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Informal Report

**DESIGN PREDICTIONS AND DIAGNOSTIC TEST METHODS
FOR HYDRONIC HEATING SYSTEMS IN ASHRAE STANDARD 152P**

J. W. ANDREWS

APRIL 1996

**Prepared for:
Office of Building Technologies,
State and Community Programs
Building Equipment Division
U.S. Department of Energy
Washington, DC 20585**

MASTER

**Energy Efficiency
and Conservation Division**

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Building Equipment Division
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Under Contract No. DE-AC02-76CH00016**

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Upton, New York 11973**

ABSTRACT

A new method of test for residential thermal distribution efficiency is currently being developed under the auspices of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE). The initial version of this test method is expected to have two main approaches, or "pathways," designated Design and Diagnostic. The Design Pathway will use builder's information to predict thermal distribution efficiency in new construction. The Diagnostic Pathway will use simple tests to evaluate thermal distribution efficiency in a completed house. Both forced-air and hydronic systems are included in the test method. This report describes an approach to predicting and measuring thermal distribution efficiency for residential hydronic heating systems, for use in the Design and Diagnostic Pathways of the test method. As written, it is designed for single-loop systems with any type of passive radiation/convection (baseboard or radiators). Multiloop capability may be added later.

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INTRODUCTION

A new method of test for residential thermal distribution efficiency is being developed under the auspices of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE). Titled "Method of Test for Steady-State and Seasonal Efficiency of Residential Thermal Distribution Systems," its ASHRAE numerical designation will be Standard 152. A draft version of the standard has been prepared by the ASHRAE committee responsible for its development. This report describes the sections of the standard that pertain to hydronic heating systems.

Pathways in Standard 152

The initial version of Standard 152 is expected to have two main approaches, or "pathways," designated Design and Diagnostic. The Design Pathway is intended to be used before the building is constructed. It uses parameters specified by the builder (including manufacturer's data) as inputs to the calculations. In some cases, default values may be used in the absence of builder's specs. The Diagnostic Pathway is intended to be used by "house doctors" and other weatherization professionals who are determining what energy-saving retrofits will be most cost-effective to do in a particular house. To the extent possible, the diagnostic pathway calculations will parallel those in the Design Pathway, but they will use measured values of quantities instead of design specifications, to the extent that these can be obtained quickly enough. In general, the criterion for the Level 2 forced-air diagnostic, which is intended to add no more than 45 minutes to a "house doctor" protocol, should be our target here as well.

Thermal Distribution Efficiency Definitions

There are two separate definitions of thermal distribution efficiency (Modera et al. 1992, Andrews 1994). These are generic definitions that apply both to air duct and to hydronic systems.

Delivery efficiency is defined as the following ratio: heat delivered to the conditioned space divided by heat input to the distribution system. Delivery efficiency can be defined on a steady-state or a seasonal basis. Because the thermal losses are expected to be a small fraction of the delivered heat, it may be useful to recast the definition in terms of the losses:

$$\text{Delivery Efficiency} = \frac{1}{1 + \frac{\text{Losses}}{\text{Delivered Heat}}} \quad (1)$$

Delivery efficiency is strictly an output-to-input ratio. It does not account for any impacts the hydronic system may have on the rest of the building. Distribution efficiency goes beyond delivery efficiency to account for system impacts. It is defined as the following ratio: the input fuel energy required to heat the house if the distribution system had no energy losses and no impacts on the boiler efficiency or building load, divided by the actual fuel energy required to heat the house. The

proposed standard uses an equipment factor and a load factor to convert delivery efficiency to distribution efficiency.

APPROACH

In order to compute the losses and the delivered heat, we will use a lumped-parameter resistance-capacitance approach. The distribution system is divided into three elements:

- Finned radiation (all of which is assumed to be in the conditioned space);
- Unfinned piping in the conditioned space;
- Unfinned piping in unconditioned space.

For sake of mathematical convenience, the third element will later be split into two parts, one comprising that portion of the piping in unconditioned spaces that is not insulated, the other that which is insulated. These four pieces of the system will be called “categories” in the proposed standard.

In line with hydronics industry terminology, the word “radiation” is used to mean the finned-tube baseboard or free-standing radiators that are used to transfer heat from the distribution system to the conditioned space. In cases where radiant heat energy is meant, the words “thermal radiation” or “radiant heat” will be used. The unconditioned space in which portions of the distribution system are located is sometimes referred to as the buffer space. Typically, hydronic piping wends its way between a basement and the living space; in this case the buffer space is the basement.

Each of the above categories of piping or radiation will be characterized by a thermal capacitance and one or more thermal resistances. Piping and radiation in the conditioned space are assumed to be in communication with the outside (by virtue of being located adjacent to a wall) as well as with the conditioned space. Piping in the buffer space is assumed to be in communication only with the space in which it is located. The following quantities are therefore needed to characterize the thermal behavior of the system:

Symbol	Definition
R_{ra}	Thermal resistance from radiation to the ambient
R_{rc}	Thermal resistance from radiation to the conditioned space
C_r	Thermal capacitance of radiation
R_{ua}	Thermal resistance from unfinned piping to outside ambient
R_{uc}	Thermal resistance from unfinned piping to conditioned space
C_u	Thermal capacitance of unfinned piping in conditioned space
$R_{b,unins}$	Thermal resistance from uninsulated piping to buffer space
$C_{b,unins}$	Thermal capacitance of uninsulated piping in buffer space
$R_{b,ins}$	Thermal resistance from insulated piping to buffer space
$C_{b,ins}$	Thermal capacitance of insulated piping in buffer space

Numerical values for any specific system will need to be calculated from the physical characteristics of the piping and radiation and, in the case of resistances to the ambient, the insulation level in the wall behind the radiation. Additional input quantities needed are as follows:

T_{boiler}	Nominal or average boiler-water temperature (equals sendout water temperature to the distribution loop)
$T_{\text{buff-design}}$	Buffer space temperature under design conditions
$T_{\text{buff-seasonal}}$	Seasonal average buffer space temperature
T_{in}	Temperature of the conditioned space
L_{design}	Design heating load of the house
L_{seasonal}	Seasonal average heating load of the house
$t_{\text{cycle-design}}$	Circulator cycle time (on+off) under design conditions
$t_{\text{cycle-seasonal}}$	Circulator cycle time (on+off) under seasonal-average conditions
Cv_{water}	Volume heat capacity of distribution fluid (usually water)
V_c	Volume flow rate of water from boiler into distribution loop

Temperatures are expressed in degrees F or C (temperature differences are in F or K), while heating loads are measured in Btu/h or W. Water flow is measured in ft³/h or m³/s. The volume heat capacity of water at typical distribution temperatures is 61 Btu/ft³-F or 4.1 X 10⁶ J/m³-K.

Still to be discussed are how we obtain the values of these quantities. The procedure in the Design Pathway will need to include reasonable means of evaluating the parameters given above for the system to be installed. For the Diagnostic Pathway, methods of measuring or estimating these quantities relatively quickly will need to be developed. Part of the development of this pathway will be a determination of which quantities need to be measured (and how accurately) and which can safely be estimated (and by what procedure). For now, let us assume that we have them and can go ahead with delivery efficiency calculations.

CALCULATION PROCEDURE

The procedure will be first to calculate the steady-state delivery efficiency and then move on to estimate the design and seasonal delivery efficiencies. In order to calculate the latter two quantities, use is made of the fact that the most efficient mode of operation is steady state, while heat transfer with the circulator off is less efficient. That is because the temperature of the radiation drops at a faster rate than that of the unfinned piping. Therefore, at any time during the cooldown (off-cycle), heat transfer from the radiation, which is almost all useful heat, is a smaller fraction of its steady-state value than is the case for heat transfer from the unfinned piping, a greater fraction of which is lost.

Step 1. Calculate UA Values

The overall heat transfer coefficients or UA values for the three portions of the distribution system, as defined above, must be calculated. This is done as follows:

Value	Expression	Definition
UA_r	$1/R_{ra} + 1/R_{rc}$	Total UA value of radiation
UA_u	$1/R_{ua} + 1/R_{uc}$	Total UA value of piping in conditioned space
UA_b	$1/R_{b,unins} + 1/R_{b,ins}$	Total UA value of piping in buffer space

Step 2. Estimate Steady-State Water-Temperature Drop Through System

An estimate of the return-water temperature to the boiler inlet is obtained using the above UA values and the ΔT between the boiler outlet and the conditioned space:

$$T_{end} = T_{in} + (T_{boiler} - T_{in}) e^{-x} \quad (2)$$

where $UA = UA_r + UA_u + UA_b$, and $x = UA / (Cv_{water} V_c)$. The log-mean temperature difference between the circulating water and the conditioned space is given by:

$$\Delta T_{ln} = (T_{boiler} - T_{in}) \frac{1 - e^{-x}}{x} \quad (3)$$

The derivation of Equation 3 is as follows. It is assumed that the heat exchange is between flowing water within the pipe and a constant-temperature heat sink outside, where for simplicity the constant temperature is taken to equal T_{in} . (Although this is not strictly true, it should be nearly so since most of the UA is within the conditioned space, and moreover the basement-to-water temperature difference is generally not greatly different, percentagewise, from the inside-to-water temperature difference.) Referring to Kays and London, *Compact Heat Exchangers* (1964), in this case it is like an evaporator or condenser so that Kays and London's Equation 2-13a for the effectiveness applies: $\epsilon = 1 - \exp(-Ntu)$, where Ntu is the number of heat transfer units AU_{av}/C_{min} , or what we have called x for simplicity. Since effectiveness is the actual heat transfer over the maximum possible, $\epsilon = (T_{boiler} - T_{end}) / (T_{boiler} - T_{in})$. Eliminating the effectiveness between these two relations yields the relation $(T_{boiler} - T_{end}) / (T_{boiler} - T_{in}) = 1 - \exp(-x)$. The definition of log-mean temperature difference is $Q = UA \Delta T_{ln}$, where Q is the heat transfer rate, but an energy balance on the entire system yields a second relation $Q = Cv_{water} V_c (T_{boiler} - T_{end})$, so with the above definition of x we obtain $\Delta T_{ln} = (T_{boiler} - T_{end}) / x$. Combining this with the equation developed in the previous paragraph yields Equation 3.

Step 3. Calculate steady-state heat transfer rates

Steady-state heat transfer rates, to the conditioned space, buffer space, and outside ambient, respectively, are expressed in energy per unit time (Btu/h or W).

$$\begin{aligned}
Q_c &= \Delta T_{ln} \left(\frac{1}{R_{rc}} + \frac{1}{R_{uc}} \right) \\
Q_a &= \Delta T_{ln} \left(\frac{1}{R_{ra}} + \frac{1}{R_{ua}} \right) \\
Q_b &= [\Delta T_{ln} + (T_{in} - T_{buff-design})] \left(\frac{1}{R_{b,unins}} + \frac{1}{R_{b,ins}} \right)
\end{aligned} \tag{4}$$

In the case of Q_a , only the temperature difference between the hydronic system and the indoors is used, not the larger one between the hydronic system and the outside. This is because the portion of the heat flow due to the temperature difference between the indoors and the outside represents through-the-wall heat transfer that would take place even in the absence of a hot-water system, so it isn't fair to charge the hot-water system with that part of the heat loss. By contrast, in the case of Q_b , the entire temperature difference between the pipe and the surrounding space is used, because the pipes are responsible for all the heat transfer from them to the buffer space.

Step 4. Calculate the steady-state delivery efficiency

This is done using the formula derived above, with the values of the Q 's calculated in Step 3.

$$\eta_{del-ss} = \frac{1}{1 + \frac{Q_a + Q_b}{Q_c}} \tag{5}$$

Step 5. Calculate cycle lengths for design and seasonal conditions

It is assumed that the hydronic system will be somewhat oversized for the load; hence under design conditions there will still be some cycling. The cycle length will affect delivery efficiency because depending on the number of cycles per hour there will be differing numbers of cooldown periods. A rationale for choosing appropriate cycle lengths is developed in Appendix 1. A spreadsheet study (Appendix 2) indicates that the bottom-line efficiencies are not strong functions of the cycle times, although the individual terms in the equation may change significantly (but in ways that tend to cancel each other out in the efficiency expression).

Step 6. Calculate relaxation times for radiation and piping

To do this exactly is complicated by the fact that radiation and piping in the conditioned space are coupled to the inside and the outside. Since these have two different temperatures, an exact solution of the differential equations will give two relaxation times interacting in a complicated manner. For

radiation, the coupling to the inside is very much stronger than that to the outside, and so the outside coupling can be ignored to first order. For piping enclosed in baseboard or in an exterior wall, the heat transfer to the outside may be a larger fraction of the whole; nevertheless, such piping will usually be inside the exterior insulation and so, in any reasonably well designed system, the heat flow to the conditioned space will be at least several times larger than the heat flow to the outside. Therefore, for piping as well as for radiation, it will not be a bad assumption to take the final relaxation temperature (after infinite time) as that of the indoors for both radiation and piping. Under these conditions, a single relaxation time for each of the four categories of piping or radiation is given by its resistance-capacitance product:

$$\begin{aligned}\tau_r &= C_r \frac{R_{rc} R_{ra}}{R_{rc} + R_{ra}} \\ \tau_u &= C_u \frac{R_{uc} R_{ua}}{R_{uc} + R_{ua}} \\ \tau_{b,unins} &= C_{b,unins} R_{b,unins} \\ \tau_{b,ins} &= C_{b,ins} R_{b,ins}\end{aligned}\quad (6)$$

Step 7. Calculate the circulator on-time for design and seasonal conditions

This will depend on the design and seasonal-average heating loads. An approximate solution can be obtained by assuming that the conditioned space receives the steady-state rate of heat transfer while the circulator is on, and then the radiation cools off completely during the circulator off-time. This ignores: (1) the need for system warmup when the circulator comes on; (2) the fact that radiation will not cool down completely; (3) heat gains from unfinned piping; and (4) heat lost from radiation to the outside during its cooldown period. The calculated circulator ontime will therefore be somewhat different from its actual value under the specified conditions. That understood, the following equations are now solved for circulator ontime under design and seasonal conditions.

$$\begin{aligned}t_{cycle-design} L_{design} &= Q_c t_{on-design} + C_r \Delta T_{ln} \\ t_{cycle-seasonal} L_{seasonal} &= Q_c t_{on-seasonal} + C_r \Delta T_{ln}\end{aligned}\quad (7)$$

The circulator off-time is then just the difference between the cycle time and the on-time.

$$\begin{aligned}t_{off-design} &= t_{cycle-design} - t_{on-design} \\ t_{off-seasonal} &= t_{cycle-seasonal} - t_{on-seasonal}\end{aligned}\quad (8)$$

In testing these Equations 7, it was found that, where there is significant unfinned piping in the conditioned space, the calculated circulator on-times tend to give final values of useful heat delivered that are somewhat higher than the input heating loads (about 10% higher in test cases). In order to bring the delivered heats into closer agreement with the loads, an additional term was added to the equations to reflect the unfinned piping in the conditioned space.

$$\begin{aligned} t_{\text{cycle-design}} L_{\text{design}} &= Q_c t_{\text{on-design}} + C_r \Delta T_{\text{in}} + z (1 - e^{-t_{\text{off-design}}}) \\ t_{\text{cycle-seasonal}} L_{\text{seasonal}} &= Q_c t_{\text{on-seasonal}} + C_r \Delta T_{\text{in}} + z (1 - e^{-t_{\text{off-seasonal}}}) \end{aligned} \quad (9)$$

where $z = C_u \Delta T_{\text{in}} R_{\text{ua}} / (R_{\text{uc}} + R_{\text{ua}})$. A difficulty with these equations is that they are not so easy to solve for the t_{on} values as the ones without the additional term. However, because the final terms are relatively small, an approximation can be made that appears to work well, namely that under design conditions the off-time is about half the cycle and under seasonal conditions it is about 90% of the cycle. The equations then become:

$$\begin{aligned} t_{\text{cycle-design}} L_{\text{design}} &= Q_c t_{\text{on-design}} + C_r \Delta T_{\text{in}} + z (1 - e^{-0.5 t_{\text{cycle-design}}}) \\ t_{\text{cycle-seasonal}} L_{\text{seasonal}} &= Q_c t_{\text{on-seasonal}} + C_r \Delta T_{\text{in}} + z (1 - e^{-0.9 t_{\text{cycle-seasonal}}}) \end{aligned} \quad (10)$$

It is now an easy matter to solve for $t_{\text{on-design}}$ and $t_{\text{on-seasonal}}$ in terms of the corresponding cycle times. The discrepancy between the input heating loads (the L 's in the above equations) and the resulting heats delivered to the conditioned space (calculated below) were reduced to about 1% in the test cases mentioned above.

Step 8. Calculate the heat flows during cooldown

Heat flows to the conditioned space, buffer space, and ambient during the circulator-off period are calculated as follows. (Heat flows during the off-cycle will be denoted by H with subscripts, and, like the steady-state heat flows denoted by the Q 's, above, will have units of Btu/h or W.) We will write only one equation for each heat flow, with the understanding that separate values for design and seasonal average conditions should be evaluated for the appropriate design and seasonal values of quantities in the equations. Heat flows are calculated for one cycle, then divided by t_{cycle} to get the hourly value.

A. Energy flow from radiation to the conditioned space. This is by far the larger of the two energy flows from radiation, and can therefore be estimated by comparing the circulator off-time to the relaxation time.

$$H_{rc} = C_r \Delta T_{\text{in}} (1 - e^{-\frac{t_{\text{off}}}{\tau_r}}) / t_{\text{cycle}} \quad (11)$$

B. Energy flow from radiation to the outside ambient. Because of the exponentially declining temperature of the radiation after circulator cutoff, we may write

$$\begin{aligned}
 H_{ra} &= \frac{\Delta T_{ln}}{R_{ra} t_{cycle}} \int_0^{t_{off}} e^{-\frac{t}{\tau_r}} dt \\
 &\approx \frac{\Delta T_{ln}}{R_{ra} t_{cycle}} \int_0^{\infty} e^{-\frac{t}{\tau_r}} dt \quad (12) \\
 &= \frac{\Delta T_{ln} \tau_r}{R_{ra} t_{cycle}}
 \end{aligned}$$

where the middle step assumes $\tau_r \ll t_{off}$, which will usually be the case.

C. Heat flow from piping to the buffer space. Here there is no need to differentiate between heat flows to different spaces.

$$H_b = (\Delta T_{ln} + T_{in} - T_{buff}) [C_{b,unins} (1 - e^{-t_{off}/\tau_{b,unins}}) + C_{b,ins} (1 - e^{-t_{off}/\tau_{b,ins}})] / t_{cycle} \quad (13)$$

D. Heat flow from unfinned piping to conditioned space and outside. This is the one difficult case, since there are two heat sinks and it is not necessarily the case that one heat flow is very much larger than the other. Nevertheless, we should be able to assume that piping in the conditioned space, whether in baseboard or buried in the wall, is inside the exterior insulation, and hence the heat flow into the conditioned space will be at least several times larger than heat flow to the outside. In this case we will not go very far wrong if we assume, as in Step 6, that the overall heat loss from the piping in the conditioned space is governed by the relaxation time calculated there:

$$H_u = C_u \Delta T_{ln} (1 - e^{-t_{off}/\tau_u}) / t_{cycle} \quad (14)$$

It remains to pro-rate this heat loss to the inside and the outside. This is done on the basis of relative thermal resistances.

$$\begin{aligned}
 H_{uc} &= H_u \frac{R_{ua}}{R_{uc} + R_{ua}} \\
 H_{ua} &= H_u - H_{uc} \quad (15)
 \end{aligned}$$

Step 9. Calculate delivery efficiency under design and seasonal conditions

This is done in a manner similar to Step 4, by comparing the losses to the useful heat, but now there are additional terms, since each loss or component of useful heat has to include both the circulator-on and the circulator-off component. Also, because the circulator is on for a finite time t_{on} , this time must be multiplied by the circulator-on heat rates calculated in Step 3.

Again, with separate calculations for design and seasonal-average conditions, the thermal losses and the useful delivered heat are given, respectively, by:

$$\begin{aligned} H_{loss} &= (Q_a + Q_b) \frac{t_{on}}{t_{cycle}} + H_{ra} + H_{ua} + H_b \\ H_{del} &= Q_c \frac{t_{on}}{t_{cycle}} + H_{rc} + H_{uc} \end{aligned} \quad (16)$$

Note that in calculating $H_{loss-design}$, the value of Q_b computed in Step 3 can be used, but for $H_{loss-seasonal}$, it is necessary to recompute Q_b using the seasonal value of T_{buff} .

Finally, the design delivery efficiency is calculated in terms of these quantities.

$$\eta_{del-design} = \frac{1}{1 + \frac{H_{loss-design}}{H_{del-design}}} \quad (17)$$

The seasonal delivery efficiency $\eta_{del-seasonal}$ is calculated using the same equation, with $H_{loss-seasonal}$ and $H_{useful-seasonal}$ substituted for the corresponding design values.

Step 10. Calculate design and seasonal distribution efficiencies

This accounts for effective regain of a portion of the heat lost to the buffer space, computed in a manner similar to that for air distribution systems.

ASSIGNING NUMERICAL VALUES TO THE INPUT QUANTITIES

To use the above formalism in the Design and Diagnostic Pathways, it will be necessary to assign numerical values to the input quantities. In the Design Pathway, these values will be taken or derived from manufacturer's specifications, building plans, and/or default values. In the Diagnostic Pathway, values will be taken from measured values. Where these are not obtainable cost-effectively, it may be necessary in some instances to use manufacturer's specifications or default values, particularly where a quantity is not a major influence on the value of delivery efficiency.

The following suggestions are made concerning how to determine the input quantities.

R_{ra} Thermal resistance from radiation to the ambient

Design and Diagnostic Pathways. It is suggested that we use $R_{ra} = R_{wall}/A_{rad}$, where A_{rad} is the area of baseboard enclosure or radiator facing the wall, and R_{wall} is the thermal R-value of the wall. Note that R_{ra} has units of F-h/Btu [K/W], while R_{wall} has units of F-h-ft²/Btu [K-m²/W]. It may be necessary to add a "form factor" to account for the fact that the wall surface will not, in general, be as hot as the radiator, while on the other hand, the influence on the wall of the hot pipes may extend beyond the enclosure box. Fortunately, this heat transfer is usually small, so an approximation should be quite adequate.

R_{rc} Thermal resistance from radiation to the conditioned space

Design Pathway. This is determined from the specs on the number of feet of finned-tube baseboard and the manufacturer's data on the heat transfer. It is suggested that the following formula be used: $R_{rc} = \Delta T/(QL)$, where Q is the stated heat transfer rate per foot of baseboard at a temperature difference ΔT between the radiation and the room, and L is the total length of finned tube used. (An analogous formula should be developed for standing radiators.)

Diagnostic Pathway. Because of the importance of this quantity, it may be advisable to measure it. After an initial period of running the circulator "flat out," the circulator is stopped and the temperature at the pipe surface of a finned-tube radiation unit is measured as a function of time. The decay rate gives the time constant, which, together with the heat capacity of the unit (see below), gives the thermal resistance.

C_r Thermal capacitance of radiation

Design and Diagnostic Pathways. This is determined from the heat capacity of the water contained in the finned-tube baseboard, plus an allowance for the copper and the fins. A general treatment for all categories of pipe is as follows. First, define the following quantities:

Cv_{pipe} Volumetric heat capacity of pipe. The value for copper is 3.4×10^6 J/m³-K or 51 Btu/ft³-F.

Cv_{water} Volumetric heat capacity of water (4.1×10^6 J/m³-K or 61 Btu/ft³-F).

Cv_{ins} Volumetric heat capacity of pipe insulation. The value for Polystyrene (0.03×10^6 J/m³-K or 0.5 Btu/ft³-F) may be considered representative.

K_{fins} Heat capacity of fins per unit length of pipe. For finned pipe, if no value for this quantity is otherwise obtainable, 200 J/K-m pipe or 0.03 Btu/F-ft pipe may be used. For unfinned pipe, $K_{fins} = 0$.

The thermal capacitance per unit length, for each of the four pipe categories, can then be obtained using the following equation:

$$K_x = \frac{\pi}{4} [(d_2^2 - d_1^2) C_{v_{ins}} + (d_1^2 - d_0^2) C_{v_{pipe}} + d_0^2 C_{v_{water}}] + K_{fins} \quad 18$$

The value d_0 of the inside diameter of the pipe is needed. It is proposed to use $d_0 = 0.9 d_1$ as a default. The capacitance per unit length is then converted to capacitance by multiplying by the length of pipe in the given category.

R_{ua} Thermal resistance from unfinned piping to outside ambient

Design and Diagnostic Pathways. For unfinned piping located next to or within an outside wall, a formula similar to that for R_{ia} is used.

R_{uc} Thermal resistance from unfinned piping to conditioned space

Design and Diagnostic Pathways. The relative conductance from unfinned piping next to or within an outside wall, to the outside and to the conditioned space, needs to be assessed for standard situations. We propose to use an effective R-value to the inside of 2 ft²-F-h/Btu [0.4 m²-K/W].

C_u Thermal capacitance of unfinned piping in conditioned space

Design and Diagnostic Pathways. Same as for C_r , except the pipe has no fins.

$R_{b,ins}$ Thermal resistance from insulated piping to buffer space

Design and Diagnostic Pathways. Here the expression for passive heat transfer from insulated horizontal cylinders is used. The formula is:

$$U_{b,ins} = \frac{2 \pi}{\left[\frac{1}{k} \ln(d_2/d_1) + \frac{2}{d_2 h_{conv+rad}} \right]} \quad 19$$

where $h_{conv+rad}$ is the convective plus radiative heat transfer coefficient of the outer pipe surface, k is the thermal conductivity of the insulation, and d_1 and d_2 are the outer pipe and insulation diameters, respectively. For bare copper pipe, a good value of $h_{conv+rad}$ is 10 W/m²-K [1.75 Btu/h-ft²-F]. For other bare pipe and for insulated pipe, $h_{conv+rad}$ can be set equal to 13.6 W/m²-K [2.4 Btu/h-ft²-F]. $R_{b,ins}$ is then set equal to $1/(U_{b,ins} L_{b,ins})$. Derivation of these quantities is based on Stamper and Koral 1979. See Appendix 3 for details.

$R_{b,unins}$ Thermal resistance from uninsulated pipe to buffer space

Design and Diagnostic Pathways. For uninsulated pipe the above equation reduces to

$$U_{b,unins} = \pi d_1 h_{conv+rad} \quad 20$$

The value of $R_{b,unins}$ is then derived from this by multiplying by the length of uninsulated pipe in the buffer space and taking the inverse of the product.

$C_{b,unins}$ and $C_{b,ins}$ Thermal capacitance of uninsulated and insulated piping in buffer space

Design and Diagnostic Pathways. Same comments as for C_u .

T_{boiler} Nominal or average boiler-water temperature

Design Pathway. Manufacturer's recommendations, with a default value.

Diagnostic Pathway. Entering water temperature to the hydronic loop is measured for a period of time near the end of the circulator ontime.

$T_{buff-design}$ and $T_{buff-seasonal}$ Buffer-space temperature at design and seasonal-average conditions

T_{in} Temperature of the conditioned space

Design Pathway. These are all to be determined as specified in the forced-air part of the Design Pathway (Section 6 of ASHRAE 1996).

Diagnostic Pathway. Measure indoor temperature. Determine buffer-space design and seasonal temperatures as in Section 6 of ASHRAE 1996

L_{design} and $L_{seasonal}$ Design and seasonal-average heating load of the house

Design and Diagnostic Pathways. Two options are provided. One permits calculation of loads using accepted ASHRAE, ACCA, or I=B=R methods. The other uses default values $L_{design} = 0.6 Q_c$ and $L_{seasonal} = 0.2 Q_c$. The value of L_{design} was chosen to agree with the fractional ontime of 0.6 for furnaces in the forced-air portion of the Standard. The one-third ratio of seasonal to design heating load was chosen on the basis of weather data studies in which the design load was taken to be proportional to 65 F minus the 99% heating design temperature and the seasonal-average load was taken to be proportional to the number of heating degree-days divided by the number of days in the heating season. The latter was taken to equal 1/24 of the number of hours with outdoor temperatures less than 65 F. Ratios of seasonal to design heating loads varied between 0.29 and 0.38 for several locations in the Northeast, Midwest, Pacific Northwest, and California. The recommended value of 0.333 falls near the middle of this range.

$t_{\text{cycle-design}}$ and $t_{\text{cycle-seasonal}}$ **Circulator cycle time (on+off) under design and seasonal-average conditions**

Design and Diagnostic Pathways. Appendix 1 calculates these.

Cv_{water} **Volume heat capacity of distribution fluid (usually water)**

Design and Diagnostic Pathways. If water, use 61 Btu/ft³-F or 4.1 X 10⁶ J/m³-K. If some other fluid is used, use appropriate value.

V_e **Volume flow rate of water from boiler into distribution loop**

Design Pathway. Use design specs, or lacking these a default value of 1.5 gallons per minute (750 lb/h or about 0.1 kg/s) might be used.

Diagnostic Pathway. The Diagnostic Pathway measures T_{boiler} and T_{end} as the inlet and outlet temperatures of the hydronic system during a period of time near the end of a circulator run. These temperatures, together with the indoor temperature, are then used to calculate ΔT_{ln} . This bypasses the need to know V_e . (It also permits V_e to be back-calculated from Equation 2, though probably not with superb accuracy.)

ΔT_{ln} **Log-mean temperature difference**

Design Pathway. In the Design Pathway, log-mean temperature difference is a derived quantity. No additional data are needed beyond those specified above.

Diagnostic Pathway. In the Diagnostic Pathway, log-mean temperature difference is calculated from the boiler-water temperature T_{boiler} and the return-water temperature T_{end} . These are measured using temperature sensors located at the inlet and outlet to the hydronic loop. The sensors should be in good thermal contact with the pipe and be covered with an insulating patch. The reason for the insulating patch is that we want as good approximations as possible to the inlet and outlet water temperatures. Ideally, one should break into the system and install thermal wells, but that would be prohibitive in this standard. The next best thing is to insulate the pipe in the region where the sensor is located. These patches will cover a small portion of the entire pipe loop, so they won't affect system performance, but they should enable one to get reasonably good estimates of the water temperatures into and out of the system.

In the course of a review of a draft of this report, a question arose concerning how T_{end} would depend on the outdoor temperature, for given values of T_{boiler} and T_{in} . The question was answered as follows. Heat transfer to the ambient Q_a is always a very small quantity, even for poorly insulated walls. Heat transfer to the buffer space may be larger, but if it's in a basement the buffer space temperature generally doesn't change that much between design and seasonal conditions. If the buffer space is a crawl space its temperature can change significantly, so in a worst-case scenario T_{end} could depend on outdoor temperature to a

significant degree. Even here, though, it seems that the delivery efficiency is not a strong function of T_{end} . Changing T_{end} is equivalent to changing ΔT_{in} and changing ΔT_{in} mostly involves a slight change in the ratio of system ontime to system offtime, with very little change in delivery efficiency. For example, in a test set of calculations, the log-mean temperature difference was arbitrarily changed by 10 F (a far larger difference than would be caused by any plausible change in basement temperature), and this resulted in only a 0.4 percentage point change in delivery efficiency. (It did change the mix of on- vs. off-time operation.) Because of this it was debated whether it was even necessary to measure T_{end} , but in the end it was decided to propose that this measurement be required. One could look at the measurement of T_{end} as a back-door measurement of the water flow rate, assuming one trusts the calculated values of the heat transfer coefficients for the various pipe sections. It is important to note that the heat loss from pipes in the buffer space to that buffer space is accounted for at the proper design and seasonal temperatures in the calculations. It's just the log-mean temperature difference that will be slightly off, but this should not matter much.

PROPOSED LANGUAGE

Proposed language for use in Standard 152 has been prepared and is given in Appendix 4. It is to be understood that this language may be changed by the standards project committee by the time it goes out for public review.

REFERENCES

1. American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) 1996. *Standards Project Committee 152P draft standard* (unpublished). 1791 Tullie Circle N.E., Atlanta, GA 30329-2305.
2. Andrews, J.W. 1994. *Conceptual Framework for the Thermal Distribution Method of Test*. BNL-61462.
3. Kays, W. M. and A.L. London. *Compact Heat Exchangers*. McGraw-Hill, 1964.
4. Modera, M.P., J. Andrews, and E. Kweiler 1992. "A Comprehensive Yardstick for Residential Thermal Distribution Efficiency." In *Proceedings of the ACEEE 1992 Summer Study on Energy Efficiency in Buildings*, 2:167-178. Washington, D.C.: American Council for an Energy-Efficient Economy.
5. Stamper, E. and R.L. Koral. *Handbook of Air Conditioning Heating and Ventilating*. New York. Industrial Press Inc., 1979.

APPENDIX 1. SELECTION OF CYCLE TIMES

One important issue concerns the length of cycle times to select in the calculation. The following theory was developed to shed light on this question.

The model of hydronic heating developed in this paper is reflected in the following picture: for a period of time t_{on} , heat is delivered to the house at a rate Q_c . The circulator then shuts down. Heat continues to be delivered, however, because of the thermal mass of the piping in the conditioned space. A good enough approximation for the present purpose is to assume that after circulator shutdown heat continues to be delivered at the rate Q_c for a time τ_r , so that in sum, heat is delivered at the rate Q_c for a time $t_{on} + \tau_r$, and then no heat is delivered for the remainder of the cycle, i.e. $t_{cycle} - t_{on} - \tau_r$.

Meanwhile, heat is leaking from the house at a constant rate L . Thus, during the initial time $0 < t < t_{on} + \tau_r$, the house is taking in heat from the hydronic system at the rate Q_c , while for the entire cycle it is losing heat to the outside at the net rate L . Since the heat lost over one complete cycle must equal the heat gained, if the average temperature of the house is to remain constant, we may write the energy balance equation

$$Q_c (t_{on} + \tau_r) = L t_{cycle} \quad (A1-1)$$

(Equation A1-1 can also be derived from Equation 7 with the aid of Equations 4 and 6.)

If we make the assumption that the house is a single thermal capacitance C_{house} , then the maximum excursion ΔT_{house} of the indoor temperature away from the setpoint can be calculated as the ratio of the maximum thermal imbalance (which happens at time $t_{on} + \tau_r$) to the capacitance:

$$\begin{aligned} \Delta T_{house} &= \frac{(Q_c - L)(t_{on} + \tau_r)}{C_{house}} \\ &= \left(L - \frac{L^2}{Q_c}\right) \frac{t_{cycle}}{C_{house}} \end{aligned} \quad (A1-2)$$

Next, if we assume that the thermostat permits a constant ΔT_{house} regardless of load, and if the thermal capacitance of the house does not vary, we may write

$$\left(L - \frac{L^2}{Q_c}\right) t_{cycle} = \text{constant} \quad (A1-3)$$

Furthermore, since Q_c is constant we may put this in as an additional factor on the left-hand side and still have it equal to a constant (albeit a different one):

$$\left(\frac{L}{Q_c} - \frac{L^2}{Q_c^2}\right) t_{cycle} = constant \quad (A1-4)$$

The form of this equation is familiar, in that the number of cycles per hour, which is inversely related to t_{cycle} , now has a parabolic dependence on load, with cycles per hour peaking when the load is one-half the steady-state capacity of the hydronic system.

To go any further it is necessary to make an assumption concerning the maximum number of cycles per hour. For furnaces, this is generally assumed to be between 3 and 6. Clearly, it will depend on the allowed temperature swing and on the amount of thermal mass in the house. Both of these can vary widely. However, for the purposes of the standard, it is proposed to select a maximum number of cycles per hour that is in the accepted range and that is also likely to provide reasonable values of $t_{on-design}$ and $t_{on-seasonal}$. By "reasonable" is meant, first, that the value in question not be less than zero, and second, that it at least be comparable to the time needed for water to pass through a typical hydronic loop. Problems with negative and too-small values of t_{on} sometimes appear in the seasonal case with short cycle times. This can easily be fixed by increasing the cycle time. Appendix 2 will show that the delivery and distribution efficiencies do not appear to be strong functions of the cycle times. Nevertheless, we would like to select values that are as reasonable as possible.

If the maximum number of cycles per hour is 4 (i.e. shortest full cycle is 15 minutes), then

$$\left(\frac{L}{Q_c} - \frac{L^2}{Q_c^2}\right) t_{cycle} = 0.0625 \quad (A1-5)$$

if time is measured in hours. Plugging in the default values $L_{design} = 0.6 Q_c$ and $L_{seasonal} = 0.2 Q_c$ yields $t_{cycle-design} = 0.26$ h and $t_{cycle-seasonal} = 0.39$ h. It is proposed to use Equation A1-4 in the standard for cases where the design heating load is externally calculated, while for the default values $L_{design} = 0.6 Q_c$ and $L_{seasonal} = 0.2 Q_c$, the round-number values $t_{cycle-design} = 0.25$ h and $t_{cycle-seasonal} = 0.4$ h are proposed. These cycle times avoided the problems discussed above in the test cases considered in Appendix 2.

APPENDIX 2. SAMPLE CALCULATIONS

A spreadsheet embodying the above equations was set up, and a test case with “reasonable” values for the input parameters was run.

Piping not in the conditioned space was assumed to be in a basement, with a basement temperature of 50 F at the coldest times of the year (design conditions) and 55 F on the average during the heating season. The boiler-water temperature, on average, was assumed to be 180 F, while the indoor temperature was set at 70 F.

The hydronic system was assumed to have 100 ft of radiation with a heat transfer rate of 5 Btu/h-F per foot of finned tubing, based on a nominal 500 Btu/h at 100 F ΔT between the circulating water and the room. This translates to a total UA value for the radiation heat transfer to the room of $5 \times 100 = 500$ Btu/h-F, or a $R_{tc} = 1/500 = 0.002$ F-h/Btu. The other heat-transfer rates (U-values) and thermal capacities were obtained from the table lookups for 3/4-in. copper pipe either uninsulated or with 1-in. polymer foam insulation. The water flow rate was taken as 1.5 gallons per minute or 750 pounds per hour. The design and seasonal cycle times were varied to determine sensitivity to these parameters. A complete list of parameters for the test case is given at the end of this Appendix.

Table A2-1 shows the dependence of design and delivery efficiency on circulator cycle time for uninsulated pipe. As can be seen, the dependence is not great even for factor-of-two variations in this parameter. Table A2-2 gives the same dependence for insulated pipe. The dependence on cycle time is negligible for insulated pipe, at least for the case studied. We thus seem to be on safe ground in picking a value for this quantity, as long as it is within a reasonable range.

Copies of the original spreadsheets may be attached to some versions of this document. If they are not in the current copy, they can be obtained from the author.

Table A2-1. Design and Seasonal Delivery Efficiency (Basement Pipes Uninsulated) for Various Circulator Cycle Times (On & Off), Hours

Cycle time (hours)	0.2	0.3	0.4	0.5	1.0
Design Efficiencies					
Delivery	0.876	0.879	0.881	0.882	0.887
Distribution	0.928	0.930	0.931	0.932	0.936
$t_{\text{on-design}}$ (h)	0.065	0.123	0.182	0.241	0.537
Seasonal Efficiencies					
Delivery	0.750	0.760	0.767	0.774	0.805
Distribution	0.846	0.853	0.859	0.864	0.885
$t_{\text{on-seasonal}}$ (h)	negative	0.001	0.019	0.037	0.132

Table A2-2. Design and Seasonal Delivery Efficiency (Basement Pipes Insulated) for Various Circulator Cycle Times (On & Off), Hours

Cycle time (hours)	0.2	0.3	0.4	0.5	0.6
Design Efficiencies					
Delivery	0.953	0.954	0.954	0.954	0.955
Distribution	0.969	0.970	0.971	0.971	0.972
$t_{\text{on-design}}$ (h)	0.065	0.123	0.182	0.241	0.537
Seasonal Efficiencies					
Delivery	0.896	0.898	0.899	0.901	0.907
Distribution	0.932	0.934	0.936	0.938	0.943
$t_{\text{on-seasonal}}$ (h)	negative	0.001	0.019	0.037	0.132

List of Parameter Values Used in Spreadsheets

Variable	Value	Units	Definition
L_r	100	ft	Length of finned radiation
$L_{r,wall}$	80	ft	Length of above against outside wall
L_u	50	ft	Length of unfinned pipe in living space
$L_{u,wall}$	40	ft	Length of above against outside wall
$L_{b,unins}$	0 or 80	ft	Length of uninsulated pipe in buffer space
$L_{b,ins}$	80 or 0	ft	Length of insulated pipe in buffer space
y	0.67	ft	Height of baseboard enclosure
I_w	11	ft ² -F-h/Btu	R-value of wall
U_{rc}	5	Btu/h-F-ft	Conductance of finned radiation/ft pipe
$U_{b,unins}$	0.4	Btu/h-F-ft	Conductance of uninsulated pipe in buffer
$U_{b,ins}$	0.1	Btu/h-F-ft	Conductance of insulated pipe in buffer
K_r	0.27	Btu/F-ft	Capacitance of finned radiation/ft pipe
K_u	0.24	Btu/F-ft	Capacitance/ft of unfinned pipe in living space
$K_{b,unins}$	0.24	Btu/F-ft	Capacitance/ft of uninsulated pipe in buffer
$K_{b,ins}$	0.25	Btu/F-ft	Capacitance/ft of insulated pipe in buffer
Cv_{water}	61	Btu/ft ³ -F	Volume heat capacity of water
V_e	12	ft ³ /h	Volume flow rate of water
T_{boiler}	180	F	Boiler-water temperature
T_{in}	70	F	Indoor temperature
$T_{b,des}$	50	F	Design basement temperature
$T_{b,sea}$	55	F	Seasonal average basement temperature
$t_{c,des}$	0.2 - 1.0	h	Design cycle time
$t_{c,sea}$	0.2 - 1.0	h	Seasonal average cycle time
F_{regain}	0.50	--	Thermal regain factor

APPENDIX 3. DERIVATION OF PIPE HEAT-TRANSFER COEFFICIENTS

Equation 19 makes use of a combined radiative-convective heat transfer coefficient $h_{\text{conv+rad}}$ governing heat transfer from the surface of a pipe. Derivation of numerical values for this quantity followed standard procedures as given in Stamper and Koral (1979), page 8-164. Radiative heat transfer is given by:

$$q_r/A = 0.174 \epsilon [(T_1/100)^4 - (T_2/100)^4] \quad A3-1$$

where q_r is the heat (in Btu) transferred via radiation per hour per square foot of pipe surface, A is the pipe surface area, ϵ is the emissivity of the pipe, T_1 is the absolute temperature (degrees R or F+460) of the pipe, and T_2 is the absolute temperature of the surroundings. Note that pipe surface refers to the outermost surface (i.e. the outer surface of the insulation, if present). Using the usual approximation, the factor in square brackets on the right-hand-side of Equation A3-1 can be expressed, very nearly, as $0.04 (T_1 - T_2)(T_{\text{av}}/100)^3$, where $T_{\text{av}} = (T_1 + T_2)/2$. The radiative heat transfer coefficient h_{rad} can then be written as:

$$h_{\text{rad}} = \frac{q_r}{A(T_1 - T_2)} = 0.00696 \epsilon (T_{\text{av}}/100)^3 \quad A3-2$$

For $T_1 = 180$ F and $T_2 = 70$ F, T_{av} equals 585 R and h_{rad} is 1.39 ϵ .

For convection from a horizontal pipe, the heat transfer rate is given by:

$$q_c/A = 1.016 d^{-0.2} T_{\text{av}}^{0.181} (T_1 - T_2)^{1.266} \quad A3-3$$

where q_c is the heat (in Btu) transferred via convection per hour per square foot of pipe surface, d is the outside diameter of the pipe (including any insulation) in inches and the other quantities are as defined above. The convective heat transfer coefficient h_{conv} can then be written as:

$$h_{\text{conv}} = \frac{q_c}{A(T_1 - T_2)} = 1.016 d^{-0.2} T_{\text{av}}^{0.181} (T_1 - T_2)^{0.266} \quad A3-4$$

Using the same values of pipe and surrounding temperature as for the radiation case, one obtains h_{conv} values equal to 1.84, 1.76, and 1.70 Btu/h-ft²-F for 1/2-in., 3/4-in., and 1-in. pipe, respectively.

The combined convective and radiative heat transfer coefficient $h_{\text{conv+rad}}$ is then just the sum of h_{conv} and h_{rad} . Stamper and Koral give emissivity values of 0.44 for (tarnished) copper pipe and 0.94 for

(oxidized) iron pipe. A value of 0.94 is given for "paint" and 0.90 for "non-metallic surfaces." Inserting $\epsilon = 0.44$ into the equations for convective and radiative heat transfer yields $h_{\text{conv}+\text{rad}}$ values of 1.84, 1.76, and 1.70 Btu/h-ft²-F for ½-in., ¾-in., and 1-in. pipe, respectively. A compromise value of 1.75 Btu/h-ft²-F (10.0 W/m²-K) was chosen for use in calculating values to be recommended for the Standard. For iron pipe and for painted and nonmetallic surfaces, an emissivity value of 0.94 was used to obtain radiative+convective heat transfer coefficients of 2.54, 2.46, 2.40, and 2.28 for ½-in., ¾-in., 1-in., and 2-in. overall diameters, respectively. A value of 2.4 Btu/h-ft²-F (13.6 W/m²-K) was recommended for use in calculations under the Standard.

APPENDIX 4. PROPOSED WORDING OF HYDRONICS SECTIONS IN STANDARD 152

Language proposed for hydronic systems in Standard 152, as of April 1996, is given below.

10.0 Hydronic Design

10.1 Instrumentation Specifications

No instrumentation required

10.2 Apparatus

No apparatus required

10.3 Test Method

No testing required

10.4 Procedure

In the Design Pathway, values for a set of thermal parameters are selected on the basis of information supplied by the builder or, where this is lacking, using default values. These parameters are then input into a set of equations that is used to derive the design and seasonal efficiencies of the hydronic system. At present, the formalism is developed for finned-tube baseboard heaters only. Other types of radiation, including in-slab radiant heating, remain to be implemented in this Standard.

10.4.1 Piping Dimensions

A single-loop hydronic distribution system is divided into four categories:

- Finned radiation (all assumed in the conditioned space);
- Unfinned piping in the conditioned space;
- Unfinned piping in buffer spaces that is not insulated.
- Unfinned piping in buffer spaces that is insulated.

Sections of each type of piping may, and usually do, alternate in succession. The sum of all pipe sections of like kind will be referred to as a category.

The following dimensional data are required for further calculations (units are m[ft]):

<u>Symbol</u>	<u>Description</u>
L_r	Total length of finned radiation
$L_{r,wall}$	Length of finned radiation mounted against exterior wall
L_u	Total length of unfinned piping in the conditioned space
$L_{u,wall}$	Length of unfinned piping mounted against exterior wall
$L_{b,unins}$	Length of unfinned piping in the buffer space that is not insulated
$L_{b,ins}$	Length of unfinned piping in the buffer space that is insulated
d_1	Outside diameter of piping
d_2	Outside diameter of insulation surrounding that portion (if any) of the piping in the buffer space that is insulated.
y	Height of convector enclosure surrounding finned radiation

10.4.2 Thermal Properties of Piping

10.4.2.1 Thermal Conductances

Piping and radiation in each of the four categories are characterized by thermal conductances or linear U-values (expressed in W/K-m pipe or Btu/h-F-ft pipe) derived either from manufacturer's data or from simple calculations. The following information is required in order to perform the calculations.

10.4.2.1.1 Finned-Tube Radiation

The value U_{rc} , which characterizes heat transfer from the finned radiation to the conditioned space, shall be obtained from manufacturer's data on heat transfer vs. pipe temperature. The temperature value closest to 80 °C (180 F) shall be used. The given heat transfer rate (in W/m pipe or Btu/h-ft pipe) shall be divided by the pipe-to-surrounding temperature difference used in the manufacturer's data sheet to obtain U_{rc} . Generally, a surround temperature of 18 °C (65 F) is used.

The heat-transfer rate U_{ra} (in W/m pipe or Btu/h-ft pipe) to the outside from finned radiation that is mounted against an exterior wall shall be estimated using the thermal R-value I_w of the outside wall, where I_w is expressed in m²-K/W [ft²-F-h/Btu]. The value of U_{ra} shall be equal to y/I_w .

10.4.2.1.2 Unfinned Piping in the Buffer Space (Table Lookup)

The U-values for insulated and uninsulated piping in the buffer space may be determined using one of two methods, table lookup and formula method. The table lookup method shall be used when the piping is one of the nominal diameters listed in this Standard, and the insulation thickness is between 0.5 and 2.0 in. [1.3 and 5.1 cm]. Currently the standard provides table lookups for ½ in., ¾ in., and 1 in. nominal inside-diameter (I.D.) copper piping. The outer diameters (O.D.) of these pipes sizes are:

- ½ in. nominal I.D.-- 0.625 in. [1.59 cm] O.D.
- ¾ in. nominal I.D.-- 0.875 in. [2.22 cm] O.D.
- 1 in. nominal I.D.-- 1.125 in. [2.86 cm] O.D.

The actual outer diameter of the pipe shall be measured and compared with these values to determine applicability of table lookup. If the specified pipe and insulation material are in Tables 10.1, the conductance values $U_{b,ins}$ and $U_{b,unins}$ shall be determined in W/K-m pipe from Table 10.1A or in Btu/h-F-ft pipe from Table 10.1B. The thermal conductivities used in developing Tables 10.1 are the same as those specified in Table 10.2. If table lookup is not applicable, formula calculation shall be applied, in accord with Section 10.4.2.1.3.

Table 10.1A. Thermal Conductances for Common Pipe Sizes, W/K-m pipe.

Insulation Material and Thickness	½-Inch Nominal I.D. Copper Pipe	¾-Inch Nominal I.D. Copper Pipe	1-Inch Nominal I.D. Copper Pipe
None	0.52	0.70	0.87
Corrugated, 1.3 cm	0.43	0.53	0.63
Corrugated, 2.5 cm	0.29	0.35	0.41
Corrugated, 5.1 cm	0.21	0.25	0.28
Molded Fiber, 1.3 cm	0.27	0.34	0.41
Molded Fiber, 2.5 cm	0.19	0.22	0.26/
Molded Fiber, 5.1 cm	0.14	0.16	0.18
Polymer Foam 1.3 cm	0.22	0.28	0.33
Polymer Foam 2.5 cm	0.15	0.18	0.21
Polymer Foam 5.1 cm	0.11	0.13	0.14

Table 10.1B. Thermal Conductances for Common Pipe Sizes, Btu/h-F-ft pipe.

Insulation Material and Thickness	½-Inch Nominal I.D. Copper Pipe	¾-Inch Nominal I.D. Copper Pipe	1-Inch Nominal I.D. Copper Pipe
None	0.30	0.40	0.50
Corrugated, ½ in.	0.25	0.31	0.37
Corrugated, 1 in.	0.17	0.21	0.24
Corrugated, 2 in.	0.12	0.14	0.16
Molded Fiber, ½ in.	0.16	0.20	0.24
Molded Fiber, 1 in.	0.11	0.13	0.15
Molded Fiber, 2 in.	0.08	0.09	0.10
Polymer Foam, ½ in.	0.13	0.16	0.19
Polymer Foam, 1 in.	0.09	0.10	0.12
Polymer Foam, 2 in.	0.06	0.07	0.08

10.4.2.1.3. Unfinned Piping in the Buffer Space (Formula)

$$U_{b,ins} = \frac{2\pi}{\left[\frac{1}{k} \ln(d_2/d_1) + \frac{2}{d_2 h_{conv+rad}}\right]} \quad 10.1$$

If the table lookup method cannot be used, then the thermal U-value of unfinned piping in the buffer space that is insulated shall be calculated using Equation 10.1, where $h_{conv+rad}$ is the convective plus radiative heat transfer coefficient of the outer pipe surface. For bare copper pipe, $h_{conv+rad}$ shall equal 10 W/m²-K [1.75 Btu/h-ft²-F]. For other bare pipe and for insulated pipe, $h_{conv+rad}$ shall equal 13.6 W/m²-K [2.4 Btu/h-ft²-F]. If the thermal conductivity k of the insulating material is not provided by the manufacturer, then an appropriate value from Table 10.2 may be selected on the basis of material specified, with corrugated sheathing as the default. The U-value of unfinned piping in the buffer space that is not insulated shall be calculated using Equation 10.2.

$$U_{b,unins} = \pi d_1 h_{conv+rad} \quad 10.2$$

10.4.2.1.4. Unfinned Piping in (or Adjacent to) the Conditioned Space

If unfinned piping in the conditioned space is wholly within the insulated envelope of the building but enclosed in baseboard units similar to those used for the finned-tube radiation, the thermal conductances U_{ua} and U_{uc} to the conditioned space and outside respectively, shall be calculated using Equations 10.3a and 10.3b respectively:

$$U_{ua} = y / I_w \quad 10.3a$$

$$U_{uc} = y / I_{uc} \quad 10.3b$$

where I_w is the thermal R-value of exterior walls against which finned radiation is mounted, in m²-K/W [ft²-F-h/Btu], and I_{uc} shall equal 0.4 m²-K/W [2 ft²-h-F/Btu]. If such piping is buried in the wall, these values shall be calculated using Equation 10.4.

$$U_{ua} = U_{uc} = \frac{2 y}{I_w + I_{uc}} \quad 10.4$$

10.4.2.2 Thermal Capacitance per Unit Length (Table Lookup)

If the system loop is composed of ½ in., ¾ in. or 1 in. nominal copper piping, the thermal capacitances per unit length, expressed in J/K-m pipe or Btu/F-ft pipe, for finned and unfinned sections, shall be determined using Table 10.3. If the table is not applicable, the formulas in Section 10.4.2.3 shall be used. Thermal capacitances per unit length shall be obtained for all four piping segments. These will be denoted by the letter K with a subscript, as follows:

K_r	Finned radiation in the conditioned space
K_u	Unfinned piping in the conditioned space
$K_{b,unins}$	Unfinned piping in the buffer space that is uninsulated
$K_{b,ins}$	Unfinned piping in the buffer space that is insulated

Table 10.2. Thermal Conductivity Values for Piping Insulation

Material Type	k in W/m-K	k in Btu/h-ft-F
Corrugated cardboard sheathing	0.069	0.04
Molded mineral fiber	0.043	0.025
Foamed rubber or polystyrene	0.035	0.02

Note: If a manufacturer's value is given in Btu/h-ft²-F and inch of thickness, multiply this value by 0.083 to obtain Btu/h-ft-F or by 0.144 to obtain W/m-K.

Table 10.3. Thermal Capacitance per Unit Length for Commonly Specified Pipes

Pipe Size and Type	Units System	J/K-m pipe	Btu/F-ft pipe
½ in. nominal Type K or L, bare, unfinned		750	0.12
¾ in. nominal Type K or L, bare, unfinned		1500	0.24
1 in. nominal Type K or L, bare, unfinned		2500	0.40
If finned, add		200	0.03
For 1 in. insulation, add		100	0.01
For 2 in. insulation, add		200	0.03

10.4.2.3. Thermal Capacitance per Unit Length (Formula)

If Table 10.3 is not applicable to the system under evaluation, the formulas in this section shall be applied. The following quantities are required for their application:

Cv_{pipe}	Volumetric heat capacity of pipe. If this is not provided, the value for copper may be used (3.4×10^6 J/m ³ -K or 51 Btu/ft ³ -F).
Cv_{water}	Volumetric heat capacity of water (4.1×10^6 J/m ³ -K or 61 Btu/ft ³ -F). Cv_{ins}
	Volumetric heat capacity of pipe insulation. If this is not provided, the value for polystyrene (0.03×10^6 J/m ³ -K or 0.5 Btu/ft ³ -F) may be used.

- K_{fins} Heat capacity of fins per unit length of pipe. For finned pipe, if no value for this quantity is otherwise obtainable, 200 J/K-m pipe or 0.03 Btu/F-ft pipe may be used. For unfinned pipe, $K_{fins} = 0$.

The thermal capacitance per unit length, for each of the four pipe categories, shall be obtained using Equation 10.5.

$$K_x = \frac{\pi}{4} [(d_2^2 - d_1^2) C_{v_{ins}} + (d_1^2 - d_0^2) C_{v_{pipe}} + d_0^2 C_{v_{water}}] + K_{fins} \quad 10.5$$

where $K_x = K_r, K_u, K_{b,ins}$ and $K_{b,unins}$ successively, with appropriate values of the d 's, C_v 's and K_{fins} inserted into Equation 10.5 for each pipe segment. In order to use this equation, the value d_0 of the inside diameter of the pipe is needed. For the purposes of this section d_0 may be assumed to equal 0.9 d_1 .

10.4.3 Other Thermal Properties

The following quantities are needed in the following sections:

<u>Symbol</u>	<u>Description</u>
$C_{v_{water}}$	Volumetric heat capacity of fluid (normally water). If not specified, a value of 4.1×10^6 J/m ³ -K [61 Btu/ft ³ -F] may be used.
V_c	Volumetric circulator flow rate. If not specified, use a default value 10^{-4} m ³ /s [12 ft ³ /h], equal to 1.5 U.S. gallons per minute.
T_{boiler}	Nominal or average boiler-water temperature. If not specified, a default value of 80 °C [180 F] may be used.
T_{in}	Indoor temperature. The default value of 21 °C [70 F] shall be used unless a different value is specified.
$t_{cycle-design}$	Circulator cycle time (on+off) under design conditions. If no specified value is available, the value 1800 s [0.5 h] shall be used.
$t_{cycle-seasonal}$	Circulator cycle time (on+off) under seasonal average conditions. Unless the system design specifies an alternative, the default value of 1100 s [0.3 h] shall be used.

10.5 Calculations

10.5.1 Thermal Parameters

10.5.1.1 Temperatures in Buffer Space

The design and seasonal temperatures of the buffer space in which piping not in the conditioned space is located ($T_{buff-design}$ and $T_{buff-seasonal}$) shall be calculated in the same way as $T_{amb,s}$ and $T_{amb,r}$ for air ducts in Section 6.2.2.

10.5.1.2 Thermal Resistances

The thermal resistance R_{rc} from finned radiation to the conditioned space shall equal $1/(L_r U_{rc})$.

The thermal resistance R_{ra} from finned radiation to the outside shall be equal to $1/(L_{r,wall} U_{ra})$. If $L_{r,wall} = 0$, then $1/R_{ra}$ shall equal zero in all equations in which it appears.

The thermal resistance R_{uc} from unfinned piping to the conditioned space shall be set equal to $1/(L_u U_{uc})$. If $L_u = 0$, then $1/R_{uc}$ shall equal zero in all equations in which it appears.

The thermal resistance R_{ua} from unfinned piping to the outside shall be equal to $1/(L_{u,wall} U_{ua})$. If $L_{u,wall} = 0$, then $1/R_{ua}$ shall equal zero in all equations in which it appears.

The thermal resistance $R_{b,unins}$ from unfinned piping to the buffer space that is not insulated shall be set equal to $1/(L_{b,unins} U_{b,unins})$. If $L_{b,unins} = 0$, then $1/R_{b,unins}$ shall equal zero in all equations in which it appears.

The thermal resistance $R_{b,ins}$ from unfinned piping to the buffer space that is insulated shall be set equal to $1/(L_{b,ins} U_{b,ins})$. If $L_{b,ins} = 0$, then $1/R_{b,ins}$ shall equal zero in all equations in which it appears.

10.5.1.3. Thermal Capacitances

The thermal capacitance C_r of finned-tube radiation shall be equal to $L_r K_r$.

The thermal capacitance C_u of unfinned piping in the conditioned space shall equal $L_u K_u$.

The thermal capacitance $C_{b,ins}$ of unfinned piping in the buffer space that is insulated shall be equal to $L_{b,ins} K_{b,ins}$.

The thermal capacitance $C_{b,unins}$ of unfinned piping in the buffer space that is not insulated shall be equal to $L_{b,unins} K_{b,unins}$.

10.5.1.4. Heat-Transfer Rates

The number of heat-transfer units in the piping system (x) shall equal $UA/(C_{v,water} V_e)$, where $UA = 1/R_{ra} + 1/R_{rc} + 1/R_{ua} + 1/R_{uc} + 1/R_{b,ins} + 1/R_{b,unins}$.

The log-mean temperature difference ΔT_{ln} between the piping and the conditioned space shall be equal to $(T_{boiler} - T_{in})(1 - e^{-x})/x$.

The steady-state heat transfer rates (W or Btu/h) from the hydronic loop to the conditioned space, buffer space, and outside ambient shall be calculated using Equations 10.6.

$$Q_c = \Delta T_{ln} \left(\frac{1}{R_{rc}} + \frac{1}{R_{uc}} \right) \quad 10.6a$$

$$Q_a = \Delta T_{ln} \left(\frac{1}{R_{ra}} + \frac{1}{R_{ua}} \right) \quad 10.6b$$

$$Q_b = [\Delta T_{ln} + (T_{in} - T_{buff-design})] \left(\frac{1}{R_{b,ins}} + \frac{1}{R_{b,unins}} \right) \quad 10.6c$$

10.5.1.5. Heating Loads

The efficiency of the hydronic system is a function of the ratio of its capacity to deliver heat and the demand for heat by the building. The heating load of the house under design and seasonal conditions is therefore a factor in the final calculation. Two options are available. If Option 1 is selected, the design heating load L_{design} shall be calculated using Chapter 25 of the ASHRAE Handbook of Fundamentals, ACCA Manual J, or I=B=R Manual H-22. However, the design heating load so calculated shall not exceed $0.8 Q_c$. The seasonal-average heating load $L_{seasonal}$ shall then be set equal to one-third the design load.

Option 2 provides default values for the heating loads. If this option is selected, the design heating load L_{design} shall be set equal to $0.6 Q_c$. The seasonal average heating load $L_{seasonal}$ shall be set equal to $0.2 Q_c$.

10.5.2 Design and Seasonal Delivery Efficiency

10.5.2.1 Relaxation times

The relaxation times or time constants τ_r , τ_u , $\tau_{b,ins}$, and $\tau_{b,unins}$, for radiation, unfinned piping in the conditioned space, and insulated and uninsulated piping in the buffer space, respectively, shall be calculated using Equations 10.7.

$$\tau_r = C_r \frac{R_{rc} R_{ra}}{R_{rc} + R_{ra}} \quad 10.7a$$

$$\tau_u = C_u \frac{R_{uc} R_{ua}}{R_{uc} + R_{ua}} \quad 10.7b$$

$$\tau_{b,ins} = C_{b,ins} R_{b,ins} \quad 10.7c$$

$$\tau_{b,unins} = C_{b,unins} R_{b,unins} \quad 10.7d$$

10.5.2.2 Circulator on-time and off-time

Circulator on-time for design and seasonal conditions shall be calculated using Equations 10.8.

$$t_{on-design} = \frac{t_{cycle-design} L_{design} - C_r \Delta T_{ln} - z (1 - e^{-0.5 t_{cycle-design} / \tau_u})}{Q_c} \quad 10.8a$$

$$t_{on-seasonal} = \frac{t_{cycle-seasonal} L_{seasonal} - C_r \Delta T_{ln} - z (1 - e^{-0.9 t_{cycle-seasonal} / \tau_u})}{Q_c} \quad 10.8b$$

where $z = C_u \Delta T_{ln} R_{ua} / (R_{uc} + R_{ua})$. The minimum allowed value for $t_{on-seasonal}$ shall equal 72 seconds [0.02 h]. The purpose of this requirement is to ensure that circulator on-times are at a minimum comparable to the time needed for a given particle of water to circulate once through a typical piping system. If this requirement is not met, the design and seasonal cycle times shall be increased in 360 s [0.1 h] increments until $t_{on-seasonal}$ does equal or exceed the above-stated minimum.

The circulator off-time shall be set equal to the difference between the cycle time and the on-time, as indicated in Equations 10.9.

$$t_{off-design} = t_{cycle-design} - t_{on-design} \quad 10.9a$$

$$t_{off-seasonal} = t_{cycle-seasonal} - t_{on-seasonal} \quad 10.9b$$

10.5.2.3 Circulator-off heat flows

Heat flows to the conditioned space, buffer space, and ambient during the off-cycle shall be calculated as follows. In each case, separate values for design and seasonal conditions shall be computed, with $t_{\text{off-design}}$ and $t_{\text{off-seasonal}}$ substituted for t_{off} in the equations in this section. Heat flows are in units of energy per unit time.

Energy flow H_{rc} from radiation to the conditioned space shall be calculated using Equation 10.10.

$$H_{rc} = C_r \Delta T_{ln} (1 - e^{-t_{\text{off}}/\tau_r}) / t_{\text{cycle}} \quad 10.10$$

Heat flow H_{ra} from radiation to the outside ambient shall be calculated using Equation 10.11.

$$H_{ra} = \frac{\Delta T_{ln} \tau_r}{R_{ra} t_{\text{cycle}}} \quad 10.11$$

Heat flow H_b from piping to the buffer space shall be calculated using Equation 10.12.

$$H_b = (\Delta T_{ln} + T_{in} - T_{buff}) [C_{b,ins}(1 - e^{-t_{\text{off}}/\tau_{b,ins}}) + C_{b,unins}(1 - e^{-t_{\text{off}}/\tau_{b,unins}})] / t_{\text{cycle}} \quad 10.12$$

Heat flow H_u from unfinned piping in the conditioned space shall be calculated using Equation 10.13.

$$H_u = C_u \Delta T_{ln} (1 - e^{-t_{\text{off}}/\tau_u}) / t_{\text{cycle}} \quad 10.13$$

The portions of this heat flow to the inside and the outside (H_{uc} and H_{ua} , respectively) shall be calculated using Equations 10.14.

$$H_{uc} = H_u \frac{R_{ua}}{R_{uc} + R_{ua}} \quad 10.14a$$

$$H_{ua} = H_u - H_{uc} \quad 10.14b$$

10.5.2.4 Loss and delivery heat rates

The rate of heat loss H_{loss} and heat delivery H_{del} over both the on- and off-cycles shall be calculated, for design and seasonal conditions, using Equations 10.15.

$$H_{loss} = (Q_a + Q_b) \frac{t_{on}}{t_{cycle}} + H_{ra} + H_{ua} + H_b \quad 10.15a$$

$$H_{del} = Q_c \frac{t_{on}}{t_{cycle}} + H_{rc} + H_{uc} \quad 10.15b$$

Note that in calculating $H_{loss-design}$ the value of Q_b computed in 10.5.1.4 can be used, but for $H_{loss-seasonal}$ it is necessary to recompute Q_b using the seasonal value of T_{buff} .

10.5.2.5 Delivery efficiency calculations

Finally, the design delivery efficiency is calculated in terms of these quantities, using Equation 10.16.

$$\eta_{del-design} = \frac{1}{1 + \frac{H_{loss-design}}{H_{del-design}}} \quad 10.16$$

The seasonal delivery efficiency $\eta_{del-seasonal}$ is calculated using the same equation, with $H_{loss-seasonal}$ and $H_{useful-seasonal}$ substituted for the corresponding design values.

10.5.3. Calculation of design and seasonal distribution efficiency

10.5.3.1 Equipment Efficiency Factor

The equipment efficiency factor F_{equip} shall equal 1.0 for hydronic systems.

10.5.3.2 Thermal Regain Factor

The thermal regain factor shall be determined using the value for the buffer space as specified in Section 6.2.9.1. This value shall then be multiplied by the fraction of lost heat that goes to the buffer space, namely $H_b/(H_b + H_{ra} + H_{ua})$, to obtain F_{regain} .

10.5.3.3 Load Factor

The design and seasonal building load factors shall be calculated using

$$F_{load,design} = \frac{1}{1 - (1 - \eta_{del,design}) F_{regain}} \quad 10.17a$$

$$F_{load,seasonal} = \frac{1}{1 - (1 - \eta_{del,seasonal}) F_{regain}} \quad 10.17b$$

10.5.3.4. Distribution efficiency calculations

Design and seasonal distribution efficiency shall be calculated from the delivery efficiencies, the equipment factor, and the load factor as specified in Section 6.3.5.

11.0 Hydronic Diagnostic

11.1 Instrumentation Specifications

11.1.1 Temperature Measurements

Temperatures at specified points in the piping system shall be measured using systems (i.e. sensor plus data acquisition system) having an accuracy of $\pm 0.25^{\circ}\text{C}$ (0.5°F).

11.1.2 Linear Dimensions

Linear dimensions of piping shall be measured to an accuracy of $\pm 2\%$. Calipers (for cross sections) and tape measure (for lengths) or equivalent devices are recommended.

11.2 Apparatus

Temperatures shall be measured at a specified point on the pipe by attaching the sensor to the pipe in good thermal contact with it. Sensors that measure inlet and outlet temperatures of the piping loop shall be isolated from the surrounding air using an insulating patch having thermal conductivity less than 0.05 W/m-K (0.03 Btu/h-ft-F), at least 1 cm (0.4 in.) thick and extending laterally at least 5 cm (2 in.) in all directions.

11.3 Test Method

The test method uses temperature measurements at two points on the piping and two points within building spaces, together with lengths and cross sections of piping segments, to evaluate the most significant quantities in the Design Pathway calculation (Section 10). That section is then used with the measured quantities to calculate the design and seasonal delivery and distribution efficiencies. The need to measure flow rates is avoided, making the method usable as a diagnostic.

11.4 Procedure

In the Diagnostic Pathway, values for a set of thermal parameters are selected on the basis of measured lengths and diameters of piping and insulation, manufacturer's data or dimensional measurements on the finned-tube radiation used in the building, and on input and output temperatures to and from the hydronic loop. These parameters are then input into the same set of equations that is used in the Design Pathway to derive the design and seasonal efficiencies of the hydronic system. At present, the formalism is developed for finned-tube baseboard heaters only. Other types of radiation, including in-slab radiant heating, remain to be implemented in this Standard.

11.4.1 Length Measurements

A diagram of the hydronic loop shall be made, with notations concerning portions of the loop that fall under each of the four classifications listed in Section 10.4.1. The linear dimensions listed in Section 10.4.1 shall be measured.

11.4.2 Sensor installation

Install temperature sensors as specified in 11.2 at the following points:

- On the section of hydronic pipe immediately adjacent to the boiler, at the inlet to the hydronic system (supply from the boiler).
- On the section of hydronic pipe immediately adjacent to the boiler, on the outlet from the hydronic system (return to the boiler).

11.4.3 Additional Temperature Measurements

One-time temperature measurements shall be made:

- Within the conditioned space, as near as practicable to the thermostat
- Within the buffer space, at the average height above the floor as the piping in that space, and as far as practicable from exterior walls, stairwells, and heat sources.

In addition, an optional test, if performed, requires two additional temperature sensors. (See Section 11.4.5 for description.)

11.4.4 Circulator-on test

Set back the thermostat at least 5°C (9°F) from normal setpoint and allow to remain there for one hour. (The homeowner can be instructed to do this before the arrival of the diagnostic team.) At the initiation of testing, increase the thermostat setting by at least 15°C (27°F), so that the circulating pump comes on. Record the hydronic inlet and outlet temperatures at least once every 30 seconds, for at least 10 minutes or until the circulator shuts down.

11.4.5. Circulator-off test (Optional)

In a diagnostic situation, it may sometimes be the case that manufacturer's specifications on the finned-tube radiation are not conveniently available. In this case, an alternative means of obtaining the necessary parameters for this component is permitted, as described in this section. Use of the information obtained is discussed in Section 11.5.

Install two temperature sensors meeting the requirements of Section 11.1.1 as follows:

- In good thermal contact with the pipe, as near as possible to the center of the longest continuous run of finned-tube baseboard within the conditioned space.
- Within the room served by the above finned-tube baseboard, at a height between 1 m and 1.5 m [3 ft and 5 ft] above the floor.

Immediately upon circulator shutdown, or after a minimum of ten minutes, decrease the thermostat setting to its lowest possible value, so that the circulator will remain off for the remainder of the test. Record the temperatures of the two sensors and the times of measurement, at least once every 30 seconds until the temperature difference between the pipe and its surroundings has decreased below 75% of its first measured value.

11.5 Calculations

All quantities defined in the Hydronic Design Pathway (Section 10) shall be computed as specified in 10.5.1, with the exception of ΔT_{lm} and, if the optional circulator-off test of Section 11.4.5 was performed, R_{tc} .

11.5.1 Log-Mean Temperature Difference

From the circulator-on test of 11.4.2, the average hydronic inlet temperature reading over the final three minutes of circulator operation shall be set equal to T_{boiler} . The average hydronic outlet temperature reading over the final three minutes of circulator operation shall be set equal to T_{end} .

The log-mean temperature difference shall be computed using Equation 11.1.

$$\Delta T_{ln} = \frac{T_{boiler} - T_{end}}{\ln\left(\frac{T_{boiler} - T_{in}}{T_{end} - T_{in}}\right)} \quad 11.1$$

11.5.2 Finned-Tube Thermal Resistance to Conditioned Space (Optional)

If the optional circulator-off test of Section 11.4.5 was performed, the temperature difference between the finned radiation and the conditioned-space temperature sensor shall be plotted on semi-log paper as a function of time. The inverse of the slope of this graph shall be set equal to the finned-radiation relaxation time constant τ_r .

If the optional circulator-off test was performed, the thermal resistance between finned baseboard and the conditioned space shall be computed using Equation 11.2.

$$R_{rc} = \frac{\tau_r}{C_r} \quad 11.2$$

11.5.3 Delivery and distribution efficiencies

Design and seasonal delivery and distribution efficiencies shall be computed as specified in 10.5.2, except that the value of ΔT_{ln} shall be determined as specified in 11.5.1, and if the optional circulator-off test was performed, the values of τ_r and R_{rc} shall be determined as specified in 11.5.2.