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Effect of Surface Radiation on Natural Convection in a Square Enclosure

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Results of an experimental study of heat transfer by natural convection and surface radiation in an air-filled square enclosure using a differential interferometer are reported. The enclosure comprises two differentially heated vertical walls and two horizontal adiabatic walls. The suppression of natural convection in the presence of surface radiation has been demonstrated experimentally. For the case of an enclosure with highly emissive walls, mean Nusselt number correlations for convection and radiation, as well as combined convection and radiation, are also presented in terms of Grashof number in the laminar range.

Nomenclature

d = width of the enclosure, m
Gr = Grashof number, $g\beta(T_h - T_c)d^3/\nu^2$
h = local heat transfer coefficient along the hot wall, W/m^2 K
$\bar{h}$ = average heat transfer coefficient along the hot wall, W/m^2 K
k_m = thermal conductivity of air at T_m, W/m K
$\bar{N}u_c$ = Nusselt number (mean) due to convection, $[(q_c d)]/[(T_h - T_c)k_m]$
$\bar{N}u_r$ = Nusselt number (mean) due to radiation, $[(q_r d)]/[(T_h - T_c)k_m]$
$\bar{N}u_t$ = total Nusselt number, $\bar{N}u_c + \bar{N}u_r$
q_c = convective heat flux entering the enclosure from the hot wall, W/m^2
q_r = radiative heat flux entering the enclosure from the hot wall, W/m^2
q_t = total heat flux entering the enclosure, $(q_c + q_r)$, W/m^2
T_c = cold wall temperature, K
T_h = hot wall temperature, K
T_m = mean temperature, $(T_h + T_c)/2$, K
β = coefficient of volume expansivity, $(1/T_m)$, K^{-1}
ε = surface emissivity of the walls (all walls have the same emissivity)
ν = kinematic viscosity of air, m^2/s

Introduction

COUPLED natural convection and radiation transport processes arise in furnaces, natural water bodies, solar energy utilization, crystal growth, etc. Even though experimental and numerical studies on natural convection in enclosures are available, very few of these studies consider the effect surface radiation has on natural convection, particularly in cases where the fluid contained in the enclosure is radiatively nonparticipating. Fluids such as air, nitrogen, and inert gases are essentially nonabsorbing and nonemitting. In fact, in the majority of the problems, the contribution due to radiation heat transfer is either absent or ignored. Interestingly, in some conjugate transport problems, radiative heat transfer often becomes substantial, even at relatively low temperatures, because the natural convection heat transfer rates are often small, particularly in gases. In some cases, radiative transport is often comparable with or even larger than the free convective heat transfer.

There are more studies on enclosures filled with participating media than nonparticipating media. Nevertheless, considering applications involving solar collectors, double-pane windows, electronic equipment cooling, etc., the contribution of surface radiation is significant, even in the presence of a nonparticipating medium such as air. An excellent review[1] on radiation convection interaction in enclosures is available. Larson and Viskanta[2] numerically investigated the transient combined laminar free convection and surface radiation in a rectangular enclosure, including wall heat conduction. They concluded that, at very high temperatures, radiation is the predominant mode of heat transfer and that the convective currents are significantly altered by radiation. However, the results were not summarized in the form of useful correlations. Bratis and Novotny[3] analytically and experimentally examined the interaction of thermal radiation and free convection in the boundary-layer regime of a vertical rectangular enclosure. Pure NH_3, N_2, and a mixture of NH_3 and N_2 for pressures up to 2 bar at a temperature level of 300 K were used as test fluids. A finite difference method of analysis was carried out by Zhong et al.[4] in a square enclosure, assuming black walls and considering two-dimensional variable properties and laminar natural convection both with and without radiation. A new correlation, based on the results of the numerical study, was also presented. It was also concluded that, while radiation is relatively insensitive to tilt angles, its effect on natural convection is considerable. The effect of surface emissivity of the walls of an enclosure was taken into consideration for the first time by Balaji and Venkateshan.[5,6] The two-dimensional analysis was carried out numerically. An important finding from their study was that surface radiation tends to suppress free convection. Correlations for Nusselt number due to free convection as well as surface radiation are presented. A notable feature of the correlations is that the surface emissivities of all the surfaces are explicitly included.

The purpose of the present study is to demonstrate experimentally the suppression of natural convection because of the presence of surface radiation in a square enclosure filled with air, having differentially heated vertical walls and adiabatic horizontal walls, and to obtain correlations for the convective and radiative Nusselt numbers. The present authors have been able to realize nearly adiabatic conditions along the top and bottom walls of the enclosure. A review of literature shows that there has been no experimental study considering the effect of surface radiation on laminar natural convection in a square enclosure containing air as the medium.

Experimental Apparatus and Procedure

The experimental apparatus used to investigate the heat transfer phenomena in the present study is an optical device, called the differential interferometer (DI). 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

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light. The description of the interferometer and procedure to calculate the local heat transfer coefficients along the hot wall of the enclosure from the measured values of the fringe deflections are found elsewhere.[7] However, for clarity, a brief description of the test cells is outlined next. Two test cells (40 × 40 × 200 mm and 60 × 60 × 300 mm) were used in the present study to cover a wide range of Grashof numbers, from $5 \times 10^4 \leq Gr \leq 1 \times 10^6$. Each test cell consisted of four vertical walls and two horizontal walls. One vertical wall formed the hot wall, and the other, the cold wall or sink. The hot wall made of aluminum consisted of a Nichrome™ wire-wound heater, energized externally by a stabilized ac power supply through a variac. The variac could be adjusted to any voltage to have a desired value of hot wall temperature. The cold wall also made of aluminum, with milled grooves that formed channels, was connected to a thermostat (Haake F Junior, Germany, ±0.05°C) to circulate water to maintain the cold wall at the desired temperature. Temperatures of hot and cold walls were measured at several locations along the depth. The thermocouples fixed along the hot and cold isothermal vertical walls showed a maximum temperature variation of ±0.2°C over the surface of the hot wall, and a maximum variation of ±0.3°C over the surface of the cold wall, at a temperature difference of 67.5°C between both walls.

In the present study, the hot wall temperature varied between 55 and 118°C, and the cold wall temperature was kept at 35°C. The top and bottom walls were made of Perspex™, ~1 in. thick, and covered on the outside by glass wool insulation about 25 cm thick, to obtain nearly adiabatic conditions. The temperatures along the inside and outside surfaces of both the top and bottom horizontal walls were also measured. Using the measured temperatures, the heat loss from the inside of the enclosure through the 1-in.-thick Perspex wall to the outside was estimated. This heat loss was found[8] to be less than ~3% of the total heat input into the enclosure from the hot wall for the case of maximum temperature difference between the hot and cold walls. The front and rear vertical walls were made of optical quality glass plates to allow monochromatic light (green, 549 nm) from the interference filter of the DI to pass through. These two walls are assumed to not transmit any low temperature [infrared (IR)] radiation out of the enclosure or into the enclosure from the outside. To have a surface emissivity of 0.85, all inside surfaces of the enclosure (except glass windows) were painted with three coats of blackboard paint. Wooden walls used by Eckert and Carlson[9] typically have a surface emissivity of 0.85, similar to the emissivity used in the present study. The maximum thickness of the surface coating was about 50 μm, and it was found that the temperature drop across the paint layer could be ignored. For achieving low emissivity for the inside surfaces of the walls, the hot and cold walls were polished to a mirror finish ($\varepsilon = 0.05$), and the top and bottom horizontal walls were covered on the inside with highly reflecting aluminum foil. The foil was laid on the horizontal walls and firmly bonded only on the sides, using an adhesive. Temperature measurements were made using chromel–alumel thermocouples (0.3-mm wire diameter) on the hot wall, cold wall, and at locations (at ~5-mm intervals) along the inside and outside of the top and bottom walls. A typical experimental run lasted more than 10 h, so that after a sufficient amount of time had elapsed, steady-state conditions prevailed. Interferograms were recorded on Kodak Academy Black and White film (200 ASA) with suitable exposure.

Results and Discussion

The interferometer was calibrated using a vertical flat plate, and local Nusselt numbers calculated from measured values of heat transfer coefficients obtained from experiments were compared with theoretical results due to Ostrach, as reported by Chapman.[10] The comparison can be termed excellent, the rms deviation of the local experimental Nusselt number values from the theoretical values along the height being about 0.27. Expressed as a percentage, the ratio of the standard deviation to the mean Nusselt number was found to be 3.94%. The thermocouples were calibrated and the correction for error was also taken into account, as per the procedure outlined in the ASTM standard.[11] The following uncertainties apply for the present work: temperature, less than ±0.04°C; length, less than ±0.02 mm; and emissivity, less than ±0.01. These uncertainties would result in an overall uncertainty[12] of less than ±5% in obtaining the total Nusselt number ($\bar{N}u_t$).

Experiments were carried out on enclosures with different hot wall temperatures, obtained by varying the electrical input to the heater provided in the hot wall. Interferograms were obtained for various cases and analyzed to evaluate the convective heat transfer coefficient and Nusselt number along the hot wall. To calculate the radiative heat transfer component, the measured temperatures of all walls were used in an enclosure analysis.[10] The required shape factors were evaluated using Hottel's crossed string method.

Certain observations, gathered from a series of controlled experiments, are worth mentioning. For a given value of electrical input to the heater, it was seen that the hot wall temperature of an enclosure with all walls having $\varepsilon = 0.85$ was lower than the corresponding case of $\varepsilon = 0.05$. The temperatures of the top and bottom adiabatic walls were higher for an enclosure with walls having $\varepsilon = 0.85$ than for the case with walls having $\varepsilon = 0.05$. This is because of the fact that the top and bottom walls interact radiatively with the hot wall. Photographs of the fringe pattern (interferogram) along the hot wall of the enclosure for $\varepsilon = 0.05$ and 0.85 are shown in Figs. 1a and 1b, respectively. Note that both photographs have the same magnification. A careful examination of the fringe pattern reveals the subtle differences in the fringe deflections along the hot wall for the two cases. As indicated on the photographs, at a common location, the fringe deflection is larger for $\varepsilon = 0.05$ than for $\varepsilon = 0.85$, for the same value of T_h. This indicates suppression of natural convection because of the presence of surface radiation.

The reason why smaller fringe deflection indicates suppression of natural convection can be explained as follows. Two coherent light beams, on interference after passage through a wedge, form parallel fringes. When an additional phase shift is introduced between the two beams, due, e.g., to a temperature field in a gas, deformation of the fringe pattern occurs. In a DI, this deformation is used as a measure of the heat flux. The DI uses a birefringent prism (Wollaston prism) to split the polarized light. The two rays pass through the test section at locations adjacent to each other. This introduces phase lag in both rays, which are proportional to the density values of the medium at these two locations, inside the test cell. The deformed fringes carry information about the difference in densities of the medium at these two locations. In an enclosure with high emissive walls ($\varepsilon = 0.85$), the presence of surface radiation tends to equilibrate the air temperature inside the enclosure.[5] In the case of highly polished walls ($\varepsilon = 0.05$), this does not happen. In the case of natural convection, the fluid is caused to move by density differences. The buoyant force is related to the fluid temperature and the coefficient of thermal expansion. Therefore, in an enclosure with high emissive walls, the presence of surface radiation reduces the buoyant force that is available for natural convection. Hence, the temperature gradient along the hot wall due to convection is affected. Therefore, the magnitude of deflection of the fringes along the hot wall is smaller for highly emissive walls compared with that for highly polished walls. Because the heat transfer coefficient is directly proportional to fringe deflection,[7] a reduction in fringe deflection indicates a reduction in heat transfer coefficient.

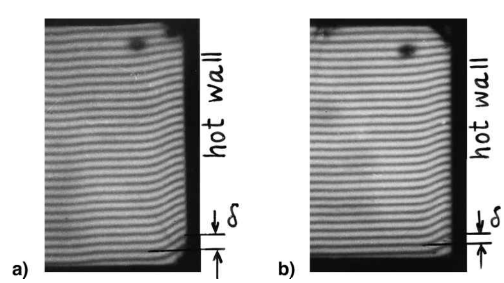

Fig. 1 Interferogram showing the deflection of fringes along the hot wall of the enclosure having walls with a) $\varepsilon = 0.05$ and b) $\varepsilon = 0.85$, respectively. $T_h = 66°C$ for both cases. Both photographs are taken to the same scale. δ indicates fringe deflection at a particular location.

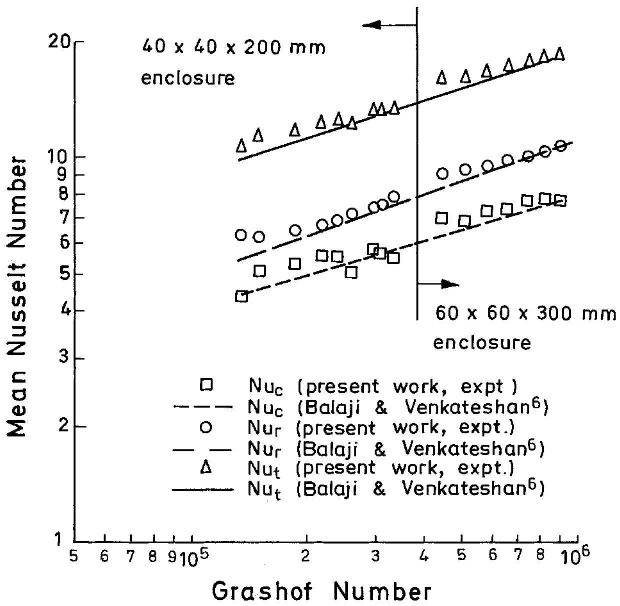

Fig. 2 Variation of mean Nusselt number with Grashof number for an enclosure with all walls coated with blackboard paint ($\varepsilon = 0.85$).

From the photographs (Fig. 1) it can also be inferred, on close examination, that the fringe deflections progressively decrease from the bottom to the top of the hot wall, indicating a gradual decrease in the wall heat flux as well as the heat transfer coefficient. This behavior was observed invariably for all of the cases considered in the present study. The local, and hence, mean value of the heat transfer coefficient for the case of an enclosure with highly polished walls ($\varepsilon = 0.05$) is higher than the case where all walls are painted with blackboard paint ($\varepsilon = 0.85$). For the case shown in the photographs, $\bar{N}u_c = 7.819$ for $\varepsilon = 0.05$ and 7.268 for $\varepsilon = 0.85$; $\bar{N}u_r = 0.464$ for $\varepsilon = 0.05$ and 9.486 for $\varepsilon = 0.85$; and $\bar{N}u_t = 8.283$ for $\varepsilon = 0.05$ and 16.754 for $\varepsilon = 0.85$. It was found that, even though surface radiation suppresses natural convection, the Nusselt number due to radiation, and therefore, the total Nusselt number, increases drastically for the enclosure whose surfaces are highly emissive. Hence, even at moderate wall temperatures ($T_h = 66°C$ in this case), neglect of contribution due to surface radiation can lead to a significant error in calculations. The results of most studies on enclosures using air that are reported in literature have used Grashof number as a parameter. In the present study, the medium inside the enclosure was air, and the Prandtl number of air varies very little with temperature. Because of these reasons, the values of mean Nusselt number obtained from the present experiments are plotted against the Grashof number in Fig. 2. The graph represents the case where all surfaces of the enclosure are painted with blackboard paint ($\varepsilon = 0.85$). The enhancement of the overall heat transfer, as evident by the increase in the value of total Nusselt number, can be clearly seen from the graph. Based on the results of the present experiments on an enclosure with highly emissive walls, the following correlations are obtained:

1) Nusselt number due to natural convection: $\bar{N}u_c = 0.149Gr^{0.29}$; correlation coefficient = 0.96; error band = ±9.6%; standard error of the estimate of $\bar{N}u_c = 0.022$.

2) Nusselt number due to surface radiation: $\bar{N}u_r = 0.141Gr^{0.317}$; correlation coefficient = 0.99; error band = ±4.9%; standard error of the estimate of $\bar{N}u_r = 0.012$.

3) Total Nusselt number: $\bar{N}u_t = 0.288Gr^{0.305}$; correlation coefficient = 0.99; error band = ±6.2%; standard error of the estimate of $\bar{N}u_t = 0.013$.

In all of these correlations, the thermophysical properties of air are evaluated at T_m, the mean of hot and cold wall temperatures. The hot wall emissivity has not been included in the correlations due to the following reasons:

1) The purpose of the study was to demonstrate experimentally the suppression of natural convection that takes place due to the presence of surface radiation.

2) The magnitude of surface radiation is higher for a highly emissive wall ($\varepsilon = 0.85$).

3) Correlations obtained from a comprehensive numerical study, incorporating emissivities of hot and cold walls, and top and bottom walls are available elsewhere.[6]

The total Nusselt number has a constant of 0.319 and index of 0.305 when correlated with Rayleigh number. Zhong et al.,[4] based on a numerical study, have correlated total Nusselt number in terms of Rayleigh number, for a particular value of $(T_h - T_c)/T_c = 2$. However, their correlation had a constant of 0.901 and index of 0.281. In the present study $(T_h - T_c)/T_c$ varies from 0.07 to 0.26, which does not come anywhere near the value of 2, used by Zhong et al.,[4] and hence, a direct comparison is not possible. However, the present results are in excellent agreement with those of Balaji and Venkateshan,[6] as shown by the comparison in Fig. 2.

Conclusion

The results of an interferometric study of an air-filled square enclosure having differentially heated vertical walls and adiabatic horizontal walls are reported. The hot and cold vertical walls and the top and bottom horizontal walls were highly emissive. A comparison of results with an enclosure having all walls highly polished shows that the presence of surface radiation suppresses natural convection heat transfer coefficient in an enclosure having highly emissive hot and cold vertical walls and adiabatic horizontal walls. However, the reduction of heat transfer coefficient is compensated by the overall increase in the total Nusselt number, due to the presence of surface radiation, in the case of an enclosure with highly emitting walls. Hence, heat transfer from the hot wall is augmented. Correlations for convective Nusselt number, radiative Nusselt number, and total Nusselt number are also presented.

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