Combined Natural Convection-Radiation in an Annulus between Two Concentric Cylinders

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ABSTRACT- In this work, we intend to solve numerically the impact of the coupling of convection-radiation for a cylindrical and an annular cylindrical configuration in order to improve this heat transfer. For that, we subject the cylinder external with a constant parietal heat flow. The numerical resolution of this system related to their limited conditions was carried out by the finite differences method making it possible to simulate the radiative effect in the annular cylindrical space. Therefore, the aim of our study is to recreate at low-temperature, radiative effects being able to improve the heat transfers in flow patterns (parallel plates and annular cylindrical spaces), representative of the ones seen in electronics or electrotechnics. The results obtained show the absorption of the black body introduced for various positions of the intermediate wall.

Key words-coupled radiation-convection, Finites differences, annular space.

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1. INTRODUCTION

Heat transfer enhancement is a permanent concern for the heating engineer and is very useful for the development of the average and high temperature exchanges. Our aim is to study the absorbing tube's influence on the field of low temperatures. The natural way, that is to say, the coupling of radiation-convection is used for its nearly gratuitousness and its very high reliability to absorb the heat produced by the electronic components or the electric machines. Our study concerns the way of enhancing this transfer under the influence of the radiation, because the airflow does not totally carry the dissipated energy in the heating resistance of a wall of a component and of a rotor-stator. A certain part is lost by the radiation of this wall towards the fluid and consequently towards the external wall of the tube. This energy is not trivial. The coupling of the convection and the radiation is one of the solutions, which thus makes it possible to improve the transfer of heat, especially in the thermics of electronics and of electrical engineering, a field where we work at low temperature. In these terms, even an improvement of a weak exchange is important, because the power dissipated by integrated circuit, microchip or on the level of the rotor-stator is unceasingly growing.

2. BIBLIOGRAPHY

the natural Convection of a fluid occupying an annular space was the subject of multiple research tasks [l-2] because of its various industrial applications: heat exchangers; medical sterilization; cooling of the electronics components and the nuclear engines. The conventional means of heat dissipation can be still improved. They are in general classical devices such as the ventilators, heat exchangers or pipes. On the other hand, N.W.Antonetti et al [3] installed heat exchangers spread out along the flow thus allowing a reduction in the temperature from 40° to 15°C. Heat is hardly transferred by a single mode, in practice, at least two modes appear simultaneously, otherwise the three modes of thermal transfer. On the other hand, Hirano et al [4] took into account the aspect absorption of the fluid namely the carbon dioxide. For instance, we can quote Mr Jacob's experimentations [5] carried out by using fluids circulating in an annular space such as ethylene, hydrogen, heated water for the calculation of the coefficient of heat transfer in laminar flow and turbulent. T.C.Chawla [6] studied the transfer of heat in a formed annular space of a motionless external cylinder and an interior cylinder turning without axial flow. Most of heat is transmitted by radiation and conduction. Onyegigu [7] was interested in the thermal transfer for the same annular space, with an interior cylinder turning heated in a motionless cooled external cylinder. R.Viskanta [8] studied the effects of the coupling radiation-convection in annular space. Toor and al [9] calculated the heat transfers by radiation between two surfaces in a simple way and having properties which depend on the direction by the use of the Monte Carlo method. Models of diffuse and specular emission and reflexion with constant and directional specular properties were treated to understand the phenomenon of the radiation of surfaces, four simple configurations allowing critical examination of the directional effects were treated. The obtained results show that under certain conditions, the choice of the model for the radiative properties of surface can be very critical at the same time for the heat flow by radiation and for calculations of the total exchange by radiation. Kaminski [10] made a comparison of the results of radiative transfer with the method P1 and the Monte Carlo method for an included conical cavity a gray and isothermal medium. Walters and Buckius [11] elaborated an algorithm, which treats the case of the diffusion by the Monte-Carlo method . They also explained the radiative transfer in the event that the medium is non homogeneous. One of the first studies of this problem was done by Baker and Kaye [12] who were interested in experimentatons in the case of an axial flow without rotation, or of a rotation of the internal cylinder without the presence of an axial flow through annular space. M.Bouafia and al [13] studied in experimentations and digitally the convective transfers between the walls of an annular space with an interior turning cylinder . Two configurations are analyzed. The cylinders surfaces are all very smooth, or the mobile wall is smooth and the other one is axially grooved. They noticed that the appearance of swirling structures in the grooved space is similar to the Taylor swirls present in a smooth space. M.Bouafia and al [13] presented numerical and experimental results characterizing the convective transfer in smooth or grooved annular spaces of the airflow, they also noticed a very satisfactory concordance of the number evolution of Nusselt for a smooth space, with those of Becker and Kaye [12]. Webb and Viskanta [14] analyzed, using a monodimensional radiative model, the natural convection caused by an external radiation in a rectangular semi-transparent medium. Yucel and al [15] considered the same problem by using the method of the discrete ordinates, which was adapted by Fiveland [16]. This method is very much used because of its flexibility and its simplicity. It is well adapted to the coupling problems, and is easy to formulate, even for multidimensional cases or complex surfaces. The methods of direct integration of the radiative transfer equation call upon various. Tan and Howell [17] adapted the method product-integral to solve the equation of the radiative transfer in a problem of natural radiation-convection inside a two-dimensional rectangular enclosure. In 1990, Chui and Raithby [18] presented an alternative (FVM) of Finit Volum Method to solve the equation of the radiative transfer, perfectly compatible with the technique of volumes of control used in the convection problems. The coupling of the natural convection and the radiation in an annular space of horizontal axis delimited by two concentric cylinders or vertically eccentrics is studied digitally. The medium taking part is gray and it absorbs, emits and diffuses in an isentropic way the radiation. The numerical model uses the control volumes method formulated in the cylindrical frame of reference adapted to the discretization, as well as equations of weight, and heat that of radiative equation, an iterative method with successive relievings is used to solve the algebraic system thus obtained. The radiation influence on the fields of the temperatures and speeds, also on the heat transfer, is presented and discussed for Rayleigh numbers of 10^3, 10^4 and 10^5 and values of the parameter of coupling radiation-conduction varying between zero and infinite. The cylindrical coordinates and the finit volum method used to solve the equation of the radiative transfer constitute an effective tool to study the coupling of the natural convection and radiation, in the case of certain geometries. Ki Wan Kim, Seung Wook Baek [19] carried out studies of the reverse analysis of the coupling of radiation-conduction between two dimensional coaxial cylinders. They showed that the thermal transfer by the conduction and the radiation is a significant mode of thermal transfer in various technologies, such as the industrial production.

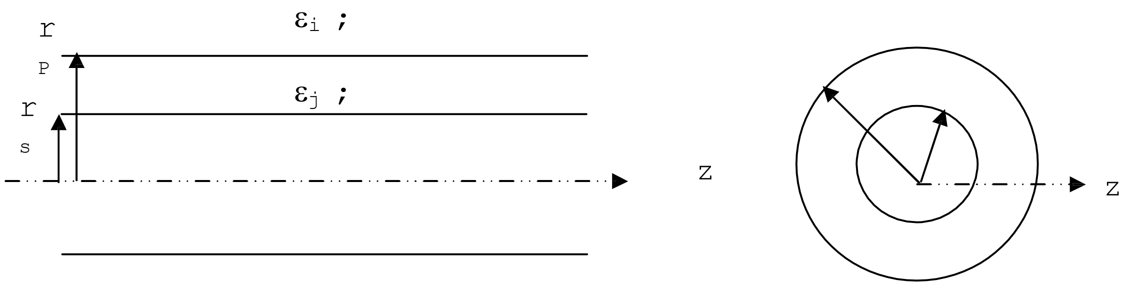
N. Aouled-Dlala and al [20] made a new technique to improve the performances of the method of the discrete ordinates by solving the problem of the coupling conduction-radiation between two concentric cylinders or spheres, This method proves more precise than the method of DOM (method of the discrete ordinates). Moreover, the results of this method in all the cases presented are obtained without additional numerical cost. Vital Dez, Hamou Sadat [21] carried out a study on the radiative transfer in a semi-transparent medium of a annulo-cylindrical space and proposed an exact analytical description of the internal radiative field inside this gray semi-transparent medium absorbing. They showed that the radiative field could be completely described by purely geometrical weighting coefficients which makes it possible to determine the field of the temperature for any thermal transfer compound with the radiation. It should be noted that no external radiation effect occurs since the only numerical discretization is purely spatial. Some examples prove that the method suggested gives exact results for the fields of the temperature and flow inside the medium. Meziane and al [22] studied the problem of the radiative exchanges between two surfaces for two cases: First , surfaces are separated by a transparent medium and then by a semi-transparent medium . In a first stage, it is the factors of form between two surfaces for a perfectly transparent medium, which are evaluated by the implementation of the Monte Carlo technique. The results obtained are compared with analytical results available through the literature. In a second phase, calculations of radiative exchanges in three types of cavities confining a M.S.T (a Semi-Transparent Medium), are carried out by these same techniques. H.A. Mohammed and al [23] studied in experiments the effect of the convection forced along an annular space. They obtained that the free convection effects tended to decrease the heat transfer at low Re number while to increase the heat transfer for high Re number. This investigation reveals that the Nusselt number values were considerably greater than the corresponding values for fully developed combined convection over a significant portion of the annulus.

3. HEAT TRANSFER MODELISATION IN AN ANNULAR CYLINDRICAL SPACE

3.1 Radiation

The dissipated energy in the heating resistance of the external cylinder is not carried in its totality by the draught. A certain part is lost by the radiation of the external wall of the intermediate tube. This energy cannot, though weak, in our case, be neglected, since we worked at low temperature. Actually, radiated energy is itself carried by the air licking the wall, so that this energy finds itself finally in the moving fluid. Knowing the emissivities of the walls, and respective surfaces, we can calculate the quantities of heat per radiation by means of the traditional law .

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



This radiant flow increases, if on the one hand the emissivities are largest, and on the other hand if the values of the wall temperatures are selected suitably.

3.2 Radiative transfers evaluation with multi-reflexions

In the preceding theory, we did not consider that of a direct transfer of heat between surfaces. In particular we could not consider the role which the third surface K (intermediate surfaces) could have, which thanks to its factor of reflexion K to return towards surface j a flow part actually coming from surface i . The taking into account of the multi-reflexions in the radiative exchanges requires in fact a more complicated treatment than that of the direct transfer. For that, we have two methods:

- method of GEBHARDT
- method of the radiosities (of POLJACK)

3.3 Problem formulation

3.3.1 The heat balance equation

The analytical method is used when the geometry and the boundary conditions of the problem to study are simple. However, when the geometry and of transfer are complicated, the numerical methods are used to solve this kind of problem. One of these methods is the finite differences method which consists in carrying out a discretization of the configuration transforming the partial derivative equations into a system of linear equations. The discretization of the variables is obtained by cutting out the field studied by a grid of form and of size adapted to each problem. After having discussed the dynamic side of the phenomenon, the objective that we fix ourselves in this chapter is to study the evolution of the thermal field for a cylindro-annular space. The equation of the heat applied to the fluid is written in the general case in coordinated cylindrical, Eq.(2):

$$U\cdot\frac{\delta T}{\delta z}+V_r\cdot\frac{\delta T}{\delta r}+V_\Omega\frac{\delta T}{\delta\Omega}=\alpha\cdot\left[\frac{\delta^2 T}{\delta z^2}+\frac{1}{r}\frac{\delta T}{\delta r}+\frac{\delta^2 T}{\delta r^2}+\frac{1}{r^2}\frac{\delta^2 T}{\delta\Omega^2}\right]+\frac{1}{r}\frac{\delta}{\delta r}(rQ_r)\qquad(2)$$

3.3.1.1 Assumptions

- It is admitted that the heat transfer is stationary $\delta T/\delta_t = 0$
- It is admitted that the fluid is incompressible, and that its physical properties are constant.
- The viscous dissipation of energy is neglected (not friction within the fluid).
- Conductivity along the tube is negligible $\frac{\delta^2 T}{\delta z^2}=0$
- The axisymmetric flow is fully developed : $V_r = V_\Omega = 0$

By taking account of the coupling of the convection and the radiation, and by neglecting the axial diffusion in front of the radial diffusion

$$\frac{\delta^2}{\delta\Omega^2}=\frac{\delta}{\delta\Omega}=0\qquad(3)$$

Under the terms of these assumptions, the equation (1) becomes:

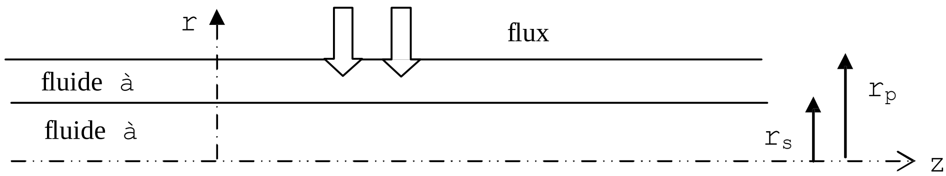
$$\rho\cdot C_P\cdot\left[U\cdot\frac{\delta T}{\delta z}\right]=\frac{1}{r}\frac{\delta}{\delta r}\cdot\left[K\cdot r\cdot\frac{\delta T}{\delta r}\right]+\frac{1}{r}\frac{\delta}{\delta r}(r\cdot Q_r)\qquad(4)$$

Where $\frac{1}{r}\frac{\delta}{\delta r}(r\cdot Q_r)=Q^*$ represent the density of radiative flow Net

The equation (2) can be written:

$$\rho\cdot C_P\cdot U\cdot\frac{\delta T}{\delta z}=\frac{1}{r}\frac{\delta}{\delta r}\cdot\left[r\cdot\left[K\cdot\frac{\delta T}{\delta r}\right]\right]+Q^*\qquad(6)$$

The equation of heat was solved by one of the finite difference methods:



With Q^* determining with the theorem of OSTRGRADSKI:

Let us pose:

$$\varepsilon_r=\frac{\sigma}{\frac{1}{\varepsilon_2}+\left[\frac{1}{\varepsilon_1}-1\right]}\qquad(7)$$

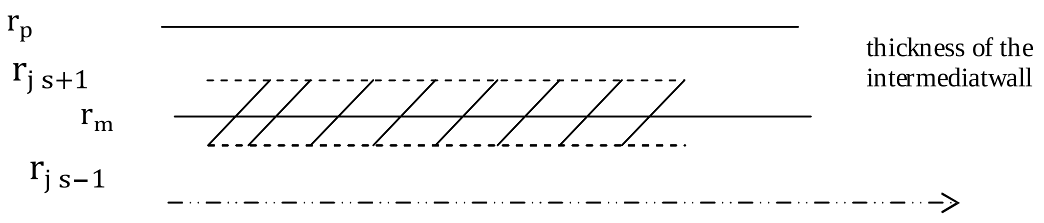
===>

$$\varepsilon_r\cdot\left(T_P^4-T_S^4\right)\cdot 2\pi r_m L=Q^*\cdot\pi\left(r_{jS+1}^2-r_{jS-1}^2\right)\cdot L\qquad(8)$$

With $r_m=\frac{r_{js+1}-r_{js-1}}{2}$

From where

$$Q^*=\frac{\varepsilon_r\cdot\left(T_P^4-T_S^4\right)}{r_{jS+1}-r_{jS-1}}\qquad(9)$$



3.3.1.2 Boundary conditions

Conditions associated with the problem are:

- z = 0 et r unspecified $T = T_0$
- z # 0 et r = 0 $\frac{8T}{8r} = 0$
- z # 0 et r = R_P $\frac{8T}{8r} = \frac{Q_{cp}}{k}$

3.3.2 Adimensionnement of the Problem

3.3.2.1 Adimensionnement of the energy equation

By taking account of the problem:

- geometry : r_s, r_p, L
- dynamic : U_0
- thermics: T, T_0
- fluid : ρ, Cp, K, μ

By introducing the adimensional variables, namely

$$8 = \frac{}{0} \;;\; R = \frac{}{p} \;;\; = \frac{}{e*2*p} \;;\; u^* = \frac{u}{u_0} \;;\; Q^* = Q_1\{3$$

====> $T = 8 * T_0$; $r = R * r_p$; $z = \frac{e*p}{2}$; $u = u^* * u_0$

By using these adimensional variables, the equation (3) becomes:

$$\rho \cdot C_P \cdot U \cdot \frac{2 \cdot T_0}{Pe \cdot r_P}\frac{\delta\theta}{\delta z} = \frac{K \cdot T_0}{r_P^2}\frac{\delta^2\theta}{\delta R^2} + \frac{K \cdot T_0}{r_P^2}\frac{1}{r}\frac{\delta\theta}{\delta R} + Q_1 \cdot \beta = \frac{K \cdot T_0}{r_P^2}\left[\frac{\delta^2\theta}{\delta R^2} + \frac{1}{r}\frac{\delta\theta}{\delta R} + \frac{r_P^2}{K \cdot T_0} Q_1\beta\right] \quad (10)$$

By replacing the PECLET number : $P * R_e = \frac{u_0 * P * c_p}{k} * 2 * r_p$

The first member of the equation (4) becomes equal to :

$$\rho \cdot C_P \cdot U \cdot \frac{2 \cdot T_0 \cdot K}{U_0 \cdot \rho \cdot C_P 2 \cdot r_P}\frac{1}{r}\frac{\delta\theta}{\delta z} = \frac{U}{U_0}\frac{K \cdot T_0}{r_P^2}\frac{\delta\theta}{\delta z} = U \cdot \frac{K \cdot T_0}{r_P^2}\frac{\delta\theta}{\delta z} \quad (11)$$

With these new data, the equation (4) becomes:

$$U \frac{K \cdot T_0}{r_P^2}\frac{\delta\theta}{\delta z} = \frac{K \cdot T_0}{r_P^2} \cdot \left[\frac{\delta^2\theta}{\delta R^2} + \frac{1}{R}\frac{\delta\theta}{\delta R} + \frac{r_P^2}{K \cdot T_0} \cdot Q_1\beta\right] \quad (12)$$

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

$$U \cdot \frac{\delta\theta}{\delta z} = \frac{\delta^2\theta}{\delta R^2} + \frac{1}{r}\frac{\delta\theta}{\delta R} + Q_1 \qquad (13)$$

With :

$$Q_1 = \frac{Q^*}{\beta} = \frac{\delta Q_r}{\delta R} = \varepsilon_r \cdot \left(T_P^4 - T_S^4\right) \cdot \frac{1}{\left(r_{j_S+1} - r_{j_S-1}\right)} \cdot \frac{r_P^2}{K \cdot T_0} \qquad (14)$$

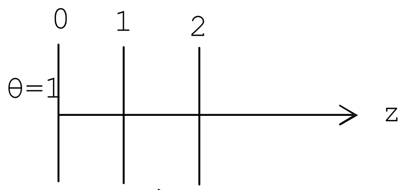
Knowing that : $\beta = \frac{r_P^2}{k * T_0}$

From where:

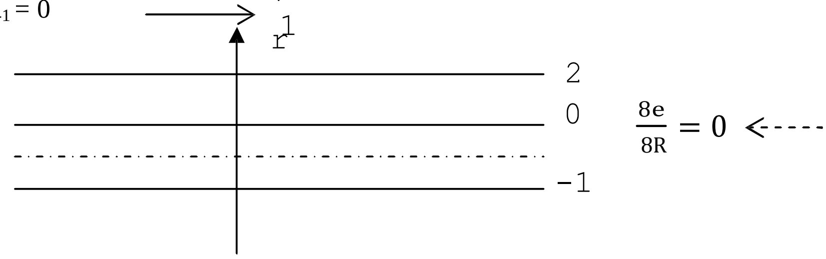
$$Q_1 = \frac{\varepsilon_r \cdot r_P^2}{K \cdot T_0} \cdot \left(T_P^4 - T_S^4\right) \cdot \frac{1}{r_{j_S+1} - r_{j_S-1}} \qquad (15)$$

3.3.2.1 Adimensionnement of the boundary conditions

- Z = 0 and R = 0 et θ = 1



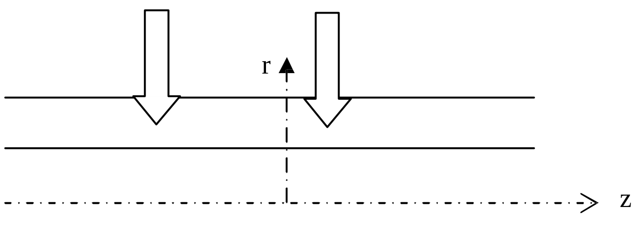
- Z # 0 and R = 0 : $\frac{\delta\theta}{\delta R} = 0$ ===> $\theta_{i,1} - \theta_{i,-1} = 0$



$A_0 = 0$; $B_0 = 1$; $C_0 = -1$; $D_0 = 0$

- Z # 0 et R = 1 : $\theta_{i,j} - \theta_{i,j-1} = Q^*_{opi} * (R_j - R_{j-1})$

$Q_c = Q_0 - Q_1 = \frac{Q_0 * r_0}{k * T_0} = \frac{\delta\theta}{\delta R}$



with :

$Q_c = \frac{k * T_0}{r_0} = \frac{\delta\theta}{\delta R}$

$A_j = -1$; $B_j = 1$; $C_j = 0$; $D_j = Q_c(R_j - R_{j-1})$

3.3.3 Resolution by finite difference methods

3.3.3.1 Grid used:

The mathematical modeling of the problem presented above, is based on the principles of the conservation of the momentum, energy and the mass. We complete the system obtained by the introduction of the boundary conditions. For very long coaxial cylinders, we can neglect the effects edge and consider that the problem is two-dimensional. The integration of the system of physical equations is based on the discretization of each equation, by the method the finite differences centered with a diagram ADI. What leads us to solve a system with matrices tri-diagonals with diagonal predominance, by using an algorithm derived from the method of Gauss. To visualize better our approach concerning the resolution of the equations we realized an algorithm of the main program(see Figure 12). We chose a uniform grid, with constant step in two directions R and Z. This type of grid facilitates the discretization of the equations considerably and makes the programming on computer easy. The following steps R and Z are different (ΔR # ΔZ).

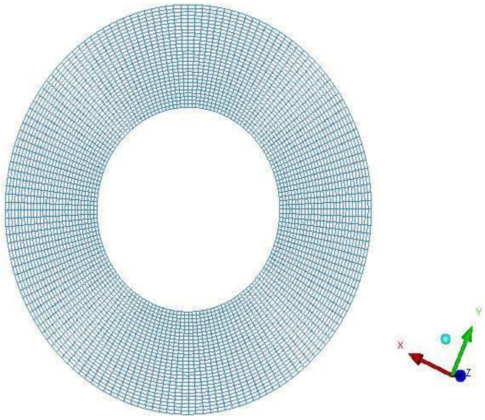


Figure 0: Grid

4. RESULTS and DISCUSSION

4.1 Validation

Contrary to the pure natural convection in cavity, there are few data in coupling natural convection radiation of surfaces for the validation of our case. The most appropriate conFigureuration seems that studied by [1, 2, 7, 17].

Let us note nevertheless that [1,2] studied a Laminar natural-convection heat transfer from a horizontal isothermal cylinder is studied by solving the Navier-Stokes and energy equations using an elliptic numerical procedure. Results are obtained for $10^6 < Ra < 10^7$. Boundary-layer solutions do not adequately describe the flow and heat transfer at low or moderate values of *Ra* because of the neglect of curvature effects. ONYEGEGBU [7] Consider an absorbing and emitting nongray Boussinesq fluid in the annular gap between two infinitely long horizontal concentric cylinders. The walls are isothermal with the inside wall maintained at a temperature T_F and the outside wall maintained at a lower temperature T_*. Results indicate that decreasing Planck number, increasing the degree of nongrayness of the fluid ***or*** increasing optical thickness increases the total heat transfer and reduces the induced buoyant flow intensity and velocities. The exact integral formulation for radiant transport and the momentum and energy balance equations are discredited by the product-integral method and finite difference method, respectively [17]. The resulting algebraic equations are solved by a non-linear SOR technique, with the Rayleigh number varying from 10^{3}, 10^4 to 10^5, and the radiation-conduction parameter ranging from 0 to ∞. The influences of radiation-conduction parameter, Rayleigh number and other parameters on flow and temperature distributions and heat transfer are discussed. In our case, we solve numerically the impact of the coupling of convection-radiation for a cylindrical and an annular cylindrical configuration by the finite differences method making it possible to simulate the radiative effect in the annular cylindrical space, by varying the ratio of radius and the value of the emissivity.

4.2 The single cylinder case

For this case of Figure, we made the report/ratio, $N_C = r_S / r_P$ towards 1, we obtain in this case of pure convection . The results are in harmony with those obtained in theory.

4.3 The annular cylindrical space case

The coupling convection –radiation, in annular space is accentuated on the intermediate wall, or the term source (Q_1) simplified makes it possible to intensify the thermal transfer. This ratio r_S / r_P varying several values in particular N_C = 0.4; 0.5 and 0.6 with various emissivities. A configuration was retained, namely the case of coaxial cylinders of ratio equal to 0.4 (very diagrammatic case of the electric motors), for a range of temperature not exceeding the 400°K, temperatures met into electrotechnical (rotor-stator) and electronics. We studied the possibility of increasing the exchange heat exchange by the influence of the radiative plate absorption (presumedly black) and the effect of the interposition of the tube within the flow.

4.3.1 Profile speed

As we can see on Figures 1 and 2, the variation of the ratio of the radius of 0.4 to 0.6 and for the same Peclet number, produces a very clear increase of the speed from approximately 36%. The profile speed is modified, which results in stronger gradients on the level of the wall and thus a better transfer of heat. This can be explained by the fact that annular space is more reduced, making
it possible for the speed to almost double value from one case to another. However, it would be necessary to consider the pressure loss, which results from the introduction of the intermediate tube (see Figure 11).

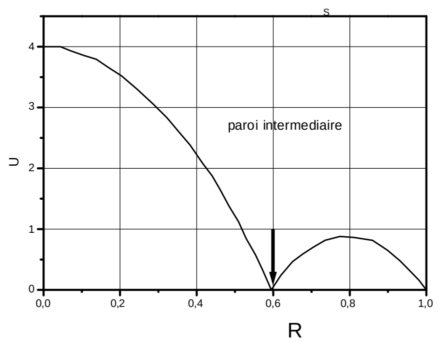


Figure 1: Profile speed
R_S=0.6

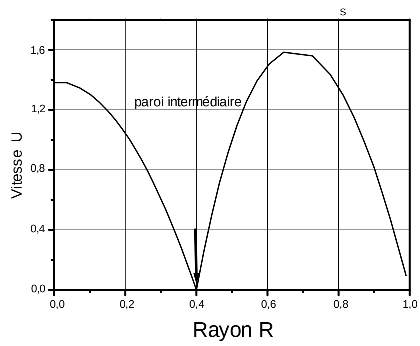


Figure 2: Profile speed
R_S=0.4

4.3.2Temperature profile

As Figures 3 show it, 4 and 5 where we traced according to the radius the change of the temperature of the wall for various values of emissivity ϵ. We notice an increase of the temperature very close to the intermediate wall in the case to a cylindrical control without absorbing tube. We can explain this increase by the fact that the wall previously passivates (body transparent ϵ=0) from the radiation point of view, absorbs and emits to a significant degree. We traced on Figures 7, 8 and 9 the change of the wall temperature θ_P of mixture θ_m and intermediary θ_S under the same parietal flow of heat, and for Peclet numbers and of the constant Boltzmann. As Figures 3 and 7 show it, the maximum of temperature of wall is obtained for a transparent tube where emissivity is null (ϵ=0).

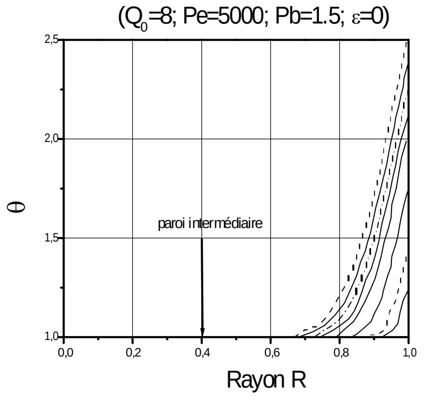


Figure 3 : Profile of the temperature
R_S=0.4 ; ε = 0.0

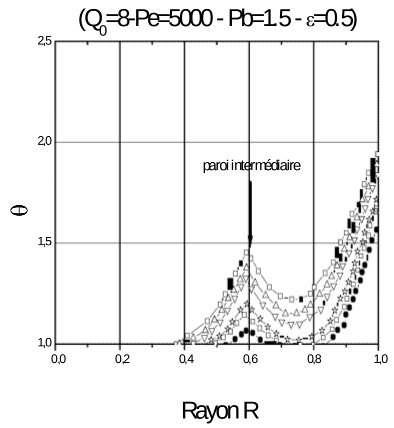


Figure 4 : Profile of the temperature
R_S=0.6 ; ε = 0.5

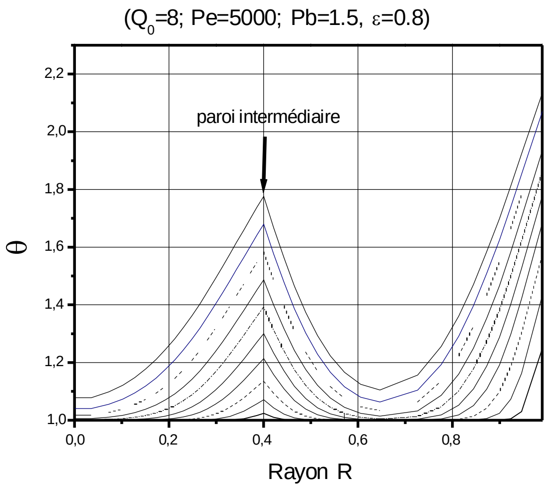


Figure 5 : Profile of temperature

Rs=0.4 ; ε = 0.8

We can notice that for an emissivity varying of ϵ=0, while passing by ϵ=0.5 with ϵ=0.8, the temperature of the wall and the intermediate wall fall appreciably . We also notice that according to spacing, namely the variation of the ratio of the radius of 0.4 to 0.6, the profile of the temperature is the same (see Figures 4 and 5). In more if emissivity grows, the temperature of the intermediate wall grows the same way. The tube within the flow, receives on behalf of the fluid heat by convection and radiation. The introduced tube thus tends to take a higher temperature that of the external wall and thus will radiate, thus evacuating the totality of received heat. The absorbing tube within the flow, although it makes much improvement in the coupling convection-radiation, introduced a pressure loss, which should not be neglected. Indeed, on Figure 6, we reproduced the change of the temperature of the wall according to the pressure loss for a constant parietal flow. We noticed that the profiles of temperature become deformed and they tend towards a form limits while increasing. Results from this stabilization of the temperatures profiles the growth of the average fluid temperature becomes equal to the growth of the wall temperature.

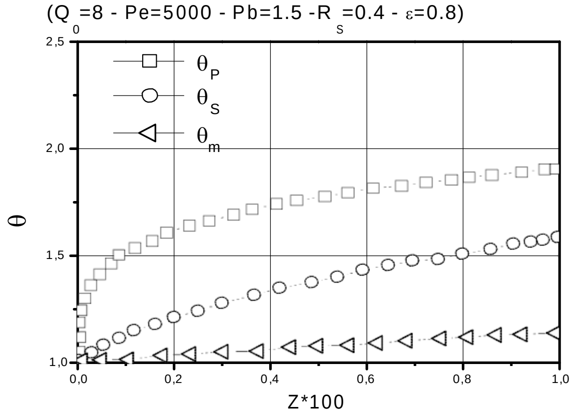


Figure 6 : Intermediate,wall, and mixture evolution of temperature Rs=0.6 ; ε = 0.8

By introducing this tube, the pressure loss grows with the detriment of the wall temperature. This is a disadvantage for the cooling of any electronics or electrotechnical component (Figure 11). With equal pressure loss, the introduction of the black tube makes the external tube cool better that is subjected to constant parietal flow. So, from the thermal point of view, the coupling of the forced convection and the radiation facilitate the dissipation of the heat of the electronic components or the electrotechnical equipment. This, was carried out at low temperature.Another representation of the distribution of the profile of the radial temperature for various lengths of the tube. The more the length of the tube increases the more the temperature decreases for the same value of emissivity (Figure 9).

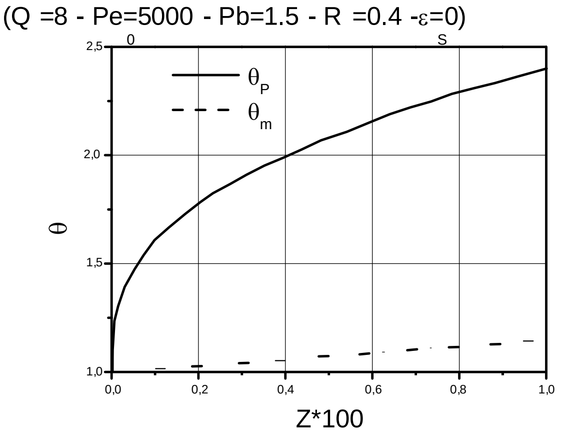


Figure 7 : Profile of temperature Rs=0.4 ; ε = 0.0

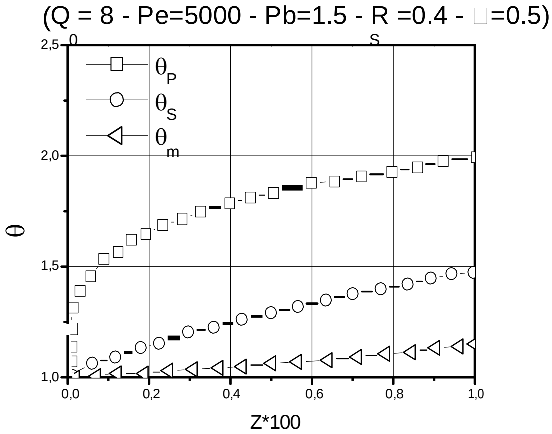


Figure 8 : Intermediate,wall, and evolution of temperature, Rs=0.6; ε = 0.5

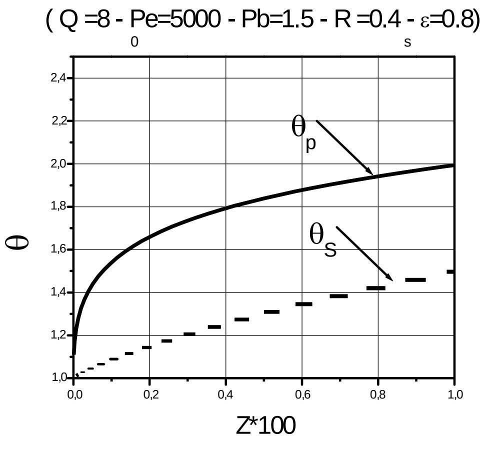


Figure 9 :Profile of a wall temperature and intermediary along the axis Z

On Figure 10, we distinguish the participation of the introduced tube into the intensification of heat, through radiative flow, indeed, the more we advance in the tube, we note the growth in radiative flow depending on convective flow . we can, starting from the temperature of the imposed wall, know radiant flow, this last cut off from imposed flow (known), will allow us to know the convection flow. Once, known convective flow, we Figure out the temperature field and we Figure once more radiant flow, then the convectiveflow and so on by iteration.

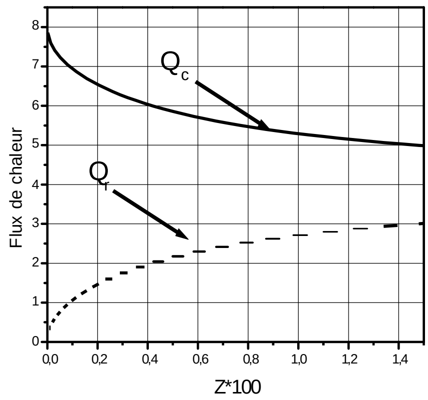


Figure 10 : Distribution of the heat flow along the axis Z

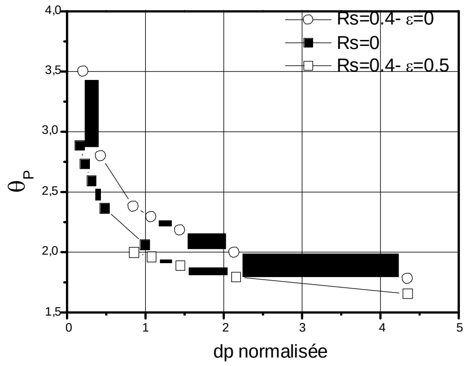


Figure 11 : Change of the temperature According to the pressure loss

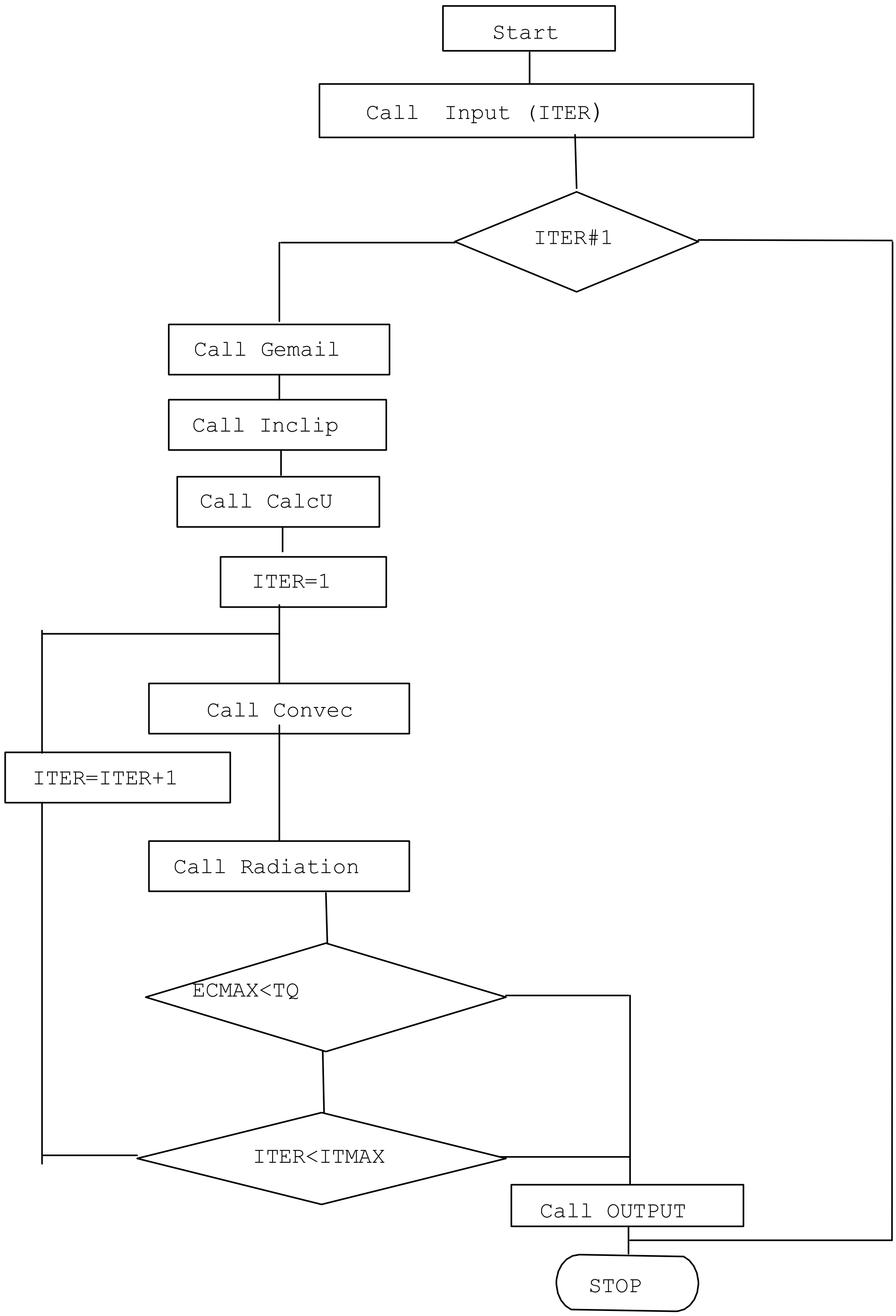


Figure 12: Synopsis of the main program

5. CONCLUSION

The work that we have just presented is above all a numerical work. The approached work in this paper made it possible to consider the possibility of intensifying the heat transfer, by the introduction of a radiant body, because the black painting which covers the introduced tube, lowers the temperature of the wall of the heated tube. We studied the distribution of the fluid masses, because this distribution plays a significant role in the way in which heat is evacuated by the flow of air. Knowing that all the electronic components and electrotechnical are sensitive to the temperature and knowing that the thermal transfer by convection and radiation is practically one of the means used to cool the integrated circuits and the electric machines. We can notice a very clear reduction in the temperature of the mixture and an increase in total exchanged heat. This increase is pointed out gradually from the entry to the exit of theannular space and tends to be maintained along the tube. This improvement is carried out thanks to the wall of the intermediate tube presumably black. This last radiates towards the fluid and the other active walls. The radiation is accentuated more and more when the variation in the temperature $T_P - T_m$ is large. The results obtained numerically confirm this, because the radiation by absorbing heat, takes part with the convection to break down the surplus of heat.

The modeling of this heat transfer for a annular cylindrical space confirms the role played by the radiation, namely the lowering of the temperature profile (see Figures).

Using the finite differences method, we solved the problem of non dimensional and axisymmetrical a flow of a coupling convection forced-radiation in dynamic mode and thermal bench, in annular cylindrical space. However, it is not possible, to affirm that in similar apparatuses (exchanging used in boxes or at the edge of satellites), that the results would be the same and that we would find with precision the coefficients and the temperatures which we measured. Because the supplies of fluid and the outfall conditions of heat are not the same. Many prospects remain to be developed and realized, namely the replacement of the air by a fluid more absorbent (CO_2, CO...), in order to better intensify the heat transfer, or to associate the two solutions together: radiative plate + fluid absorbing.

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NOMENCLATURE

QUANTITY	SYMBOL	COHERENT SI UNIT
hydraulic diameter	D_h	m
radial co-ordinate	r	m
Axial co-ordinate	z	m
radius of the external tube	r_P	m
radius of the internal tube	r_S	m
Ratio of the radius	N_C	
thermal conductivity	k	J/ms°C
length of the tube under test	Z_L	m
length of the tube upstream de Z=0	Z_D	
a number of meshs according to R ($0<R<R_P$)	J_P	
a number of meshs according to ($0<R<R_S$)	J_S	
a number of meshs according to axis Z ($0<Z<Z_L$)	L	
a number of meshs according to axis Z ($Z_D<Z<0$)	I_D	
pas de steps of the exit of the temperature profiles	IPRT	
parameter of grid en R	EVR	
parameter of grid en Z	EVZ	
density of radiative flow	Q_r	m^2/s
Emissivity (radiation)		$w/m^2{}^\circ C$
constante of STEFAN-BOLTZMANN	σ	$5.67\ 10^{-8}\ W/m^2{}^\circ K$
Viscosity dynamic (absolute)	µ	kg/m s
Temperature of wall	T_P	K
Temperature of mixture	T_m	K
thermal diffusivity	α	m^2/s
coefficient of heat transfer by convection	h_c	$w/m^2{}^\circ C$
Heat Transfer Coefficient	H_t	$W/m^2\ K$
radial speed of the air	V	m/s
axial speed of the air	U	m/s
Friction Factor	$f = r/0.5\rho u^2$	
Nusselt Number	$Nu = hd/a$	