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COMBINED HEAT TRANSFER OF NATURAL CONVECTION-CONDUCTION AND SURFACE RADIATION IN AN OPEN CAVITY HEATED BY CONSTANT FLUX

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ABSTRACT

Combined heat transfer of natural convection-conduction and surface radiation in an open cavity heated by constant flux is studied in this paper. Flow model is laminar and SIMPLE algorithm and QUICK scheme are employed. The relevant parameters are as follows, Prandtl number is 0.7 and dimensionless solid thickness is 0.2, conductivity ratio rangs from 0 to 1000, Rayleigh number ranges from 10^3 to 10^9, surface emissivity ranges from 0 to 1. The numerical results shows secondary circular formed as an effect of radiation which increased the average Nusselt number about from 54.1% to 100.3%.

INTRODUCTION

Natural convection in an open cavity is a topic of significant interest in a range of engineering applications such as building insulation, solar thermal central receivers, geothermal reservoirs, electronic cooling devices, fire spread in rooms, etc.

During the past decades, both experimental and computational studies have been conducted into the cavity-flow physics. These studies were mainly focused on flow and heat transfer of different Rayleigh numbers, aspect ratios, and tilt angles. Also, they studied the occurrence of transition to turbulence and turbulence and how the boundary conditions in the aperture are considered.

In a realistic case with any open cavity system, conduction of solid walls and surface thermal radiation always exist, and can both strongly interact with natural convection, because of the coupling between the surface temperature and the flow fields in the cavity.

Representative studies on combined heat transfer of open cavities may be categorized as natural convection with conductive wall and natural convection with surface radiation.

Lage et al.[1] studied numerically the heat transfer by natural convection and surface thermal radiation in a two-dimensional open top cavity; the numerical approach used by the authors consisted of solving separately the steady state equations of natural convection and thermal radiation, assuming a temperature distribution on the vertical adiabatic wall.

Balaji and Venkateshan[2] obtained steady state numerical results for the interaction of surface thermal radiation with free convection in an open top cavity, whose left wall was considered isothermal, and the right and bottom walls were adiabatic and their temperature distributions were determined by an energy balance between convection and radiation in each surface element of the walls; Radiation was found to enhance overall heat transfer substantially (50–80%) depending on the radiative parameters.

Dehghan and Behnia[3] studied numerically and experimentally the combination between natural convection, conduction and radiation heat transfer in a discretely heated open top cavity; the comparison of the numerical results with the experimental ones showed that the accurate prediction of the flow and temperature patterns depended strongly on the consideration of the heat transfer by radiation.

Ramesh and Merzkirch[4] made an experimental study of the combined natural convection and thermal radiation heat transfer, in a cavity with top aperture; They found out that the surface thermal radiation heat transfer in cavities with walls of high emissivities had a significant change in the flow and temperature patterns and therefore influence the natural convection heat transfer coefficients.

Singh and Venkateshan[5] present a numerical study of steady combined laminar natural convection and surface radiation heat transfer in a two-dimensional side-vented open cavity for different aspect ratios, side-vent ratios, and surface emissivities using air as the fluid medium. Results have been compared with experimental results by Ramesh and Merzkirch[4]. Surface radiation is found to alter the basic flow pattern as well as the thermal performance substantially. The numerical investigation provides evidence of the existence of thermal boundary layers along adiabatic walls of the cavity as a consequence of the interaction of natural convection and surface radiation.

Hinojosa [6] reports a numerical study of the transient and steady-state heat transfer and air flow in a horizontal square open cavity. The results show that the radiative exchange between the walls and the aperture increases considerably the total average Nusselt number, from around 94% to 125%, versus the one with no radiative exchange between walls. Also, for $Ra = 10^7$, the numerical model predicts periodic formation of thermal plumes in the bottom wall of the cavity, and their further movements explain the oscillation behavior of the Nusselt number with time.

Lauriat and Desrayaud [7] studied heat transfer by natural convection and surface radiation in a two-dimensional vented enclosure in contact with a cold external ambient and a hot internal ambient was studied numerically. Inlet and outlet openings were located at a vertical conducting side wall in contact with hot air. Special attention was given on the modeling of the flow and thermal boundary conditions applied at the side wall. The temperature difference between the two vertical facing sides of the enclosure induces downward buoyancy-driven flow of air within the enclosure and boundary layer flow along the surface immersed into the hot internal ambient. The first part of the study focuses on the conjugate problem of natural convection and wall conduction for various thermal resistances of the cold wall, considered as the main floating parameter through the present study. The effects of surface radiation are discussed in the second part of the paper. The radiative exchanges between the facing walls serve to decrease the difference in the averaged temperatures of the vertical walls. Radiation tends to significantly decrease the temperature of the hot wall while the increase in temperature of the cold wall is less important. The interplay between convection and radiation is discussed and it is shown that radiation contribution in heat transfer dominates for all of the cases investigated.

Polat and Bilgen[8] reported conjugate heat transfer in inclined open shallow cavities has been numerically studied. A thick wall facing the opening is heated by a constant heat flux, sides perpendicular to the heated wall are insulated and the opening is in contact with a fluid at constant temperature and pressure. Conjugate heat transfer by conduction and natural convection is studied by numerically solving equations of mass, momentum and energy. The governing parameters were: Rayleigh numbers, Ra from 10^6 to 10^{12}, conductivity ratio, k_r from 1 to 60, cavity aspect ratio, A=H/L from 1 to 0.125, dimensionless wall thickness, ℓ/L from 0.05 to 0.20 and the inclination angle, from 0° to 45° from the horizontal. Isotherms and streamlines are produced; heat and mass transfer is calculated. It is found that volume flow rate, is an increasing function of Ra, A, ℓ/L, , and a decreasing function of k_r. Heat transfer, Nu is an increasing function of Ra, ℓ/L, and a decreasing function of k_r. A mixed pattern is found with respect to A. In the former, Nu is an increasing function of the aspect ratio up to a critical Rayleigh number, above which the relationship changes and it becomes a decreasing function of A. In the latter case, Nu is a decreasing function at low Raleigh numbers and an increasing one at high Rayleigh numbers.

Desrayaud and Lauriat[9] studied a numerical study of natural convection generated by a cold vertical wall of an enclosure with two openings on the opposite wall of finite thickness is presented. The enclosure is connected to an infinite reservoir filled with hot air. A two-dimensional laminar flow is assumed both within the enclosure and along the side of the bounding wall immersed into the reservoir. The effects of the size of the openings, spacing between the vertical walls and thermal resistance of the bounding wall are investigated. Numerical results are discussed for aspect ratios of the enclosure and Rayleigh numbers relevant to practical applications.

Müftüoğlua and Bilgen[10] determined optimum position of a discrete heater by maximizing the conductance and then studied heat transfer and volume flow rate with the discrete heater at its optimum position in open cavities. We found that the global conductance is an increasing function of the Rayleigh number, the conductivity ratio, and a decreasing function of the wall thickness. Best thermal performance is obtained by positioning the discrete heater at off center and slightly closer to the bottom. The Nusselt number and the volume flow rate in and out the open cavity are an increasing function of the Rayleigh number and the wall thickness, and a decreasing function of the conductivity ratio. The Nusselt number is a decreasing function of the cavity aspect ratio and the volume flow rate is an increasing function of it.

Only two (Dehghan and Behnia[3], Lauriat and Desrayaud[7]) of the above studies dealt with all the three heat transfer of conduction, convection and radiation. However, they only used one-dimensional conduction of in the solid wall and did not concern the effect of wall conductivity on heat transfer.

This paper presents a numerical study into the problem of combined heat transfer of natural convection, conduction and surface radiation in an open cavity heated by constant flux. The study was based on the solution of the unsteady 2D incompressible Navier-Stokes equations by using finite volume method, and the surface thermal radiation equations are solved simultaneously. The results discussed in the present paper included the effects of Rayleigh numbers, surface emissivity, conductivity ratio, and the thickness of the conductive solid walls on the flow field.

PROBLEM AND MATHEMATICAL MODEL

1 Problem description

The schematic of the heat transfer in a 2D square open cavity considered in the present investigation is shown in Fig. 1. The opposite wall to the aperture was kept to a constant heat flux q, while the surrounding fluid interacting with the aperture was fixed to an ambient temperature T_∞. The walls perpendicular to the heated wall are conductive and insulated at their outer surfaces. The thermal fluid was assumed to be air (*Pr*=0.7) and Newtonian, and the fluid flow was considered to be laminar. The properties of the fluid were assumed to be constant, except for the density in the buoyancy force term in the momentum equations, according to the Boussinesq approximation. The thermal fluid was considered to be radiatively nonparticipating, and the surface of walls of the cavity and the aperture were considered as gray bodies. Since radiative calculations are carried out, q is fixed at 200 w/m^2 and T_∞ is fixed at 300 K. The open boundary is assumed to be a black surface at ambient temperature.

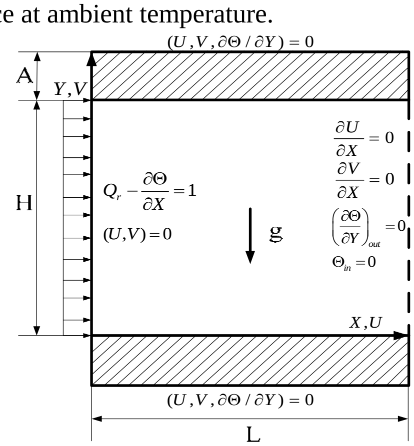

Fig. 1 Physical model

2 Governing equations

The transient-state dimensionless conservation equations governing the transport of mass, momentum, and energy in primitive variables are expressed as

$$\frac{\partial U}{\partial \tau}+U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\sqrt{\frac{\mathrm{Pr}}{\mathrm{Ra}}}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) \tag{1}$$

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\sqrt{\frac{\mathrm{Pr}}{\mathrm{Ra}}}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+\Theta \tag{2}$$

$$\frac{\partial \Theta}{\partial \tau}+U\frac{\partial \Theta}{\partial X}+V\frac{\partial \Theta}{\partial Y}=\frac{1}{\sqrt{\mathrm{Ra\,Pr}}}\left(\frac{\partial^2 \Theta}{\partial X^2}+\frac{\partial^2 \Theta}{\partial Y^2}\right) \tag{3}$$

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{4}$$

In the conductive bounding wall, the velocity is set to be zero and the conductive heat transfer is described by

$$\frac{\partial \Theta}{\partial \tau}+U\frac{\partial \Theta}{\partial X}+V\frac{\partial \Theta}{\partial Y}=\frac{k_r}{\sqrt{\mathrm{Ra\,Pr}}}\left(\frac{\partial^2 \Theta}{\partial X^2}+\frac{\partial^2 \Theta}{\partial Y^2}\right) \tag{5}$$

The above equations were nondimensionalized by defining

$$X=\frac{x}{H},\quad Y=\frac{y}{H},\quad \delta=\frac{A}{H},\quad \tau=\frac{U_R}{H}t,$$

$$U_R=\frac{\alpha}{H}\sqrt{\mathrm{Ra\,Pr}},\quad U=\frac{u}{U_R},\quad V=\frac{v}{U_R},$$

$$P=\frac{p}{\rho U_R^2},\quad \Theta=\frac{(T-T_\infty)k_f}{qH}$$

where $Ra=g\beta qH^4/k_f\upsilon\alpha$ is the Rayleigh number, $Pr=\upsilon/\alpha$ is the Prandtl number and $k_r=k_s/k_f$ is the solid to fluid thermal conductivity ratio.

3 Surface radiation calculations

When radiative interchanges among surfaces are accounted for, the thermal boundary conditions at the solid walls must include the contribution of the net radiative flux.

The general radiosity equation for the *i*th subdivision of radiative surface may be written as

$$J_i=\varepsilon_i\sigma T_i^4+\left(1-\varepsilon_i\right)\sum_{j=1}^{NF}J_jF_{ij}\qquad\left(\mathrm{i}=1,2,\cdots,\mathrm{NF}\right) \tag{6}$$

Where ε_i is the emissivity of the surface subdivision, F_{ij} is the view factor from the *i*th subdivision to the *j*th subdivision of the cavity and if i=j F=0, while NF is the total number of subdivisions along the cavity.

The net radiative heat flux (q_r) for the *i*th subdivision of any of the walls of the cavity was calculated by

$$q_{r\,i}=\sum_{j=1}^{NF}(J_i-J_j)F_{ij}\qquad\left(\mathrm{i}=1,2,\cdots,\mathrm{NF}\right) \tag{7}$$

2.4 Boundary conditions

From Fig. 1 which shows the general layout of the problem, the boundary conditions are:

(1) Along the solid-fluid interfaces of the cavity

Energy balances between convection, conduction and radiation

$$-k_r\left(\frac{\partial\Theta}{\partial Y}\right)_s=-\left(\frac{\partial\Theta}{\partial Y}\right)_f+Q_r \tag{8}$$

$$U=V=0$$

at $Y=\delta,\ 1+\delta$ and $0\le X\le 1$

where $Q_r=q_rH/k_f\left(T_H-T_\infty\right)$ is the dimensionless net radiative heat flux.

(2) Along the horizontal surface of conductive wall

$$\frac{\partial\Theta}{\partial Y}=0$$

$$U=V=0$$

at $Y=0,\ 1+2\delta$ and $0\le X\le 1$

(3) Along the vertical surface of conductive wall

$$\frac{\partial\Theta}{\partial X}=0$$

at $X=0,\ 1$ and $0\le Y\le\delta;\ 1+\delta\le Y\le 1+2\delta$

(4) Along the heated wall

$$-\frac{\partial \Theta}{\partial X} = 1$$

at $X = 0$, and $\delta \le Y \le 1+\delta$

(5) Along the opening

$$\frac{\partial \Theta}{\partial X} = 0 \text{ if } U \ge 0 \text{ or } \Theta = 0 \text{ if } U < 0$$

at $X = 1$, and $\delta \le Y \le 1+\delta$.

The definition of the local heat transfer coefficient and consequent Nusselt number is as

$$h = \frac{q}{T - T_\infty} \quad (9)$$

$$\mathrm{Nu}_Y = \frac{1}{\Theta} \quad (10)$$

The average total Nusselt number is defined as the reciprocal of the average dimensionless temperature

$$\mathrm{Nu}_t = \frac{1}{\int_{\delta}^{\delta+1} \Theta \, dY} \quad (11)$$

For the problems heated by contant flux, average total Nusselt number can also indicate the performance of heat transfer. The haigher it is, the lower the average temperature of the heated wall. Similarly, The average convective Nusselt number and the average radiative Nusselt number are defined as

$$\mathrm{Nu}_c = \frac{-\int_{\delta}^{\delta+1} \left(\frac{\partial \Theta}{\partial X} \right)_{X=0} dY}{\int_{\delta}^{\delta+1} \Theta \, dY} \quad (12)$$

$$\mathrm{Nu}_r = \frac{\int_{\delta}^{\delta+1} \left(Q_r \right)_{X=0} dY}{\int_{\delta}^{\delta+1} \Theta \, dY} \quad (13)$$

The Nusselt numbers can all be regarded as the dimensionless heat flux divided by the average difference of the dimensionless temperature.

NUMERICAL TECHNIQUES AND CODE VALIDATION

The transport equations were discretized on structured control volumes using a second-order QUICK-type scheme for the convection terms and by employing the SIMPLE pressure correction algorithm. The fully implicit scheme was used for the time discretization. The energy and momentum equations were solved by Alternating Direction Implicit (ADI) method. ADI leads to a tri-diagonal matrix which was solved with the Tri-diagonal Matrix Algorithm (TDMA). Inner iterations are used to account for the non-linearity of thze equations in every time step.

Due to the coupling of natural convection, conduction and surface thermal radiation [as indicated in Eq. (8)], the energy balance was solved at every iteration step using an iterative method. Also, the radiosities, the net radiative fluxes and the solid-fluid interface temperature were updated consequently.

In order to validate the numerical code, pure natural convection with Pr=1 in a square open cavity was solved by setting both the thermal conductivity of the wall k_r and the surface emissivity ε to 0, and the results were compared with those reported by Chan and Tien[11], obtained with an extended computational domain. In Table 1, a comparison between the average Nusselt numbers is presented, which shows that the highest percentage difference was 11.4% for Ra=10^3 and the lowest was 0.1% for Ra=10^5, with an average percentage difference of 2.8%. For all the Rayleigh numbers studied, steady state was reached.

Table 1. Comparison of the heat transfer results for the square open cavity with Pr=1

Ra	Nu		Difference
	This work	Chan and Tien [11]	
10^3	1.19	1.07	11.4%
10^4	3.43	3.41	0.6%
10^5	7.68	7.69	0.1%
10^6	15.11	15.00	0.7%
10^7	28.27	28.60	1.2%

The surface thermal radiative model combined natual convection was also validated by comparing the results with those of Hinojosa and Estrada[7]. To do so, the thermal conductivity of horizontal walls was set to 0 and the temperature of the vertical wall of the cavity was fixed at 310K, while the aperture were maintained to a constant temperature of 300K. The three walls and the aperture were considered as black bodies (with ε=1). The comparison between the average Nusselt numblers of this work and the ones of Hinojosa and Estrada is presented in Table 2. The maximum percentage difference was 9.3%.

Table 2. Comparison of steady state average Nusselt numbers for different Rayleigh numbers

Ra	Nu_c		Difference
	This work	Hinojosa and Estrada [7]	
10^3	1.30	1.19	9.3%
10^4	3.06	2.98	2.7%
10^5	6.52	6.40	1.9%
10^6	12.57	12.43	1.2%
Ra	**Nu_r**		**Difference**
	This work	Hinojosa and Estrada [7]	
10^3	1.75	1.73	1.2%
10^4	4.03	3.72	8.2%
10^5	8.11	8.02	1.1%
10^6	17.40	17.29	0.7%

Ra	Nu_t This work	Nu_t Hinojosa and Estrada [7]	Difference
10^3	3.05	2.92	4.5%
10^4	7.09	6.70	5.8%
10^5	14.63	14.42	1.5%
10^6	29.98	29.72	0.9%

In the present study, under-relaxation parameters were introduced for velocities, temperature and net radiative fluxes to control the advancement of the solution field.

Non-uniform grid in X and Y direction were used for all computations. Grid convergence was studied for the case of δ=0.2, ε=0.6, k_r=0.4 with grid sizes of 50×70, 60×84, 70×98, 80 × 112, 90 × 126, 100 × 140 at $Ra=10^6$. Grid independence was achieved within 0.008% in convective Nusselt number and 0.12% in radiative Nusselt number with grid size of 60×84. Grid size distributed within the solid domain is due to its thicknesses (i.e. for a fixed Rayleigh number with δ=0.2, it was 12 in solid and 60 in fluid).

RESULTS AND DISCUSSION

The histories of the average convective Nusselt numbers for Rayleigh numbers of 10^7,10^8, and 10^9 are shown in Figure 2 when k_r=400 and ε=0.7 . As can be seen, for all Rayleigh numbers, the total average Nusselt numbers decrease with time until steady state is reached. The average convective Nusselt numbers for $Ra=10^7$, $Ra=10^8$ and $Ra=10^9$ were 22.2, 38.6 and 66.2, respectively. For all cases in this work, the average Nusselt numbers have reached steady state.

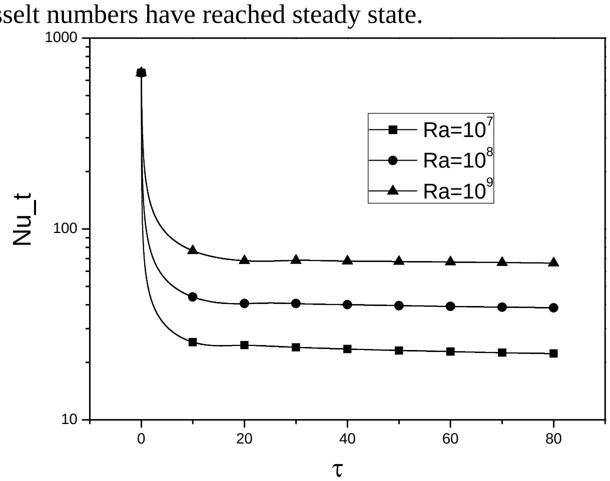

Fig. 2 History of the average convective Nusselt number for $Ra=10^7$ $Ra=10^8$ and $Ra=10^9$

1 The influence of emissivity of solid surface on flow and heat transfer

Fig. 3 shows convective Nu_c varies as a function of emissivity under different conductivity ratio with $Ra=10^6$. Average convective Nu_c decreases linearly with the increase of the emissivity of the solid surface. This can be explained by the fact that the bottom wall was heated through radiative heat exchange and it transfers heat to the entering fluid, increasing its temperature and producing a temperature gradient between the hot wall so the fluid bulk temperature near the wall decreases.

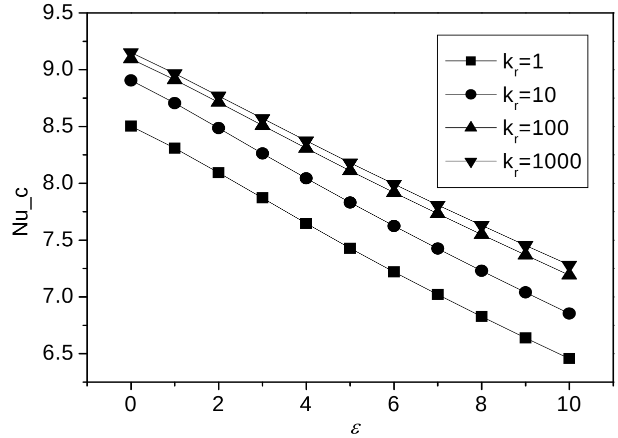

Fig. 3 Convective *Nu* as a function of emissivity with $Ra=10^6$

Table 3 shows the values of the convective, radiative, and total average Nusselt numbers, Nu_c, Nu_r, and Nu_t, as a function of Rayleigh number for 10^3 to 10^7, for thermal emisivities ε=0 (without surface thermal radiation) and ε=1 (with thermal radiation exchange between cavity surfaces). For both ε=0 and ε=1 ,Nu_t increases as the Rayleigh number increases. However, the values of Nu_c for ε=1 were lower compared with those for ε=0 as discribed above. The lowest percentage difference between Nu_c (ε=0) and Nu_c (ε=1) was -22.1% for $Ra=10^7$, and the highest was -1.9% for $Ra=10^3$. Also, it can be seen that Nu_t (ε=0) is always lower than Nu_t (ε=1) for all *Ra*; the maximum percentage difference obtained was 100.3% for $Ra=10^9$ and the minimum percentage difference was 54.1% for $Ra=10^4$. Thus, when the radiative exchange between surfaces and the conduction in the walls are considered, the total Nusselt number, Nu_t, increases between 54.1% and 100.3%. Therefore, consideration of the radiative exchange between surfaces and the conduction in the walls becomes an important factor in obtaining accurate results for horizontal open cavities.

Table 3. Comparison Nusselt numbers for different Rayleigh numbers at k_r=1000

	ε=0			ε=1			diffrerence	
Ra	Nu_t	Nu_c	Nu_r	Nu_t	Nu_c	Nu_r	Nu_t diff.	Nu_c diff.
10^3	2.09	2.09	0	3.22	2.05	1.17	54.1%	-1.9%
10^4	3.08	3.08	0	4.8	2.66	2.14	55.8%	-13.6%
10^5	5.38	5.38	0	8.3	4.43	3.87	54.3%	-17.7%
10^6	9.15	9.15	0	14.31	7.28	7.03	56.4%	-20.4%
10^7	15.17	15.17	0	24.88	11.82	13.06	64.0%	-22.1%
10^8	24.49	24.49	0	44.39	19.69	24.7	81.3%	-19.6%
10^9	39.09	39.09	0	78.28	31.29	46.99	100.3%	-20.0%

The radiative heat transfer from the cavity to the ambience is merely a linear function of the emissivity, and the conductivity ratio has little effect on it, shown as Fig. 4. It is because that the cavity is heated by a constant heat flux, the outer boundary of the conductive wall is thermal isolation, and the length-to-height ratio remains constant that make the view factor to be constant.

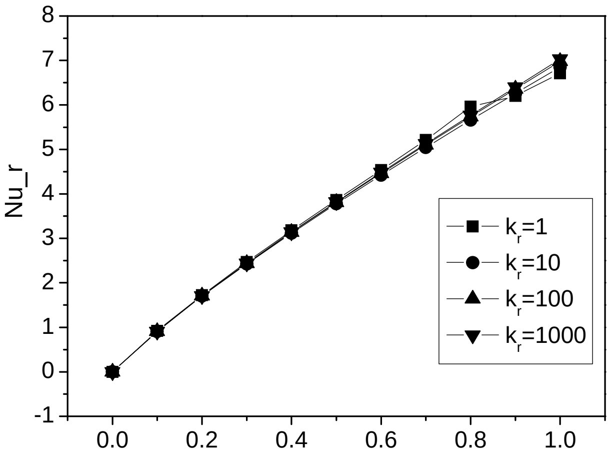

Fig.4 Average radiative *Nu* vs. emissivity under different conductivity ratio as $Ra=10^6$

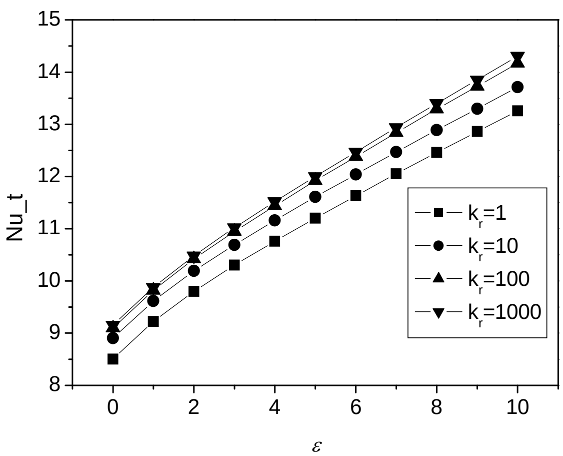

Fig. 5 Average total *Nu* as a function of emissivity at different conductivity ratio with $Ra=10^6$

Fig. 5 shows average total *Nu* variation curve as the emissivity increases at different conductivity ratio while *Ra* is kept at $Ra = 10^6$. The total heat transfer is increased by the increase of the radiative heat transfer increases as a result of the increase of emissivity. The curve indicates that average total *Nu* is a linear function of emissivity while $\varepsilon \geq 0.2$, which is consistent with Fig.2 and Fig. 3. Comparing with the cases at $\varepsilon=0$ (no radiative heat transfer), the average total *Nu* increased relatively about from 55% to 113% at $\varepsilon=1$ (all surfaces are black body).

2 The influence of conductivity on flow and heat transfer

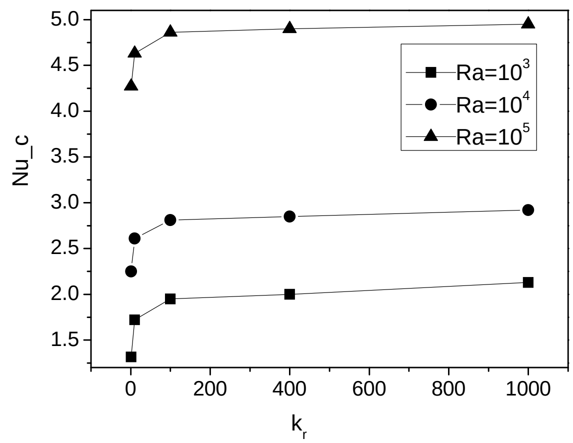

Fig.6 Average convective *Nu* vs. conductivity ratio at $\varepsilon=0.5$ and different *Ra*

The average convective *Nu* curve as a function og the conductivity ratio varying from 0 to 1000 at different *Ra* is shown in Fig. 6, and the emissivity is 0.6. The average convective *Nu* is an increasing function of conductivity ratio. The growth rate of average convective *Nu* inclines to gentle when conductivity ratio k_r exceeds 100. It indicates that the thermal conduction of conductive wall set the total heat transfer up, but its effect is close to a limit when conductivity ratio k_r exceeds 100, and the total heat transfer is hard to be enhanced by increasing the conductivity of the solid.

CONCLUSIONS

Numerical calculations were performed on combined heat transfer of natural convection-conduction and surface radiation in an open cavity heated by constant flux. The governing equations are nondimensionalized and the problem is described by six dimensionless parameters: the Rayleigh number *Ra*, thermal conductivity ratio, k_r, emissivity, ε, Prandtl number, *Pr*, aspect ratio, and thickness of conductive wall.

The effects of emissivity, Rayleigh number, and conductivity ratio on the heat transfer are investigated. The numerical results shows secondary circular formed as an effect of radiation which increased the average Nusselt number about from 54.1% to 100.3%.

NOMENCLATURE

A	thickness of conductive wall
F_{ij}	view factor between surface *i* and surface *j*
g	gravitational acceleration, m/s^2
J	radiosity, W/m^2
k	thermal conductivity, W/m k
k_r	solid to fluid thermal conductivity ratio, $=k_s/k_f$
H	cavity height, m
NF	number of subdivisions
Nu	Nusselt number
p	pressure, N/m^2
P	dimensionless pressure $= p/\rho u_R^2$

Pr Prandtl number
q heat flux, W/m^2
Q_r dimensionless net radiative heat flux $= q_r H / k_f (T_H - T_\infty)$
Ra Rayleigh number $= g\beta q H^4 / k\upsilon\alpha$
t time, s
T absolute temperature, K
u, v velocity components in x and y directions, m/s
U_R reference convective velocity, m/s
U, V dimensionless velocity components in x and y directions
x, y Cartesian coordinates, m
X, Y dimensionless Cartesian coordinates

Greek symbols
α thermal diffusivity, m^2/s
β thermal expansion coefficient, 1/K
ε surface emissivity, dimensionless
Θ dimensionless temperature
υ kinematic viscosity, m/s
ρ fluid density, kg/m^3
σ Stefan-Boltzmann constant, W/m^2 K
δ dimensionless wall thickness
τ dimensionless time

Subscripts
c convective
f fluid
R reference
r radiative
H hot wall
i,*j* *i*th and jth subdivisions
s solid
t total
∞ ambient

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