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Combined convection from isothermal cubical cavities with a variety of side-facing apertures

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Abstract—An experimental investigation of the heat transfer by combined convection from a smooth, isothermal cubic cavity with a variety of side-facing apertures is described. The study was motivated by the desire to predict the convective loss from large solar thermal–electric receivers and to understand the mechanisms which control this loss. Hence, emphasis is placed on the large Rayleigh number, *Ra*, and large Reynolds number, *Re*, regimes; on Richardson numbers, *Ri*, of the order of unity; and on large ratios of the cavity wall temperature, T_w, to the ambient temperature, T_∞. The data which are presented cover the ranges: $1 < T_w/T_\infty < 2$, $L^2/18 \leqslant A_a \leqslant L^2$, $5\times10^7 < Ra < 2\times10^{10}$, and $0.01 < Ri < \infty$.

INTRODUCTION

The problem which precipitated the title investigation is the determination of convective energy loss from large solar receivers. The determination of the convective energy loss is difficult because of: (i) the complex geometries and the complexity of the convective flow from and within open cavities, (ii) the large Rayleigh numbers, *Ra*, and large Reynolds numbers, *Re*, of interest, that is, the difficulty of experimentally modeling large buoyant and inertia forces, and (iii) the large ratios of the wall temperature to the ambient temperature, T_w/T_∞, that is, the difficulty of experimentally simulating these hot cavities without the convective energy flow being masked by radiative heat transfer. The assumption of constant properties is clearly not applicable, and the role of the additional dimensionless group, T_w/T_∞, must be established.

Some convective loss data are available from tests conducted on full scale solar receivers (see, e.g. ref. [1]). Unfortunately, these isolated data points contain large uncertainties because of the large radiative energy fluxes which are present in such tests. In addition, these data points provide little information on the important mechanisms or on the functional dependency of the convective loss on the controlling parameters. Hence, these data are of limited value in trying to predict convective losses from other cavity configurations.

On the other hand, in controlled laboratory tests with small-scale models, it is difficult to satisfy the relevant modeling laws because of the large size and the high temperature of solar receivers. For example, the excellent experimental study of convective losses from a five sided, cubic cavity by LeQuere *et al.* [2] is a pure natural study in a Rayleigh number regime which is completely outside the fully turbulent, high *Ra* regime of interest in solar receivers. Also, the ratio of the wall temperature, T_w, to the ambient temperature, T_∞, is nearly unity, whereas ratios typically between 2 and 4 are of interest in solar thermal–electric receivers. No studies to date have considered the importance of the size and location of the aperture or the influence of the temperature ratio over any significant range.

The objectives of this investigation are to understand the mechanisms which control the convective energy flow out of or into cavities for a variety of aperture configurations and to determine the Richardson number, *Ri*, at which combined inertia influences become of importance. An understanding of the mechanisms will enable the estimation of this energy transport for other geometries and provide ways of reducing this loss. The results will also be applicable to other problems such as the convective energy transport from open buildings. In this application, the large Rayleigh number and large Reynolds number regimes are again of interest, and separating the energy transported by convection from that transported by radiation is still a problem.

EXPERIMENTAL APPARATUS

Large solar receivers are modeled in the correct regimes in the title investigation by using the Cryogenic Wind Tunnel at the University of Illinois at Urbana–Champaign (UIUC). This facility is a variable ambient temperature tunnel which can operate with test section temperatures between 80 and 310 K. The rectangular test section has a height of 1.2 m and a width of 0.6 m. The maximum test section velocity is 6 m s^{-1}. Gaseous nitrogen is used as the working fluid in cryogenic tests, and air is used in experiments which are conducted with $T_\infty > 290$ K. The flow in the test section has a turbulence intensity of 1.7%; the velocity field has uniformity of 1.5%. The main advantages of this liquid nitrogen cooled facility and

NOMENCLATURE

- A area [m^2]
- AN aperture number
- b defined by equation (4)
- c_p specific heat [J kg^{-1} K^{-1}]
- f defined by equation (4)
- g acceleration of gravity, defined by equation (4) [m^2 s^{-1}]
- Gr Grashof number, $g\beta\Delta TL^3/\nu^2$
- h heat transfer coefficient [W m^{-2} K^{-1}]
- k thermal conductivity [W m^{-1} K^{-1}]
- L length [m]
- Nu Nusselt number, hL/k
- Pr Prandtl number, ν/α
- q heat flow rate [W]
- Ra Rayleigh number, $g\beta\Delta TL^3/(\nu\alpha)$
- Re Reynolds number, VL/ν
- Ri Richardson number, Gr/Re^2
- T temperature [K]
- V free-stream velocity [m s^{-1}]
- x coordinate of aperture centroid
- y coordinate of aperture centroid.

Greek symbols

- α thermal diffusivity [m^2 s^{-1}]
- β volume coefficient of expansion [K^{-1}]
- ν kinematic viscosity [m^2 s^{-1}].

Subscripts

- a aperture
- b bulk value
- B buoyancy dominated, natural convection limit
- cz convective zone
- f based on film temperature
- w wall condition or aperture width
- ∞ ambient condition.

Superscript

- * dimensionless quantity.

a more detailed description of it are provided in refs. [3, 4]. Previous natural convection data which were obtained from the UIUC facility agree well with data obtained at normal temperature levels from full scale models as well as data taken at high pressures: between 1 and 70 atm (see ref. [3]).

A cubic cavity, $L \times L \times L$ where $L = 0.4$ m, with a variety of side-facing apertures was chosen for the investigation. The cavity is mounted outside of the tunnel with its aperture plane lying in the wall of the test section. The aperture configurations with their respective aperture numbers, AN, are illustrated in Fig. 1. Seven configurations were used in the natural convection investigated [4]: $AN = 1$ (a completely open side), 2, 61, 62, 63, 12, and 18; however, only three of these were considered in the more expensive combined convection study. The five full sides of the cube are smooth, isothermal, active surfaces. The remaining portion of the sixth side, the aperture mask, is an adiabatic surface. Detailed descriptions of the

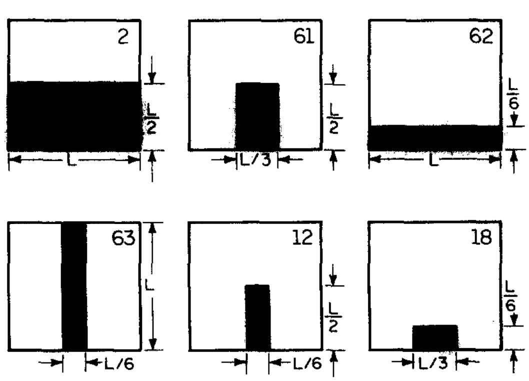

FIG. 1. Aperture configurations investigated.

model, the transient technique used to experimentally deduce the convective loss, the instrumentation, and the data acquisition equipment are provided in ref. [4].

An uncertainty analysis was performed in order to determine the accuracy of the experimental results. Following an examination of the uncertainties in the various measurements and those in the thermophysical property data of air, nitrogen and aluminum, this analysis showed that the uncertainties in Ra, Ri, and Re are 3%, and the uncertainty in the dimensionless heat loss from the cavity is 4%.

SIMILITUDE CONSIDERATIONS

The dimensionless groups which govern the heat transfer by combined convection from an isothermal cubic cavity are deduced from a dimensional analysis of the general, compressible form of the governing equations. The simplifying assumptions are: a laminar flow of a Newtonian fluid, a perfect gas, a radiatively non-participating gas, negligible viscous dissipation, and negligible work done by compression. The dependent variables, c_p/c_{pr}, μ/μ_r, and k/k_r, are general functions of only the dimensionless temperature, T/T_r, where subscript r stands for the reference temperature. With these assumptions, one obtains

$$q^* = f_1(Ra, Ri, Pr, T_w/T_\infty, L_a^*, L_w^*, x^*, y^*). \quad (1)$$

The last four groups are: the height of the aperture, the width of the aperture, and the x- and y-coordinates of the centroid of the aperture, respectively, all referenced to the characteristic length, L_c. Thus, even a simple, isothermal cubic cavity with a single, rectangular aperture gives rise to four geometrical parameters. Including the group, T_w/T_∞, enables one to arrive at equation (1) without making the Boussinesq approximation. The influences of variable properties are taken into account with the absolute temperature ratio, T_w/T_∞; hence, equation (1) is not restricted to values of this parameter near unity. The properties in all dimensionless groups are evaluated at the film temperature, $T_f \equiv (T_w + T_\infty)/2$, and (the height of the aperture $+ L/2$) is used as the characteristic length, L_c, in these groups unless indicated otherwise with the appropriate subscript. The logic for the use of this characteristic length is given in ref. [4].

Obviously, the set of dimensionless groups contained in equation (1) is not a unique set. The combination of the Rayleigh number and the Richardson number is used because the combined convection regime near the natural convection limit is of primary interest. That is, both solar central receivers and open buildings represent applications where inertia influences are hypothesized to be of secondary importance. One objective of the research is to determine if this hypothesis is correct. The Richardson number, $Ri \equiv Gr/Re^2$, is representative of the relative importance of buoyant and inertia forces. The limit as Ri approaches infinity is the natural convection limit.

Instead of trying to delineate the influences of four geometrical parameters, Clausing [1] hypothesized that (i) the region above the horizontal plane passing through the upper lip of the aperture is relatively stagnant, (ii) this plane is a nearly adiabatic plane, and (iii) the ability to heat the air inside the portion of the cavity below this plane, the convective zone, often controls the convective loss from cavities operating in the large Ra and large Ri regimes. This suggests the following. (i) The dependence on geometry can be reduced, and the cubic cavity results can be generalized, by basing the definition of h on the area in the convective zone, that is, $h \equiv q/\{A_{cz}(T_w - T_\infty)\}$. (ii) The bulk temperature in the convective zone, T_b, is close to T_∞; hence, factoring out the quantity $b \equiv (T_w - T_b)/(T_w - T_\infty)$, would reduce the dependency of the balance of the correlation on the geometrical parameters and in this manner would simplify the determination of the correlation. Note that $0 < b < 1$ and $b \to 1$ as Ra or Re approaches infinity. Hence, the Nusselt number, a dimensionless heat loss from the cavity, is defined as

$$Nu = \frac{hL_c}{k} = \frac{qL_c}{A_{cz}(T_w - T_\infty)k} \tag{2}$$

and equation (1) becomes

$$Nu = f_2(Ra, Ri, Pr, T_w/T_\infty, L_a^*, L_w^*, x^*, y^*)b. \tag{3}$$

The advantage of equation (3) is that at large Richardson numbers, f_2 is only weakly dependent on the dimensionless geometrical parameters whereas the heat loss is strongly dependent on these parameters. The usefulness of equation (3) will decrease as Ri approaches zero. At large Richardson numbers, the area above the horizontal plane passing through the upper lip of the aperture has little influence on the convective loss; hence, the data can only be correlated by using the area in the convective zone in the definition of h. On the other hand, at sufficiently small Richardson numbers, the gas in the upper region of the cavity will no longer be stagnant; therefore, it will not be possible to correlate the data with h based on A_{cz}. Determining the limit of usefulness of equation (2) is an objective of this investigation.

The function b has an interesting role. The total driving potential, $(T_w - T_\infty)$, is used in the definition of h. This is a logical choice since it is the only characteristic temperature difference in the specifications of the problem. On the other hand, the correlation of the rate of heat transfer within the cavity is possible in the classical sense only if h is based on the bulk temperature within the cavity. The function b effects this transformation. The quantity $(T_w - T_\infty)$ is used in the definition of h, but h is implicitly based on $(T_w - T_b)$ in equation (3)—$Nu = f_2 b$.

EXPERIMENTAL RESULTS

Since it has been hypothesized that buoyant forces are dominant in many practical applications involving large open cavities, an important experimental result is the determination of the Richardson number at which inertial effects become important. For this reason, the correlation applicable to the pure natural convection limit, $Ri = \infty$, is required. The correlation for the pure natural convection case, which was derived in ref. [4], is

$$Nu_B = gfb. \tag{4}$$

(i) g, a function of Ra, is the constant property correlation for the natural convection from the interior surfaces to the fluid in the convective zone which is at the bulk temperature, T_b. A gas with $Pr \cong 0.7$ is assumed. (ii) f, a function of Ra and T_w/T_∞, is the quantity which accounts for variable property influences. The limit of f, as T_w/T_∞ approaches one, is one. (iii) b, a function of Ra, T_w/T_∞ and the geometry, accounts for values of T_b which are significantly different from T_∞. Subscript B is used to denote the natural convection limit—dominant buoyant forces. The functions f, g, and b were derived for the laminar, transitional, and turbulent regimes and are reported in ref. [4]. Function b, a unique aspect of equation (4), was derived analytically. b is dependent on the rate of energy transported across the aperture; hence, an implicit function resulted which is solved iteratively to determine b. The maximum deviation of any natural convection data point from equation (4) was 20%, and 90% of the 102 data points lie within $\pm 12\%$ of this correlation. In comparison, the largest dimensional rate of heat exchange was 61 times larger than the smallest.

As the Richardson number is decreased, the data

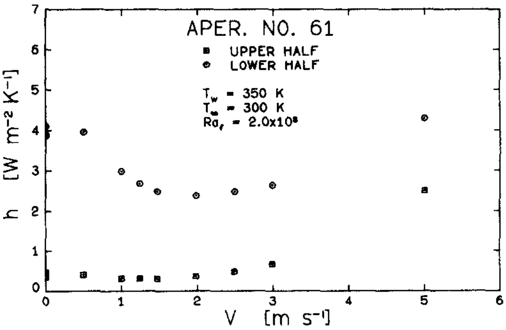

FIG. 2. The influence of free-stream velocity on h.

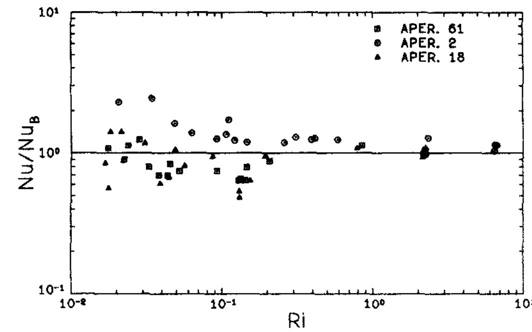

FIG. 3. Influence of Ri on Nu/Nu_B for various apertures.

will begin to deviate from equation (4) for one or more of the following reasons. (i) The forced flow will influence the temperature in the convective zone; hence, the model used to calculate b will no longer be applicable. (ii) The inertial influence on the bulk flow in the convective zone will give rise to a Reynolds number influence on the function g and perhaps also the function f. (iii) The inertial influences will be strong enough to influence the 'stagnation zone'. At this point, equations (2) and (3) will cease to be useful.

Consider first some of the dimensional combined convection data. The influence of free-stream velocity on the average heat transfer coefficient on the upper and lower halves of the cavity is shown in Fig. 2. These data are for the following conditions: aperture configuration 61, $T_w = 350$ K, $T_\infty = 300$ K, and $Ra = 2 \times 10^8$. As the velocity is increased to a critical value of approximately 2 m s^{-1}, the heat transfer from the lower half of the cavity *decreases* sharply. On the other hand, the heat transfer from the upper half of the cavity remains essentially at the pure natural convection limit. The significant decrease in the heat transfer from the lower surfaces accounts for a 35% decrease in the overall heat loss from that in the pure natural convection limit. This suggests that the wind in combination with an appropriately designed nozzle could provide an effective 'air curtain' and a means of reducing the energy transport across side-facing apertures.

More detailed data showing the influence of the free-stream velocity on the heat transfer coefficients for selected cavity surfaces are given in Table 1. Surfaces 2 and 6 are the bottom half of the two side walls closest to the aperture; surfaces 7 and 8 are the bottom and top horizontal surfaces, respectively. Results are given for aperture configuration 61 and three sets of wall and ambient temperatures. The three sets of pure natural convection data, the $V = 0$ cases in Table 1, show the expected symmetry—virtually identical values of h on surfaces 2 and 6. The heat transfer from surface 8 dominates at or near the natural convection limit. For example, in the first set of data given in Table 1, surface 8 accounts for 48% of the total convective loss, and surface 7 accounts for only 2%.

The complicated nature of the interaction between the inertial and buoyant forces is clearly evident in the independent h variations on the various surfaces of the cavity as the velocity is increased. The direction

Table 1. Influence of free-stream velocity on heat transfer from individual surfaces, aperture configuration 61

Conditions	Velocity (m s^{-1})	h_2 (Heat transfer coefficient, W m^{-2}K^{-1})	h_6	h_7	h_8	Ri_w
$T_w = 350$ K, $T_\infty = 300$ K, $Ra = 2.0 \times 10^8$	0	3.3	3.3	0.2	4.8	∞
	0.5	2.5	5.2	0.2	4.1	0.402
	1.0	1.9	4.5	0.2	3.0	0.101
	1.5	1.7	3.7	0.2	2.5	0.045
	2.0	1.7	3.4	0.2	2.3	0.025
	2.5	1.8	3.7	0.2	2.2	0.016
	3.0	2.0	4.0	0.3	2.3	0.011
	5.0	3.2	6.7	2.0	3.7	0.004
$T_w = 300$ K, $T_\infty = 200$ K, $Ra = 1.4 \times 10^9$	0	5.9	5.8	0.1	8.2	∞
	0.5	4.6	7.2	0.0	7.1	1.045
	2.0	2.8	6.0	0.0	4.3	0.065
	3.5	3.0	5.7	0.1	4.0	0.021
	5.4	4.1	7.6	0.8	5.1	0.009
$T_w = 227$ K, $T_\infty = 150$ K, $Ra = 4.2 \times 10^9$	0	6.9	6.8	0.0	9.5	∞
	0.5	5.8	8.3	0.0	8.5	1.068
	2.0	2.4	6.9	0.0	5.3	0.067
	3.4	4.0	7.6	0.0	5.6	0.023
	4.2	5.0	10.2	1.3	6.4	0.015

of free-stream flow across the aperture plane is from surface 2 to surface 6. This is the reason why the heat transfer coefficient on surface 6 becomes dominant at sufficiently high Reynolds numbers—small values of Ri (see Table 1). The variations of the heat transfer coefficients with velocity on the other lower vertical surfaces are similar, except there is not an initial increase to a local maximum. The results in Table 1 show that the heat transfer coefficient on surface 7, the upper horizontal surface, remains constant at a negligible value until relatively high velocities are reached. Specifically, the top of the cavity is oblivious to the external forced flow for Richardson numbers greater than 0.025.

The Nusselt number Nu is normalized with respect to the correlation for the limiting case of pure natural convection Nu_B, which is calculated using equation (4), and plotted vs the Richardson number in Fig. 3. The results show that for $Ri_w > 0.2$, the inertial influence is negligible. The natural convection correlation, equation (4), predicts the overall heat transfer well with an average absolute deviation between the data and the correlation of 12% if $Ri_w > 0.2$. Using the width of the aperture as the characteristic length in the Richardson number results in an even departure of the experimental data from Nu_B at $Ri \cong 0.2$ regardless of the aperture configuration. Although the inertial influences start to become significant for $Ri_w < 0.2$, the examination of the energy flow across the horizontal plane which passes through the upper lip of the aperture showed that the 'stagnation zone' above this plane persists if $Ri_w > 0.04$. Hence, the usefulness of equations (2) and (3) will cease only at Richardson numbers which are significantly less than 0.04.

CONCLUSIONS

The following conclusions are drawn from this study.

(1) In the regime $0.04 < Ri_w \leqslant \infty$, the horizontal plane passing through the upper lip of the aperture is a nearly adiabatic plane. Hence, the geometry of the portion of the cavity which lies above this plane has little influence on the convective loss, and the data can be correlated only if the definition of the heat transfer coefficient is based on the surface area which lies below this plane. This conclusion remains valid even for values of T_w/T_∞ near unity.

(2) The natural convection correlation developed in ref. [4], equation (4), provides an accurate prediction of the combined convection heat transfer from isothermal cavities if $Ri_w > 0.2$.

(3) Inertial influences can be neglected in the Richardson number regime typically of interest for solar thermal–electric receivers, the regime $1 < Ri_w < 1000$.

(4) In situations where the inertia and buoyant effects are of the same order, the convective loss is less than that for the pure natural convection case. This suggests that a two-dimensional, plane nozzle which is mounted on the exterior of a cavity receiver could provide a passive means of reducing the convective losses by creating a wind-driven air curtain.

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CONVECTION COMBINEE POUR DES CAVITES CUBIQUES ISOTHERMES AVEC UNE VARIETE D'OUVERTURES SE FAISANT FACE SUR LES COTES

Résumé—On décrit une étude expérimentale du transfert thermique par convection combinée par une cavité cubique, lisse et isotherme munie d'ouvertures latérales qui se font face. Il s'agit de prédire les pertes convectives à partir des récepteurs solaires thermo-électriques et de comprendre les mécanismes qui contrôlent cette perte. L'intérêt est porté sur les régimes de grands nombres de Rayleigh et de Reynolds, sur des nombres de Richardson Ri de l'ordre de l'unité et sur des grands rapports de la température pariétale de la cavité T_w à la température ambiante T_∞. Les données presentées couvrent le domaine: $1 < T_w/T_\infty < 2$, $L^2/18 \leqslant A_a \leqslant L^2$, $5\times10^7 < Ra < 2\times10^{10}$ et $0{,}01 < Ri < \infty$.

MISCHKONVEKTION AN WÜRFELFÖRMIGEN ISOTHERMEN HOHLRÄUMEN MIT VERSCHIEDENEN SEITLICHEN ÖFFNUNGEN

Zusammenfassung—In dieser Arbeit wird die experimentelle Untersuchung des Wärmeübergangs durch natürliche und freie Konvektion von einem würfelförmigen, glatten Hohlraum von konstanter Temperatur mit verschiedenen seitlichen Öffnungen beschrieben. Die Untersuchung entstand aus dem Wunsch, die Wärmeverluste durch Konvektion bei großen Receivern von Solarturmkraftwerken berechnen zu können und die Mechanismen zu verstehen, die diese Verluste beeinflussen. Daher wurden die Untersuchungen vor allem bei hohen Rayleigh- und Reynolds-Zahlen durchgeführt, bei einer Richardson-Zahl von ungefähr 1 und bei großen Verhältnissen von Wandtemperatur (T_w) zu Umgebungstemperatur (T_∞): $1 < T_w/T_\infty < 2$, $L^2/18 \leqslant A_a \leqslant L^2$, $5\times10^7 < Ra < 2\times10^{10}$, $0{,}01 < Ri < \infty$.

СМЕШАННАЯ КОНВЕКЦИЯ ОТ ИЗОТЕРМИЧЕСКИХ КУБИЧЕСКИХ ПОЛОСТЕЙ С МНОЖЕСТВОМ БОКОВЫХ ОТВЕРСТИЙ

Аннотация—Экспериментально исследован смешанноконвективный теплоперенос от гладкой изотермической кубической полости с рядом боковых отверстий. Целью исследования определение потерь в солнечных термоэлектрогенераторах за счет конвекции и изучение механизмов, регулирующих эти потери. Особое внимание поэтому обращается на режимы, протекающие при больших числах Рэлея, Ra, и больших числах Рейнольдса Re; при числах Ричардсона Ri порядка единицы и большой величине отношения температуры стенок резонатора T_w к температуре окружающей среды T_∞. Приведенные данные охватывают диапазоны $1 < T_w/T_\infty < 2$, $L^2/18 \leqslant A_a \leqslant L^2$, $5 \times 10^7 < Ra < 2 \times 10^{10}$ и $0{,}01 < Ri < \infty$.