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Effect of heating position on combined natural convection and non-grey gas radiation

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Abstract—**A numerical investigation was carried out for combined natural convection and non-grey gas radiation in a partially heated cylindrical enclosure. The half of the cylindrical enclosure was maintained at a hot temperature (T_h) while the second half was maintained at a cold temperature (T_c). The radiative heat transfer equation was solved through the Ray Tracing method associated to the statistical narrow bands correlated–k (SNBcK) model useful to compute the medium radiative properties. Special focus was devoted to entropy generation. The effects of heating position on temperature, velocity field and Nusselt numbers are investigated. Also its effects on the energy efficiency and exergy losses are presented and discussed in this paper.**

Keywords—**Natural convection, Radiation, Entropy Generation, SNBCK model, Ray Tracing, Cylindrical enclosure, Partial heating**

I. INTRODUCTION

Natural convection has received a big attention from researchers because of its interesting role in many engineering applications such as cooling fields, storage thermal systems, nuclear reactor, boilers, heat exchangers and solar collectors [1- 6].

Recently, the inclination angle and the partial heating through enclosures in free convection are introduced as new approaches by Sigh et al. [2] and Kefayati [3]. They proved that the negative inclined angles led to decreasing the heat transfer. Their results showed also, that the minimum entropy production occurred for inclination angles higher than 45°. Ghasemi et al. [4] studied numerically natural convection heat transfer along a cold outer circular enclosure containing a hot inner elliptic cylinder. They investigated the effect of inclination angle and the size of the inner elliptic enclosure for various Rayleigh numbers. They proved that these parameters influenced the streamlines, the isotherms and the average Nusselt number. Park et al. [5] analyzed the natural convection heat transfer around two cylinders, one is heated and the other is cooled, placed inside an enclosure symmetrically with cold walls. They showed that the distance that separated the cylinders and the enclosure lead to the appearance of local peak for Nusselt number along the cylinders and enclosure surfaces. Park et al. [6] treated the natural convection through an inclined enclosure which is contained inside a heated cylinder. They proved that the inclination angle, the cylinder radius and Rayleigh number produced changes in Nusselt number. Shyam et al. [7] analyzed the heat transfer through natural convection around a heated cylinder placed in a square enclosure. The upper and the bottom walls are insulated while the side walls are maintained at a cold temperature. They studied three cases of heating position top, middle and bottom of the cylinder. They investigated that the heat transfer is reduced when the position of the cylinder is near the top wall. Lee et al. [8] investigated two dimensional numerical simulations of natural convection through an enclosure with an inner hot cylinder located inside a cold square enclosure locally heated for different Rayleigh number (10^3, 10^4, 10^5, 10^6). Their results showed that the variation of the local heating part's size from the bottom wall presented neglecting changes in thermal and flow structures for $Ra=10^3$ and $Ra=10^4$. While it induced little variation on the convective velocity. Rabani and Talebi [9] studied natural convection heat transfer through square enclosure containing a circular cylinder by the use of thermal Lattice Boltzmann Method. Zheng et al. [10] simulated numerically turbulent flow and heat transfer performance within a rib-grooved tube heat exchanger. They proved that, the minimum value of entropy generation is obtained for a rib grooved inclination angle equal to 45°.

Thermal radiation presents an important role in heat transfer through enclosures. The majority of research studies, including coupled radiation and natural convection, are interested only on the influences of surface radiation. While for higher temperature range, the radiative heat transfer in participating gases should not be neglected. Colomer et al. [11] analyzed radiation phenomenon and natural convection through transparent and participating media within differentially heated three-dimensional cavity. They investigated the effects of Rayleigh and Planck numbers and the optical thickness.

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Mahapatra [12] treated numerical simulations of transport phenomena in steady, laminar and two-dimensional flow through a partially heated and partially cooled cavity that contained an absorbing, emitting and scattering media. Borjini et al. [13] studied numerically the effect of radiative heat transfer within three-dimensional buoyancy low placed inside a differentially heated cubical enclosure for a grey, absorbing, emitting and isotropically scattering media. They showed that the conduction–radiation parameters influenced the structure of the main flow considerably. Ben Nejma et al. [14, 15] treated a numerical analysis of entropy generation through radiative transfer within an emitting-absorbing non-grey gas, respectively in cylindrical and spherical enclosures. Using the SNBcK [14, 16] model associated with the Ray-Tracing method, they proved that the volumetric entropy generation is the most developed in heating case while the wall entropy creation is the most important in cooling configuration. Analyzing the radiative heat transfer of non-grey gas confined in a cylindrical annulus with isothermal walls. Jarray et al. [17] proved that entropy generation is greatly affected by gas and wall temperatures in which the dominance between wall radiative entropy generation and the volumetric one depends mainly on differences between gas and wall temperatures. Mazgar et al. [18] studied numerically combined effects of natural convection and non-grey gas radiation between two vertical plates with partially heated walls. They proved that there is no major influences of two-dimensional radiation to reduce the difference between the top and the bottom heating for the reported top and bottom heating for the chosen gas.
The present study is devoted to analyze numerically combined natural convection and non-grey gas radiation through a partially heated circular cylinder. The effect of heating position in the circular cylindrical is investigated. Computation of the mean Nusselt number and average entropy generation, also the energy efficiency and exergy losses are presented and discussed in this paper.

II. PROBLEM FORMULATION

1) Physical model configuration

The model configuration under study is shown in fig. 1. In this work we considered that the half of the cylinder is heated. The hot and the cold parts of the cylinder are maintained respectively at T_h and T_c temperatures. The heating position varied from range (-80°) to (90°).

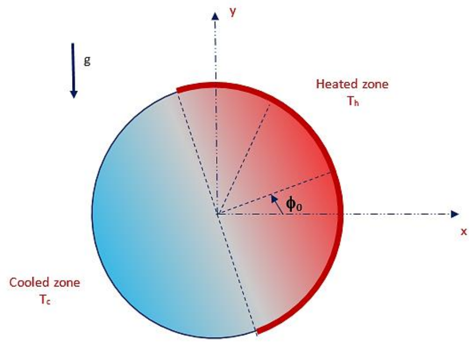

Fig.1Physical model and boundary conditions

2) Governing equations and Entropy generation

The general equations that govern natural convection are those of fluid dynamics. The equations correspond respectively to the continuity, momentum and energy equations. They are given as follows:

$$div\left(\rho\vec{u}\right)=0 \quad (1)$$

$$\rho\vec{u}\,\nabla\vec{u}=-\nabla P+\nabla\left(\mu\nabla\vec{u}\right)-\rho\vec{g} \quad (2)$$

$$\rho C_p\vec{u}\,\nabla T=\nabla\left(\lambda\nabla T\right)+Q_r \quad (3)$$

The radiative transfer equation permitted the computation of the radiative intensity through the Ray Tracing method which is based on the selection of the radiative transfer directions and their associated weights:

$$\frac{dI_\nu^i\left(r,\phi,\vec{\Omega}\right)}{dl}=-\kappa_\nu^i.I_\nu^i\left(r,\phi,\vec{\Omega}\right)+\kappa_\nu^i.I_\nu^b\left(T\left(r,\phi\right)\right) \quad (4)$$

The radiative boundary condition is given as follows:

$$I_\nu^i(R,\vec{\Omega})=\frac{1-\varepsilon}{\varepsilon}\int_\Omega I_\nu^i(R,\vec{\Omega})\mu(\vec{\Omega})d\Omega+\varepsilon I_\nu^b(T_w) \quad (5)$$

The conductive entropy generation is given as follows [1]:

$$Sc(r,\phi)=\frac{\lambda(T_h)}{T^2}\nabla^2 T \quad (6)$$

Using the SNBcK4 radiative model, the local volumetric radiative entropy production is given by the expression below [14, 16]:

$$s_V(r,\phi)=-\sum_{bands}\int_\Omega\sum_{i=1}^{4}w^i\kappa_\nu^i\left[I_\nu^b(T(r,\phi))-I_\nu^i(r,\phi,\vec{\Omega})\right]\left[\frac{1}{T(r,\phi)}-\frac{1}{T_\nu^i(r,\phi,\vec{\Omega})}\right]d\Omega\Delta\nu \quad (7)$$

Where the radiative, spectral and directional temperature is given by:

$$T_\nu(\vec{\Omega})=\frac{dI_\nu(\vec{\Omega})}{dL_\nu(\vec{\Omega})}=\frac{h.\nu}{K.Ln\left[\frac{2.h.\nu^3}{c^2.I_\nu(\vec{\Omega})}+1\right]} \quad (8)$$

And the spectral radiative entropy intensity is calculated through:

$$L_\nu(\vec{\Omega})=\frac{2K\nu^2}{c^2}\left\{(1+\frac{c^2I_\nu\left(\vec{\Omega}\right)}{2h\nu^3})ln(1+\frac{c^2I_\nu\left(\vec{\Omega}\right)}{2h\nu^3})-\frac{c^2I_\nu\left(\vec{\Omega}\right)}{2h\nu^3}ln(\frac{c^2I_\nu\left(\vec{\Omega}\right)}{2h\nu^3})\right\} \quad (9)$$

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The global volumetric radiative entropy production is then expressed by:

$$S_{Vr} = \int_0^{2\pi}\int_0^R r s_V(r,\varphi) dr d\varphi \tag{10}$$

The local radiative wall entropy generation is given by [17]:

$$S_W(\phi) = \sum_{bands}\int_{4\pi}\sum_{i=1}^{4} w^i \frac{I_v^{\,i}(R,\phi,\vec{\Omega})}{T_w(\phi)} - L_v(I_v^{\,i}(R,\phi,\vec{\Omega}))\mu(\vec{\Omega}) d\Omega \Delta v \tag{11}$$

The global radiative entropy production on the enclosure is written as follows:

$$S_W = R\int_0^{2\pi} s_W(\varphi) d\varphi \tag{12}$$

The average radiative and conductive Nusselt numbers are expressed as follows:

$$\begin{cases} Nu_{rh} = \dfrac{2.R}{\lambda(T_h).(T_h - T_c)} \dfrac{\int_{\phi_0-\frac{\pi}{2}}^{\phi_0+\frac{\pi}{2}} q_r(R,\phi) d\phi}{\pi} \\ Nu_{rc} = \dfrac{2.R}{\lambda(T_c).(T_h - T_c)} \dfrac{\int_{\phi_0+\frac{\pi}{2}}^{\phi_0+\frac{3\pi}{2}} q_r(R,\phi) d\phi}{\pi} \\ Nu_{ch} = \dfrac{2.R}{\lambda(T_h).(T_h - T_c)} \dfrac{\int_{\phi_0-\frac{\pi}{2}}^{\phi_0+\frac{\pi}{2}} q_c(R,\phi) d\phi}{\pi} \\ Nu_{cc} = \dfrac{2.R}{\lambda(T_c).(T_h - T_c)} \dfrac{\int_{\phi_0+\frac{\pi}{2}}^{\phi_0+\frac{3\pi}{2}} q_c(R,\phi) d\phi}{\pi} \end{cases} \tag{13}$$

Where: $\begin{cases} q_r(R,\phi) = \sum_{bands}\int_{4\pi}\sum_{i=1}^{4} w^i I_v^{\,i}(R,\phi,\vec{\Omega})\mu(\vec{\Omega}) d\Omega \Delta v \\ q_c(R,\phi) = -\lambda \left.\dfrac{\partial T}{\partial r}\right|_{r=R,\phi} \end{cases}$

The energy efficiency which measures the ratio of the increase in temperature heat exchange can be calculated through the expression below:

$$E_{eff} = \frac{1}{2.(Nu_{ch} + Nu_{rh})}\frac{(T_a - T_c)}{(T_h - T_c)} \tag{14}$$

The dimensionless exergy loss is expressed by:

$$L_{ex} = \frac{1}{2.(Nu_{ch} + Nu_{rh})}\frac{(S_C + S_V + S_W).T_c}{(T_h - T_c).\lambda(T_h)} \tag{15}$$

III. RESULTS AND DISCUSSION:

In figure 2, we present the profiles of the temperature, the velocity, the radiative source term, the Nusselt numbers and entropy generations for a heating position equal to 0°. We can signal that the flow moves from the side leading to the appearance of a single dissymmetric vortex. The velocity value is near zero in the center which explains the uniform profile of the temperature. We also remark the presence of vortex in the radiative source term profile. When the fluid is hot we note that the radiative term source is negative, while for the cold fluid, the radiative source term presents positive values. The radiative volumetric entropy generation presents practically the same profile as that of the radiative source term. Its importance depended on the importance of the radiative exchange. The conductive entropy production is clearly visible especially near the surfaces with higher temperature gradient. The radiative Nusselt numbers and the radiative wall entropy generations present similar aspects, consisting of two half elliptic profiles with distortions boundaries due to the view factor.

Figure 3 shows that in absence of radiation the fluid flow moves easily and rapidly in a main single vortex which explains the increasing velocity profile especially in the center of the enclosure for the heating position lower than -45°. From this position of heat, the velocity value profile decreases progressively due to the transition from two vortexes to a single one. Furthermore, the fluid flow rotation is retarded especially in the center due to the appearance of more than a single vortex. The heating position equal to 90° corresponds to the lowest value of average velocity (stratification). When thermal radiation is taken into account, the average velocity profile presented higher values compared to that computed without radiation.

In figure 4, the average temperature profiles in both cases can be classified into three zones. The first one corresponding to a decreasing profile, is followed by a stagnation phase. The third zone presented a second decreasing profile due to the stratification. The lowest value temperature is located for the heating position of 90°. Taking into account the thermal radiation, the average temperature profile presents greater values compared to those computed without radiation.

As we can see in figure 5, the convective Nusselt numbers computed through the hot wall in both cases with and without radiation presents similar behaviors. In addition, the increase in temperature gradients for heating position lower than 0° leads to a little increase in the profile of the conductive Nusselt number given through the hot wall. A decreasing profile is thus progressively shown to attend a minimum value for the heating position equal to 90° due to the stratified flow. We also note that for the cold wall portion, the corresponding conductive Nusselt number present slightly increasing profiles for heating position lower than 0°, in both cases with and without radiation. Moreover, the reduction of temperature gradient near the cold part leads to a decrease in the profiles of the corresponding conductive Nusselt numbers.

It is worth noting that due to thermodependence, the conductive Nusselt number at the cold wall portion presents the highest values compared to that computed at the heated part. As seen, the variations of heating position seems to have no effect on the evolution of radiative Nusselt numbers where the corresponding profiles at the cold wall portion present the highest values.

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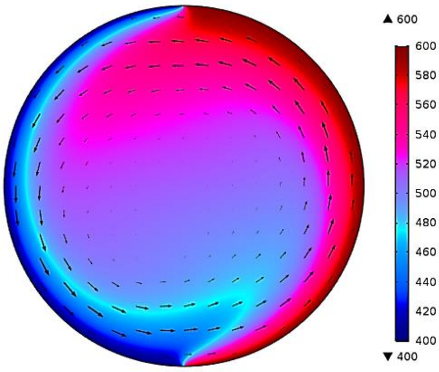
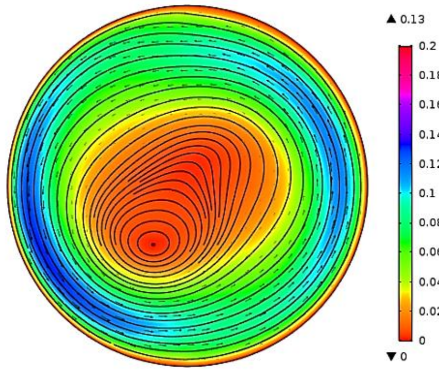
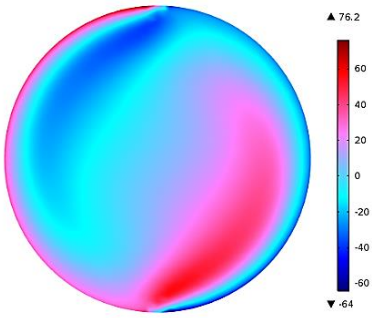
Temperature field (K)



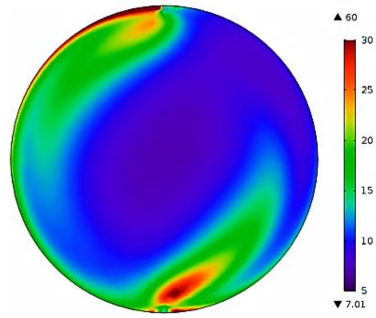
Velocity field (m/s)



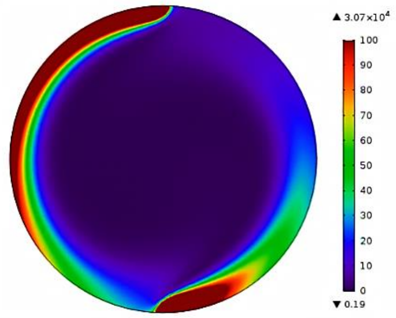
Radiative source term (kW/m³)



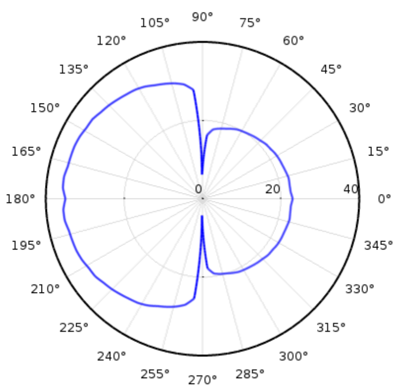
Radiative volumetric entropy generation (W/m³K)



Conductive entropy generation (W/m³K)



Radiative Nusselt number



Radiative wall entropy generation (W/m²K)

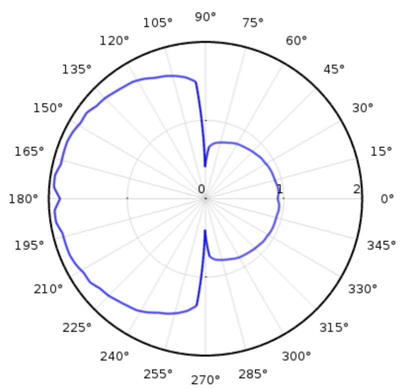


Fig.2 Effect of radiation on the different local parameters for T_h=600K, T_c=400K, R=0.025m, P_0 =1atm, ε=1, $Φ_0$=0°

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As shown in figure 6, the conductive entropy generation profiles present the same behaviors profiles in both cases, with and without thermal radiation. In addition, the entropy generated due to heat transfer is slightly increased for the heating position lower than 0° and it decreases for the heating position higher than 0° to attend the lowest value for the heating position equal to 90°. It can be also noted that the partial heating has no influence on wall and volumetric entropy production. A comparison of entropy generation components permits to conclude that the profiles of wall radiative entropy creation present the highest values, while the profiles of the volumetric entropy production present the lowest values.

As shown in figure 7, the energy efficiency seems to be independent of the negative values of the heating position in absence of radiation. For the positive ones, the energy efficiency increases progressively to attended a higher value for (90°) heating position which represents the best position to improve efficiency. When thermal radiation is taken into consideration, the energy efficiency is reduced and seems to be independent of the heating position. A close look to figure 7 allows us to conclude that the exergy loss is not affected by the heating position. In addition, the radiative contribution has no influence on the exergy losses.

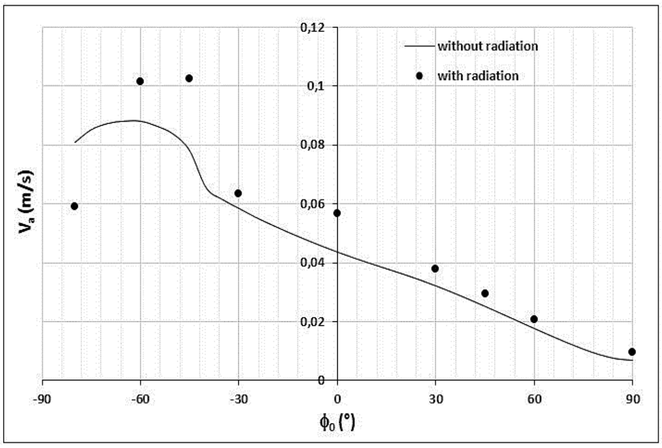

Fig. 3 Effect of heating position on the average velocity T_h=600K, T_c=400K, R=0.025m, P_0 =1atm, ε=1

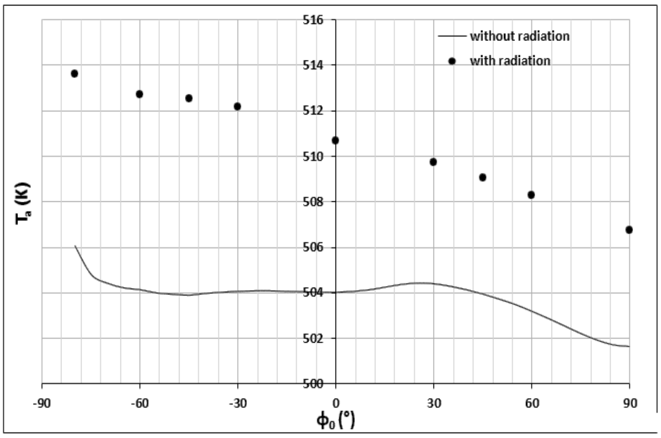

Fig. 4 Effect of heating position on the average temperature T_h=600K, T_c=400K, R=0.025m, P_0 =1atm, ε=1

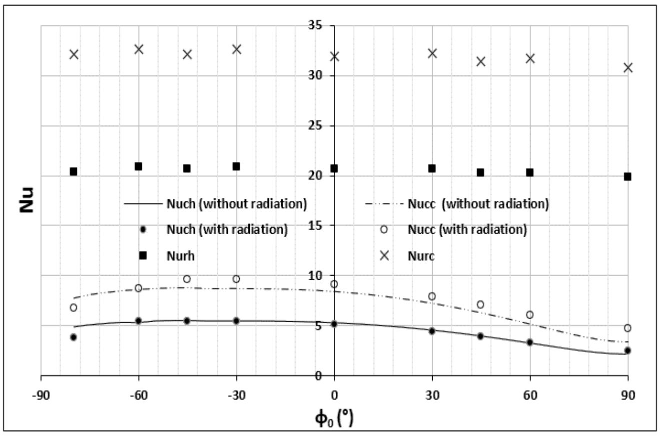

Fig. 5 Effect of heating position on the Nusselt number for T_h=600K, T_c=400K, R=0.025m, P_0 =1atm, ε=1

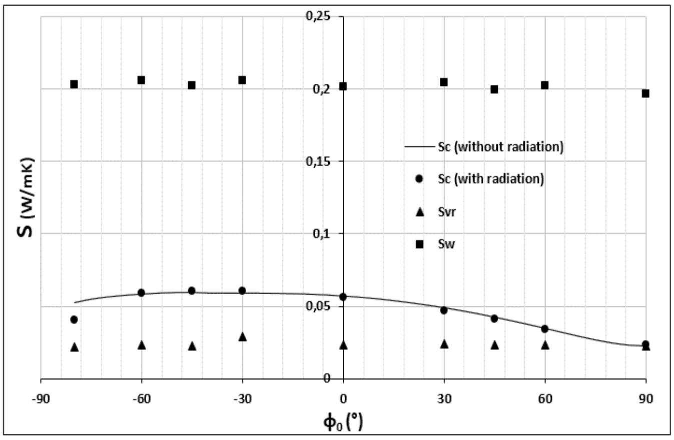

Fig. 6 Effect of heating position on the total entropy generations T_h=600K, T_c=400K, R=0.025m, P_0 =1atm, ε=1

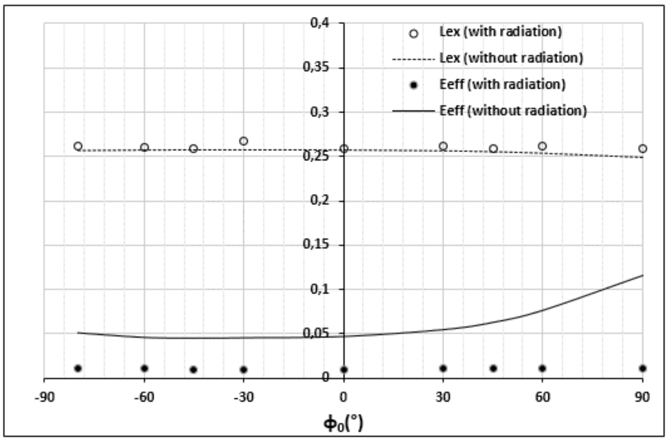

Fig. 7 Effect of heating position on the loss exergy and on the energy efficiency T_h=600K, T_c=400K, R=0.025m, P_0 =1atm, ε=1

IV. CONCLUSION:

In this work, a numerical computation of combined natural convection and non-grey gas radiation through a partially heated cylindrical enclosure is investigated. The participating media is considered as emissive, absorbent and non- scattering.

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The radiative transfer equation is resolved by the used of the Ray-Tracing method associated to the statistical Narrow band correlated-k model. The main subject of this work is to examine the radiative contribution on heat and fluid flow and to estimate entropy generation. Based on the studied results, some important conclusions can be made as follows:

- The thermal radiation enhanced the average temperature and velocity fields.
- The heating position presents significant contributions on the fluid flow structure and especially on the number and the form of the convection cells.
- The presence of radiation reduces the efficiency of the system.
- When radiation is not considered the maximal efficiency is obtained when we heat from the top.
- The exergy loss seems to be independent of the heating position and the presence of radiation.

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Nomenclature

I	: radiation intensity ($wm^{-2}sr^{-1}$)
q	: heat flux density (Wm^{-2})
R	: cylinder radius (m)
P	: pressure (atm)
T	: temperature (K)
l	: optical path (m)
w^i	: weight parameter of the Gauss quadrature point
r	: radial position
s	: local entropy generation ($WK^{-1}m^{-3}$)
S	: global entropy generation (WK^{-1})
r, φ,z	: cylindrical coordinate system
T_a	: Average temperature, $T_a = (T_c+T_h)/2$
Nur	: RadiativeNusselt number
Nuc	: Conductive Nusselt number
Sc	: Conductive Entropy generation
Q_r	: Radiative source term (W/m^{-3})
E_{eff}	: Energy efficiency
L_{ex}	: Loss exergy

Subscripts

h	: hot
c	: cold
a	: average
W	: wall
V	: volumetric
ν	: spectral

Superscripts

b	: black body
i	: partial associated grey-gas

Greek Symbols

κ	: absorption coefficient (m^{-1})
λ	: thermal conductivity (W/mK)
μ	: director cosines
$\vec{\Omega}$	: ray direction
$d\Omega$	: elementary solid angle around $\vec{\Omega}$
$\Delta\nu$	: spectral resolution (cm^{-1})

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