A HISTORY OF ASHRAE STANDARDS 152P

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ABSTRACT

The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) has been developing a standard test method for evaluating the efficiency of ducts and other types of thermal distribution systems in single-family residential buildings. This report presents an overview of the structure, function, and historical development of this test method.

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EXECUTIVE SUMMARY

The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) has been developing a standard test method for evaluating the efficiency of ducts and other types of thermal distribution systems in single-family residential buildings. This report presents an overview of the structure, function, and historical development of this test method.

The idea for a duct efficiency test method grew out of research, performed during the 1980's and early 1990's, which found astonishingly high energy losses in residential duct systems. These losses have been measured using three quite different methods. The fact that all three avenues of investigation agree lends significant weight to the credibility of the conclusion.

In 1990, the U.S. Department of Energy (DOE) commissioned a study to quantify the energy losses from thermal distribution systems on a nationwide basis and to estimate the potential for energy savings inherent in improving the efficiency of these systems in residential and small commercial buildings. The study projected that by 2020, annual energy savings of 0.87 quads could be obtained using currently available technologies, and that with improved technology, 2.11 quads could be saved each year.

Given this large potential, the DOE identified three main areas of effort by which it might assist the building and HVAC industries to achieve these savings. Three possibilities were identified:

- New technology development
- Information dissemination
- Development of a test method for thermal distribution efficiency.

Although significant progress has been made in the first two areas, the last has been the centerpiece of the program. There were several reasons for this. First, limited funding required the setting of priorities. Second, it was feared that new technology might languish if there was no way to prove its efficacy. Third, at the time there wasn't enough reliable information to make its dissemination the main goal. Fourth, the impact of test methods for equipment efficiency was recognized, and it was hoped that a thermal distribution efficiency test method could play a similar role.

Accordingly, the DOE worked with the American Society of Heating, Refrigerating, and Air-conditioning Engineers (ASHRAE) to develop the test method. In 1993, ASHRAE approved the formation of Standards Project Committee 152P (SPC152P) with the goal of developing the test method.

The stages in the development of ASHRAE Standard 152 can be summarized briefly as follows:

- Development of a conceptual framework for the standard, keyed to a definition of thermal distribution efficiency that accounts for system interaction effects as well as direct thermal losses.
- Agreement that two separate figures of merit, one reflecting design conditions and the other seasonal-average conditions, should be reported for heating and for cooling separately.
- Development of a “first-cut” version of the test method that included all of the basic building blocks found in later versions, though in unrefined form.
- Winnowing out of certain kinds of tests, in the interest of simplifying the procedure. In particular, it was decided not to try to measure the degree to which the system provides thermal comfort. An optional testing approach called the “research pathway” was eliminated to simplify the test method and avoid ambiguity.
- Sharpening of remaining issues to be considered by the committee. Salient examples of these were the tradeoff between accuracy and ease of use, whether and to what extent to allow the use of default values in lieu of measured quantities, what types of distribution systems (besides forced-air) would be included in the test method, the definition of what constitutes “conditioned space,” and the impact of ducts on air infiltration.

Most of the effort of SPC152P was focused on forced-air systems. This was motivated by the fact that these systems have achieved greater than 90% market share in new housing. The main tasks were the development of equations for delivery effectiveness, specification of how to measure air leakage from ducts, how to measure system airflow, how to quantify conductive heat losses, the treatment of temperature and humidity in the “buffer zones” in which the ducts are located, treatment of interactions between the ducts and the equipment, and interactions between the ducts and the building.

Early on, it was decided to include baseboard hot-water heating systems in the test method. This was done for several reasons. These “hydronic” systems, although fading from the picture nationally, retain a strong market niche in the Northeast. The development of a test procedure and an algorithm for calculating distribution efficiency began with the construction of a computer model for these systems. Elements of the model were then incorporated into the algorithm. Field testing determined that some of the intended measurement protocol was unworkable, and this deficiency was repaired. Finally, the algorithm was reviewed in detail by several members of the committee, after which it was, with some corrections, incorporated into the draft standard.

Consideration was given to the possible inclusion of other distribution system types. The hydronic section was expanded to include radiant flat-panel heating and sensible cooling, but not latent cooling. The development work needed for a section on refrigerant distribution systems (i.e., “mini-splits”) was completed, but was not incorporated into the

draft standard. Electric baseboard systems were simple to include, and so a section on these was added. Other suggested system types were not included, in part because of time limitations.

By 1996, a majority of the committee felt that the standard was ready to be submitted to the ASHRAE public review process. However, there was enough dissent that it was deemed advisable to consider several constructive suggestions for improvement and re-vote. In addition, instead of a traditional public review, a relatively new ASHRAE option for "public review and trial use" was requested instead. The merit of this was that it put the developing standard "on the table" in an official way and yet permitted changes to be made on a continuing basis. A motion to submit the standard for public review and trial use passed unanimously at the January, 1997 ASHRAE Winter Meeting.

Six months later, at the next ASHRAE meeting in Boston, the standard had undergone enough changes that another public review ballot was deemed necessary. Among these were minor technical changes in the Title, Purpose, and Scope (which had to be formally balloted by the committee and approved by the ASHRAE Standards Committee), as well as certain technical details, such as error allowances in the hydronic system measurements and limitations on conditions under which the House Pressure Test for duct leakage would be allowed. The upshot was the committee once again voted on a public-review draft of the standard, recommending (unanimously) public review and one year of trial use.

There were some additional delays, in part caused by staffing problems in ASHRAE and in part due to continuing efforts by the committee to refine the standard, but in 1999 a First Public Review Draft was published by ASHRAE, and comments were formally solicited. These comments were addressed by the committee and for the most part resolved.

However, three of the commenters on the First Public Review remained unresolved. Some members of the Standard 152P committee itself also had issues they felt still needed to be addressed. It was agreed, therefore, that a Second Public Review would be needed. This occurred in the summer of 2001.

The comments from the Second Public Review were addressed by the committee, and all but one commenter were resolved. The changes made to resolve most of the comments required a Third Public Review. This differed from the first two in that only the changes were subject to review, not the whole standard. In January, 2003, ASHRAE reported that the third public review had resulted in no comments.

As of this writing (October 2003) ASHRAE is considering the formal request by the committee for approval of the test method as an official ASHRAE Standard.

1 INTRODUCTION

In 1993 the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) initiated the development of Standard 152P, whose current title¹ is "Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems." Research over the preceding decade had shown that residential forced-air thermal distribution systems, i.e., ductwork, were far less efficient than had been assumed. The need was seen for a nationally recognized test method that could be used both to encourage and to verify improvements in thermal distribution efficiency.

The purpose of this history is to make available the details of how and why the test method developed as it did, to assist in the public review process and to provide an information base for any future revisions of the standard. The text will speak of "Standard 152" when the intended final product is meant, and "152P" when the subject is the current work in progress.

2 GENESIS OF ASHRAE STANDARD 152

This section discusses the formative period of Standard 152's development, from the time it was first proposed around 1992 until early 1996, when, with the abandonment of the "pathway" approach, the proposed standard assumed roughly its present form.

2.1 Evidence that Ducts Are Not Efficient

The idea for a duct efficiency test method grew out of research, performed during the 1980's and early 1990's, which found astonishingly high energy losses in residential duct systems. It is worth reviewing the evidence for this, since the value of all subsequent effort hinges on the proposition that duct energy losses are real.

The energy losses from duct systems have been measured using three quite different methods. The fact that all three avenues of investigation agree lends significant weight to the credibility of the conclusion. The three methods are:

- Measurement of individual aspects of heat and mass transfer in ducts, such as air leakage rates and temperature distributions, with calculation of the cumulative effect of these parameters on thermal distribution efficiency. This may be called the "bottom-up" approach.
- Measurement of the energy required to heat a residence using the as-found duct system, compared with the electrical energy needed to heat the same building with distributed space heaters (electric co-heating). Any difference between the two, after allowing for heat losses from the furnace and any variations in weather conditions between the times when the two tests are conducted, is attributed to energy losses from the duct system. This may be called the "top-down" approach.

¹ The original title had "Steady-State" in place of "Design" and the word "Standard" in front of "Method of Test."

- Comparison of a large sample of homes heated with electric furnaces and duct systems with another large sample heated with electric baseboard heaters. After correcting for any systematic differences between the two samples (e.g., average conditioned floor area) any remaining excess energy requirements in the houses with ducts is attributed to energy losses from the duct systems. This may be called the “statistical” approach.

The bottom-up method is by far the most commonly applied technique. At the time Standard 152 was being conceived, there was already a significant body of work (Caffey 1979; Cummings and Tooley 1989; Cummings et al. 1990; Gammage et al. 1984; Matthews et al. 1990; Modera 1989; Palmiter and Bond 1992; Parker 1989; Proctor and Pernick 1992; Robison and Lambert 1989). This work generally found seasonal-average energy losses in duct systems located in attics and vented crawl spaces in the range of 25% - 40%. In the next several years, additional work was done along similar lines (Andrews 1996; Davis et al. 1996; Jump et al. 1996; Proctor 1998; Saunders et al. 1993; Strunk et al. 1996 and 1997; Treidler and Modera 1996; Walker and Modera 1998). Results have been consistent with the earlier work.

The top-down method has been employed by a much smaller group of researchers, with the most extensive work having been done in the Pacific Northwest by researchers at Ecotope, Inc. of Seattle. The only definitive study (Olson et al. 1993, Palmiter and Francisco 1994) that had been done at the time of initial planning for Standard 152 was done on a sample of 24 homes. It found an average thermal distribution efficiency of 71%, with a low of 50% and a high of 82%. In contrast, two homes with ducts in the conditioned space had thermal distribution efficiencies of 97% and 99%. Later coheating work by this same group and by others (Andrews et al. 1996, Francisco and Palmiter 2000b, Strunk 2000) was consistent with these original findings. A variation of the top-down method compares the as-found system not with electric co-heating but with the same system after the ducts have been moved into the conditioned space. Guyton (1993) obtained seasonal energy savings in two Maryland townhouses of 34% for heating and 71% for cooling after this was done.

The statistical method has been the least widely applied technique, in part because of the need for a large sample of houses. Two companion studies (Lambert and Robison 1989; Parker 1989) found that for a sample of 200 conventionally built homes in the Pacific Northwest, half with electric furnaces and the other half with electric baseboard heating, the results, after normalization for the size of the house, implied thermal distribution losses of approximately 25%.

A survey of duct research done at the inception of development of Standard 152 (Andrews 1992) provides some additional references from the 1980's.

2.2 Energy Savings Potentials

In 1990, at the same time that the idea behind Standard 152 was beginning to be formulated, the U.S. Department of Energy commissioned a study (Andrews and Modera

1991) to quantify the energy losses from thermal distribution systems on a nationwide basis and to estimate the potential for energy savings inherent in improving the efficiency of these systems. The scope of the study included residential and small commercial buildings, but not large commercial buildings. The upper limit on floor area for “small” commercial was taken as 10,000 square feet. The study included all types of thermal distribution, not just forced air, but ducts were the main focus of the study since they were (and still are) the major opportunity for energy savings.

The approach used in the study involved four steps. The first step was to categorize the building stock in relatively homogeneous “cells” so that the possible improvements in thermal distribution could be evaluated and the resulting energy savings estimated for each cell separately. The cells were defined by a classification scheme based on variables relevant to thermal distribution:

- I. Construction Status
 - 1. Existing
 - 2. New
- II. Climate Zone
 - 1. “Frostbelt” (Northeast and Midwest Census Regions)
 - 2. “Sunbelt” (South and West Census Regions)
- III. Building Type
 - 1. Single Family
 - 2. Manufactured Home (HUD-Code)
 - 3. Multifamily
 - 4. Small Commercial
- IV. Thermal Distribution Type
 - 1. Forced Air
 - 2. Hydronic
 - 3. Built-In Electric
 - 4. Other or None
- V. Thermal Distribution Location
 - 1. In Unconditioned Space (Crawlspace or Attic)
 - 2. In Partly Conditioned Space (Basement)
 - 3. In Conditioned Space

This meant there were $2 \times 2 \times 4 \times 4 \times 3 = 192$ cells to consider. Fortunately, only a small fraction of these were important.

The second step estimated the energy use per household (residential) or unit floor area (small commercial). The third step projected the building stock out to the year 2020, with pre-1995 buildings classified as “existing” and post-1995 buildings as “new.” The reason for choosing that year was that advances in thermal distribution appropriate for new construction but not retrofit might start to make significant inroads into the market by that time. This was, perhaps, optimistic, but at the time it seemed reasonable to the authors of the study. The fourth step was to estimate energy savings from improved

thermal distribution as percentages of the annual energy use in 2020 for each cell with a significant population of buildings.

The energy savings potentials were reported in three categories. Current energy savings potentials were defined as those achievable using technologies that were already developed and understood, at least within the research community. Full savings potentials projected the possible benefits of new technology development. A third category, Undetermined potentials, was used to designate cells where little more than an educated guess was possible. These were some of the multifamily buildings and all the small commercial stock. The current savings potentials summed to 0.87 quads (1 quad = 10^{15} Btu). The full savings potentials summed to 2.11 quads. In each case, 88% of the savings was projected for forced-air systems, with the remainder for hydronic systems. For what it was worth, the undetermined potentials summed to 0.27 quads, although the uncertainty in this number was acknowledged to be very high. This study was the basis of a statement that became common in later years, namely that the annual energy savings potential for thermal distribution improvements in small buildings is between 1 and 2 quads.

2.3 The Need for a Standard Method of Test

Given that the potential energy savings from improved thermal distribution summed to a very large amount of energy nationwide, the next question for the DOE and its national laboratories was, "What, if anything, should the Federal government do about it?" Aside from the obvious possibility "Nothing," other suggestions included the development of new technologies and the development and dissemination of good information about how to "do ducts right." Over the years, significant effort has been undertaken in both of these areas. However, early on it was decided to make the centerpiece of the program the development of a national standard for determining the efficiency of ducts (and perhaps other kinds of thermal distribution systems as well). There were several reasons for this:

- Limited funding, caused by the lack of a separate line item for thermal distribution in the DOE budget, made it impossible to do everything.
- New technology, while potentially useful, might languish "on the shelf" if there were no accepted way to prove its worth.
- Information dissemination was not seen as a major driving force at the time, mainly because the amount of specific information available was limited.
- Test methods for equipment efficiency helped industry to market better products. A thermal distribution efficiency test was seen as playing a similar role.

It was recognized that no test method could cover all kinds of buildings. Large commercial buildings are so different from single-family residences that to try to develop a test method that would cover everything was simply not feasible. Small commercial buildings were within the scope of the energy savings potential study discussed above, and there was some discussion of whether to include them in the test method. In the end, it was decided to restrict the scope to residential buildings, largely because more was known about the magnitude of the energy losses in these buildings.

A nationally recognized method of test was seen as having several possible uses:

- Diagnosis of duct systems in individual homes, to determine which systems need repair and what repairs to make.
- Statistical sampling of generic approaches to duct design and construction, such as ducts in the conditioned space, various sealing methods, or inherently leak-free ducts.
- Parametric analysis of impacts on thermal distribution efficiency of changes in duct design and of equipment options such as variable capacity and fan speed.

With these expected benefits, planning for a new standard method of test was initiated in 1991. ASHRAE Technical Committee 6.3 (Forced-Air Heating and Cooling Systems) petitioned the ASHRAE Standards Committee to form a new subcommittee charged with developing such a test method. In the petition, the title, purpose, and scope were defined as follows:

Title: Standard Method of Test for Determining the Steady-State and Seasonal Efficiencies of Residential Thermal Distribution Systems

Purpose: A method of test to determine the efficiency of space heating and/or cooling thermal distribution systems under seasonal and steady-state conditions. The objective is to facilitate annual energy calculations and heating and cooling system capacity calculations for existing systems and new system design.

Scope: The standard was to apply to single-family detached and attached residences with independent forced-air, hydronic, radiant, and refrigerant distribution systems.

This was approved in 1993, and the committee, designated as Standards Project Committee 152P (SPC152P) conducted its first official meeting at the ASHRAE 1994 Winter Meeting in New Orleans, on January 23 of that year. In 1997 the word "Standard" was dropped from the title, to remove any misconception that this might be a prescriptive standard. The "Steady-State" in the title was changed to "Design," to eliminate confusion between the two terms. Finally, refrigerant distribution systems were removed from the scope, and electric distribution systems were added.

2.4 Conceptual Framework of Standard 152

It might seem that defining what is meant by thermal distribution efficiency should be a fairly simple task, but this was not the case. The most straightforward definition would be the obvious one based on an output-to-input ratio, namely the amount of heat or cooling energy delivered by the duct system divided by the amount provided to the ducts by the equipment. This concept is actually used in Standard 152 under the name "delivery effectiveness," but it is not considered the final "figure of merit" because it does not capture the impacts that the thermal distribution system may have on the equipment efficiency and on the building heating or cooling load.

The efficiency of a duct system can significantly affect the efficiency of the equipment if the equipment has two or more operating modes that differ in both capacity and

efficiency. For example, a heat pump with electric resistance backup has a high-capacity, low-efficiency mode (resistance heat) and a lower-capacity, higher-efficiency mode (compressor heating). A duct system that loses a lot of energy will cause the heat pump to go onto resistance backup at a higher outdoor temperature than one that does not have these losses. The seasonal-average efficiency of the heat pump will therefore be lower with the less efficient duct system. Furthermore, the energy penalty of this effect will be in addition to the energy lost in the duct system itself.

Thermal losses from a duct system can affect the building heating or cooling load if the ducts are located in a "buffer zone" that is thermally linked to the house. In that case, some of the losses may be "regained." An example would be heat losses to a basement, which warm the basement and thereby reduce the transfer of heat from the house to the basement. Pressure differences caused by unbalanced duct leakage can also affect the whole-house air change rate. This usually increases the building space-conditioning load, either by increasing the rate of air infiltration through the envelope, in the case of supply-dominant leakage, or by bringing in more outside air into the duct system directly, in the case of return-dominant leakage. (Occasionally, if the return-side leakage is slightly greater than the supply-side leakage, the overall air-change rate can be reduced by the effect of the leakage.)

Clearly, an alternative definition of distribution efficiency was needed, one that could account for these system effects. Fortunately, much of the needed work had already been done in the 1980's in connection with another ASHRAE project designated SP-43. This project, performed by Battelle-Columbus Laboratories under the auspices of ASHRAE Technical Committee 6.3 and funded by the U.S. Department of Energy, the Gas Research Institute, and ASHRAE, had as its major objective the determination of whether the standard test method for furnace efficiency known as Annual Fuel Utilization Efficiency (AFUE) accurately reflected the system impacts of changes in furnace efficiency. That is, the question was asked whether improving the efficiency of a furnace from the then-common value of 65% to, say, 95% would have a proportionate effect on system efficiency and actual fuel use by the homeowner. After nearly a decade of effort, the answer was unequivocally "yes" with the exception of the credit for vent dampers, which were found to be overrated.

The relevance of this for duct efficiency was that in order to pursue their objective, the researchers on this project had to define carefully all of the various energy flows in the system, including the ducts. The results of the SP-43 Project continue to be useful today and are available for study in Chapter 9 of the ASHRAE HVAC Systems and Equipment Handbook (ASHRAE 2000). Further details on results from this project are given in several published ASHRAE papers (Crisafulli et al. 1989; Fischer and Cudnik 1993; Herold et al. 1987a and 1987b; Jakob et al. 1986a, 1986b, and 1987; Locklin et al. 1987).

However helpful the SP43 project was, however, its primary focus was on furnace efficiency, not on duct efficiency. It therefore remained necessary to develop a suitable definition of thermal distribution efficiency that would take into account all of the interactive effects between the ducts, the equipment, and the load. It was also considered

desirable that the definition should be applicable to any kind of thermal distribution system, such as hydronic, radiant, or refrigerant distribution, not just forced air.

In light of these considerations, the following definition of distribution efficiency was proposed in 1992 at a meeting of building science researchers (Modera et al. 1992):

$$\text{Distribution Efficiency} = \frac{\text{Purchased Energy to Heat or Cool With No Ducts}}{\text{Purchased Energy to Heat or Cool With Ducts}} \quad (1)$$

“No Ducts” was shorthand for a hypothetical “perfect” duct system that did not gain or lose any energy via heat or mass transfer, and did not affect the efficiency of the equipment or the heating or cooling load. “Ducts” means the actual duct system under test.

“This is well and good,” one might say, “but how can one evaluate the numerator, which is based on a contrary-to-fact situation?” The response is that this equation is not intended to provide a direct means of determining distribution efficiency, but rather that it needs to be broken down into parts that can be evaluated. Discussion in Modera et al. 1992 and, from a slightly different perspective, in Andrews 1994, showed how the definition of distribution efficiency reduces to a product of factors:

$$\eta_{\text{dist}} = \text{DE} \, F_{\text{load}} \, F_{\text{equip}} \quad (2)$$

where η_{dist} is the distribution efficiency, DE is delivery effectiveness, F_{load} is the load factor, and F_{equip} is the equipment efficiency factor. Delivery effectiveness² has been defined above. The load factor is the ratio of the heating or cooling load in the absence of ducts to the load with the actual ducts in place. The equipment efficiency factor is the ratio of equipment efficiency with the actual ducts to the equipment efficiency in the absence of any impact from the duct system. (Later versions of Standard 152P have split off a “recovery factor” F_{recov} that deals with thermal regain, leaving the load factor to handle the infiltration impact.)

If the ducts cause the heating or cooling load to increase, then $F_{\text{load}} < 1$. Similarly, if the ducts cause the equipment efficiency to decrease, $F_{\text{equip}} < 1$. Under either of these undesirable conditions, F_{load} and F_{equip} are in effect “penalty factors” that assesses the ducts for their negative impacts on the rest of the system. Note, however, that F_{load} and F_{equip} can also be greater than one, i.e., η_{dist} can exceed DE. A commonly found case where F_{load} is significantly greater than one is a poorly insulated duct system in a basement. A portion of the heat losses from the ducts will have a beneficial effect, e.g. by warming the volume of air near the basement ceiling, thereby retarding conductive

² Actually, it is anachronistic to speak of “delivery effectiveness” in the 1992-94 time frame. Modera et al. called it “nominal distribution efficiency,” while early versions of Standard 152P used the term “delivery efficiency.” In order to avoid confusion, the term “delivery effectiveness” will be used throughout this discussion.

heat losses through the house floor. In such a situation, DE will generally be a quite low number, so the fact that F_{load} exceeds one does not imply that the duct system is performing well, only that it is not quite so bad as the DE value alone might seem to imply. As for F_{equip} , for conventional single-capacity equipment it may be a few percentage points greater than 1.0 because of reduced cycling losses when duct losses add to the load. For heat pumps and variable-capacity equipment, it is usually less than 1.0. The rest of the story of ASHRAE Standard 152P can be summarized as a quest for better, faster, and cheaper ways to determine values of DE, F_{load} , and F_{equip} sufficiently accurate that η_{dist} will be a useful indicator of thermal distribution efficiency.

2.5 A Question of Timing: Steady-State, Design, and Seasonal

At the outset of Standard 152P's development, it was recognized that a distinction would need to be made between system behavior under peak-load conditions and under average conditions as experienced over a heating or cooling season. The former would be used for equipment sizing and for analyzing impacts on electric utilities, while the latter would be used in annual energy and cost analyses. Thus, there would be four values of distribution efficiency for each system tested: heating design, heating seasonal, cooling design, and cooling seasonal.

At first there was some confusion between the terms "steady-state" and "design" as used in the standard, but by 1996 this was resolved by removing the term "steady-state" from the standard altogether.

After that, all calculations relating to peak-load conditions would use published data (such as temperatures and humidities) appropriate to a specified set of ASHRAE design conditions. The definition of "seasonal" was less straightforward. Initially, seasonal distribution efficiencies were based on climatic conditions averaged over the three most severe months of the heating and cooling seasons. Later, this approach was found to cause unacceptably large errors in certain climates, and the analysis was refined by using load-weighted averages over the entire season.

2.6 Present at the Creation: Inauguration of Standards Project Committee 152P

The first official meeting of the ASHRAE Standards Project Committee 152P was held on January 23, 1994 at the ASHRAE Winter Meeting in New Orleans. The technical agenda was set in terms of Task Groups within the committee that were to investigate each of five areas germane to the future development of the standard. These were:

- Building Interactions
- Equipment Interactions
- Measurement Techniques
- Computational Procedures
- Applications

A review of the discussion at this first meeting shows that it identified some major issues that have remained “on the table” for years afterward, but that it also settled some questions in ways that have remained “bedrock” features of the standard ever since.

Of the questions that were settled, three stand out. First, the definition of distribution efficiency was firmed up in the manner discussed above. The large body of previous work in the SP43 project, as well as the rationale provided in Modera et al. 1992, were sufficient to convince the committee that this fundamental question had been adequately explored and resolved.

Second, it was decided that Standard 152 would not attempt to rate thermal comfort. Rather, it would assume that an acceptable level of comfort was provided by the system, and only rate the energy efficiency at which this comfort was provided. This decision was not taken nearly as happily as the first, but instead was accepted as a matter of necessity. Even at this early stage in the development of the standard, at a time when optimism would normally be at its peak, it was recognized that trying to rate comfort would be too ambitious. Standard 152P would henceforth address energy-efficiency related issues only.

Finally, it was made explicit that Standard 152 would be purely a test method, with no attempt to be prescriptive. This was not really a decision of the Standard 152 committee but rather ASHRAE policy. Nevertheless, there has often been some confusion on this among those not closely connected to the standard itself.

Other issues were less easily dealt with. The biggest of “big picture” issues, one that has been with the committee during all of its deliberations, is the tension between the needs for accuracy and economy of use. It is fair to say that the committee has vacillated on this issue over the years, in some cases depending on the particular part of the standard under consideration and in others, perhaps, as a function of changing committee membership and attendance. All members of the committee, of course, would like to see a standard that is both accurate and quick to apply, but since these criteria are in conflict, the need for compromise has inevitably arisen. Sometimes the desire to improve accuracy has been more keenly felt, resulting in changes that would add to the time required to perform the test. At other times, the committee was motivated more by the need to speed up the test protocol enough that it would actually be used in the field.

Part of the tension between these two goals has resulted from the range of intended uses of the standard, covering both the diagnosis of individual homes (requiring speed) and of evaluating generic approaches to duct design (requiring accuracy). The initial approach to addressing this problem was to divide the standard into three “pathways,” each appropriate to an intended use:

- **Diagnostic Pathway.** For benchmarking thermal distribution systems in individual existing buildings. Based on “bottom-up” measurement methods. Must minimize time and effort.

- **Design Pathway:** For predicting performance of planned systems in buildings yet to be constructed. Intended to permit designers to specify a level of distribution efficiency, to be verified once the building was built. This also had to be fast.
- **Research Pathway:** For evaluating generic approaches to duct construction, where the need for accuracy was key and testing time was less of an issue because only a sample of the population would be evaluated. Based on “top-down” measurement methods.

A second troublesome issue has been the question of whether, and to what extent, to include “default” options in the standard. A default is a number or set of numbers that is inserted into a calculation when the required input information is missing. Defaults are legitimately used when the cost of obtaining some information exceeds its value. Clearly, however, there must be some limit on defaults, or the resulting figure of merit will be meaningless. The issue within the committee was whether to allow any defaults at all. This was intimately related to the accuracy vs. economy issue highlighted above, with those opposing defaults doing so on the basis of protecting the accuracy of the standard, and those supporting them arguing that they were needed if the standard was going to be of any practical use.

Another issue concerned the types of distribution systems to be included in the standard. It was agreed that because of their dominant position in the housing market, forced-air systems would receive most of the committee’s attention. There was also general agreement that a section of the standard should be devoted to hydronic heating systems. In principle, there was no opposition to any other systems being included as well, including refrigerant distribution (i.e., the so-called “mini-split” air conditioners), radiant heating and cooling systems, and even electric baseboards. The problem with including any of these was to identify a champion who would be both able and willing to develop and justify the needed test procedures and computational algorithms.

The precise definition of what constitutes “conditioned space” was also an item for discussion. The straightforward definition of conditioned space is any part of a building that the occupants are trying to heat or cool. Usually it is as simple as that. Nevertheless, some zones such as basements may have ambiguous characteristics. There might be no thermostat or heating registers in the basement, but it might have uninsulated ducts running through it. From this one might infer that the builder intended the basement to be heated to a temperature higher than it would have had were no ducts present, but not to the indoor set point. Is this space conditioned? Are heat gains to this space from the ducts to be considered losses?

Finally, the committee considered how to treat impacts of ducts on air infiltration. Researchers had found that operation of the system fan usually increased the air-change rate between the house and the outside, often by as much as a factor of three. Also, it was agreed that duct leaks also increased the air infiltration rate when the system fan was off, because holes in ducts add to the overall effective leakage area of the envelope. The initial consensus was that Standard 152 should account for both effects in a reasonably

accurate way, but that detailed infiltration measurements using tracer gases and the like would not be appropriate because of their cost in labor and materials.

2.7 Shaping the Standard: 1994 and 1995

As stated above, the issue of accuracy versus economy of use was addressed by distinguishing between the diagnostic and design pathways, on the one hand, and the research pathway, on the other. During the two-year period 1994-1995, the first version of the standard took shape. There are two ways to review how this process occurred. One would be to look at the minutes of the ASHRAE meetings that took place during this period. The first impression of a reviewer who was actually present at these meetings is that the minutes reflect what really happened—a meandering discussion of issues with conclusions that, in many cases, were greatly revised in following years. However, behind this scene of apparent disarray, a small group of workers put together the necessary algorithms and sets of instructions to form a usable method of test. Although what emerged was not immune to criticism, it was based on a solid conceptual framework, and most of the component parts that were hung on this framework remain in the standard today—modified, perhaps, but easily recognizable in most cases.

The following bullets summarize questions that the committee attempted to settle during this formative period, with the outcomes (as of 1996) given in brackets[]:

- Should impacts of the duct system on equipment efficiency be included explicitly in the standard? [Yes]
- Should the standard rate actual performance or give some sort of standardized rating? [It would attempt to rate actual performance.]
- Should the standard “give points” for the quality of comfort provided? [No]
- What calculation procedure should be used for delivery effectiveness? [A method that considers leakage and conductive losses together, avoiding double counting. (See Section 3.1.)]
- How should equipment size and heating/cooling loads be specified? [Equipment size would be that actually found or specified. Perhaps surprisingly, it became possible to avoid specifying the loads in accounting for effects such as infiltration impact. (See Section 4.)]
- Should Standard 152 relate in any formal way to ASHRAE standards used to measure equipment efficiency? [No]
- Should the standard give a single rating or should it be climate dependent? [It would be climate-dependent.]
- Should the standard attempt to disaggregate duct inefficiencies on the basis of whether they were caused by manufacture, system design, or installation? [No]
- How should duct leakage be measured? [Originally, the standard had three methods. Two were later dropped, and another one added. (See Section 3.2.)]
- Should the effects of pressure changes caused by closing interior doors be included? [No]

- Can the time requirement for the measurements be held to 45 minutes? [For tight ducts in new construction, possibly. For existing buildings, not unless a new and faster leakage test is developed.]
- Should the standard attempt to rate multi-zone systems? [No]
- Should a default option be allowed for determining duct surface area without actually measuring it? [The committee agreed to a formula developed by Lawrence Berkeley National Laboratory, so the answer originally was “Yes.” In the late 1990’s, the committee voted to remove all defaults from the standard, but at the next ASHRAE meeting it reconsidered and put this particular default back in. (See Section 3.4.)]
- Should the effect of duct thermal mass on off-cycle losses be included? [Yes, but in a very simple way, by assessing a flat penalty on seasonal efficiency.]
- How should thermal regain be treated? [By use of a “regain factor” as part of the load factor. Later, this was modified to account for subtleties in the way buildings may actually regain lost heat (or cooling) from ducts. (See Section 4.1.)]
- Should continuous fan operation be included? [No, for lack of research data, despite strong desire of some committee members to include it.]
- How can the water flow rate in hydronic systems be measured without breaking into the system? [A method was proposed but later modified on the basis of field tests. See Section 6.]

2.8 How Accurate Should the Standard Be?

One of the most revisited topics in committee discussions was the question of how accurate the test method needed to be, and how accurate it could be. Neither of these questions had a unique numerical answer, but the committee gradually converged on a more-or-less uniform view.

As for needed accuracy, it was agreed that at a minimum the margin of expected error should be significantly less than the difference between the actual distribution efficiency and 1.0. Otherwise the answer provided by the test method would be little better than a guess. It was, however, recognized that to impose too strict an accuracy requirement would almost certainly increase the time and labor cost of testing a system beyond what diagnosticians in the field would be willing to bear. Gradually a consensus emerged around a 5% criterion, namely that the distribution efficiency reported by the standard should in most cases be within five percentage points of the actual value. There was always a range of opinion within the committee, which might be expressed in terms of the meaning of the phrase “in most cases.” Did this mean 51% of the time, 75% of the time, or 90% of the time? This was never formalized. One result of this ambiguity was that the committee could veer toward more stringent requirements at one ASHRAE meeting and toward more relaxed criteria at the next, depending on which subset of the entire committee membership was in attendance at the time. This was evident in fluctuating decisions on matters such as what duct leakage test methods to accept or whether to include default values for some quantities that were hard to measure.

At the same time, a considerable amount of research was undertaken to quantify the level of accuracy that was possible. Salient examples of projects that pointed toward the reasonableness of the 5% criterion include Siegel et al. 2003 and Francisco and Palmiter 2000b. Efforts to improve the accuracy of duct leakage tests and the calculation of buffer-zone temperatures were recommended in the latter paper.

2.9 End of the “Pathway” Approach

Finally, it may be appropriate to give special attention to the issue that probably accounted for more committee time than any other—how to coordinate the three pathways (design, diagnostic, and research) and how to specify the measurement procedures in the research pathway in a way that would withstand scrutiny during the public review process. The main driver of this discussion was the fact that electric co-heating, which was to be the basis of the research pathway, was not in particularly wide use as a research tool, compared with more common techniques such as those used to measure duct leakage.

Considerable effort was devoted to the development of appropriate measurement techniques for the research pathway. Andrews (1993) summarized test protocols for duct efficiency that had been proposed by other researchers and placed them in categories equivalent to the “bottom-up” and “top-down” methods described in Section 2.1. It was assumed that the former would provide input to the diagnostic pathway while the latter would do the same for the research pathway. This report also described a proposed hydronic test system that could be used for both co-heating and co-cooling. A follow-on report (Andrews 1995) reviewed the literature on electric co-heating methods and assessed the accuracy of the method. This work was intended to establish a basis for co-heating and eventually co-cooling as direct methods for measuring thermal distribution efficiency that would complement and reinforce the admittedly more economical measurement of component loss factors such as duct leakage and insulation levels.

Plans were made to develop a viable co-cooling technique, which culminated in an exploratory research project (Cummings et al. 1999). Although the technique developed by these researchers showed promise, it was concluded that further development would be needed before it could be considered reliable.

Despite this work, three major considerations weighed against the research pathway as it was developing. The first was that, because co-heating was not in wide use, only a few skilled practitioners would have any success with it. This would pose a danger if Standard 152 began to produce erroneous results through misapplication. The second factor in the committee’s decision was the lack of a proven co-cooling technique. The concern was that this could lead to a tilted playing field in the competition between gas and electric equipment, since duct systems could be tested in the heating mode (mostly served by gas-fired equipment) but not in the cooling mode (mostly served by electric-powered equipment). This could lead to a mistaken public impression that ducts served by gas furnaces are less efficient than those served by electric-powered equipment, simply because most of the testing would be done on gas-fired systems.

The third factor was the concern that controversy over the research pathway would delay the adoption of the standard as a whole. The debate over the research pathway was the first instance of a question that has since become a leitmotif in the committee's deliberations: Is it better to compromise on a "good-enough" standard now, or should delays be accepted as the price of a better version later on? Naturally, one's stance on this issue was a strong function of whether one considered the current version to be adequate to serve its intended purposes.

The result was that, at the ASHRAE Winter Meeting of February 1996, in Atlanta, GA, the committee voted to drop the research pathway from the standard and to consolidate the design and diagnostic pathways, so that now there would be only a single section for each type of distribution system. This action effectively ended the formative period of Standard 152P's development. The next version of the standard, which appeared in September, 1996, had a "look and feel" that would remain essentially constant over the next five years despite changes in detail.

Remaining sections of this history will review the development of particular aspects of the standard, organized in line with Equation 2. That is, for forced-air systems, we'll first examine the evolution of Delivery Effectiveness, then the treatment of duct interactions with the load, and finally issues relating to equipment efficiency impacts. These sections will consider related issues that affect each of these factors. For example, climate influences delivery effectiveness because the temperature of the air surrounding the ducts (the "buffer zone") depends not only on the type of buffer space but also on the climate in which the house is located. Other types of distribution systems will be considered in following sections.

3 FORCED-AIR SYSTEMS: DELIVERY EFFECTIVENESS

As discussed above, delivery effectiveness is defined as the ratio of heat or cooling delivered to the load to that provided to the distribution system by the heating or cooling equipment. In forced-air systems, it accounts for direct losses due to duct leakage and heat conduction through the duct walls.

3.1 The Delivery Effectiveness Equation

To gain an understanding of the structure of Standard 152 and the function of each of its parts, the first place one should look is probably the equation used to calculate delivery effectiveness in the heating mode. This equation, which is numbered 6-28 in the current version of the standard (ASHRAE 2001), reads as follows:

$$DE = a_s B_s - a_s B_s (1 - a_r B_r) (\Delta t_r / \Delta t_e) - a_s (1 - B_s) (\Delta t_s / \Delta t_e) \quad (3)$$

where DE is the delivery effectiveness. The various quantities on the right-hand side are needed for a joint treatment of leakage and conduction losses, avoiding the double

counting that would occur if each of these factors were calculated in the absence of the other. This approach was originally developed by Larry Palmiter of Ecotope, Inc. in Seattle, Washington.

Duct leakage is quantified by the parameters a_s and a_r , which are the fractions of air entering the supply and return ducts, respectively, that reach the living space. That is, each “a” parameter is one minus the leakage fraction on its side of the duct system. It is important to note that only the leakage to or from the outside is included; leakage to inside still gets to the living space even if by unintended flow paths.

The remaining parameters in Equation 3 are needed to quantify the conductive heat losses. The B’s play a role for conductive losses similar to that of the a’s for leakage. That is, B_s is the fraction of input heat that would be retained by the supply duct and delivered to the conditioned space if there were no leakage and if the temperature of the space surrounding the ducts were the same as the indoor temperature. It is given by an exponential function:

$$B_s = \exp[-A_s / (Q_e \rho_{in} C_p R_s)] \quad (4)$$

where A_s is the surface area of the supply ducts outside the conditioned space, Q_e is the volumetric air flow rate at the system fan, ρ_{in} is the density of indoor air, C_p is the specific heat of air, and R_s is the thermal resistance of the duct, i.e., the thermal R-value. The parameter B_r is defined in the same way as B_s , with the subscripts of A and R changed from s to r.

The “delta” quantities are temperature differences. Δt_e is the temperature rise of the air passing through the furnace, heat pump, or electric resistance coil. Δt_s is the difference between the indoors and the zone surrounding the supply duct, while Δt_r is the similarly defined difference for the return duct.

The simplest situation would be one in which the buffer-zone temperatures are the same as indoors. In that case, the second and third terms of Equation 3 would equal zero, and the delivery effectiveness reduces to the product $a_s B_s$. In this case one can see how Equation 3 avoids double counting of losses. For example, if the supply leakage were 20% of the system fan flow, and if the duct would lose 20% of its heat via conduction in the absence of leakage, then $a_s = 0.8$ and $B_s = 0.8$, yielding $DE = 0.64$ or 64%, not the 60% one would obtain simply by adding the individually calculated losses.

The second and third terms of Equation 3 quantify the additional losses that occur if the buffer-zone temperatures are less than the indoor temperature. The derivation of these terms will not be reviewed here, but it may be found in Palmiter and Francisco 1997.

The delivery-effectiveness equation for cooling, 6-29 in ASHRAE 2001, is somewhat more complicated because it has to take both temperature and humidity into account.

3.2 Duct Leakage Measurement

There has been considerable evolution in the test methods for duct leakage in the standard. Initially, the plan was to use the methods embodied in ASTM Standard 1554-94, "Standard Test Methods for Determining External Air Leakage of Air Distribution Systems by Fan Pressurization." This document (ASTM 1994) specifies two different ways to measure duct leakage.

The first method in the ASTM standard was a blower-door subtraction method, in which the air flow through a blower door, necessary to pressurize (or depressurize) the house to a standard pressure is measured with the registers sealed, the registers unsealed, and finally with just the return side of the duct system sealed off from the rest. In principle, this permits the leakage flow coefficients of both the return and supply ducts to be measured. However, this method was soon abandoned because of a consensus that its level of accuracy would be insufficient for practical use except in very leaky duct systems.

The other method in the ASTM standard is based on pressurizing a sealed duct system to a standard pressure, typically 25 Pa, with a smaller version of the blower-door generically known as a duct blower. The supply and return portions of the duct system are separated by a temporary barrier (referred to as "splitting the system"), and the leakage flow coefficients on each side of the duct system are measured separately. This method found a place in the ASHRAE standard.

Early versions of Standard 152P provided three options within the diagnostic pathway for gathering the information necessary to determine delivery efficiency. One of these actually bypassed a duct leakage measurement and instead called for a direct input-output measurement. This approach, called "Method A" in the standard, specified measurements of temperatures and airflows at the system fan and at the registers. However, like the first ASTM method before it, this approach was soon abandoned as unworkable, so that as early as 1996 it was no longer a part of the standard.

The remaining two approaches did employ direct measurements of duct leakage. Later, an additional variation of one of the methods was added, bringing the number of leakage tests back up to three. These were:

- The Split-System Fan-Pressurization Test
- The House Pressure Test
- The Whole-Duct Fan-Pressurization Test

Early in 2001, the house pressure test was deleted from Standard 152P, reducing the number of tests again to two. The following sections describe these tests and discuss the issues associated with each. A final section discusses other proposed leakage tests and their current status.

3.2.1 The Split-System Fan Pressurization Test

The second approach within the original diagnostic pathway measured duct leakage using a close relative of the second ASTM method, and it has survived. As in the ASTM version, in this one the system is “split” by the temporary placement of a barrier at the system fan (or as near to it as practicable) to separate the supply and return ductwork. A modification to the ASTM protocol specified that at the same time the ducts were being pressurized with the duct blower, the house would also be pressurized to the same pressure with a blower door. The purpose of this was to eliminate leaks from the duct to the conditioned space, so that only leakage to outside would be measured. The other advance was to provide a method of pro-rating the leakage measured at the standard pressure to a pressure representative of what was actually occurring in the ducts. This was done by estimating an “operating pressure” and then pro-rating the leakage at the measured pressure to this operating pressure using the pressure-flow relationship, i.e., through multiplication by the ratio of operating pressure to standard pressure, to the power 0.6.

The operating pressure was to be determined by covering each register in turn with a blocking device, roughly the size and shape of a common baking pan, and measuring the static pressure behind the barrier. The average of these register-blocked static pressures would then be used as the operating pressure. If the number of registers was sufficiently large (>5) it was thought that the perturbation on the system caused by covering one register would be sufficiently small that the results of the measurement would be acceptable. This criterion is usually met on the supply side but not on the return side. Therefore, an alternative of using one-half the return-plenum static pressure as the operating pressure was provided for the case where there were five or fewer registers.

This test has remained largely the same during the entire period 1996-2001, with the small exception that the blocked-register or “pressure-pan” method of determining the operating pressures was dropped in favor of using one-half the plenum pressure on both the supply and return sides of the system, regardless of the number of registers. Several researchers stated that the pressure pan method tended to be biased on the high side, with two groups having included such statements in published reports (Cummings and Withers 1999, Francisco and Palmiter 1999).

The split-system fan-pressurization test has not, however, escaped controversy even after this modification. It has generally been found to be repeatable, meaning that if one tester makes several successive measurements on the same duct system, the results will generally agree to within a close tolerance. An exception to this can occur when there are no good places in the supply or return duct to attach the duct blower. It is usually attached to the return portion of the duct system at a large register, and since return registers are usually few in number, they tend to be large and connected to the main part of the duct by a channel of relatively large cross section. If a leaky system has only small return registers, however, repeatability can be compromised (Andrews et al. 1998).

Supply registers, however, tend to be smaller and are often at the ends of runout ducts of relatively low cross section. If the duct blower is attached to such a supply register and there are some large leaks in the system, the measured flow necessary to effect a pressure reading of 25 Pa at some specified location can vary considerably depending on which register is chosen for the attachment of the duct blower and where the pressure sensor is located. For this reason, the fan access opening at the air handler is often chosen as the attachment point, with the supply-return barrier placed just upstream, e.g., in the filter slot. This admittedly allocates the entire air handler cabinet to the supply side even though part of it is upstream of the fan, but it is nevertheless often done as a matter of convenience.

Beyond repeatability is the question of accuracy. Two researchers at Ecotope, Inc. of Seattle devised ways of arriving at an estimate of leakage thought to be more accurate than any of the tests under consideration. Francisco and Palmiter (1999b, 2000a, 2000b) did this by performing the fan pressurization test both with and without the blower door, and in addition measuring the air flow at the system fan and the total flow through the registers with a flow capture hood. Subtracting the register flows from the airflow at the system fan gave an estimate of the total duct leakage. This, together with the measured total-leakage flow coefficient, allows a “best estimate” for the duct operating pressure to be calculated. This is then used, along with the flow coefficient for leakage to outside, to calculate what they called a “best estimate” for that leakage. The authors then used the “best estimate” leakage values as benchmarks against which to rate the candidate tests, both the ones currently in Standard 152P and others that had been proposed. They found for 26 test cases that the leakage as measured by the fan pressurization test had an absolute average difference from the “best estimate” of 32 cfm on the supply side and 87 cfm on the return side. The return-side results included four cases where the difference exceeded 200 cfm and another three where it exceeded 100 cfm.

There has been considerable discussion within the committee of whether this “best estimate” procedure provided a reasonable benchmark to use in evaluating duct leakage tests. In a research project by Ecotope that is still ongoing in mid-2003, efforts were made in a subset of test houses to compare the method, whose name was changed from “best estimate” to “benchmark value,” with independently measured “created” leaks intentionally introduced into the duct systems.

Cummings et al. (2000) added known amounts of leakage to a duct system in each of two houses by adding bypass ducts to the outside and measuring the flows through these by independent means such as a hot-wire anemometer or a flow hood. They found that if the duct-blower measurements were combined with the measured pressure at the location of the artificially created leaks, the results of the candidate tests agreed very well with the independently measured leakage flows. However, when the pressures in the supply ducts were specified in the standard way (using pressure pans on the supply side and half the plenum pressure on the return side), the test results disagreed with the independently measured values by 60% on average. In this study, the worst offenders were on the supply side, so some of the discrepancy was probably caused by the use of the pressure pans. As mentioned above, the pressure-pan technique has since been removed from

Standard 152P. Also, the leaks were artificially created large holes and the sample size was small. Nevertheless, these results did give impetus to researchers on the lookout for improved methods of measuring duct leakage.

3.2.2 The House Pressure Test

Another method of measuring duct leakage in the standard, as of 1996, was the house pressure test. This method aims to extract values of leakage from the supply ducts to outside and to the return ducts from outside by measuring the pressure in the house (with respect to a well-vented attic) under three conditions: 1) system fan off; 2) system fan on; and 3) system fan on with the return register(s) partially blocked. Pressures in the supply and return ducts also need to be measured under each fan-on condition. The attic is used as the reference for the pressure measurements to minimize variations caused by wind gusts.

Measurements 1 and 2 permit one to tell which of the leakage rates is greater. If the pressure in the house rises when the system fan is turned on, it means that the return leakage exceeds the supply leakage, since the duct system is taking in a net amount of air from the outside and pushing it into the house, in effect “blowing it up like a balloon.” If the house pressure drops when the system fan is turned on, the reverse is true and the supply leakage is greater. So far, this only indicates whether the difference between the supply leakage (to outside) and the return leakage (from outside) is positive or negative, and for that reason it is sometimes called the dominant duct leakage test. The mathematical formalism of the house pressure test, which was developed by Mark Modera of Lawrence Berkeley National Laboratory, made use of the leakage characteristics of the house envelope to extract a numerical value for the difference between supply and return leakage, not just its sign. Moreover, by taking an additional measurement of the house pressure with the return register(s) partially blocked, it was possible to develop an additional equation that would permit not only the difference between the leakage rates, but the individual supply and return leakage rates themselves, to be calculated.

The house pressure test had the great merit of being quick and easy to do. Moreover, in some early field tests, the method appeared to give reasonable results. However, it soon came in for criticism on theoretical and practical grounds.

The main theoretical question was whether the assumptions behind the mathematics used to calculate the leakage rates were sound. One simplifying assumption was that the leakage area of the house envelope is equally divided between the floor and the ceiling, with none in the wall. Francisco and Palmiter (1997) questioned this assumption. They also raised the concern that if the ducts are in the attic and have significant unbalanced leakage, this will affect the attic pressure when the system fan comes on, so that the pressure in the attic will no longer be representative of that outside. Two years later, the same authors (Francisco and Palmiter 1999a) expanded their criticism of the leakage location assumption with a detailed analysis that showed significant deviation of the house pressure test’s value of unbalanced leakage from the actual value. In one case, for

example, they found that if the leakage area is half in the wall and the remainder equally divided between the floor and the ceiling, the unbalanced leakage given by the house pressure test would range from 80% to 95% of the actual value. If the leakage is half in the wall and half in the ceiling, with none in the floor, the house pressure test could either underestimate or overestimate the actual unbalanced leakage, in some cases by as much as 30%. Another study of the same question (Andrews 1998a) generalized the derivation of the house pressure test equations to remove the particular assumptions of where the envelope leakage occurred, and concluded that, for a wide variety of envelope leakage distributions, the generalized equations usually gave results within 5% of the ones based on leakage in the ceiling and floor only. However, many cases of greater divergence, sometimes by more than 35% were found, and it was also found that very leaky ceilings would give greater divergences more often.

The other theoretical issue concerned the sensitivity of the house pressure test to some of the measured inputs. Andrews (1997) raised the concern that the results of the house pressure test would be very sensitive to the ratio of the pressure in the return duct with the return register covered to this same quantity with the register uncovered. The sensitivity to the same ratio as measured on the supply side was predicted to be much less. Unfortunately, the expected variation of this ratio on the return side, caused by random variations in the measurement process, was predicted to be higher than that on the supply side. That is, the expected variation in the pressure ratio was greater on the side of the duct system where one could less afford it. Other concerns were that the house pressure test becomes insensitive when the house envelope is leaky and that there are particular values of the measured pressures for which the slopes of the leakage functions become vertical, resulting in exceptionally large uncertainties when these pressures happen to occur.

A final insight that emerged about this time was that the errors in the supply and return leakage rates tend to be comparable in magnitude. This implies that if the leakage is highly unbalanced, the smaller of the two leakage rates will tend to have a high percentage error even if the uncertainty in the larger leakage value is within reasonable bounds. This problem appears to be common to any type of test in which the supply and return leakage rates are measured at the same time via a protocol that extracts them from a set of simultaneous equations. (See the discussion of the delta Q test in Section 3.2.4.)

This, however, was all theory. Experimental field results soon began to come in. Cummings et al. (2000) reported, in a study of eight duct leakage configurations in two Florida houses, that the house pressure test “predicts no better than 27% average deviation from measured” leakage rates. Francisco and Palmiter (2000) stated that “the house pressure test does not perform well in many cases. Even taking extreme measures to place the return duct pressure tap in the proper location, this method performed worse than any other tested in estimating efficiency.” Strunk (2000) stated that the “house pressure tests gave unacceptably broad, nonrepeatable spreads of values in most cases.” This author qualified the statement by acknowledging that “this house posed a particularly difficult problem” for the house pressure test, and gave four reasons why that was so.

Proponents of the house pressure test were quick to point out that many of these and other similarly negative results were taken in situations that did not meet the stated criteria in Standard 152P for deciding when the house pressure test could be used. It had been recognized from the start that this test would not be appropriate in all cases.

Exceptionally leaky envelopes, high wind conditions, unfavorable placement of furnace filters, and labyrinthine return ducts with multiple registers were seen as danger signals that should warn against the use of this test, with a fan-pressurization test being the fallback option in that case. Various formulations of these caveats appeared in the standard, with revision from year to year under the stress of competing points of view in the committee.

The summation of all this is that there has been a “running battle” over the house pressure test within the Standard 152P committee almost from the beginning. Gradually, the opponents gained ground until, finally, the committee voted in January 2001 to eliminate the house pressure test from the standard. A major deciding factor in reaching consensus on this was the point that having two alternative duct leakage tests that can produce very different results would not be a good thing for a standard test method. Therefore, a choice was made between the house pressure test and the fan-pressurization test, with the latter getting the nod.

3.2.3 The Whole-Duct Fan-Pressurization Test

At the June 1996 meeting of the Standard 152P committee, various members stressed the need for a leakage test that could be performed before the house envelope was closed in, to assure that the system met a specified efficiency level. If a system performed poorly in such a test, it would be much easier to upgrade at that point than after much of the ductwork had been hidden behind partitions. It was therefore agreed that a measurement protocol would be developed that did not require the envelope to be pressurized. It was intended that this method would be used primarily in new construction.

Even though Standard 152P is not prescriptive itself, it was recognized that some regulatory bodies might specify its use in determining whether prescriptive requirements set by these agencies had been met.

The method selected is similar to the split-system fan-pressurization test in that the duct system is pressurized with a duct blower, but there are two major differences:

- The conditioned space is not pressurized with a blower door, but is at the same pressure as the outdoors.
- The duct system is not separated into supply and return zones with a temporary barrier, but is tested as a unit.

These differences reduce the time needed for the test and make it possible to perform it before the envelope is closed in, but they do introduce uncertainties into the measurement that are not present in the split-system fan-pressurization test. First, the leakage

measured is total leakage, not just that to the outside. Second, the split of the leakage between the supply and return sections is not measured.

The first difference is conservative in that the leakage to outside cannot be greater than the total leakage. If the duct system is mostly outside the conditioned space, the two numbers should be close, in which case the difference should not matter. If the builder has intentionally placed much of the ductwork inside the conditioned space and is depending on this to reduce leakage to outside, then in order for the credit for this strategy to be realized, it will be necessary to wait until the envelope is closed in and use the split-system fan-pressurization test.

In the first version of the whole-duct fan-pressurization test, it was specified that the actual pressure difference across the system fan would be equal to that specified by the manufacturer, or, should that information be missing, 125 Pa (0.5 inch water column). This was to be apportioned 60% to the return side and 40% to the supply side, this difference reflecting the collective judgment of the committee members concerning the most likely situation to be encountered. This was later (in 2001) changed to a 50-50 split.

A provision was added to the test to allow for testing before the HVAC equipment and grilles were installed. This was to say that leakage from the equipment cabinet and the register boots would be assumed to equal 5% of system fan flow. Another provision added later specified that the whole-duct fan-pressurization test could only be used if the measured duct leakage (supply plus return) is less than 10% of system fan flow (or 5% of system fan flow if the equipment and grilles were not yet installed). If the test result exceeds this limit, then a split-system test must be done.

The justification for allowing a test like this, with its admitted potential for inaccuracy in the “split” between the supply and return leakage, is that with the restriction on total leakage to 10% of fan flow, the impact on overall efficiency of even a fairly large error in the split will not affect the final figure of merit, distribution efficiency, very much.

3.2.4 Other Proposed Tests

The deletion of the house pressure test from Standard 152P was a disappointment even to those members of the committee who voted to drop it. Of the two tests left in the standard, the split-system fan-pressurization test was considered somewhat cumbersome because of the requirement for the barrier and the need for two separate tests. Moreover, although it is usually quite repeatable, its accuracy is uncomfortably close to the margin of acceptability. The whole-duct fan-pressurization test, although quicker and easier to do, is limited to relatively “tight” duct systems. A fair amount of effort has therefore gone into the development of alternative tests that would reduce the time and effort needed to do them, increase the expected level of accuracy, or both.

Variations on the House Pressure Test. Two variations of the house pressure test were suggested by a member of the Standard 152P committee (Andrews 1998b). One of these variations was identical to the original house pressure test, except that instead of partially

blocking the return register(s) in the last part of the test, the supply registers would be partially blocked instead. The motivation for this change came from the realization that the calculated leakage rates were far more sensitive to the ratio of the register-blocked to register-unblocked pressure in the return duct than to the similar ratio on the supply side, but the expected uncertainty in this ratio would be greater on the side whose registers are blocked. The question naturally presented itself, then, Could the test be done with the supply registers partially blocked instead of the return(s)? Andrews (1998) showed that the mathematics held equally well whether the supply or return registers were partially blocked. He also presented promising comparisons of the repeatability of the blocked-supply house pressure test with that of its blocked-return progenitor. However, other researchers (Francisco and Palmiter 2000, Cummings et al. 2000, Strunk 2000) did not obtain any better results with the blocked-supply test than with the blocked-return variant.

Partial blocking of the supply registers may be problematic because there are usually more of them than there are returns, because the positive supply pressure tends to blow the tape or other blocking means off the registers, and because care must be taken to insure that all the supply registers are blocked to the same degree. Although these problems might be addressed successfully, the lack of encouraging results has caused interest in this test to wane.

The other alternative to the house pressure test proposed in Andrews 1998 was a hybrid variation that combined the register-unblocked portion of the house pressure test with a version of the whole-duct fan-pressurization test that did use the blower door to zero out leakage to inside. Francisco and Palmiter (2000) found that the hybrid test performed less well than the split-system fan-pressurization in measuring supply leaks but better on return leaks, the two tests scoring similarly overall. Cummings et al. (2000) also found that these two tests performed similarly. Neither of these two research groups was especially positive about any of the tests, however, the judgment being that the hybrid test, although somewhat quicker and easier to perform than the split-system fan-pressurization test, gave results that were not noticeably more accurate.

Variations on the Whole-Duct Fan-Pressurization Test. The California Energy Commission (CEC) uses a whole-duct fan-pressurization test in Appendix F of the Title 24 Alternative Calculation Manual (CEC 1998). Appendix F uses a simplified version of Standard 152P that existed in 1998, with several defaults added to enable design calculations to be made and to reduce the measurement requirements so that duct leakage is the only quantity that must actually be measured. The CEC had closely followed the development of Standard 152, but decided it could not wait several years for the final version to be completed. In this variation of the duct pressurization test, the total leakage at 25 Pa is used. The rationale was that if total leakage is below the efficient duct credit limit of 6% of fan flow, then the leakage to outside must also be below this limit. Also, because such a low limit is allowed, the balance between supply and return leakage does not affect the duct system efficiency very much.

Also, the U.S. Environmental Protection Agency (EPA), as part of its EnergyStar program, became interested in duct efficiency. This agency had a need for as simple and quick a test as possible, and because the results were to be compared with a prescriptive standard, the test had to be very repeatable. What they settled on (CEE 2000) was a variant of the whole-duct fan-pressurization test that, like the original, measures total leakage at 25 Pa pressure, but does not do any pro-rating to actual operating pressures. A tested duct system passes the EPA acceptance criterion if the measured leakage is less than 6.25% of system fan flow for new construction or 10% of fan flow for existing duct systems.

Another variation on the whole-duct fan-pressurization is presented in a duct efficiency manual (Andrews 2001) developed for the U.S. Department of Energy. The intent was to provide a test nearly as easy to do as the CEC and EPA test but which would provide a useful estimate of the supply-return leakage split. In this test, the leakage of the whole duct is measured at 25 Pa, either without the blower door (to measure total leakage) or with it (to measure leakage to/from outside). The sum of the actual supply and return leakage rates is estimated by multiplying this measured leakage rate by the square root of one one-hundredth of the static pressure rise across the system fan. This involves some simplifying assumptions, but a Monte Carlo analysis indicated that it should give values for the sum of the leakage rates nearly as accurate as those provided by the split-system fan-pressurization test. A dominant duct leakage test is then performed, using 30 pairs of house-pressure measurements with the system fan on and with it off. Instead of attempting to quantify the unbalanced leakage (i.e., supply leakage minus return leakage) using the algorithm in the house pressure test, a simpler approach is adopted that should be more repeatable. First, it is determined whether a statistically significant proportion of the house-pressure pairs indicate return dominance (i.e., fan-on house pressure exceeds the fan-off value) or supply dominance (i.e., fan-off house pressure is the greater of the two). The number of negative differences between the fan-on and fan-off house pressure is subtracted from the number of positive differences. If the result is 10 or more, the system is considered return-leakage dominant. If the result is -10 or less, the system is considered supply-leakage dominant. If the result is between -10 and 10, the system is considered balanced. The likelihood that a balanced system would be taken as unbalanced under these criteria is 10% (binomial distribution, $N=30$, two-tailed test). The sum of the supply and return leakage rates is then apportioned 70% to the dominant side and 30% to the other side, or 50-50 if the criterion for a balanced system is met.

The Delta Q Test. The trend of the above discussion has been the search for a quicker and easier way to measure duct leakage that is also repeatable. In the meantime, however, other researchers were searching for a test that might improve accuracy while still holding the level of effort within some sort of reasonable bound. One main candidate has been the delta Q test developed by researchers at Lawrence Berkeley National Laboratory.

The delta Q test (Walker et al. 2001, 2002) uses a blower door to measure the airflow needed to depressurize or pressurize the house to ten different pressures, ranging from -25 Pa to 25 Pa. The measurements are made with the system fan off and then with it on.

At each of the ten pressures, the difference between the system-fan-on and system-fan-off air flows through the blower door fan is calculated. Each of these is called a "delta Q." The ten delta Q's are input into a set of equations that calculates a least-squares best estimate of the supply and return leakage. This test has the advantage of being fairly easy to perform, as long as the mathematics is embedded in a spreadsheet or other computer-based calculation engine. The use of a "cruise-control" blower door is recommended. Questions have been raised as to whether the mathematical assumptions underlying the delta Q test are too rigid to give reliable results in all cases. Also, as in the fan-pressurization tests for duct leakage, the results depend on what operating pressures are used on each side of the duct system. The sensitivity to these pressures is much less than in the fan-pressurization tests. Even so, it is enough to have motivated the developers of the Delta Q test to pursue means for fitting the leakage pressures using the wealth of information inherent in the Delta Q measurements. They have used fitting algorithms in some of the analyses of Delta Q tests performed in the field. There is not yet, however, general agreement on a single best algorithm to use in such a fitting procedure.

Another caveat in the delta Q test is that, like the house pressure test, the uncertainties in both the supply and return leakage rates tend to be similar in magnitude, resulting in a high percentage uncertainty in the smaller of the two leakage rates in a system where the leakage is highly unbalanced. This is not a fatal flaw for an efficiency calculation such as that in ASHRAE Standard 152P, since the uncertainties in efficiency generally depend on leakage uncertainties as fractions of system fan flow, not as fractions of the leakage rates themselves.

Additional projects have either been completed or are underway at Lawrence Berkeley National Laboratory, Ecotope, Inc., and Brookhaven National Laboratory to evaluate the delta Q test further. An analysis (Andrews 2000a, 2000e) of the Delta Q test used the Monte Carlo technique to explore the impacts of measurement errors on the test results, and compared this analysis with some initial field tests. The salient finding was that the random uncertainty in the sum of the leakage rates tends to be two to three times as large as the random uncertainty in their difference (i.e., the unbalanced leakage). A study of the Delta Q test in a laboratory duct system (Andrews 2002) found that the Delta Q test reduced measurement errors to about half those experienced with the split-system fan pressurization test. An added "wrinkle" in which hole-size information was used to estimate the leakage pressures further reduced the errors to about a third of those experienced with fan pressurization.

The Nulling Test. The other leading candidate among newly proposed tests is the nulling test developed by researchers at Ecotope, Inc., of Seattle, Washington. It is based on the concept that if the unbalanced leakage is canceled out by some artificial means of removing the extra air supplied by the duct system (if the return leakage dominates) or of replacing the air removed by the duct system (if the supply leakage dominates), then the pressure field in the envelope will revert to what it was when the system fan was off. This removal or replacement of air could be done using either a blower door or a duct blower. In any case, the means used must be capable of measuring much lower airflows

than are usually encountered in blower-door tests, unless the unbalanced leakage is unusually high.

In the second part of the nulling test, the return portion of the duct system is blocked off and a second duct blower or blower-door fan is connected to the duct system via the fan-access opening in the air-handler cabinet. The inlet side of the duct blower or blower-door fan is connected to the living space via a length of flexible duct. The system fan is energized and the duct blower or blower-door fan is adjusted until the pressure at a specified point in the supply duct is brought equal to what it was under normal system operation. The unbalanced leakage is now measured in the same way as before. However, now the unbalanced leakage is just the supply leakage, because there is no return leakage. With knowledge of the supply leakage and the unbalanced leakage under normal operation, the normal return leakage can be determined by subtraction.

The advantage claimed for the nulling test is that under the test condition, the pressure distribution within the house is essentially the same as under normal operation. This is in contrast to the delta Q test, where the house pressure varies over 50 pascals from one part of the test to another. There is some possible bias even in the nulling test, however, because the holes in the ducts are part of the envelope leakage distribution when the air handler is off, but are removed from the envelope leakage distribution when it is on (Francisco and Palmiter 1999b). It may be possible to correct for this bias and in any case it is believed to be small to moderate in most cases.

The major difficulty with the nulling test is that the second part involves a fair amount of effort, because of the need to build an alternative flow path from the living space to the air handler. If the air handler is within the living space, this task is greatly simplified. Also, for houses without return ducts, such as most manufactured housing, the nulling test can be done in one step, because there is no return leakage and the unbalanced leakage is then equal to the supply leakage. One caveat here is that this use of the nulling test would effectively consider any leakage to outside from the closet in which the air handler is located to be envelope leakage and not duct leakage.

Some early test results on the nulling test were presented in Francisco and Palmiter 1999. The test was performed in 12 houses and found to agree well with the “best estimates” of duct leakage determined as discussed in Section 3.2.1. An analysis of measurement uncertainties in the nulling test (Andrews 2000b) pointed out that although the central values of leakage obtained in the nulling test will not depend on the envelope leakage flow coefficient, the uncertainties in these values will depend on this parameter. The other major determinant of uncertainty was predicted to be the precision of the process by which the pressure during the nulling test is matched to that which prevailed with the system fan off. Errors in measuring the air flows across the envelope were expected to have only a minor impact on the overall uncertainty in the measured leakage rates.

At this writing it is too early to know whether any of these proposed tests will ever make their way into Standard 152. Good ideas for “better, faster, cheaper” ways to measure duct leakage will probably always be welcome.

3.3 System Airflow

From the perspective of distribution efficiency, knowledge of the airflow rate at the system fan is important because the efficiency penalty due to duct leakage is proportional not to the leakage rate itself but to the leakage rate as a percentage of fan flow. That is, 100 cfm of leakage is more damaging in a 600 cfm system than in a 1200 cfm system, for the simple reason that in the former case the leakage loss is a larger percentage of the delivered heat or cooling.

There are, of course, other reasons for measuring the system airflow. Inadequate airflow is a major cause of efficiency degradation in air-conditioning systems, so knowing the airflow rate often points the way to correcting problems in both the ducts and the equipment.

Unfortunately, airflow is not an easy thing to measure accurately, even under controlled laboratory conditions, and the convoluted nature of most residential duct systems, with few if any long-straight sections near the system fan, makes it even harder. With that in mind, it may be useful to review the methods that have been proposed in the Standard 152P committee to measure the airflow at the system fan.

These methods may be divided into two categories:

- Traverse methods, in which air velocity is measured at each node of a grid of points equally spaced over the duct cross section;
- Whole-duct methods, in which the volume flow rate is measured directly by some means.

Traverse Techniques. Traverses can be done using any kind of small-bore probe that is able to measure the velocity of the air without appreciably altering it. The most common instrument suggested for this is the Pitot tube, which is used to measure the static and total pressures, from which the velocity pressure is obtained by subtraction. The hot-wire anemometer is another instrument that has been used for this purpose. These methods have been criticized on the grounds that they may be time consuming, that they require a high level of knowledge and care in their execution, that they may not be appropriate for use in a duct system where the airflow is far from uniform, and that they are invasive in that several holes have to be drilled into the duct in order to access an adequate grid of test points. For these reasons, traverse methods were never seriously considered by the Standard 152P committee, even though they did find advocates from time to time among committee members and others.

Flow Hood. The other methods considered were all of the whole-duct variety. The simplest of these is to measure the total flow through the supply (or return) registers with a flow-capture hood and then to add the total leakage on the supply (or return) side of the duct system to obtain the airflow at the system fan. This is a fairly straightforward method, but it was not adopted by the committee because the primary leakage test method in Standard 152P, the split-system fan-pressurization test, measures leakage

between the ducts and the outside, not total leakage. Thus, a separate leakage test would have to be done to support this method of determining airflow. Another reason this approach was not adopted was the judgment that most flow hoods are not very accurate when measuring low flow rates (under 200 cfm) such as are found in supply registers, whereas on the return side the flow hood might alter the flow if there is only one return register and the total flow rate is relatively high.

Temperature Rise. Another method commonly used by some practitioners is the temperature-rise method, in which the energy input by the heating equipment is divided by the product of the temperature rise and the specific heat of air to obtain the flow rate. This approach was not adopted by the committee because actual equipment capacity is generally unknown, because steady-state conditions are often never reached, and because the cooling equivalent of the test is complicated by the presence of a latent component. The most important reason, however, is that the temperatures vary considerably over the duct cross section and it is impractical to completely map the temperature and velocity distributions in the air streams entering and leaving the equipment.

Pressure Matching with Duct Blower. The method that was adopted was a technique using a duct blower. The static pressure is first measured at a point in the supply duct. The return side of the system is then blocked off by a barrier (which is required anyway for the split-system fan-pressurization test) and a duct blower is attached to the system at the fan-access opening in the air handler (in the same configuration as required for the supply leakage measurement). Before the supply registers are blocked for the leakage measurement, the system fan is turned on and the duct blower motor speed is adjusted until the pressure in the supply duct is the same as under normal operation. Care must be taken that the pressure probe is not bumped or jostled between the initial test and this "pressure matching" operation.

The main reason for adopting this method was that it meshed well with the split-system fan-pressurization test and that it generally had been found to be repeatable in the field. Not very much has been published on this, however. In one test conducted by five different testers using the same duct blower (Andrews et al. 1998), the standard deviation of measured airflow rates was 3.6% of the mean value. This random error would need to be combined with the expected error in the flow measurement device itself (quoted as $\pm 3\%$ by one manufacturer) and the error in matching the pressures in order to arrive at an estimate of overall uncertainty in the airflow rate.

Flow Plate. Very recently, a new device to measure the airflow rate at the system fan has been developed under a U.S. Department of Energy research grant (Palmiter and Francisco 2000). In this technique a special plastic plate is inserted into the air stream near the system fan, with the filter slot being recommended as the preferred location. This plastic insert has several precision-machined circular holes. Copper tubes with strategically placed holes are used to sense the pressure difference across the plate, which is measured using a digital manometer. The measured flow with the plate in place is corrected to the value under normal operation, with the plate removed, by multiplying the flow rate by the square root of the ratio of pressures in the supply duct with the plate

removed and with it in place. On the basis of laboratory and field measurements, the flow plate's accuracy was placed at $\pm 7\%$ by the developers of the device. This method has been added to Standard 152P as an acceptable alternative way of determining the system fan flow.

3.4 Thermal Conduction Parameters

The thermal resistance of the duct to heat conduction across the duct walls is used in Equation 4 to calculate the temperature relaxation functions B_s and B_r . For ducts with a visible manufacturer's marking of unit-area thermal resistance or R-value, the standard specifies that this value shall be used. If there is no such marking, then the R-value is to be calculated using the methods of the ASHRAE Handbook of Fundamentals (ASHRAE 1997, pages 32.14-15). This method prescribes the use of charts, with the duct U-value depending on the air velocity in the duct. Early versions of the standard specified that for estimates based on a building plan, a film coefficient of $0.25 \text{ K}\cdot\text{m}^2/\text{W}$ or $1 \text{ h}\cdot\text{ft}^2\cdot^\circ\text{F}/\text{Btu}$ should be added to the stated R-value of the insulation, but this specific prescription is no longer in the standard. For uninsulated ducts, this R-value is close to the ASHRAE 1997 chart value for air velocities generally seen in residential ductwork, and it is suggested for this use in the Standard 152P spreadsheet maintained on an Internet Web site (<http://ducts.lbl.gov>) maintained by Lawrence Berkeley National Laboratory.

Standard 152P specifies that the total surface areas of the supply and return ducts are to be measured. This can be a time-consuming process, and it also often requires technicians to enter unpleasant areas in attics and crawl spaces. In order to eliminate this burden, a default option is currently provided within the standard, in which the duct surface area is estimated using the conditioned floor area of the house and the number of return registers. The formula and most of the supporting data are due to researchers at the Lawrence Berkeley National Laboratory (Walker 1998). This is probably the most significant default option that has remained a part of Standard 152P to the present time.

With values for the surface areas and insulation levels in the supply and return ducts, along with the airflow rate, the parameters B_s and B_r can be calculated using Equation 4 and their values inserted into Equation 3 for the delivery effectiveness.

3.5 Temperatures and Humidities in Buffer Zones

The effect of climate comes into the calculation of delivery effectiveness because the temperatures and humidities in the buffer zones are climate-dependent. These are obtained in a two-step process. The first step is a table lookup in which, for each of about 250 cities in the United States and Canada, relevant parameters are given for design (near-peak) and seasonal-average conditions. For heating, only two quantities are needed, namely the design and seasonal outdoor temperatures. For cooling there are ten parameters, these being the design and seasonal outdoor temperature, outdoor humidity ratio, indoor humidity ratio, outdoor-air enthalpy, and indoor-air enthalpy. The temperatures and (for cooling) humidities are used as inputs to formulas for the buffer-zone temperatures and humidities to be used in calculating the delivery effectiveness.

The sources of the design and seasonal outdoor temperatures and of the formulas used to calculate the buffer-zone conditions are discussed in Walker 1998.

Probably the most significant issue related to the specification of buffer-zone conditions has been over which of the following two procedures should be used:

- Approach A: Specify the temperatures and humidities in the buffer zones as they actually are, with ducts present.
- Approach B: Specify the temperatures and humidities in the buffer zones as they would be were no ducts present, and then use the thermal regain factor to correct for the resulting overestimation of energy losses.

Thermal regain is discussed in more detail below (Section 4.1). It may be summarized here as an effect by which some of the heat (or cooling) lost from the ducts may effectively be reclaimed because of interaction with the load. For example, heat lost from ducts in a basement will warm that space, which in turn will reduce the amount of heat transfer from the house to the basement, i.e., the heating load is reduced.

The argument for Approach B is that it allows one set of buffer-zone temperatures to be used for each type of buffer zone in each city or climate zone. Proponents of Approach B say that the use of an appropriate regain factor permits any inaccuracy resulting from this approach to be fixed. While it is true that the delivery efficiency values will be skewed (generally downward), the figure of merit, distribution efficiency, will come out all right.

The argument for Approach A is that it is more realistic and transparent to the user. There is something disconcerting in seeing buffer zone temperatures specified in the standard that you know are not typical of actual values.

Approach B is currently in use in Standard 152P. However, the question of whether Approach A might be better is one that has come up more often than not in meetings of the Standard 152P committee after 1996.

Perhaps the strongest proponents of Approach A have been Larry Palmiter and Paul Francisco of Ecotope, Inc. in Seattle. In a report prepared for the U.S. Department of Energy (Francisco and Palmiter 1977), they highlighted two concerns related to the question of how buffer-zone conditions are specified. The first of these suggested that the formulas in Standard 152P might be based on insufficient data to be representative of the broad spectrum of buildings and climates that span North America. The second concern related to whether changes in buffer-zone temperatures caused by duct losses should be factored in before calculating delivery effectiveness using Equation 3, that is, whether Approach A should be substituted for Approach B.

They argue for doing exactly that, and propose a model for doing the calculation. They point out that this calculation might have to be iterative, since the duct losses are dependent on the buffer-zone temperature, which is the desired output, but they say this would be acceptable. Assuming that the standard is to be implemented with software,

they are probably correct in assuming that an appropriate algorithm could be developed to do this iteration in a manner that is transparent to the user. Their method would, however, require the user to provide inputs relating to the heat-transfer parameters between the buffer zone and its surroundings. It would also require a specification of the fractional ontime, something that the standard in other respects has managed to avoid. Lastly, it would require more complex thermal modeling of the buffer zones and more effort to determine additional weather parameters that would include solar radiation effects. Because of the lack of calculation methods and appropriate weather data, Approach A was not used in the standard.

Later, Francisco and Palmiter continued this argument, using additional data gathered in a project cofunded by ASHRAE and DOE (Francisco and Palmiter 1999b). In this project, they performed field measurements of duct efficiency using both “top-down” (co-heating) and “bottom-up” (leakage measurements, etc.) methods. Their work indicated that the treatment in Standard 152 biased the distribution efficiencies upward by an average of two percentage points, because the regain factor as used in the standard overcompensated for the fact that the assumed buffer-zone temperatures were too low.

Subsequently, Francisco and Palmiter (1999a) have proposed additional changes to the way Standard 152P handles buffer-zone temperatures, with a view to correcting certain inaccuracies in the way that heat and mass balances are treated. Discussion of these proposed changes is expected to continue, but the Standard 152P committee has reached a consensus that further study is needed before attempting to incorporate them in the standard.

4 FORCED-AIR SYSTEMS: INTERACTIONS WITH THE LOAD

The two main types of interactions between ducts and the heating or cooling load that Standard 152P treats in detail are thermal regain and infiltration impacts. The third generic type of interaction, off-cycle effects, is addressed in a much less specific way.

4.1 Thermal Regain

Researchers and HVAC industry professionals took longer to recognize the significance of energy losses in duct systems than they did to address other modes of inefficiency in residences. Although the reasons for this are complex, it is perhaps possible to single out two as being of particular importance. One was the unstated assumption that air leakage from residential systems must be small because the pressures are low in comparison to those generally employed in large commercial duct systems. The other reason was the perception that most of any heat lost from the ducts would still have a beneficial effect of warming the space in which the ducts were located and thereby retarding heat loss from the main body of the house. A similar argument with basement ducts was that “heat rises, the ducts are near the basement ceiling, and so any heat lost goes into the house anyway.”

Although the blanket assurance of arguments such as those was gradually worn away, it nevertheless continued to be recognized that in most cases some of the heat (or cooling effect) that is lost from a duct system will in some sense of the word be “regained” via mechanisms such as those mentioned. In some cases, such as well-vented attics, the regain effect is sure to be small and might safely be ignored, but in other cases, such as basements, to ignore the thermal regain would be to seriously underestimate the efficiency of the duct system. The ASHRAE committee responsible for developing Standard 152 therefore had to come to grips with this issue.

The issue of thermal regain has been addressed both theoretically and experimentally, most notably in the heating mode. A theoretical treatment of thermal regain (Andrews 1994) considered the buffer space as an intermediate reservoir located between two thermal resistances, one connecting it to the house and the other connecting it to the outside. This report then went on to consider the case of two different heat sinks, for example the outside air and the ground. It then looked at some data from the ASHRAE SP-43 project to determine whether the basements that were studied showed an identifiable level of regain. These basement cases had R8 insulation in the basement walls and no insulation in the basement ceiling. Two different analysis approaches yielded regain factors (fractions of lost heat effectively recovered) of 0.59 and 0.68. A field study of basement homes in New York and Wisconsin (Strunk et al. 1997) reported a net annualized energy savings averaging 9% when ducts were sealed and insulated. This was a lower net gain than was generally expected for homes with more exposed duct systems, such as attics and crawl spaces, and the researchers stated that Standard 152P was able “to predict the average estimated percentage heating savings for the group with a very good degree of accuracy, although individual house predictions did not always match well with changes in estimated percentage savings.” This was further indication that thermal regain is a real effect, and that the factors used in Standard 152P were at least roughly correct for basement ducts.

In the earliest versions of Standard 152P, thermal regain was accounted for by the use of a simple “thermal regain factor,” F_{regain} , which was used to add back a fraction of all the energy losses from the ducts, the fraction depending only on the location of the ducts within the building. For example, attic ducts were assigned a regain factor of 0.10 while ducts in a ventilated crawlspace with an insulated building floor was given $F_{\text{regain}} = 0.17$. In the midrange of regain factors were uninsulated basements and vented, uninsulated crawl spaces, both of which were assigned $F_{\text{regain}} = 0.50$. At the high end were ducts in a basement with insulated walls ($F_{\text{regain}} = 0.75$) and buried in a slab ($F_{\text{regain}} = 0.90$). These factors and others like them were obtained through theoretical calculations using thermal resistance values judged to be typical (Walker 1998).

Two major issues subsequently emerged concerning the treatment of thermal regain. One of these was the question of whether standardized thermal regain factors that the user would pick out of a table would provide a sufficiently accurate treatment of the interaction between the ducts and the building that they were supposed to represent. Proponents of the table lookup approach argued that anything more complex would result in the standard not being used. Those who wanted a more theoretically defensible

treatment did so because they considered the existing values arbitrary and unreflective of reality in many cases. The table-of-values approach persisted for a long time, but finally in 1999 the table of values was removed and replaced by a calculated ratio: the thermal resistance between the buffer space and the conditioned space divided by the total thermal resistance of the buffer space and all surrounding spaces, whether inside or outside the building. Implicit in this change is the assumption that the user will measure or estimate the thermal resistances and convective heat flows occurring between the buffer space and its surroundings, which for diagnostic use will probably not be the case. Rather, the plug-in factors will probably continue to be used as a practical matter (see, for example CEC 1998), but the committee felt that it should not endorse, at least formally, any particular values. The wisdom of this choice is not unanimously agreed, but it is, as of this writing, the sense of the committee majority.

The second issue was more fundamental. The earliest prototype of the standard treated thermal regain as applying to all the duct losses without discrimination, but various problems soon turned up that were traced to this assumption. For example, sample runs of the Standard 152P calculation procedures seemed to indicate that distribution efficiency could improve if the return ducts were made to leak. This effect was strongest in buffer spaces with a high thermal regain factor. It was soon recognized that applying thermal regain to return leakage losses is incorrect. The November 1996 version of Standard 152P already had this correction, which was effected by allowing regain only of that portion of the losses not attributable to return leakage.

Later, a more subtle issue along the same lines arose, namely that the regain factor for the supply and return ducts can be different if they are located in different spaces. This was championed by Larry Palmiter and Paul Francisco, who proposed a change in the equations by which regain is accounted for. These changes were adopted by the committee and included in the 1999 version of the standard.

Thermal regain continues to be a subject for lively discussion within the committee, with Palmiter and Francisco having proposed additional improvements in the way it is treated. This may be revisited in a future revision of the standard, but the current consensus was to move ahead with things as they were.

4.2 Infiltration Impacts

The second major interaction between ducts and the load is the impact of unbalanced duct leakage on the air-change rate in the house. In the first two years of the standard's development, the method of computing this underwent significant oscillation, but by 1996 a particular approach had been settled upon. This was based on a comparison of several models with data, the results of which have unfortunately never been published formally. Francisco and Palmiter (1997) criticized the process of comparing the models as having considered excessively high wind speeds and excessively large unbalanced leakage rates, and recommended that the comparison of models be redone for a more restricted set of wind speeds and unbalanced leakages representative of what would be found in most houses most of the time.

The general philosophy of the method that was adopted can, however, be described. The first step is to estimate the natural infiltration rate Q_{inf} that would obtain in the absence of a duct system. Actual measurement of this rate, for example using tracer gases, was considered to be beyond the scope of Standard 152. A means of estimating it was therefore needed. The “Gordian knot” was cut by simply assuming that the natural infiltration rate would be 0.35 air changes per hour, although an option of using ASHRAE Standard 136 (1993) was provided. The unbalanced duct leakage Q_{imb} is then defined as the absolute value of the difference between the supply and return leakage rates. The next step was to calculate a “net” infiltration rate Q_{net} when the system fan is on, using these two values as inputs.

If the supply leakage exceeds the return leakage, then the operation of the system fan pulls down the pressure in the house, and this increases the infiltration rate. The effect is not a simple addition of Q_{inf} and Q_{imb} , however, but rather a complex manipulation of the neutral level. In the limit of highly unbalanced leakage, the neutral level is pushed above the ceiling, and the pressure difference across the envelope is negative everywhere. In this case, $Q_{net} = Q_{imb}$. For unbalanced leakage near zero, $Q_{net} = Q_{inf}$. Any number of mathematical models could satisfy these boundary conditions, but the one adopted had the following form:

$$Q_{net} = (Q_{inf}^{1.5} + Q_{imb}^{1.5})^{0.67} \quad (5)$$

If the return leakage dominates, then the duct system is blowing air into the house, and the situation is somewhat more complicated. If the unbalanced leakage exceeds the natural infiltration rate, then infiltration is cancelled and Q_{net} is equal to zero. If the unbalanced leakage is between zero and Q_{inf} , then a formula must be applied linking the functional values at the two end points of this domain. The formula adopted is

$$Q_{net} = (Q_{inf}^{1.5} - Q_{imb}^{1.5})^{0.67} \quad (6)$$

Once the net air-change rate is determined, the next step is to calculate its impact on the load. This was based on a calculation of the excess heating or cooling load imposed by the increased infiltration rate $Q_{net} - Q_{inf}$ (which could be a decrease if $Q_{inf} > Q_{net}$).

For the heating mode, this effect is determined as follows:

The heat delivered to the house during one unit of ontime is $E_{cap} DE$. The additional load imposed by the excess infiltration is the excess infiltration airflow multiplied by the volume specific heat of air multiplied by the indoor-outdoor temperature difference, or $(Q_{net} - Q_{inf}) \rho_{in} C_p (t_{in} - t_{out})$ in Standard 152P nomenclature. The load factor then equals one minus the ratio of the latter quantity to the former. The point to note here is that the fractional ontime drops out of the equation, because the added infiltration and the intentional heating or cooling are both operative over the same time frame, namely when the equipment and the fan are both on. This points up the fact that Standard 152P assumes simultaneous equipment and fan operation. If the fan operates continuously, for

example to provide ventilation, there will be an additional impact that will be ontime dependent. This is not included in Standard 152P at the present time.

4.3 Off-Cycle Losses

It has long been recognized that there are thermal losses from a duct system during the period when the system is not operating. Three major mechanisms for such losses have been identified:

- Cooling down of the ducts (or warming up, in the cooling mode) after system-fan shutdown at the end of the on-cycle. This is the cycling loss factor (F_{cycloss}).
- Air infiltration into the building envelope through holes in the duct system, which, during the off-cycle, act as additional holes in the envelope.
- Transfer of heat from the conditioned space to a buffer space containing ducts caused by the formation of convective loops (thermosiphoning).

Originally it was intended that Standard 152P would account for all of these effects. As time passed, the committee gradually became less ambitious in this area, until now only the first of these effects is treated within the standard, and that in a quite simplified way.

The cycling loss factor was originally specified as a table lookup containing four values as a function of the duct material (plastic flexible duct or sheet metal) and insulation level. The ratio of energy lost to energy delivered ranged from 1.2% for R-4 plastic flex duct to 4.9% for uninsulated sheet metal ducts. The seasonal distribution efficiencies for heating and cooling were reduced by these percentages to account for thermal losses during the off-cycle. No such correction was made for design conditions because the off-time is much less. By 1996, this had been simplified to a two-item table, with all nonmetallic ducts (duct board and plastic flex duct) being assigned $F_{\text{cycloss}} = 0.02$ and sheet-metal ducts receiving an F_{cycloss} value of 0.05. This remains the case today.

Off-cycle infiltration losses via leaks in the ducts was originally accounted for in Standard 152P. The calculation was fairly complicated, essentially pro-rating the air infiltration rate without ducts upward to include the effective leakage area of the ducts for the fraction of the time the system was off. This required a lot of information that one was unlikely to obtain short of making the house a long-term research project, which was never envisioned for Standard 152, even for the research pathway. By the end of 1996, the committee decided to drop any attempt to account for off-cycle infiltration losses assignable to duct leaks.

One byproduct of this decision was that the treatment of the on-cycle infiltration losses became much simpler. Specifically, it was possible to eliminate the fractional on-time as a specific parameter, because during the derivation it dropped out of the equation. The fact that it does so may seem mysterious, but it happens because the infiltration impact on distribution efficiency is a function of both the fractional on-time and the load. Assume that the infiltration losses, as determined by the unbalanced duct leakage and the natural infiltration rate of the house, are a fixed number per unit time of operation. If the load is

kept constant and the fractional on-time is increased, then the infiltration losses as a fraction of the load will also increase. On the other hand, if the fractional on-time is held constant (meaning that the infiltration losses caused by system operation are kept constant) and the load is increased, then these losses as a fraction of the load will decrease. A detailed derivation shows that the impact on distribution efficiency is a function of the ratio of the fractional on-time to the load, and this is inversely proportional to the equipment capacity. The upshot is that the formula for the load factor has the equipment capacity as one of its parameters, but not the load or the fractional on-time. This is a godsend, because the equipment capacity is generally easy to determine, as by looking at the nameplate, while the other two parameters are difficult to quantify.

The energy penalty caused by thermosiphon loops was originally included in the Standard 152P, based on research conducted at Lawrence Berkeley National Laboratory. Only one type of duct system configuration was given what amounted to a penalty factor for this effect, namely attic ducts with furnace in the garage in the heating mode. The severity of the efficiency penalty depended on the level of insulation on the ducts, ranging from a 5% penalty for R-12 (IP) insulation to a 16% penalty for R-2. This was included in the January 1996 version of the standard, but by the end of that year it had been eliminated. The reasons for doing this were, first, that it seemed unfair to restrict the penalty to only one type of duct system when others might also have such effects, and second, the amount of research was insufficient to justify including specific numbers, although it was generally understood that the effect is real.

5 FORCED-AIR SYSTEMS: IMPACTS ON EQUIPMENT EFFICIENCY

Two types of interactions between duct efficiency and equipment efficiency have been recognized, one a minor impact that occurs in most systems and the other a major impact that is associated with only a few kinds of equipment.

5.1 Equipment Cycling Efficiency

The minor impact is the effect that duct efficiency has on equipment cycling efficiency. This is explained as follows. Everything else being equal, if the duct efficiency is reduced, the heating or cooling load will increase because of the increased energy losses from the ducts. Because of the increased load, the equipment will cycle less often, and for most types of single-capacity equipment (furnaces and single-speed air conditioners), less cycling translates to greater efficiency. So when the ducts get worse, the equipment gets better.

The effect is far from an even trade, however. Changes in duct efficiency of tens of percentage points generally produce countervailing changes in equipment efficiency of a percentage point or less. This is illustrated by some results from the ASHRAE SP-43 Project (Jakob et al. 1986b). In this field-validated computer simulation, various conditions of duct leakage and insulation were studied, ranging from uninsulated ducts with leakage equal to 20% of system fan flow on both the supply and return sides to R-5

insulation and no leakage. Going from the worst to the best case improved delivery effectiveness (which they call “duct efficiency”) from 58% to 77%, a 19 percentage-point improvement, but furnace efficiency declined by only 0.7 percentage points, from 75.7% to 75.0%. That is, a 32% improvement ($77/58 = 1.32$) in delivery effectiveness was accompanied by just a 1% decline in furnace efficiency. Although this effect is probably as well established as anything in Standard 152P, the committee decided to ignore it because of its small size.

5.2 Effect of Variable Equipment Capacity

The other impact of duct efficiency on equipment efficiency occurs when the equipment has two or more operating modes that differ in both capacity and efficiency. Relevant cases include two-speed air conditioners and single-speed air-to-air heat pumps with electric resistance backup heat. In such cases, the higher-capacity mode is usually the less-efficient one. The consequence of this is that when duct losses increase, the load increases also, and the equipment is forced to deliver a greater proportion of its seasonal heat or cooling while running in its less-efficient mode. The average efficiency of the equipment is, therefore, reduced. Moreover, this efficiency penalty is over and above the penalty imposed by the increased duct losses themselves. This effect is accounted for in Standard 152P by means of the equipment efficiency factor F_{equip} , which is less than 1.0 in these circumstances.

The origin of the equipment efficiency factors in Standard 152P is explained in Walker (1998). Walker found that for variable-capacity furnaces, duct efficiency has a noticeable impact on furnace efficiency, but the effect is not a large one. For example, raising delivery effectiveness from 60% to 80% (comparable to the case studied by Jakob et al. as discussed in Section 5.1) was found to improve furnace efficiency by 2%. Note that this is in contrast to the single-capacity case discussed in Section 5.1, where a comparable increase in delivery effectiveness caused a 1% decrease in furnace efficiency. This effect was included in the standard by 1998 and remains a part of it today.

The impact of duct efficiency on the efficiency of two-speed air conditioners was found to be greater still. The same 20 percentage-point increase in delivery effectiveness was found to improve equipment efficiency by ~3.5%.

The greatest impact was found to hold for heat pumps with electric-resistance backup (strip) heat. Here, the same increase in delivery effectiveness as above was found to improve seasonal equipment efficiency by ~14%. This large increase was the result of reduced dependence on strip heat.

More recently, the effect of inadequate airflow has been added to the equipment efficiency factors. Essentially, the argument is that if the ducts are sufficiently restrictive as to cause the airflow to fall below the manufacturer’s recommended value, then the efficiency of an air conditioner will suffer, and the ducts should be penalized for it. The extent of the penalty is embodied in modified formulas for F_{equip} in both single- and

variable-capacity air conditioners. These formulas were developed by the committee on the basis of data supplied by John Proctor of the Proctor Engineering Group.

5.3 Impacts of Equipment Design on Distribution Efficiency

It is important to note that the reverse impact—equipment design affecting the efficiency of a duct system—can be very significant. There is no special factor such as F_{equip} to reflect this, however. Instead, it comes out in the calculation when the distribution efficiencies for various possible equipment choices are considered.

One such study (Andrews 2003) considered the impact on distribution efficiency of varying the capacity of furnaces and air conditioners. It found that the impact on duct leakage losses was relatively small, but that for ducts with typical (R-4 IP) insulation levels, the impact on thermal conduction losses could be quite large, often tens of percentage points for turndown ratios of two-to-one and greater. This illustrates one of the uses envisioned for Standard 152P, namely in projecting the impacts of altering various aspects of system design affecting not only the ducts but also the equipment and the load.

6 HYDRONIC SYSTEMS

An early decision of Standards Project Committee 152P was to include a section on hydronic heating systems. In view of the dominance of forced-air systems in the U.S. housing market, and in view of the widespread perception (which is probably accurate) that forced-air systems as commonly installed today sustain greater energy losses than hydronic systems, one might ask why the committee thought this would be important. There were several reasons for this:

- Although forced-air systems dominate the U.S. market, hydronic systems still serve a significant fraction of homes in the Northeast.
- If hydronic systems really are relatively efficient, they could serve to benchmark forced-air, providing a reasonable target efficiency that forced-air systems should be able to equal.
- The development of a practical, low-cost way to provide cooling with hydronic systems could permit them to make a comeback in the new housing market.

6.1 The Hydronic Simulation Model

The first step in this effort was the development of a computer model for a hydronic distribution system (Andrews 1994). The model was designed for integration into an existing simulation model developed at Lawrence Berkeley Laboratory (Treidler et al. 1993). The LBL Model was based on the widely used building simulation program DOE-2, but in addition had subprograms that provided more accurate predictions of air flows driven by pressure and temperature differences (subprogram COMIS) and duct heat- and mass-transfer dynamics (subprogram DUCTSIM). The hydronic simulation

program was designed to be substituted for DUCTSIM to enable the entire simulation engine to address hydronic systems.

Probably the most significant problem that had to be dealt with in developing a hydronic system simulation model, which is not met with in forced-air systems to any great degree, is the large thermal mass of a hydronic system. Because of this, when the circulating pump shuts off, the system continues to provide heat. Thus, a simple algorithm that shut the system off when the load was satisfied would end up overheating the house unless steps were taken to account for the extra heat residing in the system on circulator shutdown. Because the LBL Model was load-driven rather than thermostat-driven, it was not an option just to let this extra heat flow into the building, extending the time until the next call for heat. The hydronic system simulator had to anticipate the amount of extra heat that would be delivered to the load following circulator shutdown and determine whether this would satisfy the remaining heating load for the hour currently being simulated. When that condition was satisfied, the program would shut the circulator down and let the system “coast” for the remainder of the hour. This experience would prove very valuable when the time came to generate an algorithm for calculating the thermal distribution efficiency in the hydronic portion of Standard 152P.

The hydronic simulation model was run on a stand-alone basis over a one-day period for a typical residential-sized system, for cases with insulated and uninsulated piping outside the conditioned space. The distribution efficiency for uninsulated piping ranged from 78% to 90%, depending on the heating load, while for insulated piping (R-3 IP) the efficiency always exceeded 95%.

Because of funding difficulties, the subprogram was never actually integrated into the LBL Model. However, a way was developed to iterate back and forth between the two models. This was done in the following way:

1. The LBL Model provides a season's series of heating loads without a distribution system.
2. The hydronic simulation model is run with this series of loads, and provides the amounts of heat lost to the buffer zone containing any piping that is outside the conditioned space.
3. These heat losses are fed back into the LBL Model as thermal inputs to the buffer zone, and the LBL Model provides a new set of hourly heating loads as influenced by these heat losses from the pipes.
4. The hydronic simulation model is rerun with the new series of loads, and a new series of thermal inputs and outputs is generated.

This process can be iterated as many times as needed until the inputs and outputs remain stable within some criterion of variability. It was expected that one or two iterations would be sufficient.

6.2 Approach to Hydronic System Efficiency Definition

In order to develop a way to assess thermal distribution efficiency in a hydronic system, it was necessary first to conceptualize the system's operation. As a first step, it was decided to restrict the case to a single-loop one-pipe heating system in which the delivery of heat to the load was accomplished by appropriate lengths of finned-tube heat exchangers. The hydronic loop was divided into four categories of piping:

- Finned piping (generally called "radiation" in the industry);
- Unfinned piping in the conditioned space;
- Unfinned piping in the buffer zone that is uninsulated.
- Unfinned piping in the buffer zone that is insulated

Each of the above categories of piping or radiation was characterized by a thermal capacitance and one or more thermal resistances. Piping and radiation in the conditioned space were assumed to be thermally linked to the outside if they were installed against an exterior wall, although the heat losses through such walls generally turn out to be small unless the wall is uninsulated. Piping in the buffer zone was assumed to be in thermal contact only with that zone.

In addition to the lengths of each kind of piping and the resistive and capacitive characteristics of the piping, the following parameters were required in order to proceed with an efficiency calculation:

- Average boiler-water temperature;
- Buffer-zone temperatures under design and seasonal-average conditions;
- Indoor temperature;
- The design and seasonal-average heating loads of the house;
- The circulator cycle time (ontime+offtime) under design and seasonal-average conditions.
- The volume heat capacity of the circulating fluid (usually water);
- The volume flow rate of water from the boiler into the distribution loop.

A brief comparison with forced-air systems may be useful here. Perhaps the most obvious difference is that in hydronic systems, leakage may be ignored, since leaking water will demand attention long before it becomes a system efficiency issue. Second, in contrast to a furnace, which operates at the same time as the distribution system, a boiler generally cycles on and off independently of when the circulator cycles on and off. Third, measurement of the water flow rate was a tricky problem because it would not be possible, in the context of a diagnostic test, to break into the system to install a flow meter.

Some of these parameter values could be provided as inputs, but others needed to be obtained in the course of the computational algorithm. The development and operation of this algorithm will now be discussed.

6.3 Calculation Procedure

The procedure for calculating distribution efficiency in single-loop hydronic heating systems follows a logical pathway that is fully described in Andrews 1996. A summary of the procedure is given here.

1. Calculate UA Values. The first step in the process is to calculate the overall heat transfer coefficients or UA values for each of the three portions of the distribution system as defined in Section 6.2. This is done using the pipe dimensions and insulation R-values, or in the case of the finned-tube radiation, of information on the heat transfer properties of the finned-tube units themselves. This data was originally to be obtained either from manufacturer's data. Later, an alternative method was added in which the fin area and spacing could be used to determine the required information, eliminating the need to have written specifications available.

2. Calculate the Log-Mean Temperature Difference. The log-mean temperature difference between the pipe loop and the indoors is calculated next. The number of heat transfer units (NTU) is set equal to the ratio of the overall UA of the pipe loop divided by the product of the volume specific heat of the circulating fluid and the fluid flow rate.

The method of measuring the water flow rate changed over time. In the earliest versions of the standard (pre-1996), the protocol for hydronic systems actually specified the use of a flowmeter. By September 1996, this was dropped, and a series of measurements of the water temperature at the boiler outlet to the distribution loop and the inlet to the boiler from the distribution loop was used to determine the log-mean temperature difference directly, without the interim step of measuring the flow rate. This was later found to be unworkable in the field, because these temperatures tended to fluctuate unpredictably as the burner fired and then shut down.

The current method was then devised, tested in the field, and found to give workable results. In this procedure, the thermostat is turned down so that the temperatures in the pipe can relax to near-equilibrium "cold-start" values. Meanwhile, the boiler temperature is maintained in its normal range. The thermostat is then moved up well past the current room temperature, so that the circulator comes on and stays on long enough for the first slug of hot water to make its way through the system and back to the boiler. The time to breakthrough of this warm water is then used, together with the calculated interior volume of the pipe loop, to determine the water flow rate.

3. Calculate Steady-State Heat Transfer Rates. Steady-state heat transfer rates from the pipe loop to the ambient, to the buffer zone, and to the conditioned space are calculated using the thermal resistances, the log-mean temperature difference between the piping and the conditioned space, and the temperature difference between the conditioned space and the buffer zone. The last of these will have different values for design and for seasonal conditions.

4. Calculate (or Estimate) Heating Load. Evaluation of the distribution efficiency for hydronic systems requires a knowledge of the heating load. The standard calls for this to be calculated using one of three manuals in common use. It also specifies that the load shall not be set greater than 80% of the steady-state capacity of the hydronic distribution loop. Although not explicitly allowed by the standard, as a practical matter this 80% criterion can be used as a handy default, as long as its use is acknowledged.

The design heating load was set equal to 60% of the steady-state heat-transfer rate for the hydronic system. This was in line with a similar setting then being used in the forced-air section (though later the algorithm for forced-air was changed so that this factor was not needed). The seasonal-average heating load was set equal to one-third the design load. This choice was made (Andrews 1996) on the basis of weather data studies in which the design load was taken to be proportional to 65°F minus the ASHRAE 99% heating design temperature and the seasonal 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, and West census regions.

5. Calculate Delivery Effectiveness. Delivery effectiveness is calculated via several intermediate steps. First, the relaxation times of the four classes of piping are calculated using the thermal resistance and capacitance values for each. Second, the circulator on-times for design and seasonal-average conditions are calculated. The equations for this require a knowledge of the total (on plus off) cycle time. Fortunately, a parametric study showed that the delivery effectiveness is quite insensitive to the cycle time, so the standard specifies default values for these. The procedure for determining these defaults is set up to ensure that the ontime will have a minimum value sufficient to circulate water through the entire pipe loop. The equations for ontime are “forward-looking” in that they take into account the amount of heat that will be delivered to the load from cooling of the thermal mass of the system during the off-cycle.

The third calculation is of the heat flows during the on- and off-cycles. The off-cycle heat flows account for the decaying exponentials in the temperatures within the loop. For the on-cycle, steady-state heat flow rates over the calculated ontimes are assumed. No provision is made for the effect of initial warming of the pipe on circulator startup. The heat flows that can be considered losses are then summed together, as are those that represent delivered heat. The delivery effectiveness is then the delivered heat divided by the sum of delivered heat and lost heat.

6. Calculate distribution efficiency. Going from delivery effectiveness to distribution efficiency is a relatively simple matter in hydronic systems. First, any impact of the distribution system on boiler efficiency has been judged to be negligible. There could be cases where this is not so, but as long as the boiler is operating within a few percentage points of steady-state efficiency, the assumption cannot be far wrong. Second, there is no infiltration impact. Third, there is no need to worry about which losses the thermal regain applies to, as there is in forced-air systems, because in hydronic systems there is

only one kind of loss, analogous to conductive supply losses in forced-air systems. Therefore, the thermal regain factor applies to all the heat lost from the piping to the buffer space.

6.4 Hydronic Radiant Heating and Cooling

At the suggestion of members of ASHRAE Technical Committee 6.5, Radiant Heating and Cooling Systems, the hydronic section of the standard was modified to permit the inclusion of radiant panels and subfloor radiant systems served by hot water. Also, the possibility of sensible cooling using circulating chilled water was also included for these radiant systems, even though cooling cannot be done using finned-tube baseboard units. The algorithm for radiant heating and cooling is essentially the same as for baseboard heating, the heat transfer coefficients for the radiant systems, based on surface area, being derived from those for finned-tube baseboard, based on length.

7 OTHER DISTRIBUTION SYSTEM TYPES

ASHRAE Standard 152P also includes a brief section on Electric Distribution Systems. This includes electric baseboards and electric radiant panels. The inclusion of refrigerant distribution has always been envisioned to occur eventually, even though these were removed from the scope in recognition that this wasn't going to happen immediately.

7.1 Electric Distribution Systems

For electric baseboard heaters, the only loss considered is heat transfer through the wall in back of the baseboard units. This is calculated the same way as the equivalent loss factor in hydronic systems. This is usually a small number, especially since it is unusual to have electric baseboards in homes with substandard insulation levels. For electric radiant panels, a similar calculation is done of the split between delivered heat and heat lost from the back of any panels mounded against exterior walls.

7.2 Refrigerant Distribution Systems

As already discussed, it was originally intended that these systems, exemplified by the increasingly common "mini-split" air conditioners, would be covered under ASHRAE Standard 152P. When it was realized that no one was prepared to develop the necessary background information and detailed algorithms for evaluating distribution efficiency, this system type was removed from the scope. It was believed that these systems usually display quite high distribution efficiencies, but no one knew quite how to do the calculation on a basis that could be defended.

In the fall of 1999, a national laboratory was given the task by the DOE Program Manager responsible for thermal distribution to perform theoretical calculations of the thermal losses in these kinds of systems due to various mechanisms. As a starting point, a set of "Notes on Refrigerant Line Losses" by C.K. Rice of Oak Ridge National

Laboratory was used to categorize the losses by location (suction line, liquid line, or discharge line), by mechanism (thermal effects, pressure drop effects, and elevation difference between the indoor and outdoor coils), and by mode (cooling or heating). Equations for the losses from each location, mechanism, and mode were developed (Andrews 2000c) and the losses from each source quantified for a 2.5-ton system with various refrigerant-line lengths and internal cross sections. Losses caused by differences in elevation between the evaporator and condensor coils were also considered. The conclusion was that the most significant impact on delivery effectiveness was that caused by pressure drops in the suction line in the cooling mode. This could be as much as 5% for the narrowest refrigerant line studied (0.5 in. ID), but was less than 2% for the next higher size (0.625 in. ID). All other impacts were 1% or less. The total energy loss in the cooling mode was ~1% or less except for the 0.5 in. ID suction-line case, where it was ~5%. In the heating mode, the loss was <1% in all cases.

On the basis of this work, a possible section for a future version of Standard 152P was developed (Andrews 2000d). This report first extended the calculations of Andrews 2000c to include a range of system sizes and refrigerant line parameters. On the basis of these calculations, an algorithm for calculating delivery effectiveness was developed. The inputs to this calculation are refrigerant line length, outer diameters of refrigerant lines, insulation level of the line used to transport refrigerant vapor, and the rated capacity of the air conditioner or heat pump. Table lookups for pressure, temperature, and cycling impacts were developed, so that the calculation of distribution efficiency involves simple plugins to an uncomplicated formula. It should be pointed out that the resulting algorithm has not yet been validated by any laboratory or field testing.

8 THE PUBLIC REVIEW PROCESS

Any proposed ASHRAE standard must go through a process of public review before it can be considered for formal adoption as a regular standard, i.e., with the “P” removed from its designation. In the case of Standard 152P, this process has been somewhat lengthy for the following reasons:

1. Early on in the process, consideration by ASHRAE was delayed due to staffing limitations.
2. Revisions in the standard by the committee continued even after the initial decision to request public review was made.
3. Certain aspects of the standard proved to be sufficiently controversial that significant effort was required to resolve some commenters, and in addition one commenter remained unresolved even after such efforts were made.

The first effort to move towards public review occurred in a letter ballot in the autumn of 1996. Although the motion passed, there were three negative votes and enough constructive suggestions that it was decided to make changes and re-vote. Enough uncertainty was expressed about whether the standard was ready for “prime time” that an alternative suggestion was broached, namely that instead of a traditional public review,

that a relatively new ASHRAE option for “public review and trial use” be opted for instead. The merit of this suggestion was that it would put the developing standard “on the table” in an official way and yet permit changes to be made on a continuing basis. A motion to submit the standard for public review and trial use passed unanimously at the January, 1997 ASHRAE Winter Meeting in San Antonio.

Six months later, at the next ASHRAE meeting in Boston, the standard had undergone enough changes that another public review ballot was deemed necessary. Among these were minor technical changes in the Title, Purpose, and Scope (which had to be formally balloted by the committee and approved by the ASHRAE Standards Committee), as well as certain technical details, such as error allowances in the hydronic system measurements and limitations on conditions under which the House Pressure Test for duct leakage would be allowed. The upshot was the committee once again voted on a public-review draft of the standard, recommending (unanimously) public review and one year of trial use.

A year then passed, during which ASHRAE had not taken any action on the draft standard. At the June 1998 meeting in Toronto, the Standard 152P committee noted that it had been assured that making changes to this draft would not cause the standard to lose its place in the queue of proposed standards awaiting ASHRAE staff review. The reason for the delay was a temporary shortage of key staff, which was shortly to be rectified. Changes that the committee now made to the draft included required data sheets for inputs and outputs, inclusion of infiltration in the calculation of thermal regain, inclusion of temperature data within the standard to make it self-contained, specification of indoor conditions so that results from the standard would be more uniform when different users rate the same system, changes to the hydronic system measurement procedure, a minor change in the load-factor equation, and specification of separate thermal regain factors for the supply and return ducts. Provision was also made for the calculation of reference indoor humidity conditions using a simple algorithm, as discussed in Tenwolde and Walker 2002. These changes were added to the draft of the standard proposed for public review and transmitted to ASHRAE.

8.1 First Public Review

By January 1999, at the Chicago ASHRAE meeting, the Standard 152P committee learned that approval for public review was at hand. A First Public Review Draft, dated May 1999, was published by ASHRAE, and comments from a 60-day public review, ending July 8, 1999 were formally solicited. These comments were circulated to committee members and discussion took place during the summer and fall (mainly by email) on how to address them. In many cases, the comments were either editorial in nature or seen by the committee members as helpful or clarifying, and they were adopted. In other cases, discussions with the commenters resulted in solutions acceptable to both the commenter and the committee. Several comments, however, were less easily dealt with. These were as follows:

1. A comment was made to the effect that the standard was more a calculational tool than a test method. The commenter advocated removing the words “Method of Test” from the title. The committee’s response was that although there is a great deal of calculation in the standard, this is true of most other ASHRAE test methods as well. The committee took the view that the most important things that need to be measured, i.e., duct leakage and system fan flow, are actually measured in the standard, while other factors affecting thermal distribution efficiency, e.g., duct surface area and insulation level and the characteristics of the equipment and the house envelope, are adequately addressed.
2. Several commenters suggested changing the specified indoor temperatures from 68 °F for heating and 78 °F for cooling to 70 °F for heating and 75 °F for cooling. The committee saw precedents in ASHRAE for both choices but in view of uncertainty on which was better, the committee decided to keep the existing set.
3. Several comments were related to the fact that the standard did not cover many possible types of heating and cooling systems. The concern in at least some of these cases seemed to have been that the systems not covered by the standard would suffer a marketing disadvantage relative to those that were covered. The committee’s response was that it could not cover everything, but had to focus on the most common types of systems.
4. A commenter maintained that the duct leakage test methods in the standard were unreliable. The committee responded that one of the two tests—the House Pressure Test—was the subject of much soul searching within the committee itself, but the view was held forth that sufficient limitations were put on conditions under which this test would be accepted that it should be kept in the standard. (However, it was later removed.) As for the duct pressurization test, the committee expressed the view that, first, it is highly repeatable and, second, although it is subject to larger errors than one would like, it was still the best test available and “state of the art” as far as leakage testing is concerned.

In accord with ASHRAE procedure, the responses of the committee, as finally adopted, were compiled and sent to the commenters. This gave the commenters the opportunity to say whether they were satisfied or were still unresolved.

8.2 Second Public Review

After the process described above had been completed, three of the commenters on the First Public Review remained unresolved. Some members of the Standard 152P committee itself also had issues they felt still needed to be addressed. At the June, 2000 ASHRAE meeting in Minneapolis, the committee settled on the following strategy for getting the standard through the approval process:

1. Address the unresolved public review comments.
2. Get agreement on the committee.

3. Complete substantial “house cleaning” on the draft to weed out typographical errors and inconsistencies between the nomenclature and the body of the document.
4. Hold an interim meeting of the committee in August, 2000.
5. Make whatever compromises are necessary to get the standard “out the door.”
6. Vote on a Second Public Review before the next ASHRAE meeting.

The committee decided to bring whatever data could be obtained that might clarify certain issues of importance. These included:

1. Whether the House Pressure Test could be shown to produce acceptable results under some set of restricted conditions.
2. How to handle the impact of airflow on efficiency.
3. How thermal regain in attic ducts is related to attic temperature.
4. Whether there was a sufficient information base to include the effects of continuous fan operation in the standard.

The interim meeting was held in August as planned at the 2000 ACEEE Summer Study on Energy Efficiency in Buildings at Pacific Grove, California. All of the above topics were discussed, but in some cases the necessary information to make a decision was not available. Much of the time was devoted to two issues: how the equipment efficiency factor should be related to airflow through the unit, and whether the treatment of buffer zone temperatures and thermal regain should be modified.

At the January 2001 ASHRAE meeting in Atlanta, the committee approved changes to the standard based on the discussions in August and subsequent emails among the membership, and voted to request a Second Public Review. A Second Public Review Draft (ASHRAE 2001) was prepared, but the public review did not actually occur until the summer of 2001. At the June meeting of ASHRAE, therefore, the committee devoted much of its efforts to brainstorming possible directions that might be taken on future versions of the standard, once the current version is adopted.

The comments from the Second Public Review were received by committee members in October, 2001, and members worked via email on the question of how to address them, so as to be ready by January for a formal vote on its response. A summary of the comments and how they were addressed by the committee is as follows:

Commenter 1 made several suggestions for clarifying the hydronics portion of the standard. These were accepted by the committee. One additional comment, requesting a section of the standard that would deal with hybrid forced-air/hydronic systems, was rejected on the ground that the standard could already be used to evaluate the hydronic and forced-air portions of such a system independently.

Commenter 2 caught several typographical errors, sought a clarifying statement in the foreword indicating that the standard does not address thermal comfort issues, asked for substantiation of certain weighting factors in the standard, and requested an uncertainty analysis to indicate how much the results of the standard depend on the inputs. The

committee agreed to the requested changes, indicated the source of the weighting factors, and provided an analysis of the impacts of measurement uncertainties.

Commenter 3 made the following claims:

- Experts, including members of the Standard 152P committee, had shown that the standard was not accurate.
- Other long-standing test methods are better.
- The standard has been misapplied by government agencies even in advance of its acceptance by ASHRAE.
- The standard emphasizes duct airtightness to the exclusion of airflow at the system fan.
- An unproven new device for measuring system fan flow was hurriedly included in the standard.
- Thermal comfort needs to be included.

Commenter 4 submitted 29 comments on many aspects of the standard, some editorial and some substantive. Some of these were accepted immediately by the committee while others were resolved in discussions with the commenter.

The upshot was that Commenters 1, 2, and 4 indicated that their issues were resolved. In the meantime, it was also clear that it was unlikely that Commenter 3 would ever be resolved. The committee prepared a response to this commenter's statement that included the following points:

"1. Commenter states that "ASHRAE publications alone show the test methods are inaccurate and inconclusive."

"The question of accuracy usually focuses on the test for duct leakage. The duct leakage test has two parts. First, the leakage hole size is measured using fan pressurization. Second, this is pro-rated to an actual leakage rate using an estimate of the effective leakage pressure in the duct. Research reported in the ASHRAE transactions indicates that the hole-size measurement is quite accurate. Most of the uncertainty in this test is in the estimation of the leakage pressure. In Standard 152P, this is taken to equal one-half of the measured static pressure at the plenum. In cases where most or all of the leakage occurs in the plenum or near the registers, this estimate can be sufficiently far from the truth that errors of 30% to 50% can occur. On average, however, the errors in actual houses are almost certainly much less, because the leaks are usually not so concentrated.

"The critical point to remember, however, is that any error in leakage will translate into a much smaller error in the thermal distribution efficiency. For example, in a sensitivity study of Standard 152P conducted by a national laboratory (Walker 2001), distribution efficiencies for heating and cooling were calculated using Standard 152P for ducts in 3 building locations (attic, crawlspace, basement) in six climates. Cases with supply and return leakage each equal to 14%, 10%, and 5% of system fan flow were studied. If the 10% leakage

is taken as a benchmark value and the 14% and 5% leakages are taken to represent “errors” of +40% and –50%, respectively, the impacts of these “errors” on thermal distribution efficiency can be tabulated. On this basis, the average “error” in the distribution efficiency was 3 percentage points for heating and 4 percentage points for cooling. Although these are far from trivial, they are not, in the opinion of SPC152P, large enough to negate the value of Standard 152. The alternative is to have no standard at all, leaving individuals, agencies, and associations with no recourse but to devise their own standards, which are likely to be inconsistent and poorly documented.

“2. The commenter then goes on to say that the committee has “dismissed time proven methods of testing authored by ASHRAE and practiced by the HVAC and air balancing industry for decades.”

“From informal discussions with the commenter, we have learned that the method he prefers involves the use of flow hoods to measure the total airflows into the return registers and from the supply registers. A different method is then used to calculate the airflow at the system fan. The return leakage is taken to equal the difference between the system fan flow and the total flow into the return register(s). The supply leakage is taken to equal the difference between the system fan flow and the total supply-register flow. There are several problems with this method. First, it is prone to much greater errors than the fan-pressurization test because the answer is obtained by subtracting two quantities of approximately equal size. If, for example, the system fan flow is 1200 ± 100 cfm and the return-register flow is 1000 ± 100 cfm, then the leakage (assuming quadrature addition of errors) is 200 ± 140 cfm. The problem is made worse by the fact that the system fan flow and the register flows are measured using different methods that are difficult to cross-calibrate. Moreover, even if all these issues could be overcome, what is measured is total leakage flow, i.e., leakage both to the inside of the house and to the outside. What is needed is leakage to the outside only, because that is what affects efficiency.

“To sum things up so far, what the commenter wants the committee to do is reject a method that is admittedly approximate but which represents the best test that is currently available, and replace it with a test that is so inaccurate that it gives meaningless values in most cases (and often even the wrong sign).

“The remainder of the discussion refers to things the commenter said in the ‘Substantiating Statements’ section.

“3. The first three paragraphs can be summarized as a complaint that various government agencies have used parts of Standard 152P in ways that the commenter feels are inappropriate, and that the SPC152P committee should bear the responsibility for this.

“The committee’s main response is neither ASHRAE nor its committees can dictate to governments what they may and may not do. That said, it may also be pointed out that if approval of the standard does anything to affect government actions, it will be to make it more difficult to use “bits and pieces” of the standard, simply because now there will be the counterargument that if ASHRAE has specified a standard, one should use all of it or none of it. But as long as the standard is unofficial, the weight of ASHRAE approval is absent.

“4. Commenter says that “‘Tight Ducts’ have become the only focus of these programs. High static pressures and low airflow are the result of 152P’s test methods.”

“The commenter has voiced this opinion on a number of occasions, saying that Standard 152P focuses solely on duct tightness and therefore causes other problems such as low airflow to be exacerbated. This could not be farther from the truth. First, the idea that tightening ducts leads to low airflows is a fallacy. Studies at a national laboratory³ have shown that sealing ducts usually has only minimal influence on airflow.

“Second, Standard 152P specifically requires airflow to be measured. Far from being ignored, airflow is one of the two parameters that are given the most detailed scrutiny in the standard.

“Third, the commenter appears to think that Standard 152P is a prescriptive standard, i.e., that it tells people to seal ducts. This is not so. It is a method of test. True, it can be used within a prescriptive standard, but in order to assess the effect of any action, such as duct sealing, the standard must be applied twice, once before the action and again after. Each time, the airflow must be measured. Thus, if sealing ducts did affect the airflow, this would be shown in the results from the standard itself.

“Again, far from ignoring airflow, Standard 152P requires just as much care in its measurement as it does for duct leakage.

“5. Commenter states that “Although contained in the standard for years, the air handler flow plate device has just recently become available for purchase. It is experimental at best and was given credibility through this standard before it was proven. Did ASHRAE test this tool before it was allowed to be included in the proposed standard?”

“The commenter’s statement that the flow plate was “in the standard for years” may not be strictly incorrect, since it was added to the draft in February, 2000, which even at the time of the comment was two years ago. The flow plate was not, however, in the May, 1999 public-review version. Thus, in the context of the

³ It was found (Jump et al. 1996, Walker et al. 1998, and Walker et al. 1999) that when ducts were sealed in 33 homes, fan flow decreased by 2.5% on average.

nearly decade-long effort that went into this standard, the flow plate was added relatively recently, and only after extensive testing that was funded by the U.S. Department of Energy. It is the judgment of the SPC152P committee that these tests were of sufficient rigor to warrant inclusion of the device as an option for measuring the airflow at the system fan. The quoted accuracy, which is based on these tests, is $\pm 7\%$. This compares with the $\pm 5\%$ that is generally taken as a benchmark for the alternative test, which involves matching pressures using an adjustable calibrated fan. In the light of these results, the flow plate was considered sufficiently accurate for the purpose of the standard, the slightly lower accuracy well repaid by the ease of use that it introduces into the measurement protocol.

“6. The commenter states that the standard is “incomplete” and that a useful standard would need to include “performance measurement” of the system.

“These comments are made on the basis of discussion at the SPC152P meeting in Cincinnati (which the commenter attended) in which the ability of a thermal distribution system to provide thermal comfort throughout the space was considered. This is apparently what the commenter means by “performance measurement.” The committee long ago stated specifically that Standard 152 would not attempt to address the issue of thermal comfort, because to do so would require an inordinate amount of development time. The standard as proposed assumes that the system distributes the heat or cooling adequately, and simply addresses the efficiency with which it does so. Many on the committee hope that future developments may enable an enhancement of the standard (or perhaps a companion standard) to be developed that will encompass thermal comfort issues, but the committee also feels it is inappropriate to hold the current distribution efficiency standard hostage to such development.

Debates on the above issues (and others) relating to this standard (and other standards) will probably go on as long as ASHRAE continues to exist.

The changes made in the standard as a result of the Second Public Review were far more restricted in scope than those which followed the First Public Review. It was therefore decided, in accord with ASHRAE procedures, to prepare a detailed listing of these proposed changes in a form that could be used in a Third Public Review in which only these changes would be reviewed.

8.3 Third Public Review

The listing of proposed changes (in ASHRAE terminology, “Independent Substantive Changes”) was intended to facilitate a public review that would focus only on these changes, without opening up sections of the standard that were not being changed to additional review. The rationale is that these unchanged sections were already reviewed once and hence did not need to be re-reviewed.

Some of the changes were made in response to commenters' proposals. One might ask why these need to be re-reviewed. The answer is that others, who may have been satisfied with the original wording and therefore did not comment, might now reasonably object to changes made under the urging of commenters who didn't like the original version. Additionally, some changes were made after discussion within the committee. Clearly, these would need to be subject to review as well.

A summary of these changes is as follows:

- The definition of equipment capacity was changed to be more specific regarding rated capacity, and the definition of a forced-air distribution system was changed to make it more explicit.
- In the nomenclature section, unused variables were deleted, some missing variables were added, and units were added to humidity ratio for clarification.
- In the section on forced-air distribution systems, some instrument specifications were made more explicit and realistic.
- An anomaly for Alaskan summertime climates was eliminated.
- Language and equations intended to prevent the deliberate use of low airflows to increase efficiency were eliminated when a calculation showed that such a strategy would not lead to higher efficiency values.

In addition, some references were updated and minor editorial changes were made.

In January, 2003, ASHRAE reported that the third public review had resulted in no comments.

8.4 Final Approval of Standard 152

As of this writing (September 2003) ASHRAE is considering formal approval of the Standard.

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