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Energy Transport in Porous Insulator Systems II. Performance of Vertical Enclosures

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Part of
The National Program
for
Building Thermal Envelope Systems
and Insulating Materials

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ENERGY TRANSPORT IN POROUS INSULATOR SYSTEMS

II. PERFORMANCE OF VERTICAL ENCLOSURES

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Part of
The National Program for
Building Thermal Envelope Systems and Insulating Materials

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FOREWORD

This is one of a series of reports to be published describing research, development, and demonstration activities in support of the National Program for Building Thermal Envelope Systems and Insulating Materials. The National Program involves several federal agencies and many other organizations in the public and private sectors who are addressing the national objective of decreasing energy wastes in the heating and cooling of buildings. Results described in this report are part of the National Program through delegation of management responsibilities for the DOE lead role to the Oak Ridge National Laboratory.

Unresolved differences of opinion exist between the principal author and the sponsors of this project on the need for completeness in reporting experimental details and theoretical interpretations to justify the conclusions contained herein. Special attention is called to the disclaimer on the inside cover of this report.

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I. INTRODUCTION AND BACKGROUND:

The current report of findings is concerned with the thermal performance of permeable, porous insulator systems which are perfectly sealed and not subjected to any forced convective flows. Accordingly, fluid mechanical coupling of sub-elements of the insulator system derive only from natural convective effects. Of course, underlying molecular conductive and radiative transport effects are ever present.

In earlier studies[1] on the effects of air intrusion on the thermal performance of permeable, porous insulator systems, a number of effects were identified:

(a) Intrusive air flows of several kinds can serve to degrade the insulating value of various permeable insulator systems.

(b) A permeable insulator that is sealed on all sides but one can have its effective R-value degraded by fluid mechanical coupling of its interior to an air space which is contiguous to the insulator's unsealed surface.

(c) Permeable insulators which are arranged "in series," or "in parallel" or in a multiplicity of series-parallel combined arrangements are generally characterized in terms of classically employed algebraic formulae for resistors "in series," or "in parallel" or in some combined arrangement. For several cases where fluid mechanical coupling derived from forced convective air intrusion plays a role, these earlier studies[1] indicate that the classically employed R-values and formulae do not adequately characterize the performance of such thermal insulator systems.

Future reports will deal with forced convective air intrusion effects and related processes.

II. SYSTEMS STUDIED:

Studies reported herein are concerned with a limited number of insulator systems types. Low density glass fiber batts have been arranged in vertical orientations within the test section of the Stony Brook 4.0 ft. by 5.0 ft. double heat transfer gauge test apparatus. In this apparatus, the test section depth can be varied from less than one inch to dimensions of nine inches and above. The principal boundaries of the test section are defined by the innermost aluminum plates of the two 4.0 ft. by 5.0 ft. heat transfer gauges. Extensive thermocouple information on the temperatures and temperature distributions of all four heat transfer gauge plates is obtained and recorded. Individual thermocouples are read to better than $0.1^{o}F$. A schematic of the double heat transfer gauge test apparatus is shown in Figure (1). A discussion of the formulae, methods, and expected accuracy of R-value measurements derived from the double heat transfer gauge test apparatus is given in Appendix (A). Two general kinds of observations have been made: cellularization effects and convectively induced air intrusion effects. These are discussed in the following sections.

(II.a) Cellularization Effects:

A test section of six inch depth is established (4.0'x 5.0'x 0.5') in the double heat transfer gauge test section. Within this test section various arrangements of permeable and/or semi-permeable membranes

are employed to help suppress whatever natural convective effects may otherwise apply to the insulating system. The six inch depth of the test section is defined by four glass fiber batt test panels in series. Each is one and one-half inches in thickness. The one and one-half inch thick test batt panels were cut from batts of greater thickness. A schematic of the cellularization test arrangement is shown in Figure (2). Planes A, B, and C correspond to locations which could be left free of any membrane (left BARE), or where Polyethylene, TYVEK (a semipermeable membrane), or Aluminized Mylar could be placed. Polyethylene is known to be largely transparent in the 5 to 20 micron infrared (thermal) regime.[2] Spectrophotometric studies at Stony Brook have shown that a 5 mil sheet of polyethylene transmits about 90 percent of incident infrared radiation in this spectral regime. TYVEK is a spun polyolefin sheet material which provides for effusive transport of water vapor through its semipermeable body. Aluminized (two sides) Mylar is a nonpermeable sheet material that is highly reflective (infrared emissivity of the order of 0.05) in both the visible and infrared regimes. Additional data on the properties of these membrane materials are given in Appendix (B). Table (1) gives the results of a series of cellularization experiments for a mean test section temperature of 70°F and for a temperature drop across the system of 62°F. The data show that the thermal performance of this low density (ρ = 0.45 lb. per cubic foot) glass fiber sample is affected by both free convective and infrared radiative heat transfer mechanisms. The Aluminized Mylar is successful in suppressing both of these unwanted transport mechanisms. The improvement in R-value from a 6-inch non-cellularized sample (A, B, C, - Bare) to a cellularized (with Aluminized

Mylar) 6-inch sample is nineteen percent.

A brief study of the effects of "black" boundary conditions (at the test section extremes) versus "highly reflective" boundary conditions has also been carried out. For the noncellularized (A, B, C - Bare) case and for $\overline{T}$ = 55°F, ΔT = 36°F, we obtain:

Black Boundaries: R = 18.3 ft^2 hr. ^oF/Btu. ±0.2

Reflective Boundaries: R = 19.3 ft^2 hr. ^oF/Btu. ±0.2

Infrared spectrophotometric studies employing instrumentation similar to that discussed earlier (above) were used to determine the effective infrared emissivities associated with "black" and with "reflective" boundary conditions. "Black" conditions were achieved with a black paint displaying an infrared emissivity averaging 0.90. "Reflective" corresponds to a polished aluminum surface displaying an infrared emissivity of no more than 0.05.

Bases for the small uncertainties in the performance of individual heat transfer gauges are given in Appendix (A). The R-values given in this report are mean values based on the readings of the two in-line heat transfer gauges. R-value uncertainties shown are based on the mean deviation from the mean.

(II.b) Free Convectively Induced Air Intrusion Effects.

It is well known that vertical air spaces with supercritical Rayleigh numbers can give rise to substantial free convective heat transfer effects in the transverse (horizontal) direction. Less well understood are the quantitative effects that such free convection processes display when the air spaces in question are contiguous to a permeable, porous insulator.

Questions of this sort need to be answered in order to properly assess the effective R-value of a vertical cavity (perfectly sealed against forced convective effects) which is only partially filled with a permeable insulator.

Figure (3) shows the Apparatus Schematics for the study of Free-Convectively Induced Air Intrusion Effects. The double heat transfer gauge test apparatus is employed to define a nine inch test section. Arranged in series are a 1.5 inch air space, a 6.0 inch glass fiber batt, and a 1.5 inch air space. A variety of membrane samples and combinations were employed at the two air-insulator interfaces. Results to date are given in Table (2).

III. DISCUSSION OF RESULTS

Quantitative experimental results to date are detailed in Tables (1) and (2). A number of general conclusions can be drawn from these detailed observations.

(1) Cellularization of a thick vertically arranged low density glass fiber batt insulator system can be employed to increase the overall R-value of the system. The results of Table (1) indicate that cellularization is effective due to both the suppression of the modest natural convective heat transfer processes and the infrared radiative transport processes which operate. Mylar, which is aluminized on both sides is found to be best as a thermally significant cellularization barrier.

(2) The rated R-values of a permeable, porous, low-density glass fiber batt can be degraded when it is situated contiguous to

Insulator System	Measured R-Value
	$[Btu/ft^2\ {}^oF.hr]^{-1}$
A,B,C - Bare	15.6 ±0.5
A,B,C - Polyethylene	16.6 ±0.8
A,B,C - TYVEK	17.0 ±0.3
A,B,C - Aluminized Mylar	18.6 ±0.1

Table (1). Cellularization Effects for 6.0 inch Test Section and various Membrane Arrangements. $\bar{T} = 70^oF$. $\Delta T = 62^oF$. Glass Fiber Batts (4) in Series. Each is 1.5" × 48" × 60". Extreme Boundaries Covered in Aluminized Mylar for all Tests.

Cold Side	Hot Side	$\bar{T}(^oF)$	$\Delta T(^oF)$	R $ft^2 \cdot hr. ^oF/Btu$
Bare	Bare	58.3	33.3	9.5 ±0.2
		65.7	41.0	8.3 ±0.2
		73.6	49.0	7.7 ±0.3
		82.5	59.0	7.2 ±0.3
Polyethylene	Polyethylene	55.2	38.6	21.0 ±0.0
		66.0	52.0	18.0 ±1.0
Aluminized Mylar	Aluminized Mylar	54.0	34.0	19.5 ±0.6
		60.0	43.0	18.2 ±0.6
		71.0	58.0	16.5 ±0.6
TYVEK	TYVEK	57.1	38.0	18.0 ±0.8
		66.0	49.9	15.7 ±1.0

Table 2a. Measured R-Values of a glass fiber batt installation for the experimental arrangements and boundary conditions of Figure (3). Cases (a): Surface Barrier no. 1 and no. 2 identical.

Cold Side	Hot Side	$\bar{T}(^oF)$	$\Delta T(^oF)$	R $ft^2 \cdot hr. {}^oF/Btu$
Bare	Polyethylene	54.0	32.0	16.8 ±0.2
		67.6	45.0	13.0 ±1.0
Polyethylene	Bare	53.1	33.0	18.2 ±0.4
		57.0	38.0	17.0 ±0.5
		65.9	50.0	14.9 ±1.0
Bare	Aluminized Mylar	59.0	38.0	13.4 ±0.4
		65.0	44.5	12.1 ±0.6
Aluminized Mylar	Bare	55.4	35.0	17.2 ±1.0
		67.4	51.6	14.2 ±0.6
		75.0	61.5	13.3 ±1.0

Table 2b. Measured R-Values of glass fiber batt installation arrangement and boundary condition of Figure (3). Cases (b). Only one surface barrier left BARE.

Cold Side	Hot Side	$\bar{T}(^oF)$	$\Delta T(^oF)$	R $ft^2 \cdot hr. ^oF/Btu$
TYVEK	Polyethylene	54.4	34.3	21.0 ±0.6
		62.4	46.2	18.0 ±0.6
Polyethylene	TYVEK	52.3	31.6	17.5 ±0.0
		61.7	43.8	15.2 ±0.4
TYVEK	Aluminized Mylar	56.0	33.0	18.7 ±0.7
		63.0	42.0	15.3 ±1.0
		74.0	55.0	15.0 ±0.7
Aluminized Mylar	TYVEK	52.7	34.7	17.8 ±0.7
		62.6	46.0	17.0 ±0.5

Table 2c. Measured R-Values of a glass fiber batt installation arrangement and boundary conditions of Figure (3). Cases (C): Surface barriers no. 1 and no. 2 neither bare nor identical.

and fluid mechanically coupled to an air space which sustains free convective effects. Tables (2a) - (2c) detail the magnitude of these effects for a specific set of insulator arrangements. The influence of the effective air space emissivity on the results of Figure (2) are complex. Although a low air space emissivity (highly reflective boundary condition at the heat transfer gauge surfaces) generally leads to a high R-value for the air space itself, the correspondingly large temperature drop in the air space leads to more substantial free convective effects which may couple to the thermal performance characteristics of the porous glass fiber material. In essence, such a system of fluid mechanically coupled insulators must be viewed in a unified manner. The frequently invoked relation of independently operating resistance in series ($R_T = R_1 + R_2 + R_3 + \cdots$) is generally inadequate for the systems under study. This is discussed further in Appendix (C).

IV. PRACTICAL IMPLICATIONS OF FINDINGS TO DATE

(1) For perfectly sealed, vertical wall installations of glass fiber batts which are contiguous to air spaces, rated R-value degradation may result. This suggests that for unwisely insulated (underinsulated, partially bare) existing housing stock, the thermal performance of walls - free of any forced convective wind conditions - may be substantially inferior to that which was previously thought to be the case. This implies the need for quantitative reconsideration of the

estimated cost-effectiveness of wall cavity-filling reinsulation of existing buildings. In-situ thermal performance of some underinsulated walls may be more substantially improved through reinsulation than previously thought - even where the wall section is taken to be free of any forced convective wind conditions.

(2) Some existing building wall sections may have unwisely installed glass fiber batts which do not butt tightly against a non-permeable barrier at the batt end. Interferometric studies at Stony Brook of such batt-end, air-space coupling are in progress. Observations to date clearly show that air space free convective effects penetrate a bare batt-end.

(3) New construction, modern wall insulation systems should be free of the problems noted above. Nevertheless, careless or unwise construction practices may give rise to a number of such free convectively induced degradation of R-values.

(4) None of the observations reported herein refer to forced convective effects on the in-situ R-value of a permeable, porous insulator system. Nevertheless, it is found that a number of membrane multiplicities and arrangements (both semipermeable and nonpermeable) may be employed to upgrade the NO WIND thermal performance of a wall installation. In this regard, semipermeable membranes may prove thermally useful while also serving to permit effusive transport of moisture.

(5) It is anticipated that forced convective flows may further modify the in-situ thermal performance of a permeable wall

insulation system, where perfect wall sealing is not achieved and where significant wind conditions prevail. Studies of these effects are currently in progress. It is not premature to suggest, however, that a multiplicity of membrane barriers would be useful in maintaining rated R-value performance of glass fiber insulated wall systems that are subject to signficant wind conditions.

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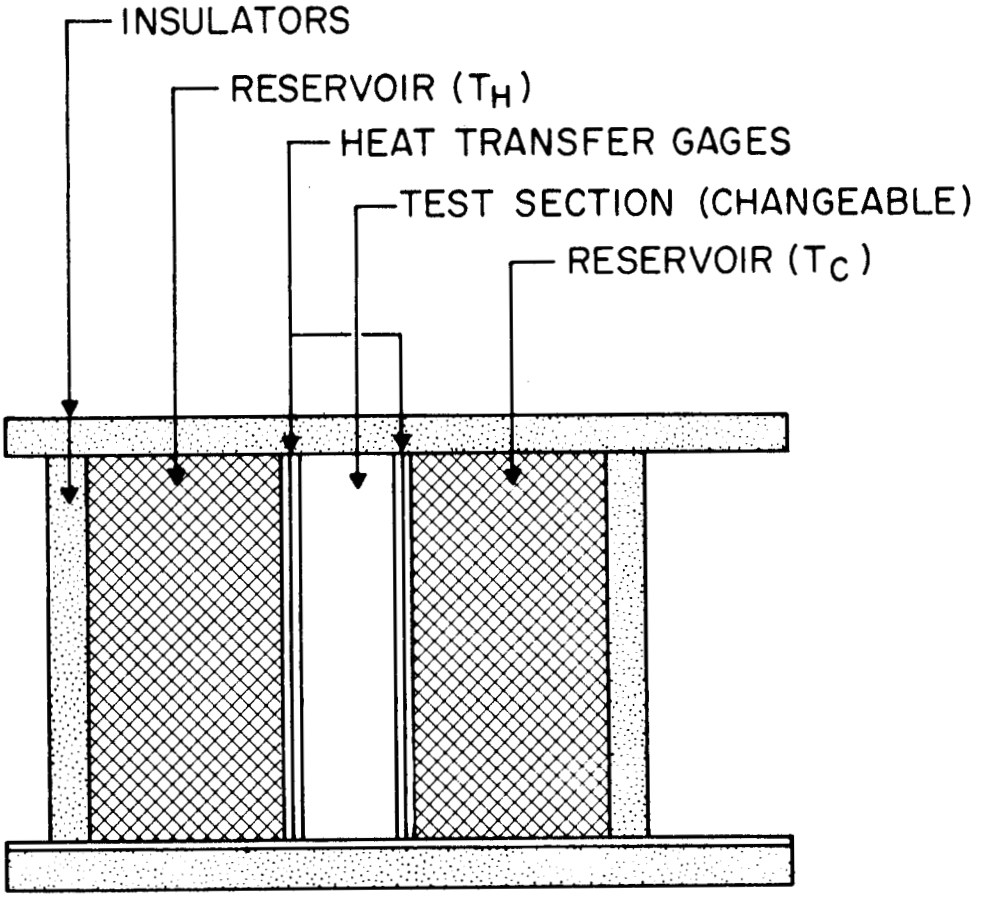
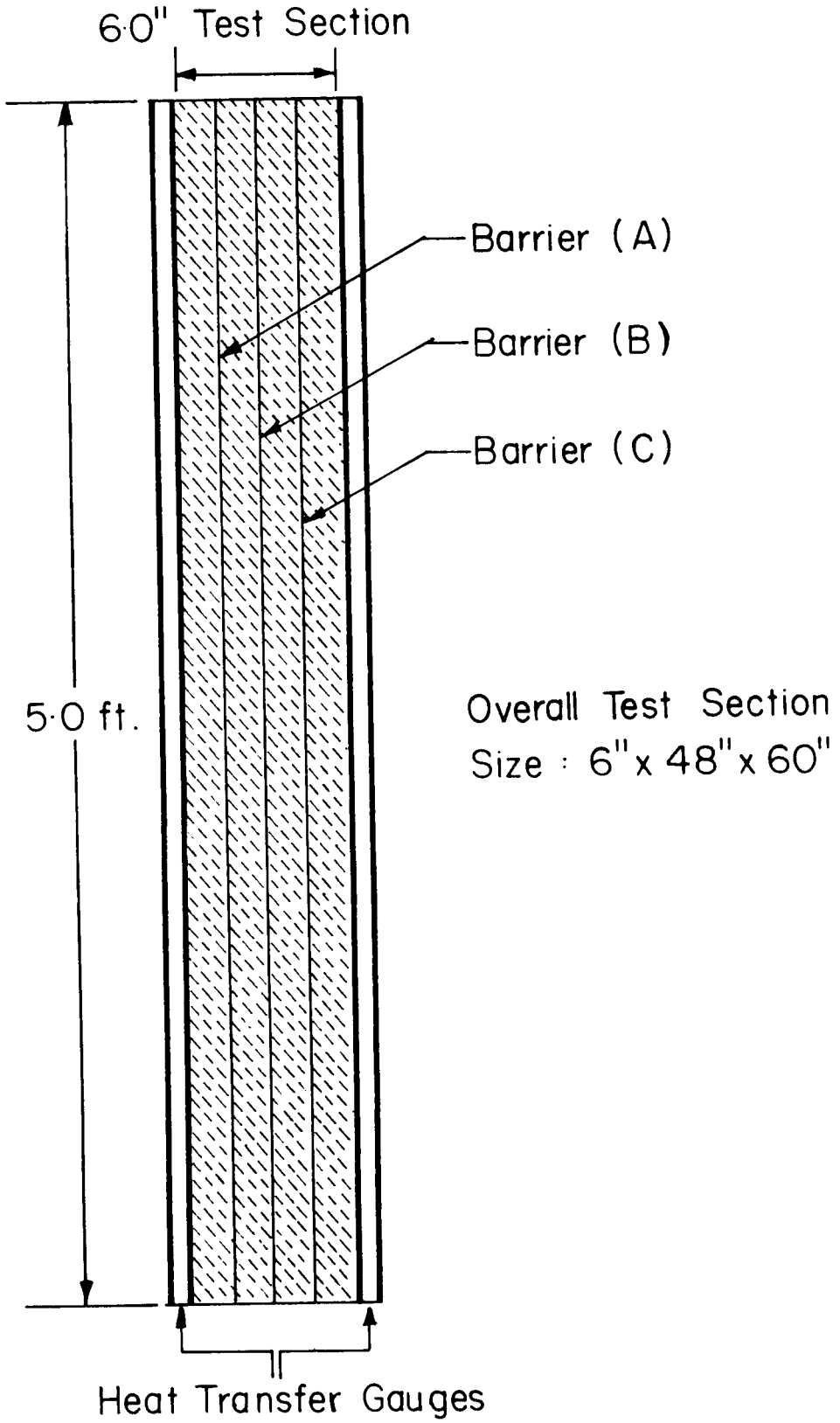


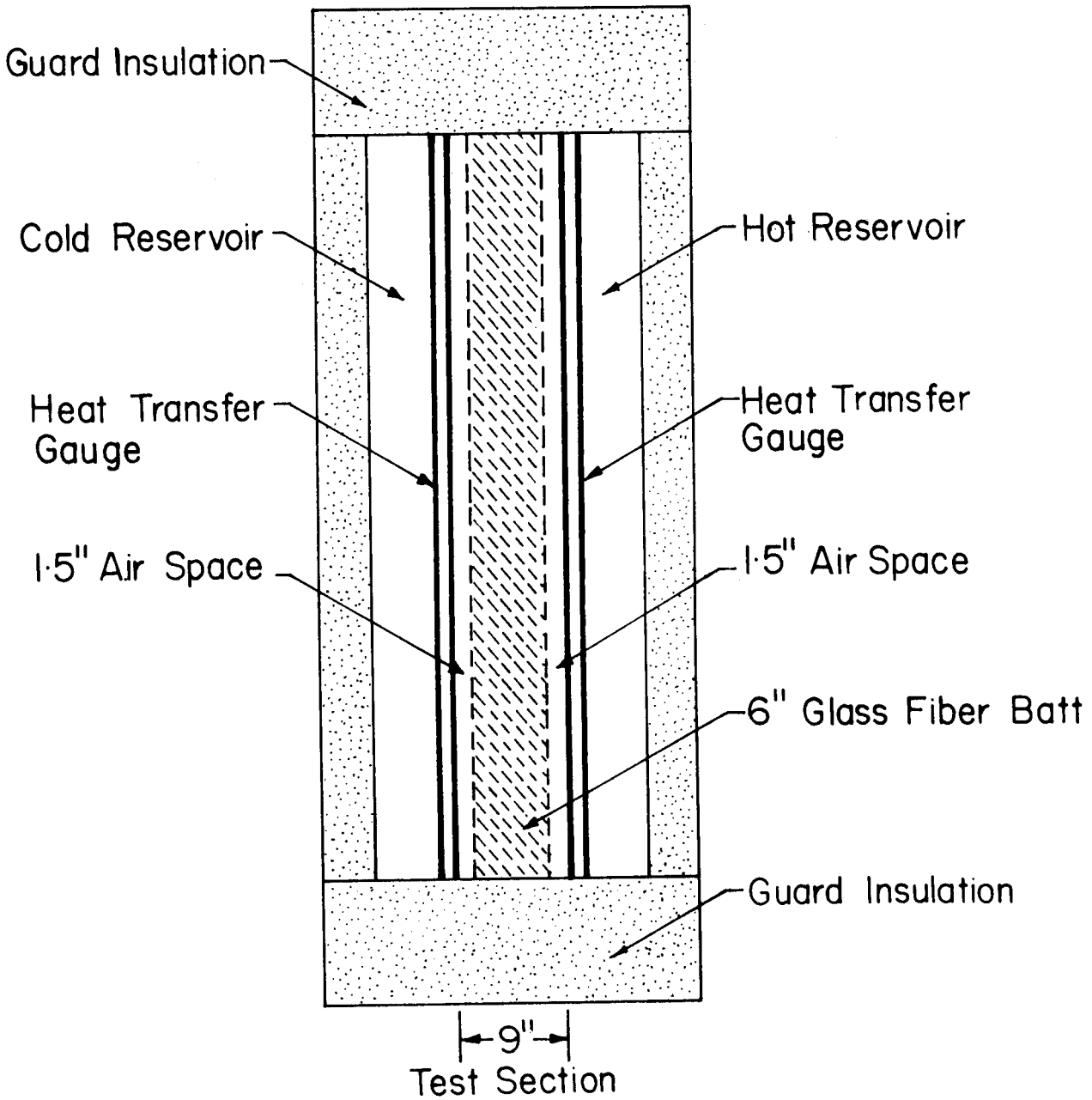
S.U.N.Y. — DOUBLE HEAT TRANSFER
GAUGE TEST APPARATUS SCHEMATICS

Figure (1)



48" x 60" x 6" Test Section Features for Vertical Glass Fiber Batt Cellularization Tests

Figure (2)



48" x 60" x 9" Test Section Arrangement for the Study of Free Convectively Induced Air Intrusion Effects on Vertical Glass Fiber Batts

Figure (3)

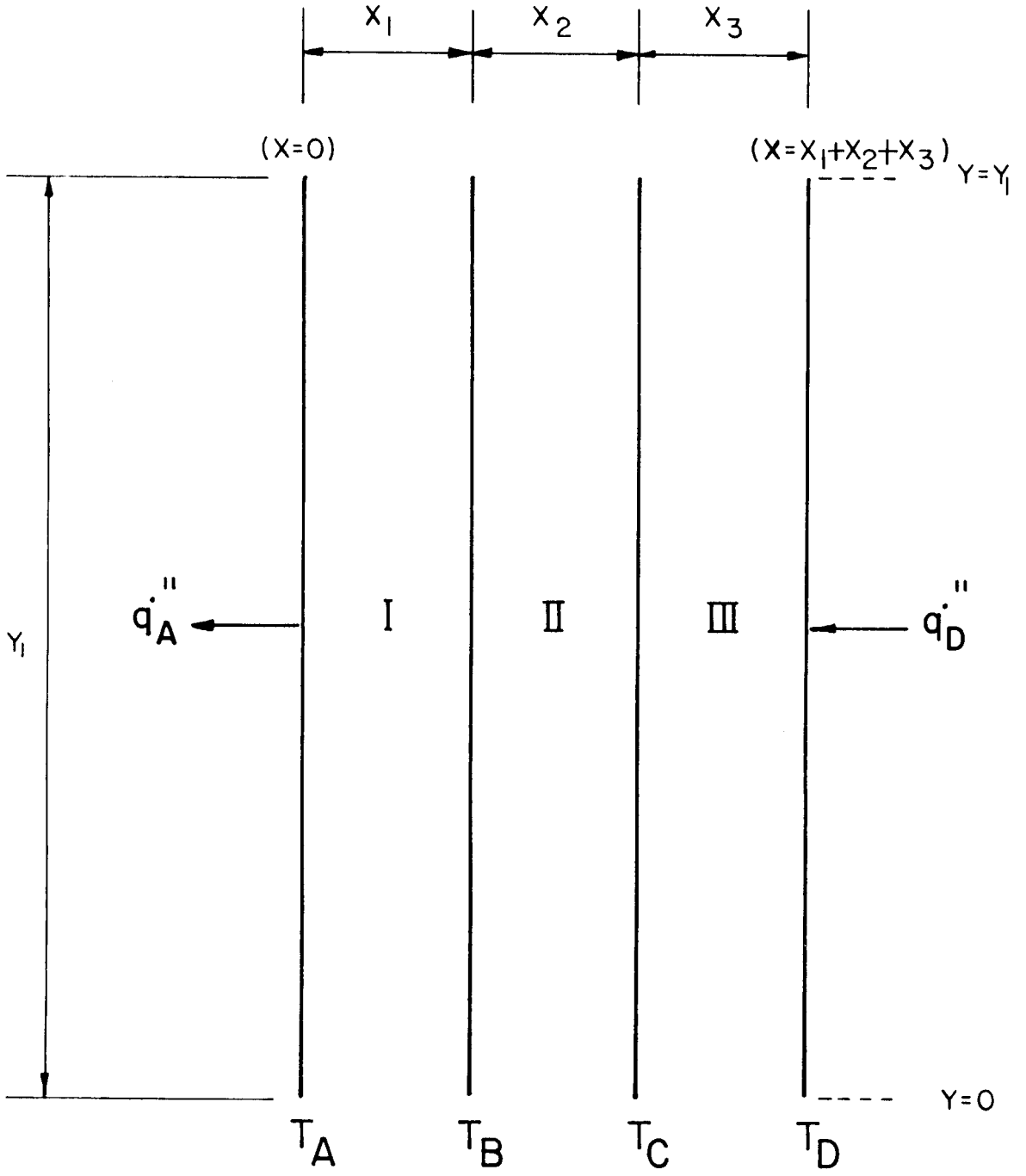


Figure (4)

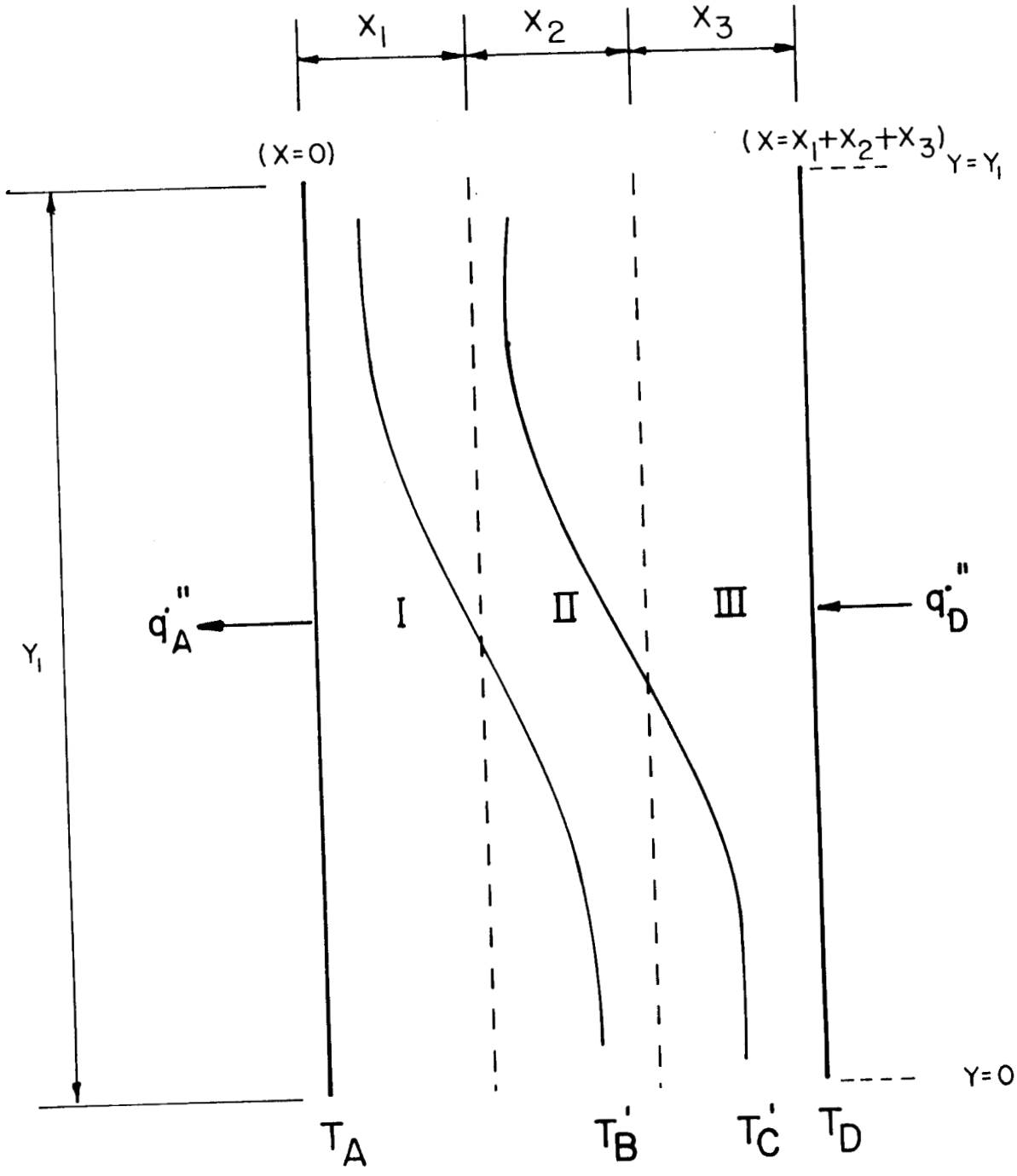


Figure (5)

APPENDIX (A)

PERFORMANCE OF AN AIR-GAP HEAT TRANSFER GAUGE

I. Introduction and Background

The use of an air-gap heat transfer gauge has been exploited at SUNY Stony Brook as a means of measuring heat transfer rates through air spaces, window systems, insulated wall sections, and other materials characterized by a significant thermal resistance. The purpose of this apendix is to discuss the accuracies expected of measurements so made.

SUNY - Stony Brook Report ETR-1277-2 provides a description of the kind of test apparatuses to which such a heat transfer gage may be applied.[3]

The general advantages of the air-gap heat transfer gauge are the following:

(1) Use of air as the gauge's insulation of interest provides a standard insulator whose thermal conductivity is well known and free of degradations which may afflict manufactured insulators.

(2) The two surfaces (of massive thermally conductive plates) which define the air-gap are of low infrared emissivity, thereby suppressing radiant heat transfer rates. With known emissivities, these rates are calculable (exactly) from fundamental physical considerations and the known surface temperatures (T_1 and T_2).

(3) The thermal conductivity of the (air) gap is known with great accuracy. The air-gap sustains no convective heat transfer due to the extremely small Rayleigh number ($\lesssim 200$). The conductive heat transfer rate is calculable (exactly) from fundamental physical considerations and the known

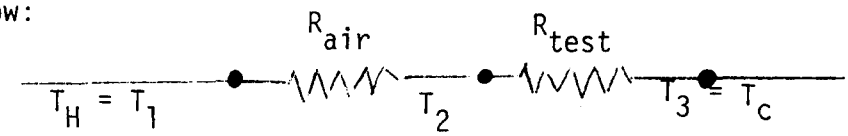
surface temperatures (T_1 and T_2).

In the following we examine some heat transfer gauge expected performances.

II. Heat Transfer Gauge

(A) Representation

The resistance network equivalent to that examined in ETR-1277-2 is shown below:



The conductive and radiative heat transfer rates (associated with the air gap) are given by:

$$\dot{q}''_{cond} = k_{air} \left(\frac{\Delta T}{\delta_a}\right)$$

and

$$\dot{q}''_{rad} = E\sigma(T_1^{\,4} - T_2^{\,4})$$

where

$$\frac{1}{E} = \frac{1}{E_1} + \frac{1}{E_2} - 1$$

E_1 = infrared emissivity of surface at $T = T_1$

E_2 = infrared emissivity of surface at $T = T_2$

k_{air} = thermal conductivity of air

δ_a = air gap dimension

$\Delta T = T_1 - T_2$

(B) Error Analysis

The uncertainty in the determination of the heat transfer rate through a heat transfer gauge is of two general kinds. Those due to random errors and those due to systematic errors. The effects of random

errors can generally be suppressed by a multiplicity of observations. Systematic errors generally imply systematic uncertainties in calculated/measured results. Thus, the measured value of the thermal resistance of a test specimen contains errors which derive from two sources:

(a) errors in the measurement of the heat transfer rate through the heat transfer gauges ($\dot{q}''_a$)

(b) errors due to transverse heat exchange (conductive leaks) with the environment (through the insulated periphery of the test section). ($\dot{q}''_b$)

In general, $\dot{q}''_b$ must be calculated or measured. For a double heat transfer test apparatus, $\dot{q}''_b$ is no larger than the difference between the readings of the two, in line, heat transfer gauges. Accordingly, the uncertainty in R_{test} associated with $\dot{q}''_b$ derives from the uncertainty in the effects of the "guard insulation".

We consider now the effects of random errors on heat transfer gauge performance:

Let $R = f(X,Y)$, where x and y are the measured quantities subject to random error. Let R_c, x_c, y_c be the true values and r, x, y the corresponding departures from them. Then

$$R = R_c + r, \qquad X = x_c + x, \qquad Y = y_c + y$$

We consider r, x, y to be random variables with zero means and respective standard deviations of σ_r, σ_x, σ_y. Expanding f(X,Y) in a Taylor's series around (x_c, y_c) gives

$$r = \left(\frac{\partial f}{\partial x}\right) x + \left(\frac{\partial f}{\partial y}\right) y$$

and

$$r^2 = \left(\frac{\partial f}{\partial x}\right)^2 x^2 + \left(\frac{\partial f}{\partial y}\right)^2 y^2 + 2\left(\frac{\partial f}{\partial x}\right)\left(\frac{\partial f}{\partial y}\right) xy$$

and having expected values

$$\sigma_r^{\ 2} = E\{r^2\} = \left(\frac{\partial f}{\partial x}\right)^2 \sigma_x^{\ 2} + \left(\frac{\partial f}{\partial y}\right)^2 \sigma_y^{\ 2}$$

It is assumed that (x) and (y) are independent ($\therefore \overline{xy} = 0$).

(C) Errors due to Uncertainties in T_1, T_2, δ_A, T_c, E and k_{air}:

If we consider a heat transfer gauge to be gold plated and to display the following characteristics (Test Case):

δ_a = (3/16)" = 0.015625 ft.

k_{air} = 0.0152 BTU/hr.ft.^{o}F

E = 0.0178

and ω_E = 0.0026

where ω_E = the uncertainty in (E). From the infrared Handbook (for Gold)

T^oK	$(E_1 = E_2)_{2\mu}$	$(E_1 = E_2)_{14\mu}$
273°K	0.038	0.0134
293°K	0.04	0.015

are employed to deduce the E = 0.0178 value. Also:

$T_c = 0^oC = 492^oR$

$T_2 = 24.8^oC = 76.64^oF = 536.64^oR$

$T_1 = 30.0^oC = 86^oF = 546^oR$

ω_x = 0.002" = 0.0001667 ft.

ω_K = 0.000304 BTU/hr.ft.^{o}F (two percent)

$\omega_{T_c} = \omega_{T_1} = \omega_{T_2} = 0.18^oF$

Thus

$$\dot{q}''_a = \dot{q}''_{cond} + \dot{q}''_{rad} = 9.1054 + 0.1812 = 9.2866 \text{ BTU/hr.ft}^2$$

Note that the radiative component to $\dot{q}''_a$ is only some 2 percent of the total. This exactly calculable component is somewhat higher for cases where polished aluminum is used instead of gold.

Also, we have

$$R = (T_2 - T_c)/\dot{q}''_a = (T_2 - T_c)/\left\{(k)\left(\frac{T_1-T_2}{\delta_a}\right) + E\sigma(T_1^4-T_2^4)\right\}$$

$$\left(\frac{\partial R}{\partial T_1}\right) = -\frac{1}{(\dot{q}''_a)^2}\left\{\frac{k}{\delta_a} + 4E\sigma T_1^3\right\}\{T_2-T_c\}$$

$$\left(\frac{\partial R}{\partial T_2}\right) = +\frac{1}{(\dot{q}''_a)^2}\left\{\frac{k}{\delta_a} + 4E\sigma T_2^3\right\}\{T_2-T_c\} + \frac{1}{\dot{q}''_a}$$

$$\left(\frac{\partial R}{\partial T_c}\right) = -\frac{1}{\dot{q}''_a}$$

$$\left(\frac{\partial R}{\partial \delta_A}\right) = \frac{+(T_2-T_c)}{(\dot{q}''_a)^2}\left\{\frac{k(T_1-T_2)}{\delta_A^2}\right\}$$

$$\left(\frac{\partial R}{\partial E}\right) = \frac{-(T_2-T_c)}{(\dot{q}''_a)^2}\left[\sigma(T_1^4 - T_2^4)\right]$$

$$\left(\frac{\partial R}{\partial k}\right) = -\frac{T_2-T_c}{(\dot{q}''_a)^2}\left(\frac{T_1-T_2}{\delta_A}\right)$$

Substitution of parameters of the problem into the above gives

$$R = 4.81 \pm 0.182$$

or an uncertainty of less than four percent. Part of this uncertainty is due to random errors in the temperature. If we average nine temperature readings we can reduce ω_R to a value of

$$\omega_r = 0.1182$$

and

$R = 4.81 \pm 0.118$ and an error of some 2.5 percent.

As noted earlier, use of a double heat transfer gauge test apparatus assures that the transverse heat loss/gain is no larger than the difference between the heat transfer rates sustained by each of the two, in line, heat transfer gauges.

(D) Accuracy of Test Methods

The performance of a properly constructed and used air-gap heat transfer gauge is expected to yield heat transfer rates whose systematic uncertainties are no more than about three percent. These uncertainties are for the individual heat transfer gauges themselves and are unrelated to $\dot{q}''_b$ values which may derive from the limited effectiveness of any "guard insulation" materials.

Shortly after completion of the manufacture of the double heat transfer gauge test apparatus, the agreement of readings provided by the two heat transfer gauges was determined for a range of boundary temperatures and for the case where the test section was "shorted out" (the heat transfer gauges were placed in good thermal contact with each other). Agreement was good to better than two percent. In other tests, the two heat transfer gauges were separated by a one inch air space (black boundaries for the test section). Again, agreement between the two heat transfer gauge readings was good to better than two percent.

Uncertainties in the thermal performance of a test sample may derive from the limited effectiveness of the "guard section" of the test apparatus. The double heat transfer gauge test apparatus is guarded by a system of commercially available foamed polyisocyanurate insulators ($R \approx 80$).

For the ranges of boundary temperature conditions and test sections reported herein, the R-value reported is based on the average of the two heat transfer gauge readings. The uncertainties reported are based on the difference in these two readings.

It may thus be concluded that the test methods examined are capable of achieving measured R-values with errors less than five percent.

It is also significant that the test section heat transfer processes need not be associated with a uniform system.[3] Window systems, insulated wall systems, etc., generally sustain three-dimensional heat transfer processes.

It is also important to note that once a test section's thermal resistance is measured satisfactorily by the methods given above, any forced convective flows imposed on the test section will generally give rise to quite <u>different</u> readings from each of the two heat transfer gauges. Once the forced convective enthalpy flow rate change is determined (from independent measurements on inlet/outlet temperatures and mass flow rate), these two quite different readings may be taken as quantitative measures of the "hot side" versus "cold side" performance of the test section. They may also be employed directly to a <u>complete</u> energy balance on the forced convectively influenced system under study.

APPENDIX (B)

SOME PHYSICAL PROPERTIES OF MATERIALS USED IN THESE EXPERIMENTS

All test materials employed in these experiments were obtained from commercial sources.

glass fiber batts: Mean Density = 0.45 lb./ft.^{3}

Polyethylene[2,6] Sheet: thickness = 0.004 inch.

$$\left\{\begin{array}{l}\text{Mean infrared transmissivity} \\ \text{in the 5-20 micron spectral} \\ \text{regime}\end{array}\right\} = 0.9$$

Mylar[7] Sheet: thickness = 0.001 inch mylar plus 0.00025 inch Aluminum on each side.

infrared emissivity $\leq$ 0.05

TYVEK Sheet: thickness = 0.005 inch Infrared emissivity and porosity not determined.

Black Paint (for boundaries): 100% Carbon Black Pigment

Infrared emissivity $\simeq$ 0.9.

APPENDIX (C)

THERMAL TRANSPORT THROUGH SYSTEMS OF RESISTANCES "IN SERIES"

It is frequently assumed that the characterization of thermal resistances to be "in series" is an obvious, geometrically determined feature of an insulator system. Figure (4) is a schematic which geometrically suggests that three insulators, (I), (II), and (III) are "in series". It is known[4] that for n independently operating thermal resistances "in series" that the total resistance, R_t, is given by

(C-1) $$R_t = R_1 + R_2 + \cdots + R_n \ .$$

If one assumes that (Figure 4):

(C-2) $$y_1 \gg \sum_i x_i \qquad \text{and}$$

(C-3) T_A, T_B, T_C, T_D each corresponds to an isothermal vertical plane, and that

(C-4) Regimes (I), (II), and (III) are each thermally open (nonadiabatic) but fluid mechanically closed (no exchange of mass) with respect to neighboring regimes, and if

(C-5) $\dot{q}''_A$ and $\dot{q}''_D$ are both constants, independent of time, and if

(C-6) the heat transfer rates (in the y-direction) at the upper and lower boundaries are negligible:

(C-7) $$\left|\dot{q}''_{y,u}\right| \sim \left|\dot{q}''_{y,d}\right| \ll \left|\dot{q}''_A\right|$$

(C-8) $$\left|\dot{q}''_{y,u}\right| \sim \left|\dot{q}''_{y,d}\right| \ll \left|\dot{q}''_D\right|$$

<u>then</u>, one may frequently accept as useful the approximation

(C-9) $$\dot{q}''_{y,u} = 0 \quad \text{and} \quad \dot{q}''_{y,d} = 0$$

For such an approximation, the overall nature of the heat transfer process is quasi-steady and one-dimensional (in the x-direction). Then

(C-10) $$\dot{q}''_A = \dot{q}''_B = \dot{q}''_C = \dot{q}''_D = \text{constant}$$

and one may define a thermal resistance[4] for each space such that

(C-11) $$\dot{q}''_x R_1 = T_B - T_A$$

(C-12) $$\dot{q}''_x R_2 = T_C - T_B$$

(C-13) $$\dot{q}''_x R_3 = T_D - T_C$$

where R_1, R_2, R_3 are the so-defined thermal resistances for regimes (I) (II), and (III), respectively.

Combinations of the above equations, (C-11) - (C-13), gives

(C-14) $$\dot{q}''_x (R_1+R_2+R_3) = T_D - T_A$$

and the summation

$$R_t = R_1+R_2+R_3$$

has the character of thermal resistances in series. Further, it is to be noted that R_1 is <u>defined</u> in terms of a uniform slab of thickness x_1 bounded by isothermal T_B at the nonpermeable barrier $x = x_1$ and T_A at at the nonpermeable barrier $x = 0$. Corresponding definitions apply to R_2 and R_3. It is notable that the physical content and heat transfer character of each of the several regimes may be quite complex and different from that of its neighbor(s). Nevertheless, the imposition of the constraints (C-2) - (C-9) permits treatment of the heat transfer problem in terms of simple resistances in series, (C-1).

However, should it not be possible to satisfy (C-2) - (C-9), then more general equations than (C-10) - (C-13) will be required to characterize the system[5].

For the studies reported herein, absence of nonpermeable membranes at the various regime interfaces (particularly those of Figure 3.) may give rise to gaseous flows across vertical interface planes. For such cases, geometrically vertical planes are generally not isothermals, and regimes (I), (II), and (III) are fluid mechanically coupled.

Accordingly, the vertical planes 0, x_1, $x_1 + x_2$, $x_1 + x_2 + x_3$, do <u>not</u> necessarily correspond to isothermals T_A, T_B, T_C, and T_D respectively.

Figure (5) shows the type of isothermal fields found by recent interferometric and temperature probe studies of multiple insulation elements which sustain air intrusion effects. Here T_A again lies in the plane $x = 0$ and T_D again lies in the plane $x = x_1 + x_2 + x_3$. T_A and T_D correspond to carefully maintained test section boundaries. However, the isothermals T_B and T_C are no longer planar. For such a case, equations (C-11) - (C-13) are not adequately descriptive of the heat transfer processes sustained by each of the regimes (I), (II), and (III).

Thus, an air-permeable thermal insulating system, made up of elements in <u>geometric series</u> and bounded by isothermals T_A and T_D, subjected to a variety of internal air intrusion effects (as found in these studies), may display a measured overall thermal resistance, R_t^*. However, R_t^* need not, in general, be equal to $R_1+R_2+R_3$.

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