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Analysis of Heat Transfer in Presence of Non gray Carbon dioxide Gas Subjected to Collimated Irradiation

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Abstract: The current study addresses the interaction phenomenon of combined natural convection with radiation in presence of radiatively active Non gray medium inside a two dimensional enclosure. The left wall is radiatively semitransparent and is subjected to collimated irradiation. All walls are assumed to be gray except, diffuse, except the semitransparent wall which is specular in nature. The variation of properties (k, ρ, μ) with temperature has been taken into consideration to make the investigation realistic. The finite volume discretization has been adopted for numerical simulation. The results are depicted through isotherm and streamline patterns. The effects of influencing parameters on Nusselt Number have also been illustrated.

Keywords: Collimated irradiation, natural convection, Non gray medium, square enclosure. variable fluid property

1. Introduction

There are many engineering applications in which collimated radiation plays an important role. In solar energy application, the primary energy input is produced by collimated radiation. Collimated laser beams are used as the radiation source for measurement systems. The analysis of radiative transfer for an incident collimated beam is also important in astrophysics and atmospheric science.

A considerable portion of heat loss from a typical residence occurs through the windows. The problem is finding an insulating material that is transparent. An examination of the thermal conductivities of the insulating materials reveal that air is a better insulator than most common insulating material. Besides, it is transparent. Therefore, it makes sense to insulate the windows with layer of air. Of course we need to use another sheet of glass to trap the air. The result is an *enclosure*. Other examples of enclosures include wall cavities, solar collectors and cryogenic chambers involving concentric cylinders or spheres.

Enclosures are frequently encountered in practice, and heat transfer through them is of practical interest. Heat transfer in enclosed spaces is complicated by the fact that fluid in the enclosure, in general, does not remain stationary. The fluid adjacent to the hotter surface rises and the fluid adjacent to the cooler one falls, setting a rotation motion within the enclosure that enhances heat transfer through the enclosure.
Present simulation has direct bearing with respect to the optimal design of solar heaters, solar ponds, building heat insulations etc.

In this study radiative transfer in a two dimensional square enclosure, with a collimated beam incident through a semitransparent wall at an arbitary incident angle, is modeled by radiation and natural convection. The participating medium in the enclosure is non gray carbon dioxide gas, it absorbs, emits radiative energy. The radiation portion of the problem is solved by using S-N discrete ordinate method.

1.1 Variable properties of carbon dioxide

Most fluids however, have temperature-dependent properties, and under circumstances where large temperature gradients exist across the fluid medium, fluid properties often vary significantly. Under many conditions, ignoring such variations may cause inaccuracies in estimating heat transfer rates. The thermo physical properties that appear in the governing equations include thermodynamic and transport properties.Thermodynamic properties define the equilibrium state of the system. Temperature, density and specific heat are such properties. The transport properties include the diffusion rate coefficients such as the thermal conductivity and viscosity.

1.2 Literature overview

As far as natural convection with incompressible fluid is concerned, the work of De-vahl Davis [1] (1983), has remained as a benchmark solution till today. Dixit and Babu [2] (2006) have done simulation of high Rayleigh number natural convection in a square cavity using the lattice Boltzmann method. Most fluids however have temperature-dependent properties, and under circumstances where large temperature gradients exist across the fluid medium, fluid properties often vary significantly. Under many conditions, ignoring such variations may cause inaccuracies in estimating heat transfer rates. Natural convection problem, involving buoyancy driven flow in a cavity, was first suggested as a suitable validation test case for CFD codes by Jones [3](1979). Chenoweth and Paolucci [4] (1985) presented exact solutions for a perfect gas using the Sutherland law for viscosity and thermal conductivity and considering the ambient fluid temperature equal to the reference temperature (mean temperature of the two plates). Chenoweth and Paolucci [5] (1986)extended the previous work to cases where the ambient fluid temperature is different from the reference temperature. The presented results in both works are valid for the temperature range between 120 K and 480 K.

The interaction of natural convection with radiation in presence of participating media finds its numerous practical applications in boiler, furnaces, gas-solid suspensions, fire spreading, building insulation systems and other high temperature applications. An excellent review has been done by Viskanta [6] (1987) on interaction of natural convection with radiation from participating media. Chang et.al [7] (1983) numerically simulated combined radiation and natural convection in two dimensional enclosures with partition. Akiyama and Chong[8] (1997)have numerically investigated the interaction of natural convection with thermal radiation of gray surfaces in a square enclosure filled with air. They confirmed that surface radiation significantly altered the temperature distribution and the flow patterns especially at higher Rayleigh numbers. Larson [9] considered fire spreading processes in buildings using a numerical approach. Desresreyaud and Lauriat [10] (1985) numerically investigated natural convection and radiation analysis based on P_1 approximation. Tan and Howell [11] (1991), used the exact integral formulation for radiative transport which was subsequently discretized by the product integration method. They also concentrated on the combined mode of transport in the presence of participating medium in a differentially heated square cavity. Collimated irradiation on to a rectangular medium was investigated by Crosbie and Schrenker [12] (1985) for isotropic scattering The work of M.F.Modest [13] (1993) is significant for understanding the phenomena of Collimated irradiation, while Kim and Lee [14] (1989) demonstrated the accuracy of the high order discrete-ordinate method by applying it to the same problem with anisotropic scattering.. The differential approximation and discrete ordinate method has been blended together considering their strength to generate the method suitable for solving the radiative transport equation employed by Mahapatra et al.[15] (2006). Nouanegue et al.[16] (2009)has investigated the conduction, convection and radiation in the 2D enclosure for heat flux boundary condition at one side. Most earlier works on collimated radiation dealt with solar radiation and other atmospheric or astrophysical application. They are, therefore, limited to one dimensional cases with uniform irradiation of a planar medium

2. Materials and Methods

2.1 Nomenclature

I, Radiation intensity [W/m^2]
I_b Black body radiation intensity (=$\sigma T^4/\pi$)[W/m^2]
I_c collimated irradiation intensity
I_d diffuse radiation intensity
k Thermal conductivity [W m^{-1} K^{-1}]
g Acceleration due to gravity
L Characteristic length [m]
q_0 collimated irradiation heat flux
q_r, q_c Radiation and convection heat flux vector [W/m^2]
RC Radiation-conduction parameter (= $\frac{\sigma T_H^3 H}{k}$)
$\hat{s}$, $\hat{s}'$ Outgoing and incoming direction
T Absolute temperature [K]
T_c Reference temperature(cold wall temperature)
u ,*v* Fluid velocities
X, *Y* co-ordinate
F_X, F_Y Body force in X and Y direction
Nu_T Total Nusselt Number
Nu_r Radiative Nusselt Number
Nu_c Convective Nusselt Number
Pr prandtl number
Ra Rayleigh Number
Cp specific heat at constant pressure
Greek Symbols:
α_a Absorption coefficient [1/m]

σ_s Scattering coefficient [1/m]
β Total extinction coefficient ($\alpha_a+\alpha_s$) [1/m]
Ω Solid angle [Sr]
ω single scattering albedo [α_s/β]
σ Stefan Boltzmann's constant [5.67X10^{-8} w $m^{-2}K^{-4}$]
ε Wall emissivity
τ Total optical depth (= βL)
θ Dimensionless temperature(T/T_c)
ρ density

Subscripts:

c Convection transfer
\r Radiation transfer
w, m wall, medium

2.2 Theory

The aim of this paper is to propose two dimensional CFD code related to natural convection and radiation in a square enclosure one side subjected to collimated radiation another fixed at medium gas temperature with variable properties of carbondioxide. Carbon dioxide absorbs infrared radiation in three narrow bands of wavelengths, which are 2.7, 4.3 and 14.99 micrometers. This means that most of the heat producing radiation escapes it. About 8% of the blackbody radiation is picked up by these finger print frequencies of co_2.

Heinz Hug showed that co_2 in the air absorbs to extinction at its peak in about 10meters.

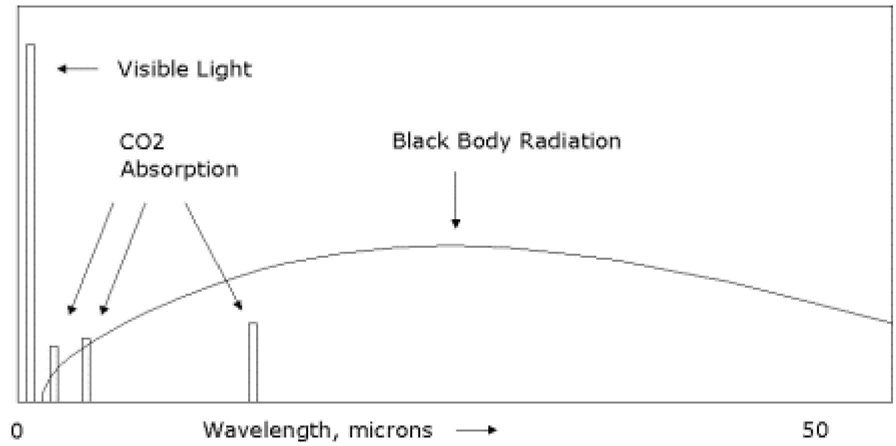


Fig-1 Measured absorption coefficient of carbon dioxide at 2.7, 4.3, 15 micron. The absorption coefficient of carbon dioxide at 15micrometer is 350m-1

The absorption coefficient of carbon dioxide at 4.3micrometer is 280m-1
The absorption coefficient of carbon dioxide at 2.7micrometer is 245m-1
For gray bands in nongray gases the behavior within the gray band is just like the behavior of a gray gas.

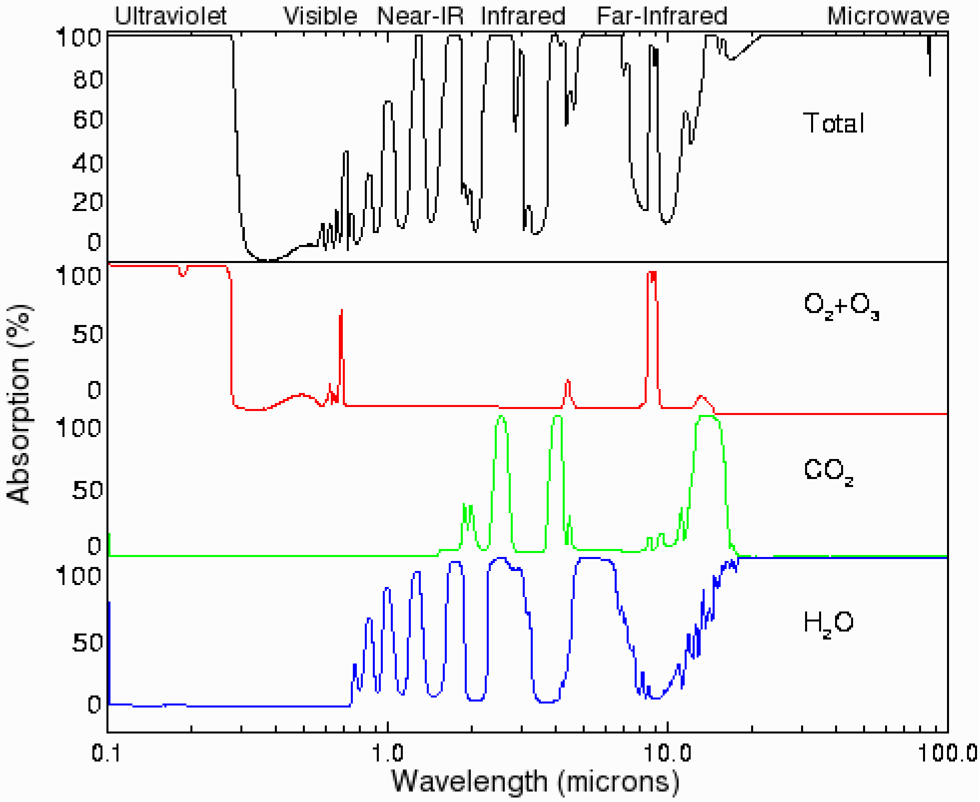


Fig. 2. Absorption of ultraviolet, visible, and infrared radiation by various gases in the atmosphere.

From Fig-2 it can be seen that most of the ultraviolet light (below 0.3 microns) is absorbed by ozone (O_3) and oxygen (O_2). Carbon dioxide has three large absorption bands in the infrared region at about 2.7, 4.3, and 15 microns

A schematic representation of the system under investigation is shown in Fig. 3, where *L* is the dimension of the enclosure which is calculated depending on the Rayleigh number. Refer Table for other parameters for calculation. The gravity vector is directed in the negative *y* coordinate direction. A general analysis is now presented for transport from vertical isothermal surface allowing for the variation of all the fluid properties. The pressure and viscous dissipation terms are neglected.

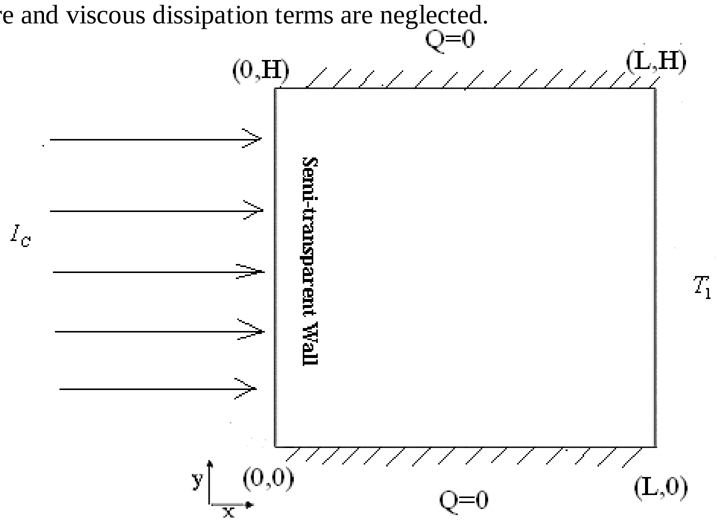


Fig-3 Schematic diagram of the Enclosure

Formulae

The equation for continuity for convective heat transfer is

$$\frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) = 0 \qquad (1)$$

Equations of motion or momentum equation in x, y, direction are

$$\rho\left(\frac{u\delta u}{\delta x} + \frac{v\delta u}{\delta y}\right) = g(\rho_\infty - \rho) + \mu\left(\frac{\delta^2 u}{\delta x^2} + \frac{\delta^2 u}{\delta y^2}\right) \qquad (2)$$

$$\rho\left(\frac{u\delta v}{\delta x} + \frac{v\delta v}{\delta y}\right) = g(\rho_\infty - \rho) + \mu\left(\frac{\delta^2 v}{\delta x^2} + \frac{\delta^2 v}{\delta y^2}\right) \qquad (3)$$

For two dimensional steady state , constant properties without internal heat generation and negligible viscous dissipation, the Energy conservation equation, can be expressed as

$$\frac{u\delta T}{\delta x} + \frac{v\delta T}{\delta y} = \propto\left(\frac{\delta^2 T}{\delta x^2} + \frac{\delta^2 T}{\delta y^2}\right) - \frac{1}{\rho C_P}(\nabla . q_r) \qquad (4)$$

When collimated irradiation is incident upon a participating medium which absorbs, emits and scatters, the angular distribution of the radiative intensities in the medium usually have very strong peaks.

The equation of transfer for an absorbing, emitting and anisotropic ally scattering medium at any location and direction(r, $\hat{s}$) is given by

$$\hat{s}\nabla I(r, \hat{s}) = kI_b(r) - \beta I(r,\hat{s}) + \frac{\sigma_s}{4\pi}\int_{4\pi} I(r,\hat{s}')\emptyset(\hat{s}, \hat{s}')d\Omega' \qquad (5)$$

The intensity at any point is considered to be diffuse part and collimated part.

$$I(r, \hat{s}) = I_c(r, \hat{s}) + I_d(r, \hat{s}) \qquad (6)$$

The collimated part of irradiation is

$$\hat{s}.\nabla_\tau I_c(r, \hat{s}) = -\beta I_c(r, \hat{s}) \qquad (7)$$

The equation of transfer for the non collimated radiation is

$$\frac{1}{\beta}\hat{s}.\nabla I_d(r, \hat{s}) = \hat{s}.\nabla_\tau I_d(r, \hat{s}) = -I_d(r, \hat{s} + \frac{\omega}{4\pi}\int_{4\pi} I_d(r,\hat{s}')\emptyset(\hat{s}, \hat{s}')d\Omega' + (1-\omega)I_b(r) + \omega S_c(r, \hat{s}) \qquad (8)$$

Boundary condition for equation (7) is

$$I_c(r_w, \hat{s}) = [1 - \rho(r_w)]q_0(r_w)\delta(\hat{s} - \hat{s_c}(r_w)) \qquad (9)$$

Boundary condition for equation (8) is

$$I_d(r_w, \hat{s}) = \epsilon I_{bw}(r_w) - \frac{\rho_{r_w}}{\pi}\left[H_c(r_w) + \int_{\widehat{n.s}<0} I_d(r_w, \hat{s}')|\hat{n}.\hat{s}'|d\Omega'\right] \qquad (10)$$

The enclosure walls are all gray and diffusely emitting and reflecting, excluding the semi transparent wall which is specular in nature.

Boundary Conditions

For Momentum equations;

u(x,0) =0 , v(x,0) = 0 , u(x, L) =0, v(x, L) = 0.

u(0, y)=0 , v(0, y) = 0, u(L, y)=0 , v(L, y) = 0

For Energy equation:

$q_0(0,y)=1353 watt/m^2$, t(x, L) = t_c = 300K,

$q_c+q_r=0$ at $0<X<L$, for y=0,H

3. Results And Discussion

For the above domain in Fig.3, left wall is considered hot wall which is subjected to collimated beam heat flux 1353watt/m2 and the cold wall is maintained at 27°C. The other two walls, i.e. top and bottom are considered to adiabatic walls i.e. heat flux $\dot{q} = 0$. In the present study, the total average Nusselt number is calculated as

$$Nu_T = Nu_r + Nu_c = \frac{q_c + q_r}{k\Delta T/L} \qquad (15)$$

The present work involves with two-dimensional analysis of fluid flow and heat transfer in the square enclosure is done using CFD (Fluent) software. The non-uniform meshing has been adopted. Mesh in the core is coarse compared to the size of mesh near the walls. All computations are conducted with 74 X 74 control volumes considering grid independent test. Solution of N-S equation as per Simple algorithm is adopted and PRESTO is chosen for solving pressure equation. Second order upwind scheme for solving momentum equation is used. Under relaxation factor of 0.3 and 0.7 are used in pressure and momentum equations respectively.

Convergence criterion is set as 10^{-5} for continuity, x-momentum and y-momentum equations and 10^{-6} for energy equation, where the convergence criterion is defined as

$$\Delta e_{max} = \frac{\left|\psi^{n} - \psi^{n-1}\right|_{max}}{\psi^{n}} \langle 10^{-6} \tag{16}$$

Variable properties of carbon dioxide is considered as per polynomial approximation and results for average Nusselt number, temperature and velocity profiles are found out.

The temperature and flow field are presented in order to bring clarity in understanding of complex conjugate heat transfer phenomena. From the governing equations it is apparent that Rayleigh number, emissivity, optical thickness are the influencing parameters. The combined effect of varying properties i.e. thermal conductivity (k), density (ρ) and viscosity (μ) has been explained through variation in Nusselt number and both isotherms pattern and streamlines pattern. The properties of working fluid is assumed to vary with temperature adopting polynomial approximation for Ra=10^4. The properties of the carbon dioxide at a temperature of 300K, are taken as the reference values. Rayleigh number is varied within 10^4 to 10^6 in order to study laminar flow regime. All surfaces of the enclosure are considered to be filled with non gray gas and of same emissivity. Effect of the surface radiation is examined by varying surface emissivity from 0.1 to 1.0. Energy balance is checked and the report with respect to heat fluxes refers to cold wall only.

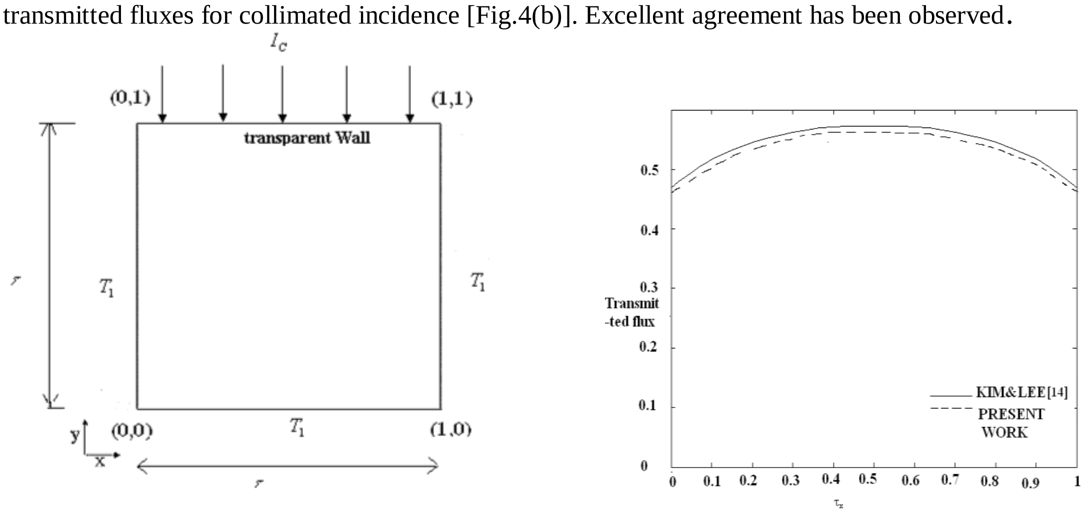
First the computer code was validated with the standard paper for convective heat transfer for differentially heated two dimensional enclosure walls by De Vahl Davis & Dixit &Babu.[2] and presented in Table -1.

Table No.1 The Effect of (Ra) on (Nu) in pure natural convection within differentially heated square enclosure

Raleigh Number	Nusselt No De-Vahl Davis[1]	Nusselt No Present code	Nusselt No Dixit &Babu[2]
Ra=10^3	1.116	1.100	1.128
Ra=10^4	2.242	2.227	2.286
Ra=10^5	4.531	4.486	4.563
Ra=10^6	9.035	8.701	8.800

It is found that the present code is compatible with the benchmark solution.

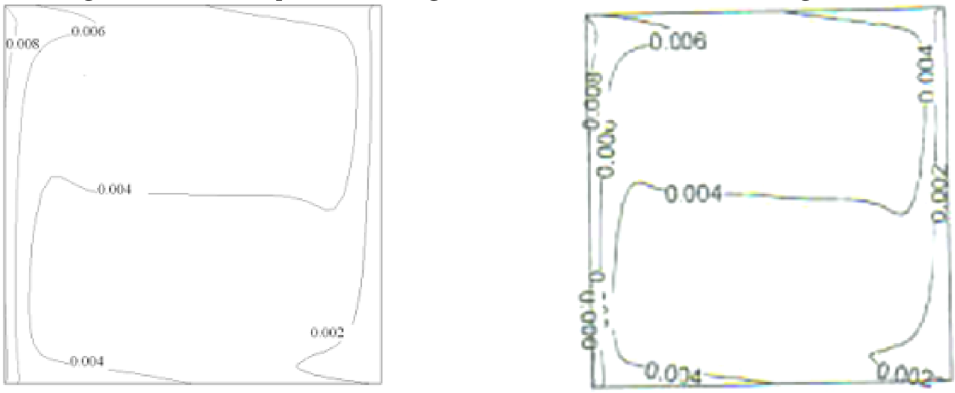
A radiatively equilibrium problem is considered with a uniform collimated irradiation incident on a transparent top wall of a two-dimensional enclosure which is shown in [Fig.4(a)] for validation purpose (i.e., same computational domain adopted by Kim and Lee [14]. The solution obtained for isotropic scattering medium and the variation of transmitted flux along the bottom wall has been validated against the work of Kim and Lee [14] for transmitted fluxes for collimated incidence [Fig.4(b)]. Excellent agreement has been observed.



(a) (b)

Fig.4 (a) Computational Test Domain for the validation purpose (i.e., in accordance with Kim and Lee[14]), (b) Variation of Transmitted Fluxes with τ along the bottom wall, for isotropic scattering media. (ε=1,ω=1,τ_X=τ_Y=1,β=1)

The software has also been used to obtain solution for combined Natural convection and radiation within differentially heated square enclosure with heat flux boundary condition. The work has been validated against Nouanegue et.al [16] and presented in fig-5.The validation is in excellent agreement.



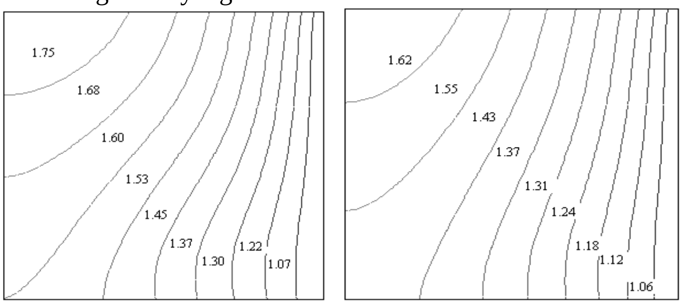
(a) present work (b) Nouanegue et.al

isotherms for A=1, ω=0.0, φ=90^0, ε=1.0, Ra=10^8

Fig- 5 Isotherms for combined natural convection and radiation with heat flux boundary condition(left wall) in square enclosure when ω=0.0, τ=1, ε=1.0, *Ra*=10^8

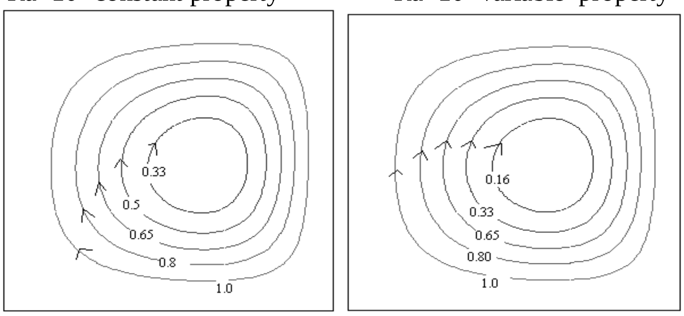
3.1 Effect of Constant and variable Fluid properties

In this section the effect of variable property has been delineated. The flow phenomenon has revealed through isotherm and streamline pattern. The Rayleigh number has been varied from 10^4 to 10^6. In the beginning effect of constant property has been described in order to sense the effect of variable properties. Normally the properties are assumed constant in order to avoid additional non linearities because of variable properties in the complex N-S equation with radiative equilibrium equation. Here flow is considered to be laminar and 2-D. It is observed from fig-6(a), (b), (c), that flow is clockwise inside the cavity. The energy received by the fluid at the hot wall is delivered at the cold wall. The insulated horizontal walls behave as energy corridors for the fluid flow. The isotherms pattern reveals that as Rayleigh number increases, the packing of isotherms near the active walls become prominent implying rise in Nusselt number. The isotherms are orthogonal at the insulated walls ensuring zero heat transfer. The stratification in isotherm pattern across the cavity has become a feature for higher Rayleigh number.



(i) (ii)

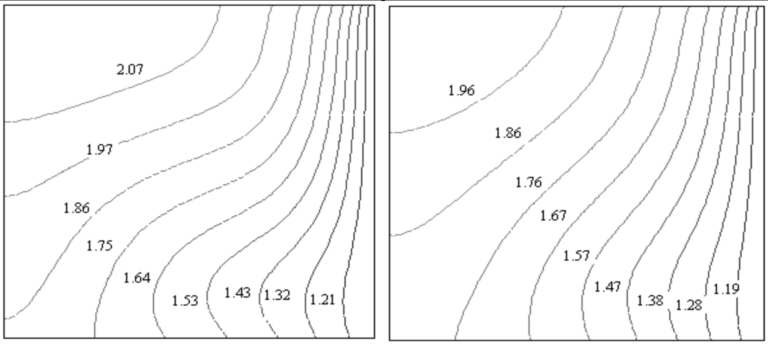
Ra=10^4 constant property Ra=10^4 variable property



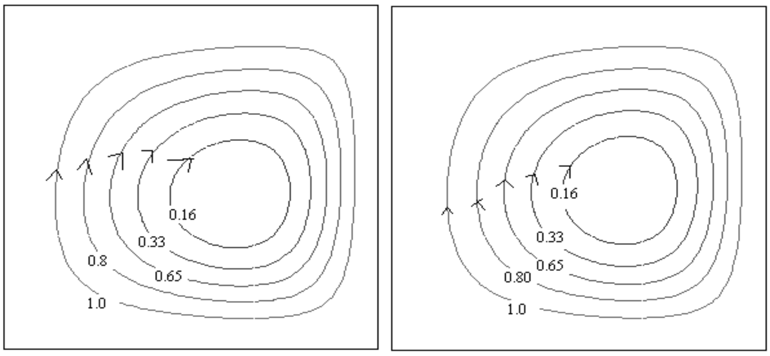
(ii) (iv)

Ra=10^4 constant property Ra=10^4 variable property

Fig.6(a) isotherms and stream line pattern for constant and variable fluid property at Ra=10^4

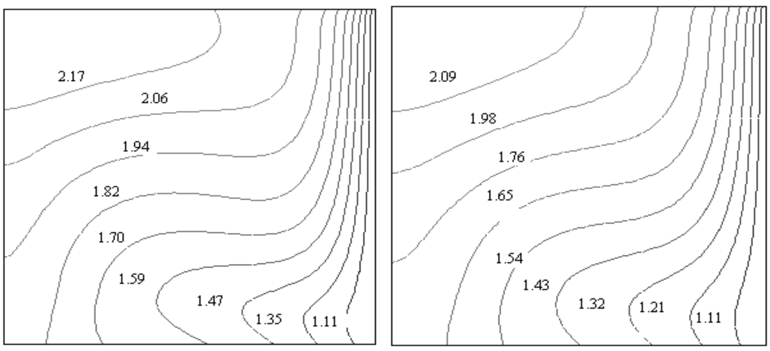


(i) Ra=10^5 constant property (ii) Ra=10^5 variable property

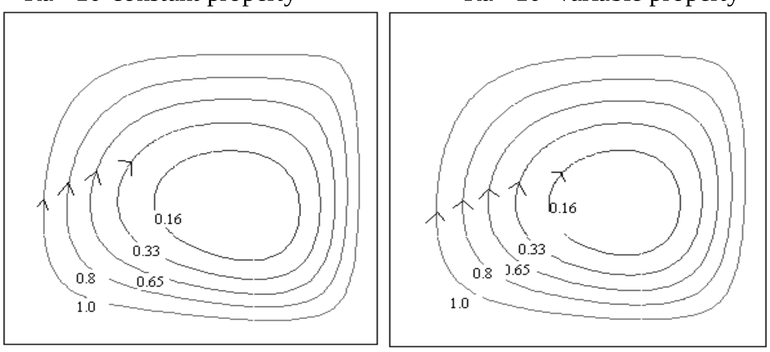


(ii) Ra=10^5 constant property (iv) Ra=10^5 variable property

Fig.6(b) isotherm and stream line pattern for constant and variable fluid property at Ra=10^5



(i) Ra= 10^6constant property (ii) Ra= 10^6 variable property



(iii) Ra= 10^6constant property (iv) Ra= 10^6 variable property

Fig.6(c) isotherms and stream line pattern for constant and variable fluid property at Ra=10^6

Fig-6 Comparison of effect of constant and varying properties (Density,thermal conductivity and viscosity)on Isotherm pattern and flow pattern for (a)Ra=$10^{4.}$ (b) 10^5 , (c) 10^6

Table-2 Effects of Constant and Variable Properties of participating medium on Nusselt Number, with τ=1, ε =0.5

Rayleigh number	Constant property		Variable property Polynomial approximation	
	Nu(radiation)	Nu(convection)	Nu(radiation)	Nu(convection
10^4	0.323011	1.312989	0.5163222	2.5976676
10^5	1.181929	2.945671	1.9274037	5.3960159
10^6	1.94851	4.50212	1.9774051	5.5985486

The graph has been plot for variation of temperature along y-direction at cold wall fig- 7of the enclosure for constant and variable properties. In the variable property collimated irradiation, the total heat transfer to the cold wall is comparatively more and the domain is subjected to less temperature difference.

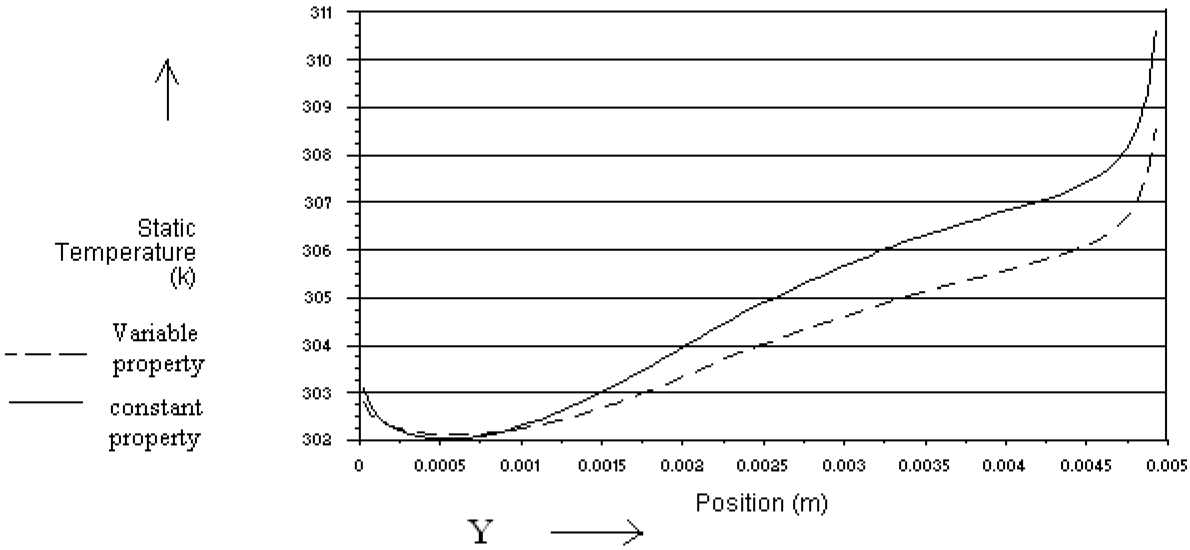


Fig-7 Temperature variation along y direction near cold wall for Ra= 10^5,costant and variable property

3.2 Effect of Variation of surface emissivity variable property (k, ρ, μ)

In order to sense the effect of surface radiation, surface emissivity has been varied considering the properties (ρ, k , μ) to be variable. At ε =0, it is observed that the heat transfer was purely convection in nature. As the emissivity of walls increased from 0.1 to 1, the radiation Nusselt Number increases and the Convective Nusselt Number decreases. The results of variation in surface emissivity are presented in table-3 .

Table-3 Effect of variation of surface emissivity on Nusselt Number for variable property (k, ρ, μ) with τ=1, at Rayleigh number *(Ra)*= 10^5

Variable parameter	Nu_r (Cold wall) variable property	Nu_c (Coldwall) variable property	Nu_T (Coldwall) variable property
ε =0	0	3.70844	3.70844
ε =0.1	0.335077	3.507756	3.84228
ε =0.5	1.255764	3.64763	4.9033951
ε =1	2.26323	3.58317	5.84640

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