Designation: C 680 – 03a

Standard Practice for Estimate of the Heat Gain or Loss and the Surface Temperatures of Insulated Flat, Cylindrical, and Spherical Systems by Use of Computer Programs[1]

This standard is issued under the fixed designation C 680; the number immediately following the designation indicates the year of original adoption or, in the case of revision, the year of last revision. A number in parentheses indicates the year of last reapproval. A superscript epsilon (ε) indicates an editorial change since the last revision or reapproval.

1. Scope

1.1 This practice provides the algorithms and calculation methodologies for predicting the heat loss or gain and surface temperatures of certain thermal insulation systems that can attain one dimensional, steady- or quasi-steady-state heat transfer conditions in field operations.

1.2 This practice is based on the assumption that the thermal insulation systems can be well defined in rectangular, cylindrical or spherical coordinate systems and that the insulation systems are composed of homogeneous, uniformly dimensioned materials that reduce heat flow between two different temperature conditions.

1.3 Qualified personnel familiar with insulation-systems design and analysis should resolve the applicability of the methodologies to real systems. The range and quality of the physical and thermal property data of the materials comprising the thermal insulation system limit the calculation accuracy.

1.4 The computer program that can be generated from the algorithms and computational methodologies defined in this practice is described in Section 7 of this practice. The computer program is intended for flat slab, pipe and hollow sphere insulation systems. An executable version of a program based on this standard may be obtained from ASTM.

1.5 The values stated in inch-pound units are to be regarded as the standard. The values given in parentheses are for information only.

1.6 *This standard does not purport to address all of the safety concerns, if any, associated with its use. It is the responsibility of the user of this standard to establish appropriate safety and health practices and determine the applicability of regulatory limitations prior to use.*

2. Referenced Documents

2.1 *ASTM Standards:* [2]

C 168 Terminology Relating to Thermal Insulating Materials

C 177 Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Guarded Hot Plate Apparatus

C 335 Test Method for Steady-State Heat Transfer Properties of Horizontal Pipe Insulation

C 518 Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Heat Flow Meter Apparatus

C 585 Practice for Inner and Outer Diameters of Rigid Thermal Insulation for Nominal Sizes of Pipe and Tubing (NPS System)

C 1055 Guide for Heated System Surface Conditions That Produce Contact Burn Injuries

C 1057 Practice for Determination of Skin Contact Temperature from Heated Surfaces Using a Mathematical Model and Thermesthesiometer

2.2 *Other Document:*

NBS Circular 564 Tables of Thermodynamic and Transport Properties of Air, US Dept of Commerce

3. Terminology

3.1 *Definitions*—For definitions of terms used in this practice, refer to Terminology C 168.

3.1.1 *thermal insulation system*—for this practice, a thermal insulation system is a system comprised of a single layer or layers of homogeneous, uniformly dimensioned material(s) intended for reduction of heat transfer between two different

[1] This practice is under the jurisdiction of ASTM Committee C16 on Thermal Insulation and is the direct responsibility of Subcommittee C16.30 on Thermal Measurements.

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[2] For referenced ASTM standards, visit the ASTM website, www.astm.org, or contact ASTM Customer Service at service@astm.org. For *Annual Book of ASTM Standards* volume information, refer to the standard's Document Summary page on the ASTM website.

temperature conditions. Heat transfer in the system is steady-state. Heat flow for a flat system is normal to the flat surface, and heat flow for cylindrical and spherical systems is radial.

3.2 *Symbols*—The following symbols are used in the development of the equations for this practice. Other symbols will be introduced and defined in the detailed description of the development.

where:

h	=	surface heat transfer conductance, Btu/(h·ft^2·°F) (W/(m^2·K)) h_i at inside surface; h_o at outside surface
k	=	apparent thermal conductivity, Btu·in./(h·ft^2·°F) (W/(m·K))
k_e	=	effective thermal conductivity over a prescribed temperature range, Btu·in./(h·ft^2·°F) (W/(m·K))
q	=	heat flux, Btu/(h·ft^2) (W/m^2)
q_p	=	time rate of heat flow per unit length of pipe, Btu/(h·ft) (W/m)
R	=	thermal resistance, °F·h·ft^2/Btu (K·m^2/W)
r	=	radius, in. (m); $r_{m+1} - r_m$ = thickness
t	=	local temperature, °F (K)
t_i	=	inner surface temperature of the insulation, °F (K)
t_1	=	inner surface temperature of the system
t_o	=	temperature of ambient fluid and surroundings, °F (K)
x	=	distance, in. (m); $x_{m+1} - x_m$ = thickness
ϵ	=	effective surface emittance between outside surface and the ambient surroundings, dimensionless
σ	=	Stefan-Boltzmann constant, 0.1714 × 10^{-8} Btu/(h·ft^2·°R^4) (5.6697 × 10-8 W/(m^2·K^4))
T_s	=	absolute surface temperature, °R (K)
T_o	=	absolute surroundings (ambient air if assumed the same) temperature, °R (K)
T_m	=	$(T_s + T_o)/2$
L	=	characteristic dimension for horizontal and vertical flat surfaces, and vertical cylinders.
D	=	characteristic dimension for horizontal cylinders and spheres
c_p	=	specific heat of ambient fluid, Btu/(lb·°R) (J/(kg·K))
h_c	=	average convection conductance, Btu/(h·ft^2·°F) (W/(m^2·K))
k_f	=	thermal conductivity of ambient fluid, Btu/(h·ft·°F) (W/(m·K))
V	=	free stream velocity of ambient fluid, ft/h (m/s)
υ	=	kinematic viscosity of ambient fluid, ft^2/h (m^2/s)
g	=	acceleration due to gravity, ft/h^2 (m/s^2)
β	=	volumetric thermal expansion coefficient of ambient fluid, °R^{-1}(K^{-1})
ρ	=	density of ambient fluid, lb/ft^3 (kg/m^3)
ΔT	=	absolute value of temperature difference between surface and ambient fluid, °R (K)
Nu	=	Nusselt number, dimensionless
Ra	=	Rayleith number, dimensionless
Re	=	Reynolds number, dimensionless
Pr	=	Prandtl number, dimensionless

4. Summary of Practice

4.1 The procedures used in this practice are based on standard, steady-state, one dimensional, conduction heat transfer theory as outlined in textbooks and handbooks, Refs **(4,5,20,21,22,30)**. Heat flux solutions are derived for temperature dependent thermal conductivity in a material. Algorithms and computational methodologies for predicting heat loss or gain of single or multi-layer thermal insulation systems are provided by this practice for implementation in a computer program. In addition, interested parties can develop computer programs from the computational procedures for specific applications and for one or more of the three coordinate systems considered in Section 6.

4.1.1 The computer program combines functions of data input, analysis and data output into an easy to use, interactive computer program. By making the program interactive, little training for operators is needed to perform accurate calculations.

4.2 The operation of the computer program follows the procedure listed below:

4.2.1 *Data Input*—The computer requests and the operator inputs information that describes the system and operating environment. The data includes:

4.2.1.1 Analysis identification.

4.2.1.2 Date.

4.2.1.3 Ambient temperature.

4.2.1.4 Surface heat transfer conductance or ambient wind speed, system surface emittance and system orientation.

4.2.1.5 *System Description*—Material and thickness for each layer (define sequence from inside out).

4.2.2 *Analysis*—Once input data is entered, the program calculates the surface heat transfer conductances (if not entered directly) and layer thermal resistances. The program then uses this information to calculate the heat transfer and surface temperature. The program continues to repeat the analysis using the previous temperature data to update the estimates of layer thermal resistance until the temperatures at each surface repeat within 0.1°F between the previous and present temperatures at the various surface locations in the system.

4.2.3 *Program Output*—Once convergence of the temperatures is reached, the program prints a table that presents the input data, calculated thermal resistance of the system, heat flux and the inner surface and external surface temperatures.

5. Significance and Use

5.1 Manufacturers of thermal insulation express the performance of their products in charts and tables showing heat gain or loss per unit surface area or unit length of pipe. This data is presented for typical insulation thicknesses, operating temperatures, surface orientations (facing up, down, horizontal, vertical), and in the case of pipes, different pipe sizes. The exterior surface temperature of the insulation is often shown to provide information on personnel protection or surface condensation. However, additional information on effects of wind velocity, jacket emittance, ambient conditions and other influential parameters may also be required to properly select an insulation system. Due to the large number of combinations of size, temperature, humidity, thickness, jacket properties, surface emittance, orientation, and ambient conditions, it is not practical to publish data for each possible case, Refs **(31,32)**.

5.2 Users of thermal insulation faced with the problem of designing large thermal insulation systems encounter substantial engineering cost to obtain the required information. This cost can be substantially reduced by the use of accurate engineering data tables, or available computer analysis tools, or both. The use of this practice by both manufacturers and users of thermal insulation will provide standardized engineering data of sufficient accuracy for predicting thermal insulation system performance. However, it is important to note that the accuracy of results is extremely dependent on the accuracy of the input data. Certain applications may need specific data to produce meaningful results.

5.3 The use of analysis procedures described in this practice can also apply to designed or existing systems. In the rectangular coordinate system, Practice C 680 can be applied to heat flows normal to flat, horizontal or vertical surfaces for all types of enclosures, such as boilers, furnaces, refrigerated chambers and building envelopes. In the cylindrical coordinate system, Practice C 680 can be applied to radial heat flows for all types of piping circuits. In the spherical coordinate system, Practice C 680 can be applied to radial heat flows to or from stored fluids such as liquefied natural gas (LNG).

5.4 Practice C 680 is referenced for use with Guide C 1055 and Practice C 1057 for burn hazard evaluation for heated surfaces. Infrared inspection, in-situ heat flux measurements, or both are often used in conjunction with Practice C 680 to evaluate insulation system performance and durability of operating systems. This type of analysis is often made prior to system upgrades or replacements.

5.5 All porous and non-porous solids of natural or man-made origin have temperature dependent thermal conductivities. The change in thermal conductivity with temperature is different for different materials, and for operation at a relatively small temperature difference, an average thermal conductivity may suffice. Thermal insulating materials ($k < 0.85$ {Btu·in}/{h·ft^2·°F}) are porous solids where the heat transfer modes include conduction in series and parallel flow through the matrix of solid and gaseous portions, radiant heat exchange between the surfaces of the pores or interstices, as well as transmission through non-opaque surfaces, and to a lesser extent, convection within and between the gaseous portions. With the existence of radiation and convection modes of heat transfer, the measured value should be called apparent thermal conductivity as described in Terminology C 168. The main reason for this is that the premise for pure heat conduction is no longer valid, because the other modes of heat transfer obey different laws. Also, phase change of a gas, liquid, or solid within a solid matrix or phase change by other mechanisms will provide abrupt changes in the temperature dependence of thermal conductivity. For example, the condensation of the gaseous portions of thermal insulation in extremely cold conditions will have an extremely influential effect on the apparent thermal conductivity of the insulation. With all of this considered, the use of a single value of thermal conductivity at an arithmetic mean temperature will provide less accurate predictions, especially when bridging temperature regions where strong temperature dependence occurs.

5.6 The calculation of surface temperature and heat loss or gain of an insulated system is mathematically complex, and because of the iterative nature of the method, computers best handle the calculation. Computers are readily available to most producers and consumers of thermal insulation to permit the use of this practice.

5.7 Computer programs are described in this practice as a guide for calculation of the heat loss or gain and surface temperatures of insulation systems. The range of application of these programs and the reliability of the output is a primary function of the range and quality of the input data. The programs are intended for use with an "interactive" terminal. Under this system, intermediate output guides the user to make programming adjustments to the input parameters as necessary. The computer controls the terminal interactively with program-generated instructions and questions, which prompts user response. This facilitates problem solution and increases the probability of successful computer runs.

5.8 The user of this practice may wish to modify the data input and report sections of the computer programs presented in this practice to fit individual needs. Also, additional calculations may be desired to include other data such as system costs or economic thickness. No conflict exists with such modifications as long as the user verifies the modifications using a series of test cases that cover the range for which the new method is to be used. For each test case, the results for heat flow and surface temperature must be identical (within resolution of the method) to those obtained using the practice described herein.

5.9 This practice has been prepared to provide input and output data that conforms to the system of units commonly used by United States industry. Although modification of the input/output routines could provide an SI equivalent of the heat flow results, no such "metric" equivalent is available for some portions of this practice. To date, there is no accepted system of metric dimensions for pipe and insulation systems for cylindrical shapes. The dimensions used in Europe are the SI equivalents of American sizes (based on Practice C 585), and each has a different designation in each country. Therefore, no SI version of the practice has been prepared, because a standard SI equivalent of this practice would be complex. When an international standard for piping and insulation sizing occurs, this practice can be rewritten to meet those needs. In addition, it has been demonstrated that this practice can be used to calculate heat transfer for circumstances other than insulated systems; however, these calculations are beyond the scope of this practice.

6. Method of Calculation

6.1 *Approach*:

6.1.1 The calculation of heat gain or loss and surface temperature requires: (*1*) The thermal insulation is homogeneous as outlined by the definition of thermal conductivity in Terminology C 168; (*2*) the system operating temperature is known; (*3*) the insulation thickness is known; (*4*) the surface heat transfer heat conductances of the system is known, reasonably estimated or estimated from algorithms defined in this practice based on sufficient information; and, (*5*) the thermal conductivity as a function of temperature for each system layer is known in detail.

6.1.2 The solution is a procedure calling for (*1*) estimation of the system temperature distribution; (*2*) calculation of the thermal resistances throughout the system based on that distribution; (*3*) calculation of heat flux; and (*4*) reestimation of the system temperature distribution. The iterative process continues until a calculated distribution is in reasonable agreement with the previous distribution. The layer thermal resistance is calculated each time with the effective thermal conductivity being obtained by integration of the thermal conductivity curve for the layer being considered. This practice uses the temperature dependence of the thermal conductivity of any insulation or multiple layer combination of insulations to calculate heat flow.

6.2 *Development of Equations*—The development of the mathematical equations is for conduction heat transfer through homogeneous solids having temperature dependent thermal conductivities. To proceed with the development, several precepts or guidelines must be cited:

6.2.1 *Steady-state Heat Transfer*—For all the equations it is assumed that the temperature at any point or position in the solid is invariant with time. Thus, heat is transferred solely by temperature difference from point to point in the solid.

6.2.2 *One-dimensional Heat Transfer*—For all equations it is assumed there is heat flow in only one dimension of the particular coordinate system being considered. Heat transfer in the other dimensions of the particular coordinate system is considered to be zero.

6.2.3 *Conduction Heat Transfer*—The premise here is that the heat flux normal to any surface is directly proportional to the temperature gradient in the direction of heat flow, or

$$q = -k \frac{dt}{dp} \tag{1}$$

where the thermal conductivity, k, is the proportionality constant, and p is the space variable through which heat is flowing. For steady-state conditions, one-dimensional heat flow, and temperature dependent thermal conductivity, the equation becomes

$$q = -k(t) \frac{dt}{dp} \tag{2}$$

where at all surfaces normal to the heat flux, the total heat flow through these surfaces is the same and changes in the thermal conductivity must dictate changes in the temperature gradient. This will ensure that the total heat passing through a given surface does not change from that surface to the next.

6.2.4 *Solutions from Temperature Boundary Conditions*—The temperature boundary conditions on a uniformly thick, homogeneous mth layer material are:

$$t = t_m \quad \text{at } x = x_m \quad (r = r_m); \tag{3}$$
$$t = t_{m+1} \quad \text{at } x = x_{m+1} \quad (r = r_m)$$

For heat flow in the flat slab, let $p = x$ and integrate Eq 2:

$$q \int_{x_m}^{x_{m+1}} dx = -\int_{t_m}^{t_{m+1}} k(t)dt \tag{4}$$

$$q = k_{e,m} \frac{t_m - t_{m+1}}{x_{m+1} - x_m}$$

For heat flow in the hollow cylinder, let $p = r$, $q = Q/(2\pi rl)$ and integrate Eq 2:

$$\frac{Q}{2\pi l} \int_{r_m}^{r_{m+1}} \frac{dr}{r} = -\int_{t_m}^{t_{m+1}} k(t)dt \tag{5}$$

$$Q = k_{e,m} \frac{t_m - t_{m+1}}{\ln(r_{m+1} / r_m)} 2\pi l$$

Divide both sides by $2\pi rl$

$$q = k_{e,m} \frac{t_m - t_{m+1}}{r \ln(r_{m+1} / r_m)}$$

For radial heat flow in the hollow sphere, let $p = r$, $q = Q/(4\pi r^2)$ and integrate Eq 2:

$$\frac{Q}{4\pi} \int_{r_m}^{r_{m+1}} \frac{dr}{r^2} = \int_{t_m}^{t_{m+1}} k(t)dt \tag{6}$$

$$Q = k_{e,m} \frac{t_m - t_{m+1}}{\frac{1}{r_m} - \frac{1}{r_{m+1}}} 4\pi$$

Divide both sides by $4\pi r^2$ and multiply both sides by $r_m r_{m+1}/r_m r_{m+1}$

$$q = k_{e,m} \frac{r_m r_{m+1}}{r^2} \frac{t_m - t_{m+1}}{r_{m+1} - r_m}$$

Note that the effective thermal conductivity over the temperature range is:

$$k_{e,m} = \frac{\int_{t_m}^{t_{m+1}} k(t)dt}{t_{m+1} - t_m} \tag{7}$$

6.3 *Case 1, Flat Slab Systems*:

6.3.1 From Eq 4, the temperature difference across the *m*th layer material is:

$$t_m - t_{m+1} = qR_m \tag{8}$$

$$\text{where} \quad R_m = \frac{(x_{m+1} - x_m)}{k_{e,m}}$$

Note that R_m is defined as the thermal resistance of the *m*th layer of material. Also, for a thermal insulation system of n layers, $m = 1,2...n$, it is assumed that perfect contact exists between layers. This is essential so that continuity of temperature between layers can be assumed.

6.3.2 Heat is transferred between the inside and outside surfaces of the system and ambient fluids and surrounding surfaces by the relationships:

$$q = h_i(t_i - t_1) \tag{9}$$
$$q = h_o(t_{n+1} - t_o)$$

where h_i and h_o are the inside and outside surface heat transfer heat conductances. Methods for estimating these conductances are found in 6.7. Eq 9 can be rewritten as:

$$t_i - t_1 = qR_i \tag{10}$$
$$t_{n+1} - t_o = qR_o$$

$$\text{where} \quad R_i = \frac{1}{h_i}, \quad R_o = \frac{1}{h_o}$$

For the computer programs, the inside surface heat transfer heat conductance, h_i, can be assumed to be very large such that $R_i = 0$, and $t_1 = t_i$ is the given surface temperature.

6.3.3 Adding Eq 8 and Eq 10 yields the following equation:

$$t_i - t_o = q(R_1 + R_2 + ... + R_n + R_i + R_o) \tag{11}$$

From the previous equation a value for q can be calculated from estimated values of the resistances, R. Then, by rewriting Eq 8 to the following:

$$t_{m+1} = t_m - qR_m \tag{12}$$
$$t_1 = t_i - qR_i, \quad \text{for } R_i > 0$$

The temperature at the interface(s) and the outside surface can be calculated starting with $m = 1$. Next, from the calculated temperatures, values of $k_{e,m}$ (Eq 7) and R_m (Eq 8) can be calculated as well as R_o and R_i. Then, by substituting the calculated R-values back into Eq 11, a new value for q can be calculated. Finally, desired (correct) values can be obtained by repeating this calculation methodology until all values agree with previous values.

6.4 *Case 2, Cylindrical (Pipe) Systems*:

6.4.1 From Eq 5, the heat flux through any layer of material is referenced to the outer radius by the relationship:

$$q_n = q_m \frac{r}{r_{n+1}} = k_{e,m} \frac{t_m - t_{m+1}}{r_{n+1} \ln(r_{m+1} / r_m)} \tag{13}$$

and, the temperature difference can be defined by Eq 8, where:

$$R_m = \frac{r_{n+1} \ln(r_{m+1} / r_m)}{k_{e,m}} \tag{14}$$

Utilizing the methodology presented in case 1 (6.3), the heat flux, q_n, and the surface temperature, t_{n+1}, can be found by successive iterations. However, one should note that the definition of R_m found in Eq 14 must be substituted for the one presented in Eq 8.

6.4.2 For radial heat transfer in pipes, it is customary to define the heat flux in terms of the pipe length:

$$q_p = 2\pi r_{n+1} q_n \tag{15}$$

where q_p is the time rate of heat flow per unit length of pipe. If one chooses not to do this, then heat flux based on the interior radius must be reported to avoid the influence of outer-diameter differences.

6.5 *Case 3, Spherical Systems*:

6.5.1 From Eq 6, the flux through any layer of material is referenced to the outer radius by the relationship:

$$q_n = q_m \frac{r^2}{r_{n+1}^2} = k_{e,m} \frac{r_m r_{m+1} (t_m - t_{m+1})}{r_{n+1}^2 (r_{m+1} - r_m)} \tag{16}$$

The temperature difference can be defined by Eq 8, where:

$$R_m = \frac{r_{n+1}^2 (r_{m+1} - r_m)}{k_{e,m} r_m r_{m+1}} \tag{17}$$

Again, utilizing the methodology presented in case 1 (6.3), the heat flux, q_n, and the surface temperature, t_{n+1}, can be found by successive iterations. However, one should note that the definition of R_m found in Eq 17 must be substituted for the one presented in Eq 8.

6.6 *Calculation of Effective Thermal Conductivity*:

6.6.1 In the calculational methodologies of 6.3, 6.4, and 6.5, it is necessary to evaluate $k_{e,m}$ as a function of the two surface temperatures of each layer comprising the thermal insulating system. This is accomplished by use of Eq 7 where *k(t)* is defined as a polynomial function or a piecewise continuous function comprised of individual, integrable functions over specific temperature ranges. It is important to note that temperature can either be in °F (°C) or absolute temperature, because the thermal conductivity versus temperature relationship is regression dependent. It is assumed for the programs in this practice that the user regresses the k versus t functions using °F.

6.6.1.1 When *k(t)* is defined as a polynomial function, such as $k(t) = a + bt + ct^2 + dt^3$, the expression for the effective thermal conductivity is:

$$k_{e,m} = \frac{\int_{t_m}^{t_{m+1}} (a + bt + ct^2 + dt^3)dt}{(t_{m+1} - t_m)} \tag{18}$$

$$k_{e,m} = \frac{a(t_{m+1} - t_m) + \frac{b}{2}(t_{m+1}^2 - t_m^2) + \frac{c}{3}(t_{m+1}^3 - t_m^3) + \frac{d}{4}(t_{m+1}^4 - t_m^4)}{(t_{m+1} - t_m)}$$

$$k_{e,m} = a + \frac{b}{2}(t_m + t_{m+1}) + \frac{c}{3}(t_m^2 + t_m t_{m+1} + t_{m+1}^2) + \frac{d}{4}(t_m^3 + t_m^2 t_{m+1} + t_m t_{m+1}^2 + t_{m+1}^3)$$

It should be noted here that for the linear case, $c = d = 0$, and for the quadratic case, $d = 0$.

6.6.1.2 When *k(t)* is defined as an exponential function, such as $k(t) = e^{a+bt}$, the expression for the effective thermal conductivity is:

$$k_{e,m} = \frac{\int_{t_m}^{t_{m+1}} e^{a+bt} dt}{(t_{m+1} - t_m)} \tag{19}$$

$$k_{e,m} = \frac{\frac{1}{b}(e^{a+bt_{m+1}} - e^{a+bt_m})}{(t_{m+1} - t_m)}$$

$$k_{e,m} = \frac{(e^{a+bt_{m+1}} - e^{a+bt_m})}{b(t_{m+1} - t_m)}$$

6.6.1.3 The piece-wise continuous function may be defined as:

$$\begin{aligned} k(t) &= k_1(t) && t_{bl} \le t \le t_l \\ &= k_2(t) && t_l \le t \le t_u \qquad t_{bl} \le t_m \text{ and } t_{m+1} \le t_{bu} \\ &= k_3(t) && t_u \le t \le t_{bu} \end{aligned} \tag{20}$$

where t_{bl} and t_{bu} are the experimental lower and upper boundaries for the function. Also, each function is integrable, and $k_1(t_l) = k_2(t_l)$ and $k_2(t_u) = k_3(t_u)$. In terms of the effective thermal conductivity, some items must be considered before performing the integration in Eq 7. First, it is necessary to determine if t_{m+1} is greater than or equal to t_m. Next, it is necessary to determine which temperature range t_m and t_{m+1} fit into. Once these two parameters are decided, the effective thermal conductivity can be determined using simple calculus. For example, if $t_{bl} \le t_m \le t_l$ and $t_u \le t_{m+1} \le t_{bu}$ then the effective thermal conductivity would be:

$$k_{e,m} = \frac{\int_{t_m}^{T_l} k_1(t)dt + \int_{T_l}^{T_u} k_2(t) + \int_{T_u}^{t_{m+1}} k_3(t)}{(t_{m+1} - t_m)} \tag{21}$$

It should be noted that other piece-wise functions exist, but for brevity, the previous is the only function presented.

6.6.2 It should also be noted that when the relationship of k with t is more complex and does not lend itself to simple mathematical treatment, a numerical method might be used. It is in these cases that the power of the computer is particularly useful. There are a wide variety of numerical techniques available. The most suitable will depend of the particular situation, and the details of the factors affecting the choice are beyond the scope of this practice.

6.7 *Surface Heat Transfer Heat Conductances*:

6.7.1 The surface heat transfer heat conductance, h, as defined in Terminology C 168, assumes that the principal surface is at a uniform temperature and that the ambient fluid and other visible surfaces are at a different uniform temperature. The conductance includes the combined effects of radiant, convective, and conductive heat transfer. The conductance is defined by:

$$h = h_r + h_c \tag{22}$$

where h_r is the component due to radiation and h_c is the component due to convection and conduction. In subsequent sections, algorithms for these components will be presented.

6.7.1.1 The algorithms presented in this practice for calculating surface heat transfer heat conductances are used in the computer program; however, surface heat transfer heat conductances may be estimated from published values or separately calculated from algorithms other than the ones presented in this practice. One special note, care must be exercised at low or high surface temperatures to ensure reasonable values.

6.7.2 *Radiant Heat Transfer Conductance*—The radiation conductance is simply based on radiant heat transfer and is calculated from the Stefan-Boltzmann Law divided by the average difference between the surface temperature and the air temperature. In other words:

$$h_r = \frac{\sigma\epsilon\,(T_s^4 - T_o^4)}{T_s - T_o} \quad \text{or} \tag{23}$$

$$h_r = \sigma\epsilon \cdot (T_s^3 + T_s^2 T_o + T_s T_o^2 + T_o^3) \quad \text{or}$$

$$h_r = \sigma\epsilon \cdot 4T_m^3 \left[1 + \left(\frac{T_s - T_o}{T_s + T_o}\right)^2\right]$$

where:

ϵ = effective surface emittance between outside surface and the ambient surroundings, dimensionless,
σ = Stefan-Boltzman constant, 0.1714×10^{-8} Btu/(h·ft^2·°R^4) (5.6697×10^{-8} W/(m^2·K^4)),
T_s = absolute surface temperature, °R (K),
T_o = absolute surroundings (ambient air if assumed the same) temperature, °R (K), and
T_m = $(T_s + T_o)/2$

6.7.3 *Convective Heat Transfer Conductance*—Certain conditions need to be identified for proper calculation of this component. The conditions are: (*a*) Surface geometry—plane, cylinder or sphere; (*b*) Surface orientation—from vertical to horizontal including flow dependency; (*c*) Nature of heat transfer in fluid—from free (natural) convection to forced convection with variation in the direction and magnitude of fluid flow; (*d*) Condition of the surface—from smooth to various degrees of roughness (primarily a concern for forced convection).

6.7.3.1 Modern correlation of the surface heat transfer conductances are presented in terms of dimensionless groups, which are defined for fluids in contact with solid surfaces. These groups are:

$$\text{Nusselt,}\quad \overline{Nu_L} = \frac{\overline{h_c}L}{k_f} \quad \text{or} \quad \overline{Nu_D} = \frac{\overline{h_c}D}{k_f} \tag{24}$$

$$\text{Rayleigh,}\quad Ra_L = \frac{g \cdot \beta \cdot \rho \cdot c_p(\Delta T)L^3}{\nu \cdot k_f} \quad \text{or} \quad Ra_D = \frac{g \cdot \beta \cdot \rho \cdot c_p(\Delta T)D^3}{\nu \cdot k_f} \tag{25}$$

$$\text{Reynolds,}\quad Re_L = \frac{VL}{\nu} \quad \text{or} \quad Re_D = \frac{VD}{\nu} \tag{26}$$

$$\text{Prandtl,}\quad Pr = \frac{\nu \cdot \rho \cdot c_p}{k_f} \tag{27}$$

where:

L = characteristic dimension for horizontal and vertical flat surfaces, and vertical cylinders feet (m), in general, denotes height of vertical surface or length of horizontal surface,
D = characteristic dimension for horizontal cylinders and spheres feet (m), in general, denotes the diameter,
c_p = specific heat of ambient fluid, Btu/(lb·°R) (J/(kg·K)),
$\overline{h_c}$ = average convection conductance, Btu/(h·ft^2·°F) (W/(m^2·K)),
k_f = thermal conductivity of ambient fluid, Btu/(h·ft·°F) (W/(m·K)),
V = free stream velocity of ambient fluid, ft/h (m/s),
ν = kinematic viscosity of ambient fluid, ft^2/h (m^2/s),
g = acceleration due to gravity, ft/h^2 (m/s^2),
β = volumetric thermal expansion coefficient of ambient fluid, °R^{-1}(K^{-1}),
ρ = density of ambient fluid, lb/ft^3 (kg/m^3), and
ΔT = absolute value of temperature difference between surface and ambient fluid, °R (K).

It needs to be noted here that (except for spheres–forced convection) the above fluid properties must be calculated at the film temperature, T_f, which is the average of surface and ambient fluid temperatures. For this practice, it is assumed that the ambient fluid is dry air at atmospheric pressure. The properties of air can be found in references such as Ref **(23)**. This reference contains equations for some of the properties and polynomial fits for others, and the equations are summarized in Table A1.1.

6.7.3.2 When a heated surface is exposed to flowing fluid, the convective heat transfer will be a combination of forced and free convection. For this mixed convection condition, Churchill **(26)** recommends the following equation. For each geometric shape and surface orientation the overall average Nusselt number is to be computed from the average Nusselt number for forced convection and the average Nusselt number for natural convection. The film conductance, h, is then computed from Eq 24. The relationship is:

$$(\overline{Nu} - \delta)^j = (\overline{Nu_f} - \delta)^j + (\overline{Nu_n} - \delta)^j \tag{28}$$

where the exponent, j, and the constant, δ, are defined based on the geometry and orientation.

6.7.3.3 Once the Nusselt number has been calculated, the surface heat transfer conductance is calculated from a rearrangement of Eq 24:

$$h_c = \overline{Nu}_L \cdot k_f / L \tag{29}$$

$$\overline{h}_c = \overline{Nu}_D \cdot k_f / D$$

where L and D are the characteristic dimension of the system. The term k_f is the thermal conductivity of air, k_a, determined at the film temperature using the equation in Table A1.1.

6.7.4 *Convection Conductances for Flat Surfaces*:

6.7.4.1 From *Heat Transfer* by Churchill and Ozoe as cited in *Fundamentals of Heat and Mass Transfer* by Incropera and Dewitt, the relation for forced convection by laminar flow over an isothermal flat surface is:

$$\overline{Nu}_{f,L} = \frac{0.6774\, Re_L^{1/2} Pr^{1/3}}{[1 + (0.0468 / Pr)^{2/3}]^{1/4}} \qquad Re_L < 5 \times 10^5 \tag{30}$$

For forced convection by turbulent flow over an isothermal flat surface, Incropera and Dewitt suggest the following:

$$\overline{Nu}_{f,L} = (0.037\, Re_L^{4/5} - 871)\, Pr^{1/3} \qquad 5 \times 10^5 < Re_L < 10^8 \tag{31}$$

It should be noted that the upper bound for Re_L is an approximate value, and the user of the above equation must be aware of this.

6.7.4.2 In "Correlating Equations for Laminar and Turbulent Free Convection from a Vertical Plate" by Churchill and Chu, as cited by Incropera and Dewitt, it is suggested for natural convection on isothermal, vertical flat surfaces that:

$$\overline{Nu}_{n,L} = \left\{0.825 + \frac{0.387\, Ra_L^{1/6}}{[1 + (0.492 / Pr)^{9/16}]^{8/27}}\right\}^2 \qquad \text{All } Ra_L \tag{32}$$

For slightly better accuracy in the laminar range, it is suggested by the same source (p. 493) that:

$$\overline{Nu}_{n,L} = 0.68 + \frac{0.670\, Ra_L^{1/4}}{[1 + (0.492 / Pr)^{9/16}]^{4/9}} \qquad Ra_L < 10^9 \tag{33}$$

In the case of vertical surfaces the characteristic dimension is the vertical height. To compute the overall Nusselt number (Eq 28), set $j = 3$ and $\delta = 0$. Also, it is important to note that the free convection correlations apply to vertical cylinders in most cases.

6.7.4.3 For natural convection on horizontal flat surfaces, Incropera and Dewitt (p. 498) cite *Heat Transmission* by McAdams, "Natural Convection Mass Transfer Adjacent to Horizontal Plates" by Goldstein, Sparrow and Jones, and "Natural Convection Adjacent to Horizontal Surfaces of Various Platforms" for the following correlations:

$$\text{Heat flow up:} \qquad \overline{Nu}_{n,L} = 0.54\, Ra_L^{1/4} \qquad 10^4 < Ra_L < 10^7 \tag{34}$$

$$\overline{Nu}_{n,L} = 0.15\, Ra_L^{1/3} \qquad 10^7 < Ra_L < 10^{11}$$

$$\text{Heat flow down:} \qquad \overline{Nu}_{n,L} = 0.27\, Ra_L^{1/4} \qquad 10^5 < Ra_L < 10^{10}$$

In the case of horizontal flat surfaces, the characteristic dimension is the area of the surface divided by the perimeter of the surface. To compute the overall Nusselt number (Eq 28), set $j = 3.5$ and $\delta = 0$.

6.7.5 *Convection Conductances for Horizontal Cylinders*:

6.7.5.1 For forced convection with fluid flow normal to a circular cylinder, Incropera and Dewitt (p. 370) cite *Heat Transfer* by Churchill and Bernstein for the following correlation:

$$\overline{Nu}_{f,D} = 0.3 + \frac{0.62 Re_D^{1/2}\, Pr^{1/3}}{[1 + (0.4 / Pr)^{2/3}]^{1/4}} \left[1 + \left(\frac{Re_D}{282\,000}\right)^{5/8}\right]^{4/5} \tag{35}$$

$$\text{All } Re_D \cdot Pr > 0.2$$

In addition, this correlation should be used for forced convection from vertical pipes.

6.7.5.2 For natural convection on horizontal cylinders, Incropera and Dewitt (p. 502) cite "Correlating Equations for Laminar and Turbulent Free Convection from a Horizontal Cylinder" by Churchill and Chu for the following correlation:

$$\overline{Nu}_{n,D} = \left\{0.60 + \frac{0.387 Ra_D^{1/6}}{[1 + (0.559 / Pr)^{9/16}]^{8/27}}\right\}^2 \qquad Ra_D < 10^{12} \tag{36}$$

To compute the overall Nusselt number using Eq 28, set $j = 4$ and $\delta = 0.3$.

6.7.6 *Convection Conductances for Spheres*:

6.7.6.1 For forced convection on spheres, Incropera and DeWitt cite S. Whitaker in *AIChE J*. for the following correlation:

$$\overline{Nu}_{f,D} = 2 + (0.4\, Re_D^{1/2} + 0.06\, Re_D^{2/3})\, Pr^{0.4} \left(\frac{\mu}{\mu_s}\right)^{1/4} \tag{37}$$

$$0.71 < Pr < 380$$

$$3.5 < Re_D < 7.6 \times 10^4$$

$$1.0 < (\mu/\mu_s) < 3.2$$

where μ and μ_s are the free stream and surface viscosities of the ambient fluid respectively. It is extremely important to note that all properties need to be evaluated based on the free stream temperature of the ambient fluid, except for μ_s, which needs to be evaluated based on the surface temperature.

6.7.6.2 For natural convection on spheres, Incropera and DeWitt cite "Free Convection Around Immersed Bodies" by S. W. Churchill in *Heat Exchange Design Handbook* (Schlunder) for the following correlation:

$$\overline{Nu}_{n,D} = 2 + \frac{0.589\, Ra_D^{1/4}}{[1 + (0.469 / Pr)^{9/16}]^{4/9}} \tag{38}$$

$$0.7 \le Pr$$

$$Ra_D < 10^{11}$$

where all properties are evaluated at the film temperature. To compute the overall Nusselt number for spheres (Eq 28) set $j = 4$ and $\delta = 2$.

7. Computer Program

7.1 *General*:

7.1.1 The computer program(s) are written in Microsoft Visual Basic and will be available as an adjunct from ASTM International.[3]

[3] This adjunct is under development and will be published upon approval.

7.1.2 The program consists of a main program that utilizes several subroutines. Other subroutines may be added to make the program more applicable to the specific problems of individual users.

7.2 *Functional Description of Program*—The flow chart shown in Fig. 1 is a schematic representations of the operational procedures for each coordinate system covered by the program. The flow chart presents the logic path for entering data, calculating and recalculating system thermal resistances and temperatures, relaxing the successive errors in the temperature to within 0.1° of the temperature, calculating heat loss or gain for the system and printing the parameters and solution in tabular form.

7.3 *Computer Program Variable Descriptions*—Descriptions of all variables used in the programs are given in the listing of the program as comments.

7.4 *Program Operation*:

7.4.1 Log on procedures and any executive program for execution of this program must be followed as needed.

7.4.2 The input for the thermal conductivity versus mean temperature parameters must be obtained as outlined in 6.6. The type code determines the thermal conductivity versus temperature relationship applying to the insulation. The same type code may be used for more than one insulation. As presented, the programs will operate on three functional relationships:

Type	Functional Relationship
Quadratic	$k = a + bt + ct^2$ where a, b, and c are constants
Linear	$k = a_1 + b_1t$; $t < t_L$ $k = a_2 + b_2t$; $t_L < t < t_U$ $k = a_3 + b_3t$; $t > t_U$ where $a1$, $a2$, $a3$, $b1$, $b2$, $b3$ are constants, and t_L and t_U are, respectively, the lower and upper inflection points of an S-shaped curve

Additional or different relationships may be used, but the main program must be modified.

8. Report

8.1 The results of calculations performed in accordance with this practice may be used as design data for specific job conditions, or may be used in general form to represent the performance of a particular product or system. When the results will be used for comparison of performance of similar products, it is recommended that reference be made to the specific constants used in the calculations. These references should include:

8.1.1 Name and other identification of products or components,

8.1.2 Identification of the nominal pipe size or surface insulated, and its geometric orientation,

8.1.3 The surface temperature of the pipe or surface,

8.1.4 The equations and constants selected for the thermal conductivity versus mean temperature relationship,

8.1.5 The ambient temperature and humidity, if applicable,

8.1.6 The surface heat transfer heat conductance and condition of surface heat transfer,

8.1.6.1 If obtained from published information, the source and limitations,

8.1.6.2 If calculated or measured, the method and significant parameters such as emittance, fluid velocity, etc.,

8.1.7 The resulting outer surface temperature, and

8.1.8 The resulting heat loss or gain.

8.2 Either tabular or graphical representation of the calculated results may be used. No recommendation is made for the format in which results are presented.

9. Accuracy and Resolution

9.1 In many typical computers normally used, seven significant digits are resident in the computer for calculations. Adjustments to this level can be made through the use of "Double Precision;" however, for the intended purpose of this practice, standard levels of precision are adequate. The formatting of the output results, however, should be structured to provide a resolution of 0.1 % for the typical expected levels of heat flux and a resolution of 1°F (0.55°C) for surface temperatures.

NOTE 1—The term "double precision" should not be confused with ASTM terminology on Precision and Bias.

9.2 Many factors influence the accuracy of a calculative procedure used for predicting heat flux results. These factors include accuracy of input data and the applicability of the assumptions used in the method for the system under study. The system of mathematical equations used in this analysis has been accepted as applicable for most systems normally insulated with bulk type insulations. Applicability of this practice to systems having irregular shapes, discontinuities and other variations from the one-dimensional heat transfer assumptions should be handled on an individual basis by professional engineers familiar with those systems.

9.3 The computer resolution effect on accuracy is only significant if the level of precision is less than that discussed in 9.1. Computers in use today are accurate in that they will reproduce the calculated results to resolution required if identical input data is used.

9.4 The most significant factor influencing the accuracy of claims is the accuracy of the input thermal conductivity data. The accuracy of applicability of these data is derived from two factors. The first is the accuracy of the test method used to generate the data. Since the test methods used to supply these data are typically Test Methods C 177, C 335, or C 518, the reports should contain some statement of the estimates of error or estimates of uncertainty. The remaining factors influencing the accuracy are the inherent variability of the product and the variability of the installation practices. If the product variability is large, the installation is poor, or both, serious differences might exist between measured performance and predicted performance from this practice.

10. Precision and Bias

10.1 When concern exists with the accuracy of the input test data, the recommended practice to evaluate the impact of possible errors is to repeat the calculation for the range of the uncertainty of the variable. This process yields a range in the desired output variable for a given uncertainty in the input variable. Repeating this procedure for all the input variables would yield a measure of the contribution of each to the overall

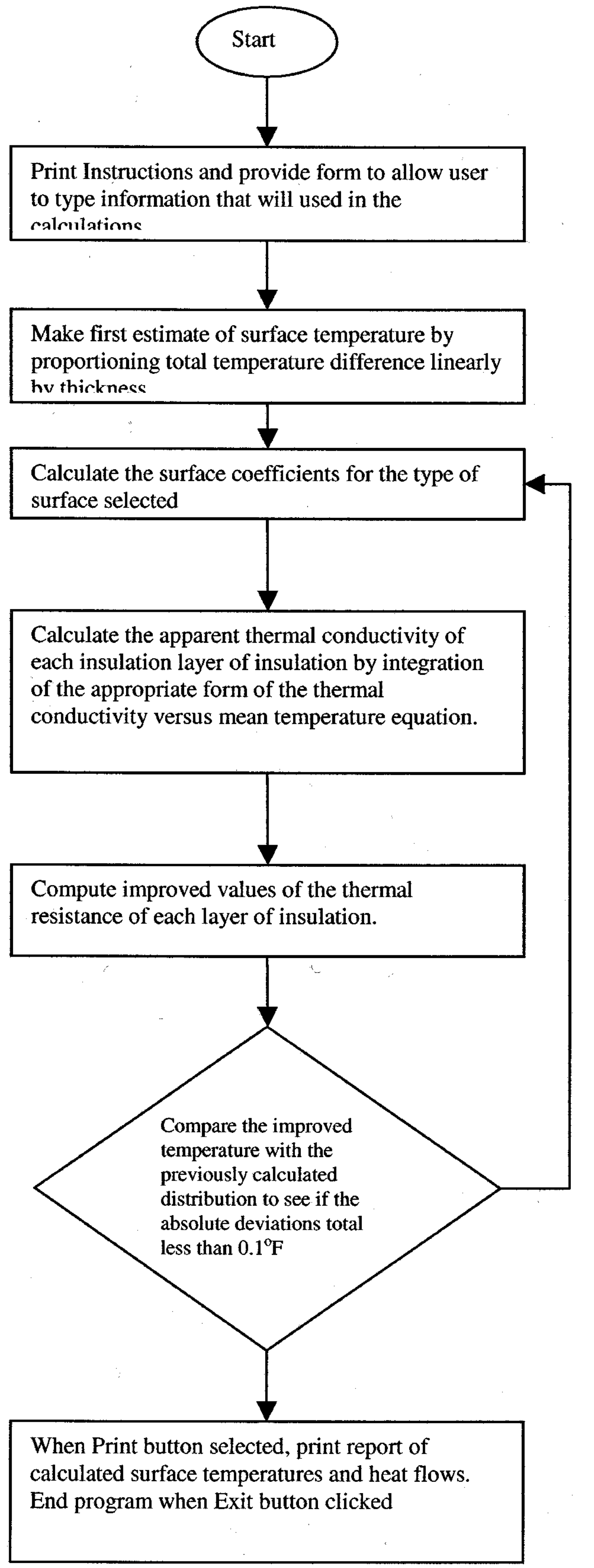

FIG. 1 Flow Chart

uncertainty. Several methods exist for the combination of these effects; however, the most commonly used is to take the square root of the sum of the squares of the percentage errors induced by each variable's uncertainty. Eq 39 from *Theories of Engineering Experimentation* by H. Schenck gives the expression in mathematical form:

$$\frac{S}{R} = \left(\sum_{i=1}^{n} \left(\left(\frac{\partial R}{\partial x_i} \right) \Delta x_i \right)^2 \right)^{1/2} \tag{39}$$

where:

S = estimate of the probable error of the procedure,
R = result of the procedure,
x_i = ith variable in procedure,
$\partial R/\partial x_i$ = change in result with respect to change in ith variable,
Δx_i = uncertainty in value of variable, i, and
n = total number of variables in procedure.

10.2 ASTM Subcommittee C16.30, Task Group 5.2, which is responsible for preparing this practice, has prepared Appendix X1. The appendix provides a more complete discussion of the precision and bias expected when using Practice C 680 in the analysis of operating systems. While much of that discussion is relevant to this practice, the errors associated with its application to operating systems are beyond the primary Practice C 680 scope. Portions of this discussion, however, were used in developing the Precision and Bias statements included in Section 10.

11. Keywords

11.1 computer program; heat flow; heat gain; heat loss; pipe; thermal insulation

ANNEX

(Mandatory Information)

A1. EQUATIONS DERIVED FROM THE NIST CIRCULAR

A1.1 Table A1.1 lists the equations derived from the NBS Circular for the determination of the properties of air as required by the practice.

A1.2 T_k is temperature in degrees Kelvin, T_f is temperature in degrees Farenheit.

TABLE A1.1 Equations and Polynomial Fits for the Properties of Air Between −100ºF and 1300ºF (NBS Circular 564, Department of Commerce [1960])

Property	Equation	Units
Thermal Conductivity, k_a	$\frac{6.325 \times 10^{-4} \cdot \sqrt{T_k}}{[1 + (245.4 \cdot 10^{=12/T_k}) / T_k]} \cdot 241.77$	Btu/(hr·ft·°F)
Dynamic Viscosity, μ	$\mu = \frac{145.8 \cdot T_k \cdot \sqrt{T_k}}{T_k + 110.4} \cdot 241.9 \times 10^{-7}$	lb/(h·ft)
Prandtl Number, *Pr*	$Pr = 0.7189 - T_f \cdot [1.6349 \times 10^{-4} - T_f \cdot (1.8106 \times 10^{-7} - 5.6617 \times 10^{-11} \cdot T_f)]$	
Volumetric Expansion Coefficient, β	$\beta = \frac{1}{1.8 \cdot T_k}$	$°R^{-1}$
Density, ρ	$\rho = \frac{22.0493}{T_k}$	lb/ft^3
Kinematic Viscosity, ν	$\nu = \frac{\mu}{\rho}$	ft^2/h
Specific Heat, c_p	$c_p = 0.24008 - T_f \cdot [1.2477 \times 10^{-6} - T_f \cdot (4.0489 \times 10^{-8} - 1.6088 \times 10^{-11} \cdot T_f)]$	Btu/(lb·ft)

APPENDIXES

(Nonmandatory Information)

X1. APPLICATION OF PRACTICE C 680 TO FIELD MEASUREMENTS

X1.1 This appendix has been included to provide a more complete discussion of the precision and bias expected when using this practice in the analysis of operating systems. While much of the discussion below is relevant to the practice, the errors associated with its application to operating systems is beyond the immediate scope of this task group. Portions of this discussion, however, were used in developing the Precision and Bias statements included in Section 10.

X1.2 This appendix will consider precision and bias as it relates to the comparison between the calculated results of the Practice C 680 analysis and measurements on operating systems. Some of the discussion here may also be found in Section 10; however, items are expanded here to include analysis of operating systems.

X1.3 *Precision*:

X1.3.1 The precision of this practice has not yet been demonstrated as described in Specification E 691, but an interlaboratory comparison could be conducted, if necessary, as facilities and schedules permit. Assuming no errors in programming or data entry, and no computer hardware malfunctions, an interlaboratory comparison should yield the theoretical precision presented in X1.3.2.

X1.3.2 The theoretical precision of this practice is a function of the computer equipment used to generate the calculated results. Typically, seven significant digits are resident in the computer for calculations. The use of "Double Precision" can expand the number of digits to sixteen. However, for the intended purpose of this practice, standard levels of precision are adequate. The effect of computer resolution on accuracy is only significant if the level of precision is higher than seven digits. Computers in use today are accurate in that they will reproduce the calculation results to the resolution required if identical input data is used.

X1.3.2.1 The formatting of output results from this has been structured to provide a resolution of 0.1 % for the typically expected levels of heat flux, and within 0.1°F (0.05°C) for surface temperatures.

X1.3.2.2 A systematic precision error is possible due to the choices of the equations and constants for convective and radiative heat transfer used in the program. The interlaboratory comparison of X1.3.3 indicates that this error is usually within the bounds expected in in-situ heat flow calculations.

X1.3.3 *Precision of Surface Convection Equations*:

X1.3.3.1 Many empirically derived equation sets exist for the solution of convective heat transfer from surfaces of various shapes in various environments. If two different equation sets are chosen and a comparison is made using identical input data, the calculated results are never identical, not even when the conditions for application of the equations appear to be identical. For example, if equations designed for vertical surfaces in turbulent cross flow are compared, results from this comparison could be used to help predict the effect of the equation sets on overall calculation precision.

X1.3.3.2 The systematic precision of the surface equation set used in this practice has had at least one through intralaboratory evaluation **(9)**. When the surface convective coefficient equation (see 6.6) of this practice was compared to another surface equation set by computer modeling of identical conditions, the resultant surface coefficients for the 240 typical data sets varied, in general, less than 10 %. One extreme case (for flat surfaces) showed variations up to 30 %. Other observers have recorded larger variations (in less rigorous studies) when additional equation sets have been compared. Unfortunately, there is no standard for comparison since all practical surface coefficient equations are empirically derived. The equations in 6.6 are accepted and will continue to be recommended until evidence suggests otherwise.

X1.3.4 *Precision of Radiation Surface Equation*:

X1.3.4.1 The Stefen-Boltzman equation for radiant transfer is widely applied. In particular, there remains some concern as to whether the exponents of temperature are exactly 4.0 in all cases. A small error in these exponents cause a larger error in calculated radiant heat transfer. The exactness of the coefficient 4 is well-founded in both physical and quantum physical theory and is therefore used here.

X1.3.4.2 On the other hand, the ability to measure and preserve a known emittance is quite difficult. Furthermore, though the assumptions of an emittance of 1.0 for the surroundings and a "sink" temperature equal to ambient air temperature is often approximately correct in a laboratory environment, operating systems in an industrial environment often diverge widely from these assumptions. The effect of using 0.95 for the emittance of the surroundings rather than the 1.00 assumed in the previous version of this practice was also investigated by the task group **(9)**. Intralaboratory analysis of the effect of assuming a surrounding effective emittance 0.95 versus 1.00 indicates a variation of 5 % in the radiation surface coefficient when the object emittance is 1.00. As the object emittance is reduced to 0.05, the difference in the surface coefficient becomes negligible. These differences would be greater if the surrounding effective emittance is less than 0.95.

X1.3.5 *Precision of Input Data*:

X1.3.5.1 The heat transfer equations used in the computer program of this practice imply possible sources of significant errors in the data collection process, as detailed later in this appendix.

NOTE X1.1—Although data collection is not within the scope of this practice, the results of this practice are highly dependent on accurate input data. For this reason, a discussion of the data collection process is included here.

X1.3.5.2 A rigorous demonstration of the impact of errors associated with the data collection phase of an operating system's analysis using Practice C 680 is difficult without a parametric sensitivity study on the method. Since it is beyond the intent of this discussion to conduct a parametric study for all possible cases, X1.3.5.3-X1.3.5.7 discuss in general terms the potential for such errors. It remains the responsibility of users to conduct their own investigation into the impact of the analysis assumptions particular to their own situations.

X1.3.5.3 *Conductivity Data*—The accuracy and applicability of the thermal conductivity data are derived from several factors. The first is the accuracy of the test method used to generate the data. Since Test Methods C 177, C 335, and C 518 are usually used to supply test data, the results reported for these tests should contain some statement of estimated error or estimated uncertainty. The remaining factors influencing the accuracy are the inherent variability of the product and the variability of insulation installation practice. If the product variability is large or the installation is poor, or both, serious differences might exist between the measured performance and the performance predicted by this method.

X1.3.5.4 *Surface Temperature Data*—There are many techniques for collecting surface temperatures from operating systems. Most of these methods assuredly produce some error in the measurement due to the influence of the measurement on the operating condition of the system. Additionally, the intended use of the data is important to the method of surface temperature data collection. Most users desire data that is representative of some significant area of the surface. Since surface temperatures frequently vary significantly across operating surfaces, single-point temperature measurements usually lead to errors. Sometimes very large errors occur when the data is used to represent some integral area of the surface. Some users have addressed this problem through various means of determining average surface temperature, Such techniques will often greatly improve the accuracy of results used to represent average heat flows. A potential for error still exists, however, when theory is precisely applied. This practice applies only to areas accurately represented by the average point measurements, primarily because the radiation and convection equations are non-linear and do not respond correctly when the data is averaged. The following example is included to illustrate this point:

(1) Assume the system under analysis is a steam pipe. The pipe is jacketed uniformly, but one-half of its length is poorly insulated, while the second half has an excellent insulation under the jacket. The surface temperature of the good half is measured at 550°F. The temperature of the other half is measured at 660°F. The average of the two temperatures is 605°F. The surface emittance is 0.92, and ambient temperature is 70°F. Solving for the surface radiative heat loss rates for each half and for the average yields the following:

(2) The average radiative heat loss rate corresponding to a 605°F temperature is 93.9 Btu/ft^2/h.

(3) The "averaged" radiative heat loss obtained by calculating the heat loss for the individual halves, summing the total and dividing by the area, yields an "averaged" heat loss of 102.7 Btu/ft^2/h. The error in assuming the averaged surface temperature when applied to the radiative heat loss for this case is 8.6 %.

(4) It is obvious from this example that analysis by the methods described in this practice should be performed only on areas which are thermally homogeneous. For areas in which the temperature differences are small, the results obtained using Practice C 680 will be within acceptable error bounds. For large systems or systems with significant temperature variations, total area should be subdivided into regions of nearly uniform temperature difference so that analysis may be performed on each subregion.

X1.3.5.5 *Ambient Temperature Variations*—In the standard analysis by the methods described in his practice, the temperature of the radiant surroundings is taken to be equal to the ambient air temperature (for the designer making comparative studies, this is a workable assumption). On the other hand, this assumption can cause significant errors when applied to equipment in an industrial environment, where the surroundings may contain objects at much different temperatures than the surrounding air. Even the natural outdoor environment does not conform well to the assumption of air temperatures when the solar or night sky radiation is considered. When this practice is used in conjunction with in-situ measurements of surface temperatures, as would be the case in an audit survey, extreme care must be observed to record the environmental conditions at the time of the measurements. While the computer program supplied in this practice does not account for these differences, modifications to the program may be made easily to separate the convective ambient temperature from the mean radiative environmental temperature seen by the surface. The key in this application is the evaluation of the magnitude of this mean radiant temperature. The mechanism for this evaluation is beyond the scope of this practice. A discussion of the mean radiant temperature concept is included in the ASHRAE Handbook of Fundamentals **(2)**.

X1.3.5.6 *Emittance Data*—Normally, the emittance values used in a Practice C 680 analysis account only for the emittance of the subject of the analysis. The subject is assumed to be completely surrounded by an environment which has an assigned emittance of 0.95. Although this assumption may be valid for most cases, the effective emittance used in the calculation can be modified to account for different values of effective emittance. If this assumption is a concern, using the following formula for effective surface emittance will correct for this error:

$$\epsilon_{eff} = \frac{A_A}{(1-\epsilon_A) / (\epsilon_A A_A + 1/A_A F_{AB} + (1-\epsilon_B) / \epsilon_B A_B} \tag{X1.1}$$

where:

ϵ_{eff} = effective mean emittance for the two surface combination,
ϵ_A = mean emittance of the surface A,
ϵ_B = mean emittance of the surrounding region B,
F_{AB} = view factor for the surface A and the surrounding region B,
A_A = area of region A, and
A_B = area of region B.

This equation set is described in most heat transfer texts on heat transfer. See Holman **(4)**, p. 305.

X1.3.5.7 *Wind Speed*—Wind speed is defined as wind speed measured in the main airstream near the subject surface. Air blowing across real objects often follows flow directions and velocities much different from the direction and velocity of the main free stream. The equations used in Practice C 680 analysis yield “averaged” results for the entire surface in question. Because of this averaging, portions of the surface will have different surface temperatures and heat flux rates from the average. For this reason, the convective surface coefficient calculation cannot be expected to be accurate at each location on the surface unless the wind velocity measurements are made close to the surface and a separate set of equations are applied that calculate the local surface coefficients.

X1.3.6 *Theoretical Estimates of Precision*:

X1.3.6.1 When concern exists regarding the accuracy of the input test data, the recommended practice is to repeat the calculation for the range of the uncertainty of the variable. This process yields a range of the desired output variable for a given input variable uncertainty. Several methods exist for evaluating the combined variable effects. Two of the most common are illustrated as follows:

X1.3.6.2 The most conservative method assumes that the errors propagating from the input variable uncertainties are additive for the function. The effect of each of the individual input parameters is combined using Taylor’s Theorem, a special case of a Taylor’s series expansion **(10)**.

$$\frac{S}{R} = \sum_{i=1}^{n} \left| \frac{\partial R}{\partial x_i} \right| \cdot \Delta x_i \quad \text{(X1.2)}$$

where:

S = estimate of the probable error of the procedure,
R = result of the procedure,
x_i = ith variable of the procedure,
$\partial R / \partial S$ = change in result with respect to a change in the ith variable (also, the first derivative of the function with respect to the ith variable),
x_i = uncertainty in value of variable i, and
n = total number of input variables in the procedure.

X1.3.6.3 For the probable uncertainty of function, R, the most commonly used method is to take the square root of the sum of the squares of the fractional errors. This technique is also known as Pythagorean summation. This relationship is described in Eq 39, Section 10.

X1.3.7 *Bias of Practice C 680 Analysis*:

X1.3.7.1 As in the case of the precision, the bias of this standard practice is difficult to define. From the preceding discussion, some bias can result due to the selection of alternative surface coefficient equation sets. If, however, the same equation sets are used for a comparison of two insulation systems to be operated at the same conditions, no bias of results is expected from this method. The bias due to computer differences will be negligible in comparison with other sources of potential error. Likewise, the use of the heat transfer equations in the program implies a source of potential bias errors, unless the user ensures the applicability of the practice to the system.

X1.3.8 *Error Avoidance*—The most significant sources of possible error in this practice are in the misapplication of the empirical formulae for surface heat transfer coefficients, such as using this practice for cases that do not closely fit the thermal and physical model of the equations. Additional errors evolve from the superficial treatment of the data collection process. Several promising techniques to minimize these sources of error are in stages of development. One attempt to address some of the issues has been documented by Mack **(11)**. This technique addresses all of the above issues except the problem of non-standard insulation k values. As the limitations and strengths of in-situ measurements and Practice C 680 analysis become better understood, they can be incorporated into additional standards of analysis that should be associated with this practice. Until such methods can be standardized, the best assurance of accurate results from this practice is tat each application of the practice will be managed by a user who is knowledgeable in heat transfer theory, scientific data collection practices, and the mathematics of programs supplied in this practice.

X2. COMMENTARY

X2.1 *Introduction and History of Practice C 680*:

X2.1.1 The history of the development of this practice has been prepared for inclusion in the document. The following discussion, while not complete, provides a brief overview of the changes that have taken place over the years since the practice was first written.

X2.1.2 The practice was originally published in 1971. A program listing written in FORTRAN was included to allow the user to be able to calculate heat losses and surface temperatures of a variety of insulated piping and equipment. The user had to have access to a computer, a method of typing the program into a usable form, then running the program to get the results. At that time the most common method of entering a program was to prepare a card deck. Each card in the deck represented a line of program code or a line of data required by the program. The deck was then read by a card reader and the program run with the output printed on a printer. There was much discussion on the choice of equations to use for the determination of the surface heat transfer coefficient. The task group finally selected a modified form of the equations published in Ref **(3)**. Langmuir was credited with equations for natural convection and a multiplier to account for forced convection. Rice and Heilman were credited for the development of equations representing heat loss from a variety of surfaces. Langmuir presents theoretical analyses of convection heat transfer from wires and plane surfaces and experimental data for plane surfaces. For wires, he refers to earlier published data on platinum wires having diameters from 0.0016 to 0.020 inches. Because of the small size of the wires, experimental convection coefficients for them cannot be applied to much larger pipes. For plane surfaces, Langmuir experimented with circular metal disks, 7½-in. in diameter. The total heat loss was measured for the disks when placed in still air at 80.3°F and heated to temperatures of 125°F to 1160°F. One of the disks was made of pure polished silver, which had a very low emittance. The emittance for this disk was estimated from the theoretical Hagen-Rubens equation, and the radiation heat transfer was calculated and subtracted from the total heat loss to give the natural convection coefficient. The convection heat losses for a vertical surface were compared with a theory by Lorenz. Langmuir noted that changing the numerical coefficient from 0.296 to 0.284 would give good agreement with his measured data. He noted that convection from a horizontal surface facing upwards was about 12 % larger than for a vertical surface (actually, his data indicates the percentage to be closer to 10 %). For a horizontal surface facing downwards, he states that the convection is about one-half as great as that facing upwards (his numbers indicate a factor of 0.45 rather than 0.5).

X2.1.3 To investigate the effect of air currents, Langmuir made measurements on a 7½-in. diameter vertical disk of "calorized" steel. The steel disk was heated to 932°F. Heat loss measurements were made in still air and then when subjected to the wind produced by an electric fan. Wind speeds of 6.0, 8.3, and 9.2 miles per hour were used. From these data, he derived the factor used in the practice. Since these data were taken with one geometry, one surface size, and one surface temperature, it is not obvious that his correlation can be generalized to all other conditions. Langmuir does note that Kennelly had found a similar factor for the effect of wind on small wires. Instead of the factor of 1.277, Kennelly obtained a factor of 1.788. Kennelly's wires were less than 0.0275 in. in diameter.

X2.1.4 Heilman measured the total heat loss from nominal 1-in., 3-in., and 10-in. bare steel pipes. The pipes were surrounded by still air at 80°F. Data were obtained for the 1-in. pipe for pipe temperatures from about 200°F to about 650°F. For the 3-in. pipe, the temperature range was 175 to 425°F, and for the 10-in. pipe, the range was 125 to 390°F. He made independent measurements of the emittance, calculated the radiation heat loss, and subtracted this from the total heat loss to obtain the convective heat loss. He obtained his correlation from dimensional reasoning and analysis of this data.

X2.1.5 Heilman also measured the total heat loss from 1-in. and 3-in. vertical pipes with heights of 3 feet. These data led to the factor of 1.235 to be used his correlation. For plane vertical surfaces, he used three heavily silver-plated and highly polished brass disks. The silver plating and polishing greatly reduced the radiation heat loss. The three plates had diameters of 3.47, 6.55, and 9.97 inches, and corresponding thicknesses of 0.758, 0.80, and 0.90 inches. From data on these disks, he derived the factor of 1.394. He suggested that "further investigational work should be carried out on larger plane surfaces than were used during this investigation." For horizontal plates, he relied upon experimental data of Griffiths and Davis on a 50-in. square plate. They found the convection upward from a horizontal plate to be 28 % higher than for a vertical plate, and the convection downward to be about 34 % less than that for a vertical plate. Heilman applied these percentage changes to the factor of 1.394 to obtain factors of 1.79 and 0.89 for the horizontal plate facing upward and downward, respectively. Heilman's paper deals only with still air conditions, and thus his equations do not contain any reference to wind speed. The multiplication of Heilman's equation by Langmuir's wind factor appears to have been made later by Malloy.

X2.2 The next major revision occurred in 1982. The program was rewritten in the BASIC programming language to make it more readily available to users of desktop personal computers, since BASIC came with the operating system. There were no major changes in the methodology or the equations used to determine the surface heat transfer coefficients.

X2.3 The 2002 revision represents a major change in the determination of the surface heat transfer coefficient. After the work of Langmuir, Rice, and Heilman, many improved correlations of more extensive sets of data have been published. Prominent heat transfer texts by McAdams, Holman, and Incropera and DeWitt all list recommended correlations, see Refs **(21-29)**. Correlations presented by Holman and by Incropera and DeWitt are very similar. In general, the correlations by Incropera and DeWitt are used in this revision. There was also discussion on the use of the ISO equations.

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