flow equations were solved using finite different schemes. Finite Element Method (FEM) was used to compute the thermal stress fields. Different materials on the thermal stress values have been discussed. As a result, maximum thermal stress occurs in the aluminum material because of high thermal conductivity of that material. Thermal stresses are directly depends on the thermal conductivity of the material. The maximum thermal stress ratio positions inside the tube have been indicated as MX for all investigated cases. Maximum thermal stress values are obtained at top surface of the groove for all the materials.

Keywords: *Heat transfer, thermal stress, grooved tube, different materials and FEM.*

1. Introduction

Heat transfer in pipe has very wide applications in industry. Heat exchangers are some of the most widely used applications. Heat exchangers are devices that allow heat transfer between two fluids that have different temperatures without physical contact between them. Heat transfer increment is generally crucial for some reasons, such as saving energy and material resources. Therefore, compact heat exchangers that weight and high level of performance, must be designed for heat exchanger industry. The heat transfer enhancement has been analyzed by various methods in order to have compact heat exchangers Kakac et al. Ligrani et al have compared heat transfer augmentation techniques. Heat transfer augmentation techniques for several geometries and boundary conditions have been examined in previous works Adachi et al, Zimparov et al , Wang et al, Yang et al. Enhanced surfaces have been successfully used in the heat transfer industry to obtain more compact and efficient units Zimparov et al. Heat transfer enhancement technology has been widely used in heat exchangers in the refrigeration, automotive and process industries among others Webb et al , Bergles et al, Herman et al state that grooved channels are frequently en countered, including on heat transfer surfaces of heat exchangers during cooling of electronic equipment and in nuclear reactor cores as well as in biomedical and aerospace applications. There has been some work on grooved surfaces in channels or tubes Ghadder et al, Brognaux et al, Graham et al , Cavallini et al , Wirtz et al , Goto et al , Park et al, Herman, Wang et al , Izumi et al. The heat transfer increase on pressure drop in the helically grooved, horizontal micro-fin tubes were investigated Wang et al. Many kinds of internally grooved tubes have been investigated to improve the performance of air conditioning heat exchanger for HFC refrigerants Wirtz et al, Greiner et al, Park et al . Herman et al have shown that grooved or fluted tubes are widely used in modern heat exchangers Graham et al expressed heat transfer and friction characteristics for single phase flow in single grooved and cross grooved micro-fin tubes. They have found that the micro-fin tubes and augmented heat transfer in a symmetrically grooved channel have been analyzed have enhanced heat transfer to as high as 1.8 times that of smooth tubes. Three dimensional flow and augmented heat

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transfer in a symmetrically grooved channel have been analyzed by Greiner et al. They have carried out their analysis for $180 \leq Re \leq 1600$ with constant temperature boundary condition. Lorenz et al also investigated the distribution of the local convection coefficient in a transverse grooved channel. The tube is heated externally while the flow passes through it. Thus a high temperature gradient is introduced across the tube. This high temperature gradient causes thermal stresses that some times play an important role in the case of tube failure. Therefore, thermal stresses must be analyzed in grooved tubes. In the tube material, thermal stress calculations are very difficult by using the analytical method due to its complicated configuration and boundary conditions Lui el al. The large majority of existing grooved tube investigations focus only on heat or mass transfer. They consider grooving geometry, grooving materials, grooving configuration, channel geometry, and other parameters. Only a few resent studies examine the thermal stress characteristics through passages with grooved surfaces. There are some researches that have been published related to thermal stresses in the tube Lui et al, Al-Zaharnah el al. The thermal stresses induced by temperature differences in the tube of heat transfer equipment were investigated by Lui el al. The thermal stresses have been analyzed in smooth tube for different boundary conditions Al-Zaharnah et al. Ozceyhan et al has been analyzed the thermal stresses by considerinf the temperature boundary conditions. L. Syam Sundar et el. has discussed the thermal stress distributions by introducing the fluid to wall heat transfer coefficient and he also estimate the thermal stresses at various pitch of the groove and he observed the maximum stresses are obtained at the P=d condition.

The objective of the present investigation is to analyze the stress distribution of the different tube materials of the internal grooved tube. For all the materials P=d condition is considered. The flow is assumed to be fully developed flow. Constant wall temperature boundary condition on the outer circumference of the tube and inlet, outlet and fluid to tube heat transfer coefficient applied inside the tube for the estimation of the thermal profiles. Thermal results are incorporated for the estimation of the stresses developed in the tube material.

From the FEM analysis, copper material tube has the higher thermal stress values when compared to the other tube materials, because of high thermal conductivity of that material.

Analysis

The numerical solution of heat conduction and other related processes have been expressed in mathematical form; generally in terms of differential equations that we shall encounter express a certain conservation principle. Each equation employs a certain physical quantity as its dependent variable and implies that there must be a balance among various factors that influence the variable. If the dependent variable is denoted by φ, the general differential equation is

$$\frac{\partial}{\partial x}\left(K_x\frac{\partial\phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_y\frac{\partial\phi}{\partial y}\right)+\frac{\partial}{\partial z}\left(k_z\frac{\partial\phi}{\partial z}\right)+\lambda\phi+Q=0 \qquad (1)$$

where λ is a function of ϕ and Q is the source term and K_x, k_y, k_z being the respective material properties in the x, y, z directions. The above equation is referred to as Helmholtz equation, the special forms of which include various scalar field problems related to heat conduction, fluid flow, seepage, electric-magnetic fields ,etc, Eq [1] on substituting the respective terms into the general 3-d equation for heat conduction.

$$\frac{\partial}{\partial x}\left(K_x\frac{\partial\phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_y\frac{\partial\phi}{\partial y}\right)+\frac{\partial}{\partial z}\left(k_z\frac{\partial\phi}{\partial z}\right)+Q=\rho Cp\frac{\partial T}{\partial t} \qquad (2)$$

where k is thermal conductivity of the material, T is temperature; Q is internal heat generation per unit volume. ρ is the density of the material; C_p is specific heat of the material; T is time. If the material considered were isotropic, then $K_x=k_y=k_z=$ constant. For steady state analysis, the above conditions make the eq [2] as

$$K\left\{\frac{\partial^2 T}{\partial_x{}^2}+\frac{\partial^2 T}{\partial_y{}^2}+\frac{\partial^2 T}{\partial_z{}^2}\right\}+Q=0 \qquad (3)$$

$$K\,\Delta^2 T = 0 \text{ (Laplace Equation)} \qquad (4)$$

Equation [3] for 2-d heat conduction

$$K\left\{\frac{\partial^2 T}{\partial_x{}^2}+\frac{\partial^2 T}{\partial_y{}^2}+\right\}+Q=0 \qquad (5)$$

Eq. [5] is the governing equation for 2-d steady state heat conduction with heat source or sink.

Derivation of finite element equations using Galerkin approach

Galerkin's method using sets of governing equations develops an integral form. It is usually presented as one of the weighted residual methods. For our discussion, let us consider a general representation of a governing equation on a region C-D-E.

$$Lu = P \qquad (6)$$

For the 2-D steady state heat conduction, the governing equation is

$$\frac{\partial}{\partial x}\left(K_x\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_y\frac{\partial T}{\partial y}\right)-Q=0 \qquad (7)$$

The exact solution seeds to satisfy eq [1] at every point x if we seek an approximate solution u it introduces an error ε(x), is called the residual:

$$\varepsilon(x)=L.u-p \qquad (8)$$

The approximate methods revolve around setting the residual to weighting function w_i to zero.

$$\int_v W_i(L.u-p)dv=0 \text{ For i=1 to n} \qquad (9)$$

The choice of the waiting equation W_i leads to various approximation methods in the Galerkin method the waiting function W_i is chosen from the basis functions used for constructing u. Let u represented by

$$u=\sum_{i=1}^{n}Q_i.G_i$$

Where G_i, i = 1 ton are basis function usually polynomials of x, y, z. Here, we chose the waiting function to be a linear combination of the basis functions G_i. Consider the arbitrary function ϕ given by

$$u = \sum_{i=1}^{n} \phi_i . G_i$$

where the coefficient ϕ_i are arbitrary except for requiring that ϕ satisfy homogeneous boundary conditions where u is prescribed.

Galerkin method can be stated as follow:

Chose basis functions G_i. Determine the coefficient Q_i in $u = \sum_{v}^{n} Q_i . G_i$ such that $\int_v W_i (L.u - p) dv = 0$ for every ϕ of the type $\phi = \sum_{i=1}^{n} \phi_{i.} G_i$ φ, where the coefficients φ_i are arbitrary except for requiring that ϕ satisfy homogeneous boundary conditions the solution of the resulting equations for Q_i then yields the approximate solution u. Applying the similar procedures for equation (2) the error introduced ε(T) by seeking an approximate solution.

$T_{(x,y)}^{(e)} = \sum_{i=1}^{r} N_i . T_i$. Where r is the number of nodes per element is,

$$\varepsilon(T) = \frac{\partial}{\partial x}\left(k_x \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_y \frac{\partial T}{\partial y}\right) - Q = 0 \quad (10)$$

Setting the above residual to zero by weighted, we get

$$\int_{\Omega(e)} W_i \left\{ \frac{\partial}{\partial x}\left(k_x \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_y \frac{\partial T}{\partial y}\right) - Q \right\} = 0 \quad (11)$$

In the galerkin approach the weight wi is equal to the shape function Ni. Substituting this change above equation becomes

$$\int_{\Omega(e)} N_i \left\{ \frac{\partial}{\partial x}\left(k_x \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_y \frac{\partial T}{\partial y}\right) - Q \right\} = 0 \quad (12)$$

In order to reduce the requirement on continuity on T applying following integration by parts rule to the above equation.

$$\int_{\Omega(e)} u.(\Delta.v) d\Omega^{(e)} = \int_{\Omega(e)} u.v.n d\Omega^{(e)} - \int_{\Omega(e)} u.v.n d\Omega^{(e)}$$

Where $u = N_i, n = n_x i + n_y j$

$$v = \left(k_x \frac{\partial T}{\partial x}\right) i + \left(k_y \frac{\partial T}{\partial y}\right) j = 0$$

we obtain,

$$\int_{u(e)} N_i \left[k_x \frac{\partial T}{\partial x} n_x + k_y \frac{\partial T}{\partial y} n_y \right] d\tau^{(e)} - \int_{u(e)} \Delta N_i \left[k_x \frac{\partial T}{\partial x} i + k_y \frac{\partial T}{\partial y} j \right] d\Omega^{(e)} - \int N_i Q d\Omega^{(e)} \quad (13)$$

for isotropic material,

$$\int_{u(e)} N_i K \left[\frac{\partial T}{\partial x} n_x + k_y \frac{\partial T}{\partial y} n_y \right] d\tau^{(e)} - \int_{u(e)} \Delta N_i K \left[\frac{\partial T}{\partial x} i + \frac{\partial T}{\partial y} j \right] d\Omega^{(e)} - \int N_i Q d\Omega^{(e)} \quad (14)$$

The above equation is written in factorial form as

$$\int N_i [k \ \Delta T n] d\tau^{(e)} - \int \Delta N_i [R \Delta N_i] d\Omega^{(e)} + \int N_i Q d\Omega^{(e)} = 0 \quad (15)$$

on τ1 , T=f(x,y)

on τ2, $K \frac{\partial T}{\partial x} n_x + \frac{\partial}{\partial y} n_y + q = h(T - T_\alpha) = 0$

On substitution of the above boundary conditions into the vectorial form equation we get

$$\iint K \left\{ \frac{\partial N_i}{\partial x} \frac{\partial N_i}{\partial x} + \frac{\partial N_i}{\partial y} \frac{\partial N_i}{\partial y} \right\} dxdy \{T\} = \iint_{\Omega(e)} N_i Q dxdy - \int_{\tau 2} N_i \{q + h(T - T\alpha\} d\tau_{(e)}^2 \quad (16)$$

The LHS of the above equation is represented as

$$K_{i,j}^{(e)} = K\{T\} + K_h \{T\}, where K_h = \int_{\tau_2^{(e)}} N_i h N_i d\tau_2^{(e)}$$

And the remaining term in equation (16) will go to the RHS force matrix {fi}. This is the equation for one element. For the whole continuum all these elemental equations are assembled into a global matrix form as $[k_t]$ {T} + $[k_h]$ {T} = {F}. To solve this global matrix essential boundary condition is applied and the equations are solved using Gauss - quadrature integration formula.

Governing stress equations Fauple et al

$$\sigma_\theta = \frac{E.\alpha}{(1-\zeta) r^2} \left\{ \frac{r^2 + r_i^2}{r_0^2 - r_i^2} \int_{ri}^{r0} T(r) r.dr + \int_{ri}^{r} T(r).r.dr - T(r).r^2 \right\}$$

$$\sigma_\theta = \frac{E.\alpha}{(1-\zeta) r^2} \left\{ \frac{r^2 + r_i^2}{r_0^2 - r_i^2} \int_{ri}^{r0} T(r) r.dr + \int_{ri}^{r} T(r).r.dr - T(r).r^2 \right\}$$

$$\sigma_\theta = \frac{E.\alpha}{(1-\zeta)} \left\{ \frac{2}{r_0^2 - r_i^2} \int_{ri}^{r0} T(r) r.dr + T(r) \right\}$$

The effective stress according to Von-Mises theory Fauple et al

$$\sigma_{eff} = \left(\sigma_\theta^2 + \sigma_r^2 + \sigma_x^2 - (\sigma_\theta \sigma_r + \sigma_\theta \sigma_x + \sigma_r \sigma_x) \right)^{1/2}$$

The assumptions used in the present investigation.

- The flow is steady turbulent and two-dimensional.
- Thermal conductivity of the tube material does not change with temperature.
- The tube material is homogeneous and isotropic.
- The heat transfer coefficient does not change along the length of the tube.

Heat transfer coefficient is calculated based on the Nussult's equation of fully developed flow for three different water velocities of 0.3 m/sec, 0.4 m/sec, 0.5 m/sec. Calculations have been performed for four different grooving pitch values.

The physical values of the tube material (Commercial steel)

Length L=0.072m r_i =0.009m

r_o=0.011m T_{inlet} = 300K T_{outlet} = 450K t = 2mm.

Table: 1 Thermo-mechanical properties of the different tube materials.

	Mat.	**(ρ)Kg/m³**	**(α)1/K**	**CpJ/Kg K**	**(K)W/m K**	**E (Gpa)**
1	Al	2707	22.5e-6	896	204.2	70
2	C.I	7272	17.03e-6	420	52.0	205
3	Br	8522	20e-6	385	25.9	110
4	Cu	8954	17e-6	383	386	110
5	St	7833	11.7e-6	465	53.6	210

2. Mathematical Modeling

Figure 1a represents the investigated geometry. In heat transfer, two things to be considered, first one is conduction heat transfer takes place inside the tube and second one is convection heat transfer takes place from the inner surface of the tube to the fluid. Commercial steel material is considered as tube material. The temperature distribution of the inside tube is solved by the two-dimensional heat conduction equation. The analysis is based on the steady state, two-dimensional continuity, energy and momentum equations.

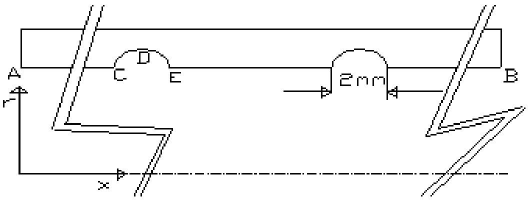


Fig: 1 (a). Physical coordinates and boundary conditions of the internally grooved

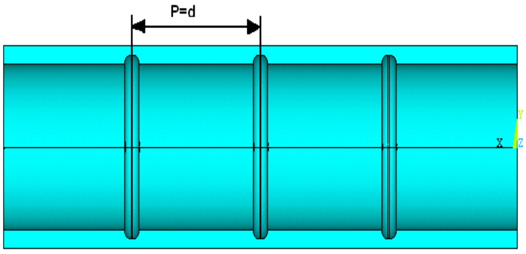
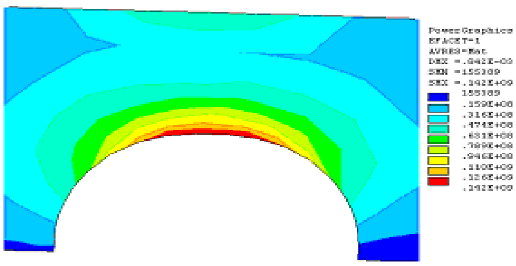
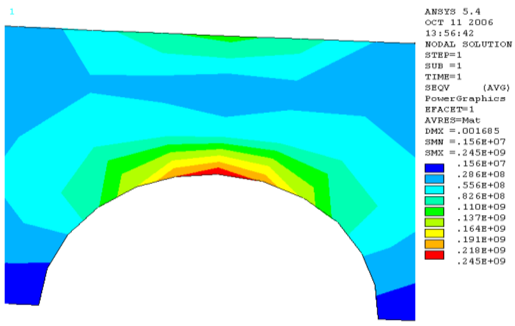
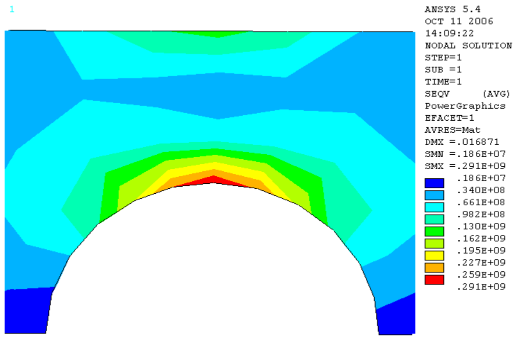


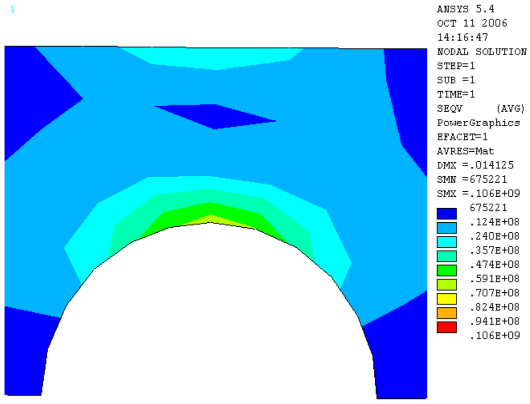
Fig: 1(b). Mechanical model developed in the ANSYS software

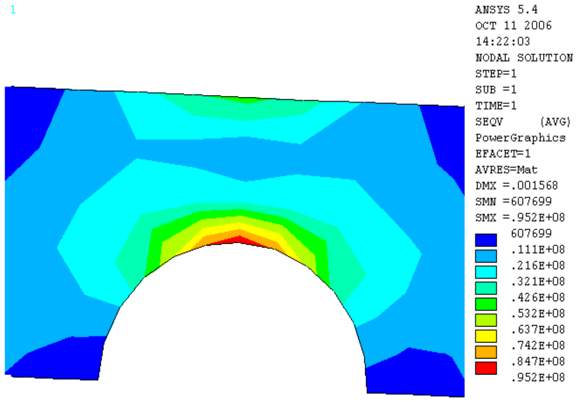
3. Results & Discussion











4. Conclusions

Temperature and thermal stress distributions inside the tubes for three different velocities and four different grooving distances are presented. Thermal stresses in smooth tubes have been analyzed for comparison tubes with grooved surfaces. The maximum thermal stresses occur near the grooved parts of the tubes for all water inlet velocities because of the higher temperature gradient, thermal stress increases when the velocity of water increases. Maximum thermal stress ratios has occur in p = d cases for 0.5m/sec water inlet velocity. The maximum thermal stress inside the tube depends on the distance between two groves, and the flow of the fluid. The types of grooves and the distance between them must be considered to avoid thermal stress effects inside the tubes, and surface modifications must be applied to surface where the maximum thermal stresses occur. This study provides ideas on the design of new heat exchangers using internal grooved surface for heat transfer augmentation.

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