Thermal Scale Modeling of Radiation-Conduction-Convection Systems

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An investigation is made of thermal scale modeling applied to radiation-conduction-convection systems with particular emphasis on the spacecraft cabin atmosphere/cabin wall thermal interface. The thermal similitude criteria, scaling techniques and the results of an experimental investigation are presented. The "Modified Material Preservation," "Temperature Preservation," "Scaling Compromises" and "Nusselt Number Preservation" scale modeling techniques and their inherent limitations and problem areas are described The compromised scaling techniques of mass flux preservation and heat-transfer coefficient preservation were investigated further and show promise of giving adequate thermal similitude while preserving both gas and temperature in the scale model. The use of these compromised scaling techniques was experimentally demonstrated in tests of full scale and ¼ scale models. It is concluded that either mass flux or heat-transfer coefficient preservation may result in adequate thermal similitude depending on the system to be modeled. Heat-transfer coefficient preservation should give good thermal similitude for manned spacecraft scale modeling applications.

Nomenclature

A = area
c = specific heat
C, C_1 = constants
$\mathbf{e}$ = unit vector
F = radiation exchange factor
g = acceleration of gravity
Gr = Grashof number $(\rho^2 g\beta L^3 T_0/\mu^2)$
h = heat-transfer coefficient
k = thermal conductivity
K = conductance per unit area to environment
L = characteristic length
Nu = Nusselt number (hL/k)
M = molecular weight or thermal mass per unit area
P = pressure
P_* = nondimensional pressure $(\rho L^2 P/\mu^2)$
Pr = Prandtl number $(\mu c/k)$
q_s = heating rate per unit area
q_v = heating rate per unit volume
Q = heating rate
Re = Reynolds number $(\rho \bar{v} L/\mu)$
t = time
T = temperature
T_0 = characteristic temperature
v = velocity
v_* = nondimensional velocity $(v/\bar{v})$
w = mass flow rate
α = heat-transfer area/cross-sectional flow area
β = coefficient of thermal expansion
δ = thickness
θ = nondimensional temperature (T/T_0)
θ_f = nondimensional fluid temperature $(T_f - \bar{T}_f)/T_0$
μ = viscosity
ρ = density
σ = Stefan-Boltzmann constant
τ = nondimensional time $(kt/\rho c L^2)$

Operators
∇ = gradient operator
∇_* = nondimensional gradient operator

Subscripts
e = environment
f = fluid
g = gas or gravity
m = model
n = surface normal
o = characteristic value
p = prototype
r = reference
$(\bar{\ })$ = average values

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Introduction

MOST spacecraft thermal scale modeling studies have dealt with systems involving only radiation and conduction modes of heat transfer. These studies range from verification of scale modeling techniques using relatively simple configurations[1–4] to thermal scale modeling of an actual spacecraft concept.[5] Thermal scale modeling of manned spacecraft involves convection as well as radiation and conduction modes of heat transfer. The similitude criteria and discussions of possible scaling techniques have been presented in the literature,[6,7] however, no experimental verification of the scaling techniques has been reported for manned spacecraft. A thermal scale modeling investigation of free convection in heated enclosures has been reported[8] in which an 8 × 8 × 8 ft prototype room with a convector heater and a ¼ scale model were built and tested. Radiation-conduction scaling criteria were used for temperature preservation in the ¼ scale model. No attempt was made to modify the convective heat transfer and air at atmospheric pressure was used in both model and prototype.

In order for thermal scale modeling to be a practical tool for manned spacecraft applications effective thermal scale modeling techniques must be developed for the cabin atmosphere/spacecraft cabin wall thermal interface. This paper describes the development and demonstration of practical thermal scale modeling techniques applicable to radiation-conduction-convection systems with particular emphasis on the cabin atmosphere/cabin wall thermal interface.

Scale Modeling Criteria

The thermal scale modeling criteria may be developed from the equations which govern the system thermal energy balance. The governing equations when written in nondimensional form result in dimensionless groups of parameters. These dimensionless groups must remain invariant for thermal similitude to exist between similar systems. These equations can be written in nondimensional form as:

Heat conduction

$$\partial\theta/\partial\tau = \nabla_*{}^2\theta + (L^2 q_v/kT_0) \quad (1)$$

Boundary condition at solid/fluid interface

$$(q_s/\sigma T_0{}^4) - (k/\sigma T_0{}^3 L)(\mathbf{e}_n \cdot \nabla_*\theta)_w = \int_j (\theta^4 - \theta_j{}^4)dF_j - (k_f/\sigma T_0{}^3 L)(\mathbf{e}_n \cdot \nabla_*\theta_f)_w \quad (2)$$

or for two-dimensional conduction

$$(q_s/\sigma T_0{}^4) + (k\delta/\sigma T_0{}^3 L^2)(\nabla_*{}^2\theta - \partial\theta/\partial\tau) = \int_j (\theta^4 - \theta_j{}^4)dF_j - (k_f/\sigma T_0{}^3 L)(\mathbf{e}_n \cdot \nabla_*\theta_f)_w \quad (3)$$

Thermal similitude within the solid elements of geometrically similar systems having identical radiative surface properties is achieved by keeping the following dimensionless groups invariant: heat conduction—$(k/\sigma T_0{}^3 L)$ or for two dimensional conduction—$(k\delta/\sigma T_0{}^3 L^2)$, volume heat sources—$(L^2 q_v/kT_0)$, surface heat flux—$(q_s/\sigma T_0{}^4)$, heat transfer to fluid—$(k_f/\sigma T_0{}^3 L)$. The invariance of this last dimensionless group assumes that thermal similitude exists for the fluid elements of the systems as well as for the solid elements.

The fluid elements of the system are governed by the energy, momentum and continuity relationships. These relationships may be written in nondimensional form as

Energy (no internal heat sources or viscous dissipation)

$$(k/\rho c)(\rho c/k)_f\, \partial\theta/\partial\tau + RePr\mathbf{v}_* \cdot \nabla_*\theta_f = \nabla_*{}^2\theta_f \quad (4)$$

Momentum (incompressible flow with buoyancy effects)

$$(k/\rho c)(\rho c/k)_f(Re/Pr)\, \partial\mathbf{v}_*/\partial\tau + Re^2\mathbf{v}_* \cdot \nabla_*\mathbf{v}_* = -\nabla_* P_* + Re\nabla_*{}^2\mathbf{v}_* - Gr\theta_f\mathbf{e}_g \quad (5)$$

Continuity (incompressible flow)

$$\nabla_* \cdot \mathbf{v}_* = 0 \quad (6)$$

Thermal (and dynamic) similitude within the fluid elements of similar systems then requires the following dimensionless groups to remain invariant: Reynolds number—$Re = \rho\bar{v}L/\mu$, Prandtl number—$Pr = \mu c/k$, Grashof number—$Gr = \rho^2 g\beta L^3 T_0/\mu^2$, fluid/solid transients—$(k/\rho c)(\rho c/k)_f$. The last dimensionless group $(k/\rho c)(\rho c/k)_f$ when kept invariant preserves the relationship between the transient response of the fluid and that of the solid. However, for thermal scale modeling applications involving the manned spacecraft cabin atmosphere-cabin wall interface, the fluid (gas) transients have negligible effect on the spacecraft thermal response. Consequently the invariance of this group is not required for thermal similitude in this case.

Scale Modeling Techniques

Four scaling techniques are considered for the cabin atmosphere-cabin wall thermal interface. These are the Modified Material Preservation, Temperature Preservation, Scaling Compromises and Nusselt Number Preservation techniques. Table 1 gives a comparison of these techniques. The Modified Material and Temperature Preservation techniques are straightforward attempts to meet the scaling criteria. The Scaling Compromises technique attempts to achieve thermal similitude without completely satisfying the scaling criteria. The Nusselt Number Preservation technique uses the thermal scale model to experimentally determine the Nusselt number as a function of Reynolds and Grashof numbers and uses these results in conjunction with a thermal math model to determine the prototype performance.

Modified Material Preservation

The material preservation technique used for radiation-conduction systems keeps the same materials in model and prototype and meets the scaling criteria by increasing the temperature and heat flux in the model. The scaling criteria for systems involving convective heat transfer may be met by a modified material preservation technique which accounts for the variation of gas thermal conductivity with temperature. This technique requires the scale model temperature and heat fluxes to be increased to a greater extent than the normal material preservation technique. The required model to prototype temperature ratio is given by

$$T_m/T_p = (k_g/L)_m{}^{1/3}(L/k_g)_p{}^{1/3} \quad (7)$$

The modified material preservation technique also requires the wall material to be changed (or in the case of two-dimensional wall conduction, the proper selection of wall thickness) to meet the scaling criteria at the higher temperature.

Since the Prandtl number does not vary substantially with temperature and since the Reynolds number may be adjusted to the desired value, their preservation in the scale model is easily accomplished. For equal Grashof numbers in the model and prototype the scale model pressure must be increased such that

$$P_m/P_p = (\mu k_g{}^{1/3}/g^{1/2}L^{11/6})_m/(\mu k_g{}^{1/3}/g^{1/2}L^{11/6})_p \quad (8)$$

Table 1 Comparison of scaling techniques

Scaling technique	Thermal similitude	Model temperature	Gas in model	Simulation of prototype gravity field: 0-g	Simulation of prototype gravity field: 1-g
Modified material preservation	Exact	Increased[a] over prototype	Same as prototype	Good prospects	Increased pressure required
Temperature preservation	Exact	Same as prototype	Thermal[b] conductivity scaled	Reduced pressure required	Good prospects
Scaling compromises	Approximate[c]	Same as prototype	Same as prototype	Reduced pressure required	Good prospects
Nusselt number preservation	Partial[d]	Approximately same as prototype	Same as prototype	Good prospects	Increased pressure required

[a] Excessive temperature and heat fluxes in the scale model limits use of "Modified Material Preservation."
[b] The availability of suitable gases limits the use of "Temperature Preservation."
[c] Degree of similitude depends on scaling compromise used and system being scaled.
[d] The use of "Nusselt Number Preservation" requires a verified math model for the radiation-conduction aspects of the system.

The use of nonmetals (seals, insulation, etc.) in the scale model effectively limits the use of the Modified Material Preservation technique to scale ratios greater than about 0.5.

Temperature Preservation

The temperature preservation technique maintains equal temperature in the model and prototype and meets the scaling criteria by requiring a reduced thermal conductivity for the model materials. This technique has been used successfully for radiation-conduction systems at various scale ratios, and has considerable flexibility for two-dimensional conduction systems where the model material thickness need not be scaled geometrically. The application of this technique to systems involving convective heat transfer depends on the availability of gases with substantially lower thermal conductivities than that of the prototype gas. The scale ratios which are possible using selected gases to simulate air at 70°F, are depicted in Fig. 1.

As with the modified material preservation technique, the preservation of Reynolds in the scale model is easily accomplished and the Prandtl number variation between gases is generally insignificant. The preservation of the Grashof number in the model requires the model pressure to be set such that

$$P_m/P_p = (\mu/Mg^{1/2}L^{3/2})_m/(\mu Mg^{1/2}L^{3/2})_p \tag{9}$$

The model scale ratio and pressure ratio are fixed by the gas in the prototype and the gas chosen for the model. For example a prototype using air at atmospheric pressure could be simulated with a $\frac{3}{8}$ scale model with Freon 11 at pressure of about 0.5 atm.

Application of the temperature preservation technique to thermal scale modeling of systems involving convective heat transfer must be evaluated separately for each particular system of interest. Practical problems that might arise in using this scaling technique include: gas availability, gas toxicity, compatibility between gas and model materials, liquefaction of the gas and lack of reliable thermophysical property data for the gas.

The application of the temperature preservation scaling technique to the cabin atmosphere-cabin wall thermal interface is severely limited by the availability of suitable gases. Of the gases considered, only Freon 11, Krypton and Xenon allow scale ratios less than 0.5. Krypton and Xenon are rare gases and are not readily available. Freon 11 is available, however, it liquifies at 75°F at atmospheric pressure. Low-pressure operation of a $\frac{3}{8}$ scale model would be feasible using Freon 11 since its boiling point can be reduced to about -30°F by lowering the pressure to 1 psia.

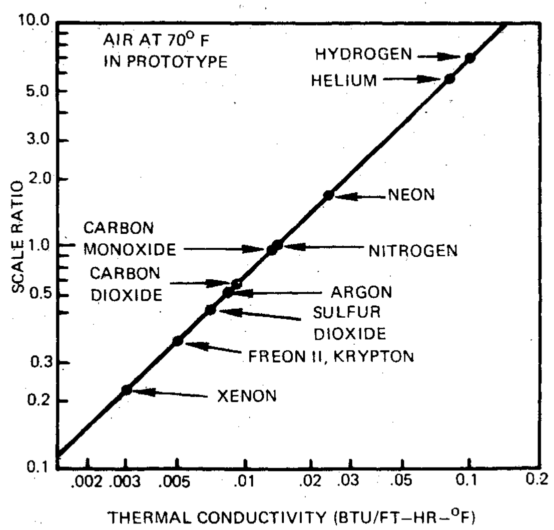

Fig. 1 Candidate scale ratios for temperature preservation.

Scaling Compromises

The problems associated with the modified material preservation and temperature preservation scaling techniques limits their usefulness. Consequently other techniques must be sought for thermal scale modeling of systems involving-convection.

By allowing scaling compromises it may be possible to achieve adequate thermal similitude while preserving both gas and temperature in the scale model. Thermal similitude for the wall is achieved by preserving the convective heat-transfer term in Eq. (2). This term may be written as

$$(k_g/\sigma T_0{}^3L)(\mathbf{e}_n \cdot \nabla_* \theta_g)_w = q_{\text{conv}}/\sigma T_0{}^3 \tag{10}$$

Where q_{conv} is the convective heat-transfer rate per unit area of surface. The convective scaling criteria can then be written as

$$(q_{\text{conv}})_m/(q_{\text{conv}})_p = (T_0{}^4)_m/(T_0{}^4)_p \tag{11}$$

The necessary and sufficient conditions for satisfying Eq. (11) are, of course, given by the scaling criteria. However, two techniques which approximately satisfy Eq. (11) are suggested by rewriting this equation with the convective heat transfer in terms of either heat-transfer coefficient and temperature difference between gas and wall, or mass flux and gas temperature change through the system, i.e., either

$$h_m(T_g - T_w)_m/h_p(T_g - T_w)_p = (T_0{}^4)_m/(T_0{}^4)_p \tag{12}$$

or

$$(\rho c\bar{v}\Delta T_g)_m/(\rho c\bar{v}\Delta T_g)_p = (T_0{}^4)_m/(T_0{}^4)_p \tag{13}$$

Temperature preservation using the same gas in model and prototype requires that both heat-transfer coefficient and mass flux be preserved, i.e.,

$$h_m/h_p = (\rho\bar{v})_m/(\rho\bar{v})_p = 1 \tag{14}$$

Even though the heat-transfer coefficient and mass flux cannot be preserved simultaneously, preservation of either may result in adequate thermal similitude.

Preservation of the heat-transfer coefficient may be achieved if its dependence on the system parameters is known. In practical applications the heat-transfer coefficient is usually based on analytical, semiempirical or empirical results for simple flowfields (e.g., flow over a flat plate). These results typically relate the Nusselt number to the Reynolds number for forced convection cases by an equation of the form

$$Nu = CRe^nPr^m \tag{15}$$

and to the Grashof number for free convection cases by an equation in the form

$$Nu = C_1Gr^{n_1}Pr^{m_1} \tag{16}$$

The value of the constants in these equations depend on the flow conditions (turbulent or laminar), boundary conditions and the system geometry. If the system to be modeled can be described by these relationships and the constants are known, then the heat-transfer coefficient can be preserved. Preservation of the heat-transfer coefficient is most easily achieved when both model and prototype have either laminar or turbulent flow since only the exponents (n) are needed to determine the scaling criteria. Otherwise the constants (C) and exponents (m) must also be known.

The mass flux is easily preserved by adjusting the Reynolds number in the model such that

$$Re_m = (L_m/L_p)Re_p \tag{17}$$

Preservation of either mass flux or heat-transfer coefficient results in some loss of thermal similitude. If the mass flux is preserved then the ratio of forced convection heat-transfer coefficients between model and prototype is given, for identical flow regimes, by

$$h_m/h_p = (L_p/L_m)^{1-n} \tag{18}$$

and conversely, if the forced convection heat-transfer coefficient is preserved, then the ratio of mass fluxes between model and prototype is given by

$$(\rho\bar{v})_m/(\rho\bar{v})_p = (L_m/L_p)^{1-n/n} \tag{19}$$

Mass flux preservation tends to preserve the gas temperature change through the system, whereas, heat-transfer coefficient preservation tends to preserve the temperature difference between the wall and gas. The degree of thermal similitude achieved with the mass flux and heat-transfer coefficient preservation scaling techniques may be approximated using the following simplified analysis.

Consider gas flow through an enclosure which is subject to uniform surface heating and coupled to the external environment through a uniform conductance per unit surface area. The thermal balance for the enclosure is given by

$$q_s = K(T - T_e) + h(T - T_g) \tag{20}$$

The average gas temperature may be related to the inlet gas temperature by equating the convective heat transfer rate to the energy transport rate

$$h(T - T_g) = (\rho\bar{v}c/\alpha)(T_g^{out} - T_g^{in}) \tag{21}$$

Setting, $T_g = (T_g^{out} + T_g^{in})/2$ and solving for T_g gives

$$T_g = \frac{T_g^{in} + (\alpha h/2\rho\bar{v}c)T}{1 + (\alpha h/2\rho\bar{v}c)} \tag{22}$$

Using this result in Eq. (20) and solving for T yields

$$T - T_r = \frac{T_g^{in} - T_r}{1 + K(1/h + \alpha/2\rho\bar{v}c)} \tag{23}$$

where the reference temperature T_r is the enclosure temperature when there is no gas flow and is given by

$$T_r = q_s/K + T_e \tag{24}$$

Assuming the free and forced convection heat-transfer coefficients to be additive the heat-transfer coefficient may be written as

$$h = (k_g/L)(CRe^nPr^m + C_1Gr^{n_1}Pr^{m_1}) \tag{25}$$

Using this definition of h and writing $\rho\bar{v}c$ in terms of $RePr$, Eq. (23) becomes

$$T - T_r = \frac{T_g^{in} - T_r}{1 + (KL/k_g)[\alpha/2RePr + 1/(CRe^nPr^m + C_1Gr^{n_1}Pr^{m_1})]} \tag{26}$$

The prototype and scale model temperatures can be written (for $Pr = 1$ and preservation of the free convection heat-transfer coefficient) as

Prototype

$$(T_p - T_r/T_g^{in} - T_r) = \{1 + (KL_p/k_g)[\alpha/2Re_p + 1/(CRe_p^n + C_1Gr_p^{n_1})]\}^{-1} \tag{27}$$

Scale Model (Mass Flux Preservation)

$$\frac{T_m - T_r}{T_g^{in} - T_r} = \left\{1 + (KL_p/k_g)\left[\frac{\alpha}{2Re_p} + \frac{1}{C(L_m/L_p)^{n-1}Re_p^n + C_1Gr_p^{n_1}}\right]\right\}^{-1} \tag{28}$$

Scale Model (Heat-Transfer Coefficient Preservation)

$$\frac{T_m - T_r}{T_g^{in} - T_r} = \left\{1 + (KL_p/k_g)\left[\frac{\alpha}{2(L_m/L_p)^{1-n/n}Re_p} + \frac{1}{CRe_p^n + C_1Gr_p^{n_1}}\right]\right\}^{-1} \tag{29}$$

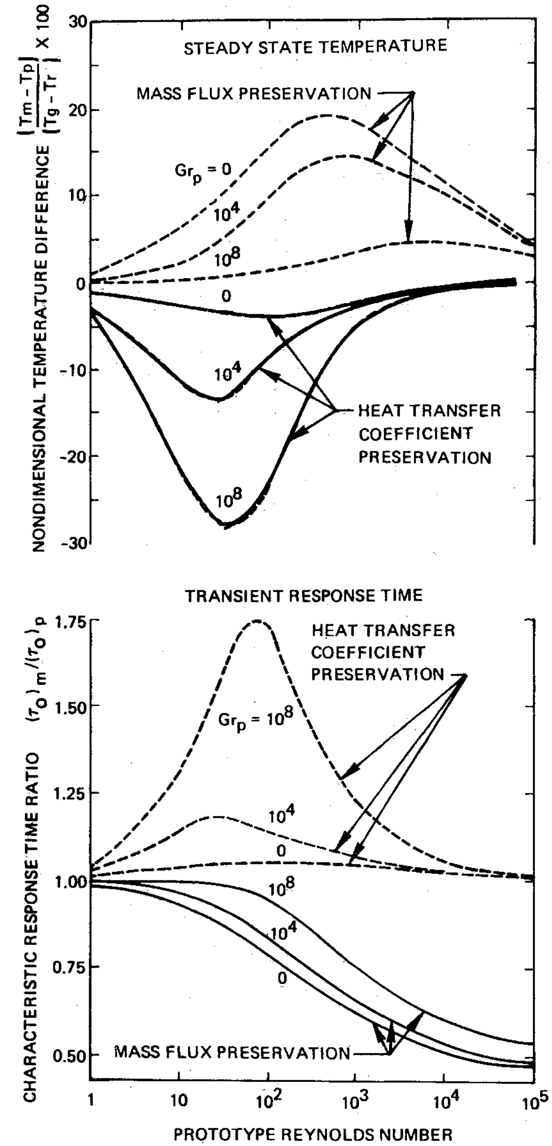

Fig. 2 Effect of scaling compromises in a $\frac{1}{5}$ scale model spacecraft.

Figure 2 shows the temperature differences between $\frac{1}{5}$ scale model and prototype spacecraft as a function of Reynolds and Grashof numbers. These results are based on the simplified analysis applied to laminar flow ($n = 0.5$, $n_1 = 0.25$ and $C_1 = C = 0.33$) through a cylinder with length equal to radius ($\alpha = 4$). The heat leak term (KL_p/k_g) was set equal to 10.

The degree of thermal similitude achieved with either scaling technique is greatest at large Reynolds numbers where the wall temperature approaches the gas temperature and at small Reynolds numbers where the wall temperature approaches the reference temperature. The heat-transfer preservation technique gives the best results for pure forced convection ($Gr = 0$) and mass flux preservation gives the best results when free convection dominates.

These scaling compromises also affect the transient response of the spacecraft cabin wall. The transient response may be calculated by substituting $T_r - (M/K)\,dT/dt$ for T_r in Eq. (26) and solving for T, assuming $T = T_r$ at $t = 0$

$$T - T_r = \left(\frac{T_g^{in} - T_r}{1 + \eta}\right)\left[1 - \exp - \left(\frac{1+\eta}{\eta}\right)\left(\frac{K}{M}\right)t\right] \tag{30}$$

where

$$\eta = (KL/k_g)[\alpha/2RePr + 1/(CRe^nPr^m + C_1Gr^{n_1}Pr^{m_1})] \tag{31}$$

The characteristic response time is then

$$t_0 = M\eta/K(1+\eta) \quad (32)$$

Noting that (K/M) corresponds to $(k/\rho c L^2)$ in the scaling criteria, Eq. (32) may be written in terms of nondimensional time as

$$\tau_0 = \eta/(1+\eta) \quad (33)$$

The characteristic response time ratio between model and prototype for changes in convective heat transfer is then given by

$$(\tau_0)_m/(\tau_0)_p = \eta_m(1+\eta)_p/\eta_p(1+\eta)_m \quad (34)$$

This response time ratio for the $\frac{1}{4}$ scale model spacecraft parameters is shown in Fig. 2 as a function of Reynolds and Grashof numbers for the two scaling techniques. As would be expected the relative model response is slow with heat-transfer coefficient preservation and fast for mass flux preservation. The heat-transfer coefficient preservation technique generally gives better transient simulation except for the lower Reynolds numbers when free convection effects dominate. The calculated results for this simple model indicate that the prototype temperature remains bounded by the scale model temperatures obtained using mass flux and heat-transfer coefficient preservation.

Whether mass flux preservation or heat-transfer coefficient preservation gives better thermal similitude depends on the nature of the system being modeled. Systems with mass flow rates large enough to give a small gas temperature change should be modeled well using the heat-transfer coefficient preservation technique. If the free convection mode of heat transfer dominates then the mass flux preservation technique should give good results.

Nusselt Number Preservation

The Nusselt number preservation scaling technique would use the thermal scale model to experimentally determine the Nusselt number for the system as a function of Reynolds and Grashof numbers. (If the model uses a different fluid, then Prandtl number effects would also have to be determined.) This functional relationship would then be used in conjunction with a thermal math model to predict the prototype performance.

The Nusselt number is preserved if thermal and dynamic similitude exist between the fluid elements of the systems. Thermal and dynamic similitude may be achieved by preserving Re and $(Gr\theta_{fw})$. This keeps the nondimensional velocity v_* and fluid temperature θ_f/θ_{fw} invariant for similar systems [see Eqs. (4) and (5)]. The Reynolds number is easily preserved in the scale model, however preservation of the $(Gr\theta_{fw})$ term depends on the magnitude of the convective heat transfer. In the limit of no convective heat transfer $(\theta_{fw})_m = (\theta_{fw})_p$ and in the limit of only convective heat transfer $(\theta_{fw})_m = (L_m/L_p)(\theta_{fw})_p$. Consequently the model pressure required to preserve the $(Gr\theta_{fw})$ term (i.e., the free convection effects) is within the limits

$$(L_p/L_m)^{3/2} \leq (P_m/P_p) \leq (L_p/L_m)^2 \quad (35)$$

The Nusselt number preservation technique avoids the problems inherent in the other scaling techniques, i.e., high temperatures and heat fluxes required for the modified material preservation, suitable gas for temperature preservation, and the uncertainties of scaling compromises. However thermal scale modeling using Nusselt number preservation requires the use of experimental research techniques whereas using the other scaling techniques requires only fabrication and testing of the scale model.

Experimental Investigation

Since the Nusselt number preservation scaling technique and the compromised scaling techniques of mass flux preservation and heat-transfer coefficient preservation allow a greater range of scale modeling applications, these techniques were chosen for an experimental investigation. The objective of the investigation was to experimentally demonstrate the application of these scaling techniques to thermal scale modeling of systems involving radiation, conduction and convection. Full scale and $\frac{1}{4}$ scale models of a radiation-conduction-convection system were fabricated and tested. The $\frac{1}{4}$ scale model was tested at various conditions that allowed the different scaling techniques to be evaluated. This paper, however, only covers the investigation of the compromised scaling techniques.

Model Configuration

The model configuration chosen for the experimental investigation is shown in Fig. 3. The model basically consists of two concentric cylinders with end plates closing off the annular region between the cylinders. Thermal gradients are generated by heating the center section of the inner cylinder and cooling the center of the outer cylinder. The heating is provided by an electrical heater mounted on the inside of the inner cylinder and the cooling by a cooled fin attached to the center of the outer cylinder. The exposed surfaces in the annular region are painted black to enhance the radiation heat transfer and to provide the same radiation properties for full and $\frac{1}{4}$ scale models. Forced convection is provided by airflow in the annular region. Air is supplied to and exhausted from the model through air distribution plugs in the air inlet and outlet sections of the inner cylinder.

The radiation, conduction and convection modes of heat transfer are important in the thermal response of the models. The relative importance of each mode of heat transfer varies with the test conditions. While the model configuration was not meant to represent a manned spacecraft configuration, the airflow pattern and the gas velocity are similar to those of manned spacecraft. The model symmetry minimizes the instrumentation requirements and allows the axially asymmetric convective heat-transfer effects to become apparent.

The basic structure of the full scale model was made from 6061-T6 aluminum and that for the $\frac{1}{4}$ scale model from 304 stainless steel. The conductive heat-transfer scaling criteria was met to within 10% in the $\frac{1}{4}$ scale model. Table 2 gives the basic model dimensions. The interior model surfaces were painted with two coats of Fullers Metal Etch Primer (3811 black with 3816 primer). Emissivities of painted

Table 2 Model dimensions (in.)

	Full scale 6061-T6 aluminum	$\frac{1}{4}$ Scale 304 stainless steel
Inside length	24	6.0
Outer cylinder		
Inside diameter	24	6.0
Thickness	0.063	0.040
Inner cylinder		
Outside Diameter	8.0	2.0
Heater section		
Length	4.0	1.0
Thickness	0.125	0.085
End sections		
Length	2.0	0.5
Thickness	0.125	0.085
Thin section		
Thickness	0.050	0.032
End plates		
Thickness	0.375	0.250
Horizontal cooling fin		
Outside diameter	30.13	8.0
Thickness	0.071	0.063

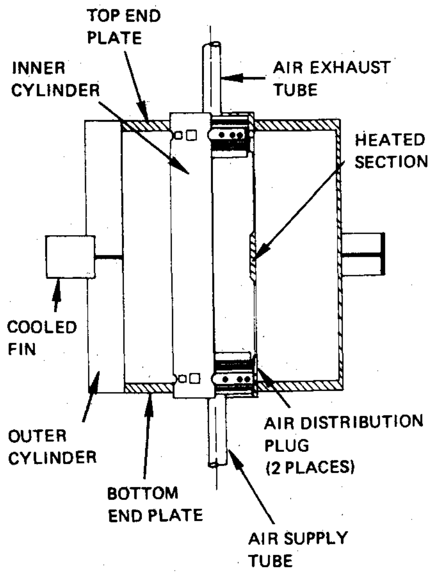

Fig. 3 Model configuration.

samples were measured with a Lions Emissometer before and after a baking cycle of 4 hr at 400°F. The emissivity for the aluminum sample was 0.88 before baking and 0.83 after baking, the corresponding values for the stainless steel sample were 0.85 and 0.88.

Calibrated chromel-constantan thermocouples (AWG #36 wire) were used to instrument the models. These thermocouples were attached to 12 axial locations on the inner and outer cylinders, 4 radial locations on the top and bottom end plates, the center of the cooling fin and the cooling tube inlet and outlet locations. Thermocouples were also used to measure the air inlet and outlet temperatures. The thermocouples are located at the center of the model nodes which were defined for the thermal math model used in the Nusselt number preservation scaling technique. Figure 4 shows the location of these nodes.

Prior to testing the models were insulated with polyurethane foam. The models were centered in cylindrical molds and the insulation was foamed between the models and the molds. The foam on the full scale model was about 4 in. thick and that on the ¼ scale model was sized to preserve the one-dimensional heat transfer through the foam.

Model Tests

The full scale model was tested at atmospheric pressure for both free convection and forced convection conditions. The free convection tests were made at five heating rates (nominal values of 294, 224, 187, 141 and 93 w). The forced convection tests used three heating rates (nominal values of 294, 224, and 187 w) and three airflow rates (nominal values of 4.80, 1.20 and 0.30 lb/min). The flow rate tests were repeated at the highest heating rate with the flow direction through the model reversed (i.e., flow from top to bottom).

The ¼ scale model tests correspond to those for the full scale model. The free convection Nusselt number was assumed to be proportional to the fourth root of the Grashof number (typical for laminar flow free convection). Based on this assumption, the free convection heat transfer is preserved in a ¼ scale model by requiring $P_m = P_p/2$. Consequently the ¼ scale model tests were made at a nominal pressure of ½ atm. The heating rates were set such that $Q_m = Q_p/16$. The flow rates were adjusted to preserve the mass flux and the heat-transfer coefficient. The mass flux was preserved by setting the flow rate such that $w_m = w_p/16$. Assuming the forced convection to be represented by laminar flow over a flat plate (i.e., Nusselt number proportional to the square root of the Reynolds number), the heat-transfer coefficient was preserved by setting the flow rate such that $w_m = w_p/64$. The power input to the heaters, provided by a d.c. power supply, was determined by measuring voltage and current. The fin was cooled by water circulated through an ice bath. The fin temperature was maintained at 33.5±1°F for all tests. The model airflow rates were measured using calibrated "rotometers." The room temperature inlet air was nominally at 68°F.

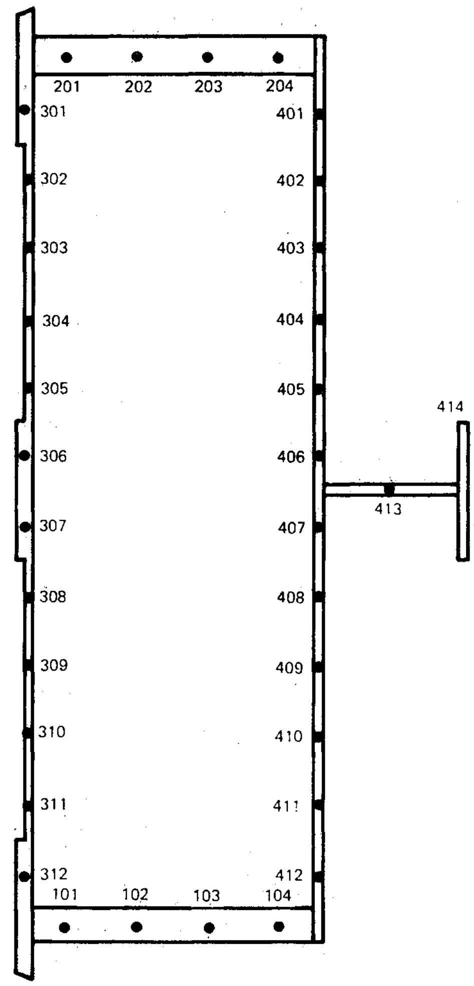

Fig. 4 Model node locations.

Discussion of Test Results

Free Convection Tests

Figure 5 shows the temperature correlation between ¼ scale and full scale models for the free convection tests. Even though the correlation is quite good, the full scale model temperatures are consistently larger than those of the ¼ scale model. The maximum temperature difference increases from about 3.5°F for the lowest heating rate case to about 13°F for the highest heating rate case. These temperature differences are probably caused by slightly different emissivities in ¼ scale and full scale models. The correlation between the models shows that radiation and conduction as well as free convection heat-transfer processes can be preserved in a thermal scale model.

Forced Convection Tests

The correlations between the ¼ scale and full scale models, at the high heating rate, are shown in Fig. 6 for the high, intermediate and low flow rate cases. The ¼ scale model data are shown for both mass flux and heat-transfer coefficient preservation techniques. These correlations show that preservation

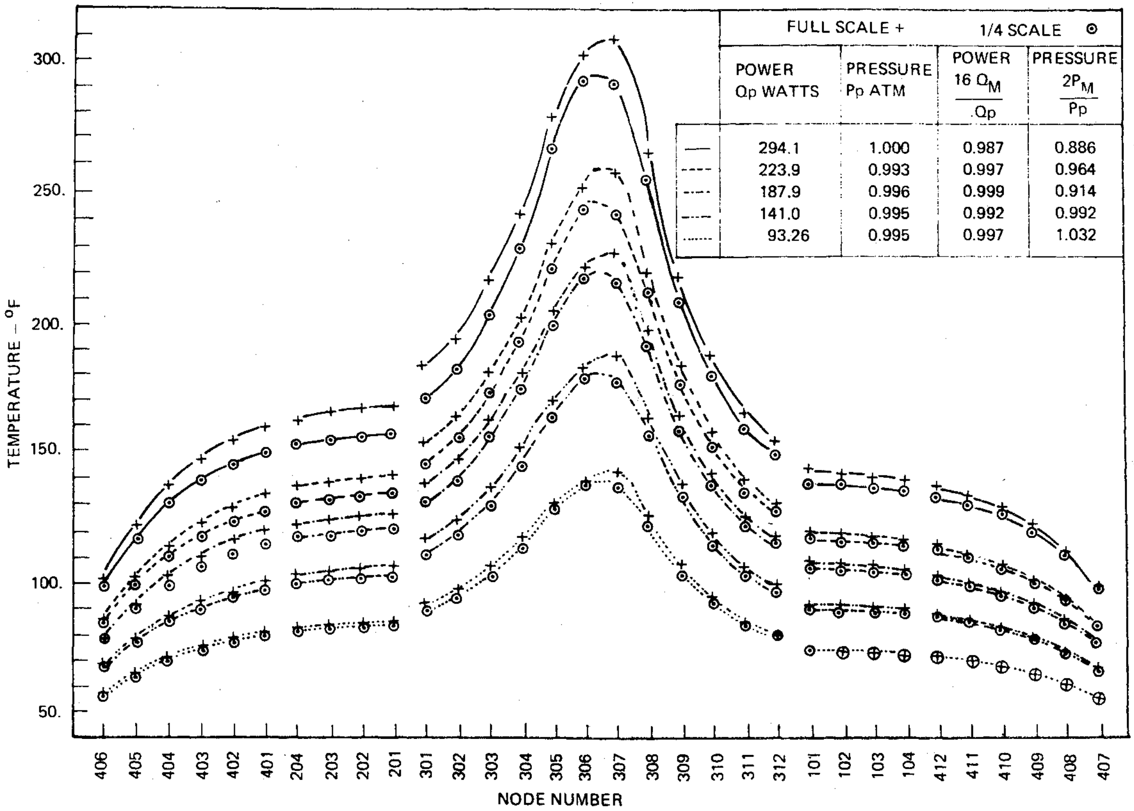

Fig. 5 Free convection correlations.

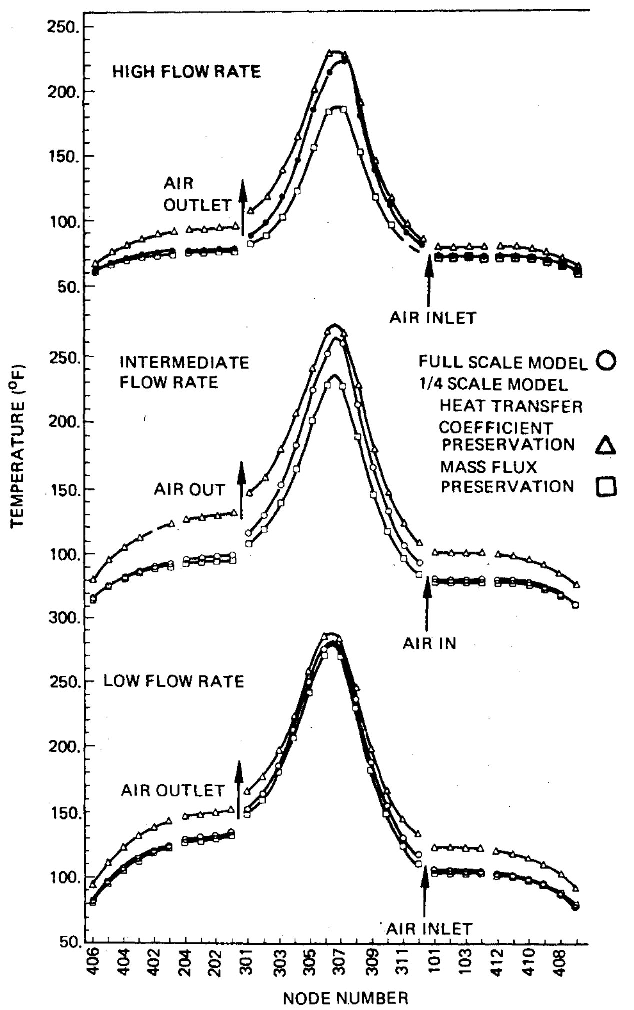

Fig. 6 Forced convection correlations.

of mass flux results in $\frac{1}{4}$ scale model temperatures generally lower than the full scale model while preservation of heat-transfer coefficient results in generally higher model temperatures.

Even though the simplified analysis presented earlier is a poor representation of the experimental model, the over-all effects of the scaling compromises are predicted fairly well by this analysis. Figure 7 shows the comparison between the simplified analysis applied to the model configuration and the average test results. The analysis assumed laminar flow, $(KL_p/k_g) = 50$, $\alpha = 8$ and $Gr_p = 10^8$. These conditions are representative of the model tests. The test temperature differ-

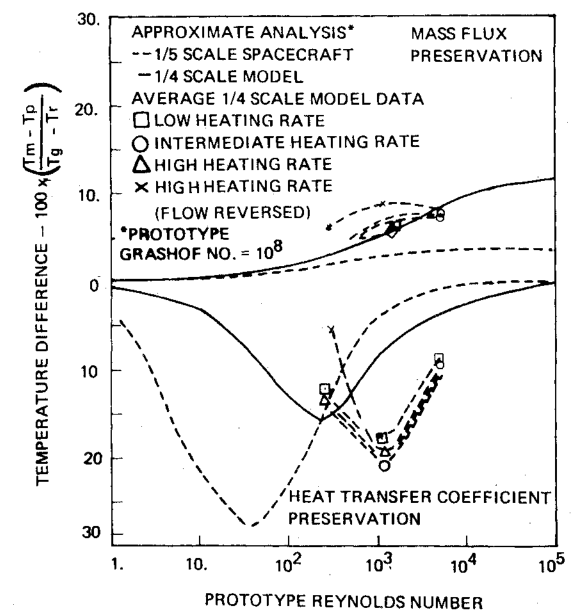

Fig. 7 Temperature differences between model and prototype due to scaling compromises.

ence data were based on the average model temperatures with the reference temperature, for a particular heating rate, taken as the average full scale model temperature under free convection conditions at the same heating rate. Also shown in Fig. 7, for comparison, are the calculated results for the $\frac{1}{5}$ scale model manned spacecraft discussed earlier. Since the simplified analysis is more representative of a manned spacecraft configuration than of the experimental model configuration the calculated results should give a good indication of the temperature differences to be expected in thermal scale modeling a manned spacecraft. The results shown in Fig. 7 indicate that, at a typical manned spacecraft Reynolds number of 10^4, better thermal similitude would be achieved in a $\frac{1}{5}$ scale spacecraft than that achieved in the $\frac{1}{4}$ scale model of the present investigation.

Conclusions

Thermal similitude may be achieved in thermal scale modeling of radiation-conduction-convection systems by using the compromised scaling techniques of either mass flux or heat-transfer coefficient preservation. The best thermal similitude is achieved with mass flux preservation when free convection effects dominate and with heat-transfer coefficient preservation when forced convection effects dominate. The degree of thermal similitude achieved with either technique depends on the system being modeled. This was illustrated by a simplified analysis and verified in the experimental investigation. The heat-transfer coefficient preservation scaling technique should give better thermal similitude than mass flux preservation in thermal scale modeling applications for manned spacecraft. The simplified analysis indicates that very good thermal similitude (both transient and steady state) can be achieved with a $\frac{1}{5}$ scale model spacecraft using heat-transfer coefficient preservation.

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