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Informal Report

EFFECT OF AIRFLOW AND HEAT INPUT RATES ON DUCT EFFICIENCY

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

Reducing the airflow and heat input rates of a furnace that is connected to a duct system in thermal contact with unconditioned spaces can significantly reduce thermal distribution efficiency. This is a straightforward theoretical calculation based on the increased residence time of the air in the duct at the lower flow rate, which results in greater conduction losses. Experimental tests in an instrumented residential-size duct system have confirmed this prediction. Results are compared with the heat-loss algorithm in ASHRAE Standard 152P. The paper concludes with a discussion of possible remedies for this loss of efficiency in existing systems and optimal design strategies in new construction.

INTRODUCTION

Recent theoretical work on interactions between heating/cooling equipment and duct systems has drawn attention to what could be a significant energy penalty associated with the application of modulating furnaces to duct systems with conductive heat losses to the outside. The concern can be expressed most simply in terms of residence time. If, say, a modulating furnace is capable of operating at 50% of the nominal heat output and airflow rate for which the duct system was designed, then when the furnace is operating in this low-capacity mode, air from the furnace will require twice as long to be delivered to the registers as it would under the high-capacity mode. This will result in significantly greater heat losses at the low-flow condition unless steps are taken to mitigate these losses.

Much research on duct energy losses has been done over the past two decades. Three quite different methods have been used to quantify the effect of thermal distribution on system efficiency. The most common method has been to measure the individual factors that contribute to energy losses in ductwork (Cummings and Tooley 1989; Gammage et al. 1984; Matthews et al. 1990; Modera 1989; Parker 1989; Palmiter and Bond 1992; Proctor 1998; Proctor and Pernick 1992; Robison and Lambert 1989; Treidler and Modera 1996). A second method has been to compare, on a seasonal basis, the heating energy consumption of two large samples of homes, where one sample is heated with electric furnaces and duct systems and the other sample uses electric baseboard heating (Lambert and Robison 1989). The third method is electric co-heating. The co-heating system measures the "bare" heating load of a house, which is then compared with the energy used by the ducted heating system (Andrews et al. 1996; Francisco and Palmiter 2000; Olson et al. 1993; Palmiter and Francisco 1994; Strunk 2000). All these methods have led to the same general conclusion—that duct systems in unconditioned spaces typically lose 25% to 40% of the energy output from the space-conditioning equipment, with leakage and conductive losses contributing comparable amounts.

As an initial benchmark consistent with these findings, let us assume that 15% of the input heat is lost via conduction under full-capacity operation. This means that 85% of the heat is retained. Then, under the half-capacity operating mode of a modulating furnace, one would expect that this rate of loss to occur twice, so that ~ $(1 - 0.85^2)$ X 100 or 28% of the furnace heat output would be lost in the ducts, an increase in these losses of 13 percentage points.

The energy losses from duct leakage, on the other hand, should not change very much under furnace modulation. If the airflow rate is reduced by 50%, the pressures in the ducts will drop to ~ 25% of their values when the equipment is operating at full capacity, since the airflow scales approximately as the square root of the driving pressure. Duct leakage generally relates to pressure with an exponent that may be somewhat greater than 0.5, with 0.6 being taken as a standard value in most analyses, including ASHRAE Standard 152P. This means that the leakage rate would reduce to ~43% of its value under full-capacity operating conditions. Assuming that in its low-capacity mode the furnace must operate twice as long to satisfy the load, then the energy impact of the leakage would be twice this 43% or ~86% as great as under full airflow.

Assuming, again as a benchmark, that 15% of the input heat is lost via leakage under full-capacity operation, this means that 0.86 X 15% or ~13% of the input heat would be lost via leakage under the half-capacity condition, an improvement of ~2 percentage points. (It should be noted that not all researchers agree that the difference in exponents between the airflow within a duct and the duct leakage flows is significant. Thus, this analysis of the leakage impact represents a best-case scenario for the modulating furnace.) The result is that, under conditions that at all approximate what are typically found, the negative impact on conduction losses will greatly outweigh any positive impact on leakage losses.

Although the above reasoning is straightforward, a judgment was made that it needed to be confirmed not only by more detailed analyses but also by testing under controlled laboratory conditions. An expected side benefit of this testing was to determine how accurately the algorithm used in ASHRAE Standard 152P to calculate conductive losses would predict the measured values. This paper reports on such testing that has recently been carried out at a national laboratory.

DUCT LOSSES IN ASHRAE STANDARD 152P

A new standard method of test for thermal distribution efficiency, ASHRAE Standard 152P (ASHRAE 2001a), has been developed. It addresses both heating and cooling with ducts and in some types of non-ducted systems. The approach used in the standard is first to calculate the Delivery Effectiveness (DE), which is the ratio of the heat or cooling delivered by the duct system to that put into the ducts by the equipment. Various system interaction effects are then considered, e.g., effective regain of lost heat caused by its warming the space surrounding the ducts, impacts on air infiltration to the house caused by unbalanced duct leakage, and impacts of duct characteristics on equipment efficiency. These result in a final figure of merit, the Distribution Efficiency (η_{dist}), which may be higher or lower than DE.

This paper considers forced-air heating. For this application, DE is given by the following equation in Standard 152P:

$$DE = a_s B_s - a_s B_s (1 - a_r B_r) \Delta T_r / \Delta T_e - a_s (1 - B_s) \Delta T_s / \Delta T_e \qquad (1)$$

where a_s and a_r are the ratios of airflow at the register(s) to airflow at the system fan on the supply and return sides, respectively; B_s and B_r are factors that account for conductive losses on the supply and return sides; ΔT_s and ΔT_r are the temperature differences between the conditioned space and the "buffer zones" (e.g., attic, crawl space, basement) surrounding the supply and return ducts; and ΔT_e is the temperature rise in the air as it flows through the heating equipment. The a's are obtained from measurements of duct leakage and system fan flow. The B's are calculated from duct surface area and insulation levels according to the relation

$$B_x = \exp[-A_x/(Q_e C_{vx} R_x)] \qquad (2)$$

where (with x = s or r for supply or return ducts) A_x is the duct surface area, R_x is the thermal resistance of the duct, Q_e is the volume airflow rate at the system fan, and C_{vx} is the volume specific heat of the air in the duct. The ΔT's are obtained from system fan flow, climate data, characteristics of the buffer zones, and equipment specifications.

In the context of Standard 152P, it is important to note that the effect under investigation here, which is an impact of equipment characteristics on the efficiency of the duct system, must be carefully distinguished from the opposite type of impact, that of duct design on equipment efficiency. The former effect, which is studied in this paper, arises directly from Equation 1 (and a companion equation for cooling) without the need for any other factors. The latter effect, which occurs primarily with variable-capacity air conditioners and heat pumps, is embodied in an equipment efficiency factor, F_{equip}. For equipment with a single operating mode, F_{equip} is set equal to 1.0 in Standard 152P, even though there are some effects that, if accounted for, would yield slightly different values for this parameter (Gu et al. 2003). For variable-capacity equipment, a dependence of equipment efficiency on duct characteristics is embodied in the seasonal distribution efficiencies reported by Standard 152P, but a study that supported the development of Standard 152P states that "furnaces are not very sensitive to poor duct systems that increase the operating time at high capacity" (Walker 1998). Again, the effect discussed here does not depend on F_{equip}, and is predicted by the Standard 152P algorithm even when F_{equip} is set equal to a constant 1.0.

THEORETICAL PREDICTIONS AND SIMULATION RESULTS

In order to obtain better benchmark predictions of the efficiency impact of furnace modulation than the "back of envelope" calculations in the Introduction, a parametric study was undertaken for a sample case, using ASHRAE Standard 152P. The version of the standard resident on a national laboratory web site (www.ducts.lbl.gov) was used.

The example case was a 2000 ft^2 (186 m^2) house located in Pittsburgh, PA. The ducts were sized in conformity with typical values of surface area found in Walker 1998. Assuming two return registers, this gave 540 ft^2 (50 m^2) of supply duct and 200 ft^2 (19 m^2) of return duct. The duct material was specified as having low thermal mass (duct board or flexible duct), which meant that Standard 152P assessed a 2% efficiency penalty across the board for off-cycle losses. A base-case heating plant was selected as a gas furnace with a 70 000 Btu/h (20.5 kW) output rate and a system fan flow rate of 1200 cfm (0.57 m^3/s). This is consistent with a 54 °F (30 K) temperature rise in the furnace. A half-capacity mode was simulated by cutting both the heat output rate and the airflow rate by 50%.

Two sets of cases were run. In the first set, the ducts were perfectly insulated (R=10000 IP) but leaked. Three leakage levels were considered:

- Zero leakage
- Low leakage (5% of system fan flow in the supply and return systems at base-case conditions).
- "Typical" leakage (17% of system fan flow, supply and return, base case.)

One factor had to be calculated outside of the Standard 152P formalism, namely the variation of duct leakage with system fan flow rate. As discussed above, the system fan flow rate is expected to vary approximately as the 0.5 power of the driving pressure caused by the fan. The pressures in the duct are proportional to the fan pressure. Duct leakage is generally assumed to vary at a slightly higher power of the pressures in the duct; this exponent is taken as 0.6 in Standard 152P. This implies that the duct leakage will vary as the 0.6/0.5 or 1.2 power of the system fan flow rate. Thus, high flow rates produce slightly higher fractional leakage rates than lower flow rates. We would expect, then, to find the distribution efficiency to be slightly greater for lower flow rates in a duct system with leakage but no thermal transmission losses. Again, however, there is not a complete consensus that the assumed difference between the two exponents is real; the calculated leakage benefit from furnace modulation may be illusory.

Table 1 shows the seasonal distribution efficiencies for this first set of cases. Thermal regain was assumed to be negligible (as it is for ducts in vented attics or vented crawl spaces with insulated ceilings). Under these conditions, the distribution efficiency was nearly identical to the delivery effectiveness.

It can be seen that furnace output has at most a small effect on distribution efficiency, assuming that the system fan flow rate is varied in proportion. For the highlighted cases, representing "typical" leakage and 50% modulation with no thermal regain, the gain in distribution efficiency is 2 percentage points, in line with the value obtained in the Introduction.

In the second set of cases, the ducts were assumed to have zero leakage, but the insulation was varied. Three insulation levels were considered:

- "Perfect" insulation, approximated by an R-value of 10,000.
- "High" insulation, specified as R-8.
- "Typical" insulation, specified as R-4.

(IP units for thermal resistance R-values [ft^2-h-°F/Btu] are used throughout this paper. To convert to SI units [K-m^2/W], multiply by 0.176.)

Table 2 shows the seasonal distribution efficiencies for the cases considered in this set of runs, assuming zero thermal regain. Note that an insulation of R-8 is midway in heat-retention value between R-4 and infinity, since the heat-loss rate is proportional to the inverse of the R-value.

Table 1. Seasonal Distribution Efficiencies for Perfectly Insulated Ducts with Varying Furnace Outputs, Fan Flow Rates, and Duct Leakage Rates. Thermal Regan Factor = 0.0

Furnace Output (Btu/h [kW]) System Fan Flow (cfm [m^3/s]) Duct Leakage at Base Case	70 000 [20.5] 1 200 [0.57] (Base Case)	35 000 [10.2] 600 [0.28] (50% Output)	17 500 [5.1] 300 [0.14] (25% Output)
No Leakage	0.98	0.98	0.97
5% of Fan Flow @ 1200 cfm	0.91	0.92	0.92
17% of Fan Flow @ 1200 cfm	**0.76**	**0.78**	0.80

Table 2. Seasonal Distribution Efficiencies for Leak-Free Ducts with Varying Furnace Outputs, Fan Flow Rates, and Insulation Values. Thermal Regan Factor = 0.0

Furnace Output (Btu/h [kW]) System Fan Flow (cfm [m^3/s]) Duct Insulation Level	70 000 [20.5] 1 200 [0.57] (Base Case)	35 000 [10.2] 600 [0.28] (50% Output)	17 500 [5.1] 300 [0.14] (25% Output)
"Perfect" Insulation (R-10000)	0.98	0.98	0.97
"High" Insulation (R-8)	0.90	0.83	0.69
"Typical" Insulation (R-4)	**0.83**	**0.69**	0.46

For ducts with less-than-perfect insulation, the impact of the furnace capacity and system fan flow rate is very significant. For the highlighted case with R-4 insulation and 50% modulation with no thermal regain, the loss in distribution efficiency is 14 percentage points, in line with the value obtained in the Introduction.

More-detailed simulations were carried out by Walker (2001), using an hour-by-hour simulation model. For six climate locations ranging from Miami to Minneapolis, he found reductions in heating distribution efficiency ranging from 10 to 12 percentage points resulting from 50% furnace modulation when the ducts were located in a vented attic. An older study (Andrews and Krajewski 1985) using a simpler model obtained results consistent with these.

Although modulating furnaces do not necessarily reduce heat input and airflow rate in exact proportion, the assumption of proportional reduction was used as a reasonable benchmark, from which some departure may be seen in practice. Later, the suggestion was made that increasing the airflow in the low-capacity mode might be beneficial. The amount of variation from proportionality is limited by the need to provide comfortable delivered-air temperatures in both the high- and low-capacity modes. Still, many modulating furnaces do not reduce airflow as much as they do heat output when they switch to low capacity. Because the Web-based calculator used in the above study was temporarily unavailable as the final draft of this paper was being written, the predicted impact on delivery effectiveness of raising the system airflow in the low-capacity mode from 600 cfm [0.28 m^3/s] to 900 cfm [0.42 m^3/s] was studied using Equation 1 directly. It was found that for the typical R-4 insulation value there was some benefit, but the effect was not large, averaging 0.7 percentage points for the four possible leakage combinations involving 5% or 17% of system fan flow on the supply and return sides. This did not include the effect on air infiltration when the supply and return leakage rates are unequal (unbalanced leakage), which would usually be a negative impact.

LABORATORY TEST FACILITY

The tests were performed in a thermal distribution test facility (Figure 1) comprising an instrumented residential-size duct system located within a high-bay work space that can be opened to the outside. In addition to the work reported here, this facility was also used to assess the accuracy of duct leakage testing (Andrews 2002). The supply duct system consists of a main trunk duct splitting into two branch ducts, with four runouts emerging from each branch. All the ducts are of sheet-metal construction, the trunk and branch ducts being rectangular and the runouts round. Most of the supply ducts are located on a mezzanine halfway between the floor and the ceiling of the high bay. The single return duct is of rectangular sheet metal. The duct surface area is 395 ft^2 [36.7 m^2] on the supply side and 127 ft^2 [11.8 m^2] on the return side. The supply-duct area equals the default value in ASHRAE Standard 152P for a house with 1460 ft^2 [130 m^2] of heated floor area.

The fan speed is continuously variable, with total airflow ranging from near zero to ~1500 cfm [0.7 m^3/s]. A True-Flow™ flow plate was left in the system at the return plenum. This allowed easy measurement of airflow to ± 2% (Palmiter and Francisco 2000) instead of ± 7% if the flow plate is inserted only when measuring airflow. The added pressure drop caused by the flow plate was treated as if it were caused by an elbow in the return duct system. The system flow rate was calculated from the flow-plate reading, corrected for temperature and pressure variations, plus any leaks into the air handler between the flow plate and the system fan.

Heat was supplied to the duct system by four electric heaters, each of nominal 5 kW capacity. These can be activated manually in any desired combination. The heating load is simulated by an 8 ft X 4 ft X 8 ft high [2.4 m X 1.2 m X 2.4 m high] enclosure (the "register box"), roughly the size of a walk-in cooler. The supply registers are in the ceiling near one end of the box, while the return register is near ground level, at the other end. The register box was configured to act as a "heat dump" via continuous removal of heated air coming from the supply registers and substitution of an equivalent amount of cooler outdoor air.

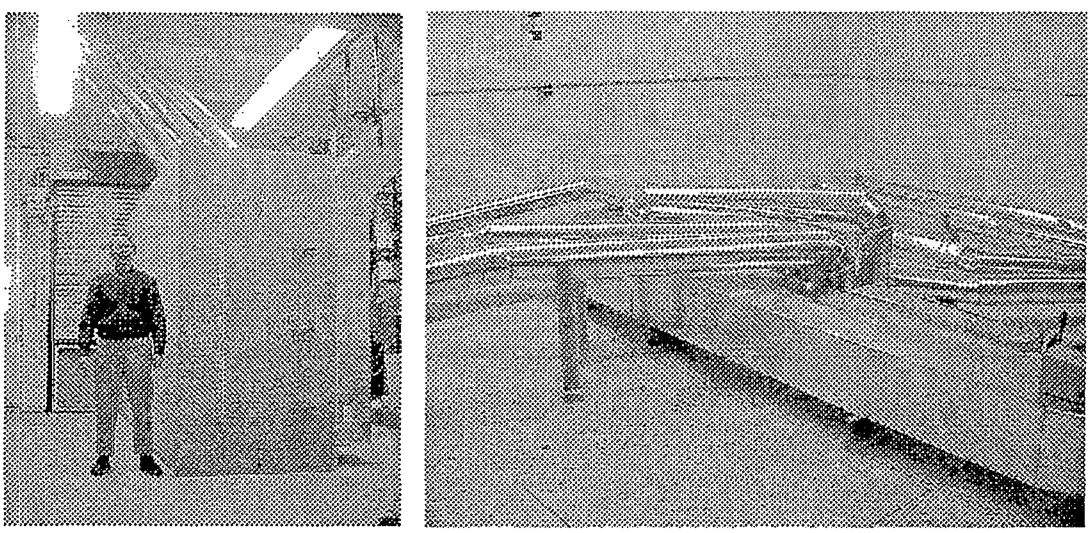

Figure 1. Test Facility. Left: Register Box. Right: Ducts on Mezzanine.

Within the high-bay enclosure, most of the supply duct system, the fan, the heaters, and the control station are located on a mezzanine, while the register box, the last several feet of each supply runout, and the return intake are on the ground floor. Because of the large thermal mass of the high-bay structure, the air temperature within it, which serves as the "buffer-zone temperature" seen by the ducts, varies slowly. During a two-hour run this temperature typically changed by ~5 °F [3 K] when the ducts were not insulated and ~2 °F [1 K] after insulation was added.

Air leakage from the duct system was reduced to near zero by sealing all seams with mastic. Measured leakage using the duct airtightness test in Standard 152P was ~2.5% of system fan flow on the supply side and ~1% of fan flow on the return side.

In addition to the flow plate, instrumentation consisted of the following. Pressures were measured with static-pressure probes linked to a digital manometer. Temperatures at the inlet and outlet of each duct section, as well as points within the room and outside, were measured using thermocouples linked to a data acquisition system. The inlet temperature at the heaters was measured by an averaging grid of nine thermocouples. Input power to the resistance heaters was measured at a particular driving voltage (~465 V) by means of a calibrated power meter. This was then corrected for the small variations in line voltage from one day to the next using Ohm's law.

MEASUREMENT OF DELIVERY EFFECTIVENESS

It was decided to perform the experiments in such a way that only the supply-side conductive heat losses needed to be accounted for. This was done by taking the indoor temperature to be the inlet temperature at the air handler and the temperature difference between the indoors and the buffer space to be the difference between that temperature and the average air temperature near the ducts. The main reason for this choice was that the supply ducts, being both larger in surface area and having a much greater temperature difference between the interior and the surrounding space, would in any case be responsible for most of the losses. Return losses in this system were generally too small to measure reliably with the instrumentation currently in place. The study was further focused on conductive losses as opposed to air leakage, since this was the main area of concern as far as the efficiency impact of furnace modulation was concerned.

It was desired to measure DE values over a range of temperature differences between the conditioned space, i.e., the air temperature at the air-handler/heater inlet, and the buffer space containing the ducts, i.e., the average temperature inside the high bay in the vicinity of the ducts. This temperature was allowed to float under the influence of outdoor conditions as modified by internal heat gains and the large thermal mass of the enclosing high-bay structure. During most of the tests it was between 50 and 70 °F [10 and 21 °C]. The temperature of the air entering the air handler was manipulated by varying the amount of heat dumped from the register box. The temperature difference between the air-handler inlet and the space surrounding the ducts was varied over a range of ~50 °F [30 K] for each of the six test conditions defined by choosing one of three insulation levels (no insulation, one layer of R-4 duct wrap, or two layers of R-4 duct wrap) and either of two operating modes (full capacity, i.e., ~20 kW heat input and ~1200 cfm [0.6 m^3/s] airflow, or half capacity, i.e., ~10 kW heat input and ~600 cfm [0.3 m^3/s] airflow).

As stated above, the definition of DE is heat delivered at the registers divided by heat input to the duct system. This can be expressed as:

$$DE = Q_e C_v (T_{reg} - T_{in}) / H_{heaters} \quad (3)$$

where Q_e is the airflow rate in ft^3/h [m^3/s], C_v is the volume specific heat in Btu/ft^3-°F [J/m^3-K], T_{reg} is the average temperature at the registers in °F [°C], T_{in} is the inlet temperature at the air handler, and $H_{heaters}$ is the heat input rate of the heaters, in Btu/h [W]. (The average in T_{reg} should ideally be flow-weighted, but because the register flows were nearly equal, a simple average was used.)

An alternative method could have been used, based on the temperature rise of the air passing through the heaters. That would be represented by the formula

$$DE = (T_{reg} - T_{in}) / (T_{heaters} - T_{in}) \quad (4)$$

where $T_{heaters}$ would be the flow-weighted average air temperature leaving the heaters. A comparison of Equations 3 and 4 shows that their relative accuracy is governed by the ability to measure airflow and heater input versus the ability to measure the temperature of the air leaving the heaters. It was found that both the flow and temperature distributions of this air were very uneven, despite efforts to smooth them, and the use of Equation 4 was abandoned in favor of Equation 3.

Each data point was based on a two-hour run in the test facility. An attempt was made to keep conditions as nearly steady-state as possible for the duration of each run. As a precaution, the first twelve minutes of data were eliminated for each run, and the remainder of temperature samples, taken at one-minute intervals, were divided into 12-minute segments. A correction was made for any variation in duct temperature from one of these segments to the next, in terms of heat deposited into or extracted from the thermal mass of the duct. These corrections were generally small.

EXPERIMENTAL UNCERTAINTIES

Estimation of expected experimental uncertainties was based on analysis of Equation 3. Assuming the errors in the measured variables to be independent of one another, the fractional uncertainty in DE can be estimated by quadrature addition of the fractional uncertainties in the airflow rate, the input power, and the temperature difference between the air entering the heaters and the air leaving the supply registers. The uncertainty in airflow rate was taken to be ~2% on the basis of testing performed by the developers of the flow-plate device, while that in the heaters, whose power was measured by trained technicians using a calibrated meter, was estimated at less than 1%. Each of the temperatures, T_{reg} and T_{in}, was an average of 8 or 9 thermocouple measurements. Taking the uncertainty of each individual measurement to be ~1 °F [0.6 K], this produces an estimated uncertainty in each average of ~0.3 °F [0.2 K] and in their difference of ~0.5 °F [0.3 K]. The error bars shown on the data points reflect these estimates.

RESULTS

The salient results of the experiment are shown in Figures 2 – 4. DE is plotted against the temperature difference between the buffer zone and the indoors. As one would expect, DE decreases in a nearly linear fashion as this temperature difference increases. In each of these figures, the data points indicate measured DE values, with the squares representing full-capacity operation and the circles half-capacity data.

The lines represent the DE predictions of ASHRAE Standard 152P as represented by Equation 1. Solid lines represent the full-capacity case and dashed lines the half-capacity case. The prediction of Standard 152P depends, of course, on the thermal resistance of the duct insulation, as indicated by separate lines for different R-values in these figures. In this experiment, Equation 1 simplified considerably, since there is essentially no supply leakage and return losses are not included in the calculation:

$$DE = B_s - (1 - B_s)\,\Delta T_s / \Delta T_e \qquad (5)$$

Figure 2 shows the results for uninsulated ducts. Generally, an R-value of 1 (in IP units) is used for uninsulated ductwork. These data appear to indicate that a somewhat higher R-value may be warranted. Moreover, the effective R-value appears to be slightly higher at the low-capacity condition. There, R = 1.5 gave the best fit to the standard, while at full capacity, R = 1.2 worked best. (Lauvray [1978] measured R-values of ~1.6 for 10-inch [0.25 m] square sheet-metal ducts at airspeeds in the range used here.) The effect of this is to reduce somewhat the negative impact of modulation on DE, relative to the prediction of Standard 152P at a constant R-value. Nevertheless, the reduction in DE is still severe, ranging from 15 to 20 percentage points.

Figure 3 shows the case of nominal R-4 insulation. This consisted of fiberglass duct wrap with an outer layer of reflective foil. Here, the Standard 152P predictions were upheld almost exactly for the half-capacity case, using the nominal thermal resistance value. At full capacity, however, a slightly smaller R-value, around 3.5, gives a better fit, and there appears to be some deviation in the slope of the line as well. One might expect a somewhat lower R-value to hold at the higher flow rate, caused by a greater "scouring effect" on the interior duct surface, raising the inside film coefficient of heat transfer. Typically, a reduction in DE of ~10 percentage points was experienced when switching from full-capacity to half-capacity operation, with the loss in efficiency being greater, on both an absolute and relative basis, for higher temperature differences between the indoors and the buffer zone.

Finally, Figure 4 shows results for the case where two layers of R-4 duct wrap were used. Here again, there is a significant reduction in DE when operation is changed from the full-capacity to the half-capacity mode. Typically, this reduction was again ~10 percentage points.

A surprising result, at least to the author, was that the apparent R-value of the added insulation did not come up to expectations. Instead of approximating an R-value close to the nominal 8, the data agreed with Standard 152P with R equal to ~5. Taken literally, this would mean that adding insulation to a duct system would not be as effective in reducing conductive heat losses as might be supposed simply by applying the nominal R-values.

An obvious first place to look for a cause of this phenomenon is the well-known decreasing effectiveness of additional insulation surrounding cylindrical tubes. A calculation was performed that compared the effective thermal resistance of a given thickness of insulation on a flat surface with the same thickness surrounding a tube. An expression for the outward rate of heat flow per unit length of a cylinder (Carlslaw and Jaeger 1973) was used to calculate a ratio of the effective thermal resistance based on the inner surface area of the cylinder to the thermal resistance of the same insulation thickness as a flat layer (obtained by letting the radius of the cylinder go to infinity). This ratio was then used to adjust the effective R-value of insulation around the cylinder from the nominal to the actual value. Using this procedure, it was found that for a 7-inch [0.18 m] diameter tube (the size of the runout ducts in this experiment) an insulating material that is one inch thick and nominally R-4 (IP units) would have an effective R-value of 3.4. Similarly, a two-inch layer of the same material would have an effective R-value of 6.2, i.e., an incremental R-value for the second inch of 2.8. Thus, it would appear that this explains part of the reduced effectiveness of insulation in this experiment.

A second possible source of reduced insulation value might lie in a relatively subtle squeezing of the insulation layer as it was put on, despite attempts to avoid doing this. It should be noted that squeezing the insulation can have two bad effects: it reduces the thickness of the material and it may also increase its thermal conductivity. The thickness reduction affects the R-value linearly. The effect on conductivity is less straightforward. Most types of insulation have a minimum in the curve of thermal conductivity vs. density (ASHRAE 2001b), and we do not know what the effect of a density change would be on the insulation we used. However that might be, if the material was squeezed by 10%, i.e., if the effective thickness of the nominal R-8 insulation ended up being 1.8 inches rather than the nominal 2 inches, and if the thermal conductivity was increased by 10% as a result of this squeezing, these effects in combination with the cylindrical effect considered in the previous paragraph would reduce the effective R-value of the nominal R-8 insulation to 5.1. This is close to the value actually observed. Although we do not know if such squeezing occurred despite our attempts to avoid it, it may explain some of the residual difference between our observed thermal resistance values for the double layer of duct wrap and the calculated value after accounting for the increased surface area as discussed above.

DISCUSSION

The main result of this work was to confirm the theoretical prediction that modulating the airflow and heat input rates would reduce the efficiency of the duct system. The observed reductions were somewhat less than what ASHRAE Standard 152P would predict under the assumption of constant duct thermal resistance. A secondary result was that the benefit of adding insulation to the duct, particularly in going from R-4 to R-8, was much less than would be predicted on the basis of the nominal R-values alone. Part of this reduced R-value is explained by the expanding surface area of each incremental layer of insulation added to a cylinder. The rest of it may have been caused, at least in part, by a modest squeezing of the insulation when the additional layer was added, despite precautions taken to avoid doing this.

The observed reductions in DE in going from full- to half-capacity operation are sufficiently large that efforts to mitigate the impacts are strongly indicated. Possible approaches can be briefly described for new construction and for existing systems.

In new construction, the best solution is to place the ductwork in the conditioned space. Then there will be no conductive losses under either operating mode, and the problem is essentially solved. Moreover, by eliminating duct losses, the ducts can be made smaller in cross section than otherwise would be necessary. That in turn will make it easier to "hide" the ductwork in an aesthetically acceptable fashion. It is important to note, however, that ducts which appear to be in the conditioned space may nevertheless have one or more hidden airflow paths to the outside. It is important to test ducts for outside air leakage even when it is hard to imagine how there could be any.

If this solution is not available for any reason and it is nevertheless decided to use a modulating furnace, every effort should be made to minimize the thermal losses from the ducts through some combination of the following actions:

- Minimize the total length of ducts outside the conditioned space, for example by placing the registers against interior walls.
- Use low-loss fittings to enable the duct cross section to be minimized, consistent with design pressure-drop requirements.
- Avoid any design conservatism that could lead to the ducts being oversized.
- Locate the ducts in a semi-conditioned space such as a basement.
- Superinsulate the ducts. This may work in some cases, e.g., burying attic ducts in a thick layer of loose-fill insulation. However, simply specifying additional layers of duct wrap may be significantly less effective than might be predicted using the nominal R-values stamped on the insulating material.
- If it is expected that the system will be operating in the high-capacity mode a relatively small fraction of the time, consider sizing the ducts so that airspeeds in the runouts under high-capacity operation will be at the "maximum" values in ACCA Manual D (ACCA 1995). This will minimize duct surface area and residence time of the air within the ducts and yet keep airspeeds well within the "recommended" values under low-capacity operation, i.e., most of the time.

Advanced concepts for mitigating the problem involve the use of dampers or control bladders to block off some of the ducts or decrease their cross section during low-capacity operation, thereby decreasing the residence time of air in the ducts. Commercial feasibility of these ideas remains to be demonstrated.

The problem in existing systems is less amenable to solution within the constraint of retaining the duct system that is already in place. The obvious approach of adding insulation to the outside of the ducts, in addition to being less than nominally effective as discussed above, is likely to be costly in terms of both materials and labor. One new idea is to add insulation internally to the runout ducts via some process that would have to be developed. The motivation here is that internal insulation is more effective, on a per-inch basis, than external insulation, and the runout ducts are often more difficult to retrofit externally than the trunk ducts, being both less accessible and requiring greater "technician crawl" per unit area of duct surface. Internal insulation would reduce the residence time of the air in the runouts, though it would also increase the system-wide pressure drop. That might be acceptable if high-capacity operation only occurs for a relatively small fraction of the time, or if the existing duct system is oversized to begin with.

SUMMARY

This paper can be summarized as follows. Reducing the heat input and airflow rates in a forced-air thermal distribution system, as will occur when a modulating furnace operates in its low-capacity mode, can seriously detract from the ability of the duct system to deliver heat efficiently to the living space, if (as is usually the case) the duct design is such that conductive heat losses occur. The problem can be avoided in new construction by placing the ducts within the conditioned space. In existing systems, retrofit of modulating furnaces in systems with ducts outside the conditioned space is probably not advisable unless steps are taken to reduce the conductive heat losses from the in-place ducts.

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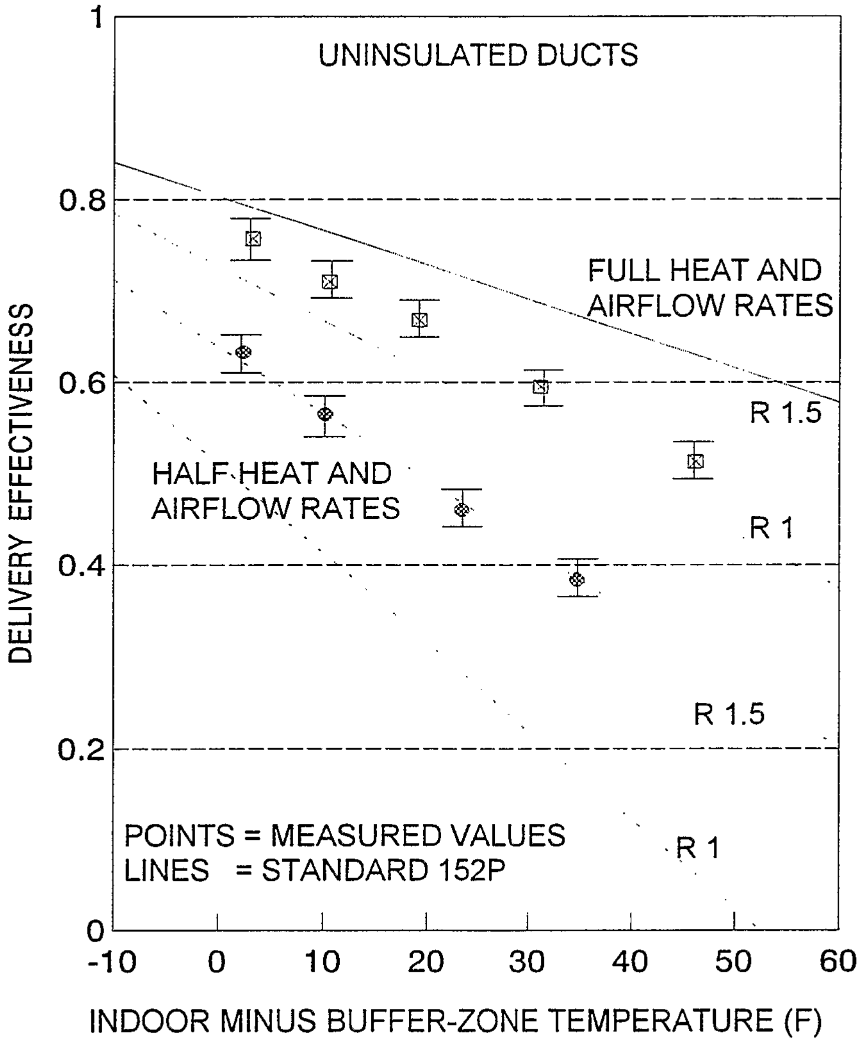

Figure 2. Delivery Effectiveness for Uninsulated Ducts

(Note: Insulation R-values are in IP units. To convert to SI, multiply by 0.176. The range of the horizontal axis in SI units is –6 K to +33 K.)

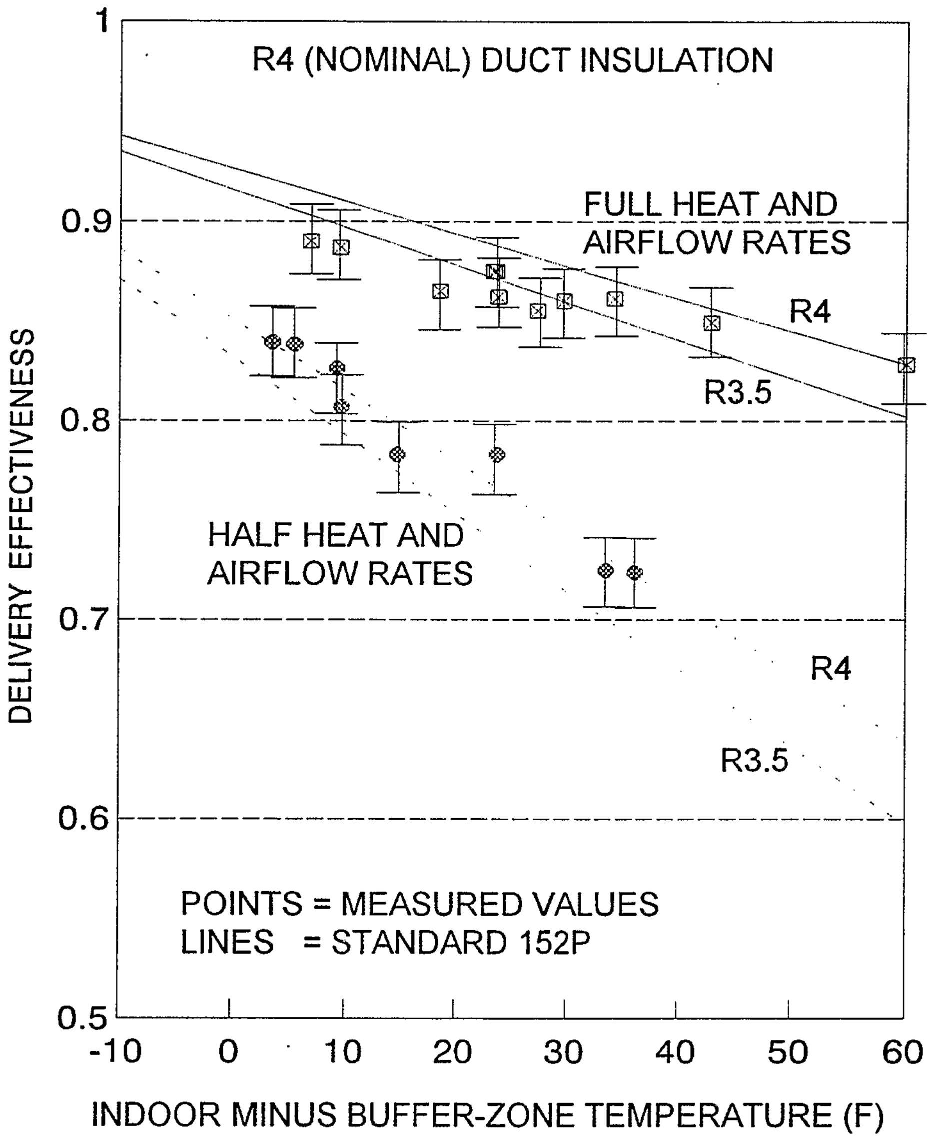

Figure 3. Delivery Effectiveness for R4 (Nominal) Duct Insulation

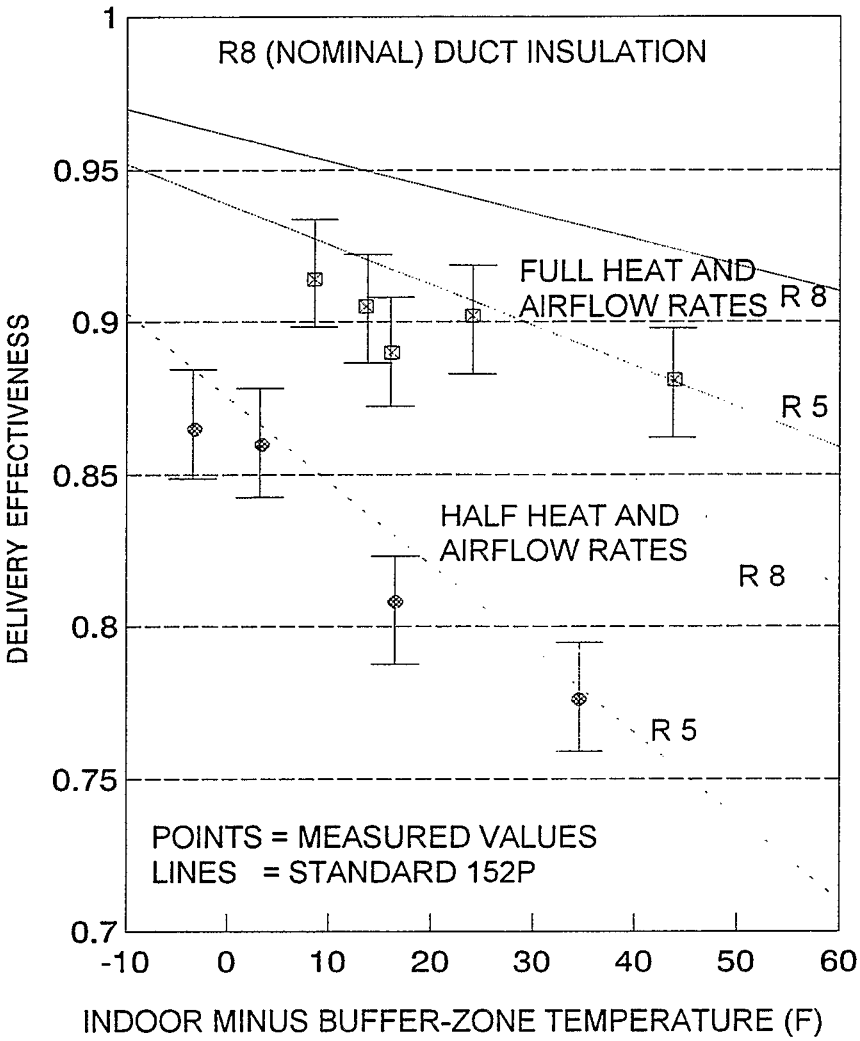

Figure 4. Delivery Effectiveness for R8 (Nominal) Duct Insulation