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Combined convective and radiative heat transfer in side-vented open cavities

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Abstract

Steady, combined laminar natural convection and surface radiation heat transfer from side-vented open cavities with top opening have been studied for different aspect ratios, surface emissivities and side-vents, using air as the medium. A single, isothermal, heated vertical wall formed the heat source for the cavity, and all other walls were insulated. An optical measurement technique using a shearing interferometer with two Wollaston prisms was used to obtain local values of the convective heat transfer coefficient along the heated wall of the cavity. The radiative heat transfer rates were estimated by a numerical procedure, by means of enclosure analysis using the radiosity-irradiation formulation. Parametric studies, for a range of parameters of practical interest, were carried out to understand the effect of interaction of surface radiation on natural convection. Results of a large number of systematic, controlled experiments that were carried out are presented, highlighting the effect of interaction. © 2001 Elsevier Science Inc. All rights reserved.

Keywords: Interferometry; Natural convection; Radiation; Cavity; Electronic equipment cooling

1. Introduction

Natural convection flows caused by buoyancy forces occur in many thermal engineering systems. Combined natural convection and surface radiation exchange between surfaces, bounding a radiatively non-participating medium such as air, occur in numerous engineering applications, such as solar energy collection devices, energy efficient buildings, double-pane windows, cooling of electronic equipment and so forth. Despite significantly lower values of the convective heat transfer coefficient, cooling by natural convection using air is preferred, for example, in numerous electronic cooling applications because of its low cost, inherent reliability, simplicity and noiseless method of thermal control. Since all real surfaces have a non-zero emissivity, there exist radiative interactions among various surfaces that form an enclosure, a cavity or a corner. Therefore, in natural convection systems, even a moderate temperature differential gives rise to significant radiation effects such that the radiative and convective contributions may become competitive and thus influence the total heat transfer rate. It is therefore unrealistic to ignore surface radiation interactions in situations such as those mentioned earlier. In other words, problems involving the interaction of natural convection and surface radiation are of practical interest.

An open cavity can be defined as a cavity with an opening at the top. Heat transfer from an open cavity with different side-vents becomes important because of its application in the area of electronic equipment cooling. Though very simple in design, this geometry presents interesting physics governing the flow and heat transfer from the walls that form the cavity. The radiative interaction between the walls of the cavity and its effect on natural convection heat transfer are of fundamental importance in the design of such systems. The development of thermal boundary layers along the highly emissive adiabatic walls of the cavity provides proof for the existence of radiative interactions among the walls that form the cavity. Hence, neglecting the effects of surface radiation particularly for a cavity with highly emissive walls leads to inaccurate results.

The problem of multi-mode heat transfer is of considerable engineering interest. The importance of the effect of the interaction of surface radiation and natural convection can be found in earlier studies on enclosures by Larson and Viskanta (1976), Balaji and Venkateshan (1993a), Ramesh and Venkateshan (1999a,b) and Yu and Joshi (1999) and studies on channels by Carpenter et al. (1976), Sparrow et al. (1980), Yamada (1988) and Moutsoglou et al. (1992). In fact, some of the above studies also address the effect of conduction heat transfer along with combined natural convection and surface radiation heat transfer.

Studies on heat transfer in open cavities are relatively few. Chan and Tien (1985) numerically analyzed a shallow open

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Notation

A	aspect ratio (d/H)
d	cavity width (m)
g	acceleration due to gravity (m/s^2)
Gr	Grashof number $(g\beta(T_h - T_\infty)H^3/\nu^2)$
H	cavity height (m)
k	thermal conductivity (W/m K)
k_m	thermal conductivity of air at T_m (W/m K)
Nu_c	average convective Nusselt number $(q_c H/(T_h - T_\infty)k_m)$
Nu_r	average radiative Nusselt number $(q_r H/(T_h - T_\infty)k_m)$
Nu_t	total Nusselt number $(Nu_c + Nu_r)$
q_c	convective flux entering the cavity from the left wall (W/m^2)
q_r	radiative flux entering the cavity from the left wall (W/m^2)
Ra	Rayleigh number $(g\beta(T_h - T_\infty)H^3/\nu\alpha)$
T	temperature (K)
ΔT	temperature difference ($T_h - T_\infty$, °C)
w	height of the side-vent (m)
W	dimensionless height ratio of the side-vent (w/H)
x	coordinate along cavity height (m)
y	coordinate along cavity width (m)

Greeks

α	fluid thermal diffusivity (m^2/s)
β	isobaric coefficient of volumetric thermal expansion (K^{-1})
ε	surface emissivity of the walls
ν	fluid kinematic viscosity (m^2/s)
θ	dimensionless temperature $((T - T_\infty)/(T_h - T_\infty))$
ξ	dimensionless distance along height (x/H)
ψ	dimensionless distance along width (y/H)

Subscripts

b	bottom wall
c	convection
h	hot wall
m	mean
r	right wall, radiation
t	total
∞	ambient

cavity having two horizontal insulated walls and one vertical wall that was assumed to be isothermal. The cavity aperture opened on one side to the surroundings at a lower temperature. Isotherms and streamline plots were obtained for a cavity of aspect ratio of 0.143, for Rayleigh numbers from 10^3 to 10^6. Showole and Tarasuk (1993) provide results of experiments using a Mach-Zehnder interferometer, and also numerical results of an open cavity having all walls maintained at the same temperature. Different aspect ratios (0.25, 0.5, 1) and different inclination angles (0°, 30°, 45°, 60°) were considered in their study, for Rayleigh numbers from 10^4 to 5×10^5. Photographs of isotherm patterns and numerically obtained plots along with correlations for Nusselt number for different inclination angles are also presented. Balaji and Venkateshan (1993b, 1994) reported the results of a numerical study of the interaction of surface radiation with free convection in an open cavity by considering the left wall to be isothermal, whereas the right wall and the bottom walls were allowed to attain a temperature that was determined by a balance between convection and radiation. The heat transfer by natural convection and surface radiation in a two-dimensional cavity with one side heated (left wall), one side (right wall) and bottom insulated, having an open top was investigated numerically by Lage et al. (1992). The effect of combined natural convection, conduction and radiation heat transfer in a discretely heated open cavity was considered by Dehghan and Behnia (1996). They concluded that the radiation heat exchange in the cavity has a significant influence on the thermal and flow fields.

A review of the above available literature provides enough evidence of the need for systematic experimental studies in the area of interactive multi-mode heat transfer and fluid flow in open cavities. The presence of side-vents in these cavities provides additional ports for airflow into or out of the cavity. It is therefore informative to explore, in detail, the effect of surface radiation on natural convection and thereby the total heat transfer rate in side-vented open cavities for a range of parameters. The objective of the present study therefore is to obtain better a physical insight into the interaction of multi-mode heat transfer on the thermal and flow fields by experimentally investigating the effect of surface emissivity, aspect ratio and different side wall opening on the convective and radiative heat transfer rates from the cavity for a range of parameters of practical interest.

2. Experimental apparatus and procedure

The open cavity test cell consists of two vertical walls and one horizontal wall. One vertical wall, maintained isothermal ($T_h > T_\infty$), forms the left wall, which is the heat source for the cavity. The other vertical wall, parallel to the left wall is an insulated wall, called the right wall. The bottom wall, which is horizontal, is also an insulated wall. Fig. 1 shows the sectional view of the side-vented open cavity test cell. The left wall,

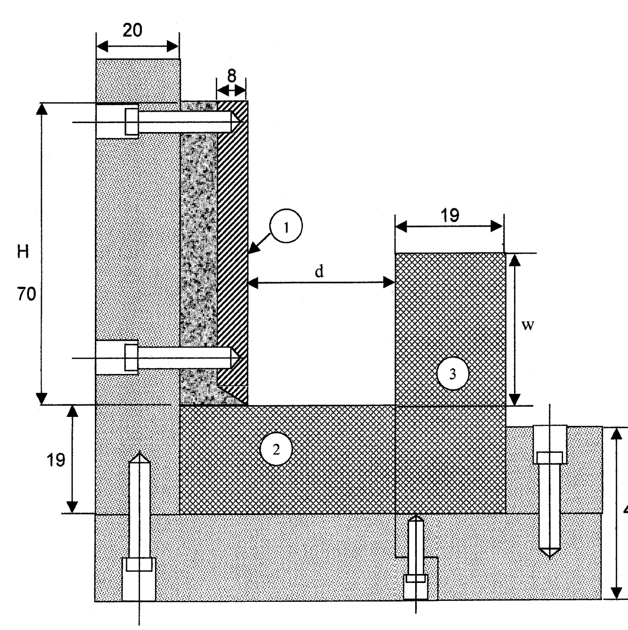

Fig. 1. Sectional view of the test cell (1 – left wall, 2 – bottom wall, 3 – right wall). All dimensions shown in the figure are in mm.

made of aluminum, has the following dimensions: 70 mm height, 8 mm thickness and 350 mm depth. It is provided with a foil heater on one face, energized by means of stabilized electric ac supply through a variable transformer. Different voltage settings of the variable transformer provide different temperatures for the wall. The right and bottom walls were made of 19 mm thick particleboard ($k = 0.078$ W/m K) insulation material. The bottom wall was of different widths in order to obtain two aspect ratios (0.25 and 0.5) for the cavity. The right wall, also made of particleboard is of different heights so as to enable different heights of the side-vents for the cavity. Type T thermocouples were used for measuring temperatures along all the walls. Temperatures were measured at four locations along the left wall and locations at 5 mm intervals along the right and bottom walls. The walls of the cavity were coated with blackboard paint in order to obtain a high value of surface emissivity of 0.85. In order to have highly reflecting walls of very low emissivity of 0.05, the left wall was highly polished to a mirror finish and the insulated walls were covered with a very thin aluminum foil.

The optical instrument used in the present study is a Wollaston prism shearing (or schlieren) interferometer. Being non-intrusive, optical methods provide an accurate description of the flow field and thus enable faithful replication of true phenomena. The heart of the interferometer is a shearing element (Wollaston prism), the use of which produces the required physical separation between the interfering beams of light. Fig. 2 shows the schematic of the optical setup used in the present study. Light from a mercury vapor lamp (S) is allowed to pass through a pinhole, color filter, polarizer and a system of lenses, after which it falls on the Wollaston prism (WP1). The parallel beam of monochromatic light available between the spherical mirrors M1 and M2 provides the test section. The light that passes through the Wollaston prism (WP2) carries information about the temperature gradients in a test cell suitably positioned in the test section. The interferogram is recorded by means of a CCD camera. The purpose of the interferometer is twofold: (i) to enable flow visualization and (ii) to obtain local convective heat transfer coefficients from an interferogram. The visualization of the development of thermal boundary layers on the walls that form the cavity can be accurately obtained by means of the infinite fringe setting of the shearing interferometer. Besides, the inherent property of this interferometer directly provides the gradient of the temperature in the thermal boundary layer along the walls of the cavity, which is measured as the fringe shift. Thus, the use of a shearing interferometer provides a direct measurement of the local convective heat transfer coefficients.

In order to obtain steady-state conditions, it was found that a typical experimental run required about 5 h. As a precaution, even after this time, all temperatures were checked for any significant variations over a period of time (say 15 min) to verify whether steady-state conditions prevailed before recording data. The interferograms and temperatures were recorded only under steady-state condition. The interferograms were analyzed by means of a computer program developed specifically for this purpose, in order to obtain the fringe deflections along the hot wall. These values of fringe deflections were used to evaluate the local convective heat transfer coefficient from the left wall of the cavity. The average value of the convective heat transfer coefficient was obtained from the local values by integration. Detailed exposition on shearing interferometers is available in Merzkirch (1987) and its application to heat transfer studies can be found in Black and Carr (1971) and Sernas and Fletcher (1970). The relatively long dimension (350 mm depth) compared to the height of the vertical wall (70 mm) ensured two-dimensional conditions inside the test cell. The experiments are carried out in the laminar range only, in a separate chamber in the laboratory. This enabled isolation from air currents and other disturbances that could otherwise be detrimental for natural convection as well as the optical setup. The overall uncertainty in the estimated value of the total Nusselt number was found to be within ±5%, determined by methods outlined in Coleman and Steele (1989).

The total heat transferred from the left wall of the cavity is obtained as the sum of the convective and radiative heat transfer components. The convective part is evaluated using the heat transfer coefficient obtained from the interferogram. The radiative part is evaluated numerically, using the measured temperatures of all the walls of the cavity and the required radiation shape factors. The shape factors were evaluated using Hottel's crossed string method. Using the measured temperatures, surface emissivities and shape factors, the radiative heat transfer from the left wall of the cavity is

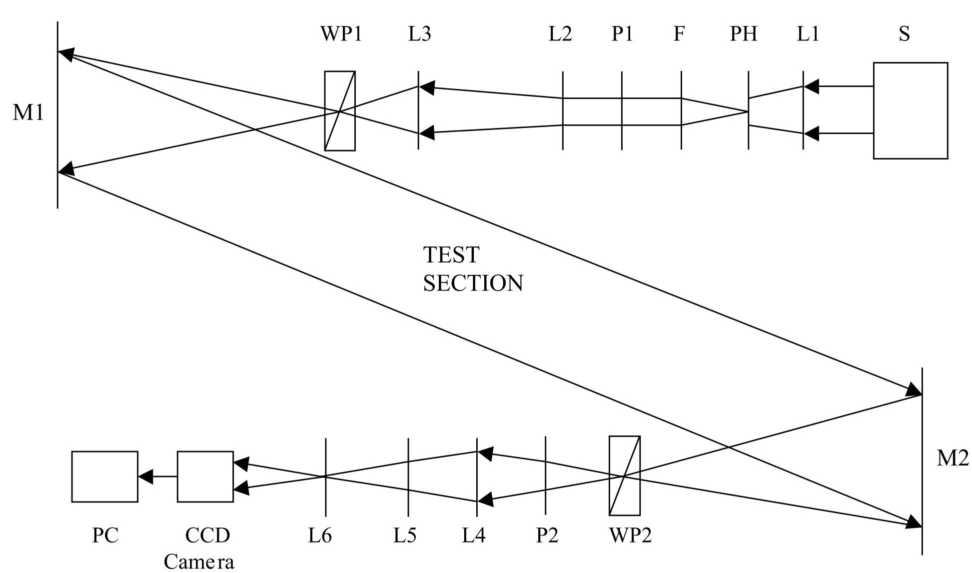

Fig. 2. Optical setup of the schlieren interferometer (S – light source, L1…L6 – lenses, PH – pinhole, F – color filter, P1 and P2 – polarizers, WP1 and WP2 – Wollaston prisms, M1 and M2 – spherical mirrors).

estimated by accounting for radiant heat exchange among all the walls of the cavity, using the radiosity-irradiation formulation (Sparrow and Cess, 1978) for enclosure analysis.

3. Results and discussion

The thermocouples were calibrated and the error estimated as per ASTM Standard (1989). The calibrated thermocouples were assembled in grooves made in the walls, using a high conducting cement. Before embarking on exhaustive systematic parametric studies, the interferometer was calibrated by comparing the values of the local Nusselt number, obtained based on experimentally measured fringe shifts along a hot, isothermal vertical flat plate with the theoretical values obtained from the similarity solution due to Ostrach (1953). This exercise was essential in order to have confidence in the correspondence between measured values of wall temperatures and the associated local fringe deflections along the height of the vertical plate. The comparison was found to be excellent; the rms deviation of the local experimental Nusselt number variation along the height from the theoretical values was found to be 0.29. In terms of the mean value of the Nusselt number, this translates to an error bar of $\pm 4\%$.

Table 1 shows the range of parameters considered in the present study. The combined mode heat transfer in the present geometry provides interesting insight underscoring the importance of the interaction of natural convection and surface radiation. The total heat transfer from the cavity is influenced by: (i) the temperatures of the walls, (ii) emissivity of the walls, (iii) the aspect ratio of the cavity and (iv) the opening provided by the side-vent of the cavity. In what follows, a detailed discussion on the effect of these quantities on the temperature distribution along the insulated walls, the local convective and radiative fluxes along the left wall and the competing modes of heat transfer for different cases considered in the study is given.

Table 1
Range of parameters considered in the present study

Parameter	Range
T_h (°C)	45–80
T_∞ (°C)	20–25
$\varepsilon_h = \varepsilon_r = \varepsilon_b$	0.05, 0.85
Ra	10^5–10^6
H (m)	0.07
d (m)	0.035, 0.0175
w (m)	0.035, 0.0175
A	0.25, 0.5

Fig. 3 shows the graph of the variation of dimensionless temperature with dimensionless distance (height) along the insulated right wall for two different cavities. The two cavities considered here have the same side-vent ($W = 0.5$) but different aspect ratios and wall surface emissivities. The graph shows the effect of surface emissivity and aspect ratio on the right wall temperatures. It can be seen that the average temperature of the right wall is always higher than the temperature. For the case shown in the graph, the average temperature of the right wall is found to be higher than the ambient by about 9°C for a highly emissive cavity with $A = 0.25$, whereas for a cavity with low emissive walls, the average right wall temperature was found to be higher by 3°C than the ambient. The increase in the right wall temperature was found to be dependent on the available temperature differential ($T_h - T_\infty$), surface emissivity of the walls and the width of the cavity or the aspect ratio. It was found from experiments that higher values of the temperature differential, surface emissivity and low aspect ratio of the cavity provided higher values of the average right wall temperature and thereby higher temperatures above the ambient. The variation of temperature along the right wall from bottom to top was found to be linear for all cases except for the case of the cavity with low emissive walls ($\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.05$) and low aspect ratio ($A = 0.25$) considered in the present study. For a cavity with highly emissive walls, higher right wall temperatures result because of increased radiation interchange among the surfaces that form the cavity. Different locations along the walls that form the cavity interact radiatively to varying amounts among themselves thereby affecting the heat transfer within the cavity. Radiation interchange becomes severe as the surface emissivity and the temperatures of the walls increase. For a cavity with

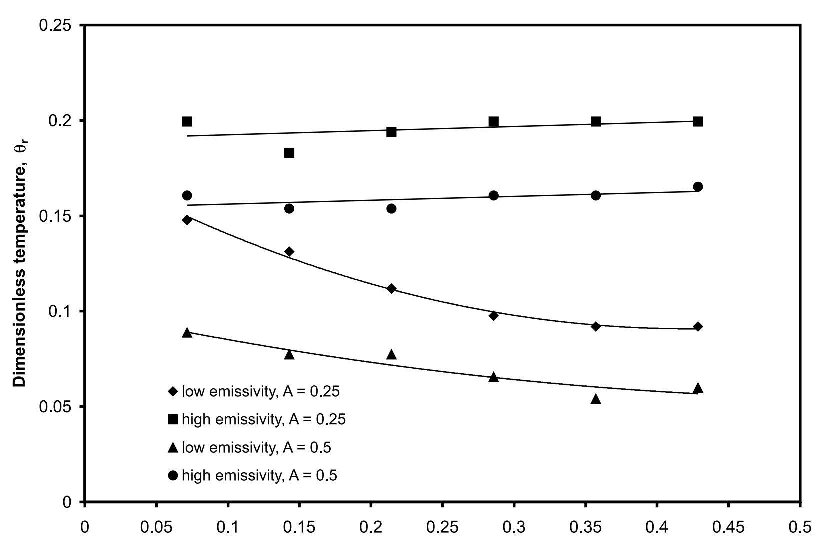

Fig. 3. Effect of surface emissivity and aspect ratio on temperature variation along the right wall ($W = 0.5$, low emissivity – $\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.05$, high emissivity – $\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.85$, $\Delta T = 43.7$°C).

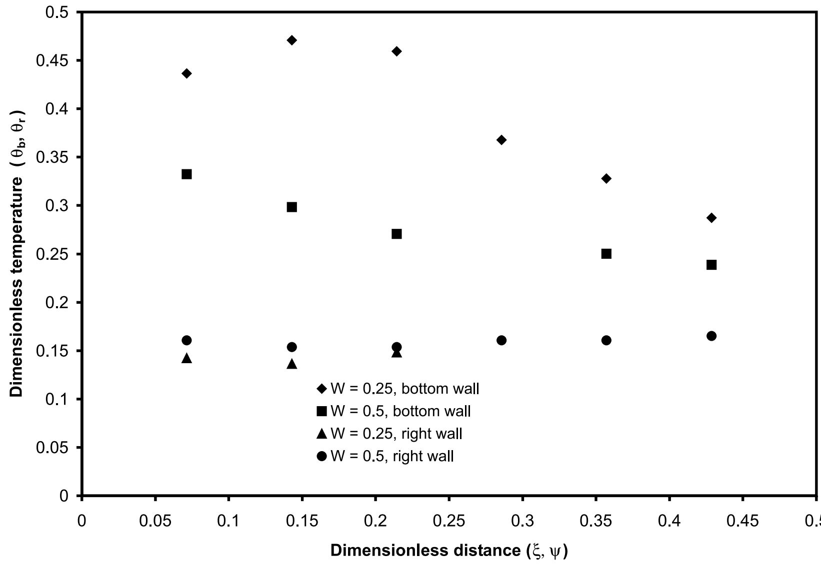

Fig. 4. Effect of side-vent on bottom and right wall temperatures ($\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.85$, $\Delta T = 43°C$).

highly emissive walls, lower aspect ratios also result in higher right wall temperatures.

The height of the side-vent has an influence on the temperature distribution along the bottom and right wall. This can be seen in Fig. 4, which shows the variation of dimensionless temperature along the bottom wall and right wall (side-vent) for two representative cases. Both cavities have the same value of high surface emissivity ($\epsilon_h = \varepsilon_b = \varepsilon_r = 0.85$) but different heights of the side-vents. For the same value of temperature differential maintained in both cases, as the height of the side-vent increases, it was found that the average bottom wall temperature was lower for a cavity with higher side-vent than that of the corresponding case for a cavity with lower height of the side-vent. Interestingly, because of surface radiation, as the height of the side-vent increases, the right wall was found to be at a slightly higher average temperature. The effect of side-vent on the local convective and radiative heat flux along the hot, isothermal left wall of the cavity is shown in Fig. 5. The graph shown here is for a representative case, encountered in the present study. For a cavity with low emissive walls ($\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.05$), the average convective heat flux along the hot wall was found to be higher for a cavity with higher side-vent than the corresponding case of the cavity with a lower side-vent. As can be seen from the graph, the average radiative heat flux was found to be not significantly different for the two cases. For a cavity with highly emissive walls ($\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.85$), it was found that the height of the side-vent has negligible influence on the convective and radiative heat fluxes along the left wall. However, if one were to compare two cavities with the same height of the side-vent, but with different surface emissivities, it was found that the surface emissivity of the cavity walls has a profound effect on the convective and radiative heat fluxes along the hot wall. This can be inferred from Fig. 6. As can be seen in Fig. 6, the higher value of surface emissivity causes a reduction of the convective heat transfer from the left wall, but at the same time, causes

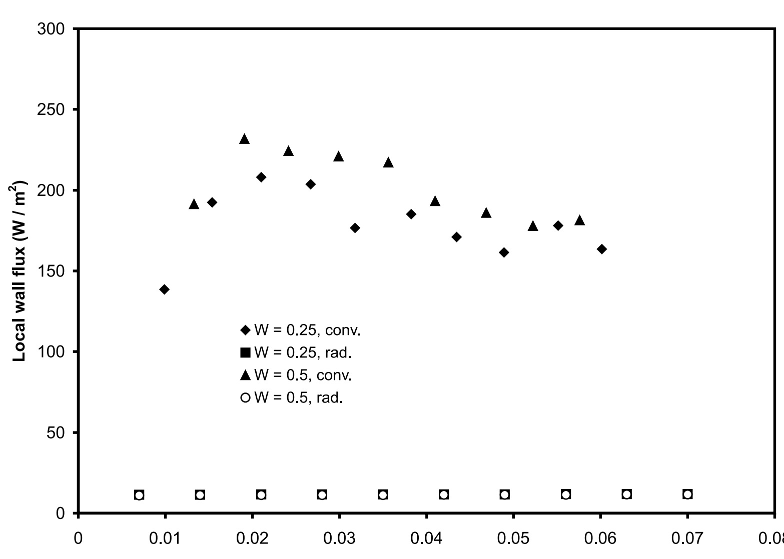

Fig. 5. Effect of side-vent on the local flux along the left wall ($\Delta T = 34.3°C$, $A = 0.5$, $\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.05$).

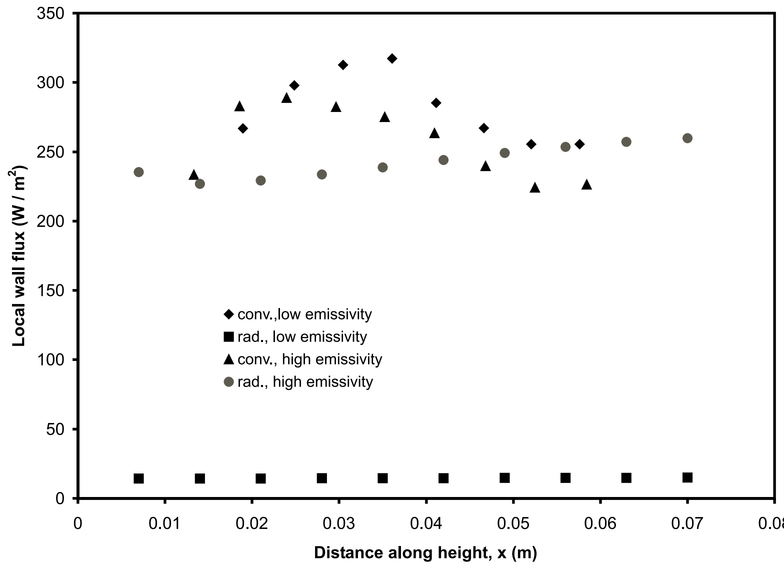

Fig. 6. Effect of surface emissivity on the local flux along the left wall ($\Delta T = 43.1°C$, $A = 0.5$, $W = 0.5$).

increased radiative heat transport from the left wall, because of intense surface radiation interchange (due to higher emissivity) among the walls of the cavity. It is known that natural convection heat transfer takes place because of buoyancy, wherein the fluid is caused to move because of density differences. The buoyant force is related to the fluid temperature differential and the coefficient of thermal expansion. In a cavity with highly emissive walls, the walls get heated due to surface radiation. Thus the air that flows along the walls of the cavity gets heated due to convection. In other words, convective heating of the air takes place due to radiative heating of the walls. Thus the buoyant force available for natural convection is reduced. Hence the drop in convective heat transfer coefficient for a cavity with highly emissive walls. For the cavity with low emissive walls, the contribution of radiation to the total heat transfer rate can be seen to be very small, as compared to that of the cavity with highly emissive walls.

From the discussion of results so far, it can be inferred that the presence of surface radiation has a significant effect on the thermal transport mechanism in a cavity. The extent to which the natural convection is affected by surface radiation is found to be a very strong function of surface emissivity of the walls that form the cavity. It would be of interest, therefore, to examine the relative contribution of natural convection and surface radiation heat transfer to the total heat transfer rate from the left wall of the cavity. Fig. 7 shows the variation of the percentage of heat transferred with Rayleigh number for two sets of cavities having the same aspect ratio and side-vents but different surface emissivities. The ordinate shows the percentage of heat transferred both by natural convection as well as surface radiation. The percentages have been computed with respect to the total heat transferred from the left wall of the cavity. As can be seen from the graph, for cavities with low emissive walls, natural convection is the dominant mode of heat transfer (approx. 95%) and only about 5% of

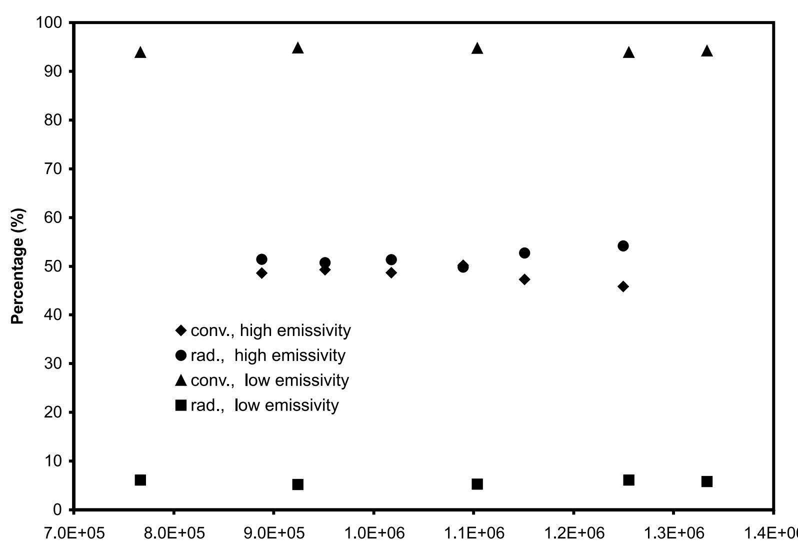

Fig. 7. Effect of surface emissivity on the mode of heat transfer from the left wall ($A = 0.5$, $W = 0.5$).

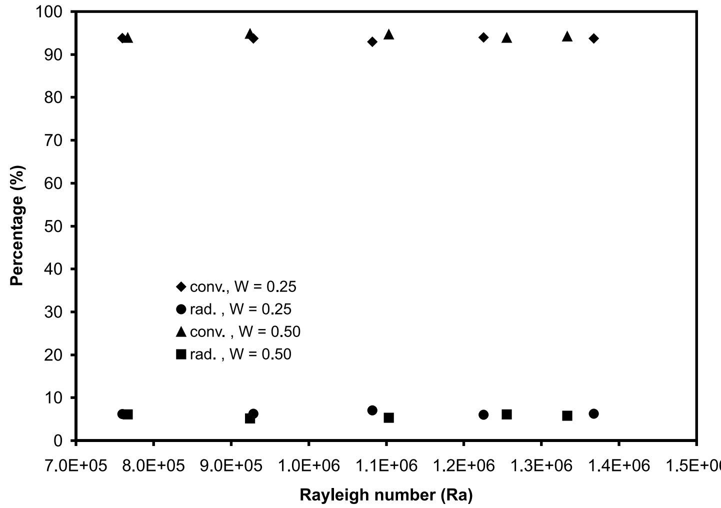

Fig. 8. Effect of side-vent on the mode of heat transfer from the left wall ($A = 0.5$, $\varepsilon_h = \varepsilon_b = \varepsilon_r = 0.05$).

heat is transferred by surface radiation. For cavities with highly emissive walls, both natural convection and surface radiation appear to be competing. Convection accounts for about 48% and radiation accounts for about 52% of the total heat transferred. Fig. 8 shows the effect of the height of the side-vent on the relative modes of heat transferred as a function of the Rayleigh number for a cavity with low emissive walls. It can be seen from the graph that the height of the side-vent does not have any significant effect on the relative modes of heat transfer from the left wall. This is found to be true for a similar cavity of $A = 0.25$ also. For the range of parameters considered in the present study, the effects of aspect ratio and the height of the side-vent were too small to be considered significant, in comparison to the effect of surface emissivity, in controlling the relative modes of heat transfer.

4. Conclusions

Results of interferometric studies on the interaction of natural convection and surface radiation heat transfer on side-vented open cavities investigating the effect of surface emissivity, height of the side-vent and aspect ratio of the cavity are presented. For cavities with highly emissive walls, it is found that, even for a moderate temperature differential typically encountered in practical applications, radiation heat transfer due to the high surface emissivity of the walls that form the cavity has a significant effect on the natural convection heat transfer that exists in such systems. The effects of the height of the side-vent and the aspect ratio of the cavity on the relative contributions of the heat transfer rate were found to be not significant compared to the effect of surface emissivity. For cavities with low emissive walls, natural convection was found to be the dominant mode of heat transfer, accounting for about 95% of the total heat transfer rate. For highly emissive cavities, both surface radiation and natural convection were found to have a competing behavior in transferring heat from the left wall of the cavity, with each mode contributing between about 45–55% of the total heat transfer rate. Therefore, in situations such as this, the effect of surface radiation has to be accounted for in order to obtain accurate estimates of the heat transfer rates.

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References

Annual Book of ASTM Standards, 1989. American Society for Testing and Materials 14.03. ASTM, Philadelphia, USA, pp. 96–97.

Balaji, C., Venkateshan, S.P., 1993a. Interaction of surface radiation with free convection in a square cavity. Int. J. Heat Fluid Flow 14 (3), 260–267.

Balaji, C., Venkateshan, S.P., 1993b. Discussion on natural convection with radiation in a cavity with open top end. ASME J. Heat Transfer 115, 1085–1086.

Balaji, C., Venkateshan, S.P., 1994. Interaction of radiation with free convection in an open cavity. Int. J. Heat Fluid Flow 15 (4), 317–324.

Black, W.Z., Carr, W.W., 1971. Application of a differential interferometer to the measurement of heat transfer coefficients. Rev. Sci. Instr. 42 (3), 337–340.

Carpenter, J.R., Briggs, D.J., Sernas, V., 1976. Combined radiation and developing laminar free convection between vertical flat plates with asymmetric heating. ASME J. Heat Transfer 98, 95–100.

Chan, Y.L., Tien, C.L., 1985. A numerical study of two-dimensional laminar natural convection in shallow open cavities. Int. J. Heat Mass Transfer 28 (3), 603–612.

Coleman, H.W., Steele Jr., W.G., 1989. Experimentation and Uncertainty Analysis for Engineers. Wiley, New York, USA (Chapter 3).

Dehghan, A.A., Behnia, M., 1996. Combined natural convection conduction and radiation heat transfer in a discretely heated open cavity. ASME J. Heat Transfer 118, 56–64.

Lage, J.L., Lin, J.S., Bejan, A., 1992. Natural convection with radiation in a cavity with open top end. ASME J. Heat Transfer 114, 479–486.

Larson, D.W., Viskanta, R., 1976. Transient combined laminar free convection and radiation in a rectangular enclosure. J. Fluid Mech. 78, 65–85.

Merzkirch, W., 1987. Flow Visualization, second ed. Academic Press, New York, pp. 180–188.

Moutsoglou, A., Rhee, J.H., Won, J.K., 1992. Natural convection radiation cooling of a vented channel. Int. J. Heat Mass Transfer 35 (11), 2855–2863.

Ostrach, S., 1953. An analysis of laminar free-convection flow and heat transfer about a flat plate parallel to the direction of the generating body force. NACA Report 1111.

Ramesh, N., Venkateshan, S.P., 1999a. Effect of surface radiation on natural convection in a square enclosure. AIAA J. Thermophys. Heat Transfer 13 (3), 299–301.

Ramesh, N., Venkateshan, S.P., 1999b. Effect of surface radiation and partition resistance on natural convection heat transfer in a partitioned enclosure: an experimental study. ASME J. Heat Transfer 121, 616–622.

Sernas, V., Fletcher, L.S., 1970. A schlieren interferometer method for heat transfer studies. ASME J. Heat Transfer 92, 202–204.

Sparrow, E.M., Cess, R.D. (Ed.), 1978. Radiation Heat Transfer. Hemisphere Publishing Corporation, New York, pp. 90–92.

Sparrow, E.M., Shah, S., Prakash, C., 1980. Natural convection in a vertical channel: I. Interacting convection and radiation. II. The vertical plate with and without shrouding. Numer. Heat Transfer 3, 297–314.

Showole, R.A., Tarasuk, J.D., 1993. Experimental and numerical studies of natural convection with flow separation in upward-facing inclined open cavities. ASME J. Heat Transfer 115, 592–605.

Yamada, Y., 1988. Combined radiation and free convection heat transfer in a vertical channel with arbitrary wall emissivities. Int. J. Heat Mass Transfer 31 (2), 429–440.

Yu, E., Joshi, Y.K., 1999. Heat transfer in discretely heated side-vented compact enclosures by combined conduction, natural convection and radiation. ASME J. Heat Transfer 121, 1002–1010.