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FUTURE DIRECTIONS FOR THERMAL DISTRIBUTION STANDARDS

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ABSTRACT

This report details development paths for advanced versions of ASHRAE Standard 152, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Efficiency. During the course of conversations within the ASHRAE committee responsible for developing the standard (SPC152P), three areas of development for Standard 152 were proposed: 1) extend the scope of the standard to include thermal comfort variables; 2) extend the scope of the standard to include small commercial buildings; and 3) improve the existing standard with respect to accuracy and economy of effort. Research needs associated with each of the three options are identified.

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EXECUTIVE SUMMARY

This report details development paths for advanced versions of ASHRAE Standard 152, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Efficiency. During the course of conversations within the ASHRAE committee responsible for developing the standard (SPC152P), three areas of development for Standard 152 were proposed:

- Extend the scope of the standard to include thermal comfort variables.
- Extend the scope of the standard to include small commercial buildings.
- Improve the existing standard with respect to accuracy and economy of effort.

Thermal distribution efficacy has become a common term for what most of those in the field of thermal distribution mean by a figure of merit that takes account of the system's ability to provide desired levels of thermal comfort at all times and at all locations within the building. A precise definition of efficacy, however, has yet to be agreed upon. Two approaches are being considered. One depends on measures of delivered comfort such as temperature and humidity in the various rooms or zones of the building, while the other uses energy quantities in a manner similar to that employed by ASHRAE Standard 152, but with the aim of quantifying comfort rather than efficiency.

The second major area for possible extension of ASHRAE Standard 152 is for small commercial buildings. Research in Florida and California has indicated that the energy losses from air distribution in small commercial buildings may be even greater than those typically found in residences. However, the residential version of Standard 152 may need to be altered in order to apply to small commercial buildings. Primary areas of concern include the separation of air and thermal barriers (not an issue in residences), active ventilation and exhaust systems (usually a minor issue in residences), the larger airflows found in some commercial buildings (even ones considered "small"), possible need to determine loads based on factors such as occupancy (usually a minor factor in residences), and possible need to consider interactions between the ducts and the building that are significantly different from those found in residences.

The third area is the further refinement of the residential standard. It is common practice within ASHRAE to begin consideration of the next version of a standard soon after the current one is approved. Such continuing maintenance assures that a standard remains up-to-date and relevant to the needs of the users. Areas where such refinement might be undertaken include better duct leakage tests, more accurate treatment of interactions between distribution systems and the zones in which they are located, inclusion of continuous fan operation in the standard, improved algorithms for duct-equipment interactions, "tweaking" of the calculations for conductive losses, revisiting the question of infiltration and off-cycle losses, inclusion of zoning impacts, and addition of new system types to the standard.

Each section of the report considers possible research efforts that might be taken in support of these activities.

1.0 INTRODUCTION

This report details development paths for advanced versions of ASHRAE Standard 152, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Efficiency.(ASHRAE 2001)

During the course of conversations within the ASHRAE committee responsible for developing the standard (SPC152P), three areas of development for Standard 152 were proposed:

1. Extend the scope of the standard to include thermal comfort variables.
2. Extend the scope of the standard to include small commercial buildings.
3. Improve the existing standard with respect to accuracy and economy of effort.

2.0 THERMAL DISTRIBUTION EFFICACY

Thermal distribution efficacy has become a common term for what most of those in the field of thermal distribution mean by a figure of merit that takes account of the system's ability to provide desired levels of thermal comfort at all times and at all locations within the building. A precise definition of efficacy, however, has yet to be agreed upon.

2.1 Issues in Defining Efficacy

Two possible approaches to the definition of thermal distribution efficacy have been identified. One of these would construct a measure of efficacy in terms similar to those in which thermal distribution efficiency has been defined, namely by comparing two energy quantities. The other, perhaps simpler approach, is to look at the variability of some measure of delivered comfort across the rooms or zones of the building.

A start toward the first type of definition was made in the course of preparing this report. It is discussed further below. At the June 2003 meeting of ASHRAE, a discussion within the Standards Subcommittee of TC 6.3 concentrated on the second approach. Because of the latter's relative simplicity, it will be discussed first. After that, an exposition of an energy-comparison method is given.

2.2 Efficacy Based on a Measure of Delivered Comfort

In the discussions at the 2003 ASHRAE Annual Meeting, two different approaches were proposed. One of these would use just the measured temperatures (or temperatures and humidities for cooling), while the other would compare delivered heating or cooling to each room with the calculated load. This would, of course, require that airflows as well as temperatures and humidities be measured at the registers.

2.2.1 Efficacy Based on Temperature and Humidity Only

A very simple method of defining thermal distribution efficacy would be based on a measure of the variability of the delivered comfort conditions (temperature, humidity) at the supply registers. In the heating mode, the delivered air temperature after a set interval of continuous system operation would be measured. This interval might be set to capture the typical conditions on average during part-load operation. Alternatively, it might try to duplicate steady-state operation. Or, as in the case of thermal distribution efficiency, two values might be given, one relating to design conditions and the other to seasonal-average conditions.

However that might be, it would be desirable to benchmark the condition for which efficacy is 1.0 (or 100%) and those for which it is zero. The 100% efficacy condition is perhaps most easily visualized, in the heating mode, as a situation in which all the delivered temperatures are equal.

What then would constitute zero efficacy? This is a more difficult question that has no obvious answer. It seems we need some kind of benchmark against which to rate the variability of the observed delivered temperatures. One possibility might be to say that zero efficacy would be a condition in which the standard deviation of the delivered temperatures is equal to the difference between the average delivered temperature and the setpoint temperature of the thermostat. For example, suppose there are six supply registers with measured delivered air temperatures of 100, 105, 110, 115, 120, and 125 °F, and the setpoint temperature is 70 °F. The standard deviation of the delivered air temperatures is 9.3, while their average is 112.5 °F. The efficacy defined in this way would then be 1 – 9.3/(112.5-70) = 0.78.

The cooling mode presents the complication that humidity and temperature are both important. One could perform a similar operation on the measured humidities at the registers. In this case, should one use absolute water vapor content or relative humidity (RH)? Once that decision is made, how should the humidity information be merged with the temperature information? Perhaps the delivered enthalpy could be used as a measure of the delivered-air condition at each register. Then, in a similar manner to the above suggestion, the standard deviation of the delivered enthalpies could be compared with the difference between the average delivered enthalpy and the enthalpy of the intended condition within the living space (e.g., the set-point temperature and 50% RH).

For example, suppose that the measured conditions at the six registers in the cooling mode are as shown in Table 1. The temperatures and RH values would be measured at each register, and then the enthalpies would be calculated. The standard deviation of the register enthalpies is 1.53 while the difference between the average register enthalpy and the intended indoor-air enthalpy is 28.4 – 21.7 = 6.7. The calculated efficacy would then be 1 – 1.53/6.7 = 0.77.

Table 1. Sample measured register conditions for simple efficacy, cooling.

Register or Space	Temperature (°F)	Relative Humidity	Enthalpy (Btu/lb air)
	(measured)		(calculated)
1	55	80%	21.3
2	60	70%	22.8
3	55	75%	20.7
4	65	60%	24.3
5	50	90%	19.5
6	55	85%	21.7
Average			21.7
Indoor setpoint	75	50%	28.4

2.2.2 Efficacy Based on Delivered Heating or Cooling Compared with Room Load

The second suggestion was to compare the heating or cooling delivered at each register with the load that this register must meet. This would require that the airflow at each register be measured, in addition to the temperature and humidity. For each register, the delivered cooling in Btu/hr would then be calculated by multiplying the delivered airflow by the density of air (either at the delivered condition or, with a minor loss of accuracy, at standard conditions) and then by the difference between the delivered enthalpy and the enthalpy of air at the desired room conditions. Table 2 illustrates such a procedure. For example, the Register 1 airflow of 130 cfm is first converted to 7800 ft^3/hr and multiplied by the standard density of 0.075 lb/ft^3 and then by the difference between 21.3 and 28.4, yielding a delivered cooling rate of 4154 Btu/hr.

The ratio of the delivered cooling to the load would be calculated for each register. The standard deviation of these ratios would then be calculated. Using the same philosophy as in the previous example, an efficacy of 1.0 or 100% would be defined as that condition for which all these ratios are equal. Zero efficacy would be defined as the condition where this standard deviation equals the average value of the ratio. In the case illustrated in Table 2, the standard deviation of the delivered to load cooling ratios is 0.196, leading to an efficacy of $1 - 0.196/1.135 = 0.83$

This second method, although more complicated, has the virtue of relating the delivered heat or cooling to the needs of the various spaces that are conditioned. One deficiency, however, remains, namely that there is no consideration of whether the latent-sensible split is what it should be. This could be fixed by calculating the delivered and load sensible and latent cooling rates separately and calculating a contributing factor from each to an overall efficacy, or alternatively of defining two separate efficacies.

It should be recognized that under this definition of efficacy, fixing a leaky, poorly insulated duct system might result in a drop in efficacy if the existing duct system

Table 2. Sample measured register conditions for load-related efficacy, cooling.

Regi-ster or Space	Temp . (°F)	RH (%)	Enthalpy of air (Btu/lb)	Airflow (CFM)	Deliv'd Cooling (Btu/hr)	Room Load Btu/hr	Ratio
	measured		calculated	measured	calculated		
1	55	80%	21.3	130	4154	3500	1.19
2	60	70%	22.8	170	4284	4500	0.95
3	55	75%	20.7	150	5198	5000	1.04
4	65	60%	24.3	100	1845	2000	0.92
5	50	90%	19.5	150	6008	4000	1.50
6	55	85%	21.7	200	6030	5000	1.21
Avg			21.7		4586	4000	1.135
Indoor setpoint	75	50%	28.4				

happened to match the cooling needs of the rooms. Fixing the ducts would then unbalance the system. This is not necessarily a drawback, though. It could serve as a signal that the system needs to be rebalanced.

Another possible outcome is that the efficacy wouldn't change much, even though the energy efficiency from fixing the ducts is greatly improved. For example, if all the ducts were improved to the point that they delivered air at the same condition as the best duct in the Table 2 example (i.e., 50 °F, 90% RH), then the delivered cooling would increase from 27,516 to 36,048 Btu/hr, a 31% improvement, presumably without any increase in energy input. However, the standard deviation of the delivered-to-room load ratios would now be 0.236, leading to an efficacy of 1 – 0.236/1.502 or 0.84. This is not much different from the value before the duct upgrade. This simply means that in this particular case, fixing the ducts did not improve the balance of the system even though it greatly reduced the amount of running time needed to meet the load.

Table 3. Sample measured register conditions for load-related efficacy, after ducts are fixed, cooling example.

Regi-ster or Space	Temp (°F)	RH (%)	Enthalpy of air (Btu/lb)	Airflow (CFM)	Deliv'd Cooling (Btu/hr)	Room Load Btu/hr	Ratio
	measured		calculated	measured	calculated		
1	50	90%	19.5	130	5206	3500	1.49
2	50	90%	19.5	170	6808	4500	1.51
3	50	90%	19.5	150	6008	5000	1.20
4	50	90%	19.5	100	4005	2000	2.00
5	50	90%	19.5	150	6008	4000	1.50
6	50	90%	19.5	200	8010	5000	1.60
Avg			19.5		6008	4000	1.502
Indoor setpoint	75	50%	28.4				

2.3 Efficacy Based on a Ratio of Energy Quantities

The other approach to efficacy would be to obtain values of input energy needed to heat or cool the house uniformly and compare these with the actual heating or cooling energy, in a manner reminiscent of the way thermal distribution efficiency is defined in ASHRAE Standard 152. Although there probably is a connection between this approach and the load-based approach described in the previous section, it is far from obvious that they are the same.

To arrive at a definition of thermal distribution efficacy, it might be useful to consider what was done in defining thermal distribution efficiency. There, the actual distribution system under evaluation was benchmarked against a heating/cooling system whose thermal distribution component had no direct thermal losses and had no impact on either the equipment efficiency or the load. The ratio of the input (fuel or electrical) energy needed to run this "perfect-distribution" system to that required to run the actual system was defined as the thermal distribution efficiency.

In a similar fashion, in order to define thermal distribution efficacy, it might be possible to benchmark the actual system against a system that provides uniform temperature and humidity throughout the house. However, the definition of efficacy will likely require some care. Considerations such as the following will need to be addressed:

1. If portions of the house are maintained at higher temperatures (heating) or lower temperatures/humidities (cooling) than are required for comfort, this represents "wasted effort." It seems clear that this should be charged against the system.
2. If portions of the house are maintained at lower temperatures (heating) or higher temperatures/humidities (cooling) than are required for comfort, this represents inadequate space conditioning. One imagines that this should also be charged against the system, but perhaps not in the same way.
3. If the system maintains thermal comfort equivalent to that provided by a uniform space conditioning system, but at different temperature or humidity conditions, this should be permitted and any energy savings should be credited to the system. Possible ways this could be done might include:
--Radiant heating systems that provide equivalent mean radiant temperatures at lower sensible temperatures.
--Zoned systems that "underheat" or "undercool" spaces at times when they are unoccupied.
--Systems that enhance thermal comfort in the cooling mode through the use of ceiling fans or other air-moving devices.

The following issues related to these objectives can be noted:

1. In what form should thermal distribution efficacy be presented?
2. How does one do the accounting of the excess use of energy for overconditioning, and the penalty for underconditioning?
3. How does one decide whether the level of thermal comfort provided by an alternative system is really equivalent to the benchmark?

4. How does one test for the above conditions without requiring a daylong (or longer) test protocol that clearly would be beyond the scope of what contractors would accept as a field diagnostic?

2.3.1 Thermal Distribution Efficacy Definition

If thermal comfort is to be incorporated into a test method such as ASHRAE Standard 152, it probably will need to be encapsulated into a single figure of merit. Otherwise, one is likely to be led into an "apples and oranges" situation in which one criterion shows up positive and another negative, and there is no clear-cut decision on whether one system is "better" than another. Of course, it may be that no such decision is possible in many cases, but then one could argue that developing a thermal comfort test method is simply a bad idea.

Assuming that a figure of merit is to be developed, the next choice may be whether to base it on energy quantities or on some other quantities related to thermal comfort. The argument for using energy quantities is similar to the argument often used in economics for translating things that are priceless (such as human life) into money "equivalents" as the only way to go forward. The argument against this is that thermal comfort has nothing inherently to do with energy, and a figure of merit based on other criteria, such as predicted mean vote or predicted percent dissatisfied, should be used instead.

Without intending to preclude the latter option, the rest of this section will explore the feasibility of basing a thermal comfort figure of merit on energy quantities.

Let us first review how thermal distribution efficiency is currently defined in Standard 152. The obvious way to define it would be as a ratio of heat or cooling delivered to the conditioned space by the distribution system to the heat or cooling input to the distribution system by the equipment. This ratio is a useful quantity (which Standard 152 calls delivery effeciveness). However, it is not an adequate definition of distribution efficiency because it does not account for interactions between the distribution system and either the equipment or the load. In order to include these interactions, thermal distribution efficiency, η_{dist}, is actually defined as:

$$\eta_{dist} = \frac{\text{Energy needed to heat/cool the house if the distribution system were "perfect"}}{\text{Energy used to heat/cool the house with the existing distribution system}} \qquad (1)$$

A "perfect" distribution system is defined as one that has no losses due to heat or mass flows into or out of the system, and no impacts on the heating/cooling load or the efficiency of the equipment. By "energy needed" or "energy used" is meant the input energy at the threshold, i.e., the heat content of the fuel if the system uses gas or oil, or the energy content of the electricity if it uses an electric-powered heat pump or air conditioner. A great deal of thought went into this definition, and it has stood the test of time.

However, the definition does have one failing that some might consider important, and that is a certain amount of ambiguity about what it means to "heat/cool the house." Informally, the members of the Standard 152 committee have always spoken of the definition as "assuming" that uniform thermal comfort is provided. The measurements and calculations in Standard 152 are consistent with this assumption. However, if it is now desired to get into the issue of thermal comfort explicitly, a little more precision is going to be needed. Should the numerator and denominator in the definition of η_{dist} be taken to apply to the non-uniform comfort provided by the actual system, and to the same non-uniform level of comfort as provided if the distribution system were "perfect"? Or is the definition to be taken as projecting what the energy use values for both the actual and perfect systems would be if they did provide uniform thermal comfort?

In either case, it would seem that the overall distribution system performance rating should be a ratio of the purchased energy required to heat/cool the house with a perfect distribution system providing uniform thermal comfort to that required to provide the actual non-uniform level of thermal comfort with the actual distribution system. A proviso needs to be added here, that no system should benefit from any reduced energy requirement deriving from its less-than-perfect comfort performance.

Let us make the definitions per the following Table 4. In each case, the Quantity is the purchased energy needed to heat/cool the house to either uniform thermal comfort or the actual level of thermal comfort, with either a perfect distribution system or the actual distribution system.

Table 4. Criteria for defining quantities related to thermal distribution efficacy

Quantity	Thermal Comfort Level	Distribution System Quality
$E_{perf,U}$	Uniform	Perfect
$E_{perf,nonU}$	Non-Uniform (i.e., actual)	Perfect
$E_{actual,U}$	Uniform	Actual
$E_{actual,nonU}$	Non-Uniform (i.e., actual)	Actual

Thermal distribution efficiency, then, might have either of the following definitions, each of which is consistent with Equation 1:

Interpretation A: $$\eta_{dist} = E_{perf,U} \,/\, E_{actual,U} \qquad (2)$$

Interpretation B: $$\eta_{dist} = E_{perf,nonU} \,/\, E_{actual,nonU} \qquad (3)$$

In either case, we would like to have the distribution system performance factor, defined as the product of the distribution system efficiency η_{dist} and the distribution system efficacy (which we'll call φ_{dist}), equal to a ratio with perfect distribution and uniform comfort in the numerator and actual distribution and actual comfort in the denominator:

$$\eta_{dist}\,\varphi_{dist} = E_{perf,U} \,/\, E_{actual,nonU} \qquad (4)$$

This implies two different possible definitions for φ_{dist}, depending on which interpretation, A or B, is used for η_{dist}.

Interpretation A: $\varphi_{dist} = E_{actual,U} / E_{actual,nonU}$ (5)

Interpretation B: $\varphi_{dist} = E_{perf,U} / E_{perfl,nonU}$ (6)

Comparing the two interpretations of efficacy (Equations 5 and 6), one may intuitively prefer Equation 5, since it deals with the actual system, whereas if Equation 6 is used, one must project what perfect systems would do hypothetically. This leads us to choose Interpretation A. This interpretation is also consistent with the above-cited rubric in the Standard 152 committee of "assuming that the system provides uniform comfort" even if it doesn't.

It is recognized that there may be arguments in the other direction. For the present, however, we will adopt Equation 5 as the definition of efficacy, subject to one major modification as discussed below.

Let us now look into how Equation 5 would deal with the three conditions set forth previously, namely over-conditioning, under-conditioning, and alternative conditioning.

2.3.2 Over-Conditioning

If the system over-conditions parts of the house, it will presumably require more energy than if it provides uniform thermal comfort. Under these circumstances, $E_{actual,U}$ will be a smaller number than $E_{actual,nonU}$, making $\varphi_{dist} < 1$. The system is penalized for the excess energy it used to over-condition the space.

2.3.3 Under-Conditioning

If the system under-conditions portions of the house, e.g., leaves some rooms too cold in the heating mode, then it is likely that the actual system will use less input energy than it would if it had to condition the house uniformly. That is, $E_{actual,nonU}$ will be less than $E_{actual,U}$, which will result in $\varphi_{dist} > 1$. The system would be given a credit for providing too little space conditioning. In the limit where the system doesn't function at all, its efficacy would go to infinity. Clearly this can't be right. One way to fix this is to add a proviso that any energy "saved" by under-conditioning a space must be added back to $E_{actual,nonU}$. For a system whose only deficiency was that it under-conditioned some spaces, i.e., there was no over-conditioning, this would result in $\varphi_{dist} = 1$.

This might or might not be what we want. Intuitively, one may want to penalize a system that under-conditions some spaces, but probably not by as much as one would penalize a system that over-conditions some spaces, since the two systems may be equally uncomfortable, but the former will at least save energy while the latter squanders it. It may be desirable, therefore, to add some kind of additional factor here. However, since any such decision will be to some extent arbitrary, we will not attempt to define one.

The above discussion leads to a slightly modified version of Equation 5:

$$\varphi_{dist} = E_{actual,U} \,/\, (E_{actual,nonU} + \Delta E_{undercond}) \qquad (7)$$

where $\Delta E_{undercond}$ is any additional purchased energy that would be required to make up for the under-conditioning by the actual system.

2.3.4 Alternative Conditioning

By alternative conditioning is meant any thermal distribution strategy that can provide a level of thermal comfort equal to that represented by uniform temperature and humidity throughout the space, but at the cost of less input energy. The major examples of such strategies are expected to be:

- Use of variables other than ambient temperature and humidity to provide comfort. For example, radiant heating systems may provide thermal comfort at lower air temperatures than forced-air systems because the mean radiant temperature is higher. A system that makes use of air movement as part of its overall comfort-cooling strategy may achieve comfort at higher air temperatures than a standard air-conditioning system.
- A system that intentionally under-conditions unoccupied spaces may use less energy than one that maintains uniform conditions throughout the building, yet because these under-conditioned spaces are not occupied, no loss of thermal comfort is experienced.
- A system that employs a lower temperature setpoint at night in the heating mode may be considered to provide thermal comfort equal to one that uniformly heats the space at all times, if the occupants like such conditions for sleeping, as some in fact do.

In all of these cases, no energy add-back would be required, that is, $\Delta E_{undercond}$ would be set equal to zero.

2.4 What would an efficacy test method look like?

It is too early to speculate in much detail on how one would measure thermal distribution efficacy, but at a minimum it would probably involve measurement of temperatures, humidities, and perhaps other comfort variables at various places in the building while the system is operating. An obvious question here is this: To what extent will these parameters differ at different times of the day or at different periods within the heating or cooling season? If the efficacy that is derived from any set of measured quantities is found to vary widely depending on the precise timing of the test, the result will be of little value.

2.5 How would an efficacy figure of merit be used?

In other words, why do we need it? Perhaps the salient use would be to sort out competing claims of various system types which purport to provide better thermal comfort than the standard type of system. Examples would include modulating systems, radiant systems, systems incorporating ceiling fans or other in-room air-moving devices, and continuous or extended fan operation. These are almost certain to be controversial subjects. The deliberations in any committee charged with creating an efficacy test method are likely to be lengthy, involved, and perhaps heated at times. Still, if successful, such an effort may resolve some of the issues that currently are addressed, if at all, in piecemeal and ad-hoc ways.

2.6 Research needs

The research needed to develop a thermal distribution efficacy standard would appear to be relatively modest. It may be useful to divide it into three parts:

1. Detailed development of a thermal distribution efficacy definition. This would include both design and seasonal efficacy values.

2. Development of a measurement protocol. This would not only include prescriptions and uncertainty limits for measuring register air temperatures, humidities, and airflows, but also criteria for operation of the system prior to and during the tests. Consideration would need to be given to the issue of whether and to what extent simultaneous measurements at the different registers would be needed (adding complexity and cost) or time delays from one to the next could be tolerated (possibly impacting accuracy and repeatability).

3. Field testing of the protocol. Shakedown tests of draft protocols would be needed to sort out inevitable problems and concerns that nearly always arise when an idea is put into practice. More than one iteration of field testing and protocol improvement might be needed.

3.0 SMALL COMMERCIAL BUILDINGS

The second major area for possible extension of ASHRAE Standard 152 lies in the area of small commercial buildings. Research in Florida and California has indicated that the energy losses from air distribution systems in small commercial buildings are likely to be comparable to or even greater than those typically found in residences. Results of this work may be found in Cummings and Withers 1998; Cummings et al. 1996; Delp et al. 1998a, 1998b, and 1997; Westphalen and Koszalinski 1999; Withers and Cummings 1998; Withers et al. 1996; and Xu et al. 20000. A "fact sheet" has been prepared for the U.S. Department of Energy (Andrews et al. 2002) on many of the topics germane to a small-commercial test method. To gain an appreciation for the issues that are likely to arise, it may be useful to compare and contrast the characteristics of thermal distribution systems in small commercial buildings with those found in single-family residences, on the one hand, and large commercial buildings, on the other.

3.1 Similarities and Differences Between Residential and Small Commercial

The first and most obvious question, once this classification system is proposed, is: "What (other than size) makes a small commercial building different from a large commercial building, and where does one draw the line between the two?" A short answer to this would have two parts. First, small commercial buildings are like single-family residences and small apartment buildings in that their heating and cooling loads are dominated by heat transfers across the envelope, while in large buildings, internal heat generated within the core of the building tends to dominate, producing cooling loads even in winter in cold climates. Second, the HVAC systems in small commercial buildings are usually similar to those in residences in that they tend to be composed of a heating/cooling equipment unit mated to a duct system that serves the entire building, or at least a large zone within such a building, while large buildings often have very different (and usually more complex) types of systems involving large central chillers, chilled water loops, dual-duct systems, variable air volume delivery, economizers, terminal reheat, and so on.

There is no precise dividing line between a small commercial building and a large one. However, as a rule of thumb, 20,000 square feet of conditioned floor space is often taken as the upper limit for a small commercial building.

Despite the similarities between residential and small commercial buildings cited above, there are significant differences as well. These include: 1) in residential buildings, the air barrier and the thermal barrier are usually in the same location within the envelope, whereas in small commercial buildings these barriers are often separate, especially in the ceiling space; and 2) small commercial air distribution systems usually have separate provisions for inducting and exhausting air from or to the outside, such provisions usually being rudimentary or lacking in residential systems. Additionally, the total airflow rate in a small commercial system is very often greater than the 1000 to 1500 cfm typically found in a residential system.

Additional differences may stem from the very different types of use patterns found in small commercial buildings. These may affect both design and seasonal loads as well as the efficiency with which a given HVAC system can deliver comfort. The great variety of such loads provides an uncertainty factor that is not present in residential systems, even though the latter are not entirely homogeneous in their own right. Probably some simplification will be required in dealing with this, the question of course being, What is the optimal tradeoff between simplicity and reality?

Finally, the types and sizes of heating and cooling equipment are more varied in small commercial buildings than they are in residences. Some small commercial buildings have systems that look very much like residential ones, but there are also systems such as unitary rooftop units that are not often found in residences. This in itself may not present any serious issues, but the details of the equipment's operation may, especially if it is provided with a number of operating modes in which things like airflow and equipment capacity vary. The simple functions for equipment efficiency factors that are used in the residential version of Standard 152 may be quite inadequate to cover the small commercial case. Of course, in large commercial buildings it is expected that the relationship between equipment and the distribution system will become vastly more complicated and varied. The question is to what extent this can be simplified to handle the small-commercial case with acceptable accuracy. These comparisons are summarized in Table 5.

Table 5. Characteristics of three building types.

Building Type / Characteristic	Residential	Small Commercial	Large Commercial
Type of heating and cooling load	Envelope dominated	Envelope dominated	Core dominated
HVAC system type	Central unit + ducts	Central unit + ducts	Complex system
Location of air barrier and thermal barrier	Together	May be separate, particularly in the ceiling space	May be separate, though may be a less critical issue if core dominates.
Active outside air induction/exhaust	Absent or rudimentary	May include outside air, exhaust air, and makeup air system.	Usually includes extensive ventilation & exhaust systems.
Airflow rates	1000-1500 cfm	1000-10000 cfm	10000 cfm and up
Occupancy and use effects	Variability not a serious concern	Highly variable	Highly variable
Duct-equipment interactions	Simple formulas for Fequip	Situation can be more complex	Situation is usually more complex

3.2 Areas where Standard 152 may need to be altered

These considerations may lead naturally to an enumeration of areas where the residential version of Standard 152 may need to be altered in order to be appropriate for small commercial buildings.

3.2.1. Location of ducts relative to air and thermal barrier

Buildings need to control the transfer of heat and air between the inside and the outside. Heat transfer is limited mostly by insulation, while air exchange is limited by the physical structure. Commercial buildings usually have suspended T-bar ceilings (often called "drop ceilings"), which provide a convenient space for electrical and mechanical services, including the ductwork. This means that the overhead portions of the air and thermal barriers can either be together, at the ceiling or at the roof, or separate, with the thermal barrier at the ceiling and the air barrier at the roof.

Five possible ceiling-space configurations have been defined (Andrews, Cummings, and Modera 2002). The first is typical of residential housing. It has a tight gypsum-board ceiling and a vented attic. In residences, the ductwork is often placed in the vented attic space, though it may be elsewhere. This type represents only 2% of the small commercial building stock.

The second configuration is like the first, except that it has a suspended T-bar ceiling instead of gypsum board. It is the most common. Air leakage through this type of ceiling tends to be quite high, a definite minus when it comes to energy efficiency. Also a minus is the placement of the ducts in a very hot and humid location. Because of these two factors, uncontrolled airflows can produce large impacts upon energy use, ventilation rates, and indoor humidity.

The third configuration is like the second except that the ceiling space is not vented. This puts the ducts inside the air barrier, which is good, although they are still outside the thermal barrier, which is bad. During the cooling season, the ceiling space tends to be very hot and dry. Uncontrolled airflows increase energy use but not ventilation rates or humidity levels.

The fourth and fifth configurations differ only in whether or not there is a drop ceiling. In both cases the insulation is at the roof plane and the space below the roof is unvented. Both are very forgiving of uncontrolled airflow as long as the ductwork is inside the building. (If the ducts are on top of the roof, all bets are off!) Duct leakage and unbalanced return air have little impact upon energy use, ventilation rates, and indoor humidity, because conditions in the space below the roof deck are not greatly different from those in the rooms.

3.2.2. Impact of active ventilation and exhaust systems

Forced-air systems in most small commercial buildings have at least three basic parts: 1) the **return air** duct, which transports room air to the air handler; 2) the **air handler**, which has a blower to move the air and whatever devices are used to impart or extract heat from the air stream; and 3) the **supply air** duct, which distributes the conditioned air throughout the building. In some systems, the return air can be anything from a single large grille upstream of the air handler to a complete duct system connected to every room in the building. In contrast, the supply air nearly always has runouts leading to every conditioned room. In small commercial buildings, many forced-air systems have an **outdoor air** duct leading from the outside to the return side of the ductwork; this is used to provide ventilation. Ventilation may also be provided by a separate **exhaust air** system consisting of a duct and fan that blow air out of the building. Finally, there may also be a **make-up air** system blowing air into the building, used where necessary to balance out all the other airflows.

In contrast to residences, where forced flow of air into or out of the building (e.g., because of exhaust fans) is usually a relatively minor part of the overall airflow picture, in small commercial buildings the outdoor-air, exhaust-air, and make-up air systems may at times exceed the system airflow inside the building. The impacts of these systems therefore need to be taken into account in a more sophisticated way than just sealing off the openings, as is typically done in residential tests.

Perhaps the most important single factor characterizing these airflows is "unbalanced return air." Return air is unbalanced when the amount of air drawn from a zone for return to the central heating/cooling equipment does not equal the amount of supply air delivered to that zone. This imbalance creates pressure differences between various parts of the building when such a zone is isolated from other zones by barriers such as walls and closed interior doors. The most common form of unbalanced return air occurs when return grilles are located only in the central zone and interior doors are closed. Supply air is delivered to the closed rooms, so they experience positive pressures. Negative pressure occurs in the central zone because the supply air is prevented from returning to the central return when interior doors are closed.

Return air can also be unbalanced when a firewall divides a ceiling space that functions as a return plenum. Good practice calls for "transfer windows" through the firewalls (with fire dampers), but in many cases these transfer windows are missing, are undersized, or have closed dampers.

3.2.3. Test methods for larger airflows

Airflow test equipment, such as blower doors, duct blasters, and flow plates, have been marketed in sizes appropriate for single-family residences, with airflows up to ~2000 cfm at the system fan and ~5000 cfm across the envelope (at ~50 Pa) during blower-door tests. Many small commercial buildings experience much larger airflows over parts of the system and, during blower-door tests, across the envelope. Researchers have resorted to the use of two or more blower doors, duct blasters, etc. to do these tests. Alternatively, a blower door can be rigged up as a duct blaster to provide larger airflows as needed. Whether such expedients will prove adequate in diagnostic situations, where time and ease of use are critical variables, is an open question. Some consideration of more expedient measurements in small commercial buildings would seem to be warranted.

3.2.4. Determination of design and seasonal loads

The residential version of Standard 152 has managed to avoid the need to evaluate heating and cooling loads. It is not clear that a small commercial version would be able to do the same. The design calculations in the residential standard are based on steady-state assumptions, while the seasonal calculations are based on analyses performed on generic residential buildings in several climates. In the small commercial case, high internal loads and variable occupancy schedules might make it necessary to account for such variations that could impact both the design and seasonal distribution efficiency values. Much of this work might be done via computer simulation, perhaps using the same computer programs that have been applied to the residential case. Field surveys of small commercial buildings of various types could also be done to assess the need to specify the intended use of the building in any distribution efficiency test method.

3.2.5. Specification of equipment efficiency factors

The equipment efficiency factor Fequip is used in Standard 152 to account for any impact of the distribution system on the efficiency of the equipment. In the residential version of the standard, Fequip is set equal to 1.0 for the most common types of equipment, namely single-capacity furnaces and air conditioners. Two categories of equipment, however, were seen as requiring explicit formulas for non-unity values of Fequip. These are heat pumps with electric-resistance backup heat and variable-capacity air conditioners. The reason that Fequip differs significantly from 1.0 in these cases is that the efficiency of the duct system affects the relative amounts of time that the equipment spends in its higher efficiency mode (compressor heating and low-speed operation, respectively) compared with its lower efficiency mode (resistance backup and high-speed operation, respectively). An inefficient duct system forces the equipment into its lower efficiency mode more often, compounding the impact of the lower duct efficiency by itself. Thus, values of Fequip for these systems are generally less than 1.0.

In the small commercial area, there are types of equipment and applications not usually found in residential applications. Is the efficiency of a rooftop unit affected by the duct system more than say, a residential furnace? Is the impact of an economizer cycle affected by the state of the duct system? These and other aspects of equipment-duct interactions have not been extensively explored, and would need to be before any small commercial test method could be promulgated.

3.3 Research Needs

The need for research in the small-commercial area is probably the greatest of any of the three areas discussed in this report. The following list summarizes what are perhaps the most pressing needs:

- Develop a logical approach toward the inclusion of the impacts of ceiling-space configuration on overall system efficiency. (This will include a consistent treatment to sort out which of these impacts are appropriate to charge against or credit to the distribution system.)

- Develop the algorithms needed to include the impact of subsystems designed to bring in or expel across the envelope amounts of air that are significant in relation to natural airflow rates and system fan flow rates.

- Develop or modify airflow measurement techniques and duct leakage test methods applicable to the larger airflows in small commercial buildings.

- Quantify the need to determine loads,including internal loads and variable occupancy schedules.

- Determine the conditions under which equipment efficiency factors other than 1.0 would need to be used in small-commercial systems.

4.0 IMPROVEMENTS TO THE RESIDENTIAL STANDARD

The third area identified for possible extension/improvement of ASHRAE Standard 152 is a future upgrade to the current residential standard.

4.1 Why improvements are desirable

A common occurrence during the development of Standard 152 was a recognition that compromises would have to be made if anything was to be accomplished within a reasonable time frame. These compromises were generally of two kinds:

- Acceptance of an approximate treatment (or what more vocal critics might sometimes characterize as "wrong") of a relevant phenomenon, for which no more nearly exact solution was available. The most important items in this category are almost certainly duct leakage test methods and the method of calculating buffer-zone temperatures.
- Agreement to defer inclusion of an aspect of residential thermal distribution efficiency because no one was able to provide a sufficiently credible treatment of it. The salient example here is continuous fan operation.

The following discussion provides a list of suggested areas for improvement in the current standard, within the context of its scope, namely residential thermal distribution efficiency.

4.2 Better Duct Leakage Tests

There is general agreement that a duct leakage test, more accurate than the current fan-pressurization test yet requiring no more time to do, would be highly desirable. There are currently two leading candidates: Delta Q and Nulling. The Delta Q test, developed by researchers at Lawrence Berkeley National Laboratory, uses a blower door to measure the airflow into (or out of) the envelope needed to maintain ten different house pressures under two conditions, system fan on and system fan off. These data are fed into a least-squares fitting routine, the output of which is putative values of supply and return leakage to/from outside under normal operating conditions. The Nulling test, developed by researchers at Ecotope, Inc., of Seattle, uses a calibrated fan (blower door or duct blower) to reset the house pressure, with the system fan on, to what it was with the system fan off. To get both supply and return leakage values, the test must be done in two stages, one with the as-found system and one with a temporary airflow path that bypasses the return duct.

Two major areas of concern have been raised with respect to the Delta Q test. The first is whether the test has an inherent bias. Research on this question is ongoing. The other question is the extent to which it is desirable (and feasible) to obtain a better estimate of the duct pressures than simply assuming some set fraction of the measured pressures at the plenums. Two approaches to this have been suggested, one involving the use of a mathematical algorithm to extract information on the leakage pressures from the Delta Q data themselves, and the other using independently measured values for the leakage hole size to iterate a better solution. Research on these possibilities is also underway.

The Nulling test appears to provide results that are generally quite accurate. However, it also tends to be time consuming. Even the developers of the test agree that it is likely to remain primarily a research tool rather than a viable diagnostic test for use by technicians in the field. One suggestion, however, has been made that may allow this test to find a diagnostic use. This is duct systems that lack returns, such as are typical in manufactured housing. For a system with only supply ducts, the first part of the Nulling test, which provides the difference between the supply and return leakage rates, is relatively easy to do. If there are no return ducts, then the first part of the Nulling test gives the supply leakage directly and the more difficult second part need not be done.

The upshot is that an advanced version of Standard 152 might have a variant of the Delta Q test for use with systems having supply and return ducts and the Nulling test for systems with supply ducts only. However, since both the Nulling and Delta Q tests require that the house envelope be completed, the fan pressurization test might be retained for use in testing ducts in new construction that has not yet been closed in.

Field and laboratory evaluations of these tests can be found in Andrews 2002, Francisco et al. 2002 and 1998, Francisco and Palmiter 2001, and Walker et al. 2002.

4.3 Interactions with Buffer Zones

A great deal of work on the question of rating the temperatures in buffer zones (i.e., areas outside the conditioned space in which ducts are located) has been done by Paul Francisco and Larry Palmiter of Ecotope, Inc.(Francisco and Palmiter 1999, 1998, 1997). One important issue concerns the choice of whether to calculate a baseline temperature assuming no ducts are present, and then to adjust the thermal regain fraction to account for the fact that ducts will alter this temperature, or alternatively to attempt to rate the most likely actual temperature in this space with the ducts in place. Other concerns relate to the question of what to do when more than one type of buffer space is present. The equations relating to this issue can get quite complicated, and the research community is far from unanimous on the best approach.

Related to this issue is the method for calculating the thermal regain fraction in both the heating and cooling mode. Significant advances were made on this during the development of the current standard, such as the recognition that thermal regain is affected by supply-side conduction and leakge and by return-side conduction, but not by return-side leakage. Related to this is the question of whether default values for thermal regain should be included in the standard. The argument in favor of this is that thermal regain is notoriously difficult to calculate "from scratch." The argument against is that default values are likely to be significantly in error.

The entire area of interactions between ducts and buffer zones is probably of roughly equal importance to the need for improved duct leakage test methods.

4.4 Continuous Fan Operation

Next in probable importance is the question of impacts on distribution efficiency of various ventilation strategies, of which continuous fan operation is probably the salient example. Various reasons have been advanced for operating the system fan other than when the furnace is on (or soon after shutdown, for purging). These generally relate to schemes for improving ventilation. The problem with such strategies is that they carry a significant energy penalty, both in terms of fan-motor power and increased air leakage in the duct system. Research to investigate this issue by means of computer simulation has recently been completed, and field studies are also underway. Impacts of other ventilation strategies, such as the use of an outside-air branch duct into the return, may also need to be addressed.

An ASHRAE research project (RP-1165) used computer modeling to investigate the impact of continuous fan operation on system efficiency (Gu et al. 2003a). The authors found that continuous fan operation could have a significant impact on the overall energy efficiency of the system. They developed the concepts of Energy Use Ratio and Distribution Efficiency Ratio to quantify these effects. The report should provide a basis for a first-order treatment of continuous fan and other ventilation strategies within the context of an updated Standard 152. Additional work is being performed by researchers at Lawrence Berkeley National Laboratory. When this is completed, it should provide a complementary piece of the puzzle.

4.5 Duct-Equipment Interactions

It is generally recognized that duct design can influence the efficiency of heating and cooling equipment, and also that equipment operating parameters can affect on the efficiency of the duct system. These two interactions are handled very differently in Standard 152. The impact of duct design on equipment efficiency is characterized by an equipment efficiency factor, Fequip. This is defined as the ratio of equipment efficiency when connected to a "perfect" duct system to the equipment efficiency with the actual duct system. Less-than-perfect ducts can improve equipment efficiency (usually by just a percent or two) because as duct losses increase the load increases, causing the equipment to cycle less frequently. In the other direction, variable-capacity equipment can experience a significant drop in efficiency as duct losses increase, because the increased load causes the equipment to run in its high-capacity mode (which is generally the less efficient mode) a greater fraction of the time. This effect applies to heat pumps with electric resistance backup as well as to two-speed air conditioners.

On the other hand, the impact of equipment design on duct efficiency is captured within the delivery effectiveness algorithm without needing a special factor to account for it. This is because the equipment's operating characteristics influence the duct system by means of parameters such as airflow rates and pressure drops that are already accounted for in this algorithm. A modulating furnace, for example, is expected to incur a lower distribution efficiency in the low-capacity mode—unless the ducts are located in the conditioned space—because the longer residence time of the air in the ducts increases conductive losses to the buffer space.

Computer modeling of the influence of duct design parameters on equipment efficiency factors has been carried out in ASHRAE research project RP-1165 (Gu et al. 2003b). Its results are only in rough agreement with the algorithms for Fequip currently in Standard 152.

For heat pumps, as an example, the formula for Fequip in Standard 152P is linear, with DE as the independent variable. Gu and coworkers found that the actual relationship was more complex and that the product of DE and Fload would make a better independent variable.

The question of whether and to what extent the existing formulas for Fequip should be modified is open.

4.6 Duct Conduction Parameters

Two factors influencing thermal conduction losses in duct systems have been discussed above: thermal regain and duct-to-buffer-zone temperature differences. Two other parameters are also important: duct surface area and duct insulation R-value. Recognizing that measuring the surface area of a duct system in real houses is time consuming, the Standard 152 committee included a default option for estimating the surface area on the basis of information on house size and number of return registers. This algorithm is based on data collected by researchers and reported in the ASHRAE Transactions. It is recognized, however, that the estimates of duct surface area obtained in this way are of necessity far from perfect. The question poses itself, therefore, whether an effort to improve the default algorithm may be justified. The alternative, requiring direct measurement in every case, would probably severely limit the utility of the standard in all but research-oriented situations.

The question of estimating duct insulation thermal resistance values is probably of lesser difficulty. Usually the nominal R-value is imprinted on the insulation, if present. Alternatively, a trained technician should be able to tell the difference between uninsulated ducts, R-4, and R-8. There is a question of whether the actual R-value is truly represented by the nominal value. Probably the actual value of "R-4" is somewhat lower that that, and there is some evidence that "R-8" may sometimes be significantly less than its nominal value.(Andrews 2003) Although this does not appear to be a major issue, some refinement of the specifications of insulation values for ducts may be warranted.

4.7 Infiltration and Off-Cycle Losses

Currently, Standard 152 accounts for thermal losses caused by impacts of the duct system on air infiltration through the house envelope. It also accounts, in a crude way, for off-cycle losses. Significantly, losses caused by air infiltration through duct leaks during the off-cycle is not considered. Thermal mass effects—heating or cooling of the ducts during the off cycle—is accounted for only roughly, with a penalty factor of 2% or 5% on seasonal distribution efficiency for non-metallic and metallic ducts, respectively. Thermosiphoning losses have also received a fair degree of attention by researchers but are not included in the current standard. The open questions here are, first, to what extent a better algorithm for on-cycle infiltration impacts may be needed and, second, whether any effort should be expended on better treatments of off-cycle losses.

4.8 Zoning Impacts

The final area for possible improvement in Standard 152 as it relates to forced-air systems can be summed up under the rubric of "side effects," i.e., effects caused by operation of the system other than in a standard mode. One such operating variation, continuous fan operation, has already been discussed because of its likely importance. Other operational variations would include internal pressurization from isolation of zones, i.e., the impacts of zone pressurization caused by door closings, and the effects on energy efficiency of zoning strategies.

4.8.1 Isolation of Zones

The effect of forced-air distribution systems on the pressure distribution within a house has long been a subject of research and discussion. It is generally recognized that systems in which certain rooms have supply registers but no returns and that, moreover, can be isolated from the rest of the house, are likely to have higher rates of air change with the outside than systems in which such imbalances do not occur. Whether and to what extent an accounting of such effects should be included in an updated standard is an appropriate subject for discussion.

4.8.2 Impacts of Intentional Zoning Strategies

Zoning strategies usually have one or both of the following objectives: to improve thermal comfort and/or to save energy. These two objectives are intertwined and, at least to some extent, in opposition. Therefore, there are likely to be problems incorporating zoning impacts into a test method such as ASHRAE Standard 152 that does not include thermal comfort within its purview. It is possible that zoning strategies intended to save energy without significantly compromising thermal comfort could be included. It might, however, be better to leave zoning impacts within the area of thermal distribution efficacy, as discussed in Section 2 of this report.

4.9 Additional System Types

Finally, the question of what system types besides forced air should be added to the standard needs to be addressed. All through the public review process, several commenters lamented the fact that this or that system was not included in the standard. Forced-air systems received most of the Standard 152 committee's attention because they have captured most of the market in American housing. Hydronic heating systems were included because they appear in a significant fraction of existing homes in one region of the country (the Northeast). Radiant heating systems, whether based on hot water or electricity, were included as well, and the algorithm was extended to include sensible cooling using hydronic radiant panels. Electric baseboard systems were also included, not only because they are common in some regions of the country but also because their treatment was particularly simple. Two additional system types have been proposed by significant numbers of formal and informal commenters: hydronic cooling other than radiant-sensible, and refrigerant distribution systems (mini-splits). Detailed calculations in support of a section on the latter have been done (Andrews 2001a, 2001b), and a proposed section for the standard has actually been written. The question of whether additional system types should be included in the next version of the standard needs to be addressed.

4.10 Research Needs

In many cases, the research needed to upgrade Standard 152 in the residential area has already been done. What remains to be done is to develop simplified algorithms to reflect these latest results that: 1) significantly upgrade the accuracy of what is in the current standard; 2) do not add significant difficulties in performing the tests; and 3) attract a general consensus that they should be added to the standard.

The areas where such upgrading can be done without much additional research include:

- Inclusion of Delta Q and Nulling test methods, either as alternatives or as replacements for the existing duct pressurization test.
- Review of the current method of accounting for buffer-zone interactions and the literature on this topic to achieve a step increase in the accuracy of the algorithm used to account for these effects.
- Review of the research on continuous fan operation conducted by FSEC, LBNL, and others to arrive at an acceptable algorithm for including its effects in the standard.
- Updating, in light of recent research, of the formulas for equipment efficiency factors for heat pumps and multi-speed air conditioners.
- Improvement of the treatment of infiltration and off-cycle losses to include off-cycle leakage impacts of ducts and more precise quantification of off-cycle thermal-mass effects than the current 2%/5% penalty on seasonal distribution efficiency.
- Development of algorithms needed to bring hydronic cooling and refrigerant distribution within the purview of the standard.

A few areas still need further research. These include:

- Determination of the extent to which uncritical use of nominal insulation thermal resistance values may cause errors in thermal distribution efficiency.
- Further work on relationships between duct surface area and other, more easily measured, parameters.
- Quantification of the energy impacts of zone isolation.
- Quantification of the energy impacts of intentional zoning strategies.

5.0 SUMMARY AND CONCLUSIONS

No consensus has been reached to date on which of the above three areas is most important to pursue. Yet it is unlikely that sufficient effort can be brought to bear to push all three forward to the point of acceptance by ASHRAE within a time horizon of three to five years. As a result of discussions held at the June, 2003 ASHRAE meeting in Kansas City, a compromise approach has been settled upon. In this approach, work on these aspects would be divided into separate subcommittees, so that people with an interest in one area but not others would not have to sit through discussions that are of no interest to them. The plan would be as follows:

1. Upon the approval by ASHRAE of the current Standard 152P (thus removing the "P"), TC 6.3 would consider petitioning ASHRAE to reestablish a Standard 152 committee to consider possible upgrades to the current residential standard. This kind of maintenance and continuing upgrade is common practice for other standards, and does not reflect in any way a judgment that the current standard is inadequate, only that it is, like other standards, far from perfect. Upgrades designed to improve accuracy, extend the scope of the standard to additional distribution system types, and to reduce the level of effort needed to do the tests would all be included within the purview of the upgrading process.

2. Use the newly established Standards Subcommittee of TC 6.3 as a venue for discussion on possible new standards (or new sections of the existing standard) relating to thermal distribution efficacy and to small-commercial thermal distribution efficiency.

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