Standard Practice for Determination of Heat Gain or Loss and the Surface Temperatures of Insulated Pipe and Equipment Systems by the Use of a Computer Program

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 The computer programs included in this practice provide a calculational procedure for predicting the heat loss or gain and surface temperatures of insulated pipe or equipment systems. This procedure is based upon an assumption of a uniform insulation system structure, that is, a straight run of pipe or flat wall section insulated with a uniform density insulation. Questions of applicability to real systems should be resolved by qualified personnel familiar with insulation systems design and analysis. In addition to applicability, calculational accuracy is also limited by the range and quality of the physical property data for the insulation materials and systems.

1.2 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:
C 168 Terminology Relating to Thermal Insulating Materials
C 335 Test Method for Steady-State Heat Transfer Properties of Horizontal Pipe Insulation
C 585 Practice for Inner and Outer Diameters of Rigid Thermal Insulation for Nominal Sizes of Pipe and Tubing (NPS System)
E 691 Practice for Conducting an Interlaboratory Study to Determine the Precision of a Test Method

2.2 ANSI Standards:
X3.5 Flow Chart Symbols and Their Usage in Information Processing
X3.9 Standard for Fortran Programming Language

3. Terminology

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

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 coefficient, \( \text{Btu/(h·ft}^2·\text{°F})(\text{W/(m}^2·\text{K}) \)
- \( k \) = thermal conductivity, \( \text{Btu-in.}/(\text{h·ft}^2·\text{°F})(\text{W/(m}·\text{K}) \)
- \( k_a \) = constant equivalent thermal conductivity introduced by the Kirchhoff transformation, \( \text{Btu-in.}/(\text{h·ft}^2·\text{°F})(\text{W/(m·K}) \)
- \( Q_t \) = total time rate of heat flow, \( \text{Btu/h} (\text{W}) \)
- \( Q_l \) = time rate of heat flow per unit length, \( \text{Btu/h·ft} (\text{W/m}) \)
- \( q \) = time rate of heat flow per unit area, \( \text{Btu/h·ft}^2 (\text{W/m}^2) \)
- \( R \) = thermal resistance, \( (\text{°F·h·ft}^2)/\text{Btu} (\text{K·m}^2/\text{W}) \)
- \( r \) = radius, in. (m)
- \( t \) = local temperature, °F (K)
- \( t_i \) = temperature of inner surface of the insulation, °F (K)
- \( t_a \) = temperature of ambient fluid and surroundings, °F (K)
- \( x \) = distance in direction of heat flow (thickness), in. (m)

4. Summary of Practice

4.1 The procedures used in this practice are based upon standard steady-state heat transfer theory as outlined in textbooks and handbooks. The computer program combines the functions of data input, analysis, and data output into an...
4.2 The operation of the computer program follows the procedure listed below:

4.2.1 Data Input—The computer requests and the operator inserts information that describes the system and operating environment. The data include:

4.2.1.1 Analysis Identification.
4.2.1.2 Date.
4.2.1.3 Ambient Temperature.
4.2.1.4 Surface coefficient or ambient wind speed, insulation system surface emittance, and orientation.
4.2.1.5 System Description—Layer number, material, and thicknesses.

4.2.2 Analysis—Once input data is entered, the program calculates the surface coefficients (if not entered directly) and the layer resistances, then uses that data to calculate the heat losses and surface temperatures. The program continues to repeat the analysis using the previous temperature data to update the estimates of layer resistance until the temperatures at each surface repeat with a specified tolerance.

4.2.3 Once convergence of the temperatures is reached, the program prints a table giving the input data, the resulting heat flows, and the inner surface and external surface temperatures.

5. Significance and Use

5.1 Manufacturers of thermal insulations express the performance of their products in charts and tables showing heat gain or loss per lineal foot of pipe or square foot of equipment surface. These data are presented for typical operating temperatures, pipe sizes, and surface orientations (facing up, down, or horizontal) for several insulation thicknesses. The insulation surface temperature is often shown for each condition, to provide the user with information on personnel protection or surface condensation. Additional information on effects of wind velocity, jacket emittance, and ambient conditions may also be required to properly select an insulation system. Due to the infinite combinations of size, temperature, humidity, thickness, jacket properties, surface emittance, orientation, ambient conditions, etc., it is not practical to publish data for each possible case.

5.2 Users of thermal insulation, faced with the problem of designing large systems of insulated piping and equipment, encounter substantial engineering costs to obtain the required thermal information. This cost can be substantially reduced by both the use of accurate engineering data tables, or by the use of available computer analysis tools, or both.

5.3 The use of analysis procedures described in this practice can also apply to existing systems. For example, C 680 is referenced for use with Procedures C 1057 and C 1055 for burn hazard evaluation for heated surfaces. Infrared inspection or in situ heat flux measurements are often used in conjunction with C 680 to evaluate insulation system performance and durability on operating systems. This type analysis is often made prior to system upgrades or replacements.

5.4 The calculation of heat loss or gain and surface temperature of an insulated system is mathematically complex and because of the iterative nature of the method, is best handled by computers.

5.5 The thermal conductivity of most insulating materials changes with mean temperature. Since most thermal insulating materials rely on enclosed air spaces for their effectiveness, this change is generally continuous and can be mathematically approximated. In the cryogenic region where one or more components of the air condense, a more detailed mathematical treatment may be required. For those insulations that depend on high molecular weight, that is, fluorinated hydrocarbons, for their insulating effectiveness, gas condensation will occur at higher temperatures and produce sharp changes of conductivity in the moderate temperature range. For this reason, it is necessary to consider the temperature conductivity dependence of an insulation system when calculating thermal performance. The use of a single value thermal conductivity at the mean temperature will provide less accurate predictions, especially when bridging regions where strong temperature dependence occurs.

5.6 The use of this practice by both manufacturers and users of thermal insulations will provide standardized engineering data of sufficient accuracy for predicting thermal insulation performance.

5.7 Computers are now readily available to most producers and consumers of thermal insulation to permit the use of this practice.

5.8 Two separate computer programs are described in this practice as a guide for calculation of the heat loss or gain, and surface temperatures, of insulated pipe and equipment 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. Both programs are intended for use with an “interactive” terminal. With 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, prompting user response. This facilitates problem solution and increases the probability of successful computer runs.

5.8.1 Program C 608E is designed for an interactive solution of equipment heat transfer problems.

5.8.2 Program C 608P is designed for interactive solution of piping-system problems. The subroutine SELECT has been written to provide input for the nominal iron pipe sizes as shown in Practice C 585, Tables 1 and 3. The use of this program for tubing-systems problems is possible by rewriting subroutine SELECT such that the tabular data contain the appropriate data for tubing rather than piping systems (Practice C 585, Tables 2 and 4).

5.8.3 Combinations of the two programs are possible by using an initial selector program that would select the option being used and elimination of one of the k curve and surface coefficient subroutines that are identical in each program.

5.8.4 These programs are designed to obtain results identical to the previous batch program of the 1971 edition of this practice. The only major changes are the use of an interactive terminal and the addition of a subroutine for calculating surface coefficient.

5.9 The user of this practice may wish to modify the data
input and report sections of the computer program presented here to fit individual needs. Also, additional calculations may be desired to include other data such as system costs or economic thickness. No conflict with this method in making these modifications exists, provided that the user has demonstrated compatibility. Compatibility is demonstrated using a series of test cases covering the range for which the new method is to be used. For those cases, results for the heat flow and surface temperatures must be identical, within the resolution of the method, to those obtained using the method described herein.

5.10 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 would provide an SI equivalent of the heat-flow results, no such “metric” equivalent is available for the other portions of the program. To date, there is no accepted metric dimensions system for pipe and insulation systems for cylindrical shapes. The dimensions in use in Europe are the SI dimension equivalents of the American sizes, and in addition have different designations in each country. Therefore, due to the complexity of providing a standardized equivalent of this procedure, no SI version of this practice has been prepared. At the time in which an international standardization of piping and insulation sizing occurs, this practice can be rewritten to meet those needs. This system has also been demonstrated to calculate the heat loss for bare systems by the inclusion of the pipe/equipment wall thermal resistance into the equation system. This modification, although possible, is beyond the scope of this practice.

6. Method of Calculation

6.1 Approach:

6.1.1 This calculation of heat gain or loss, and surface temperature, requires (1) that the thermal insulation be homogeneous as outlined by the definition of thermal conductivity in Terminology C 168; (2) that the pipe size and equipment operating temperature be known; (3) that the insulation thickness be known; (4) that the surface coefficient of the system be known, or sufficient information be available to estimate it as described in 7.4; and (5) that the relation between thermal conductivity and mean temperature for the insulation be known in detail as described in 7.3.

6.1.2 The solution is a computer procedure calling for (1) estimation of the system temperature distribution, (2) calculation of the thermal resistances throughout the system based on that distribution, and (3) then reestimation of the temperature distribution from the calculated resistances. The iteration continues until the calculated distribution is in agreement with the estimated distribution. The layer thermal resistance is calculated each time with the equivalent thermal conductivity being obtained by integration of the conductivity curve for the layer being considered. By this technique, the thermal conductivity variation of any insulation or multiple-layer combination of insulations can be taken into consideration when calculating the heat flow.

6.2 Development of Equations—The development of the mathematical equations centers on heat flow through a homogeneous solid insulation exhibiting a thermal conductivity that is dependent on temperature. Existing methods of thermal conductivity measurement account for the thermal conduction, convection, and radiation occurring within the insulation. After the basic equations are developed, they are extended to composite (multiple-layer) cases and supplemented with provision for heat flow from the outer surface by convection or radiation, or both.

6.3 Equations—Case 1, Slab Insulation:

6.3.1 Case 1 is a slab of insulation shown in Fig. 1 having width \( W \), height \( H \), and thickness \( T \). It is assumed that heat flow occurs only in the thickness of \( x \) direction. It is also assumed that the temperature \( t_j \) of the surface at \( x_j \) is the same as the equipment surface temperature and the time rate of heat flow per unit area entering the surface at \( x_1 \) is designated \( q_1 \). The time rate of heat flow per unit area leaving the surfaces at \( x_2 \) is \( q_2 \).

6.3.1.1 For the assumption of steady-state (time-independent) condition, the law of conservation of energy dictates that for any layer the time rate of heat flow in must equal the time rate of heat flow out, i.e., there is no net storage of energy inside the layer.

6.3.1.2 Taking thin sections of thickness \( \Delta x \), energy balances may be written for these sections as follows: Case 1:

\[
(WHq)|_x - (WHq)|_{x + \Delta x} = 0 \quad (1)
\]

Note: 1—The vertical line with a subscript indicates the point at which the previous parameter is evaluated. For example: \( q_1|_{x + \Delta x} \) reads the time rate of heat flow per unit area, evaluated at \( x + \Delta x \).

6.3.1.3 After dividing Eq 1 by \( \Delta x \), and taking the limit as \( \Delta x \) approaches zero, the differential equation for heat transfer is obtained for the one-dimensional case:

\[
(d/dx)q = 0 \quad (2)
\]

6.3.1.4 Integrating Eq 2 and imposing the condition of heat flow stability on the result yields the following:

\[
q = q_1 = q_2 \quad (3)
\]
6.3.1.5 When the thermal conductivity, \( k \), is a function of local temperature, \( t \), the Fourier law must be substituted in Eq 2. Fourier’s Law for one-dimensional heat transfer can be stated mathematically as follows:

\[
q = -k \frac{dt}{dx}
\]  

(4)

Therefore,

\[
\frac{d}{dx} q = (d/dx)(-k \frac{dt}{dx}) = 0
\]  

(5)

6.3.1.6 To retain generality, the functionality of \( k \) with \( t \) is not defined at this point, therefore, Eq 5 cannot be integrated directly. The Kirchhoff transformation \(^5\) allows integration by introducing an auxiliary variable \( u \) and a constant \( k_a \) defined by the differential equation as follows:

\[
k_a \frac{du}{dx} = k \frac{dt}{dx}
\]  

(6)

This equation must be satisfied by the following boundary conditions:

\[
u = t_i \text{ at } x = x_i
\]

\[
u = t_2 \text{ at } x = x_2
\]

6.3.1.7 Rederiving Eq 4 in terms of Eq 6, integrating, and imposing the boundary conditions for the transformation yields the following:

\[
q_i = \frac{t_1 - t_2}{\frac{x_1 - x_2}{k_a}}
\]  

(7)

6.3.1.8 Eq 7 is in a familiar form of the conductive heat transfer equation used when thermal conductivity is assumed constant with local temperatures. To evaluate the equivalent thermal conductivity, Eq 6 is solved for \( k_a \). Separating variables in either equation and integrating through the boundary conditions, the following general relation is obtained:

\[
k_a = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} k \, dt
\]  

(8)

Evaluation of the integral in Eq 8 can be handled analytically where \( k \) is a simple function, or by numerical methods where \( k \) cannot be integrated. Particular solutions of Eq 8 are discussed in 6.5.

6.3.2 The equations for heat flow through a single-layer insulation can now be extended to the multiple layer or composite insulation case. Consider Fig. 2 as a multiple-layer extension of the simple case. The figure shows the composite system with insulations having different thermal conductivities.

6.3.2.1 Equations can be written for each additional layer analogous to Eq 7. With the entire system at stability and assuming no temperature drop across layer interfaces, the equation is written as follows:

\[
q_{i+1} = \frac{t_i - t_{i+1}}{\frac{x_i - x_{i+1}}{k_{a_{i+1}}}}
\]  

(9)

Note 2—The generalized index, \( i \), denotes any interface within the system.

6.3.2.2 It is useful at this point to introduce the concept of thermal resistance, that is, the heat flow per unit area given simply by a temperature difference divided by the corresponding thermal resistance. The heat flow per unit area at the outer surface, \( q_n \), is calculated as follows:

\[
q_n = \frac{(t_i - t_{i+1})}{R_{i+1}}
\]  

(10)

where:

\[
R_{i+1} = \frac{(x_i - x_{i+1})}{k_{a_{i+1}}}
\]  

(11)

6.3.3 Characterization of the heat flow from the systems can be completed by developing an expression for the rate of heat flow per unit area at the outer solid surfaces. For this purpose, the following definition of the surface coefficient is employed:

\[
h = q_RGB
\]  

(12)

or

\[
q_n = \frac{(t_n - t_a)}{(1/h)}
\]  

(13)

Because of the similarity between Eq 10 and Eq 13, Eq 13 can be rewritten as follows:

\[
q_n = \frac{(t_n - t_i)}{R_i}
\]  

(14)

where:

\[
R_i = (1/h)
\]  

(15)

6.3.4 The surface coefficient, \( h \), is a complex function of the properties of the ambient fluid, surface geometry, the temperatures of the system, the surface finish, and motion of the ambient fluid. Equations used by this practice for estimating the surface coefficient are discussed in 7.4.

6.3.4.1 Summing the series of equations from 6.3.2 including equations from 6.3.3 yields the following expression for the heat flow through the entire composite system:

\[
q_n = \frac{(t_1 - t_n)}{R_s}
\]  

(16)

where:

\[
R_s = R_{1,2} + R_{2,3} + \ldots + R_{n-1,n} + R_s
\]  

(17)

6.3.4.2 Setting the heat flow per unit area through each element, \( q_r \), equal to the heat flow through the entire system, \( q_n \),
shows that the ratio of the temperature across the element to the
temperature difference across the entire system is proportional
to the ratio of the thermal resistance of the element to the total
thermal resistance of the system or in general terms.

\[
\frac{(t_i - t_{i+1})}{(t_1 - t_0)} = \frac{1}{R_{i,i+1}/R_s}
\]  

Eq 17 provides the means of solving for the temperature
distribution. Since the resistance of each element depends on
the temperature of the element, the solution can be found only
by iteration methods.

6.4 Equations—Case 2, Cylindrical Sections:

6.4.1 For Case 2, Figs. 3 and 4, the analysis used is similar
to that described in 6.3, but with the replacement of the
variable x by the cyclindrical coordinate, r. The following
generalized equation is used to calculate the conductive heat
flow through a layer of a cylinder wall.

\[
q_{i+1} = \frac{r_i - t_{i+1}}{R_{i,i+1}} \ln \left( \frac{r_i r_{i+1}}{k_{a,i+1}} \right)
\]  

(18)

Note the similarity of Eq 9 and Eq 18 and that the solution
of the transformation equation for the radial heat flow case is
identical to that of the slab case (see Eq 8).

6.4.2 As in Case 1, calculations for slabs, simplification of
the equations for the heat loss may be accomplished by
defining the thermal resistance. For pipe insulations, the heat
flow per unit area is a function of the overall thermal
resistance of the system or in general terms.

The outer radius, \( r_o \), of the insulation system is chosen for this
purpose. The heat flow per unit area for cylinders, calculated at
the outer surface, \( r_o \), is:

\[
q_o = (t_i - t_{i+1})R_{i,i+1}
\]  

(19)

where:

\[
R_{i,i+1} = \frac{r_i}{k_a} \ln \left( \frac{r_i r_{i+1}}{k_{a,i+1}} \right)
\]  

(20)

6.4.3 The concept of surface resistance used in an analysis
similar to 6.3.3 and 6.3.4 permits introduction of the definition
of the heat transfer as a function of the overall thermal
resistance for the cylindrical case as follows:

\[
q_{i+1} = \frac{t_i - t_{i+1}}{R_{i,i+1}} \ln \left( \frac{r_i r_{i+1}}{k_{a,i+1}} \right)
\]  

(17)

Case 2

\[ q_o = (t_i - t_{i+1})R_i \]  

(21)

where:

\[ R_i = R_{i,2} + R_{i,3} + R_{i,4} + ... + R_{i,n-1} + R_i \]

Note 3—In some situations where comparisons of the insulation
system performance is to be made, basing the areal heat loss on the inside
surface area, which is fixed by the pipe dimensions, or on the heat loss per
unit length, is beneficial. The heat loss per unit area of the inside surface
is calculated from the heat loss per unit area of the outside surface by
multiplying by the ratio of the outside radius to the inside radius. For
calculation of the heat loss per linear foot from the heat loss per outside
area, simply multiply by the outside area per foot or \( 2\pi r_o \). For Case 2, the
annulus, results are normally expressed as the time rate of heat flow per
unit length, \( Q_1 \), which is obtained as follows:

\[ Q_1 = 2\pi r_o q_o = 2\pi r_o (t_i - t_2)/R_i \]  

(22)

6.5 Calculation of Effective Conductivity:

6.5.1 In Eq 11-22 it is necessary to evaluate \( k_a \) as a function
of temperature for each of the conductive elements. The
generalized solution in Eq 8 is as follows:

\[ k_{a,i+1} = \frac{1}{(t_i - t_{i+1})} \int_{t_i}^{t_{i+1}} k dt \]

6.5.2 When \( k \) may be described in terms of a simple
function of \( t \), an analytically exact solution for \( k_a \) can be
obtained. The following functional types will be considered in
the examples (see 9.1-9.4).

6.5.2.1 If \( k \) is linear with \( t \), \( k = a + bt \) and

\[ k_a = \frac{1}{(t_{i+1} - t_i)} \int_{t_i}^{t_{i+1}} (a + bt) dt = a + b \left( \frac{t_{i+1} + t_i}{2} \right) \]  

(23)

where \( a \) and \( b \) are constants.

6.5.2.2 If

\[ k = e^{at+bt} \]

then:

\[ k_a = \frac{1}{(t_{i+1} - t_i)} \int_{t_i}^{t_{i+1}} e^{at+bt} dt \]

and evaluating the integral yields:

\[ k_a = \left[ \frac{1}{(t_{i+1} - t_i)} \right] \left[ e^{at_{i+1}+bt_{i+1}} - e^{at_i+bt_i} \right] \]  

(24)

where \( a \) and \( b \) are constants, and \( e \) is the base of the natural
logarithm.
6.5.2.3 If 
\[ k = a + bt + ct^2 \]
then:
\[ k_e = \frac{1}{(t_{i+1} - t_i)} \int_{t_i}^{t_{i+1}} (a + bt + ct^2) \, dt \]
and evaluating the integral yields:
\[ k_e = a + \frac{b}{2} (t_{i+1} + t_i) + \frac{c}{3} (t_{i+1} - t_i)^3 \] (25)
where \( a, b, \) and \( c \) are constants.

6.5.3 When the relationship of \( k \) with \( t \) is more complex and does not lend itself to simple mathematical treatment, a numerical method may 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 on the particular situation, and the details of the factors affecting the choice are beyond the scope of this practice.

7. Input Data

7.1 In general, data input is in accordance with ASTM Standards or American National Standards. The source of other required data is noted.

7.2 Dimensions of Pipe and Pipe Insulation:

7.2.1 Only nominal pipe sizes and insulation thicknesses are required as input data. The actual dimensions of both pipe and pipe insulation are obtained by the computer from a software file based on Practice C 585 during the calculation.

7.3 Thermal Conductivity Versus Mean Temperature:

7.3.1 The data describing the relationship of thermal conductivity to mean temperature are obtained in accordance with Test Methods C 177, C 335, or C 518, as appropriate for the product.

7.3.2 To describe accurately the relationship of thermal conductivity to mean temperature for thermal insulations, especially those exhibiting inflection points due to condensations of the insulating gases, thermal conductivity tests at small temperature differences are required. The minimum temperature differences used will depend on the vapor pressure to temperature of the gases involved, and the accuracy of the test apparatus at small temperature differences. Sufficient tests must be made to characterize the conductivity versus mean temperature relationship over the desired temperature range.

NOTE 4—ASTM Committee C-16 is currently developing recommendations for preparing thermal conductivity curves for use in systems analysis. Although the exact procedures are beyond the scope of this practice, caution should be exercised. The use of experimental data to generate curves must include consideration of test sample geometry, temperature range of data, test temperature differentials, thickness effects, test boundary conditions, and test equipment accuracy. Especially important is that the test data should cover a temperature range of conditions wider than those of the analysis, so that the data is interpolated for the analysis rather than extrapolated.

7.4 Surface Coefficients:

7.4.1 The surface coefficient, \( h \), as defined in Definitions C 168, assumes that the surroundings (fluid and visible surfaces) are at uniform temperature and that other visible surfaces are substantially perfect absorbers of radiant energy. It includes the combined effects of radiation, conduction, and convection.

7.4.2 In many situations surface coefficients may be estimated from published values (2).

7.4.3 Procedures for Calculating Surface Coefficients—Where known surface coefficients are not available, this practice provides a calculational procedure to estimate the surface coefficient. This calculation is based on the assumption of heat flow from a uniformly heated surface. This assumption is consistent with those used in developing the remainder of this practice. In simple terms, the surface coefficient equations are based on those commonly used in heat transfer analysis. A detailed discussion of the many heat flow mechanisms is present in several texts (3, 4, 5) or similar texts.

7.4.4 Analysis Configurations—Several convective conditions have been identified as requiring separate treatment when calculating the surface coefficient. The first is the two geometries treated in this method, that is, flat (equipment) and circular cylinder (pipe). Another case identifies the two air flow systems common to most applications. Free convection is defined as air motion caused by the buoyancy effects induced by the surface-to-air temperature difference. This case is characterized by low velocity, and, for most cases, includes any situation where the local air velocity is less than 1 mph (0.5 m/s). Forced convection is where some outside agent causes the air movement. For high air velocities, convection is the dominant mechanism of heat flow from the surface. The radiative heat flow surface coefficient is calculated separately and added to the convection losses since for a vast majority of cases, this mechanism operates independently of the convective transfer.

7.4.5 Surface Coefficient Calculation—Summary of Method—The convection coefficient calculation subroutine, SURCOF, developed for this practice, estimates the magnitude of the convection coefficient based upon the equations for the given set of geometric conditions and temperature-dependent air properties. The radiative component is also determined and added to yield the net surface coefficient. All equations used in the analysis (3) were experimentally developed. The equations used are briefly described in 7.4.7-7.4.9.

7.4.6 Alternative equation sets have been developed to calculate the surface heat transfer coefficients. These equation sets often include parameters in addition to those used in the development of the SURCOF subroutine described in this practice. These additional parameters are used to extend the data set to a wider range of conditions or better fit the data available. Use of these alternate equation sets instead of the SURCOF subroutine equation set is permitted, providing adequate documentation is provided and similarity of results is demonstrated under the exposure conditions covered by the SURCOF documentation (See Appendix X1) (3).

7.4.7 Convection:

7.4.7.1 Forced Convection—One of the major contributors to surface heat transfer is the convection of air across a surface where some difference exists between their temperatures. Not only is the rate of heat flow controlled by the magnitude of the temperature difference but also by the speed of the air flow as
it passes the surface. Since convection is a complex phenomena and has been studied by many researchers, many empirically developed equations exist for estimating the surface coefficients. One of the simpler to apply and more commonly used system of equations is that developed by Langmuir (6). His equations were developed for conditions of moderate temperatures which are most commonly seen in cases of insulated piping or equipment systems. For the condition of the natural convection of air at moderate temperature Langmuir proposed the following equation:

\[ Q_c = 0.296(t_s - t_a)^{1.25} \]  

(26)

where:
- \( Q_c \) = heat transferred by natural convections, Btu/ft² (J/m²),
- \( t_s \) = temperature of surface, °F (°C), and
- \( t_a \) = temperature of ambient, °F (°C).

7.4.7.2 Modifications for Forced Convection—When the movement of the air is caused by some outside force such as the wind, forced ventilation systems, etc. Langmuir (6) presented a modifier of Eq 26 to correct it for the forced convection. This multiplier was stated as follows:

\[ h_{cv} = \frac{V + 68.9}{68.9} \sqrt{1.00 + 1.277 \times \text{Wind}} \]  

(27)

where \( V \) is the bulk air velocity (ft/min). In a more commonly presented form where the velocity is miles per hour, this correction term reduces to

\[ h_{cv} = 1.25 \times \frac{V}{68.9} \sqrt{1 + 1.277 \times \text{Wind}} \]  

(28)

This equation will work for both forced and free convection because when Wind equals zero, the equation returns to its original form.

7.4.7.3 Convection for Geometric Variations—Further research by Rice and Heilman (7) refined the technology of Langmuir to account for changes in air film properties (density, thickness, viscosity) with the air film mean temperature. Also their refinements provided corrections to the equation form for geometric size, shape, and heat flow directions that permit use of the basic form of Langmuir’s (6) equation for a host of conditions. The result of their research yields the following equation set which forms the basis for the surface coefficient routines used in this practice.

\[ h_s = C \times \left( \frac{1}{d} \right)^{0.2} \times \left( \frac{1}{t_{avg}} \right)^{0.181} \times \Delta t^{0.266} \times \sqrt{1 + 1.277 \times \text{Wind}} \]  

(29)

where:
- \( h_s \) = convective surface coefficient, Btu/h·ft²·°F (W/(m²·K)),
- \( d \) = diameter for cylinder, in. (m). For flat surfaces and large cylinders \( d > 24 \), use \( d = 24 \),
- \( t_{avg} \) = average temperature of air film, °F (°C) = \( \frac{(t_s + t_a)}{2} \), and
- \( \Delta t \) = surface-to-air temperature difference, °F (°C), \( = (t_s - t_a) \).

7.4.7.4 The values of constant \( C \) are shown in Table 1 as a function of shape and heat flow condition.

7.4.8 Radiative Component—In each previous case, the radiative exchanges are for the most part independent of the convection exchange. The exception is that both help to determine the average surface temperature. The radiation coefficient is simply the radiative heat transfer rate, based upon the Stefan-Boltzman Law, divided by the average surface-to-air temperature difference. Thus the relationship can be expressed as the following:

\[ h_{rad} = \frac{E_{miss} \times 0.1713 	imes 10^{-8} (t_a + 459.6)^4 - (t_s + 459.6)^4}{(t_s - t_a)} \]  

(30)

where:
- \( E_{miss} \) = effective surface emittance (includes ambient emittance) and
- \( 0.1713 \times 10^{-8} \) = Stefan-Boltzman Constant (Btu/(h·ft²·°R⁴).

7.4.9 Overall Coefficient—Once the radiation and convection coefficients are determined for the specific case under investigation, the overall coefficient is determined by adding the two coefficients together.

\[ h = h_{cv} + h_{rad} \]  

(31)

8. Computer Programs

8.1 General:

8.1.1 The computer programs are written in Basic Fortran in accordance with ANSI X3.9.

Note 5—Identical versions of these computer programs have been successfully compiled and run on two processors. Only minor modifications necessary for conformance to the resident operating system were required for operation.

8.1.2 Each program consists of a main program and several subroutines. Other subroutines may be added to make the program more applicable to the specific problems of individual users.

8.1.3 The programs as presented call for the use of an interactive terminal connected in real-time to a computer. The computer controls the terminal interactively with program-generated instructions and questions transmitted to the terminal. Alternatively a second device could be used for display or printing of computer messages. The final report can be displayed or printed on the message destination device or may be directed to a line printer or other hard copy unit. This is the usual device used for the final report when a cathode ray tube is used as the input terminal.

<table>
<thead>
<tr>
<th>TABLE 1 Shape Factors—Convection Equations</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Shape and Condition</strong></td>
</tr>
<tr>
<td>Horizontal cylinders</td>
</tr>
<tr>
<td>Longer vertical cylinders</td>
</tr>
<tr>
<td>Vertical Plates</td>
</tr>
<tr>
<td>Horizontal plates, warmer than air, facing upward</td>
</tr>
<tr>
<td>Horizontal plates, warmer than air, facing downward</td>
</tr>
<tr>
<td>Horizontal plates, cooler than air, facing upward</td>
</tr>
<tr>
<td>Horizontal plates, cooler than air, facing downward</td>
</tr>
</tbody>
</table>
8.2 **Functional Description of Program**— The flow charts, shown in Figs. 5 and 6 are a schematic representation of the operational procedures of the respective programs. They show that logic paths for reading data, obtaining actual system dimensions, calculating and recalculating system thermal resistances and temperatures, relaxing the successive errors in

<table>
<thead>
<tr>
<th>Insulation Type</th>
<th>Functional Relationship Employed</th>
<th>Coefficients and Constants</th>
<th>Correlation Coefficient</th>
<th>F value</th>
<th>Standard Error of Estimates</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type 1 (Fig. 11)</td>
<td>$k = a + bt + cT$</td>
<td>$a = 0.400$; $b = 0.105 \times 10^{-3}$; $c = 0.286 \times 10^{-6}$</td>
<td>0.999</td>
<td>550</td>
<td>0.0049</td>
</tr>
<tr>
<td>Type 2 (Fig. 10)</td>
<td>$lnk = a + bt$</td>
<td>$a = -1.62$; $b = 0.213 \times 10^{-2}$</td>
<td>0.999</td>
<td>2130</td>
<td>0.0145</td>
</tr>
<tr>
<td>Type 3 (Fig. 12)</td>
<td>$k = a_1 + b_1T; t \leq TL$</td>
<td>$a_1 = 0.201$; $b_1 = 3 \times 10^{-3}$; $T = 0.999$</td>
<td>148</td>
<td>0.00165</td>
<td></td>
</tr>
<tr>
<td>Type 4 (Fig. 12)</td>
<td>$k = a_2 + b_2T; TL &lt; T &lt; TU$</td>
<td>$a_2 = 0.182$; $b_2 = -0.39 \times 10^{-3}$</td>
<td>0.997</td>
<td>187</td>
<td>0.00094</td>
</tr>
<tr>
<td>Type 5 (Fig. 12)</td>
<td>$k = a_3 + b_3T; T \geq TU$</td>
<td>$a_3 = 0.141$; $b_3 = 0.37 \times 10^{-3}$</td>
<td>0.993</td>
<td>69.3</td>
<td>0.00320</td>
</tr>
</tbody>
</table>

**TABLE 2 Regression Analysis of Sample Data for Examples 1 to 4**

---

![FIG. 5 Flow Diagram of the Computer Program C 680E for Insulated Equipment Systems](image)
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. The flow chart symbols are in accordance with ANSI X3.5.

8.3 Computer Program Variable Description—The description of all variables used in the programs are given in the listing of each program as comments. The listings of the mainline programs and the applicable subroutines are shown in Fig. 7Fig. 8Fig. 9.

8.4 Program Operation:

8.4.1 Logon procedures and any executive program for execution of this program must be followed as needed.

8.4.2 The input for the thermal conductivity versus mean temperature parameters is obtained as described in 7.3. (See the thermal curves depicted in Figs. 10-12.) 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 program will operate on the three functional relationships:
Type Code   Functional Relationship
1     \[ k = a + bt + ct^2 \] where \( a, b, \) and \( c \) are constants.
2     \[ k = e^{at^b} \] where \( a \) and \( b \) are constants and \( e \) is the base of the natural logarithm
3     \[ \begin{align*}
    k &= a_1 + b_1 t; \quad &T_L < t < T_U \\
    k &= a_2 + b_2 t; \quad &T_L < t < T_U \\
    k &= a_3 + b_3 t; \quad &t > T_U
\end{align*} \]
   \( a_1, a_2, a_3, b_1, b_2, b_3 \) are constants. \( T_L \) and \( T_U \) are, respectively, the lower and upper inflection points of an S-shaped curve.

Additional or different relationships may be programmed but require modifications to the program.

8.4.3 For multiple number entry in a free field format, all numbers must be separated by commas.

9. Illustration of Examples

9.1 General:

9.1.1 Four examples are presented to illustrate the utility of the program in calculating heat loss or gain and surface temperature. Most practical insulation design problems implicitly or explicitly call for such calculations. Three insulating materials, having equations forms for Types 1, 2, and 3, are considered. The fourth example illustrates a combination of these three materials.

Note 6—The curves contained herein are for illustration purposes only and not intended to reflect any actual product currently being produced.
9.1.2 Sample data relating thermal conductivity to mean temperature data for the three insulating materials are shown in Figs. 10-12. Least-square estimates of the regression curve for each sample data set produced a satisfactory fit to one of the program’s functional types. The information in Table 2 was obtained from the regression analysis (least-squares fit) on each material.

9.2 Example 1:

9.2.1 Consider application of a Type 2 insulation to the flat vertical surfaces of a piece of hot equipment. The operating temperature is 450°F (232°C). The equipment is located out-doors in an area where the winter design ambient temperature is 10°F (−12°C). Determine the insulation thickness required to maintain the heat losses below 35 Btu/h·ft² (110 W/m²).

9.2.2 Assuming the system faces virtually blackbody surroundings at the design ambient temperature, the surface coefficient may be obtained from the ASHRAE Handbook of Fundamentals (2). The value given for a nonreflective surface in a 15-mph (6.7-m/s) wind (winter) is 6.00 Btu/h·ft²·°F (34 W/m²·K).

9.2.3 From Table 2 for the material designated as Type Code 2, the two coefficients required for the equation are $a = -1.62$ and $b = 0.00213$. 

FIG. 7 (continued)
9.2.4 From past experience, it is estimated that the required thicknesses will fall in the range from 4.0 to 5.0 in. (101 to 127 mm). This range will be covered in increments of 1/2 in. (3 mm).

9.2.5 The resulting programing and analysis is given in Fig. 13 where 4.5 in. (114 mm) is the least thickness to maintain heat loss below 35 Btu/h·ft² (110 W/m²).

9.3 Example 2:

9.3.1 Determine the minimum nominal thickness of Type 1 pipe insulation required to maintain the surface temperature of a horizontal 3-in. (76-mm) iron pipe below 130°F (54°C). Consider a pipe temperature of 800°F (427°C). The ambient temperature is 80°F (26°C).

9.3.2 Assuming the piping is located in a large room with surrounding surfaces at ambient temperature and that the emissivity of the system is not significantly different from that of bare steel pipe (0.9), the surface coefficient could be estimated from the ASHRAE Handbook of Fundamentals (2).

Because the thicknesses to be chosen will provide a surface temperature about 50°F (28°C) above the 80°F (26°C) ambient, the 50° column is entered. The system diameter (insulation size) is not known since it will depend on the insulation thickness. For the first calculation, and the estimated insulation diameter, 9 in. (229 mm), 1.76 Btu/(h·ft²·°F) (10 W/m²·K), will be used. The thicknesses chosen as a result of the first calculation will provide a basis for reestimating the surface temperature. (continued)
coefficients. These can be refined if a more rigorous treatment of pipe temperature-thickness combinations that satisfy the surface temperature criterion is required.

9.3.3 Referring to Table 2, for the material designated as Type 1, the required constants for the thermal conductivity equations are:

- \( a = 0.400 \)
- \( b = 0.105 \times 10^{-3} \)
- \( c = 0.286 \times 10^{-6} \)

9.3.4 From experience, the nominal insulation thicknesses of 2, 2.5, and 3 in. (51, 64, and 76 mm) are estimated to include the range of solutions.

9.3.5 The solutions for this problem are given in Fig. 14 where 3.0 in. (76 mm) is shown to maintain a surface temperature below 130°F (54°C).

9.4 Example 3:

9.4.1 Example 3 is a repeat of Example 2 except that the internal surface coefficient routine in the program C 680P2 is used.

9.4.2 Assume the same ambient and operating conditions, but the program calculates the surface coefficient from a flow of 0 mph (0 m/s) and a surface emittance of 0.9 instead of choosing from a handbook.

9.4.3 The results of this analysis (Fig. 15) yield approximately the same answer as 9.3 and provide for more realistic
ambient input conditions and no time loss from interpolation of the reference tables.

9.5 Example 4—Multiple Layers:

9.5.1 Determine the heat loss and surface and interface temperatures of an insulated 4-in. (110-mm) pipe operating at 600°F (315°C), insulated with 3 in. (76 mm) of Type 1 material, 2-in. (51-mm) thick layer of Type 2 material and 11/2-in. (13-mm) thick layer of Type 3 material at an ambient temperature of − 100°F (−73°C). The wind speed is 5 mph (3.2 m/s) and surface emittance is 0.9.

9.5.2 Referring to Figs. 10-12, to obtain the material properties, the required constants are:

\[
\begin{align*}
  a_1 &= 0.201 \\
  b_1 &= 0.39 \times 10^{-3} \\
  a_2 &= 0.182 \\
  b_2 &= -0.39 \times 10^{-3} \\
  a_3 &= 0.141 \\
  b_3 &= 0.37 \times 10^{-3}
\end{align*}
\]

(a) Transition Temperatures for Type 3:
9.5.3 The interactive communication record and calculated results are shown in Fig. 16.

10. Report

10.1 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:

10.1.1 Name and other identification of products or components,

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

10.1.3 The surface temperature of the pipe or surface,

10.1.4 The equations and constants selected for the thermal conductivity versus mean temperature relationship,
FIG. 7 (continued)

C 680

0187  DO 251 J=1, NLAYER  C 680 325

0188  I=MAT(J)  C 680 326

0189  IF(INSK(1,10), GT, 2.5) GO TO 247  C 680 328

0191  IF(INSK(1,11), GT, 2.5) GO TO 245  C 680 329

0193  WRITE(IP, 244) INSK(1,2), INSK(1,3), INSK(1,4)  C 680 331

0194  244 FORMAT(1, A)  C 680 332

0195  * * T(4, T*)  C 680 333

0195  GO TO 251  C 680 334

0196  245 WRITE(IP, 244) INSK(1,2), INSK(1,3)  C 680 335

0197  246 FORMAT(1, * * T('4, T*))  C 680 337

0198  GO TO 251  C 680 338

0199  247 WRITE(IP, 249) INSK(1,2), INSK(1,3), INSK(1,4)  C 680 339

0200  248 FORMAT(1, * + ('1, T'))  C 680 341

0201  WRITE(IP, 249) INSK(1,5), INSK(1,6), INSK(1,7), INSK(1,8)  C 680 342

0202  249 FORMAT(1, * + ('1, T'))  C 680 344

0203  WRITE(IP, 250) INSK(1,8), INSK(1,9), INSK(1,10)  C 680 345

0204  250 FORMAT(1, * + ('1, T'))  C 680 347

0205  251 CONTINUE  C 680 348

0206  WRITE(IP, 254)(1)  C 680 349

0207  254 FORMAT(1, * + ('1, T'))  C 680 350

0208  WRITE(IP, 255)(1)  C 680 351

0209  255 FORMAT(1, * + ('1, T'))  C 680 352

0210  IF(MISS, LT, 0.0) GO TO 262  C 680 353

0211  WRITE(IP, 260)  C 680 354

0212  260 FORMAT(1, 'NO')  C 680 355

0213  WRITE(IP, 261)  C 680 356

0214  261 FORMAT(1, 'NO')  C 680 357

0215  WRITE(IP, 262)  C 680 358

0216  262 FORMAT(1, 'NO')  C 680 359

0217  WRITE(IP, 263)  C 680 360

0218  263 FORMAT(1, 'NO')  C 680 361

0219  WRITE(IP, 271)  C 680 362

0220  271 FORMAT(1, 'NO')  C 680 363

0221  WRITE(IP, 280)  C 680 364

0222  280 FORMAT(1, 'INSULATION CONDUCTIVITY, RESISTANCE')  C 680 365

0223  281 FORMAT(1, 'INSULATION CONDUCTIVITY, RESISTANCE')  C 680 366

0224  WRITE(IP, 281)  C 680 367

0225  281 FORMAT(1, 'INSULATION CONDUCTIVITY, RESISTANCE')  C 680 368

0226  WRITE(IP, 282)  C 680 369

0227  282 FORMAT(1, 'INSULATION CONDUCTIVITY, RESISTANCE')  C 680 370

0228  WRITE(IP, 283)  C 680 371

0229  283 FORMAT(1, 'INSULATION CONDUCTIVITY, RESISTANCE')  C 680 372

0224  DO 283 I=1, NLAYER  C 680 373

0225  WRITE(IP, 282)  C 680 374

0226  WRITE(IP, 281)  C 680 375

0227  WRITE(IP, 280)  C 680 376

0228  WRITE(IP, 283)  C 680 377

0229  WRITE(IP, 282)  C 680 378

FIG. 7 (continued)

C 680 379

C 680 380

C 680 381

C 680 382

C 680 383

C 680 384

C 680 385

C 680 386

C 680 387

C 680 388

C 680 389

C 680 390

FIG. 7 (continued)
10.1.5 The ambient temperature and humidity, if applicable,
10.1.6 The surface coefficient and condition of surface heat
transfer,
10.1.6.1 If obtained from published information, the source
and limitations,
10.1.6.2 If calculated or measured, the method and signifi-
cant parameters such as emittances, fluid velocity, etc.,
10.1.7 The resulting outer surface temperature, and
10.1.8 The resulting heat loss or gain.

10.2 Either tabular or graphical representation of the results
of the calculations may be used. No attempt is made to
recommend the format of this presentation of results.

11. Precision and Bias

11.1 The precision of this practice is a function of the
computer equipment used to generate the calculational results.
In many typical computers normally used, seven significant
digits are resident in the computer for calculations. Adjust-
ments 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, has been structured to provide a
resolution of 0.1 % for the typical expected levels of heat flux
and within 0.1°F (0.05°C) for surface temperatures.
11.2 Many factors influence the accuracy of a calculational procedure used for predicting heat flux results. These factors include computer resolution, 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.

11.3 The computer resolution effect on accuracy is only significant if the level of precision is less than that discussed in 11.1. Computers in use today are accurate in that they will reproduce the calculation results to the resolution required if identical input data is used.

11.4 The most significant factor influencing the accuracy 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 test data accuracy. 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 or the installation is poor, or both, serious differences might exist between measured performance and predicted performance from using this practice.

11.5 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 uncertainty. Repeating this procedure for all the input variables would yield a measure of the contribution of each to the overall 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 32 (8) gives the expression in mathematical form:

\[
S = \left( \sum_{i=1}^{n} \left( \frac{\Delta F_i}{F_i} \right)^2 \right)^{1/2}
\]
where:

\( S \) = estimate of the probable error of the procedure,
\( R \) = result of the procedure,
\( x_i \) = \( i \)th variable in procedure,
\( \Delta x_i \) = uncertainty in value of variable, \( i \), and
\( n \) = total number of variables in procedure.

11.6 In summary, the use of this system of equations in this practice for design and specification of insulation systems since 1971 has demonstrated the applicability and useful precision and bias expected when using 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 is beyond the primary C 680 scope. Portions of this discussion, however, were used in developing acceptance of the calculational procedures, the specific applicability should be defined for each insulation system installation at the time of its design.

11.7 Appendix X1 has been prepared by ASTM Subcommittee C16.30, Task Group 5.2, responsible for preparing this practice. The appendix provides a more complete discussion of the precision and bias expected when using C 680 in the analysis of operating systems.
the Precision and Bias statements included in Section 11.

12. Keywords

12.1 block; computer program; heat flow; heat gain; heat loss; pipe; thermal insulation

FIG. 8 (continued)
FIG. 8 (continued)
```
C 680
0193 WRITE(IW,231)I                   C680 325
0194 231 FORMAT(1800)                 C680 326
   +/'0' CONDUCTIVITY OF LAYER .13, ' IS LESS THAN 0.01'/'     C680 327
   +/'15', 'CHECK YOUR INPUT VALUES'/'201, 'PROGRAM TERMINATED'/'  C680 328
   +'*****************************************************'  C680 329
0195 GO TO 299                         C680 330
C 680 331
0196 232 R(I)=D(I)/DOUT(NLAYER)/2*DLOG(DOUT(I)/DIAGN(I))/K(I)  C680 332
0197 RSUM=RSUM+R(I)                     C680 333
0198 233 CONTINUE                       C680 334
C 680 335
0199 Q=T(I)-TAMB/RSUM                   C680 336
0200 TSUM=0                             C680 337
0201 DO 234 I=1,NLAYER                   C680 338
0202 TINT=T(I)-0.5R(I)                   C680 339
0203 TSUM=TSUM+ABS(T(I+1)-TINT)         C680 340
0204 T(I+1)=TINT                        C680 341
0205 234 CONTINUE                       C680 342
0206 IF (TSUM .GT. 0.1) GOTO 220         C680 343
0206 QLIFE=Q(3)*1.1599*DOUT(NLAYER)/12.  C680 344
C 680 345
C 680 346
C 680 347
C 680 348
C 680 349
C 680 350
C 680 351
0209 WRITE(IP,241)TITLE                 C680 352
0210 240 FORMAT(1800)                   C680 353
   +/'HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED PIPE SYSTEM/')  C680 354
C 680 355
0211 WRITE(IP,242)DATE                  C680 356
0212 241 FORMAT(1800)                   C680 357
C 680 358
0213 WRITE(IP,243)                      C680 359
0214 242 FORMAT(1800)                   C680 360
   +/'THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED')  C680 361
C 680 362
0215 WRITE(IP,243)                      C680 363
0216 243 FORMAT(1800)                   C680 364
   +*/ THIS ANALYSISAVED/')  C680 365
C 680 366
0217 DO 251 J=1,NLAYER                   C680 367
0218 I=IMAT(J)                          C680 368
C 680 369
0219 IF(INSK(I,1).GT.2.5) GO TO 247      C680 370
0221 IF(INSK(I,1).GT.1.5) GO TO 245      C680 371
C 680 372
0222 WRITE(IP,244)INSK(I,2),INSK(I,3),INSK(I,4)  C680 373
0224 244 FORMAT(1800)                   C680 374
C 680 376
0225 245 WRITE(IP,246)INSK(I,2),INSK(I,3)  C680 377
0227 246 FORMAT(1800)                   C680 378
   +/'TYPE 2 MATERIAL, K= EXP(3.74. +'E10 3. * T)/')  C680 379
```

FIG. 8 (continued)
FIG. 8 (continued)

0240 254 FORMAT(’PIPE SERVICE TEMPERATURE,F’,1X,F6.0) C680 399
0242 255 WRITE(12,PI3)TAMB C680 400
0243 255 FORMAT(’AMBIENT TEMPERATURE,F’,2X,F5.0) C680 401
C 0244 256 IF(EMISS LT 0.0) GO TO 262 C680 402
0246 256 WRITE(12,PI4)EMISS C680 404
0247 259 FORMAT(’EMITTANCE’,3X,F6.2) C680 405
0248 259 WRITE(12,PI5)WIND C680 406
C 0250 261 WRITE(12,PI6)SURFC C680 408
0250 262 WRITE(12,PI6)SURFC C680 409

FIG. 8 (continued)

0251 263 FORMAT(’SURFACE COEFFICIENT USED BTU/HR SF F’,7X,F6.2) C680 410
C 0252 270 WRITE(12,PI7)DOLFR C680 411
0253 271 FORMAT(’TOTAL HEAT FLUX, BTU/HR SF’,1X,D10.1) C680 412
C 0254 280 WRITE(12,PI8) C680 413
0255 280 FORMAT(’LAYER MATERIAL INSULATION CONDUCTIVITY, RESISTANCE’,C680 414
4X,TEMPERATURE, F’) C680 415
0256 281 WRITE(12,PI9) C680 416
0256 281 FORMAT(’NO. SIZE BTU/HR SF F HR SF F/BTC’,C680 417
4X) C680 418
0257 281 FORMAT(’INSIDE OUTSIDE’) C680 419
0258 282 WRITE(12,PI10)DO289 C680 420
0259 282 FORMAT(’DO 289 I=1,NLAYER’,C680 421
4X) C680 422
0259 282 WRITE(12,PI11)MAT(I),INSIZ(I),THK(I),R(I),T(I),T(I) C680 423
0260 282 WRITE(12,PI12)INSIZ(I),THK(I),R(I),T(I),T(I) C680 424
0261 283 CONTINUE C680 425
C 0262 283 CONTINUE C680 426
0262 290 FORMAT(’/ 290 /’ C680 427
0263 290 DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS?’ C680 428
*YES. /’ ENTER 0 FOR NO’/7X,’1C’ C680 429
0264 290 READ(12,PI13)ANS C680 430
0265 290 IF (ANS NE 0) GOTO 129 C680 431
C 0266 299 CALL EXIT C680 432
0267 299 CALL EXIT C680 433
0268 END C680 434
FIG. 9 Computer Listings—Support Subroutines: SURCOF-Surface Heat Flow Coefficient; KCURVE-Equivalent Thermal Conductivity; SELECT-Nesting Insulation Sizing for Pipes
FIG. 9 (continued)

C LAST REVISION MADE ON 11/13/81
C PROGRAM KURVE
C
C
0001 SUBROUTINE KURVE (NLAYER, MAT, INSK, T, K)
C
C THIS ROUTINE CALCULATES THE THERMAL CONDUCTIVITY OF EACH LAYER OF
C INSULATION USING THE MATERIAL K-CURVE PARAMETERS AND INNER AND
C OUTER TEMPERATURES. THE ROUTINE IS EMPLOYED SUCCESSIVELY AS INNER
C AND OUTER TEMPERATURES ARE RECOMPUTED UNTIL A STABLE THERMAL
C EQUILIBRIUM IS REACHED
C
C VARIABLES USED IS THIS ROUTINE-
C
C C = TEMPERATURE OF COLD SIDE OF INSULATION LAYER, F
C H = TEMPERATURE OF HOT SIDE OF INSULATION LAYER, F
C I = INDEX VARIABLE
C INSK(I,J) = INSULATION K-CURVE PARAMETER ARRAY
C K(I) = THERMAL CONDUCTIVITY, K, OF LAYER I
C MAT(I) = MATERIAL NO. OF LAYER I
C NLAYER = NUMBER OF INSULATION LAYERS
C T(I) = INNER TEMPERATURE OF LAYER I, F. THE OUTER
C TEMPERATURE OF LAYER I IS THE INNER TEMPERATURE
C OF THE NEXT LAYER
C TL = LOWER TEMPERATURE BOUND OF REGION II OF
C MATERIAL TYPE 3.
C TU = UPPER TEMPERATURE BOUND OF REGION II OF
C MATERIAL TYPE 3.
C
0002 DIMENSION T(I), MAT(7)
0003 REAL K(7), INSK(10,9)
C
0004 DO 510 J=1,NLAYER
0005 I=MAT(J)
C
0006 IF(INSK(I,1).GE.2.5) GO TO 502
0007 IF(INSK(I,1).GE.1.5) GO TO 501
C
0010 500 K(I)=INSK(I,2)+INSK(I,3)*((T(I)+T(J))/2.+INSK(I,4)*T(J)**3-
C *(T(J)**5))/(3*(T(I)-T(J)))
0011 GO TO 510
C
0012 501 K(I)=EXP(INSK(I,2)+INSK(I,3)*T(J))+EXP(INSK(I,2)+INSK(I,3)*T(J))
C *(T(I)**5))/(INSK(I,1)*T(I)+T(J))
0013 GO TO 510
C
0014 502 IF (T(J).GE.T(I)) GO TO 503
0015 H=T(I)
0016 C=T(J)
0017 C=T(J)
0018 GO TO 504
0019 503 H=T(I)
0020 C=T(I)
C
0021 504 TL=INSK(I,4)

FIG. 9 (continued)
FIG. 9 (continued)
FIG. 9 (continued)
FIG. 9 (continued)
FIG. 10 Sample Data—Type 2 Material

FIG. 11 Sample Data—Type 1 Material

FIG. 12 Sample Data—Type 3 Material
RUN EQUIP1

ASTM C-680 RECOMMENDED PRACTICE FOR THE DETERMINATION OF HEAT FLOW AND SURFACE TEMPERATURES OF MULTIPLE-LAYERED EQUIPMENT INSULATION SYSTEM FOR AN INTERACTIVE INPUT/OUTPUT COMPUTER TERMINAL.

ENTER TITLE - 60 CHARACTER LIMIT

SAMPLE PROBLEM 1
ENTER DATE - ANY FORMAT

NOVEMBER 24, 1981
ENTER AMBIENT TEMPERATURE, F

10
TYPICAL SURFACE COEFFICIENT IS 1.65.
IF COEFFICIENT IS TO BE CALCULATED FROM EMITTANCE AND WIND SPEED ENTER 0
OTHERWISE ENTER SURFACE COEFFICIENT TO BE USED.

6.00
UP TO 10 THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS MAY BE USED.
THEY ARE OF 3 TYPES. THE TYPES ARE:

MATERIAL CODE 1 = K = A + B * T + C * T**2
MATERIAL CODE 2 = K = EXP( A + B * T )
MATERIAL CODE 3 = K = A1 + B1 * T, FOR T < TL
                K = A2 + B2 * T, FOR TL < T < TU
                K = A3 + B3 * T, FOR TU < T

WHERE A, B, AND C ARE THE COEFFICIENTS OF THE EQUATIONS, AND T IS THE MEAN TEMPERATURE.
ENTER MATERIAL TYPE CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 1
2
ENTER A, B FOR MATERIAL CODE 2.
-1.62, 0.00213
ENTER MATERIAL TYPE CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 2
0
ENTER NUMBER OF INSULATION LAYERS - MAXIMUM OF 7
1
ENTER LAYER INFORMATION FROM THE EQUIPMENT SURFACE TO THE AMBIENT SURFACE
ENTER INSULATION NO. AND INSULATION THICKNESS FOR LAYER NO. 1
1, 1.4
ENTER EQUIPMENT SERVICE TEMPERATURE, F
450

FIG. 13 Sample Problem 1
SAMPLE PROBLEM 1

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED EQUIPMENT PER ASTM C-680

THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS:

\[ K = 600 \times 10^6 + 0.213 \times 10^{-2} \times T \]

EQUIPMENT SERVICE TEMPERATURE, F 450
AMBIENT TEMPERATURE, F 10

SURFACE COEF. USED: BTU/HR SF F 6.00

TOTAL HEAT FLUX, BTU/HR SF, 36.5

<table>
<thead>
<tr>
<th>LAYER NO.</th>
<th>MATERIAL</th>
<th>INSULATION</th>
<th>CONDUCTIVITY, BTU/HR SF F</th>
<th>RESISTANCE, HR SF F/BTU</th>
<th>TEMPERATURE, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td>4.00</td>
<td>0.337</td>
<td>11.88</td>
<td>450.00</td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION OR LAYER SCHEDULE.
ENTER 0 FOR NO
1 FOR YES

ENTER NUMBER OF INSULATION LAYERS – MAXIMUM OF 7
1

ENTER LAYER INFORMATION FROM THE EQUIPMENT SURFACE TO THE AMBIENT SURFACE

ENTER INSULATION NO. AND INSULATION THICKNESS FOR LAYER NO. 1
1: 4.5

ENTER EQUIPMENT SERVICE TEMPERATURE, F
450

FIG. 13 (continued)
SAMPLE PROBLEM 1

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED EQUIPMENT PER ASTM C-680

THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS.

TYPE 2 MATERIAL: \[ K = \exp(-1.6200 + 0.213E^{-02} \times T) \]

| EQUIPMENT SERVICE TEMPERATURE, \(^\circ\)F | 450.0 |
| AMBIENT TEMPERATURE, \(^\circ\)F | 10.0 |
| SURFACE COEF. USED, BTU/HR SF \(^\circ\)F | 6.00 |
| TOTAL HEAT FLUX, BTU/HR SF | 32.5 |

<table>
<thead>
<tr>
<th>LAYER</th>
<th>MATERIAL</th>
<th>INSULATION</th>
<th>CONDUCTIVITY, BTU IN/HR SF (^\circ)F</th>
<th>RESISTANCE, HR SF.F/ETU</th>
<th>TEMPERATURE, (^\circ)F</th>
<th>INSIDE</th>
<th>OUTSIDE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>4.50</td>
<td>0.337</td>
<td>13.37</td>
<td>450.00</td>
<td>15.92</td>
<td></td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION OR LAYER SCHEDULE.
ENTER 0 FOR NO
1 FOR YES

0

>
RUN PIPE2

ASTM C-680 RECOMMENDED PRACTICE FOR THE DETERMINATION OF HEAT FLOW AND SURFACE TEMPERATURES OF MULTIPLE-LAYERED INSULATED PIPE FOR AN INTERACTIVE INPUT/OUTPUT COMPUTER TERMINAL.

ENTER TITLE - 60 CHARACTER LIMIT

SAMPLE PROBLEM 2

ENTER AMBIENT TEMPERATURE, F

80

TYPICAL SURFACE COEFFICIENT IS 1.65.

If coefficient is to be calculated from emittance and wind speed enter 0 otherwise enter surface coefficient to be used.

1.76

Up to 10 thermal conductivity vs. mean temperature equations may be used.

They are of 3 types. The types are:

Material code 1 - \( K = A + B \times T + C \times T^2 \)

Material code 2 - \( K = \exp(A + B \times T) \)

Material code 3 - \( K = A_1 + B_1 \times T \), for \( T < T_L \)

\( K = A_2 + B_2 \times T \), for \( T_L < T < T_U \)

\( K = A_3 + B_3 \times T \), for \( T_U < T \)

Where \( A, B, \) and \( C \) are the coefficients of the equations, and \( T \) is the mean temperature.

Enter material type code (or 0 if all entered) for insulation no. 1

1

Enter \( A, B, C \) for material type 1.

0.0400, 0.105, -0.03, 0.285, -0.06

Enter material type code (or 0 if all entered) for insulation no. 2

0

Enter number of insulation layers - maximum is 7

1

Insulation thicknesses of 1 inch to 4 inches can be entered in increments of 0.5 inch.

Enter layer information from the pipe surface to the ambient surface

Enter insulation material no. and insulation thickness for layer no. 1

1, 2.0

Enter nominal pipe size per ASTM C-585

3.0

Enter pipe service temperature, F

800

FIG. 14 Sample Problem 2

34
SAMPLE PROBLEM I

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED PIPE SYSTEMS PER ASTM C-680

THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS:

TYPE I MATERIAL: X = 0.400 + 3.105E-03 + T + 0.356E-06 * T^2

| NOMINAL IRON PIPE SIZE, IN | 3.00 |
| ACTUAL PIPE DIAMETER, IN   | 3.500|
| PIPE SERVICE TEMPERATURE, F | 800  |
| AMBIENT TEMPERATURE, F     | 80.  |
| SURFACE COEFFICIENT (BTU/HR-FT-F) | 1.5 |
| TOTAL HEAT FLOW, BTU/HR-LF | 220.3|

<table>
<thead>
<tr>
<th>LAYER NO</th>
<th>MATERIAL</th>
<th>INSULATION CONDUCTIVITY</th>
<th>RESISTANCE</th>
<th>TEMPERATURE, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>3.00 1.20</td>
<td>0.524</td>
<td>9.67</td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION, OR LAYER SCHEDULE?
ENTER 0 FOR NO
1 FOR YES.

1

ENTER NUMBER OF INSULATION LAYERS - MAXIMUM IS 7

1

INSULATION THICKNESSES OF 1 INCH TO 4 INCHES CAN BE ENTERED IN INCREMENTS OF 0.5 INCH.

ENTER LAYER INFORMATION FROM THE PIPE SURFACE TO THE AMBIENT SURFACE

ENTER INSULATION MATERIAL NO. AND INSULATION THICKNESS FOR LAYER NO. 1

1 1 2.5

ENTER NOMINAL PIPE SIZE PER ASTM C-680

3.0

ENTER PIPE SERVICE TEMPERATURE, F

800

FIG. 14 (continued)
C 680

SAMPLE PROBLEM 2

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED PIPE SYSTEMS PER ASTM C-680

THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS

<table>
<thead>
<tr>
<th>TYPE</th>
<th>MATERIAL</th>
<th>( R = 0.400 + 0.105e-03 \times T + 0.296e-06 \times T^2 )</th>
</tr>
</thead>
</table>

| NOMINAL IRON PIPE SIZE, IN | 3.00 |
| ACTUAL PIPE DIAMETER, IN  | 3.300 |
| PIPE SERVICE TEMPERATURE, F | 800 |
| AMBIENT TEMPERATURE, F     | 80 |
| SURFACE COEFFICIENT USED BTU/HR SF F | 1.76 |

TOTAL HEAT FLOW, BTU/HR LF. | 202.5 |

<table>
<thead>
<tr>
<th>LAYER</th>
<th>MATERIAL</th>
<th>INSULATION CONDUCTIVITY, BTU IN/HR SF F</th>
<th>RESISTANCE, HR SF F/ BTU</th>
<th>TEMPERATURE, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO</td>
<td>NO.</td>
<td>SIZE</td>
<td>BTU IN/HR SF F</td>
<td>HR SF F/ BTU</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>3.00 X 2.50</td>
<td>0.522</td>
<td>7.46</td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION, OR LAYER SCHEDULE?

ENTER 0 FOR NO
1 FOR YES.

1

ENTER NUMBER OF INSULATION LAYERS - MAXIMUM IS 7

1

INSULATION THICKNESSES OF 1 INCH TO 4 INCHES CAN BE ENTERED IN INCREMENTS OF 0.1 INCH.

ENTER LAYER INFORMATION FROM THE PIPE SURFACE TO THE AMBIENT SURFACE

ENTER INSULATION MATERIAL NO. AND INSULATION THICKNESS FOR LAYER NO. 1

1 x 3.0

ENTER NOMINAL PIPE SIZE PER ASTM C-585

3.0

ENTER PIPE SERVICE TEMPERATURE, F

800

FIG. 14 (continued)
SAMPLE PROBLEM 2

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED PIPE SYSTEMS PER ASTM C-680

THERMAL CONDUCTIVITY VS MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS:

\[ K = 0.400 + 0.100E-03 \times T + 0.286E-06 \times T^{1/2} \]

| NOMINAL IRON PIPE SIZE, IN. | 3.00 |
| ACTUAL PIPE DIAMETER, IN. | 3.500 |
| PIPE SERVICE TEMPERATURE, F | 800. |
| AMBIENT TEMPERATURE, F | 80. |
| EMITTANCE | 0.99 |
| WIND SPEED, MPH | 0.0 |
| SURFACE COEFFICIENT USED, BTU/HR SF F | 1.76 |
| TOTAL HEAT FLOW, BTU/HR LF. | 102.7 |

<table>
<thead>
<tr>
<th>LAYER NO.</th>
<th>MATERIAL</th>
<th>INSULATION THICKNESS, IN.</th>
<th>CONDUCTIVITY, BTU/HR SF F</th>
<th>RESISTANCE, HRF</th>
<th>TEMPERATURE, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>3.00 1.300</td>
<td>0.520</td>
<td>9.76</td>
<td>800.00 121.24</td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION, OR LAYER SCHEDULE?
ENTER 0 FOR NO
1 FOR YES.

1

ENTER NUMBER OF INSULATION LAYERS - MAXIMUM IS 7

1

INSULATION THICKNESSES OF 1 INCH TO 4 INCHES CAN BE ENTERED IN INCREMENTS OF 0.5 INCH.
ENTER LAYER INFORMATION FROM THE PIPE SURFACE TO THE AMBIENT SURFACE

ENTER INSULATION MATERIAL NO. AND INSULATION THICKNESS FOR LAYER NO. 1
1    3.5

ENTER NOMINAL PIPE SIZE PER ASTM C-585
3.5

ENTER PIPE SERVICE TEMPERATURE, F
800

FIG. 14 (continued)
RUN PIPE

ASTM C-680 RECOMMENDED PRACTICE FOR THE DETERMINATION OF HEAT FLOW AND SURFACE TEMPERATURES OF MULTIPLE-LAYERED INSULATED PIPE FOR AN INTERACTIVE INPUT/OUTPUT COMPUTER TERMINAL.

ENTER TITLE - 60 CHARACTER LIMIT

SAMPLE PROBLEM 3
ENTER DATE - ANY FORMAT

NOVEMBER 24, 1991
ENTER AMBIENT TEMPERATURE, F
80
TYPICAL SURFACE COEFFICIENT IS 1.65.
IF COEFFICIENT IS TO BE CALCULATED FROM EMITTANCE AND WIND SPEED ENTER 0
OTHERWISE ENTER SURFACE COEFFICIENT TO BE USED.
0
TYPICAL EMITTANCE IS 0.9.
TYPICAL WIND SPEED IS 0 MPH.
ENTER EMITTANCE, WIND SPEED, AND PIPE ORIENTATION CODE:
1 FOR VERTICAL PIPE RUN
2 FOR HORIZONTAL PIPE RUN
0.9*0.02
SIGNIFICANT SYSTEM DIMENSION (VERTICAL HEIGHT, AVERAGE HORIZONTAL DIMENSION,
OR INSULATION SURFACE DIAMETER); IF UNKNOWN ENTER 0.
0.75
UP TO 10 THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS MAY BE USED.
THEY ARE OF 3 TYPES. THE TYPES ARE:
MATERIAL CODE 1 - K = A + B * T + C * T**2
MATERIAL CODE 2 - K = EXP(A + B * T)
MATERIAL CODE 3 - K = A1 + B1 * T, FOR T < TL
K = A2 + B2 * T, FOR TL < T < TU
K = A3 + B3 * T, FOR TU < T
WHERE A, B, AND C ARE THE COEFFICIENTS OF THE EQUATIONS, AND T IS THE MEAN TEMPERATURE.
ENTER MATERIAL CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 1
1
ENTER A, B, C FOR MATERIAL TYPE 1.
0.400*0.105E-03+0.296E-06
ENTER MATERIAL TYPE CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 2
0
ENTER NUMBER OF INSULATION LAYERS - MAXIMUM IS 7
1
INSULATION THICKNESSES OF 1 INCH TO 4 INCHES CAN BE ENTERED IN INCREMENTS OF 0.5
INCH.
ENTER LAYER INFORMATION FROM THE PIPE SURFACE TO THE AMBIENT SURFACE
ENTER INSULATION MATERIAL NO. AND INSULATION THICKNESS FOR LAYER NO. 1
1:2.0
ENTER NOMINAL PIPE SIZE PER ASTM C-585
3.0
ENTER PIPE SERVICE TEMPERATURE, F
500

FIG. 15 Sample Problem 3
SAMPLE PROBLEM 3

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED PIPE SYSTEMS PER ASTM C-680

THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS:

\[ k = 0.400 + 0.102 \times 10^{-3} \times T + 0.28 \times 10^{-6} \times T^2 \]

| NOMINAL IRON PIPE SIZE, IN. | 3.00 |
| ACTUAL PIPE DIAMETER, IN. | 3.500 |
| PIPE SERVICE TEMPERATURE, F | 800 |
| AMBIENT TEMPERATURE, F | 80 |
| EMITTANCE | 0.90 |
| WIND SPEED, MPH | 0.0 |
| SURFACE COEFFICIENT USED, BTU/HR SF | 1.92 |
| TOTAL HEAT FLUX, BTU/HR LF. | 231.9 |

<table>
<thead>
<tr>
<th>LAYER NO.</th>
<th>INSULATION MATERIAL NO.</th>
<th>INSULATION THICKNESS, IN.</th>
<th>CONDUCTIVITY, BTU IN/HR SF F</th>
<th>RESISTANCE, HR SF F/BTU</th>
<th>INSIDE TEMPERATURE, F</th>
<th>OUTSIDE TEMPERATURE, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>3.00 X 2.00</td>
<td>0.523</td>
<td>5.68</td>
<td>800.00</td>
<td>140.47</td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION, OR LAYER SCHEDULE?
ENTER 0 FOR NO
1 FOR YES.

1

ENTER NUMBER OF INSULATION LAYERS – MAXIMUM IS 7

1

INSULATION THICKNESSES OF 1 INCH TO 4 INCHES CAN BE ENTERED IN INCREMENTS OF 0.1 INCH.

ENTER LAYER INFORMATION FROM THE PIPE SURFACE TO THE AMBIENT SURFACE

ENTER INSULATION MATERIAL NO. AND INSULATION THICKNESS FOR LAYER NO. 1

1 x 2.5

ENTER NOMINAL PIPE SIZE PER ASTM C-585

3.0

ENTER PIPE SERVICE TEMPERATURE, F

800

FIG. 15 (continued)
Sample Problem 3

November 24, 1981

Heat flow and surface temperatures of insulated pipe systems per ASTM C-680

Thermal conductivity vs. mean temperature equations used in this analysis:

Type 1 Material: \( K = 0.400 + 0.105E^{-0.2} + 7 + 0.286E^{-0.06 \cdot T^2} \)

| Nominal Iron Pipe Size, in. | 3.00 |
| Actual Pipe Diameter, in. | 3.500 |
| Pipe Service Temperature, F | 800.0 |
| Ambient Temperature, F | 80.0 |
| Emittance | 0.90 |
| Wind Speed, MPH | 0.0 |
| Surface Coefficient Used, BTU/hr SF F | 1.83 |
| Total Heat Flux, BTU/hr LF | 200.0 |

<table>
<thead>
<tr>
<th>Layer No.</th>
<th>Material</th>
<th>Insulation Conductivity, BTU/hr SF F</th>
<th>Resistance, HR SF F/BTU</th>
<th>Temperature, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>2.00 x 2.50</td>
<td>0.121</td>
<td>7.46</td>
</tr>
</tbody>
</table>

Do you want to re-run this program with a different thickness, insulation, or layer schedule?
Enter 0 for no. 1 for yes.

Enter number of insulation layers - maximum is 7
1
Insulation thicknesses of 1 inch to 4 inches can be entered in increments of 0.5 inch.
Enter layer information from the pipe surface to the ambient surface.

Enter insulation material no. and insulation thickness for layer no. 1
1.0
Enter nominal pipe size per ASTM C-680
3.5
Enter pipe service temperature, F
800.0

Fig. 15 (continued)
SAMPLE PROBLEM 2

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED PIPE SYSTEMS PER ASTM C-680

THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS:

TYPE 1 MATERIAL: \( k = 0.400 + 0.105E^{-03} \times T + 0.236E^{-06} \times T^2 \)

| NOMINAL IRON PIPE SIZE, IN. | 3.00 |
| ACTUAL PIPE DIAMETER, IN.   | 3.500 |
| PIPE SERVICE TEMPERATURE, F | 300.0 |
| AMBIENT TEMPERATURE, F      | 90.0 |
| SURFACE COEFFICIENT USED, BTU/HR. SF. F | 1.76 |
| TOTAL HEAT FLUX, BTU/HR. LF. | 182.7 |

<table>
<thead>
<tr>
<th>LAYER NO.</th>
<th>MATERIAL</th>
<th>INSULATION SIZE</th>
<th>CONDUCTIVITY, BTU IN/HR. SF. F</th>
<th>RESISTANCE, HR. SF. F/BTU</th>
<th>TEMPERATURE, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>3.00 X 3.00</td>
<td>0.520</td>
<td>9.26</td>
<td>800.00</td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION, OR LAYER SCHEDULE?
ENTER 0 FOR NO
1 FOR YES.

0

FIG. 15 (continued)
SAMPLE PROBLEM 3

NOVEMBER 24, 1981

HEAT FLOW AND SURFACE TEMPERATURES OF INSULATED PIPE SYSTEMS PER ASTM C-680

THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS USED IN THIS ANALYSIS:

TYPE 1 MATERIAL: \( K = 0.400 + 0.1055E^{-03} \times T + 0.286E^{-06} \times T^2 \)

<table>
<thead>
<tr>
<th>NOMINAL IRON PIPE SIZE, IN.</th>
<th>3.00</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACTUAL PIPE DIAMETER, IN.</td>
<td>3.500</td>
</tr>
<tr>
<td>PIPE SERVICE TEMPERATURE, F</td>
<td>800</td>
</tr>
<tr>
<td>AMBIENT TEMPERATURE, F</td>
<td>80</td>
</tr>
</tbody>
</table>

EMITTANCE 0.90
MIND SPEED, MPH 0.0
SURFACE COEFFICIENT USED, BTU/HR. SF. F 1.70

TOTAL HEAT FLUX, BTU/HR. LF. 166.0

<table>
<thead>
<tr>
<th>LAYER</th>
<th>MATERIAL</th>
<th>INSULATION</th>
<th>CONDUCTIVITY</th>
<th>RESISTANCE</th>
<th>TEMPERATURE, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO.</td>
<td>NO.</td>
<td>SIZE</td>
<td>BTU/HR. SF. F</td>
<td>HR SF. F/BTU</td>
<td>INSIDE</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>OUTSIDE</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>3.00 x 3.50</td>
<td>0.519</td>
<td>11.62</td>
<td>800.00</td>
</tr>
</tbody>
</table>

DO YOU WANT TO RE-RUN THIS PROGRAM WITH A DIFFERENT THICKNESS, INSULATION, OR LAYER SCHEDULE?
ENTER 0 FOR NO
1 FOR YES.

0

FIG. 15 (continued)
RUN PIPE

ASTM C-680 RECOMMENDED PRACTICE FOR THE DETERMINATION OF HEAT FLOW AND SURFACE TEMPERATURES OF MULTIPLE-LAYERED INSULATED PIPE FOR AN INTERACTIVE INPUT/OUTPUT COMPUTER TERMINAL.

ENTER TITLE - 60 CHARACTER LIMIT

SAMPLE PROBLEM 4
ENTER DATE - ANY FORMAT

NOVEMBER 24, 1981
ENTER AMBIENT TEMPERATURE, F
-100.0
TYPICAL SURFACE COEFFICIENT IS 1.65.
IF COEFFICIENT IS TO BE CALCULATED FROM EMITTANCE AND WIND SPEED ENTER 0
OTHERWISE ENTER SURFACE COEFFICIENT TO BE USED.
0
TYPICAL EMITTANCE IS 0.9.
TYPICAL WIND SPEED IS 0 MPH.
Enter EMITTANCE, WIND SPEED, AND PIPE ORIENTATION CODE:
1 FOR VERTICAL PIPE RUN
2 FOR HORIZONTAL PIPE RUN
0.9*5.0*2
SIGNIFICANT SYSTEM DIMENSION (VERTICAL HEIGHT, AVERAGE HORIZONTAL DIMENSION, OR INSULATION SURFACE DIAMETER); IF UNKNOWN ENTER 0.
0
UP TO 10 THERMAL CONDUCTIVITY VS. MEAN TEMPERATURE EQUATIONS MAY BE USED.
THEY ARE OF 3 TYPES. THE TYPES ARE:
MATERIAL CODE 1 - K = A + B * T + C * T^2
MATERIAL CODE 2 - K = EXP( A + B * T )
MATERIAL CODE 3 - K = A1 + B1 * T, FOR T < TL
K = A2 + B2 * T, FOR TL < T < TU
K = A3 + B3 * T, FOR TU < T
WHERE A, B, AND C ARE THE COEFFICIENTS OF THE EQUATIONS, AND T IS THE MEAN TEMPERATURE.
ENTER MATERIAL TYPE CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 1
1
ENTER A, B, C FOR MATERIAL TYPE 1.
0.400+0.105E-03+0.286E-06
ENTER MATERIAL TYPE CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 2
2
ENTER A, B FOR MATERIAL CODE 2.
-1.52+2.12E-03
ENTER MATERIAL TYPE CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 3
3
FOR MATERIAL TYPE 3:
ENTER A1, B1, TL
0.301+0.00039,-25.0
ENTER A2, B2, TU
0.182,-0.00038,50.0
ENTER A3, B3
0.141+0.00037
ENTER MATERIAL TYPE CODE (OR 0 IF ALL ENTERED) FOR INSULATION NO. 4

FIG. 16 Sample Problem 4
APPENDIX

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 11.

X1.2 This appendix will consider precision and bias as it relates to the comparison between the calculated results of the C 680 analysis and measurements on operating systems. Some of the discussion here may also be found in Section 11; 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 practice 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. The Rice Heilman adjustments (7) to the Langmuir’s equations (6) is one commonly used equation set. If two different equations 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 coefficient equation set used in this practice has had at least one thorough intralaboratory evaluation (9). When the surface convective coefficient equation (see Eq 30) 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. Eq 30 is widely used and accepted and will continue to be recommended until evidence suggests otherwise.

X1.3.4 Precision of Radiation Surface Equations:

X1.3.4.1 The Stefan-Boltzman equation for radiant transfer is widely applied, but still debated. 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 could 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 of 0.95 versus 1.0 shows 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 C680 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 test data accuracy. The remaining factors influencing the accuracy are the inherent variability of the product and the variability of the 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 temperatures. 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:

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 610°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:

The average radiative heat loss rate corresponding to a 610°F temperature is 93.9 Btu/ft²/ h.

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/hr/ft². The error in assuming the averaged surface temperature when applied to the radiative heat loss for this case is 8.6%.

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 C680 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 this 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 C680 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 the new effective surface emittance will correct for this error:

\[
\varepsilon_{\text{eff}} = \frac{A_A}{(1 - \varepsilon_A)\varepsilon_A A_A + 1/A_B F_{AB} + (1 - \varepsilon_B)\varepsilon_B A_B}
\]  

where:
- \( \varepsilon_{\text{eff}} \) = effective mean emittance of the two surface combination,
- \( \varepsilon_A \) = mean emittance of the surface \( A \),
- \( F_{AB} \) = view factor for the surface \( A \) and the surrounding region \( B \),
- \( \varepsilon_B \) = mean emittance of 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 radiative heat transfer. See Holman (4), p. 305.

X1.3.5.7 Wind Speed—Wind speed, as used in the Langmuir’s (6) and Rice Heilman (7) equations, 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 C680 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).

\[
\frac{S}{R} = \sum_{i=1}^{n} \frac{\partial R}{\partial x_i} \cdot \Delta x_i
\]  

where:
- \( S \) = estimate of the probable error of the procedure,
- \( R \) = result of the procedure,
- \( x_i \) = \( i \)th variable of the procedure,
- \( \partial R/\partial x_i \) = change in result with respect to a change in the \( i \)th variable (also, the first derivative of the function with respect to the \( i \)th variable),
- \( \Delta x_i \) = 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 the following equation:

\[ S = \left( \sum_{i=1}^{n} \left( \frac{\partial R}{\partial x_i} \cdot \Delta x_i \right)^2 \right)^{1/2} \]  

(X1.3)

X1.3.7 Bias of C680 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 are 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 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 C680 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 that 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.

REFERENCES