Title
Technical Background for Default Values used for Forced Air Systems in Proposed ASHRAE Standard 152P

Journal
ASHRAE Transactions, 104(1B)

Author
Walker, Iain S.

Publication Date
1998-09-01

LBNL 40588

Technical Background for Default Values used for Forced Air Systems in Proposed ASHRAE Standard 152P

Published in ASHRAE Transactions, Vol. 104, Pt. 1.

Iain S. Walker

Abstract
ASHRAE Standard 152P (Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems) includes default values for many of the input parameters required to calculate delivery system efficiencies. These default values have several sources: measured field data in houses, laboratory testing, simple heat transfer analyses, etc. This paper will document and discuss these default values and their sources for forced air systems.

1 Introduction

Proposed ASHRAE Standard 152P is a method of test for estimating the efficiency of HVAC energy distribution within residential buildings. In order to be of use to as wide an audience as possible, it contains default values for many of the parameters used in the calculation procedure. The default values were chosen to represent typical values so that they can be used in distribution system design. 152P includes forced air, hydronic, electric and refrigerant systems. This paper concentrates on forced system defaults, but the defaults for design and seasonal temperatures apply to all system types.

2 Design and Seasonal Temperatures and Enthalpies

One of the key parameters used in the standard is the outside temperature because it determines the temperature that distribution systems outside the conditioned space are exposed to. The following calculations provide a method for determining appropriate seasonal outdoor temperatures (and enthalpies for cooling calculations) from design outdoor temperatures. This method uses hourly weather data, weighted by system ontime, to determine seasonal conditions. The length of the season is determined by the number of Heating Degree Days (HDD) or Cooling Degree days (CDD).

2.1. Design Temperatures

Design temperatures for 152P are taken from the ASHRAE Fundamentals Handbook. The handbook gives heating season dry bulb and cooling dry and wet bulb design temperatures. The design values are 2.5% of the heating and cooling seasons. The Heating season is December, January and February (2160 hours) and the cooling season is June through September (2928 hours).

2.2 Seasonal Temperatures

Rather than have the user of 152P determine seasonal weather conditions, the following analysis provides a method of converting design conditions to seasonal conditions. This analysis determines the seasonal conditions from standard weather data files and determines the average difference between seasonal and design temperatures. TMY (NCDC (1980)) data of hourly temperatures were used to calculate seasonal temperatures for heating and cooling seasons. These temperatures are weighted by indoor to outdoor temperature difference so as to simulate system ontime weighting because distribution system loss calculations require the temperature whilst the equipment is operating. It was assumed that building load was proportional to indoor-outdoor temperature difference and that system ontime was proportional to building load. Three example locations were chosen (Los Angeles, Atlanta and New York) that cover a range of weather conditions. The season length was determined by examining NOAA (1980) records for monthly Heating Degree Days (HDD) and Cooling Degree Day (CDD) data. The criteria for determining the season were: from a base of 65°F (18°C), a heating month was assumed if there were more than 150 HDD (in °F). Table 1 shows the results of these calculations.

For heating the indoor temperature was assumed to be 70°F (20°C). Averaging the three locations gives a seasonal temperature **9°C (16°F) higher than the design temperature.**

For cooling a similar procedure to determine dry bulb did not work because most cooling load also depends on solar gains, and outdoor conditions rarely (if ever) produce a net load on the building for the assumed indoor conditions (26°C (78°F), 45%RH). Therefore, a cutoff outdoor temperature of 20°C (70°F) was used instead for averaging outdoor conditions, i.e., all hours with an outdoor temperature above 20°C (70°F) were averaged. Averaging the results from all three cities gives seasonal dry bulb temperature about **9°C (16°F) lower than design temperature.**

2.3 Humidity Calculations

Rather than attempt to seasonally average the humidity conditions, the standard gives the following specifications, and instructions and examples for calculating the enthalpy for duct locations. This is because each duct location has different air surrounding it, thus requiring a different enthalpy calculation. The philosophy for these calculations is to assume that there are no sources or sinks for moisture and therefore the humidity ratios are preserved in different duct locations and only the dry bulb temperature changes. It is also assumed that outdoor relative humidity is the same for design and seasonal conditions.

For example, attics will have the same air as outside (in terms of water vapor content) due to their relatively high ventilation rates, but the dry bulb temperature will be different, however, ducts in basements or exterior walls will tend to have air from inside the building surrounding them (again at a different dry bulb temperature). The source of the air (inside or outside) determines the humidity ratio. Together with the design dry bulb temperature this determines the design enthalpy conditions.

For seasonal conditions there are two calculation methods, depending on the duct location:

1. Ducts exposed to outside air: The outdoor seasonal relative humidity is assumed to be the same as the outdoor design relative humidity. The outdoor seasonal humidity ratio is then determined from this design RH and the seasonal dry bulb temperatures. This

seasonal humidity ratio is used with seasonal duct location temperatures to calculate the seasonal enthalpies at the duct locations.

2. Ducts exposed to indoor air: The indoor humidity ratio is calculated from the indoor wet and dry bulb temperatures. This indoor humidity ratio is used together with the seasonal dry bulb temperature at each duct location to calculate the enthalpy.

3 Design and Seasonal Conditions for Distribution System Locations

The temperatures of each distribution system location are determined relative to the outdoor design temperature to capture climatological differences between building locations. Design conditions are defined to be 2.5% of the season, where the season is as defined in ASHRAE fundamentals Handbook (see section 2.1). The distribution system locations in the standard can be found in Table 11.

3.1 Attics

Calculations are given for well vented and poorly vented attics. These base case temperatures are then corrected for the presence of temperature mitigation factors: radiant barriers, low emissivity exterior coatings and tiled roof systems. The attic temperatures are based on measured attic data from three sources:

- Source 1 - Parker et al. (1997)
- Source 2 - Walker (1993)
- Source 3 - Rose (1997)

These data were chosen because they cover a sufficient time period that seasonal calculations could be made.

3.1.1 Source 1 Attic Measurements

Parker et al. (1997) have analyzed 25 houses in Florida (for summer cooling conditions). 10 Houses could be characterized as well vented attics, with design attic temperatures 12°C (22°F) warmer than outside. Four houses were not well vented and had attics 20°C (36°F) warmer at design conditions. The results for the poorly vented attics are higher than the source 2 and 3 results, due to different attic construction and solar gains.
For seasonal conditions, the source 1 data showed that poorly vented attics were about 10°F (6°C) hotter than outside and vented attics about 5.5°F (3°C) hotter. In addition, white painted roofs averaged slightly cooler than ambient conditions.

3.1.2 Source 2 Attic Measurements

Walker (1993) measured attic temperatures in two attics from 1990 to 1992. Temperatures were measured in the attic air, joists, trusses, attic floor, and in four locations in each pitched roof surface. Attic 1 had no intentional venting and Attic 2 had soffit vents and mushroom cap vents to meet the 1:300 rule of thumb for ratio of vent area to attic floor area. In addition, Attic 2 was equipped with a power fan ventilator for the second winter of testing. More details can be found in Forest and Walker (1992), Forest and Walker (1993), and Walker and Forest (1995). Only small differences (1°C (2°F)) were found between the two years results so they were averaged together here.

The seasonal temperatures were selected to have ambient temperatures > 20°C and are weighted by (20°C-ambient temperature) to simulate ontime weighting. The indoor temperatures were not used for weighting because these houses had no cooling systems. Table 2 summarizes the temperature differences between the attics and ambient temperatures.

3.1.3 Source 3 Attic measurements

A building research laboratory with multiple attic test sections was used to monitor attic performance with a variety of venting strategies, insulation, roof covering etc. The results of tests for four of the test sections are summarized in Table 2.

3.1.4 Summary of Attic air temperatures to be used in 152P

The attic temperatures to be used in 152P were determined by looking for consensus between the above studies. The differences between the results are due to different solar gains, venting arrangements, climates, and attic construction.
For heating, the source 2 and 3 results are close enough that choosing one or the other is not significant. Also, the differences in venting do not produce significant differences in temperature. Therefore, there is no differentiation between vented and unvented attics for heating conditions.
For cooling, the following points summarize the rationale used to select appropriate temperatures:

- *Well vented attic, design conditions*: The range of results was only 3°C (6°F), and the source 1 and 2 results were in good agreement. Therefore the source 1 and 2 data were chosen for this case.
- *Poorly vented attic, design conditions*: In this case, the source 1 and 3 results were the same, with the source 2 results significantly lower, so the source 1 and 3 results were used.
- *Well vented attic, seasonal conditions*: The source 1 and 3 results were fairly close to each other. Taking an average of these results gives attics that are 5°C (9°F) warmer than ambient conditions. The source 2 result was significantly higher, presumably due to differences in solar gain due to longer solar exposure times for northern climates.
- *Poorly vented attic, seasonal conditions*: Like the well vented attic case, the source 1 and 3 results were fairly close to each other. Taking an average of these results gives attics that are 8°C (15°F) warmer than ambient conditions. Again, the source 2 result was significantly higher.

3.1.5 Summary of attic temperature mitigation methods

There are several methods of reducing summer attic temperatures in attics. The methods given credit in 152P are Radiant Barriers (RB), low absorbtivity exterior coatings and the use of tile roofs. In all cases the credit can only be applied to cooling conditions and to well vented attics (vent area/plan area > 1/300). In addition, only RB's that are truss mounted receive credit due to possible longevity problems with attic floor RB's. The magnitude of the credit was initially determined for radiant barriers. The magnitude of the other mitigation methods was then set equal to the RB credit for simplicity. The following sections show how the RB credit was determined and how close the other methods are to the effect of RB's.

3.1.5.1 Radiant Barrier effect on attic air temperatures and duct top surface temperatures.

The following is a summary of some existing research and publications discussing radiant barrier effects. The publications used here are listed in the bibliography. There are a few RB performance effects about which almost all researchers and practitioners agree:

- For heating RB's have a small effect and can be neglected.
- For cooling, to have a significant effect, the attic must be well vented.
- For duct surface temperatures to be reduced the RB must be between the ducts and the roof.
- RB's at the underside of the roof are referred to as Truss Radiant Barriers (TRB).
- It is assumed that foil backed roofing has the same effect as an independent RB. This requires confirmation.

Most research has concentrated on the reduction of heat flow through ceilings rather than attic temperature reduction. The heat flux data can be analyzed to determine the equivalent attic air (or attic floor) temperature changes that would produce these changes in ceiling heat transfer. Therefore, this includes reduction in both radiation and convection heat transfer. X, the fractional reduction in ceiling heat flow is used in Equation 1 to find the reduced attic temperature for supplies, $t_{amb,s}$. X is calculated for both peak (design) and average (seasonal) effects.

$$t_{amb,s} = (1 - X)t_{atticnoRB} + Xt_{in} \qquad \textbf{(1)}$$

where $t_{atticnoRB}$ is the design or seasonal temperature, and t_{in} is the design indoor temperature. The bibliography lists many useful references for RB effects. Here we will use the results of Levins and Karnitz (1987). The temperature implied from the changes in ceiling heat transfer is the effective temperature at the top of the insulation in the ceiling. This temperature includes convection form the attic air and radiation from the other interior attic surfaces and is the correct surface temperature to use for supply duct losses. For supplies:

- The average ceiling heat flow (used for seasonal calculations) was reduced by 30%, therefore X=0.30, and the effective ambient temperature for the supply ducts is:

$$t_{amb,s} = 0.7t_{atticnoRB} + 0.3t_{in} \qquad \textbf{(2)}$$

- The difference between design and seasonal conditions was found by looking at the difference between peak and average heat flow for an RB on the attic floor because this data was not available for the TRB case. The peak ceiling heat flow (used for design calculations) was reduced by 39% with the attic floor RB. The average reduction in ceiling heat flow with attic floor RB was 35%. This implies a peak reduction about 5% greater than the average reduction. Assuming we can apply the same 5% reduction to the other RB results, means that for design conditions X=0.35, and the effective ambient temperature for the supply ducts is :

$$t_{amb,s} = 0.65t_{atticnoRB} + 0.35t_{in} \qquad \textbf{(3)}$$

The return losses are a combination of leakage at the attic air temperature and conduction losses at the combined air/radiation temperature used for supplies. The ambient temperature for the return ducts ($t_{amb,r}$) is assumed to be an average of the change in air temperature and the change in surface temperature due to radiation reduction. If we assume equal contributions of leakage and conduction/radiation heat transfer, then we can average the air temperature with the air/radiation surface temperature given above for the supplies. The attic air temperature for a vented attic with an RB was typically 3°C (5°F) lower than without the RB.

For Returns:

- For seasonal calculations:

$$t_{amb,r} = \frac{1.7t_{atticnoRB} + 0.3t_{in} - 3}{2} \qquad \textbf{(4)}$$

- For design calculations:

$$t_{amb,r} = \frac{1.65t_{atticnoRB} + 0.35t_{in} - 3}{2} \qquad \textbf{(5)}$$

3.1.5.2 Effect of low emissivity outer coatings:

The measurements presented by Rose (1992) showed approximately 2.5°C (5°F) reduction in attic air temperature and 10°C (18°F) lower sheathing temperature for attics with low absorbtivity exterior coatings. Parker et al. (1997) analyzed 25 houses in Florida for summer cooling conditions. This data set contained seven houses with low absorbtivity (<0.4) exterior coatings that averaged 1.4°C (3°F) lower attic temperatures than outdoor air temperatures under design conditions. A single house was tested with and without white painted shingles, and the design attic temperatures were changed from 10.5°C (19°F) warmer than outside to 1.8°C (3°F) cooler than outside.

The following example calculations were used to determine if the attic temperature reductions from RB’s above could also be applied to reduced absorbtivity exterior coatings by using Rose’s results.

Given t_{in}=26°C (78°F) and t_{out} = 31°C (88°F), then Section 3.1.4 gives a seasonal $t_{atticnoRB}$=34°C (93°F) and design $t_{atticnoRB}$ =43°C (109°F).

Using Equation 2 for seasonal conditions, we get: $t_{amb,s}$=32°C (90°F). This is a 2°C (4°F) reduction in seasonal temperature which seems reasonable compared to the results of Rose and Parker et al. Using Equation 3 for design conditions, we get: $t_{amb,s}$=37°C (99°F). This is a reduction of 6°C (11°F) from $t_{atticnoRB}$.

Additional data from Parker (1997), for seasonal temperatures in attics with tiled roofs or white painted roofs showed temperature differences between the attic and outside of 2°C (5°F) and 0°C (0°F) respectively. The change from unaltered attics was 0.5°C (1°F) and 3°C (5°F) respectively. This additional data appears to agree fairly well with the changes predicted in the example calculation given above.

For the attic air temperature used for the return duct calculations in Equations 11 and 12, the reduction of 3°C (6°F) of attic air temperature is close to the 2.5°C (5°F) reduction measured by Rose, and so this correction for return temperature for RB's can also be applied to low emissivity outer coatings.
Given the uncertainty in the measurements, averaging procedure, geographical variations etc., it is reasonable to use the same relationships (Equations 2 through 5) for both RB's and reduced emissivity exterior coatings.

3.1.5.3 Tile Roof Attic Measurements

Proctor (1997) looked at four houses in desert conditions (Nevada), and found that the attics were not much warmer (4°C (8°F)) at design conditions and only 1°C (2.5°F) warmer than outside over a season. These results imply that tile roofs should be given the same attic temperature credit as radiant barriers.

3.2 Garage Temperatures

Garage temperatures were calculated using two different methods. The first method is from an algorithm provided by Parker (1991). A simple empirical relationship was derived to match the predictions of garage temperatures from a computer program. The garage temperature is calculated from the indoor and outdoor temperatures and includes a 24 hour diurnal cycle as well as correcting for the time of year for solar insolation effects. The results of method one are given in Table 3.
The second method is also from Parker (1997) where measured outdoor and garage temperatures for a single garage in Florida were used to determine mean garage to outdoor temperature differences for design conditions.

Calculation procedure for method 1:
garage median temperature [MED] = $0.813(T_{out}) + 0.360(T_{in})$
garage minimum temperature [MIN] = $0.645(T_{out}) + 0.502(T_{in})$
garage maximum temperature [MAX] = $0.950(T_{out}) + 0.083(T_{in})$
The garage temperature is then given by:

$$t_{garage} = MED - \left(|MED - MIN| \cos\left(\frac{2\pi(hour - PC)}{24} \right) \right) \quad \textbf{(6)}$$

where hour is the time of day and PC is a phase correction, given by:

$$PC = 3.5 + 0.0192|182.5 - JulianDate|$$

The measured data showed that the garage temperature is about 0.5 °C (1°F) warmer than outside at cooling design conditions. For heating conditions, the garage is about 4°C (7°F) warmer at design conditions. These measured values show smaller differences between the garage and outside than the values given in Table 3. However, these measured results are for a different climate, so direct comparisons are difficult. Because the measured and predicted values are not too different this is not critical.
For garages in 152P, the following values (based on the results of Method 1) are used:

Heating Design : $t_{garage}=t_{design}+9°C$ (16°F)
Heating Seasonal : $t_{garage}=t_{seasonal}+7°C$ (13°F)
Cooling Design : $t_{garage}=t_{design} + 3°C$ (5°F)
Cooling Seasonal : $t_{garage}=t_{seasonal} + 5°C$ (9°F)

3.3 Basement Temperatures

Basement temperatures were calculated from simple steady-state energy balances based on thermal resistance (U) and surface area (A). The basement calculations are based on an example basement where the house has a square plan 10m X 10m (33ft X 33ft). The basement walls are 1.25m (4 ft) above grade and 1.25m (4 ft) below grade. The ceiling area (Ac) is equal to basement floor area (Af) of 100 m^2 (1060 ft^2). Above grade basement area (Aa) = 50 m^2 (530 ft^2). Below grade basement area (Ab)= 150m^2 (1600 ft^2). For infiltration flows an effective UA is $UA_{infiltration}=24$ for 0.35 ACH. This is the same as assumed for the house in 152P and is the minimum requirement for ASHRAE Standard 62.
The basement temperature is given by a UA weighted average of its surroundings:

$$t_{basement} = \frac{t_{in}UcAc + t_{design}\left(UaAa + UA_{infiltration}\right) + t_{ground}UbAb}{UcAc + UaAa + UbAb} \quad \textbf{(7)}$$

For the following cases, the appropriate values of A and U are used in the above equation. The results have been simplified by converting to more rational fractions and removing small terms.

3.3.1 Uninsulated basement

Uc=3.3W/m^2C (R2) for 1 cm plywood, Ua=Ub=3.6 W/m^2C (R2) for 20 cm (8 inches) of concrete.

$$t_{basement} = \frac{3t_{in} + 5t_{ground} + 2t_{design}}{10} \quad \textbf{(8)}$$

3.3.2 Insulated basement ceiling

Uc=0.43 W/m^2C (approximately RSI 2.5 (R15) insulation)

$$t_{basement} = \frac{3t_{ground} + t_{design}}{4} \quad \textbf{(9)}$$

3.3.3 Insulated basement walls

The basement surface area is separated into walls below grade and the floor. Below grade there is 50 m^2 (530 ft^2) wall with U=0.31 (based on R15 insulation plus the effect of the ground around the foundation) and 100 m^2 (1060 ft^2) of uninsulated floor with U=3.6 W/m^2C (R2) for 20 cm (8 inches) of concrete. The uninsulated ceiling has Uc=3.3W/m^2C (R2) for 1 cm plywood.

$$t_{basement} = \frac{t_{in} + t_{ground}}{2} \quad \textbf{(10)}$$

3.4 Crawlspace temperatures

Crawlspace temperatures are calculated the same way as the basement temperatures, using simple steady-state energy balances. For crawlspaces, a floor plan of 10mX10m (33ftX33ft) is used with a 1m (3.3ft) high crawlspace. The walls of the crawlspace and the floor are made of plywood (approximately RSI 0.3 (R-2)). The U value (0.57 W/m^2°C (0.1 Btu/hft^2°F)) for the dirt floor is from ASHRAE Fundamentals Handbook (1985) p.23.15. This dirt floor U value is for the heat transfer through a concrete floor on the ground, but is used here for convenience for the crawlspace floor. The crawlspace ventilation was assumed to be 1 ACH for an unvented crawlspace and 5 ACH for a vented crawlspace. The vented crawlspace ventilation rate is based on the work of Palmiter and Bond (1994). Note that for the crawlspace the ground temperature for basements was not used because the top surface of the ground for a crawlspace is directly exposed to the house and ambient conditions.

Ua_{dirt}=57 W/°C (30 Btu/h°F), Ua_{floor}=330 W/°C (170 Btu/h°F), $UA_{infiltration}$=28 W/°C (15 Btu/h°F), UA_{walls}=132 W/°C (67 Btu/h°F)

Then:

$$t_{crawlspace} = \frac{t_{in}(330) + t_{out}(57+28+132)}{330+57+28+132} \approx \frac{3t_{in} + 2t_{out}}{5} \quad \textbf{(11a)}$$

At 5 ACH, $UA_{infiltration}$=140 W/°C (73 Btu/h°F)

$$t_{crawlspace} = \frac{t_{in}(330) + t_{out}(57+140+132)}{330+57+140+132} \approx \frac{t_{in} + t_{out}}{2} \quad \textbf{(11b)}$$

With the crawlspace walls and the house floor insulated to RSI 2.5 (R-15):

UA_{floor}=43 W/°C (22 Btu/h°F), UA_{walls}=17 W/°C (9 Btu/h°F)

$$t_{crawlspace} = \frac{t_{in}(43) + t_{out}(57+28+17)}{43+57+28+17} \approx \frac{t_{in} + 3t_{out}}{4} \quad \textbf{(12a)}$$

At 5 ACH, $UA_{infiltration}$=140 W/°C (73 Btu/h°F)

$$t_{crawlspace} = \frac{t_{in}(43) + t_{out}(57+140+17)}{43+57+140+17} \approx \frac{t_{in} + 5t_{out}}{6} \quad \textbf{(12b)}$$

For crawlspaces with uninsulated walls, but the house floor is insulated:

$$t_{crawlspace} = \frac{t_{in}(43) + t_{out}(57+28+132)}{43+57+28+132} \approx \frac{t_{in} + 5t_{out}}{6} \quad \textbf{(13a)}$$

At 5 ACH:

$$t_{crawlspace} = \frac{t_{in}(43) + t_{out}(57+140+132)}{43+57+140+132} \approx \frac{t_{in} + 8t_{out}}{9} \quad \textbf{(13b)}$$

3.5 Manufactured House Belly Pan Temperatures

Tyson et al. (1996) measured five houses with belly pan ducts in Alabama. Example results show that the belly pan temperature is close to indoor conditions. Therefore, for duct calculations it is assumed that the belly pan temperature is the same as indoors. Other computer model studies (Cummings (1996)) have shown that regain is high (averaging 62%) for belly pan ducts. This reinforces the assumption of using indoor temperature for belly pan ducts because a high regain implies good thermal communication between inside and the belly pan compared to outside and the belly pan. There may be substantial variations from this simple assumption due to the range of insulation used in belly pans (approximately R-7 to R-33) and the relative airtightness of the exterior of the belly to the interface between the belly and the house. These effects are not included in the standard for simplicity and because more research needs to be done to support a more complex calculation method.

4. Estimation of Ground Temperatures

An estimate of ground temperature is required for basement and crawlspace temperature calculations. A simple approach is to assume that the ground temperature is the average outdoor air temperature for the year (The complex three dimensional change in losses with depth is beyond the scope of 152P). The depth at which this is true depends on the climate, so the variation of temperature with depth is ignored. The average yearly temperature is not typically known or used by building designers, nor is it readily available. A first order approximation to the average yearly outdoor temperature would be to average the summer and winter design conditions. Table 4 shows the average $t_{winter,2.5\%}$ and $t_{summer,2.5\%}$ (the same design temperatures used in the rest of 152P) for 12 locations in the U.S. The design temperatures are taken from ASHRAE Fundamentals (1993) Chapter 24, Table 1. Average outdoor air temperature data is from NOAA (1980) and from TMY (NCDC 1980) data files. This table shows that averaging $t_{winter,2.5\%}$ and $t_{summer,2.5\%}$ gives reasonable estimates for ground temperature. It gives average outdoor air temperatures close to the averages from the NOAA and TMY data.

5. Default Duct Surface Area Estimation Method

The duct surface areas are those outside conditioned space and include plenum surfaces. The duct surface area estimation method was based on measured field data from 69 systems. Extra details about these systems can be found in Andrews (1996), Jump, Walker and Modera (1996) and Modera (1993). All duct surface areas are based on outside diameter, which includes the insulation thickness. Only the single story houses are used in this analysis. For these houses, all their ducts were exposed. The two story houses had hidden ducts of unknown surface area in walls, chases and floor spaces. Analysis of the two story houses indicated that they had about 30% less exposed duct surface area for the same floor area. In 152P the fraction of exposed duct is a separate input. Therefore this section concentrates on the total duct surface area which was only available for the single story houses. This reduced the data set to 45 of the 69 houses. The first parameter tested was the dependence of duct surface area on floor area. As expected, there was a strong correlation, with larger houses having larger duct systems. To remove the dependence on size of house, all the supply duct surface areas (A_s) are normalized by dividing by the house floor area (A_{floor}). Other parameters that were considered in determining duct surface area were:

1. Age of duct system (not always the same as house age).
2. Duct material type - Sheet metal or Flex duct
3. Equipment (furnace or A/C) location - Central or Outside wall (outside wall includes garages).
4. Register location - either central or perimeter (with respect to floor plan).
5. Number of registers.
6. Duct system topology. This is determined by the ratio (Y) of number of registers to number of connections to the supply plenum. For 0<Y<1.5 the duct system is an "Octopus" (i.e. it has almost as many plenum connections as registers). For 1.5<Y<6 the duct system is a "Tree", in which a few plenum connections split into branches to each register. For Y>6 the duct system is a "Trunk" in which there are only one or two plenum connections by large diameter ducts, which then have many smaller ducts along their length. It was found that counting registers and connections in the above manner is an effective method for characterizing duct topology, even though duct topology does not influence duct surface area.

A subset of 54 houses (it does not include Andrews houses), was used to determine which of these parameters had a significant effect on the duct surface area. These 54 houses include both one and two story houses. The evaluation was done by performing linear regression between the parameter of interest and the duct surface area. For some of the methods that showed little linear correlation, an average value of the duct surface area to floor area ratio was calculated. The standard deviation of the average compared to the average was then used to determine if a simple algebraic average value could be used as a correlation. For system age, equipment location, register location and duct system topology, the regression and averaging results were poor. Note that only supply duct surface areas were examined because the returns are very simple in the duct systems analyzed here. Table 5 summarizes the results.

5.1 Supply Duct Calculations

The parameters for surface area from Table 5 are:

- Floor area.
- Number of registers.
- Duct material type - Sheet metal OR Flex duct

The supplies were analyzed three different ways:

1. The supply duct area depends on floor area, the number of registers, and the duct material.
2. The supply duct area depends on the number of registers and floor area.
3. The supply duct area depends on floor area only.

These four options were rated by the average absolute difference between the measured duct surface area and the predicted duct surface area. This number gives the user an estimate of the average uncertainty for predicting duct surface area for an individual house. In all the analysis procedures, the duct surface area is normalized by the floor area, and it is the dependence of this ratio on the parameters discussed in points 1 to 4 above that will be discussed. All the coefficients, mean differences and absolute mean differences are expressed in percentage points (the same units as the ratio of duct surface area to floor area). In other words, the differences are not percentages of the measured value.

5.1.1 Option 1
Equation 14 was used to determine A_s/A_{floor} as a function of number of registers (N_{supply}) and duct type - flex or sheet metal. The systems were split into two groups:
1. Flex duct
2. Sheet metal.

$$\frac{A_s}{A_{floor}}[\%] = A + B(N_{supply}) \tag{14}$$

where A and B were determined by least squares fitting to each group of measured data, and are given in Table 6. The average absolute error (AE) was calculated using:

$$AE = \frac{\sum_{i=1}^{N} |\text{Measured} - \text{Predicted}|}{N} \tag{15}$$

where N is the number of houses in each group, Measured is the measured area, and Predicted is found using Equation 14.

5.1.2 Option 2
The differentiation between duct types was removed so that the only parameters were floor area and number of registers. The measured data were least squares fitted to a Equation 14, resulting in the coefficients in Table 6.

5.1.3 Option 3
This option just uses the mean measured values. Averaging all the systems (and removing zero size systems for returns), gives the values in Table 6.

5.1.4 Supply Duct Summary
The results in Table 6 indicate that there is little reduction in prediction error by including factors other than floor area alone.

5.2 Return Duct Surface Area
The returns for these houses are much simpler than the supplies. Therefore the number of possible dependent parameters is much smaller. For example, duct system topology is undefined for many return systems because they only have 1 or 2 return ducts. In addition, 20 out of 69 systems had no return ducts. In these houses the return was comprised of an air handler unit that was connected directly to the equipment. Two options are examined below.
In 152P, the default duct areas are total duct areas. The returns for two story houses do not include ducts inside the conditioned space, in walls or chases. To account for this, 152P uses the results for single story houses only for the defaults. The user of the standard must then determine the fraction of this total in the various duct locations, including inside conditioned space. Analysis of the two story systems indicates that about 40% of returns for two story houses are not exposed.

5.2.1. Changing return duct surface area with the number of registers and number of stories.

Equation 16 was used to describe the variation with number of registers and stories:

$$\frac{A_r}{A_{floor}}[\%] = C(N_{return}) \qquad \textbf{(16)}$$

A least squares fit to the measured surface areas gives C=5. The mean absolute error was 2.1. Note that N_{return} is for the duct part of a system only. For furnaces connected directly to a return plenum the registers in the plenum are NOT counted in N_{return}. Therefore, systems having ONLY a plenum (or air handler stand) have N_{return}=0, i.e. NO return surface area. Equation 16 has application limits due to the limited nature of the data set used to develop the correlations. The largest number of returns for single story houses was five. Any single story house with more than 5 returns should use five returns in Equation 15 to determine duct surface area. Using more than 5 results in unrealistic predictions.

6. Building Plan and Default system fan flow

6.1 Building Plan

The manufacturers fan flow rating specified in the building plans is reduced by 15%. This is based on the field measurements made by various committee members.

6.2 Default

The default fan flow is a function of the floor area of the building such that a larger building will have a larger fan flow (corresponding to bigger equipment and higher building loads). From the houses studied by Jump et al. (1996) the measured fan flows averaged 0.64 cfm/ft^2 (11m^3/hour m^2) for air conditioning (AC) and heat pump (HP) systems. The large house to house variation means that no simple correlation with number of stories, duct material, system topology etc. could be found. Therefore, simply using a mean value is sufficient. Note that 0.64 cfm/ft^2 (11m^3/hour m^2) is much lower than the 1 cfm/ft^2 (18m^3/hour m^2) used in many energy calculations. This reflects poor installation practices that restrict the system flows and dirty heat exchangers or filters. This lower value is also supported by data from surveys sponsored by the California Energy Commission (private communication, April 1997) which used equipment manufacturers specifications. These specifications gave an average of 0.77 cfm/ft^2 (14m^3/hour m^2). This is slightly higher than the measured results but was not measured directly and so does not take into account reductions in flow below manufacturers specifications due to poor duct installation or system design.

7. Default Duct Leakage as a Fraction of Fan Flow

The bibliography lists references that discuss leakage flows as a fraction of fan flow. In order to keep 152P calculations simple, and because any correlations of leakage with other house parameters are unclear, it is reasonable to choose a single leakage fraction for both supply and return. Most of the references listed in the bibliography indicate leakage rate of 10% to 20% of fan flow, therefore it is reasonable to choose the value of 17% found by Jump et al. (1996).

8. Cyclic Losses

The impact of duct thermal mass on the energy delivered at the registers is included in the building load factor used to calculate distribution efficiency. This impact depends on the materials used for the ducts. Results from simulations of an attic-only central plenum duct system in Sacramento, CA for the month of January (Modera and Treidler (1995)) have been used to estimate the cyclic losses. Due to the limited range of duct system parameters exercised in the simulations, the results are not sufficient to allow for a complex cyclic loss factor to be estimated. For simplicity, the correction is determined only as a function of duct material. The correction to the delivery effectiveness is 2% for non-metallic (plastic flex duct or duct board) ducts and 5% for sheet metal.

9. Duct and equipment interactions

9.1 Changes in equipment performance with reduced fan flows.

Reductions in equipment efficiency associated with duct-induced reductions in flow across the heat exchanger are accounted for by the flow factor in 152P. The flow factor is a simple reduction in equipment performance to encourage proper design procedures. An 8% reduction in equipment efficiency was found by Rodriguez et al. (1995) for flow reduction of 15% (as used as the building plan default in 152P). This reduction of 8% in equipment performance was for orifice control systems. Rodriguez et al. found that there was little equipment performance change for TXV systems or furnaces when fan flows were reduced. Therefore, for cooling F_{flow}=1.0 if designed according to ACCA (1997) and F_{flow}=0.92 for orifice control without duct system layout or design calculations. For heating or TXV controlled cooling F_{flow}=1.0. This correction factor does not include the effects of other potential flow reducing devices, such as electronic air filters, because it is assumed that the HVAC system designer will account for this when sizing the system fan.

9.2 Variable Capacity Equipment

9.2.1 Delivery Effectiveness, DE

For design calculations it is assumed that the equipment is properly sized (e.g., using ACCA manuals), and at design conditions the equipment should be operating in its high capacity mode. Therefore, DE is calculated using the high capacity and matching system fan flow for design conditions.

For seasonal conditions, system operation is partly at high capacity and partly at low capacity. The fraction of time spent in high and low capacity modes to meet seasonal loads was based on manufacturers specifications and HPSF and SEER calculation methods. The DE was calculated for seasonal temperature (and humidity) conditions for both high (DE_{hi}) and low (DE_{lo}) capacity modes. The seasonal $\overline{DE}$ is then a weighted average of the high and low capacity results. The weighting is given by the fraction of time (T) in each mode:

$$\overline{DE} = DE_{hi}\frac{T_{hi}}{T_{total}} + DE_{lo}\frac{T_{lo}}{T_{total}} \quad \textbf{(17)}$$

For example, for a typical air conditioner:

$$\overline{DE} = DE_{hi}\,0.18 + DE_{lo}\,0.82 \tag{18}$$

Equation 18 indicates that DE_{lo} dominates for seasonal calculations. The following examples show that just using DE_{lo}, rather than the weighted average suggested by Equation 17, is acceptable.

For poor ducts: DE_{hi}=0.47, DE_{lo}=0.40. $\overline{DE}$ =0.41

For typical ducts: DE_{hi}=0.63, DE_{lo}=0.56. $\overline{DE}$ =0.57

For good ducts: DE_{hi}=0.82, DE_{lo}=0.79. $\overline{DE}$ =0.795

9.2.2 Distribution System Efficiency, η_{dist}

In addition to the effect on DE shown above, an additional factor is required that accounts for reduction of AFUE, SEER or HPSF equipment efficiency for seasonal calculations. At design conditions the load factor is calculated for high capacity (and associated fan flow) only. For seasonal conditions, however, the load factor must be calculated for both high and low capacities and have the same weighting method as for DE in Equation 17. An equipment factor, F_{vc}, is used to account for poorer duct systems making the equipment operate for a longer time in high capacity mode. The calculations below show how F_{vc} is estimated.

9.2.2.1 Derivation and example calculations for variable capacity equipment efficiency derating factor F_{vc}

Equation 19 shows how the equipment efficiency, η_{equip}, is determined from the high and low capacity efficiencies, η_{hi} and η_{lo}, and the fraction of time the equipment operates at high capacity, T_{hi}/T_{total}.

$$\eta_{equip} = \eta_{lo} + \frac{T_{hi}}{T_{total}}\left(\eta_{hi} - \eta_{lo}\right) \tag{19}$$

The duct losses change the ratio of T_{hi} to T_{total} to be $\left.\frac{T_{hi}}{T_{total}}\right|_{duct}$.

F_{vc} is defined as the ratio of equipment efficiency with a duct system, $\eta_{equip,duct}$, to that without (i.e., the rated equipment efficiency):

$$F_{vc} = \frac{\eta_{equip,duct}}{\eta_{equip}} = \frac{\eta_{lo} + \left.\frac{T_{hi}}{T_{total}}\right|_{duct}\left(\eta_{hi} - \eta_{lo}\right)}{\eta_{lo} + \frac{T_{hi}}{T_{total}}\left(\eta_{hi} - \eta_{lo}\right)} \tag{20}$$

The ratio of T_{hi} to T_{total} changes with distribution system efficiency. The distribution of the number of hours at a given building load are typically unknown. For simplicity, it is assumed that the number of hours at each load is the same. This assumption makes the problem linear, as shown in Equation 21.

$$\left.\frac{T_{hi}}{T_{total}}\right|_{duct} = 1 - DE\left(1 - \frac{T_{hi}}{T_{total}}\right) \quad \textbf{(21)}$$

F_{vc} can then be written in terms of T_{hi}/T_{total} at which the equipment was rated:

$$F_{vc} = \frac{1 + \left(1 - DE\left(1 - \frac{T_{hi}}{T_{total}}\right)\right)\left(\frac{\eta_{hi}}{\eta_{lo}} - 1\right)}{1 + \frac{T_{hi}}{T_{total}}\left(\frac{\eta_{hi}}{\eta_{lo}} - 1\right)} \quad \textbf{(22)}$$

Calculation F_{vc} requires: the ratio of high to low capacity equipment efficiencies, the ratio of T_{hi} to T_{total} at which the equipment efficiency is specified, and the DE calculated in 152P. The effect of F_{vc} increases with larger differences between high and low efficiency and poorer duct systems.

For furnaces T_{hi}/T_{total} is typically 0.05 and η_{hi}/η_{lo}=0.90. Using these values, Equation 22 can be simplified to:

$$F_{vc} = 0.905 + 0.095DE \quad \textbf{(23)}$$

For a good duct system, with DE = 0.90 this results in F_{vc} = 0.990. For a poor duct system with DE = 0.60 then F_{vc} = 0.96. These results shows that the furnaces are not very sensitive to poor duct systems that increase the operating time at high capacity.

For AC systems the SEER rating procedure can be used to estimate the change in equipment efficiency to account for the added load of duct losses. From available manufacturer's data, the ratio of high capacity (E_{hi}) to low capacity (E_{lo}) is about 2. The SEER rating method uses binned temperature data for climate zone 4 in rating equipment. The number of hours in each 5°F (2.5°C) bin from the 65°F (18°C) base to 105°F (41°C) is specified. Assuming E_{hi} meets the maximum load at 105°F (41°C) (a temperature difference of 40°F (22°C)), then E_{lo} meets the load at a temperature difference of 20°F (11°C) or an outside temperature of 85°F (30°C). The number of hours at low capacity is then found by adding up all the hours in the SEER distribution below 85°F (30°C). From the SEER calculation procedure:

$$\frac{T_{lo}}{T_{total}} = (65-70)\text{bin} + (70-75)\text{bin} + (75-80)\text{bin} + (80-85)\text{bin}$$

$$\frac{T_{lo}}{T_{total}} = 0.214 + 0.231 + 0.216 + 0.161 = 0.822$$

Therefore $\frac{T_{hi}}{T_{total}} \cong 0.18$.

The ratio of high to low capacity efficiencies (EER) can be found from manufacturers data. Analyzing manufacturers output and consumption data to determine EER for high and low capacity gives typical $EER_{hi}/EER_{lo} \cong 0.8$. Substituting these values into Equation 22 we get a simplified version for use with AC equipment:

$$F_{vc} = 0.83 + 0.17DE \quad \textbf{(24)}$$

This approximation to Equation 22 is very good (within one percentage point for F_{vc}).

For heat pumps, the HPSF tests also use a bin temperature method. Because this calculation is for heating, the bins and the distribution are different from the AC calculations above. Using the same approximate factor of two between high and low capacities (for cooling), and assuming that the heat pump will exactly meet the maximum load at maximum temperature difference. The maximum temperature difference for standard (zone 4) conditions is 65°F-(-10°F) = 75°F (42°C). At half load (i.e. all low capacity operation) the temperature difference would be 75/2=37.5°F (21°C). The difference between 65°F (18°C) and half load conditions is 27.5°F (15°C). Adding up the fraction of time in each bin up to 27.5°F (the nearest bin is 25-30°F) from HPSF rating calculations:

$$\frac{T_{lo}}{T_{total}} = (65-60)\text{bin} + (60-55)\text{bin} + (55-50)\text{bin} + (50-45)\text{bin}$$
$$+ (45-40)\text{bin} + (40-35)\text{bin} + (35-30)\text{bin} + (30-25)\text{bin}$$

$$\frac{T_{lo}}{T_{total}} = 0.132 + 0.111 + 0.103 + 0.093 + 0.100 + 0.109 + 0.126 + 0.087 = 0.861$$

Therefore $\frac{T_{hi}}{T_{total}} \cong 0.14$.

Using the same EER_{hi} and EER_{lo} ratio as for AC, we can substitute this value of T_{hi}/T_{total} into Equation 24 to get a simplified version:

$$F_{vc} = 0.82 + 0.18DE \quad \textbf{(25)}$$

Given how close the AC and HP results are (Equations 24 and 25) the same relationship can be used for these pieces of equipment.

For heat pumps using strip heat in high capacity mode, the EER_{hi} to EER_{lo} ratio can be estimated by assuming that EER_{hi} =1.0 (by definition for electric resistance heat), and by assuming EER_{lo} = 2.5. Substituting these values into Equation 22 gives:

$$F_{vc} = 0.44 + 0.56DE \quad \textbf{(26)}$$

This shows how the use of strip heat has a significant impact on heat pump performance. Duct losses (reflected by reduced DE) force the system to operate with the strip heat on (with associated low EER) and results in much reduced heat pump performance.

9.3 Summary of duct and equipment interactions

Table 7 summarizes the duct and equipment interactions. In Table 7, hi cap and lo cap refer to the use of high or low equipment capacity and the corresponding fan flow rates in 152P calculations. Note that fan flow effects must still be combined with these variable capacity effects to get the total equipment factors for 152P.

10. Thermal Regain

The reduction in building load due to regain of duct losses by means of reduced duct zone temperature differentials is based upon the relative thermal resistances of the buffer-space/conditioned-space interface and the total thermal resistance of the buffer-space. The thermal regains are calculated from the following relationship:

$$F_{regain} = \frac{UA_c}{UA_{total}} \quad \textbf{(27)}$$

where UA_c is the UA value for the interface between the conditioned space and the buffer space, and UA_{total} is the total UA value for the buffer space. These UA values include thermal conduction across the interface and air infiltration.

10.1 Attics

Duct losses will change the temperature of the surroundings in such a way that duct losses may be reduced due to the ambient air being closer to the duct temperature. This was included in the thermal regain effect by including the UA of the duct system in the regain calculations. This UA should include both conduction and leakage effects, but for simplicity only conduction losses are included in these example calculations. In addition, this analysis will look at supply conduction only because this is the dominant source of conduction losses. The magnitude of the change in attic temperatures due to duct losses can be seen in the houses tested by Jump et al. (1996). The effect is most clearly seen in heating mode with a 3°C (5°F) increase in attic-outside temperature difference after the ducts were retrofitted.

As a first estimate of the thermal regain factor including reduced duct losses, the UA of the ducts is combined with the UA of the ceiling. The duct surface area is assumed to be 27% of floor area (152P default for a single story) with either R4 or R6 insulation.

The following example calculation shows how the attic regain factors were estimated. The attic has plywood sheathing approximately R2 (RSI 0.3) and a 1:1 pitch, 10m X 10m (33ft X 33ft) floor area, R30 (RSI 5) ceiling gives an equivalent U value of 0.2. The UA values are:

$UA_{ceiling}$=100 x 0.2=20 W/k (10 Btu/h°F)

UA_{duct}=ceiling area x 0.27% x RSI 1=100 x 0.27 x 1 = 27 W/k (14 Btu/h°F)

$UA_{atticexterior}$=640 W/k (330 Btu/h°F)

$UA_{infiltration}$ = Cp_{air} x Attic Volume x ACH/3600 = 1000 x 250 x ACH/3600

At 1 ACH, $UA_{infiltration}$ = 69 W/k (36 Btu/h°F). It is assumed that all the infiltration is to outside.

$$F_{regain} = \frac{UA_{ceiling} + UA_{duct}}{UA_{ceiling} + UA_{duct} + UA_{atticexterior} + UA_{infiltration}} = \frac{20+27}{20+27+640+69} = 0.06$$

(28)

This calculation was repeated for a range of ceiling and duct insulation and attic ventilation rates and the results are summarized in Table 8. The results in Table 8 show that a regain factor of 0.10 is typical (in the middle of the range) and so this will be used as F_{regain} for the attic in 152P.

10.2 Garages

The example calculation garage is 2.5m (8ft) high with a floor plan of 7m X 5m (23ft X 16ft) (double garage) with one wall attached to the house (5m X 2.5m (16ft X 8ft) , insulated to R15 (RSI 2.5)) and the other walls are plywood (R2 (RSI 0.3)). Assuming 1 ACH ventilation rate gives F_{regain}=0.02. With R4 (RSI 0.6) walls, F_{regain} increases to 0.03. These calculations for garages are dominated by the large uninsulated surface are of the garage, and the fact that the wall between the garage and the house is insulated.
If the house to garage wall is not insulated and all the garage walls are R4 (RSI 0.3), then F_{regain} becomes 0.11. Given this wide range of F_{regain} (0.02 to 0.11) and the small magnitudes, a middle value of F_{regain}=0.05 is used in 152P.

10.3 Crawlspaces

As for the attic, the crawlspace walls are assumed to be plywood that is approximately R-2 (RSI 0.3). The floor of the house is also assumed to be of plywood of the same thickness. The crawlspace walls are 1m (3.3 ft) high. The U value (0.57 W/m^2°C (0.1 Btu/hft^2°F)) for the dirt floor is the same as for crawlspace temperature calculations in Section 3.4. Table 9 summarizes the calculated crawlspace regain factors for a range of insulation locations and crawlspace infiltration rates. Adding duct losses to crawlspace regain did not significantly change the regain factors (unlike for attics).

10.4 In/Under slab ducts

For this calculation it was assumed that the area above the slab to the house is the same as the area below the ducts to the ground. With the same U value for the dirt under the slab as for the crawlspace, and using U=4.7 W/m^{2}C (0.8 Btu/hft^2°F) for 15 cm (six inches) of concrete above the ducts: **F_{regain} =0.83**
With insulation under the slab and the ducts in the slab (i.e. on the house side of the insulation): **F_{regain} =0.90**

10.5 Basements

This example calculation uses the same U value for the floor as for crawlspaces. Based on comments in ASHRAE Handbook of Fundamentals, the U value for the basement walls below grade is doubled. Above grade, the basement walls are 20 cm (eight inches) of concrete with U=0.72 W/m^{2}C (0.13 Btu/hft^2°F) . The floor of the house is uninsulated plywood and the basement walls are above and half are below grade. Table 10 gives F_{regain} for a range of insulation locations and basement ventilation rates.

10.6 Exterior walls

It is assumed that the ducts are located such that the thermal resistance to outside is the same as the thermal resistance to inside and $\mathbf{F_{regain}=0.5}$.

10.7 Belly Pans in manufactured houses:

Computer modeling (Cummings (1996)) showed regain averaging 62% for five Florida houses, plus two additional houses in North Carolina. This is a similar result to the above regain for crawlspaces with uninsulated floors. For simplicity in this standard it is assumed that the belly pan location is the same as a crawlspace.

10.8 Summary of default thermal regain factors

Default thermal regain factors are summarized in Table 11. This is the table used in 152P.

11 Summary

This paper has shown how the default values for forced air systems in proposed ASHRAE standard 152P were determined. These defaults were based on field measurements, simple heat transfer analyses, modeling, and analysis of weather data. An approximate ranking of the importance of these parameters can be estimated from the sensitivity of the distribution system efficiency calculated using the standard to each parameter. The following list is in approximately decreasing order of importance:

1. local climate
2. system location
3. duct leakage
4. system fan flow
5. duct surface area
6. thermal regain
7. interactions with equipment (includes variable capacity effects)
8. cyclic losses

Acknowledgements

This work was supported by the Assistant Secretary for Conservation and Renewable Energy of the U.S. Department of Energy, Office of Building Technologies, Existing Buildings Research Program, under Contract No. DE-AC03-76SF00098.

The research reported here was funded in part by the California Institute for Energy Efficiency (CIEE), a research unit of the University of California. Publication of research results does not imply CIEE endorsement of or agreement with these findings, nor that of any CIEE sponsor.

The author would like to thank the following organizations for providing source data and data analysis: the Alberta Home Heating Research Facility at the University of Alberta in Canada, Florida Solar Energy Center, and the Small Homes Council – Building Research Council at the University of Illinois.

References

ACCA. 1997. ACCA Manual D - Residential Duct Systems. Air Conditioning Contractors of America. Washington, D.C.

Andrews, J. 1996. Field Comparison of Design and Diagnostic Pathways for Duct Efficiency Evaluation. Proc.1996 ACEEE Summer Study, Vol. 1. pp. 21–31.

ASHRAE. 1993. 1993 ASHRAE Handbook - Fundamentals, 1993. Atlanta. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1989. *ASHRAE Standard 62-1989 Ventilation for Acceptable Indoor Air Quality*. Atlanta. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1997. *ASHRAE Standard 152P (Proposed) 1997 Method of Test For Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems*. Atlanta. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Cummings, J., 1996. Computer model results for manufactured house belly pan temperatures. Private Communication. August 1996.

Forest, T.W., and Walker, I.S. 1991. Field Study on Attic Ventilation and Moisture. Proc. CANCAM '91, University of Manitoba, Winnipeg, Canada.

Forest, T.W., and Walker, I.S. 1992. Attic Ventilation Model. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings V, pp. 399-408.

Forest, T.W., and Walker, I.S. 1993. Attic Ventilation and Moisture. Canada Mortgage and Housing Corporation report.

Jump, D., Walker, I. and Modera, M. 1996. Field measurements of Efficiency and Duct Retrofit Effectiveness in residential Forced Air Distribution Systems. Proc.1996 ACEEE Summer Study. Vol. 1. pp. 147-157.

Levins, W.P. and Karnitz, M.A. 1987. Cooling Energy measurements of Single-Family Houses with Attics Containing Radiant Barriers in Combination With R-11 and R-30 Ceiling Insulation. ORNL/CON-226. Oak Ridge National Laboratory report.

Modera, M. P. 1993. Characterizing the performance of Residential Air Distribution Systems. Energy and Buildings. Vol. 20. pp. 65-75.

NCDC. 1980. Climatography of U.S., #81, Monthly Normals of Temperature, Precipitation and Heating and Cooling Degree days 1951-1980. National Climatic Data Center, Federal Building, Asheville, NC.

NOAA (National Oceanic and Atmospheric Administration - U.S. Dept. of Commerce). 1980. Local Climatological Data - annual summaries for 1980. Environmental data and information service. National Climatic Center. Asheville, N.C.

Palmiter, L. and Bond, T. 1994. Modeled and Measured Infiltration PhaseII, A Detailed case Study of three Homes. EPRI Report TR-102511. Electric Power Research Institute. Palo Alto, CA.

Parker, D.S., Sherwin, J.R., and Gu, L., 1997. Monitored Peak Attic Air Temperatures in Florida Residences: Analysis in Support of ASHRAE Standard 152P. FSEC-CR-944-97. Florida Solar Energy Center, Cocoa, Fl.

Parker, D., 1991. Monitored Energy Use Characteristics of a Florida Residence. FSEC-RR-23-91. Florida Solar Energy Center, Cocoa, FL.

Proctor, J., 1997. Measured Attic Temperatures in Desert Conditions. Presentation at ASHRAE 152P committee meeting. January 1997.

Rodriguez, A.G., O'Neal, D.L., Bain, J.A., and Davis, M.A. 1995. The Effect of Refrigerant Charge, Duct Leakage, and Evaporator Air Flow on the High Temperature Performance of Air Conditioners and heat Pumps. Energy Systems Laboratory, Dept. of Mechanical Engineering, Texas Engineering Experiment Station, Texas A&M University.

Rose, W.B. 1992. Measured Values of Temperature and Sheathing Moisture Content in Residential Attic Assemblies. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings V, Clearwater Beach, Florida, December 1992. pp. 379-390.

Rose, W.B. 1997. Measured Attic Temperatures. Personal communication.

Tyson, J.,Gu, L., Anello, M., Swami, M.V., Chandra, S. and Moyer, M., 1996. Manufactured Housing Air Distribution Systems; Field Data and Analysis for the Southeast. FSEC-CR-875-96. Florida Solar Energy Center. Cocoa, Fl.

Walker, I.S., and Forest, T.W. 1995. Field Measurements of Ventilation Rates in Attics., Building and Environment, Vol. 30. No.3. pp. 333-347. Pergamon Press.

Walker, I.S., 1993. Attic Ventilation, Heat and Moisture Transfer. Ph.D. dissertation. University of Alberta. Edmonton. Alberta. Canada.

Attic radiant barrier and reduced emissivity coating bibliography

Abrantes, V. 1985. Thermal Exchanges Through Ventilated Attics. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings III. Clearwater Beach. Florida. December 1985. pp. 296-308.

Carlson, J.D., Christian, J.E., and Smith, T.L. 1992. In Situ Thermal Performance of APP-Modified Bitumen Roof Membranes Coated with reflective Coatings. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings V, Clearwater Beach. Florida. December 1992. pp. 420-428.

Chen, H., Larson, T., and Erickson, R.W. 1992. The Performance of an Attic Radiant Barrier for a Simulated Minnesota Winter. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings V. Clearwater Beach. Florida. December 1992. pp. 114-122.

Griggs, E.I, and Shipp, P.H. 1988. The Impact of Surface Reflectance on the Thermal Performance of roofs: An Experimental Study. ORNL/TM-10699. Oak Ridge National Laboratory. Oak Ridge, TN.

Levins, W.P. and Karnitz, M.A. 1986. Cooling Energy measurements of Unoccupied Single-Family Houses with Attics Containing Radiant Barriers. ORNL/CON-200. Oak Ridge National Laboratory. Oak Ridge, TN.

Levins, W.P. and Karnitz, M.A. 1987. Cooling Energy measurements of Single-Family Houses with Attics Containing Radiant Barriers in Combination With R-11 and R-30 Ceiling Insulation. ORNL/CON-226. Oak Ridge National Laboratory. Oak Ridge, TN.

Levins, W.P. and Karnitz, M.A. 1988. Heating Energy measurements of Single-Family Houses with Attics Containing Radiant Barriers in Combination With R-11 and R-30 Ceiling Insulation. ORNL/CON-239. Oak Ridge National Laboratory. Oak Ridge, TN.

Levins, W.P., Karnitz, M.A., and Hall, J.A. 1989. Moisture Measurements in Single-Family Houses with Attics Containing Radiant Barriers. ORNL/CON-255. Oak Ridge National Laboratory. Oak Ridge, TN.

McDonald, J.M., Courville, G.E., Griggs, E.I., and Sharp, T.R. 1989. A Guide for Estimating Potential Energy Savings from Increased Solar Reflectance of a Low-Sloped Roof. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings IV. Orlando. Florida. December 1989. pp. 348-357.

Ober, D.G. 1989. Full-Scale Radiant Barrier Tests in Ocala, Florida: Summer 1988. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings IV. Orlando. Florida. December 1989. pp. 286-293.

Parker, D.S., Sherwin, J.R., and Gu, L. 1997. Monitored Peak Attic Air Temperatures in Florida Residences: Analysis in Support of ASHRAE Standard 152P. FSEC-CR-944-97, Florida Solar Energy Center, Cocoa, Fl.

Parker, D.S. 1997. Seasonal Attic temperatures. personal communication.

Rose, W.B., 1992. Measured Values of Temperature and Sheathing Moisture Content in Residential Attic Assemblies. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings V. Clearwater Beach. Florida. December 1992. pp. 379-390.

Wilkes, K.E. 1989. Thermal Modeling of Residential Attics with Radiant Barriers: Comparison with Laboratory and Field Data. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings IV. Orlando. Florida. December 1989. pp. 294-311.

Wu, H. 1989. The Effect of Various Attic Venting Devices on the Performance of Radiant Barrier Systems in Hot Arid Climates. Proc. ASHRAE/DOE/BTECC/CIBSE Thermal Performance of the Exterior Envelopes of Buildings IV. Orlando. Florida. December 1989. pp. 261-270.

Duct Leakage bibliography

Cummings, J.B., Tooley, J., and Dunsmore, R. 1990. Impacts of Duct Leakage on Infiltration Rates, Space Conditioning Energy Use, and Peak Electrical Demand in Florida Homes. Proc. 1990 ACEEE Summer Study. Pacific Grove. Ca.. August 1990. American Council for an Energy Efficient Economy, 1001 Connecticut Ave. N.W., Suite 335, Washington, D.C. 20036.

Downey, T., and Proctor, J. 1994. Blower Door Guided Weatherization Test Project, Final Report. Proctor Engineering Group Report for Southern California Edison Customer Assistance Program.

Jump, D.A., Walker, I.S. and Modera, M.P. 1996. Field Measurements of Efficiency and Duct Retrofit Effectiveness in Residential Forced air Distribution Systems. Proc. 1996 ACEEE Summer Study. Vol. 1. pp. 147-155.

Modera, M.P., and Wilcox, B. 1995. Treatment of Residential Duct Leakage in Title-24 Energy Efficiency Standards. CEC contract report. California Energy Commission.

Modera, M.P. and Jump, D.A. 1995. Field Measurement of the Interactions Between Heat Pumps and Attic Duct Systems in Residential Buildings. Proc. ASME International Solar Energy Conference. March 1995.

Table 1. Estimating differences between design and seasonal temperatures using 2.5% design values

Location	**Winter Dry Bulb, °C [°F]**			**Summer dry bulb, °C [°F]**		
	Design (97.5%)	TMY season avg.	Difference	Design (2.5%)	TMY season avg.	Difference
Los Angeles, CA.	4 [39]	12 [54]	8 [14]	32 [90]	21.7 [71]	10.3 [19]
Atlanta, GA.	-6 [21]	4 [39]	10 [18]	33 [91]	24.4 [76]	8.6 [15]
New York, NY	-9 [16]	0.5 [33]	9.5 [17]	32 [90]	23.4 [74]	8.6 [15]
Average Differences			9 [16]			9 [16]

Table 2. Summary of Attic to Ambient Temperature Differences, °C [°F]

	Source 1		Source 2		Source 3	
	Well vented	Poorly vented	Well vented	Poorly vented	Well vented	Poorly vented
Cooling Design	12 [22]	20 [36]	12 [22]	15 [27]	9 [16]	20 [36]
Cooling Seasonal	3 [6]	6 [10]	12 [22]	17 [31]	7 [13]	9 [16]
Heating Design	-	-	5 [9]	7 [13]	6 [11]	7 [13]
Heating Seasonal	-	-	0 [0]	1 [2]	2 [4]	2 [4]

Table 3. Difference between outside and garage Temperatures, °C [°F]

	Heating			Cooling		
	Atlanta	New York	Los Angeles	Atlanta	New York	Los Angeles
Design	10 [18]	9 [16]	9 [16]	2 [4]	3 [5]	4 [7]
Seasonal	7 [13]	7 [13]	6 [11]	5 [9]	4 [7]	5 [9]

Table 4. Using average design conditions to estimate ground temperatures, °C [°F]

Location	$t_{winter,2.5\%}$	$t_{summer,\ 2.5\%}$	$\frac{t_{winter,2.5\%} + t_{summer,2.5\%}}{2}$	Average outdoor air temperature	
				NOAA	TMY
Fairbanks, Alaska	-44 [-47]	26 [15]	-9 [16]	-3 [27]	-3.5 [26]
Phoenix, Arizona	1 [34]	42 [108]	21.5 [71]	21 [70]	22 [72]
Oakland, California	2 [36]	27 [81]	14.5 [58]	14 [57]	
Athens, Georgia	-6 [21]	33 [91]	13.5 [56]	16 [61]	
Boise, Idaho	-12 [10]	34 [93]	11 [52]	11 [52]	11 [52]
Chicago, Illinois	-17 [1]	33 [91]	8 [46]	9 [48]	10 [50]
New Orleans, Louisiana	1 [34]	33 [91]	17 [63]	20.5 [69]	20 [68]
St. Louis, Missouri	-14 [7]	34 [93]	10 [50]	13 [55]	13 [55]
NY,NY	-9 [16]	31 [88]	11 [52]	12 [54]	12 [54]
Bismark, N. Dakota	-28 [-18]	33 [91]	2.5 [37]	5 [41]	5 [41]
Dallas, Texas	-6 [21]	38 [100]	16 [61]	19 [66]	
Seattle, WA.	-3 [27]	28 [82]	12.5 [55]	10 [50]	10 [50]

Table 5. Duct surface area significant parameters

Parameter	Surface area dependence
Age of duct system	NO
Equipment location	NO
Register location	NO
Duct system topology	NO
Duct material type	YES - Flex duct systems have 50% more normalized surface area than sheet metal.
Number of registers	YES - Normalized duct area increases with increasing number of registers
Floor Area	YES - bigger houses have bigger systems

Table 6. Supply duct surface area coefficients

Option	Duct Type	**A**	**B**	Average Absolute Error, AE	Number of houses in group
1	Flex	**19**	**1.3**	9	18
1	Sheet metal	**9**	**1.6**	7	27
2	-	**13**	**1.5**	8.4	45
3	-	**27**	**-**	8.4	69

Table 7. Variable Capacity Equipment Effect in ASHRAE 152P

152P parameter	Furnaces	AC	HP	HP in strip heat mode
DE design	hi cap	hi cap	hi cap	Hi cap
DE seasonal	low cap	low cap	low cap	Low cap
η_{dist} design	hi cap	hi cap	hi cap	Hi cap
η_{dist} seasonal	$F_{VC} = 0.905 + 0.095DE$	$F_{VC} = 0.82 + 0.18DE$	$F_{VC} = 0.82 + 0.18DE$	$F_{VC} = 0.44 + 0.56DE$

Table 8. Attic thermal regain factors

F_{regain}	Infiltration (ACH)	Ceiling Insulation	Duct Insulation
0.06	1	R-30	R6
0.05	5	R-30	R6
0.04	10	R-30	R6
0.08	1	R-30	R4
0.06	5	R-30	R4
0.05	10	R-30	R4
0.11	1	R-15	R4
0.08	5	R-15	R4
0.06	10	R-15	R4
0.16*	1	R-15	R4
0.11*	5	R-15	R4
0.07*	10	R-15	R4

* - for the smaller attic with 1:5 pitched roof

Table 9. Crawlspace regain factors

F_{regain}	Infiltration rate (ACH)	Insulation
0.63	0.1	None
0.60	1	None
0.5	5	None
0.41	10	None
0.36	0.1	R-15 house floor and crawlspace walls
0.30	1	R-15 house floor and crawlspace walls
0.17	5	R-15 house floor and crawlspace walls
0.11	10	R-15 house floor and crawlspace walls
0.18	0.1	R-15 house floor
0.16	1	R-15 house floor
0.12	5	R-15 house floor
0.08	10	R-15 house floor

Table 10. Regain Factors for basements

$\mathbf{F_{regain}}$	Ventilation Rate [ACH]	Insulation
0.55	0	Uninsulated
0.51	0.35	Uninsulated
0.78	0	R15 insulated basement walls
0.74	0.35	R15 insulated basement walls
0.32	0	R15 basement walls and house floor
0.27	0.35	R15 basement walls and house floor

Table 11. 152P Thermal Regain Factors

Location	Thermal Regain Factor [F_{regain}]
Attic	0.10
Garage	0.05
Crawlspace, Unvented, Uninsulated	0.60
Crawlspace, Unvented, Insulated Building Floor and crawlspace walls	0.30
Crawlspace, Unvented, Insulated Floor only	0.16
Crawlspace, Vented, Uninsulated	0.50
Crawlspace, Vented, Insulated Building Floor and crawlspace walls	0.17
Crawlspace, Vented, Insulated Floor only	0.12
Under Slab	0.90
Uninsulated Basement	0.50
Insulated-Ceiling Basement	0.30
Insulated-Wall Basement	0.75
Exterior Walls	0.5