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COMBINED NATURAL CONVECTION AND RADIATION HEAT TRANSFER IN A RECTANGULAR ENCLOSURE IN THE PRESENCE OF A POLYDISPERSION AND A NON-PARTICIPATING GAS

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SUMMARY

In this investigation, combined heat transfer due to natural convection and radiation in a vertical rectangular enclosure containing a fluid with a polydispersion is studied. The four walls are assumed to be gray, and diffusely emitting and reflecting. The fluid within the enclosure walls consists of a radiatively transparent gas with water droplets. The water droplets absorb, emit, and scatter radiation. A finite control volume formulation and a two-dimensional discrete ordinates method were used to solve the governing equations. Results are presented to illustrate the effects of radiative heat transfer on the natural convection flow pattern within the enclosure and the heat transfer across the vertical walls.

INTRODUCTION

The problem of combined convection and radiation heat transfer in the presence of polydispersions has received particular attention in applications related to the cooling system of fast reactors, combustion chambers, and environmental processes. Recently, the need to cool electronic equipment in sealed enclosures has added interest to the field.

A similarity relationship was derived by Close and Sheridan [1] for the study of natural convection in enclosures filled with a saturated gas-vapor mixture in the presence of a polydispersion of fine water droplets (a fog). Radiation, however, was not included in that study.

Recent studies of combined natural convection and radiation in rectangular cavities show that the topics of interaction of surface radiation and gray gas participation with natural convection in the absence of scattering have been fairly covered [2-6]. Although it is known to over predict the interaction of natural convection and radiation, most of the work done has been based on the P-1 approximation [4, 7-11]. In one case, a one-dimensional radiation model [5] was used. In only a few cases, the effects of isotropic scattering and non-gray gas (using different spectral techniques) have been examined (mainly CO_2) [7-12]. The method of discrete ordinates has seldom been used in these types of problems [6]. A recent method [13] has yet to be fully tested in situations involving convection-radiation problems. To the best of our knowledge, the complete problem of interaction of natural convection with radiation in rectangular enclosures in the presence of an anisotropically scattering medium remains an unsolved problem. Furthermore, the discrete ordinates method (more accurate than the P-1 approximation) has only been used, in these type of problems, to solve the basic case of participating gray media.

It is of interest to study the effect, normally neglected, of the polydispersion in the overall heat transfer process when radiation is considered. In this paper, the combined effect of natural convection and radiation heat transfer is studied for a two-dimensional rectangular enclosure containing a non-participating gas that is saturated with a Mie-anisotropically scattering polydispersion. Following the work of Close and Sheridan, the equations of conservation of mass, momentum, and energy are rewritten in a form similar to those for a single component. After discretization of the control volume, the resulting elliptic problem is solved by means of an iteration procedure involving a finite control volume analysis and a two-dimensional radiation analysis using the discrete ordinates method. The radiative properties of the droplet cloud are calculated using the Mie theory.

Results showing the relative effects of considering convection alone, combined radiation and convection, and the presence or not of a radiatively participating medium are presented.

ENCLOSURE DESCRIPTION

The system under study is shown in Fig. 1 and consists of a square enclosure with sides of length L. The vertical walls are isothermal at temperatures T_h and T_c, where $T_h > T_c$, and the horizontal walls are adiabatic. The enclosure walls are gray and diffusely emitting and reflecting. The fluid within the enclosure consists of a saturated air-water vapor mixture that is assumed to be radiatively transparent gas and water droplets. The water droplets absorb, emit, and scatter radiation, where the scattering is anisotropic. The flow is laminar, and gravitational acceleration acts parallel to the isothermal walls. Except for the mixture density in the buoyancy term, the mixture properties are assumed to be constant, and the Boussinesq approximation applies. At a given spatial location, the mixture and water droplets are assumed to have the same velocity and temperature. The characteristics of the water droplets are described later.

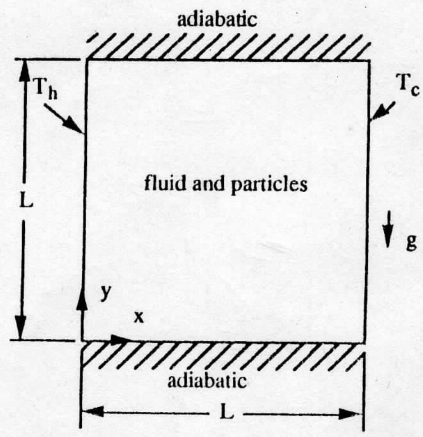


Fig. 1 Schematic of enclosure.

CONSERVATION EQUATIONS

Conservation Equations for a Single Component Fluid. The steady-state conservation equations for a single component fluid in dimensionless form are as follows [14]:

Continuity:

$$\frac{\partial}{\partial \xi}\left(\bar{u}\right) + \frac{\partial}{\partial \eta}\left(\bar{v}\right) = 0 \qquad (1)$$

x-momentum:

$$\frac{\partial}{\partial \xi}\left(\bar{u}\,\bar{u}\right) + \frac{\partial}{\partial \eta}\left(\bar{u}\,\bar{v}\right) = \Pr\left[\frac{\partial}{\partial \xi}\left(\frac{\partial \bar{u}}{\partial \xi}\right) + \frac{\partial}{\partial \eta}\left(\frac{\partial \bar{u}}{\partial \eta}\right)\right] - \frac{\partial \bar{P}}{\partial \xi} \qquad (2)$$

y-momentum:

$$\frac{\partial}{\partial \xi}\left(\bar{v}\ \bar{u}\right) + \frac{\partial}{\partial \eta}\left(\bar{v}\ \bar{v}\right) = \mathrm{Pr}\left[\frac{\partial}{\partial \xi}\left(\frac{\partial \bar{v}}{\partial \xi}\right) + \frac{\partial}{\partial \eta}\left(\frac{\partial \bar{v}}{\partial \eta}\right)\right] - \frac{\partial \bar{P}}{\partial \eta} + \mathrm{Ra}\,\mathrm{Pr}\,\theta \quad (3)$$

Energy:

$$\frac{\partial}{\partial \xi}\left(\bar{u}\ \theta\right) + \frac{\partial}{\partial \eta}\left(\bar{v}\ \theta\right) = \frac{\partial}{\partial \xi}\left(\frac{\partial \theta}{\partial \xi}\right) + \frac{\partial}{\partial \eta}\left(\frac{\partial \theta}{\partial \eta}\right) - \frac{\bar{Q}}{\bar{N}\ (\psi_h - 1)} \quad (4)$$

where ξ and η are dimensionless distances in the x and y directions, respectively; the velocities are normalized with α/L; the pressure is normalized with $\rho\ \alpha^2/L^2$; θ is given by $(T - T_c)/(T_h - T_c)$; Pr is the Prandtl number, Ra is the Rayleigh number; $\bar{N}$ is the conduction to radiation ratio $(k/L\ \sigma\ T_c^3)$; and $\psi = T/T_c$. The fluid properties are denoted by ρ for density, α for thermal diffusivity, and k for thermal conductivity. T is the temperature of the fluid. In Eq. (4), the dimensionless divergence of the radiative heat flux is represented by $\bar{Q}$ $(= L\ \nabla \cdot q_r/\sigma\ T_c^4)$, where q_r is the radiative heat flux vector and σ is the Stefan-Boltzmann constant.

The flow boundary conditions are zero velocities at all surfaces. The thermal boundary conditions become at

$$\xi = 0, \theta = 1\ ;\ \xi = 1, \theta = 0 \quad (5)$$

$$\eta = 0, \frac{\partial \theta}{\partial \eta} = \frac{\bar{q}_r(0)}{\bar{N}\ (\psi_h - 1)}\ ;\ \eta = 1, -\frac{\partial \theta}{\partial \eta} = \frac{\bar{q}_r(1)}{\bar{N}\ (\psi_h - 1)} \quad (6)$$

where the dimensionless radiant surface flux is $\bar{q}_r$ $(= L\ q_r/\sigma\ T_c^4)$.

Conservation Equations for the Mixture. The same equations presented for a single component fluid apply to the mixture provided that saturated conditions exist throughout the cavity, that the cloud of droplets (with the characteristics of a fog) circulates within the gas, and that the effect of the liquid droplets on density and viscosity is negligible. Under these circumstances, the properties and dimensionless groups of the equivalent single component fluid are evaluated as [1,15,16]:

Thermal conductivity: $$k = k_m + \rho_m(1 + m)\ D\ h_{fg}\frac{d}{dT}\left(\frac{m}{1+m}\right) \quad (7)$$

Specific heat: $$C = \frac{dh_m}{dT} - h_l\frac{dm}{dT} + \frac{\rho_l}{\rho_d}\frac{dh_l}{dT} \quad (8)$$

Thermal expansion coefficient: $$\frac{\chi}{\chi_m} = \left[1 - \frac{m\ (M_v - M_d)}{1 + m}\ \frac{h_{fg}}{R\ T}\right] \quad (9)$$

where subscripts m, l, d, and v denote mixture without mass transfer, liquid, non-condensing component, and vapor, respectively; m is the mass ratio of vapor to non-condensing component; D is the diffusivity of gas-vapor mixture; h_{fg} is the enthalpy of vaporization; M stands for molecular weight; and R is the universal gas constant.

Prandtl number: $$\mathrm{Pr} = \frac{\rho_d\ C\ v_m}{k} \quad (10)$$

where v is the kinematic viscosity.

Rayleigh number: $$\mathrm{Ra} = \frac{g\ \chi\ \Delta T\ L^3}{v_m\ \alpha} \quad (11)$$

The properties for the mixture were evaluated using the mass ratio and the individual properties of the vapor and the non-condensing component.

Heat Transfer. The overall heat transfer across the enclosure is expressed in terms of the overall Nusselt number defined by

$$Nu_t = Nu_c + Nu_r \quad (12)$$

The average Nusselt number for natural convection is expressed by

$$Nu_c = -\int_0^1 \left.\frac{\partial \theta}{\partial \xi}\right|_{\xi=0} d\eta \quad (13)$$

and the average radiation Nusselt number is evaluated from

$$Nu_r = \frac{1}{\bar{N}\ (\psi_h - 1)}\int_0^1 \bar{q}_r(\eta)\ d\eta \quad (14)$$

At the cold wall, the minus sign is omitted for the natural convection relation and inserted for the radiation relation. By conservation of energy across the cavity, the overall Nusselt numbers at the hot and cold walls must be equal. Note that although an attempt has been made to separate the overall Nusselt number contributions due to natural convection and radiation, Nu_c is nonetheless dependent on Nu_r, that is, radiant exchange within the enclosure affects the temperature of the adiabatic walls and, therefore, the velocity fields. In turn, the velocity fields affect the temperature gradients at the isothermal walls that are used in the calculation of Nu_c.

RADIATION MODEL

One of the motivations of this paper is to study the influence of a cloud of particles, namely, a fog of water droplets, on the overall heat transfer processes that take place in the enclosure when radiative transfer and natural convection are considered simultaneously. For this purpose, the gas mixture is considered non-participating and, therefore, all the radiation effects are due to surface radiation and to the presence of the polydispersion. The radiation problem to be solved is a two-dimensional problem in the presence of an absorbing, emitting, and anisotropically scattering media surrounded by gray, diffuesely emitting and reflecting walls.

The monochromatic problem. The governing equation of monochromatic radiative transfer is [17]

$$\mu\frac{dI}{dx} + \delta\frac{dI}{dy} + \gamma\frac{dI}{dz} = -\beta\ I + (1 - \omega_o)\ \beta\ I_b + \frac{\omega_o\ \beta}{4\pi}\int_0^{4\pi} I(\zeta,\omega_i)\ \Phi(\lambda,\omega,\omega_i)\ d\omega_i \quad (15)$$

where μ, δ, and γ are direction cosines; I is the radiant intensity; β is the extinction coefficient; I_b is the blackbody intensity; ω_o is the single scattering albedo; ζ is the line of sight of incident radiation; ω_i is the incident solid angle; and Φ is the scattering phase function. Unless otherwise noted, all radiant energy and property quantities are understood to be monochromatic. The discrete ordinates method [18-20] was used to solve Eq. (15) for the intensity. For this method, Eq. (15) is discretized and rewritten, for two-dimensions, as:

$$I_i^p = \frac{|\mu_i|\ A_{n,s}\ I_i^{xr} + |\delta_i|\ A_{e,w}\ I_i^{yr} + a_r\ (S_1+S_2)\ \Delta V_p}{|\mu_i|\ A_{n,s} + |\delta_i|\ A_{e,w} + a_r\ \beta\ \Delta V_p} \quad (16)$$

subject to:
$$I_i^p = a_r\ I_i^{xe} + (1 - a_r)\ I_i^{xr}$$
$$= a_r\ I_i^{xe} + (1 - a_r)\ I_i^{yr} \quad (17)$$
$$= a_r\ I_i^{ze} + (1 - a_r)\ I_i^{zr}$$

where: $$S_1 = (1 - \omega_o)\ \beta\ I_b^p \Rightarrow S_1 = a\ I_b^p \quad (18)$$

$$S_2 = \frac{s}{4\pi} \sum_j w_j I_i^p \Phi_{ij} \qquad (19)$$

In Eqs. (16) to (19), s is the scattering coefficient; a is the absorption coefficient (= β - s); a_r is the spatial interpolating weight; w_j are the weights for the integration procedure; $A_{n,s}$, $A_{e,w}$, and ΔV_p are, respectively, the north-south and east-west areas and the volume of the grid element, where Eq. (16) is being applied; I_i^p is the outgoing intensity in the discrete direction i at the center (p) of that grid element; and superscripts xr and xe indicate reference (where energy originates) and end (where energy arrives) faces for the coordinate direction x.

The phase function is represented in terms of Legendre polynomials of order k:

$$\Phi_{ij} = \sum_{k=0}^{N} (2n+1)\, b_k\, P_k(\mu_i\mu_j, \delta_i\delta_j, \gamma_i\gamma_j) \qquad (20)$$

where b_k are coefficients for the series expansion of the phase function in terms of Legendre polynomials P_k and N is the number of terms.

The boundary conditions for Eq. (16) are at

$$x = 0 \qquad I_i = \varepsilon I_b + \frac{(1-\varepsilon)}{\pi} \sum_{\mu_j < 0} w_j |\mu_j| I_j \qquad \text{for } \mu_i > 0 \qquad (21a)$$

$$x = L \qquad I_i = \varepsilon I_b + \frac{(1-\varepsilon)}{\pi} \sum_{\mu_j > 0} w_j \mu_j I_j \qquad \text{for } \mu_i < 0 \qquad (21b)$$

$$y = 0 \qquad I_i = \varepsilon I_b + \frac{(1-\varepsilon)}{\pi} \sum_{\delta_j < 0} w_j |\delta_j| I_j \qquad \text{for } \delta_i > 0 \qquad (21c)$$

$$y = L \qquad I_i = \varepsilon I_b + \frac{(1-\varepsilon)}{\pi} \sum_{\delta_j > 0} w_j \delta_j I_j \qquad \text{for } \delta_i < 0 \qquad (21d)$$

where ε is the surface emittance. The summations in Eq. (21) are performed over the incoming directions for each surface.

The net radiation fluxes leaving the enclosure walls are evaluated at

$$x = 0 \text{ and } x = L \qquad q_{r,x} = \sum_j w_j \mu_j I_j \qquad (22a)$$

$$y = 0 \text{ and } y = L \qquad q_{r,y} = \sum_j w_j \delta_j I_j \qquad (22b)$$

The divergence of the radiative flux for each control volume is calculated from

$$\nabla \cdot q_r = a\left(4\pi I_b^p - \sum_j w_j I_j\right) \qquad (23)$$

The spectral integration problem. Equation (16) is formulated for a single wavelength (or a gray media). In order to account for the spectral variation of optical properties, an integration of Eq. (16) over the entire spectrum is needed. To this effect, the spectrum was divided into M rectangular wavelength bands, where the radiative properties, evaluated at the wavelength (λ) corresponding to the center of each band, are assumed constant over each band. Integrated quantities are found as the summation over all bands of the individual contributions for each band. Equations (22) and (23) become at

$$x = 0 \text{ and } x = L \qquad q_r(y) = \sum_{\lambda=1}^{M} \left(\sum_j w_j \mu_j I_j\right)_\lambda \qquad (24a)$$

$$y = 0 \text{ and } y = L \qquad q_r(x) = \sum_{\lambda=1}^{M} \left(\sum_j w_j \delta_j I_j\right)_\lambda \qquad (24b)$$

The boundary conditions in Eq. (24) are combined with those in Eq. (6) to evaluate the temperature distribution along the horizontal walls. The total divergence of the radiative flux is calculated from summing Eq. (23) over all bands to yield

$$\nabla \cdot q_r = \sum_\lambda \left[a\left(4\pi I_p^b - \sum_j w_j I_j\right)\right]_\lambda \qquad (25)$$

A computer program (an S-4 implementation of the discrete ordinates method) was written to solve Eqs. (16) to (23) subject to the boundary conditions given by Eq. (21) and to perform the spectral integration.

WATER DROPLET PROPERTIES

Number distribution. The cloud of droplets is assumed to have the number distribution characteristics shown in Fig. 2, where n is the number of particles of radius r, N is the total number of particles per unit volume, and w is is the water content. The particle number density is given by the following relation:

$$n = 3.52\, r^3\, e^{-2r/3} \qquad (26)$$

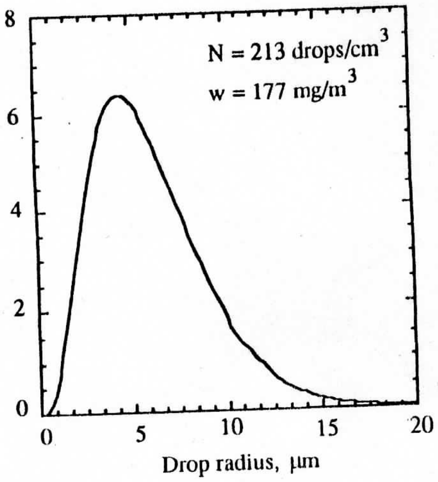


Fig. 2 Characteristics of the polydispersion.

Table 1. Spectral Data

Wavenumber Range, cm^{-1}	Scattering Coefficient, m^{-1}	Extinction Coefficient, m^{-1}
0→ 189 0.0	0.0	
189 → 211	0.00726	0.02394
211 → 236	0.00815	0.02442
236 → 268	0.00957	0.02575
268 → 310	0.01246	0.02915
310 → 367	0.01557	0.03382
367 → 450	0.01790	0.03816
450 → 583	0.01939	0.04071
583 → 833	0.01418	0.03401
833 → 1500	0.02050	0.03069
1500 → 2500	0.03214	0.03903
500 → ∞	0.0	0.0

Under this conditions, the polydispersion has characteristics similar to those of a fog [21].

Optical properties and wavelength discretization. The spectrum was subdivided into twelve bands as shown in Table 1 (M = 12). Detailed calculations were performed for ten of those bands, concentrated in the region between 5 and 50 µm, where most of the radiant energy is concentrated. For each one of these bands, and for the polydispersion represented in Fig. 2, the parameters of interest are calculated from the following relations:

$$\beta = \int_{r_{min}}^{r_{max}} \sigma_{ext}(r) \frac{dn(r)}{dr} \, dr \qquad (27)$$

$$s = \int_{r_{min}}^{r_{max}} \sigma_{sca}(r) \frac{dn(r)}{dr} \, dr \qquad (28)$$

$$\Phi(\theta) = \frac{1}{s} \int_{r_{min}}^{r_{max}} \sigma_{sca}(r) \, \Phi(\theta,r) \frac{dn(r)}{dr} \, dr \qquad (29)$$

The extinction (σ_{ext}) and scattering (σ_{sca}) cross-sections were evaluated from the exact Mie calculations, for a single water droplet of radius r, using wavelength dependent refractive and absorpive

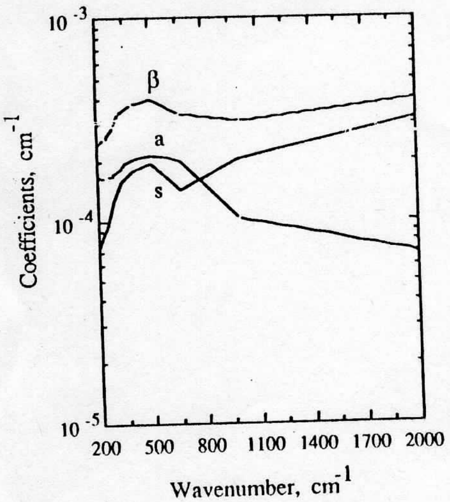


Fig. 3 Volume extinction, absorption, and scattering coefficients for the polydispersion.

indices tabulated by Kondratyev [22] and the computational procedure of Dave [23]. Expansion of the phase function in a series of Legendre polynomials to find the b_k terms in Eq. (20) was performed by means of a procedure similar to that described by Kumar [24].

Results of applying Eqs. (27) to (29) to the polydispersion presented in Fig. 2 are shown in Fig. 3 and Table 1. A qualitative comparison with similar results evaluated by Curran [25] for a C-1 cloud shows excellent agreement.

NUMERICAL PROCEDURE

Numerical solutions for the fluid flow and temperature patterns were obtained using the control-volume formulation and the SIMPLER algorithm described by Patankar [26]. The distribution of the control volumes were skewed along all surfaces in order to resolve accurately large velocity and temperature gradients. The control volumes for the flow and radiation models were identical. Convergence of the numerical solution was checked by performing overall mass and energy balance.

With the exception of the radiation model, where the divergence of the radiative flux given by Eq. (25) is treated as a heat sink term, the code is identical to that used by House, et al. [14]. The accuracy of the calculations for mixture velocities and temperatures is identical to that reported in the cited literature, and is not repeated here.

The resulting computer code for the radiation model was tested, where possible, against known results from other authors [13, 27-29]. With the exception of cases involving highly reflecting wall, where the code (as reported for other S-4 discrete ordinates implementations [27]) tends to over predict irradiations, the results obtained were always very good. Constant wall emittances of 0.8 are used in the present work and, therefore, very good results for both heat fluxes and irradiations can be expected. As an example of the accuracy of the radiation model, representative results for the nondimensional heat transfer rate at the hot surface are displayed in Fig. 4 for an square enclosure containing an isotropically scattering, non-absorbing medium. For the two cases presented (black and gray walls) the emissive power is one for the hot wall and zero for the other walls. The notation of ANDISORD refers to the current radiation model. Tests were performed to check the correct coding of the spectral problem. For these tests, results from a single band problem were compared with the results obtained after subdividing the single band into several rectangular partial bands and integrating. The results were in good agreement.

Convective Nusselt numbers, under the effects of radiation, at the hot and cold wall of a black square enclosure containing a non-scattering gray medium have been recently reported by Kassemi [30]. The combined radiation-natural convection code used in this study was tested against Kassemi [30]. The results of these tests show discrepancies, particularly for the convective Nusselt numbers at the cold wall. These discrepancies accentuate, and become significant, as radiation becomes the dominant mode of heat transfer. Unfortunately, Kassemi does not report radiation Nusselt numbers, and, therefore, energy conservation can not be verified. In all the cases tested with the present model, conservation of energy was satisfied.

Although not required to be, the grids for the flow and radiation models are identical. The results reported were produced using a nonuniform grid with 40 control volumes in each of the x- and y-directions. The grid generation algorithm placed a greater number of control volumes nearer to the surfaces where large gradients occur.

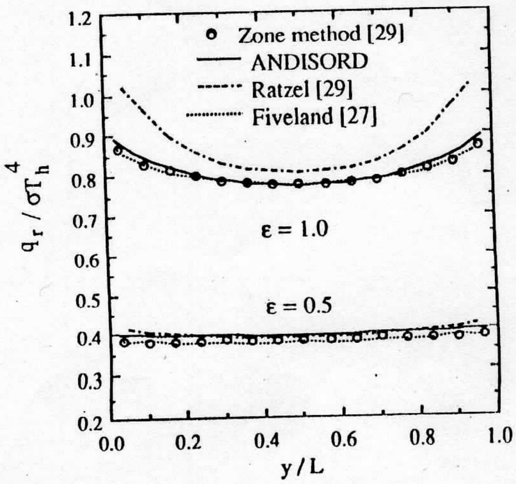


Fig. 4 Radiation model comparisons.

The computations were performed on an ENCORE computer system.

RESULTS AND DISCUSSION

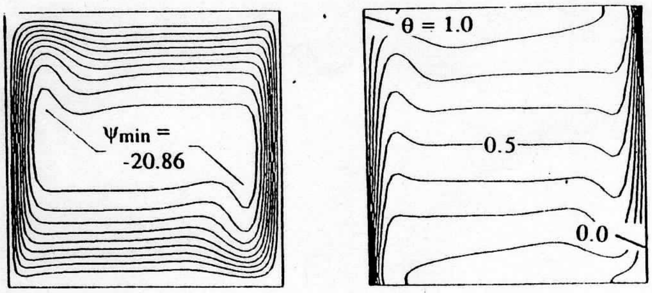
Cases. Two sets of problems were solved, each set consisting of four cases: convection only for pure air, convection only for the mixture air-vapor, convection plus wall radiation for the mixture, and convection plus radiation with the mixture as a participating medium. For both set of problems the length of the cavity was 0.15 m, the average temperature was 300 and 350 K for sets I and II respectively, and the temperature difference between the hot and the cold wall was 5 K. The other properties are given in Table 3.

Table 3. Properties*.

	Air			Mixture		
Set	Pr	Ra, 10^6	k_a, 10^{-3}	Pr	Ra, 10^5	k
I	0.707	1.54	26.3	1.284	15.27	0.1304
II	0.700	1.08	30.0	0.404	8.48	1.8680

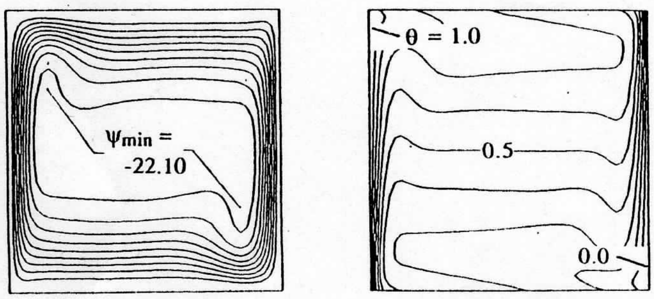
* Units for k are W/m-K

Streamlines and isotherms. Figures 5–8 show the streamlines and isotherms obtained from the solution of the two sets of problems described above. For both convection plus wall radiation for the mixture, and convection plus radiation with the mixture as a participating medium, the streamlines and isotherms are similar, and therefore, only one set is presented. The streamlines and isotherms for air alone are available elsewhere [14] and are not repeated here.



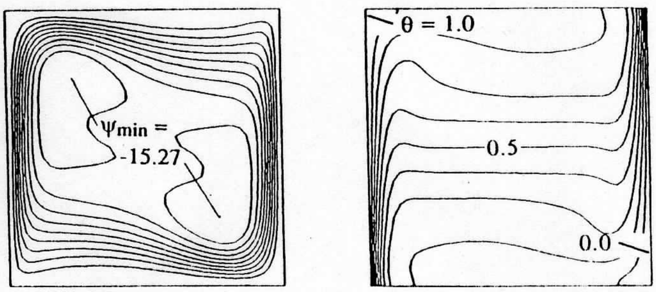
(a) Streamlines (b) Isotherms

Fig. 5 Set I. Mixture. Natural convection alone.



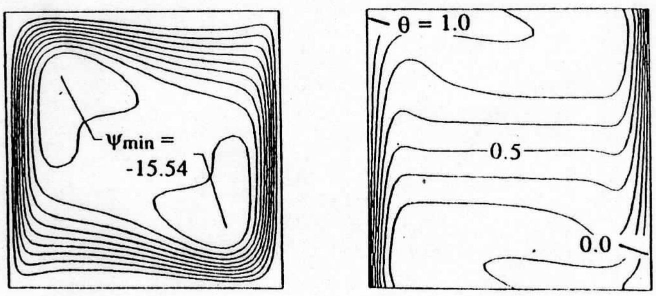
(a) Streamlines (b) Isotherms

Fig. 6 Set I. Mixture. Natural convection–surface radiation.



(a) Streamlines (b) Isotherms

Fig. 7 Set II. Mixture. Natural convection alone.



(a) Streamlines (b) Isotherms

Fig. 8 Set II. Mixture. Natural convection–surface radiation.

In Figures 5-8, Ψ_{min} is the minimum stream function value. Eleven equally spaced isotherms or stream functions are shown in each plot. In all the cases presented, the temperature and flow fields show symmetry and boundary layer characteristics. These results are due to the weak absorbing characteristics of the mixture, the small temperature difference, and the assumption of constant properties throughout the enclosure. When surface radiation is present, (Figures 6 and 8), the characteristic "s" curves appear as a consequence of the radiation exchange at the insulated walls.

Nusselt numbers. Tables 4 and 5 show the Nusselt numbers obtained from the solution of the two sets of problems given in Table 3. Results of applying Eq. (31) (a commonly used correlation [31] for problems involving natural convection alone), are given , for comparison purposes, in Table 4. Recalling the sharp difference in thermal conductivity between air and mixture shown in Table 3, it can be inferred from table 4, that the presence of the mixture produces a drastic improvement in the heat transfer by natural convection.

$$Nu_c = 0.18\left(\frac{Pr}{0.2 + Pr}\,Ra\right)^{0.29} \tag{31}$$

Table 4. Nusselt numbers for natural convection alone.

Set	Air		Mixture	
	Eq. (31)	This study.	Eq.(31)	This study.
I	10.43	9.97	10.72	10.21
II	9.40	9.03	8.392	8.09

Table 5. Nusselt numbers for the mixture.

Set	With surface radiation.			With surface radiation and participating media		
	Nu_c	Nu_r	Nu_t	Nu_c	Nu_r	Nu_t
	Hot wall					
I	9.498	3.543	13.04	9.502	3.547	13.05
II	7.962	0.395	8.357	7.963	0.395	8.359
	Cold wall					
I	9.522	3.519	13.04	9.532	3.516	13.05
II	7.965	0.393	8.357	7.966	0.392	8.359

As reported by other authors [12], Nu_c at the cold wall is always higher than Nu_c at the hot wall when radiation is present. Tables 4 and 5 show that Nu_c at the hot wall, for combined radiation-natural convection, is always smaller than Nu_c for natural convection alone. This result has been reported previously [12] for gray gases, and for the range of Rayleigh numbers used in this study.

For the first set of problems (I), radiation accounts for approximately 27% of the total heat exchange, while for the second set of problems (II), the contribution of radiation to the total heat exchange is less than 5%. This relatively small influence of the radiation on the overall heat exchange [8,12] is due to the temperature difference ratio (Ψ_h -1) which, in set I, is in the order of 0.017 and in set II, in the order of 0.014.

It can be seen in Table 5 that the influence of the participating medium (polydispersion) in the overall heat exchange is negligible. This is due to the small water content (177 mg/m^3) and insignificant optical thickness for the problems under consideration. Similar behavior has been reported [32] for sodium droplets in the same domain of optical thickness and droplet distribution.

That energy is conserved in the numerical procedure can be verified from total Nusselt numbers at the hot and cold walls.

CONCLUSIONS

In this paper, the combined effect of natural convection and radiation heat transfer were studied for a two-dimensional rectangular enclosure containing a non-participating gas that is saturated with a Mie-anisotropically scattering polydispersion. Following the work of Close and Sheridan, limitations were imposed on temperature difference and liquid water content and the equations of conservation of mass, momentum, and energy were rewritten in a form similar to those for a single component. A finite control volume analysis and a two-dimensional radiation model using the discrete ordinates method were used to solve the governing equations. The radiative properties of the droplet cloud were calculated using the Mie theory.

The results of this study show that natural convection is highly enhanced when the mixture is present; the contribution of radiation to the overall heat exchange decreases rapidly with increasing average temperature (for the imposed temperature difference of 5 K, surface radiation accounts for less than 30% of the total heat transfer when the average temperature is 300 K and less than 5% when the average temperature is 350 K); for the water content and droplet size distribution required for this study, the polydispersion has negligible influence on the radiation process.

Flow and temperature patterns, as well as Nusselt numbers, *show qualitative agreement with other studies. Quantitative comparisons with similar studies are impossible at this point, due to* the lack of published work in this area.

The methodology and computer codes described in this work represent a powerful tool for the study of combined radiation-natural convection heat transfer in the presence of particulate media.

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