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EFFECT OF A RADIATIVE INTERMEDIATE WALL IN AN ANNULAR-CYLINDRICAL SPACE

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Abstract- The electronic and electrical components are subjected to heat transfer phenomena during their performances. In this regard we propose to treat numerically the heat transfer improvement in a cylindrical and annular-cylindrical geometry. By combining a structured mesh (O-grid) and the finite volume method, the resolution of the convection-radiation coupling is done by using the P1 algorithm. The aim of our study is to show that the radiative effects could improve the heat exchanges thanks to the absorption of the black body, positioned at different places of the intermediate wall. We validated our simulation results with those of Le Dez and et al [1].

Keywords: Cylindrical Geometry, Radiation-Forced Convection Coupling, Finite Volumes, Radiative Model P1.

1. INTRODUCTION

Improving heat transfer is of a primary importance as it ensures a good functioning of electronic and electrical equipment's. This avoids power loss due to heat dissipation. When current flows through the system, heat is generated. This heat is even more important when a strong electric current circulates. Therefore, it is important to ensure that the system does not exceed the specific temperature allowed by the engineer. If the evacuation of heat is less than its production, the system will tend to overheat; which will lead to malfunction and even irreversible damage. Radiation-convection coupling is used for its ability to absorb heat produced by electronic components or electrical machines. It is one of the solutions for improving heat transfer, especially in areas where the temperature must be low. Under these conditions, even a slight improvement in the exchange becomes important.

Industries develop and generate new complex and scalable methods to meet their needs. T.H. Dostalizade, et al, in their work, propose a new method of learning that allows engineers to acquire new skills in order to approach problems in the field of engineering [1]. The phenomenon of heat transfer in electronics and electrical engineering was the subject of many scientific works. Among the researchers to study this phenomenon we will cite:

Vital Le Dez, et al, studied the radiative transfer in a translucent medium inside a cylindrical annulus. They propose an analytical method for the internal radiation field. They concluded that this can be fully described by geometric weights [2]. N.Y. Mammadov, et al, have analyzed and compared the thermal stabilities in walls with and without thermal insulation, under well-defined conditions. The results show that when the lower density layer is located on the outside, the temperature fluctuations on the inner surface are halved. Proper arrangement of wall structures can increase thermal stability against radiation influence [3].

Baker and Kaye, experimentally studied the case of an axial flow through an annular space, with or without rotation of an internal cylinder. In this study, the heat transfer data are represented by Reynolds and Taylor numbers through Equations which are easy to solve. The flows in the annular space with a rotating inner cylinder were defined by measuring the radial temperature distribution [4]. Convective transfer between annular space walls with rotating inner cylinders has been investigated experimentally and numerically M. Bouafia, et al. Two cases have been studied; in the first one, the surfaces of the cylinders are smooth, while in the second case, the movable wall is smooth and the other is grooved axially. They observed the appearance of vortex structures in the grooved air gap, similar to the Taylor vortices present in a smooth air gap [5]. Their results showed a very satisfactory concordance with those of Becker, et al [4].

Ki Wan Kim, et al, have done an inverse radiation-conduction analysis to estimate the thermal properties or to deduce them. They show that the coupling radiation-conduction is important in various technologies. In the cylindrical geometry studied, they found that the entire heat flux on the walls is calculated when the boundary conditions are known [6]. Jalil, et al, have studied the transfer of heat in an annular channel, composed of two concentric cylinders, where the outer is fixed and heated, while the inner is rotating and adiabatic. They took as experimental parameters, an air velocity from 2 to 6 m/s, a calorific heat flux from 600 to 1200 W/m^3 and an internal cylinder rotation velocity from 0 to 1500 rpm [7].

N. Aouled Dlala, et al, used a new method that gives an accurate formula of the angular derivative and avoids approximation to limits. This method is more precise than the discrete ordinate method. The results are obtained without any additional digital costs [8]. With the resolution of the dominant equations by the pressure correction method, A. Shaija, et al the authors have numerically studied the coupling of external radiation with natural convection in a horizontal cylindrical annulus [9]. The authors noticed that external radiation reduces convective heat transfer in the annulus. They also highlighted the symmetry of flow and thermal fields with respect to the vertical midline; as well as when the conductivity increases the temperature in the solid decreases.

Ming Jeng, et al, propose the experimental study of Taylor-Couette-Poiseuille flow inside a ribbed channel located inside a rotating cylinder. The inlet and outlet of the coolant in the channel were designed in such a manner that, the Nusselt number is minimal at the inlet and then increased in the axial direction. Heat transfer is improved by adding fins to the inner cylinder for a number of Renolds between 300 and 600 **[**10]. Lanical, et al**,** in their study, the authors aimed to understand with simplified models, the flows and heat exchanges in electrical machines. They studied a Taylor-Couette-Poiseuille flow in a hydro-generator. They performed numerical simulations and compared their results with those obtained experimentally [11].

Fenot, et al**,** studied the impact of airflow on the convective transfer in the air gap of the slotted rotor of a four-pole asynchronous motor is [12]. They found that the rotational speed of the rotor was the cause of the convective transfer. Laurent Cadet, showed that the form factor influenced the flow and heat transfer in a cavity by retaining all the dimensionless quantities associated with the convective problem [13]. By treating the radiation implicitly through a temperature field imposed on the walls, resulting from measurements in the small cavity, he observed that the decrease in the longitudinal shape ratio in a realistic configuration gives:

• An increase in the central thermal stratification
• An increase of the convective exchanges at the walls (Nu)
• An increase of the overall kinetic energy
• intensify the turbulent fluctuations
• decrease the maxima velocities

M. Jami, et al, numerically studied the phenomenon of natural convection-radiation coupling in a square cavity containing a cylinder. Only the surface radiation has been taken into consideration, and the radiative surfaces are supposed to be gray and isotropic in emission/reflection. A hybridization of the Boltzmann method and the finite difference method is used to analyze the thermal and dynamic behavior of air. They analyzed and discussed the impact of Rayleigh number and cylinder size on heat transfer and airflow [14].

A. MERABET and al, studied a 2D modeling on a square cavity filled with air, where horizontal walls are supposed to be adiabatic, while the verticals are subjected to constant temperatures. The purpose of this study is to highlight the effect of bulk radiation on natural convection by considering the effect of absorption coefficients on the dynamic field structure. This study is carried out for different radiation models (DO, DTRM, P1 and Rosseland) and for a single Rayleigh number equal to 5×105 [15].

S. Al Jabair, et al, in their study, the natural convection was simulated in a concentric annulus consisting of an inclined outer square cold chamber and a heated inner cylinder [16]. Numerical results are given for Rayleigh numbers from 103 to 106 and seven different tilt angles for the outer square boundary from 0° to -180°. They found that the aspect ratio and the Rayleigh number have their importance in the choice of the flow model and the thermal field. They also found that the angle of inclination had an impact on heat transfer, regardless of the Rayleigh number.

F. Hajji, et al, studied laminar forced convection for steady liquid flow between concentric rotating cylinders. They found that the irreversibility created by fluid friction is ignored compared to the entropy created by conduction. Furthermore, they found that the entropy production was hardly affected by the angular velocity of the inner cylinder [17]. K. Kahveci, his study concerns unstable mixed convection in a horizontal ring composed of two concentric cylinders at constant temperature. The inner cylinder is heated and rotates at a constant angular velocity. For a narrow annular space, the simulation results showed the instability of the convective flow for large values of Ra, and this instability becomes more important when the Re increases [18].

Xing Yuan, et al, propose the study of natural convection in horizontal concentric annulus of different geometries. By maintaining the interior and exterior surfaces at a constant temperature, the results show an increase in the rate of heat due to the radiation of the surfaces and the presence of corners of certain geometries. Surface radiation has an important role in heat transfer when the reference temperature is increased [19]. The radiation-convection coupling in a vertical annulus has been simulated by B.Y. Isah, et al. For the resolution of the governing Equations, they used the method of perturbations. They used the Roseland approximation to describe the radiative heat of the energy equation. They highlighted the existence of friction and Nusselt number, respectively on the surfaces, exterior for interior cylinder and interior for exterior cylinder [20].

Isayah and Basant K. Jha, studied the combined effect inside a vertical annulus, of radiation-natural convection coupling [21]. They used the perturbation method to solve the energy and momentum equations. In order to study the radiant flow described by the energy equation, they used Roseland's approximation. The combination of radiation with laminar convection in an eccentric annulus, with participating media have been studied by Kang Luo, et al. They varied the eccentricity, the Rayleigh and Reynolds numbers as well as the convection-radiation parameters. For complicated geometries, this combination is easy and has good handling [22].

R. Cherfi, et al, have numerically treated the mixed convection around a finned cylinder heated from inside [23]. They noted that the increase of R_e, and the decrease of Nu; affect the heat exchange between the outer and inner cylinders. Moreover, for a low Reynolds number, vortices appear near the fins. M.A. Medebber, et al, their work focuses on natural convection-radiation coupling in an air-filled cylinder with gray surfaces [24]. The results obtained showed that the temperature distribution, the flow pattern when R_a is high as well as the average Nu depended on surface radiation. They also found that an increase in emissivity leads to a rapid increase in average Nu. For a high emissivity, radiation plays a major role.

X. Chen, et al, investigated the radiation-convection coupling in an annular space containing foam. Two case studies were conducted; the first is that the heat flow is constant at the inner wall, while the outer one is adiabatic. In the second case, they did the reverse. They also studied the effect of the structure of the foam as well as the radius ratio of the rings. The results show that increasing the porosity and pore size promotes radiative heat transfer, whereas it decreases with increasing radiation rate. They also found that as porosity increased, the average Nusselt number decreased [25].

B. Ould Said, et al, numerically simulated the radiation-natural convection coupling in a vertical annular space containing air [26]. The important parameters in this study are R_a and ε. In order to solve the governing equations and the radiative heat equation, they used the finite volume method and the DOM method respectively. They found that increasing of R_a number and surface emissivity ε lead to an increase in the average Nu.

2. MODELING HEAT EXCHANGE IN AN ANNULAR-CYLINDRICAL SPACE

In an annular-cylindrical space, a part of the energy dissipated by the outer cylinder is lost through radiation from the outer wall of the inner cylinder. This quantity of energy cannot be neglected in the case where the ΔT temperature is low, due to the heating intensity which does not exceed four amperes. The energy dissipated by radiation is carried away by the fluid in motion, thus sweeping the wall. The quantity of heat by radiation is given as follows [27]:

$$Q_{ray} = \sigma S_i \left[\frac{1}{\frac{1}{\varepsilon_i} + \frac{S_i}{S_j} + \left(\frac{1}{\varepsilon_j} - 1 \right)} \right] \left(T_i^4 - T_j^4 \right) \tag{1}$$

Figure 1 shows the geometry of the computational domain in front and left view.

2.1. Thermal Balance

In the presence of simple geometry and transfer, the analytical method is well suited, whereas the numerical methods are more suitable for solving more complicated problems.

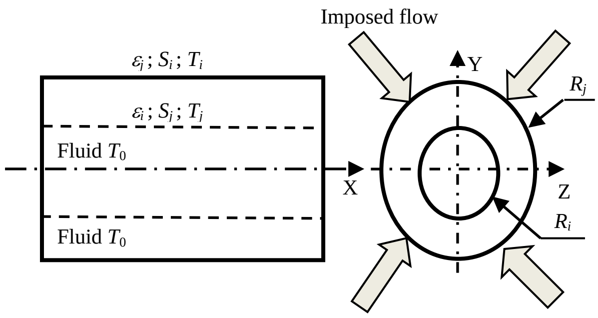


Figure 1. The computational domain

Among these methods, we have chosen that of finite volumes to discretize the geometric configuration. We chose it for its simplicity and applicability to all geometries (including complex geometries). It is also advantageous, since it:

- facilitates the linearization of energy source terms (if they are not),
- It allows to, conserve mass and momentum, in the computational domain.
- It allows easier treatment of heterogeneous media.

The objective of our study is to see how evolves the thermal field in an annular-cylindrical space. In cylindrical coordinates the heat equation is written:

$$U \frac{\delta T}{\delta x} + V_r \frac{\delta T}{\delta r} + V_\Omega \frac{\delta T}{\delta \Omega} = \alpha \left| \frac{\delta^2 T}{\delta x^2} + \frac{1}{r} \frac{\delta T}{\delta r} + \frac{\delta^2 T}{\delta r^2} + \frac{1}{r^2} \frac{\delta^2 T}{\delta \Omega^2} \right| + \frac{1}{r} \frac{\delta}{\delta r} \left(r Q_r \right) \tag{2}$$

Using the following assumptions:

- The heat exchange is stationary $\delta T/\delta t = 0$
- The conductivity $\frac{\delta^2 T}{\delta x^2} = 0$
- The flow is axisymmetric $V_r = V\Omega = 0$.

Taking into account that:

$$\frac{\delta^2}{\delta \Omega^2} = \frac{\delta}{\delta \Omega} = 0 \tag{3}$$

The Equation (1) becomes:

$$\rho c_p \left[U \frac{\delta T}{\delta x} \right] = \frac{1}{r} \frac{\delta}{\delta r} \left[K r \frac{\delta T}{\delta r} \right] + \frac{1}{r} \frac{\delta}{\delta r} \left(r Q_r \right) \tag{4}$$

where, $\frac{1}{r} \frac{\delta}{\delta r} \left(r Q_r \right) = Q^*$ represent the density of the net radiative flux. Equation (2) can be written:

$$\rho c_p U \frac{\delta T}{\delta x} = \frac{1}{r} \frac{\delta}{\delta r} \left[r \left[k \frac{\delta T}{\delta r} \right] \right] + Q^* \tag{5}$$

with, Q^* to be determined using Ostrgradski's theorem. By taking:

$$\varepsilon_r = \frac{\sigma}{\frac{1}{\varepsilon_2} + \left[\frac{1}{\varepsilon_1} - 1 \right]} \tag{6}$$

We will have:

$$\varepsilon_r \left(T_p^4 - T_s^4 \right) 2 \pi r_m L = Q^* \pi \left(r_{j\,s+1}^2 - r_{j\,s\text{-}1}^2 \right) L \tag{7}$$

$$r_m = \frac{r_{j\,s+1} - r_{j\,s-1}}{2} \tag{8}$$

By substitution we will have:

$$Q^* = \frac{\varepsilon_r \left(T_p^4 - T_s^4\right)}{r_{j\,s+1} - r_{j\,s-1}} \tag{9}$$

2.2. Boundary Conditions

Our boundary conditions are as the following.
Taking as dimensionless variables:

$$\theta = \frac{T}{T_0};\ R = \frac{r}{r_p};\ Z = \frac{4z}{P_e 2r_p};\ U = \frac{U}{U_0};\ Q^* = Q_1\beta$$

By substituting these variables in Equation (4) we will have

$$\rho c_p U \frac{2T_0}{P_e r_p}\frac{\delta\theta}{\delta z} = \frac{kT_0}{r_p^2}\frac{\delta^2\theta}{\delta R^2} + \frac{kT_0}{r_p^2}\frac{1}{r}\frac{\delta\theta}{\delta R} + \\ +Q_1\beta = \frac{kT_0}{r_p^2}\left[\frac{\delta^2\theta}{\delta R^2} + \frac{1}{r}\frac{\delta\theta}{\delta R} + \frac{r_p^2}{kT_0}Q_1\beta\right] \tag{10}$$

Taking the Peclet number $P_e = P_r R_e = \frac{U_0 \rho\, c_p}{k} 2r_p$ the Equation (4) becomes:

$$\rho c_p U \frac{2T_0 k}{U_0 \rho\, c_p 2r_p}\frac{1}{r}\frac{\delta\theta}{\delta z} = \frac{U}{U_0}\frac{kT_0}{r_p^2}\frac{\delta\theta}{\delta z} = \\ = U^* \frac{kT_0}{r_p^2}\frac{\delta\theta}{\delta z} \tag{11}$$

$$U\frac{kT_0}{r_p^2}\frac{\delta\theta}{\delta z} = \frac{kT_0}{r_p^2}\left[\frac{\delta^2\theta}{\delta R^2} + \frac{1}{r}\frac{\delta\theta}{\delta R} + \frac{r_p^2}{kT_0}Q_1\beta\right] \tag{12}$$

After simplification, the Equation (5) becomes equal to:

$$U\frac{\delta\theta}{\delta z} = \frac{\delta^2\theta}{\delta R^2} + \frac{1}{r}\frac{\delta\theta}{\delta R} + Q_1 \tag{13}$$

Knowing that

$$Q_1 = \frac{Q^*}{\beta} = \frac{\delta Q_r}{\delta R} = \int_r \left(T_p^4 - T_s^4\right)\frac{1}{\left(r_{j\,s+1} - r_{j\,s-1}\right)}\frac{r_p^2}{kT_0} \tag{14}$$

And $\beta = \frac{r_p^2}{kT_0}$ then,

$$Q_1 = \frac{\int_r r_p^2}{kT_0}\left(T_p^4 - T_s^4\right)\frac{1}{\left(r_{j\,s+1} - r_{j\,s-1}\right)} = \\ = \beta\varepsilon_r\left(T_p^4 - T_s^4\right)\frac{1}{\left(r_{j\,s+1} - r_{j\,s-1}\right)} \tag{15}$$

2.3. Mesh Used

We opted for a refined mesh on the R direction and uniform according to the length of the cylinder (Figure 2). This mesh helps to discretize the equations and facilitates the simulation. The steps of iterations ΔR and ΔX are taken different.

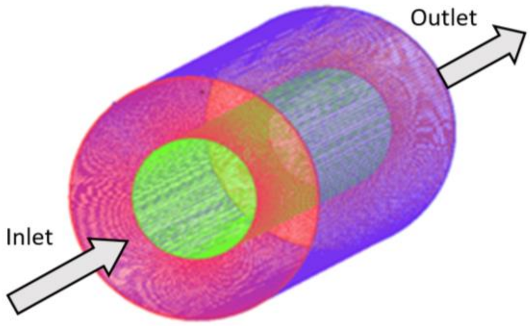


Figure 2. The mesh representation

3. THE MODEL VALIDATION

We validate our model with the work of [28]. There are two cases. In the first case we took as radius ratio r=0.5 and K=1000 m^{-1} (Figure 3). In the second case we modified the value of the radius ratio r=0.2 and maintained that of K=1000 m^{-1} (Figure 4).

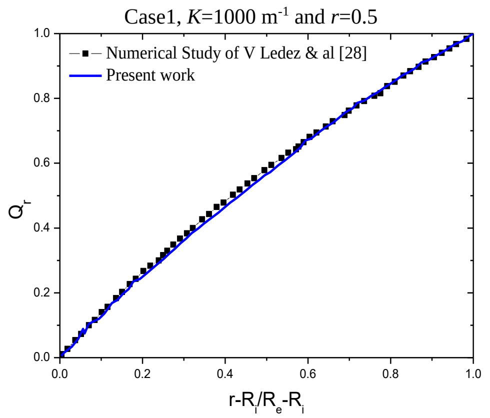


Figure 3. The radiative flux density variation Q_r according to the radius r (case 1)

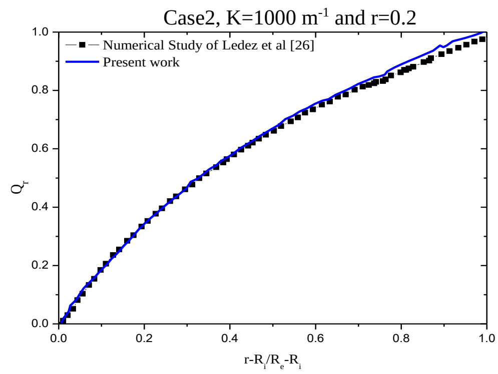


Figure 4. The radiative flux density variation Q_r according to the radius r (case 2)

In both cases, we notice that there is a good agreement of our results with those obtained by [28]. We can clearly see that the two curves have the same appearance and the error between them is very small.

4. RESULTS AND DISCUSSION

4.1. Case of the Cylinder Alone

For a ratio, $N_c = r_s / r_p = 1$, in this case we will have pure convection.

4.2. Case of the Annular-Cylindrical Space

For temperatures not exceeding 400 °K, we studied the improvement of heat exchange in an annular-cylindrical space by the absorption effects of the supposedly black radiative plate and the location of the cylinder in the flow. The radius ratio N_c= 0.4, 0.5 and 0.6 was varied with different emissivity, in order to find the adequate configuration which allows good heat exchange.

4.2.1. Velocity Profile

Figures 5 and 6 show the Velocity profile for radius ratios N_c = 0.4 and 0.6 with the same Peclet number.

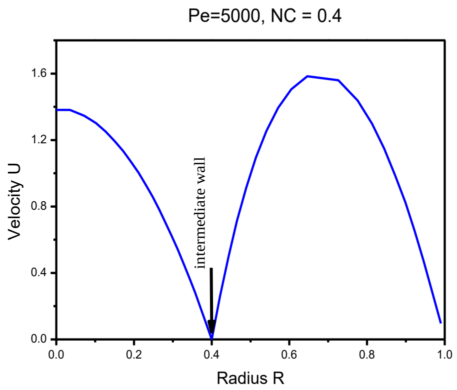


Figure 5. Velocity Profile according to R for N_c = 0.4

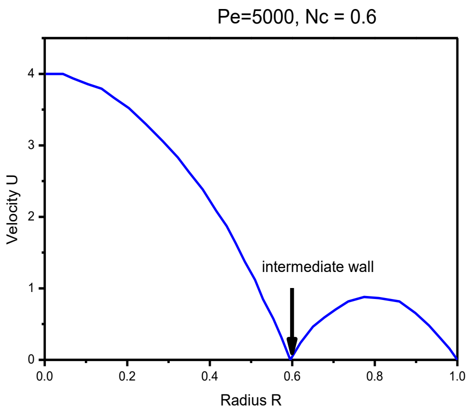


Figure 6. Velocity Profile according to R for N_c = 0.6

For the same Peclet number, by varying the radius ratio from 0.4 to 0.6, the annular space is reduced, then a very marked increase in velocity near the wall, this induces the presence of strong gradients and consequently a good transfer of heat.

4.2.2. Temperature Profile

Figures 7 and 8 show the temperature evolution of the wall, along the radius R for different emissivity ε and at different length positions.

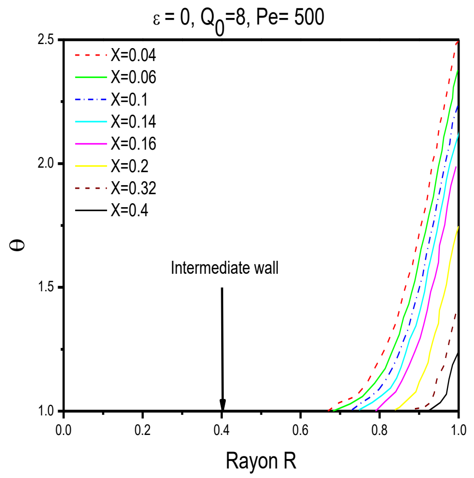


Figure 7. temperature profile according to R

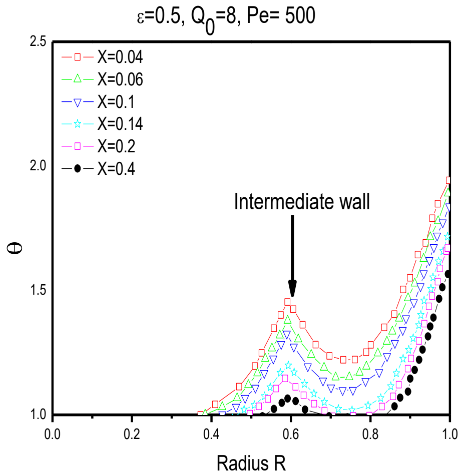
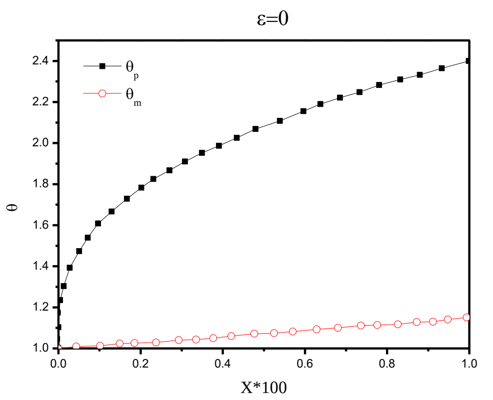


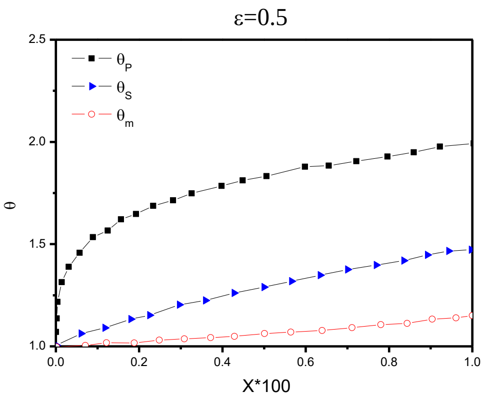
Figure 8. temperature profile according to R

From Figures 7 and 8, we notice that the temperature of the walls and partitions decreases significantly when the emissivity is between 0 and 0.8. We also note that for radius ratios of 0.4 and 0.6, the temperature distributions have the same appearance without the emissivity being zero. As emissivity increases, so does the temperature of walls, outer and separation. Near the middle wall, the increase of temperature is due to the radiative passivity of that wall. When the emissivity increases the intermediate wall absorbs and strongly emits. The cylinder inside the flow absorbs heat by convection and radiation, and since it's hotter than the outer wall, it radiates out, taking all the heat away.

Despite the improvement in the convection-radiation coupling brought by the cylinder, it introduces significant pressure losses. Figures 9, 10 and 11 represent the evolution of the temperatures of the wall θ_P, of the mixture θ_m and the intermediate θ_S, according to the length, for the number of Peclet P_e=5000, Q_0=8 and r_s=0.4 constant and ε variable. The cylinder inside the flow, receives heat by convection and radiation, as its temperature is higher than that of the external wall, it will radiate and thus evacuate all of the heat.



Figures 9. Evolution of the temperature as a function of the length with $\varepsilon = 0$



Figures 10. Evolution of the temperature as a function of the length with $\varepsilon = 0.5$

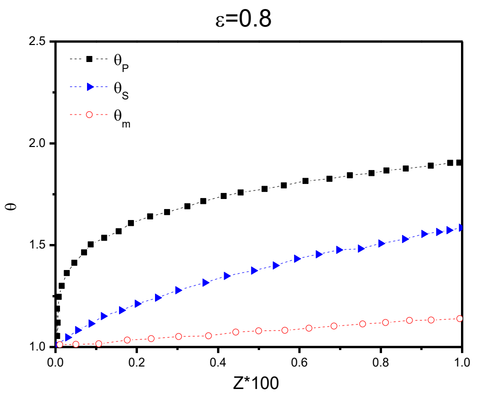


Figure 11. Evolution of the temperature as a function of the length with $\varepsilon = 0.8$

The results show that when the emissivity ε =0 the wall temperature is maximum. Furthermore, we note that for different emissivity, it is the temperature of the wall which is the highest, then that of the intermediate and of the mixture. We've also noted that there is a gradual increase in the temperature in the inlet and at the outlet of the annular space.

5. CONCLUSION

This numerical study treats the way which we can use to intensify the heat transfer by introducing a black body, which decrease the temperature of the heated wall.

We found that the temperature of the mixture tends to decrease, while the temperature of the walls increases, favoring heat flow exchange. The increase in the heat exchanged takes place gradually in the annular space from the inlet to the outlet of the tube. The introduction of the intermediate cylinder supposed to be black improves the heat transfer, because it radiates towards the fluid and the active wall. Radiation is important when the temperature difference between the wall and the mixture is also important. Radiation by absorbing the heat in the convection dissipates the excess heat.

The advantage of this method is in the use of the finite volume method for the discretization of the computational domain. This method is fairly simple and is applied to all geometries. It allows us to solve the problem of an axisymmetric flow of a radiation- forced convection coupling in dynamic and thermal mode, in an annular-cylindrical space.

NOMENCLATURES

1. Acronyms

P	Wall
r	Radiation
f	Fluid
in	Inlet
out	Outlet
t	Total

2. Symbols / Parameters

D_h: hydraulic diameter (mm)
r_p: Outer radius (mm)
r_s: Inner radius (mm)
N_c: Radius ratio
C_p: Constant pressure specific heat, [J/Kg K]
K: Thermal conductivity (J / m.s.°C)
L: Length of the test tube (mm)
Q_r: Density of the radiative flux
h_t: Coefficient of total heat exchange (w / m^2.C)
h_c: Coefficient of convective heat transfer (w / m^2.C)
h_r: Radiation heat exchange coefficient (w / m^2.C)
N_u: NUSSELT number
R_a: Rayleigh number
P_r: Prandtl number
R_e: Reynolds number
P_e: Peclet number
θ_m: Temperature of the mixture (°C)
θ_s: Wall temperature (°C)
θ_f: Flow temperature of the cross-sectional area (°C)
f: Coefficient of friction
V: Air radial velocity (m/s)
U: Air axial velocity (m/s)
r: Radial coordinate (mm)
x: Axial coordinate (mm)
X: Length of the tube (mm)
α: Thermal diffusivity (m^2/ s)
Σ: STEFAN-BOLTZMANN constant
ε: Emissive power
λ: Thermal conductivity (w / m.°C)
υ: Kinematic viscosity (kg / m.s)
ρ: Fluid density (kg / m^3)

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