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The Influence of Media Properties, Geometric and Operational parameters on the Thermal Performance of Bilayered Composite Cylinder

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Abstract

This paper presents the investigation of the conjugate heat transfer in a fluid conveying bilayered composite cylindrical pipe subjected to external convection. The problem is considered a two-dimensional problem in each of the three media. A finite difference scheme is employed to discretize the differential equations for the three media and conjugated at the interfaces. Codes were written in C-language which provided a fast solution to the models. The effects of the operational parameters (Peclet and Biot numbers), the media properties (pipe-to-fluid and laminate-to-pipe thermal conductivity ratios) and the geometric parameter (laminate-to-pipe thickness ratio) were investigated on the thermal performance – the Nusselt number of the composite pipe. These parameters and properties were found to influence the thermal performance to varying degrees significantly.

Keywords: Biot number; Peclet number; pipe-to-fluid thermal conductivity ratio; laminate-to-pipe thermal conductivity ratio; Nusselt number

1. Introduction

Thermal management is a very crucial subject in heat transfer which has a very wide application in electronic cooling, micro-electro-mechanical systems (MEMS), crude oil transportation in extreme conditions such as in high-temperature offshore fields, nuclear energy, aerospace, chemical and process engineering. Laminate composite

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structures have been considered in numerous engineering applications for their design flexibility, increased strength-to-weight ratio, stability under thermal loading, corrosion resistance, high-fatigue strength and impact resistance.

Comprehensive literature can be found in [Refs 1, 2]. Gufan et al. [3] in their study on the thermal performance of a structure containing three media considered a one-dimensional problem. The composite materials were subjected to the Stefan-Boltzmann law at the interface to prove the global unique existence in time of the classical solution. On conjugate analysis in the multilayered composite pipe, Alves et al. [4] studied the heat transfer in the solid media as a one-dimensional problem in the radii coordinate and the conveyed fluid as a one-dimensional case in the axial coordinate. The transport analysis was solved by writing an algorithm in FORTRAN 90 environment while the structural analysis was handled by computational fluid dynamics (CFD) software – ANSYS version 11. Su et al. [5, 6] studied the transient conjugate heat transfer in multi-layered composite cylinders with heat generation with specific application to the production of oil and gas in deep and ultra-deep offshore environment. The problem also was considered as a one-dimensional case. The focus of their numerical study was the effect of composite on the steady and transient state of the fluid bulk temperature.

In the present study, flow in bilayered composite cylindrical pipe involving convective boundaries is conceived as a two-dimensional conjugate heat transfer problem that captures low Reynold number, and low Prandtl number flows.

2. Problem formulation

The schematic diagram of the single regional composite circular pipe and the coordinate system considered is shown in Fig. 1. The composite medium consists of 2 - concentric cylindrical layers. Each layer is considered to be homogeneous, isotropic and with constant thermal properties. The physical problem investigates a fluid with temperature T_f in laminar flow with inlet velocity u_{in} flowing into a composite circular pipe comprising two layers in perfect thermal contact. The finitely long structure has an inner diameter of d_i, the thickness of the first layer is (r_1 - r_i) and that of the second layer is ($r_o - r_1$). The structure is surrounded by ambient temperatureT_∞. The temperature of the fluid at entry is T_{in} and uniform. The thermal condition of the laminate at x = 0 and x = L are assumed to be adiabatic. Heat is transferred from the flowing fluid into the first layer and between the layers by conduction and from the second layer and the environment by convection. Assumptions adopted are as follows: (1) the flow is steady and the fluid is incompressible; (2) the radiative heat transfer, body and electrostatic forces, are negligible; (3) negligible heat generation due to viscous dissipation occurs; and (4) there is no slip flow and no temperature jump.

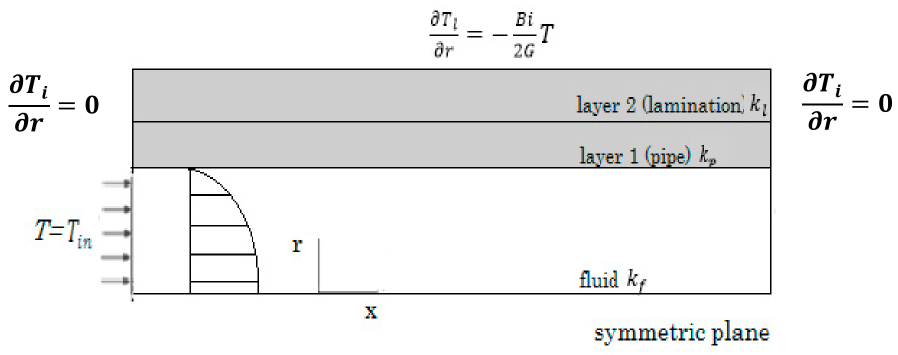

Fig. 1: Schematic representation of the physical problem.

The governing differential equations and boundary conditions in non-dimensional form are as listed below

In the fluid region

$$\frac{\partial}{r^* \partial r^*}\left(r^* \frac{\partial \theta_f}{\partial r^*}\right) + \frac{1}{\mathrm{Pe}^2}\left(\frac{\partial^2 \theta_f}{\partial x^{*2}}\right) - (1 - r^{*2})\frac{\partial \theta_f}{\partial x^*} = 0 \tag{1}$$

The above equation is solved with the following boundary equations

at $x^* = 0, \quad 0 \le r^* \le 1, \quad \theta_f = 1$ (2a)

at $x^* = 1, \quad 0 \le r^* \le 1, \quad \frac{\partial \theta_f}{\partial r^*} = 0$ (2b)

at $r^* = 0, \quad 0 \le x^* \le 1, \quad \frac{\partial \theta_f}{\partial r^*} = 0$ (2c)

At $r^* = 1, \quad 0 \le x^* \le 1, \quad \frac{\partial \theta_p}{\partial r^*} = \frac{1}{k_{pf}} \frac{\partial \theta_f}{\partial r^*}$ (2d)

$\theta_p = \theta_f$ (2e)

In the pipe and laminate regions

$$\frac{\partial}{r^* \partial r^*}\left(r^* \frac{\partial \theta_i}{\partial r^*}\right) + \frac{1}{\mathrm{Pe}^2}\left(\frac{\partial^2 \theta_i}{\partial x^{*2}}\right) = 0 \tag{3}$$

Where $i = p, l$ represent the first (pipe) and the second (laminate) layers.
The above equation is used to solve in first and second layer of the bi-layered pipe using different boundary conditions as follows

at $x^* = 0, \quad 0 \le r^* \le 1, \quad \frac{\partial \theta_i}{\partial r^*} = 0$ (4a)

at $x^* = 1, \quad 0 \le r^* \le 1, \quad \frac{\partial \theta_i}{\partial r^*} = 0$ (4b)

at $r^* = 1, \quad 0 \le x^* \le 1, \quad \frac{\partial \theta_p}{\partial r^*} = \frac{1}{k_{pf}} \frac{\partial \theta_f}{\partial r^*}$ (4c)

$\theta_p = \theta_f$ (4d)

at $r^* = 1 + t_p, \quad 0 \le x^* \le 1, \quad \frac{\partial \theta_l}{\partial r^*} = \frac{1}{k_{lp}} \frac{\partial \theta_p}{\partial r^*}$ (4e)

$\theta_p = \theta_l$ (4f)

at $r^* = 1 + +t_p + t_l, \ 0 \le x^* \le 1, \quad \frac{\partial \theta_l}{\partial r^*} = -\frac{Bi}{2G}\theta_l$ (4g)

where $G = 1 + +t_p + t_l$

The dimensionless variables are defined below

$\theta = \frac{T - T_\infty}{T_{in} - T_\infty}$; Pe = $Re*Pr$= $\frac{2r_i u_m \rho_f c_{pf}}{k_f}$; $Bi = \frac{h_0 r_0}{k_l}$; $k_{pf} = \frac{k_p}{k_f}$; $k_{lp} = \frac{k_l}{k_p}$; $t_p = \frac{r_1 - r_i}{r_i}$; $t_l = \frac{r_0 - r_1}{r_i}$; $t_{lp} = \frac{t_l}{t_p}$

$$r^* = \frac{r}{r_i};\quad x^* = \frac{x}{Pe*r_i}$$

Where, T, T_{in}, T_∞ are the instantaneous, fluid inlet and ambient temperatures respectively; r_i, r_l and r_o are the radii of the inner wall at the wall-fluid interface, pipe-laminate interface and the outer surface of the laminate, respectively. u_m is the fluid mean velocity; ρ_f and c_{pf} are the fluid density and specific heat capacity respectively; k_f, k_p and k_l are the thermal conductivities of the fluid, pipe and laminate respectively. The dimensionless parameters are noted as follows; θ is dimensionless temperature, Pe is the Péclet number, Bi is the Biot number, k_{pf} the ratio of the thermal conductivity of pipe-to-fluid, while k_{lp} is the ratio of the thermal conductivity of the laminate-to-pipe, t_l and t_p are the thickness ratios of the laminate and pipe respectively, $t_{lp}$is the thickness ratio of the laminate-to-pipe, Re and Pr are the Reynolds number and Prandtl number respectively.

The definitions for the bulk temperature, heat flux and Nusselt number respectively are given below

$$\theta_b = 4\int_0^1 r^*(1-r^{*2})\theta_f dr^*;\ q'_{wi} = -\left(\frac{\partial\theta_f}{\partial r^*}\right)_{r^*=1};\ Nu = \frac{d_i \frac{\partial T_f}{\partial r}\Big|_{r=r_i}}{T_b - T_{pi}} \tag{5a-c}$$

Where T_b, T_f and T_{pi} are the bulk, fluid and fluid-pipe interfacial temperature, θ_f and $r^*$are the dimensionless fluid temperature and dimensionless radius.

3. Solution Methodology

The governing equations for the media with the boundary conditions (Eq. 1-4) were discretized using the central difference scheme. The Gauss-Seidel iteration is used to obtain the temperature distribution with the initial set to 0.001. This assumption is based on the fact that the pipe and the ambient temperatures are assumed to be close/ the same at steady state. The iterations continue until the convergence condition is reached. The code written in C-language was developed based on the Gauss-Seidel iteration as shown in the flow chart, Fig. 2 which gives a very low computational time.

4. Results and Discussion

The experimental and numerical results of the local Nusselt number of Li et al [7] for the case of Re = 100, $t/d_i = 0.076, k_{pf} = 26.81$ was used for the validation of our code as shown in Fig 3. In the code Bi was set to zero to account for adiabatic condition and the geometric parameters were cast in dimensionless form for comparison purposes. The results of the current study are in good agreement with both the experimental and numerical results of Li et al. [7].

Presented in this section are the influence of operational parameters – Peclet and Biot numbers; media properties – pipe-to-fluid and laminate-to-pipe thermal conductivity ratios; and geometric parameter – laminate-to-pipe thickness ratio on the thermal performance of a fluid conveying bilayered composite pipe subjected to convective boundaries. The results are presented in Fig. 4a – e for a pipe thickness ratio of 0.3; the Peclet numbers of 10, 30 and 100; Biot numbers of 0, 50 and 100; k_{pf} of 0.5, 5 and 50; k_{lp} of 0.001, 0.1 and 1.0 and t_{lp} of 0.01, 0.1 and 1.0.

Fig. 4a shows that the thermal performance increases with increase in the Peclet number and that it varies between the proximities of the constant temperature and the constant heat flux conditions. Fig. 4b shows that increase in the Biot number reduces the thermal performance of the structure. The zero Biot number corresponds to the constant heat flux condition while the large value is equivalent to the constant temperature condition. It can be seen that for low Peclet Number flows especially in thickwalled pipes such as in microtubes, the effect of fluid and wall axial conductions can lower the thermal performance [7]. In this case, for adiabatic condition, Nusselt number was less

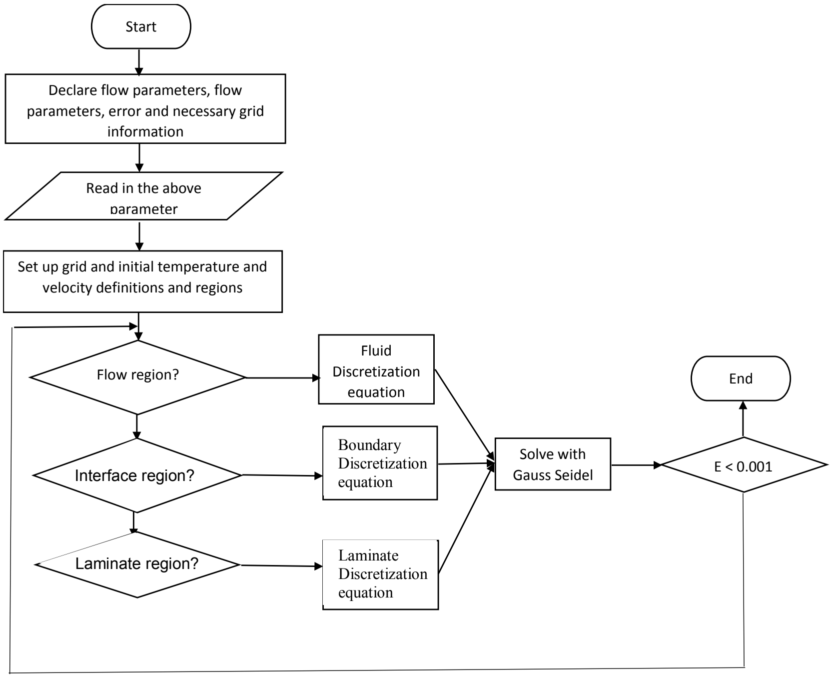

Fig. 2: The flow chart for the solution

than 4.364. The effect of the thermal properties of the pipe and fluid media are presented in Fig. 4c. These show that higher k_{pf} enhances the thermal performance. The effect of k_{lp} is presented in Fig. 4d. Increase in k_{lp} reduces the thermal performance of the structure while very low k_{lp} indicates the insulating ability of the configuration which corresponds to constant heat flux case. The effect of geometric configuration is presented in Fig. 4e. Increase in t_{lp} has shown to increase the thermal performance of the bilayered pipe. This result can serve as a useful tool for the transportation of different fluids through multilayered pipes.

5. Conclusion

The study presents the analysis of a steady conjugate heat transfer problem involving laminar flow through a bilayered composite cylindrical pipe. A simple and fast numerical procedure in C-language was used to solve the conjugated problem for laminar flow including two-dimensional fluid and wall conduction.

The effects of operational parameters such as Peclet and Biot numbers; media properties such as pipe-to-fluid and laminate-to-pipe thermal conductivity ratios and geometric parameter – the laminate-to-pipe thickness ratio are investigated on the Nusselt number. Results show that 1) increase in the Peclet number, k_{pf} and t_{lp} increase the thermal performance of the laminated structure while the increase in the Biot number and k_{lp} result in a decrease. 2) High Peclet number, zero Biot number and low k_{lp} correspond or are in close proximities of the constant heat flux condition. 3) Low Peclet number, high Biot number, low k_{pf} (=5) corresponds to or are in close proximities of the constant temperature condition.

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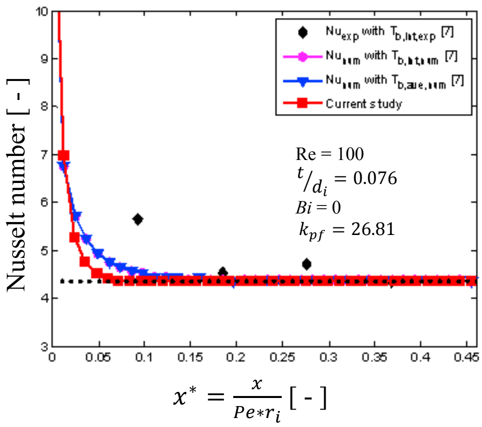

Fig. 3: Comparison of the local Nusselt number with the experimental and numerical results of Li et al. [7].

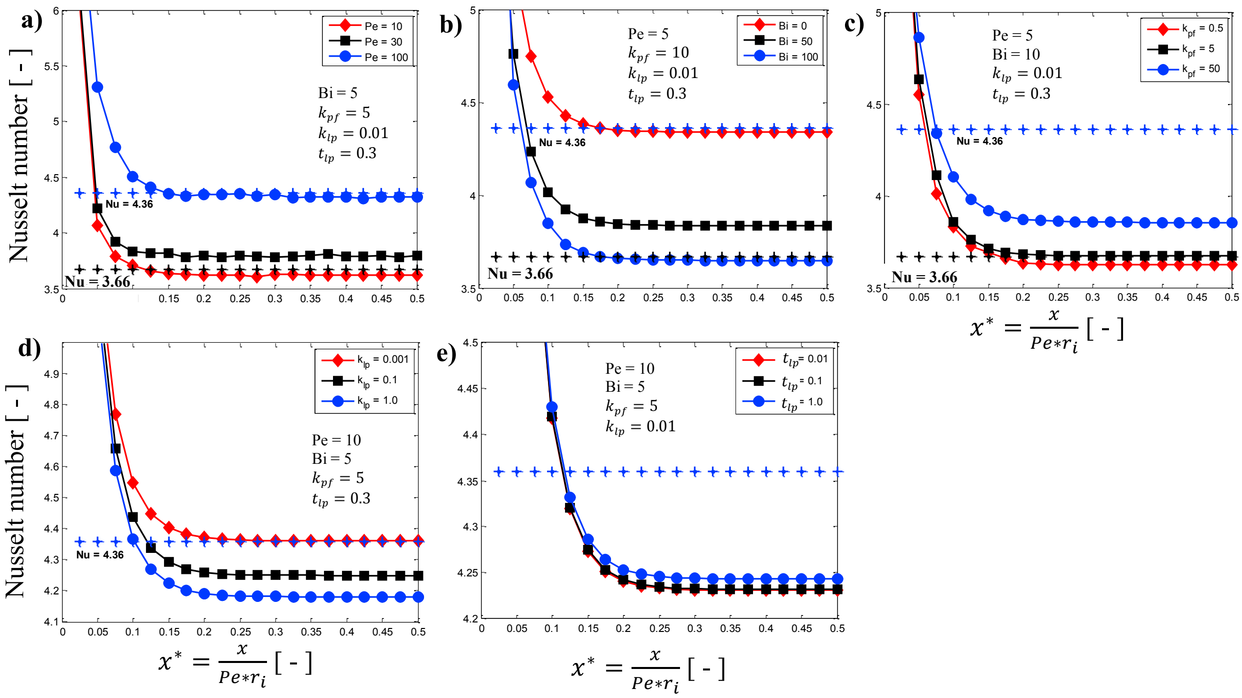

Fig 4: The effects of a) Peclet number, b) Biot number, c) pipe-to-fluid thermal conductivity ratio, d) laminate-to-pipe thermal conductivity and e) laminate-to-pipe thickness ratio on the Nusselt number.