Natural convection heat transfer within enclosures at reduced pressures
by Peter Kevin Brown

A thesis submitted in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in Mechanical Engineering
Montana State University

Abstract:
Natural convection heat transfer in air within enclosures has been investigated over the pressure range of 2670-86,180 Pa (20-646.4 mm Hg), Dimensionless correlations have been generated from the data; The best correlation found included a correction for the air density: Nul = .342 RaL^1/4 (ρ/ρatn,)^.129 where L is the hypothetical gap width. The Rayleigh number in the experiments ranged over 1x10^3 - 2x10^6. The geometries used were cylinder-cube (inner body-outer body) and cube-cube, with the bodies mounted concentrically in both cases. Temperature profiles at four positions (0°, 34°, 80°, and 160° from the upward vertical) were measured for the cylinder-cube case. The thickening of the boundary layer at low pressures and the region of constant temperature between the bodies at high Ra were clearly observed.

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NOMENCLATURE

Symbol	Description
a	Any characteristic length
b	Boundary layer length on the inner body
C, C_{1-4}	Empirically determined constants
c_p	Constant pressure heat capacity
d	Length of a side of a cube
$D_i (D_o)$	Diameter of a sphere of surface area equal to that of the inner (outer) body
g	Acceleration due to gravity
g_x	Gravitational acceleration in the direction of the x coordinate
Gr	The Grashof number, $g\beta(T_i-T_o)a^3\rho^2/\mu^2$
$\overline{h}$	Average heat transfer coefficient, $q''/(T_i-T_o)$
k	Thermal conductivity of air
L, L', L''	Gap widths as defined on p. 23-24
$\mathring{m}_i$ ($\mathring{m}_o$)	Mass flow rate in the inner (outer) region of the boundary layer
mm Hg	A unit of pressure, 760 mm Hg = 1 atmosphere
M	$(T_m-T_s)/(T_\infty-T_s)$
Nu	The Nusselt number, $\overline{h}a/k$
P	Pressure of air at experimental conditions
Pa	A unit of pressure, 101,325 Pa = 1 atmosphere
Pr	The Prandtl number, $\mu c_p/k$

Symbol	Description
q or q_{conv}	Heat transferred by natural convection
q_{cond}	Heat transferred by conduction through a stagnant fluid under similar conditions
q''	Heat transferred by convection per unit area
Q	q_{conv}/q_{cond}
r	Radius of curvature
r^*	Dimensionless radius defined on p.19
R_i (R_o)	Radius of a sphere of volume equal to that of the inner (outer) body
Ra	The Rayleigh number, Gr·Pr
S	Length of the x coordinate along a body
T	Temperature
T_i (T_o)	Inner (outer) body temperature
T_s	Temperature at the wall
T_∞	Temperature far away from the wall
T^*	Dimensionless temperature defined on p.19
T_m	Temperature at y_m or temperature in the fluid between the inner and outer equivalent conduction layers
$\overline{T}_i$ ($\overline{T}_o$)	Average temperature in the inner (outer) region of the boundary layer
$\overline{T}_m$	Average temperature between the inner and outer equivalent conduction layers
ΔT_r	A reference temperature difference
u	Velocity component in the direction of x
u_{max}	Maximum velocity in the profile

Symbol	Description
x	Coordinate along the body in the direction of flow
y	Coordinate out of the body normal to x
y_m	Location of the velocity maximum in the profile
β	Thermal expansivity of air $(1/V)(dV/dT)$, where V is the volume; for an ideal gas $\beta = 1/T$
Γ_i (Γ_o)	Mass flow rate per unit depth in the inner (outer) region of the boundary layer
δ	Boundary layer thickness or any one of three gap widths L, L', or L"
$\Delta(x)$ or $\Delta_\ell(x)$	Local equivalent conduction layer thickness
$\overline{\Delta}$ or $\overline{\Delta}_\ell$	Average equivalent conduction layer thickness
η	Dimensionless y coordinate, y/δ
μ	Viscosity of air
ρ	Density of air in the experimental conditions
ρ_{atm}	Density of air at atmospheric pressure and standard temperature (298.15°K)

ABSTRACT

Natural convection heat transfer in air within enclosures has been investigated over the pressure range of 2670-86,180 Pa (20-646.4 mm Hg). Dimensionless correlations have been generated from the data. The best correlation found included a correction for the air density:

$$Nu_L = .342\ Ra_L^{1/4}\ (\rho/\rho_{atm})^{.129}$$

where L is the hypothetical gap width. The Rayleigh number in the experiments ranged over 1×10^3 - 2×10^6. The geometries used were cylinder-cube (inner body-outer body) and cube-cube, with the bodies mounted concentrically in both cases. Temperature profiles at four positions (0°, 34°, 80°, and 160° from the upward vertical) were measured for the cylinder-cube case. The thickening of the boundary layer at low pressures and the region of constant temperature between the bodies at high Ra were clearly observed.

CHAPTER I

INTRODUCTION

The phenomenon of natural convection heat transfer within enclosures has received a great deal of attention in recent years. There is a growing demand for an understanding of this phenomenon in such areas as nuclear design, electronic packaging, space heating, and solar collector design. The majority of this effort has been experimental. The nonlinearity and coupling of the governing differential equations of continuity, momentum, and energy have made analytical solutions difficult to find. Those few that exist apply only to relatively simple geometries, such as concentric cylinders or spheres.

In the present study, natural convection heat transfer in air from an isothermal (heated) inner body to an isothermal (cooled) outer body was investigated. More specifically, rates of heat transfer were measured for various temperature differences between the bodies and different pressures of air in the test space. The outer body was cubical; the inner bodies were a vertical cylinder with hemispherical end caps and a cube.

The primary purpose of this investigation was to find an empirical correlation for the heat transferred by natural convection in air from a body to its enclosure as a function of air pressure (i.e., vacuum). The Rayleigh number in these experiments ranged over 10^3- 2×10^6. Since evacuation of the cavity surrounding a heated device, such as a solar collector tube, is an effective way to reduce heat loss [1-3], such a relationship would be quite useful. The correlations found were tested against previous results. In addition, a correlation based on the method of Raithby and Hollands [4] was developed and tested against the data.

Temperature profiles were also measured for some of the heat transfer conditions. The profiles are useful for elucidating general trends, such as the thickening of the boundary layer at low pressure. In addition, they may provide a means of verification for any future numerical or analytical solutions which may be advanced.

CHAPTER II

LITERATURE REVIEW

Natural convection phenomena fall broadly into two categories: convection in an infinite fluid bath, or external convection, and convection within an enclosure, or internal convection. The following review is intended to provide a background for the present investigation and is not a complete survey of research in natural convection.

Dimensional analysis has shown [5,6] that external natural convection may be correlated by

$$Nu_a = f(Gr_a, Pr)$$

where a is some characteristic dimension and

$$Nu_a = \frac{\bar{h}\ a}{k} \qquad \text{(Nusselt number)},$$

$$Gr_a = \frac{g\beta\ \Delta T\ a^3\ \rho^2}{\mu^2} \qquad \text{(Grashof number)},$$

and $$Pr = \frac{\mu c_p}{k} \qquad \text{(Prandtl number)}.$$

Often the effects of Gr_a and Pr can be combined by

$$Nu_a = f(Ra_a), \text{ where}$$

$$Ra_a = Gr_a \cdot Pr \qquad \text{(Rayleigh number)}.$$

In this investigation, Ra_a will be varied largely by variations in ρ (i. e., pressure of the air) and also by variations in a and in ΔT.

Elenbaas [7] developed some early correlations for external natural convection heat transfer from a vertical cylinder:

$$Nu_{d,w}\exp(-2/Nu_{d,w}) = 0.6Ra_{d,w}^{1/3}/Ra_{h,w}^{1/12}$$

and from a horizontal cylinder:

$$Nu_{d,w}\exp(-2/Nu_{d,w}) = 0.16Ra_{d,w}^{1/3}/g(Ra_{d,w})$$

where $g(Ra_{d,w})$ is a function presented graphically. The subscript d refers to the diameter and h to the height of the cylinder, and w means that the fluid properties are to be evaluated at the wall temperature.

King [8] correlated external natural convection from several geometric shapes, including ones similar to those used in this investigation, by

$$Nu_D = 0.60\ Ra_D^{1/4} \text{ for } 10^4<Ra_D<10^9$$

where $$\frac{1}{D} = \frac{1}{\text{vertical dimension}} + \frac{1}{\text{horizontal dimension}} .$$

Holman [6] reports a correlation for a vertical cylinder of

$$Nu_h = 0.59Ra_h^{1/4} \text{ for } \frac{D}{h} \geq \frac{35}{Gr_h^{1/4}}$$

where D is the diameter and h the height of the cylinder.

Lienhard [9] found that laminar external convection could be correlated well by a balance of buoyancy and viscous forces on the body.

Using velocity and temperature profiles for a flat plate, he obtained

$$Nu_b = 0.52Ra_b^{1/4}$$

where b is the length of the thermal boundary layer on the body.

There have been several experimental and analytical investigations of external convection to air at low Ra (low pressure of air, small characteristic dimension, or small ΔT). Saunders [10] performed early experiments on the pressure dependency of convection in air. He found that Nu_h for a vertical flat plate falls well above accepted correlations for $Ra_h < 10^5$. This is seen as a leveling off of Nu to near unity as a pure conduction regime is entered at low pressure. Kyte, Madden, and Piret [11] extended low pressure measurements to a vacuum in which the mean free path of the air molecules becomes comparable to the physical body dimensions. They found for vertical wires that

$$Nu_{D'} = \frac{2}{\ln[1+4.47/(Ra_{D'}\frac{D'}{h})^{0.26}]}$$

in the molecular flow regime. Here h is the height of the wire (cylinder) and D' is the diameter plus twice the mean free path of the gas.

Gryzagordis [12] reports that a vertical plate in air at low pressures obeys the standard correlation

$$Nu_h = 0.555Ra_h^{1/4}$$

down to $Ra_h=10$, in contrast to Saunders [10] and to Suriano and Yang [13]. The latter study suggests that the Nusselt number should fall above this correlation for Ra_h lower than about 500. The interested reader is referred to the literature for further details.

Analytical and numerical solutions of external natural convection are based on solving the coupled differential equations of continuity, momentum, and energy in the boundary layer approximation [5]. The solutions usually take the form of series solutions in Ra for the temperature and stream functions. Chiang, Ossin, and Tien [14] solved the case of an isothermal sphere and presented graphical results of velocity and temperature profiles and local Nusselt numbers. Lin and Chao [15] and Saville and Churchill [16] presented similar results for two-dimensional and axisymmetric cases. Sparrow and Gregg [17] solved the equations for a vertical cylinder and developed a condition under which it may be correlated by the flat plate result:

$$\frac{2^{1/2}}{Gr^{1/4}}(h/D) \leq 0.15.$$

Minkowycz and Sparrow [18] used the technique of local non-similarity to solve the case of a vertical cylinder. The cases solved ranged from near the flat plate solution to a factor of four deviation from it. The reader is directed to the literature for further information. Finally, Churchill and Churchill [19] have presented a correlating equation for all geometries of external natural convection, in both laminar and turbulent flow:

$$Nu^{1/2} = Nu_o^{1/2} + \left\{\frac{Ra/300}{[1+(0.5/Pr)^{9/16}]^{16/9}}\right\}^{1/6} .$$

Values are tabulated for Nu_o, the Nusselt number in the limit as Ra approaches zero, and characteristic lengths are given for various geometries. In addition, corrections are outlined to account for non-Newtonian fluids and simultaneous heat and mass transfer.

The great majority of work in internal natural convection has been experimental. An early study by Elenbaas [20] with parallel vertical plates led to the following correlation:

$$Nu_D = \frac{1}{2}\frac{D}{h} Ra_D[1-\exp(-35h/DRa_D)]^{3/4}$$

where D is the plate spacing and h is the height. Holman [6], for the same geometry, gives the following correlation:

$$Nu_D = 0.197Ra_D^{1/4} (D/h)^{1/9} \quad \text{for} \quad 6x10^3 \leq Ra_D \leq 2x10^5$$

with the lengths defined as before. In an analytical study of parallel plates, Batchelor [21] recommends using

$$Nu_D = 0.48Ra_D^{1/4} (h/D)^{3/4} \quad \text{for} \quad \frac{Ra_D}{500} > \frac{h}{D} .$$

Newell and Schmidt [22] made an analytical study of long rectangular enclosures with adiabatic top and bottom surfaces and isothermal walls at two different temperatures. The resulting correlations were

$$Nu_D = 0.0547 Gr_D^{0.397} \quad , \quad \frac{h}{D} = 1 \quad ,$$

and
$$Nu_D = 0.155 Gr_D^{0.315} (h/D)^{-0.265} \quad , \quad 2.5 \leq \frac{h}{D} \leq 20 \; .$$

Randall, Mitchell, and El-Wakil [23] experimented with rectangular enclosures tilted at various angles and reported

$$Nu_D = 0.118 [Ra_D \cos^2(\phi - 45°)]^{0.29} \; .$$

In these experiments, ϕ, the angle from the horizontal, varied from 45-90°, the aspect ratio h/D varied from 9-36, and Ra_D varied from 2.8×10^3-2.2×10^5.

Flack, Konopnicki, and Rooke [24] performed heat transfer experiments with an isoceles triangular enclosure with an adiabatic bottom face and isothermal rising walls of different temperatures. The apex angle was varied so that the aspect ratio (height/base width) ranged from 0.29-0.87. They correlated their results by

$$Nu_l = c_3 (Gr_l)^{c2} + 1.589/\cos\theta$$

where θ is the angle of the rising walls from the horizontal and l is their length. The constants are tabulated for values of θ. The trigonometric term in the correlation is to account for conduction near the apex, where the isothermal surfaces are close together.

Scanlan, Bishop, and Powe [25] measured convection between concentric spheres. They defined an effective thermal conductivity

$$\frac{k_{eff}}{k} = \frac{q(r_o - r_i)}{4\pi k \Delta T r_i r_o}$$

and recommend the following correlation:

$$\frac{k_{eff}}{k} = 0.117 Ra_L^{0.276} \quad , \quad 1.4 \times 10^4 < Ra_L < 2.5 \times 10^6$$

where L is the gap width $r_o - r_i$.

Kuehn and Goldstein [26-28] have analytically and experimentally treated the case of concentric and eccentric horizontal cylinders. They present correlations based on the ratio of actual convection heat transfer to that which would be conducted through a stagnant film layer under the same conditions. For air they present

$$Q = q_{conv}/q_{cond} = 0.159 Ra_L^{0.272} \quad , \quad 2.1 \times 10^4 \leq Ra_L \leq 9.6 \times 10^4$$

where L is the gap width. $L/D_i = 0.8$ in these experiments. Itoh, Fujita, Nishiwaki, and Hirata [43] propose a correlation for concentric cylinders based on a different characteristic length:

$$Nu = 0.20 Ra^{1/4} \quad , \quad Ra \geq 7.1 \times 10^3$$

where $$Nu = \frac{\bar{h}_i r_i \ln(r_o/r_i)}{k} = \frac{\bar{h}_o r_o \ln(r_o/r_i)}{k}$$

and $$Ra = \frac{g\beta\Delta T[\sqrt{r_i r_o}\ \ln(r_o/r_i)]^3\ c_p \rho^2}{k\rho}\ .$$

There exist few studies of internal natural convection in air at low pressures. Mack and Hardee [29] have examined this problem for concentric spheres, and Mack and Bishop [30] have done so for concentric cylinders. These are analytical studies in which power series solutions in Ra are developed for the temperature and stream functions. Graphical results are presented for velocity and temperature profiles and local Nusselt numbers. The reader is referred to the literature for additional information.

Koshmarov and Ivanov [31] have published an experimental investigation of natural convection between concentric cylinders at very low pressures. Over the range of $10^{-3} < Gr_L < 10^4$ they correlate the convection results by

$$Q = A \left\{\frac{2}{1+T_o/T_i}\right\}^{(1-A)}$$

where $$A = \left\{1+\frac{19}{3}\,\frac{Kn_i}{\ln(D_o/D_i)}\right\}^{-1}\ .$$

Temperatures here are absolute. Kn_i is the Knudsen number, defined by

$$Kn_i = \frac{\text{mean free path}}{D_i}\ .$$

Recently, Marker and Leal [32] have reported an analytical result for convection in shallow vertical annuli (three-dimensional flow):

$$Nu_h = \frac{A}{\gamma \ln(\frac{1+\gamma}{\gamma})} \quad 1+2.76\times10^{-6} \frac{(Ra_h A)^2}{\gamma \ln(\frac{1+\gamma}{\gamma})^2}$$

where $$A = \frac{h}{r_o - r_i} \quad \text{and} \quad \gamma = \frac{Ar_i}{h} .$$

Raithby and Hollands [4] have developed an analysis similar to, but more refined than, Lienhard's [9] boundary layer analysis. The method involves a balance of bouyancy and viscous forces very close to the wall, along with an assumed form for the temperature profile. New correlations generated by this method include

$$Q = \frac{0.317 \ln(D_o/D_i)}{L^{3/4}(D_i^{-3/5}+D_o^{-3/5})^{5/4}} Ra_L^{1/4}$$

for concentric cylinders and

$$Q = \frac{0.61 L^{1/4} Ra_L^{1/4}}{D_o D_i (D_i^{-7/5}+D_o^{-7/5})^{5/4}}$$

for concentric spheres. This method will be gone into in greater detail later (see Chapter V).

The present investigation is essentially a continuation of work by Warrington [33] with the same apparatus. His work involved a wide variety of enclosure geometries and fluids with a large range of Prandtl numbers. The best correlation overall from his data was

$$Nu_L = 0.425 Ra_L^{0.234} (L/R_i)^{0.498} .$$

For the cylinder-cube geometry (inner body-outer body) the best correlation was

$$Nu_b = 0.593Ra_b^{0.240}(L/R_i)^{0.434}$$

and for the cube-cube geometry,

$$Nu_L = 0.322Ra_L^{0.244}(L/Ri)^{0.466}Pr^{0.0185} .$$

The better correlations for air in all geometries were

$$Q = 0.468Ra_L^{0.172} (L/R_i)^{0.167} ,$$

$$Nu_L = 0.612Ra_L^{0.207} (L/R_i)^{0.508} ,$$

and

$$Nu_b = 0.570Ra_b^{0.228} (L/R_i)^{0.162} .$$

In these equations, L is the gap width between hypothetical concentric spheres of volumes equal to the actual volumes of the inner and outer bodies. R_i is the radius of such an inner sphere. b is the distance traveled by the thermal boundary layer on the actual inner body, assuming no flow separation. These correlations will be tried with the present data and compared to any new correlations found.

CHAPTER III

EXPERIMENTAL APPARATUS AND PROCEDURE

The apparatus used in this investigation (see Figure 3.1) was a cubical test space 26.67 cm (10.5 in) along an inner side, fabricated from 1.27 cm (0.5 in) type 6061 aluminum. A water jacket enclosure of the same material surrounded this cube; it measured 35.56 cm (14.0 in) along an inner side. The two enclosures were fastened together with machine screws and sealed with silicone rubber sealant. Access was gained into the test space by removing the water jacket lid and a 25.4 cm (10.0 in) circular cover in the top face of the test space. The water jacket lid was sealed with a gasket; the circular cover was flanged and sealed with an O-ring and high-vacuum grease. The 3.2 cm (1.25 in) channels separating the two enclosures contained cooling water. The water flow was separately adjustable along each of the six cube faces. The cooling water was collected from the test apparatus through a drain manifold and pumped through a chiller apparatus, into an insulated storage tank, and from there through the supply manifold and back into the test apparatus.

Two heated inner bodies were used in this investigation: a cylinder with hemispherical end caps, 11.43 cm (4.5 in) in diameter and 22.61 cm (8.9 in) in overall length; and a cube measuring 12.70 cm (5.0 in) along a side. The inner bodies were each mounted on a 1.27 cm (0.5 in) stainless steel stem of .159 cm (.0625 in) wall thickness. The leads to the thermocouples and heater tapes on the inner body passed through this

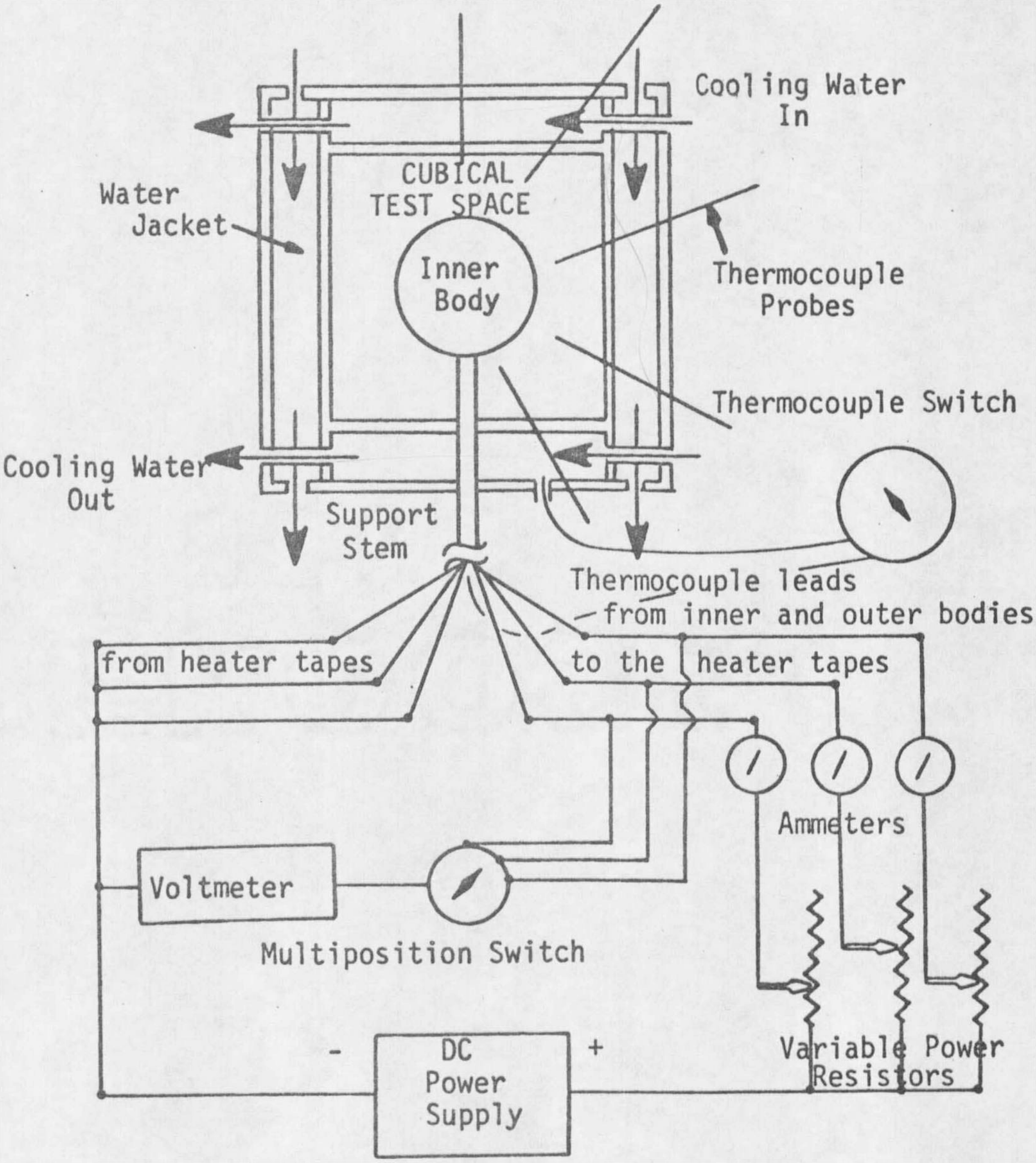

Figure 3.1 Schematic of Heat Transfer Apparatus With Supporting Instrumentation

stem. The stem in turn passed through centrally located holes of 1.27 cm (0.5 in) diameter in the bottom faces of the test space and water jacket enclosures. The holes were grooved and fitted with O-rings to seal against the stem. The stems were insulated with heat-shrinkable tubing to minimize convection heat loss to the working fluid. Flow visualization pictures taken in a similar apparatus (but with a transparent face into the test space) have clearly shown this to be effective [33]. The stem passed through a 1.27 cm (0.5 in) Conax compression fitting beneath the water jacket enclosure. This fitting eliminated leakage of air between the shrink tubing and the stem and allowed the inner body to be positioned vertically within the system.

Heat was supplied to the surface of the inner bodies by a series of heater tapes of .51 cm (.2 in) thickness, .32 cm (.126 in) width, and 28.87 ohms/m (8.8 ohms/ft) resistance per unit length. Input voltages to the tapes were individually variable by means of Ohmite rheostats (0-35 ohms, 150 watts, 2.07 amperes maximum) connected in series with the tapes. The inner bodies were constructed of sheet copper, of either .064 cm (.025 in) (cylinder) or .318 cm (.125 in) (cube) thickness. Insulating material was packed inside the inner bodies to cut down on convection within them. There were thus three factors which contributed to maintaining an isothermal heated surface on the inner bodies: 1. Individually controllable heater tapes attached to separate areas of the surfaces; 2. Low convective activity within the bodies; and 3. Fabrication from a high thermal conductivity material which promotes a "conduction smearing" effect.

Temperatures were monitored in these experiments by means of copper-constantan thermocouples and a United Systems Corp.-Digitec 268 digital millivoltmeter. There were 25 thermocouples epoxied .318 cm (.125 in) from the inner surface of the cubical test space enclosure. The number of thermocouples per face of the cube varied from 3 to 7. All thermocouples on a given face were connected in parallel, since it has been found that the temperature across any face varied no more than .8°C (1.5°F) [33].

Thermocouples were mounted flush on the inner surface of the cylindrical inner body and .165 cm (.065 in) beneath the outer surface of the cubical inner body. All thermocouples were flattened for about 2.5 cm (1 in) along the inner bodies and the outer body to minimize conduction heat transfer along the leads and resultant error in temperature measurement. Thermocouples were positioned on the inner body so that at least one thermocouple lay within each heater tape winding.

Temperature profiles between the inner and outer bodies were measured with thermocouples epoxied in the ends of .16 cm (.0625 in) o.d. stainless steel tubes. Four such temperature probes were used in this investigation, positioned at 0°, 34°, 80°, and 160° from the upward vertical. The 80° probe was mounted in a vertical plane passing diagonally through the center of the cube; the other three probes were in a vertical plane through the center and perpendicular to two opposite vertical faces. The stainless steel probes were positioned within probe holders, each of which consisted of a .95 cm (.375 in) o.d. stainless steel tube with a .159 cm (.0625 in) i.d. Conax compression fitting welded on the outer end and .95 cm (.375 in) SAE threads cut in the inner end. The probe passed through a .18 cm (.07 in) hole in the inner end of the probe holder, and was sealed and held in place by the Conax fitting. The cubical test space enclosure had holes drilled and threaded to accept the probe holes. These holes were seated and sealed against the holders with O-rings approximately .25 cm (.1 in) from the inner surface of the cube. The probes passed through .19 cm (.075 in) holes in these seats and thus into the test space. The probe holders passed through 1.02 cm (.4 in) holes in the water jacket enclosure and were sealed there with O-rings.

To perform experiments, the inner body was positioned within the test space, the thermocouple leads were connected to the millivoltmeter along with a standard reference at 0°C (32°F), and the heater tape leads were connected to the power supply. The circular cover and water jacket lid were attached and sealed, and all temperature probes were positioned within their holders. The test space was evacuated to the desired vacuum by means of a Sargent-Welch mechanical vacuum pump. Above 1 mm Hg, pressures were monitored by a mercury U-tube manometer and below 1 mm Hg by a General Electric Thermocouple Vacuum Gauge. Power was applied to the heater tapes and adjusted until the desired isothermal condition was reached on the inner body. Simultaneously, cooling water flow rates were adjusted so that an isothermal condition was obtained on the outer body. Typically, temperature variation across either of the bodies ranged from .5-1.1°C (1-2°F) once steady state conditions were reached, which took about 2 hours. The data then taken were: thermocouple readings on the inner and outer bodies, voltage and current in each heater tape circuit, and air pressure within the test space. These raw figures were converted to inner and outer body temperatures and total power consumed (i.e., total heat transfer rate, assuming negligible power loss in the leads to the heater tapes).

The natural convection heat transfer must be obtained from the total heat transfer by subtracting off the radiation between the bodies and conduction down the stem. This correction was made by evacuating the system over a period of two days to pressures in the range of 3-13 Pa (25-100 microns Hg), where convection presumably does not occur [37]. Fifteen heat transfer data points were taken in this pressure range with temperature differences between the bodies ranging from 11.1-47.2°C (20-85°F). A plot of heat transfer against temperature difference resulted in a straight line. The equation of this line was then used in subsequent tests to remove radiation and stem conduction contributions.