Measured and Modeled Duct Efficiency in Manufactured Homes: Insights for Standard 152P

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ABSTRACT

Modeling of delivery efficiency was performed using three levels of combining measured and default input parameters and compared to measured data from seven manufactured homes. Using values based on all measured data provided modeled efficiency results that were closest to short-term coheat efficiency results. As individual measured parameters were replaced by estimated or default values suggested by the draft version of proposed ASHRAE Standard 152P, the agreement with measured efficiency results worsened. The primary parameters that were varied were the leakage to outdoors and the temperatures in the buffer spaces in which the ducts were located. All of the models gave results that were, on average, within 8 percentage points of measured results. However, simple modifications to the way in which the estimated or default values are obtained would improve the model predictions. It should be noted that many potential default values were not used and that the comparisons were for a very simple type of house, as manufactured homes do not have return duct systems. Additional suggestions are made on ways to improve the determination of input parameters for modeling paths suggested by Standard 152P that were not compared to measured results in this paper.

INTRODUCTION

There has recently been an increasing effort to develop mathematical models for predicting the thermal efficiency of forced-air distribution systems in residences. Such a model needs to take into account losses due to both air leakage and conduction, as well as the complex interaction between these two types of losses. Two models are currently in use, one by the authors (Palmiter and Francisco 1997) and the one being used in the current draft version of ASHRAE Standard 152P (ASHRAE 1996), as implemented in the spreadsheet distributed with the draft.

Both of these models are designed to predict two different measures of duct efficiency. The first, and more simple, of these is known as the delivery efficiency. This is defined as the fraction of heat produced by the equipment that is delivered through the registers. The second measure of efficiency is known as the distribution efficiency, which takes into account conditioning energy that crosses the building envelope via paths other than through the registers, such as leaks that enter through holes in the building or by conduction. System efficiency also takes into account the change in house load that results from a change in the whole-house infiltration rate due to unbalanced leakage on the supply side compared to the return side.

However, several of the important model inputs are difficult to measure accurately using current technology and techniques, such as the flow through the air handler. Other parameters can be measured fairly accurately but not easily, such as temperatures in buffer zones like attics and crawl spaces. As a result, various measurement techniques and/or default assumptions can be used to obtain values for many of the model input parameters. This paper provides insights on potential problems and suggests improvements to some of these techniques and assumptions.

For simplicity, the modeling in this paper is restricted to the case of delivery efficiency. Both the Palmiter/Francisco model and the Standard 152 model have the same basic equation for this measure of efficiency, so this restriction limits comparisons to the methods by which the input parameters are obtained. When extended to distribution efficiency, the models have different functional forms, so the results would be different even if all of the same input values were used.

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Modeling is done with three different combinations of measured and default parameters, and the results are compared to measured results from seven manufactured homes tested in late winter 1996 in Eugene, Oregon (Siegel et al. 1997). These homes were not selected randomly and should not be considered as representative of homes in general, either manufactured or site-built. These homes were selected to have large duct leakage to the outdoors (in this case 250 cubic feet per minute [cfm] at 50 Pa duct pressurization) and were taken from a list of homes that were signed up for the utility's retrofit program. Of those houses that met the screening criteria, the final selection was made based on when the homeowners could leave their homes for the testing.

TEST HOME DESCRIPTIONS

The homes tested in the field study are older manufactured homes built from the 1970s through 1986. Table 1 gives pertinent house characteristics, and Table 2 shows characteristics of the ductwork. There were actually eight homes included in the study; however, due to a failure of the data collection system, some of the measurements pertinent to this paper are unavailable for Site 1. The tables, therefore, are restricted to the remaining seven homes.

The homes all have crawl spaces that are surrounded by wood or metal skirting. A layer of insulation (typically about R-7) called a "belly blanket" runs under the floor joists and above the crawl space, creating a buffer zone in which main trunk ducts, duct runouts, and plumbing are located. The belly blanket serves as the primary thermal barrier for floor heat loss. The main trunk ducts in these homes are made of light gauge sheet metal and are normally uninsulated. Homes with multiple sections have one or more crossover ducts, which are usually insulated to about R-4 or R-8 and are located in the crawl space. These crossover ducts take air from the section in

TABLE 1
Test Home Characteristics

Site	Year Built	# Sections	Floor Area		Volume	
			ft^2	m^2	ft^3	m^3
2	1973	2	1015	94	6913	196
3	1979	2	1347	125	10200	289
4	unknown	1	990	92	7840	222
5	1974	2	1387	129	10399	294
6	unknown	2	1703	158	13044	369
7	1986	1	831	77	6397	181
8	1982	2	1383	128	10128	287
Avg.			1237	115	9274	267

TABLE 2
Test Home Duct Characteristics

Site	Surface Area		Effective R-value				Location[1]
			$(ft^2 \cdot h \cdot °F)$ /Btu		$(m^2 \cdot °C)$ /W		
	ft^2	m^2	Pre	Post	Pre	Post	% in Belly Space
2[2]	270.3	25.1	1.69	1.76	0.30	0.31	84
3	313.1	29.1	1.60	1.60	0.28	0.28	90
4	156.3	14.5	14.56	14.56	2.56	2.56	100
5	313.2	29.1	1.54	1.63	0.27	0.29	90
6	389.7	36.2	3.64	4.21	0.64	0.74	90
7	153.0	14.2	7.00	7.00	1.23	1.23	100
8	343.3	31.9	1.66	1.71	0.29	0.30	35
Avg.	277.0	25.7	4.53	4.64	0.80	0.82	84

1. Based on surface area
2. These are the pre-retrofit values for surface area and location. One of the two crossover ducts at this site was replaced wi h a larger one. The post-retrofit surface area is 276.7 ft^2 (25.7 m^2), and 82% of the ducts are in the belly.

which the furnace is located to the trunk duct in the other section of the home. Most of the homes have "transverse" floors, which means that the floor joists run across the short dimension of the home with the trunk ducts underneath the joists. The other type of flooring in manufactured homes is the "longitudinal" floor, where the joists run along the long dimension of the home and the trunk ducts are located between the joists.

All of the homes are heated by downflow electric furnaces. The heating systems do not contain a return system but rather have only a return grille in the door of the furnace closet and a filter on the top of the furnace. (One home has a positive operating system, which is a return-side dampered pipe to the outdoors that provides supplemental air for the furnace during operation; this was sealed off for the tests.) In most homes, after passing through the heating elements, the air flows into a 5-inch-deep "T" and is forced to make two right-angle turns into the main trunk duct, which is generally either a 4 in. × 10 in. or 5 in. × 12 in. rectangular cross section. In the majority of homes, the nearest heating register is less than 5 feet away from the furnace.

Five of the seven homes have two sections, and all of these homes have the transverse floor. All of these homes have crossover ducts, which are usually either 8 in. or 10 in. diameter round duct. Three of these homes, denoted Sites 3, 5, and 6, all have one crossover that is located at the furnace such that some of the hot air goes directly from the furnace to the opposite side of the house. Site 2 has two crossover ducts, one at each end of the house, resulting in a rectangular duct system. Site 8 has a splitter box. In this home, the furnace is in the middle of the house and the supply air flows down into two crossover ducts, one to each side of the house. Site 8 also has a badly torn belly blanket, with the portion at the end of one side completely ripped away. This is at the location where one of the crossover ducts was supposed to connect to the trunk ducts; however, prior to the retrofit, the crossover duct was disconnected from the trunk ducts and blew hot air directly into the crawl space. Because of the condition of the belly blanket, 65% of the duct surface area was assumed to be in a vented crawl space with an insulated floor.

The two remaining homes have only one section and, hence, no crossover ducts. One of the two single-wide homes, designated as Site 4, has a longitudinal floor, while the other (Site 7) has a transverse floor. Both of these homes have additional insulation under the floor and around the ducts; at Site 4 the utility installed insulation to R-19 under the floor due to a torn belly blanket, and at Site 7 there was additional insulation under and around the ducts such that the estimated R-value around the ducts is R-7.

The effective R-values listed in Table 2 are the reciprocal of the surface area-weighted U-factors for the main trunk ducts and the crossover ducts.

FIELD MEASUREMENTS

The primary goals of the field testing were twofold: first, to evaluate the savings achieved through an aggressive duct-sealing retrofit, and, second, to acquire sufficient auxiliary data to model the duct efficiency for comparison with measured results. The first goal was addressed by making a complete set of measurements on each home both pre- and post-retrofit. The direct measurements of duct efficiency included a steady-state heat delivery efficiency test and an overnight measurement of overall distribution efficiency. Both of these tests included measurement of interior air temperatures in five to eight zones (usually rooms with supply registers), supply register temperatures, the temperature at the return grille, the supply plenum temperature, the outside temperature, the crawl space temperature, and the temperature of the belly space at several locations. The electricity consumption was measured with clamp-on true power watt meters. Register flows were measured with a calibrated flow hood. Subsequent analysis showed that the supply plenum temperatures were unreliable; the final analysis used supply plenum temperatures calculated from the temperature at the return grille, measured watt consumption by the furnace, and the air-handler flow rate.

The measured steady-state heat delivery efficiency test used data obtained after about 45 minutes of furnace operation. The measured temperature difference between each supply register and the return grille was combined with the measured airflow through the register to compute the heat delivered to the house. The total heat delivered was divided by the measured power consumption by the furnace to obtain the delivery efficiency.

The distribution efficiency was obtained by alternating between heating the home with the furnace and with interior resistance space heaters. The ratio of the power consumption of the space heaters to that by the furnace is the distribution efficiency.

Duct leakage was directly measured at several pressure differences with a duct pressurization leakage tester. These tests included both total supply system leakage (i.e., to both outside and inside) and, by using a blower door to adjust the pressure between the home and the duct system to zero, the duct leakage to outside. Because the leakage of interest is that which occurs when the furnace is operating normally, it is necessary to extrapolate the resulting power law leakage curves to operating conditions. With this in mind, upstream static pressures were measured at each register with a long pitot tube, and the average of these measurements was used as the static pressure under operating conditions. The pre- and post-retrofit leakage to outside as a fraction of air-handler flow for each home is illustrated in Figure 1, which shows that, on average, the retrofits were successful in reducing leakage by nearly 83%. In addition, the blocked flow pressure was measured at each register with a pressure pan, as suggested in Standard 152P. Due to the geometry of the furnace-duct connection, it was not possible to measure a meaningful

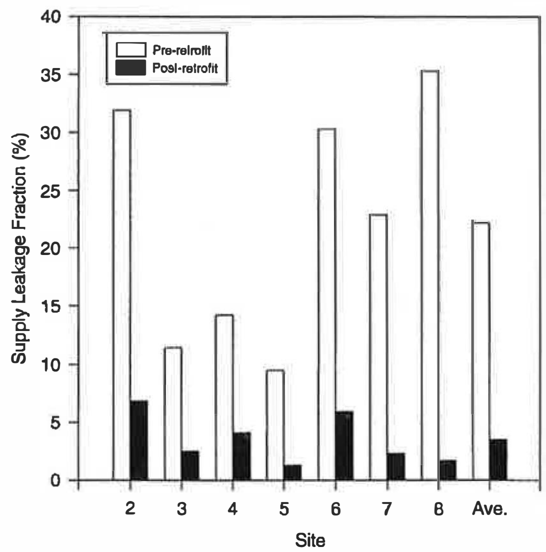

Figure 1 *Pre- and post-retrofit leakage fractions.*

supply plenum pressure. Pressures were also measured across the envelope with the air handler on and off.

The air-handler flow was estimated by several methods including direct measurement with a duct tester. The most consistent method was to add the total supply duct leakage at operating static pressure to the sum of the register flows. The resulting air-handler flows were used for all subsequent analysis.

The house leakage was measured by an automated, multipoint blower door test, both with the registers open and with the registers sealed.

MODELING

For simplicity, modeling was restricted to the case of delivery efficiency. Both the Palmiter/Francisco model and the Standard 152P model have the same basic equation for this measure of efficiency, so this restriction limits comparisons to the methods by which the input parameters are calculated. It also removes the need to estimate a regain factor or the natural infiltration rate of the homes. The equation for delivery efficiency is

$$\eta_{ss-hde} = \alpha\beta - \alpha(1-\beta)\frac{\Delta T_s}{\Delta T_e}$$

where

η_{ss-hde} = steady-state heat delivery efficiency,

α = the fraction of the air-handler flow that is delivered to the house,

β = the conduction efficiency of the ducts,

ΔT_s = the temperature difference between inside and the space where the ducts are located, and

ΔT_e = the temperature rise across the furnace.

This equation is valid for any set of consistent temperature units.

All modeling was done with many of the same measured input parameters, including indoor and outdoor temperatures, air-handler flow, furnace power consumption during operation, duct surface area, and duct insulation level. Since the air-handler flow and furnace consumption were the same for all modeling, the temperature rise across the furnace was also the same. The main variations between the inputs to the models are the duct leakage to outdoors, the method by which the conduction losses are calculated, and the temperature of the zone in which the ducts are located.

Three levels of modeling were done. The first, referred to as Model A, used all measured parameters, including duct leakage to outside (as described in the previous section) and buffer space temperatures. The belly space and crawl space temperatures were combined to take into account that the trunk ducts, located in the belly space, were generally uninsulated, while the crossover ducts, located in the crawl space, were insulated. In addition, a detailed calculation of the conduction efficiency β was used, as opposed to using the overall flow through the duct system and an area-weighted U-factor.

The second model, referred to as Model B, also used the measured duct leakage to outside. The conduction efficiency was based on the overall flow through the duct system combined with the total duct surface area and an area-weighted U-factor. More importantly, it used some of the default Standard 152P duct locations to calculate the effective buffer space temperature. Because Standard 152P does not have belly space as one of the duct location options, external wall was chosen for this portion of the ductwork since this was deemed to most closely resemble an actual belly space. The remainder of the ducts were said to be in a ventilated crawl space with an insulated floor.

Model C used all of the same parameters as Model B with the exception of duct leakage to outdoors. In this case, the duct leakage was estimated using the house depressurization diagnostic described in Standard 152P. This procedure combines measured pressures across the building envelope, both with the air handler off and on, with a one-point blower door test on the home with the registers unsealed and the house depressurized by 25 Pa relative to outdoors. While the duct leakage to the outside that results from this procedure is not a default value, it is an indirect measurement since no direct testing of the ducts occurs. Further, some parameters, such as the power law exponents from the leakage curves, are default values.

COMPARISON OF MODEL PARAMETERS

Models A and B were run using the measured duct leakage to outdoors at operating static pressure. For the Model C blower door house depressurization test described in the previous section, the coefficient of the power law equation is estimated using an assumed exponent of 0.65. To get the leakage at operating conditions, the pressure difference across the building envelope is measured with the air handler off and on. The leakage is then calculated using

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TABLE 3
Pertinent Flows

Site	Pertinent Flows (cfm)									
	Measured Air- Handler Flow		Measured Flow Through Registers		Measured Leakage to Outside		Measured Leakage to Inside		Leakage to Outside using Blower Door	
	Pre	Post	Pre	Post	Pre	Post	Pre	Post	Pre	Post
2	804	794	525	633	256	54	22	107	295	89
3	1030	1024	888	959	117	25	25	40	150	0
4	1015	1088	822	987	144	45	49	56	129	0
5	1580	1693	1376	1607	151	22	53	64	103	0
6	1043	794	672	719	316	47	55	28	381	94
7	688	693	504	648	157	16	27	29	206	0
8	802	1047	518	1029	283	18	0	0	301	0
Avg.	994	1019	758	940	203	32	33	46	224	26

$$Q_{sleak+rleak} = \frac{1}{2}C_{env}(sign(2\Delta P_{off} - \Delta P_{on})|\Delta P_{on} - 2\Delta P_{off}|^{n_{env}} - sign(\Delta P_{on})|\Delta P_{on}|^{n_{env}}) \quad (1)$$

where

$Q_{sleak+rleak}$ = difference between supply and return duct leaks to outdoors (cfm or m^3/s),

C_{env} = power law leakage coefficient (cfm/Pa^n or m^3/(s·Pa^n)),

ΔP_{on} = pressure across the building envelope when the air handler is on (Pa),

ΔP_{off} = pressure across the building envelope when the air handler is off (Pa),

n_{env} = power law leakage exponent.

Since manufactured homes do not have return duct systems, this equation produces the supply duct leakage to outdoors. Standard 152P specifies that the pressure differences across the envelope with the air handler on and off are to be measured across the ceiling to the attic to minimize the effect of wind. However, as manufactured homes do not have attics, these measurements were taken across the door to the home with the external pressure tap placed in a bucket of sand.

Table 3 shows air-handler flow, register flow, measured leakage to inside and outside, and the leakage to outside based on the one-point blower door test as described above. The zero values in the post-retrofit case for the blower door based leakage reflect situations where the pressure differences across the envelope with the air handler off and on were the same to within the noise level of the measurement device. In the pre-retrofit case, the leakage to outside used in Model C averages about 10% greater than the measured value.

The leakage efficiency (α) used in the models is simply the ratio of the applicable duct leakage to outside to the measured air-handler flow. The air-handler flow for all model cases is the sum of the supply register flows plus the total measured leakage at operating conditions, including leaks to inside.

The conduction efficiency (β) used in Model A is calculated as follows: a different conduction efficiency is calculated for each section of ductwork based on the flow through the section and the physical dimensions and insulation level of the section. The conduction efficiency for each register is calculated by multiplying the conduction efficiencies of all sections leading to the register, and the resulting register conduction efficiencies are flow-weight-averaged to provide the overall duct system conduction efficiency. The conduction efficiency used in Models B and C is based on the overall flow through the duct system combined with the total duct surface area and an area-weighted U-factor.

Table 4 shows the parameters α and β used in the various models. The α's for Model C average about 4% less than those used in Models A and B since, on average, the duct leakage is larger. Except for Sites 4 and 7, which have additional insulation around the ducts, the β's tend to be low due to the uninsulated ducts in the belly space. The β values used in Models B and C are slightly lower than those used in Model A. This difference has little effect on the calculated efficiencies.

Temperatures were measured in both the belly space and crawl space of each home. Using the airflow rates through the ducts in each space and standard conductive heat transfer theory, an overall effective buffer space temperature can be calculated. This buffer space temperature is used for Model A. For Models B and C, belly and crawl space temperatures are calculated using weighted averages of the indoor and outdoor temperatures as specified in Standard 152P for exterior walls and ventilated crawl spaces with insulated floors, respectively. The resulting temperatures are then weighted by the fraction of duct surface area in each space to get an overall effective buffer space temperature.

Table 5 shows the indoor and outdoor temperatures, as well as the overall effective buffer space temperatures, both as

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TABLE 4
Leakage and Conduction Efficiencies

Site	Pre-Retrofit (%)				Post-Retrofit (%)			
	α		β		α		β	
	Models A & B	Model C	Model A	Models B & C	Models A & B	Model C	Model A	Models B & C
2	68.1	63.3	78.0	75.4	93.2	88.8	81.3	79.5
3	88.6	85.4	82.5	81.5	97.5	100.0	83.7	82.8
4	85.8	87.3	98.9	98.8	95.9	100.0	99.1	99.0
5	90.5	93.5	89.1	88.4	98.7	100.0	90.3	89.5
6	69.7	63.4	86.7	86.3	94.1	88.1	89.1	88.8
7	77.1	70.0	96.1	96.1	97.7	100.0	96.9	96.9
8	64.7	62.4	73.8	69.1	98.3	100.0	84.3	83.5
Avg.	77.8	75.1	86.4	85.1	96.5	96.7	89.2	88.6

TABLE 5
Pertinent Temperatures

Site	Pertinent Temperatures (°F)							
	Inside		Outside		Buffer		Buffer, 152	
	Pre	Post	Pre	Post	Pre	Post	Pre	Post
2	73.5	75.8	41.3	44.6	61.0	66.4	55.4	58.0
3	82.8	81.6	42.0	41.4	63.4	64.8	60.8	59.9
4	90.5	91.5	43.0	42.1	82.0	80.6	66.7	66.8
5	80.3	77.4	33.5	36.3	62.8	57.4	50.1	55.3
6	82.7	80.6	39.0	49.6	64.9	63.8	59.2	63.9
7	85.9	87.5	57.1	54.7	78.2	72.4	71.5	71.1
8	79.2	86.4	57.2	57.8	76.5	63.3	62.6	64.9
Avg.	82.1	83.0	44.7	46.6	69.8	67.0	60.9	62.8

calculated from measured belly and crawl space temperatures and as estimated by Standard 152P. Compared to the values estimated by Standard 152P, the effective buffer space temperatures based on measured belly and crawl space temperatures average about nine degrees Fahrenheit warmer pre-retrofit and four degrees warmer post-retrofit. This is due in large part because Standard 152P does not take duct losses into account when calculating buffer space temperatures.

Table 6 shows the temperature differences used in modeling delivery efficiency, the temperature rise across the equipment (ΔT_e), and the temperature difference between inside and the buffer space in which the ducts are located (ΔT_s). This shows that, on average, ΔT_s based on Standard 152P is almost twice as large as measured values, and in one case (Site 4) the value from Standard 152P is nearly ten times the measured value. On average, this nearly doubles the temperature-dependent efficiency loss in the models. However, the error in overall efficiency will only be large when β is relatively low (i.e., there are large conduction losses). The average measured ΔT_s increases by about half in the post-retrofit case due to the reduction in air leakage to the buffer space, while the average value based on Standard 152P is almost unchanged.

COMPARISON OF MODELED DELIVERY EFFICIENCIES

The measured and modeled steady-state delivery efficiencies are compared in Table 7. For the pre-retrofit case, the modeled efficiencies are lower, on average, than the measured values. Model A is low by about 2.7 percentage points, Model B by 5.3 percentage points, and Model C by about 7.4 percentage points, where a percentage point corresponds to one percent of the furnace output. The results are about the same based on mean absolute errors. In the post-retrofit case, Model A agrees with the measured values on average, Model B is about 1.6 percentage points low, and Model C is about 1.3 percentage points low. However, based on mean absolute errors, Model A is off by about 3.7 percentage points, Model B by 4.5 percentage points, and Model C by about 5.9 percent-

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TABLE 6
Pertinent Temperature Differences

Site	ΔT_e (°F)		ΔT_s (°F)			
			Measured Buffer		Std. 152P Default Buffer	
	Pre	Post	Pre	Post	Pre	Post
2	47.6	35.7	10.9	9.5	18.1	17.8
3	51.3	50.7	18.0	16.5	22.0	21.7
4	47.2	48.7	2.6	7.6	23.7	24.7
5	43.7	41.1	16.8	19.8	25.2	22.1
6	65.3	66.1	16.1	16.8	23.6	16.7
7	45.8	61.0	5.9	13.4	14.4	16.4
8	63.9	49.7	3.5	22.2	16.5	21.5
Avg.	52.1	50.4	10.5	15.1	20.5	20.1

TABLE 7
Measured and Modeled Delivery Efficiencies

Site	Delivery Efficiency (%)								Difference from Measured (%)					
	Meas.	Pre-Retrofit Model			Meas.	Pre-Retrofit Model			Pre-Retrofit Model			Pre-Retrofit Model		
		A	B	C		A	B	C	A	B	C	A	B	C
2	55.6	49.2	45.0	41.8	75.2	71.2	64.6	61.5	-6.4	-10.6	-13.8	-4.0	-10.6	-13.7
3	66.8	67.3	65.2	62.8	73.7	76.4	73.6	75.4	0.5	-1.6	-4.0	2.7	-0.1	1.7
4	88.1	84.7	84.3	85.7	96.6	94.8	94.5	98.5	-3.4	-3.8	-2.4	-1.8	-2.1	1.9
5	76.7	75.8	73.9	76.4	85.6	83.2	82.8	83.8	-0.9	-2.8	-0.3	-2.4	-2.8	-1.8
6	65.7	57.9	56.7	51.6	85.5	81.2	80.9	75.7	-7.8	-9.0	-14.1	-4.3	-4.6	-9.8
7	74.6	73.6	73.1	66.4	83.9	94.0	93.9	96.1	-1.0	-1.5	-8.2	10.1	10.0	12.2
8	47.3	47.0	39.5	38.1	76.3	75.7	75.1	76.4	-0.3	-7.8	-9.2	-0.6	-1.2	0.1
Avg.	67.8	65.1	62.5	60.4	82.4	82.4	80.8	81.1	-2.7	-5.3	-7.4	0.0	-1.6	-1.3
Mean absolute difference from measured									2.9	5.3	7.4	3.7	4.5	5.9

age points. All of these are weighted heavily by the large disagreement at Site 7. We have no explanation for this disagreement.

As one might expect, as the inputs move progressively from measured values to defaults, the errors grow increasingly worse. One would also expect the post-retrofit modeled results to agree more closely with measured data since the air leakage and its associated uncertainties are greatly reduced.

The efficiencies are compared graphically in Figure 2 for the pre-retrofit case and in Figure 3 for the post-retrofit case. It is interesting to look at the impacts of some of the larger discrepancies in model inputs. For example, ΔT_s is much too large in Models B and C in the pre-retrofit case. At Site 8, where the ducts are uninsulated (and thus β is large), Model B is lower than Model A by 7.5 percentage points. However, at Site 4, Model B is only 0.4 percentage points lower than Model A. This is because the ducts were well insulated at Site 4, and thus the $(1-\beta)$ term in the model is very small.

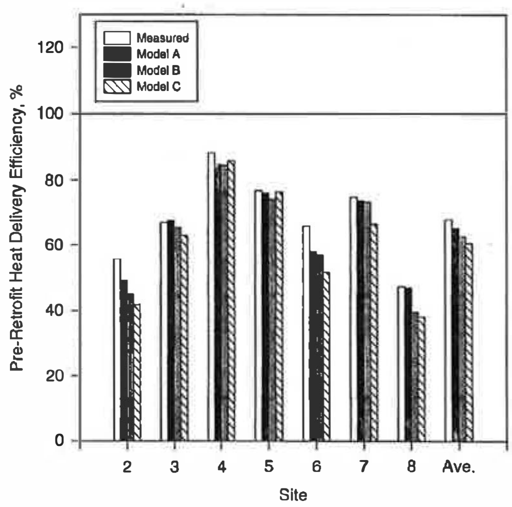

Figure 2 Comparison of measured and modeled pre-retrofit delivery efficiency.

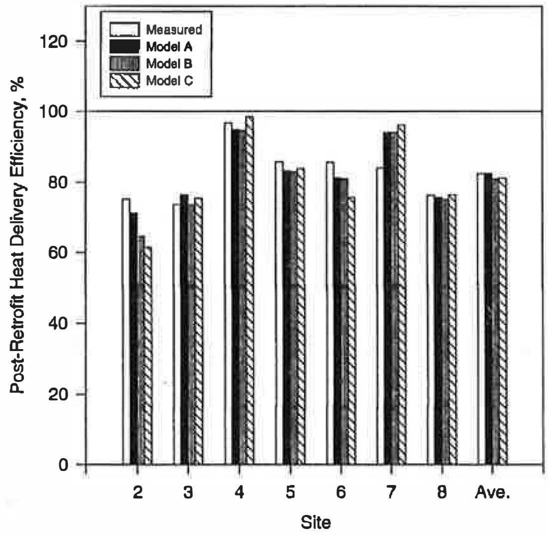

Figure 3 *Comparison of measured and modeled post-retrofit delivery efficiency.*

One of the important potential uses of modeling duct efficiency is to evaluate actual or potential savings from retrofits. Table 8 shows the measured savings and compares these results to the savings predicted by the models, and Figure 4 presents this information graphically. Because energy use is proportional to the reciprocal of fractional efficiency, the percentage savings is given by $(1-\eta_{pre}/\eta_{post}) \times 100$. For this paper, the measured savings are based on the raw results. For instance, the effects of a reduction in furnace capacity due to cycling on the limit switch are ignored, as well as differences in indoor and outdoor temperature before and after retrofit. Because the models underestimate the pre-retrofit efficiency more than the post-retrofit efficiency, on average they produce greater savings than from the measured data. Model A comes the closest, with predicted savings being about 19% higher than the measured values. Model B and Model C overpredict the savings by about 28% and 45%, respectively.

Although the sample size is too small to draw general conclusions about the adequacy of the models, we investigated whether the models show any statistically significant bias compared with the measured values. Figure 5 shows a scatter plot of the Model A efficiencies versus the measured values. The straight line is a one-one line. A simple linear regression gives a slope of 1.059 and an offset of -0.058. A test of whether the slope equals 1.00 and the offset equals 0.0 indicated a 54% probability of getting values this discrepant if the model was in fact unbiased. Another test performed an iteratively reweighted robust regression on the same data; the slope was 1.0029 and the offset was -0.024.

Figures 6 and 7 show the same scatter plots for Models B and C, respectively. For these models, the statistical tests showed increasing evidence of bias, with the results for Model C almost significant at the 5% level.

ADDITIONAL CONSIDERATIONS FOR STANDARD 152P

In addition to the previously discussed factors affecting the modeling of delivery efficiency, there are other details about Standard 152P worthy of discussion.

Measurement of House Leakage for Use in Estimating Duct Leakage

In the previous discussion of Model C, whole house leakage (including the ductwork) at 25 Pa depressurization was used in calculating the leakage coefficient for Equation 1. The essential idea behind this diagnostic technique for measuring leakage is to use the house as a kind of calibrated orifice for estimating the duct leakage. Simple reasoning shows that when the air handler is on, the portion of leakage in the ductwork is irrelevant. There is a pressure change across the envelope caused by unbalanced flows to the home from the duct

TABLE 8
Measured and Modeled Savings Due to Retrofit (Based on Delivery Efficiency)

Site	Savings (%)				Difference from Measured (%)		
	Measured	Model A	Model B	Model C	Model A	Model B	Model C
2	26.1	30.9	30.3	32.0	4.8	4.2	5.9
3	9.4	11.9	11.4	16.7	2.5	2.0	7.3
4	8.8	10.7	10.8	13.0	1.9	2.0	4.2
5	10.4	8.9	10.7	8.8	-1.5	0.3	-1.6
6	23.2	28.7	29.9	31.8	5.5	6.7	8.6
7	11.1	21.7	22.2	30.9	10.6	11.1	19.8
8	38.0	37.9	47.4	50.1	-0.1	9.4	12.1
Avg.	18.1	21.5	23.2	26.2	3.4	5.1	8.0
Mean absolute difference from measured					3.8	5.1	8.5

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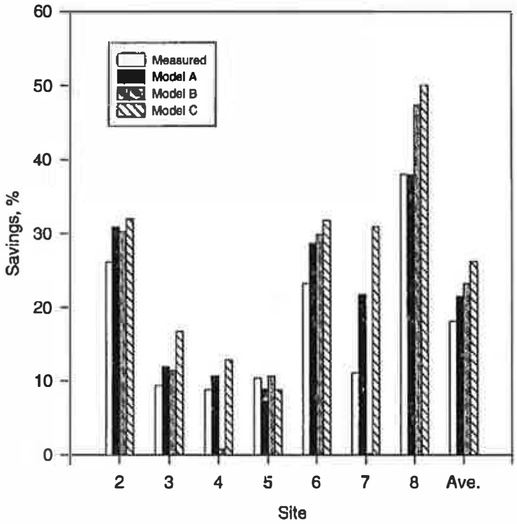

Figure 4 *Comparison of measured and modeled retrofit savings estimates.*

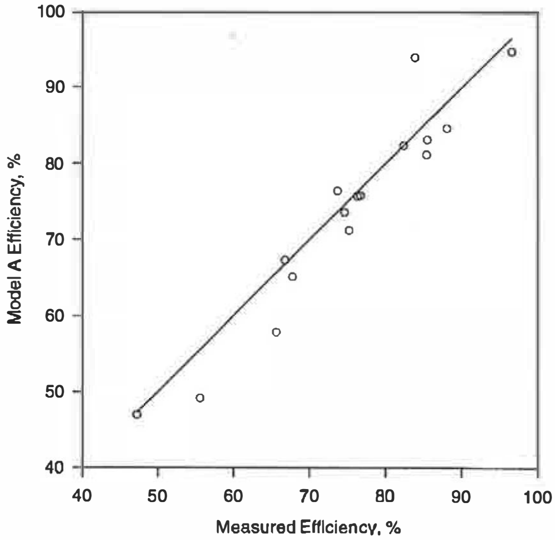

Figure 5 *Comparison of Model A to measured delivered efficiency.*

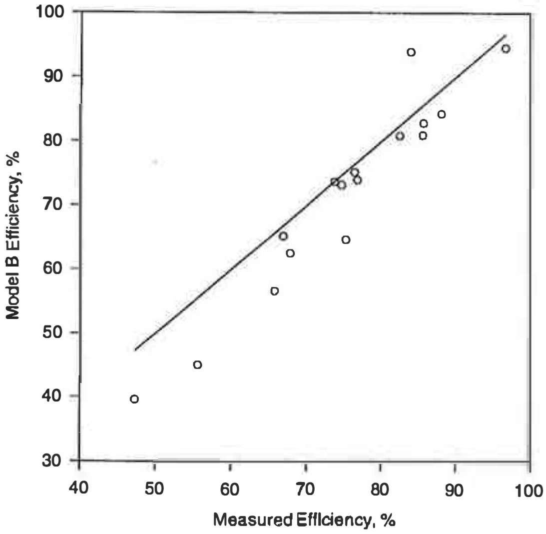

Figure 6 *Comparison of Model B to measured delivered efficiency.*

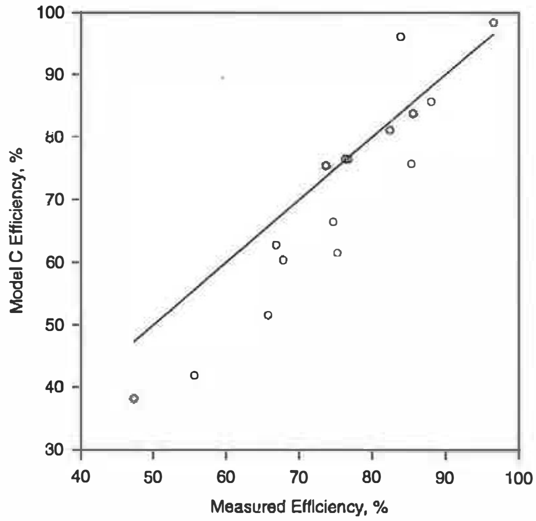

Figure 7 *Comparison of Model C to measured delivered efficiency.*

system. The observed pressure change is due solely to the leakage characteristics of the envelope. By including the holes in the ducts in the diagnostic, the estimated duct leakage tends to be too high. It would be preferable to use pressurization data measured with the registers sealed. To the extent that there is large leakage from the duct system to the interior, even the value with registers sealed will be too large.

The effect of using sealed versus unsealed pressurization test data is illustrated in Table 9. Only the pre-retrofit results are shown since there are only two non-zero values for the pressure method in the post-retrofit case and the leakage is generally quite small. The first column gives the measured value of leakage to outside. The next two columns show the leakage from the unsealed test as used previously in Model C along with the leakage from the sealed test. On average, the estimated duct leakage from the depressurization test is reduced by about 15% by sealing the registers. The next two columns show that the consequent change in α is about 5%. The next two columns show the change in efficiency due to the change in α (the efficiencies change in the same ratio as α). These efficiencies use the β's and buffer zone temperatures of Model B. The last two columns compare the differences from the measured efficiency for the two cases. At every site, the discrepancy is reduced when the sealed tests are used. On average the bias is reduced from −7.4% to −4.5%.

The α's resulting from the sealed test are much closer to the measured values than are the ones from the unsealed test. This agreement is encouraging because the two sets of α's

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TABLE 9
Effects of Sealing Registers for House Pressurization Leakage Diagnostic

Site	Leakage to Outside at Operating Conditions (cfm)			α(%)		Delivery Efficiency (%)		Difference from Measured (%)	
	Measured	Unsealed	Sealed	Unsealed	Sealed	Unsealed	Sealed	Unsealed	Sealed
2	256	295	275	63.3	65.8	41.8	43.4	-13.8	-12.2
3	117	150	132	85.4	87.1	62.8	64.1	-4.0	-2.7
4	144	129	118	87.3	88.4	85.7	86.7	-2.4	-1.4
5	151	103	96	93.5	93.9	76.4	76.8	-0.3	0.1
6	316	381	293	63.4	71.9	51.6	58.5	-14.1	-7.2
7	157	206	182	70.0	73.5	66.4	69.7	-8.2	-4.9
8	283	301	224	62.4	72.1	38.1	44.0	-9.2	-3.3
Avg.	203	224	189	75.1	79.0	60.4	63.3	-7.4	-4.5
Mean absolute difference from measured								7.4	4.5

TABLE 10
Furnace Power Consumption and Operating Static Pressure Comparisons

Site	Average Furnace Power Consumption (W)					Average Register Pressure (Pa)			
	Nameplate	Steady-State		Cycling		w/ Pitot Tube		Blocked Flow	
		Pre	Post	Pre	Post	Pre	Post	Pre	Post
2	17300	12107	8964	10984	8804	10.0	10.0	9.0	17.9
3	17500	16720	16439	14314	14383	9.7	12.7	20.1	13.0
4	15700	15149	15206	13385	13200	14.0	19.0	30.5	24.3
5	22500	21843	22026	18525	18589	16.0	17.3	30.3	35.2
6	22500	21547	16615	17484	15272	10.8	7.3	9.1	10.8
7	15200	9982	13361	11709	12236	14.2	18.2	17.7	33.5
8	16500	16225	16498	14652	14828	10.3	14.3	9.9	32.4
Avg.	18171	16225	15587	14436	13902	12.1	14.1	18.1	23.9

were estimated using completely independent methods. We also calculated the efficiencies that would result from using all the parameters from Model A but substituting the α's from the pressure diagnostic test. We found the predicted efficiencies based on the sealed test agreed about as well with the measured efficiencies as did Model A, while those from the unsealed test were, on average, biased low by about three percentage points.

Furnace Output

For a fixed flow rate, duct efficiency can be strongly affected by the output capacity of the furnace. Under the steady-state conditions considered in this paper, one might expect the unit to have an output close to the rated capacity. However, in some of these homes, the combination of capacity, flow rate, and elevated indoor temperature resulted in cycling of the high-temperature limit switch. The values of the measured average output near the end of the steady-state test are compared with the nominal output rating from the nameplate in Table 10. Due primarily to high-limit cycling, the actual power output is about 11% less than the nameplate rating. This situation results in a lower duct efficiency than would have been obtained under full power. Other reasons output capacity may differ from the nameplate rating include having a burned out element or a defective sequencer and having a replacement element of different wattage from the original.

Under normal cycling conditions, there is an even more important output capacity effect due to the fact that electric furnaces and electric resistance backup heaters for heat pumps are frequently controlled by sequencers. Individual heating elements are usually in the 4 to 5 kW range. On a typical unit, the fan comes on simultaneously with some of the heaters; there is then a delay of 30 to 60 seconds before the next set of elements comes on. There is considerable variation among furnaces in the sequencing logic. For instance, 15 kW furnaces may start up at 10 kW and sequence to 15 kW, or they may start up at 5 kW and sequence to 15 kW, or they may sequence in three separate 5 kW increments. This has potentially important impacts on duct efficiency. In the initial part of the on-cycle, the reduced capacity leads to rather low delivery efficiency, while in the final phase, the effect is to scavenge heat stored in the ducts and thus improve efficiency. More importantly, under conditions of low load relative to total capacity, the unit will satisfy the thermostat before reaching full output. The furnace behaves as a furnace of two-thirds or even one-third of the rated capacity, much like a modulating burner. The duct efficiency can be substantially degraded under low part-load conditions.

The measured values of average heat output during normal cycling at the seven manufactured homes are also shown in Table 10 for comparison. These values are averaged only over the time when the fan is on. The cycling power is about 11% less than the steady-state power and about 22% less than the nameplate rating.

Measurement of Duct System Static Pressure Under Operating Conditions

Another suggested method to obtain duct leakage at operating conditions in Standard 152P is to use the same duct pressurization test as was used for Model A but to obtain the operating static pressure of the duct system with a pressure pan. In this method, a pressure pan is placed over each register while the air handler is operating, and the measured pressures across the pressure pan are averaged. However, this effectively measures the pressure at each register when that register is sealed. Sealing one register increases the pressure everywhere in the duct system, so the static pressure as measured with the pressure pan will be higher than the pressure that actually exists during normal operation.

The final four columns in Table 10 compare the average operating static, as measured in the field study using a long pitot tube placed facing upstream in the main trunk at each register, with the average pressure obtained with the pressure pan, or "blocked flow," technique. On average, in the pre-retrofit case, the blocked-flow pressure measurements result in an operating static pressure that is about 50% higher than the average static pressure as measured with a pitot tube. Since the standard suggests using an exponent of 0.6 to get the leakage to outside, this will result in a leakage 27% greater than that obtained using the pitot tube. In the post-retrofit case, the discrepancy becomes worse, with the blocked-flow pressure averaging about 70% higher than the pitot tube pressure measurements, which results in a leakage that is 37% greater.

DISCUSSION

Before discussing the results, it is well to note several important caveats. First, with a sample size this small, one should be cautious not to overinterpret the results. It is possible that similar tests on an additional 100 manufactured homes would lead to quite different conclusions regarding some of the comparisons. Second, the homes discussed here represent a very specialized type of duct system (i.e., no return ducts, no insulation on most of ducts, ducts located in insulated and partially sealed belly space, presence of crossover ducts in zone of very different temperature, etc.). Some of the conclusions and comparisons might come out quite differently for homes without these special features. Third, we discuss only a small number of comparisons here. For instance, with the field measurements that were made we have about five ways of estimating the air-handler flow, about four ways of estimating the duct leakage to outside, several ways of estimating conduction losses, several ways of estimating buffer zone temperatures for the belly and crawl space temperatures, and several ways of combining these temperatures. The potential range of values for the estimated efficiency is larger than that indicated by the few comparisons discussed here. Also, incorporation of additional default values (i.e., outside temperature) will contribute even larger variations.

Given the simplicity of the engineering model leading to the efficiency equation and the difficulty and attendant uncertainties in measuring and estimating the model parameters, the level of agreement between the measured and modeled efficiencies is surprisingly good, especially for Model A. This model—or its variant using α's from the pressure diagnostic test—delivered parameters closest to those derived from direct measurement. It should be noted that no parameter tuning was done to improve agreement with the measured values, although in some instances, large discrepancies led to the discovery of mistakes in the calculations that were subsequently corrected.

Although modeled efficiencies from Models B and C are still fairly close to the measured efficiencies, the discrepancies are of noticeable size for purposes such as estimating the loss fraction. We would argue that, in general, the loss fraction is a more important parameter because it measures the potential for energy conservation measures applied to distribution systems. For instance, if two methods produce estimated efficiencies of 80% and 85%, they differ by only five percentage points. However, the losses are 20% and 15%, respectively. If some conservation measures could eliminate these losses, the apparent cost-effectiveness of these measures would differ by 25%. Unfortunately, as illustrated by Table 8, it is unlikely that the relative error in losses can ever be estimated better than about 15%, even with improved measuring techniques.

As one would expect, there is a clear trend toward progressively more inaccurate results as more default assumptions and parameters are incorporated into the efficiency estimates. This suggests caution in accepting results based largely on generic default values until they can be confirmed by field measurement on a suitably large sample of representative homes.

The comparisons presented earlier in this paper provide a basis for some suggested improvements to Standard 152P. These can be summarized as follows:

1. Measure the output of electric resistance furnaces and backup units. This will avoid situations where the output differs from the nameplate due to replaced elements of

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different rating and will limit switch cycling, low voltage, and nonfunctional elements.

2. Seal the registers when doing the pressurization test to get an envelope flow coefficient for use in the envelope pressure diagnostic method. This helps to eliminate a fairly large bias in the estimated leakage. It is a trivial change in the standard and does not require much time in the field.
3. Use the static tap on a long, thin pitot tube to measure the upstream static pressure at each register rather than using the blocked pressure. The measurements are equally easy either way, and the pitot tube yields less biased results.
4. Improve the calculation of buffer zone temperatures so that the increase due to duct losses is properly accounted for. This is more difficult to implement than the previous suggestions, as it requires a change to the model equations in the standard. The solution requires iteration (Palmiter and Francisco 1997). The current buffer zone temperatures are very badly biased for the uninsulated duct in exterior wall scenario.

Finally, we cannot overemphasize the great need for further field measurement to verify the general reasonableness of Standard 152P procedures over a wide range of housing types, climates, duct system types, duct locations, and heating and cooling equipment types. The blind application of simplified models with many generic defaults may result in estimated efficiencies with unacceptably large biases.

SUMMARY

Modeling of delivery efficiency was performed using three levels of combined measured and default input parameters and compared to measured data from seven manufactured homes. Using values based on all measured data provided results that were closest to measured data of any of the models. As individual measured parameters were replaced by estimated or default values, the agreement with measured results worsened. All of the models gave results that were, on average, within eight percentage points of measured results, but it should be noted that many potential default values were not used and that the comparisons were for a very simple type of house, as manufactured homes do not have return duct systems. One would expect that in more complex houses and as more default values are used, agreement would worsen even further.

The primary parameters that were varied were the leakage to outdoors and the temperatures in the buffer spaces in which the ducts were located. On their own, substituting estimated or default values for these parameters worsened agreement with measured results by a few percentage points. However, simple modifications to the way in which these values are obtained would improve the model predictions. For example, either measured buffer space temperatures or some modification to the calculation process to account for the impact of duct losses on buffer space temperatures would provide better results. Also, in the case of using a blower door depressurization test along with measured pressure differences across the envelope with the air handler off and on to estimate duct leakage, sealing the registers for the blower door test can lead to a significant improvement in the measured leakage to outdoors and, hence, a better model prediction.

There are also additional suggestions that do not affect the results presented in this paper. For example, measuring the furnace output rather than using the nameplate output to calculate the temperature rise across the furnace can greatly affect the temperature-dependent portion of the duct efficiency, since in many cases the output of the furnace under operating conditions, or even at steady state, can be very different from the rated output. Another is to measure the duct system static pressure under operating conditions with a long pitot tube rather than a pressure pan, since using a pressure pan inherently changes the pressure in the entire system.

UNIT CONVERSIONS

To make this paper easier to read, some data were presented in only one set of units. This section provides the appropriate conversion factors:

Flow: cfm = 2118.88 m^3/s
Temperature: °F = 1.8°C + 32
Temperature difference: F = 1.8°C
Power: W = Btu/3.413
Pressure: Pa = in. $H_2O/248.84$

REFERENCES

ASHRAE. 1996. BSR/ASHRAE Standard 152P, Method of test for determining the steady-state and seasonal efficiencies of residential thermal distribution systems. November 1996 draft version. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Palmiter, L., and P.W. Francisco. 1997. *Development of a practical method for estimating the thermal efficiency of residential forced-air distribution systems*. Mountain View: Electric Power Research Institute. Report TR-107744.

Siegel, J., B. Davis, P. Francisco, and L. Palmiter. 1997. *Measured heating system efficiency retrofits in eight manufactured (HUD-code) homes*. Mountain View: Electric Power Research Institute. Report TR-107737.

Measurements of Window Air Leakage at Cold Temperatures and Impact on Annual Energy Performance of a House

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ABSTRACT

This study was initiated to determine the extent of cold temperature air leakage from operable windows available in today's marketplace and the impact that this has on the energy consumption of a house. During the heating season, changes in the window's leakage characteristics, as a result of thermal and pressure effects, were to be included.

At two laboratories, air-leakage tests down to −30°C were performed on 35 windows, enough to reach some general conclusions about performance.

The majority of windows met or exceeded the highest levels of air leakage performance of Canadian window standards at normal temperatures, and many did very well even at the lowest temperatures tested. Increased leakage at cold temperatures was attributed to design more than frame material; vertical sliders (including tilt-in) exhibited the worst performance under all conditions.

Data were used to quantify the impact on the annual energy performance of a house. The Canadian Standards Association's (CSA) window performance indicator for houses, the CSA Energy Rating (ER) number (CSA 1993), was evaluated for the case of variable air infiltration with outdoor climate. Results were encouraging; for the majority of windows the impact on the rating was negligible, as long as leakage was not excessive at normal temperatures. For others, lessons were learned about materials and designs that could be used to improve product performance at extreme temperatures.

INTRODUCTION

Excessive window air leakage during the heating season can result in three undesirable consequences for a homeowner: reduced comfort from cold drafts, increased demand on the house heating system, which may not be capable of meeting load requirements, and increased energy consumption. This study attempts to characterize real-world leakage for a range of production windows and quantify the influence on the energy consumption of a house. It is presumed that proper installation practice for both operable and fixed windows will eliminate other site-specific leakage associated with windows.

The impact of windows on house energy performance can be shown to result from three window energy fluxes: solar energy entering interior spaces, thermal energy transmitted through the window to or from the interior, and mass transfer of air to or from the inside through (or past) window parts. Although the latter component is usually considerably less than the first two, for well constructed and installed windows, it has not been easy to estimate, and its contribution in severe conditions, such as in cold climates, needed to be evaluated.

Annual energy performance estimates for fenestration are part of Canadian Standards Association (CSA) standards in Canada and National Fenestration Rating Council (NFRC) standards in the U.S. These in turn are referenced by building energy codes to set minimum levels of energy performance (Henry 1995). The standards attempt to account for infiltration effects through all weather conditions experienced during the heating season; however, window leakage data generally have been available only for specific test conditions, usually 20°C and one specific pressure difference. Similarly, energy simulation programs, whether performing hourly calculations or applying a bin approach, simplify window air leakage calculations, often only applying wind and stack effects to estimated overall house leakage.

Some manufacturers in Canada claimed that their products were not fairly rated by the CSA Energy Rating (ER) number since their products maintained low air leakage under winter conditions while competitors' products distorted and

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leaked excessively. Testing was needed to see if these claims were justified and if there was a significant impact on the established ER numbers.

THE TESTING PROGRAMS

To date, the majority of window air-leakage data has been collected in tests following ASTM E283, which specifies a test method for ideal conditions, for instance, 20°C on both sides of the specimen. Results have been shown to be quite reproducible and have been useful to compare products, albeit under ambient conditions for which air leakage usually is not of concern. Since it has been well known that temperature gradients can cause shrinkage and distortion of window components, leakage characteristics under more extreme summer and winter conditions would be more useful.

In 1991, a new standard, ASTM E1424-91, was put forward to determine rate of air leakage through building envelope components as a function of temperature and pressure differentials. Calibration requirements for test equipment, however, are still under development. The only previous reported work on window air leakage at any temperature other than standard conditions was that carried out by Kehrli in 1989. He tested a few windows at low temperatures following a cycle of both low and elevated temperatures. Results suggested some very real air leakage problems with the few windows tested but failed to identify causes or quantify energy impacts.

As support to CSA standards development, the government of Canada commissioned two separate studies, one at a commercial testing laboratory in Montreal and the other at the National Research Council (NRC), to look into changes in window air leakage as a result of both temperature and pressure differentials. These projects were initiated to test representative windows from the marketplace to determine the extent of the problems and to make recommendations with respect to the ASTM interim standard for testing under such conditions. If warranted, additional test requirements could be added to CSA standards to better account for air leakage in annual window energy performance estimates.

The test method evaluated and used (in a modified form) in both cases was ASTM E1424-91. Changes to the procedure were validated by means of inter-laboratory comparisons with a few selected windows.

Windows Tested

In all, five windows were tested at NRC and 30 at the commercial testing laboratory. These were representative of the broad range of operable window types sold in Canada: casement, vertical sliding, horizontal sliding, tilt-turn, and awning. They were off-the-shelf units obtained from local suppliers. Frames and sashes were constructed of aluminum, vinyl, fiberglass, wood, and wood with vinyl cladding. A varied assortment of hardware (individual locks, multi-point locks, snubbers, etc.) and weather stripping was used on the products. A brief description of each is contained in the table of results. Windows were mounted in testing surround panels according to manufacturers' normal instructions for mounting in walls.

Tests at NRC

Windows were mounted in a surround panel with the weather side facing an environmental chamber capable of providing temperature and pressure differences with the room side of up to 40 K and up to 1500 Pa (Elmahdy 1995). An air-leakage apparatus was installed on the room side along with instrumentation in accordance with ASTM E1424-91. Details of the test apparatus and test procedure are described in the reference.

Five windows were tested at room-side temperature of 20°C and weather-side temperatures of 20°C, −5°C, and −20°C and six pressure differentials: 0, 50, 75, 100, 200, and 300 Pa.

Generally, all windows showed increasing leakage with increased temperature and pressure differential. Tests showed the greatest increase with pressure differential but went far beyond average pressures experienced in houses during heating seasons. Two of the casement windows actually showed slightly decreased leakage at greater temperature differential, possibly a result of members closing with shrinkage. Although no trends could be identified with window type (casement, vertical slider, horizontal slider, etc.) or frame material, it was noted that measurements on one particular window were accurately repeatable after a period of time. Air-leakage measurements per unit crack length, at 75 Pa pressure differential, are included in Table 1.

Tests at Commercial Laboratory

Windows were mounted in a mask wall between two chambers, one maintained at 20°C, the other at the test condition temperatures (NRCan 1995). Laminar flowmeters were used to measure leakage rate from the "inside" chamber at a set ΔP. A schematic of the test setup is shown in Figure 1. Further details of instrumentation, window mounting, and test procedures are contained in the reference.

Air leakage measurements per unit length of crack are listed in Table 1 for the 22 windows at ΔP = 75 pascals and three outdoor temperatures (T = −30°C, 0°C, and 20°C). These values span the normal range of conditions expected during the heating season in Canada. Additional data were also obtained at an elevated temperature of 50°C and at ΔP = 300 pascals but are not included here. Subsequent to the initial test, additional data were reported (CSA 1996) and are included in Table 1 as A-I: 23 to A-I: 30.

Careful observation during testing revealed the following information:

- Exterior lock-side corners of casement windows warped at cold temperatures by as much as 3.5 mm in relation to the frame. The magnitude was as much a function of design (number and location of locking points, profile rigidity, etc.) as sash material (wood, aluminum, vinyl, etc.)

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TABLE 1
Summary of Air Leakage Test Results for 35 Windows

Sample	Type	Material	Hardware	Measured Infiltration per Unit Crack Length (m^3/h·m) at Indoor Air 20°C and ΔP = 75 Pa for Outdoor Air (°C) −30	0	20	Infiltration Heat Loss/yr (MJ/m) New/Old	Change in ER
A-I: 1	Casement	f: wood/PVC clad	2 locks	1.10	0.65	0.45(A3)	6.1/3.8	−0.4
		s: PVC	1 snubber					−0.1
A-I: 2	Casement	all wood	2 locks	0.25	0.25	0.20 (A3)	2.1/1.7	
			1 snubber					
A-I: 3	Casement	f: wood/PVC clad	2 locks	0.45	0.21	0.18 (A3)	2.1/1.5	−0.1
		s: PVC	1 snubber					
A-I: 4	Casement	all PVC	2 locks	0.18	0.07	0.06 (A3)	0.7/0.5	0.0
			1 snubber					
A-I: 5	Casement	all PVC	mp locking (3)	0.12	0.12	0.06 (A3)	1.0/0.5	−0.1
			1 snubber					
A-I: 6	Casement	all PVC	3 locks	0.23	0.21	0.16 (A3)	1.8/1.3	−0.1
			2 snubbers					
A-I: 7	Casement	all PVC	mp locking (3)	2.84	0.90	0.59 (A2)	10.3/5.0	−0.6
			1 snubber					
A-I: 8	Casement	all PVC	mp locking (3)	1.35	0.28	0.27 (A3)	3.8/2.3	−0.2
			2 snubbers					
A-I: 9	Casement	same as A-I: 8 with a larger bulb weatherstrip		0.83	0.27	0.24 (A3)	3.1/2.0	−0.2
A-I: 10	Casement	same as A-I: 8 but	2 locks	0.58	0.52	0.47 (A3)	4.4/3.9	−0.1
			2 snubbers					
A-I: 11	Casement	aluminum	mp locking (2)	2.72	1.07	0.82 (A2)	11.3/6.9	−0.7
A-I: 12	Casement	aluminum	2 locks	0.45	0.37	0.34 (A3)	3.2/2.9	−0.1
A-I: 13	Casement	aluminum	mp locking (3)	0.78	0.39	0.25 (A3)	3.8/2.1	−0.2
			3 snubbers					
A-I: 14	Casement	f: alum/wood/PVC		1.09	0.72	0.58 (A2)	6.6/4.9	−0.3
		s: alum/PVC						−0.1
A-I: 15	Tilt-turn	PVC	5 locks	0.58	0.40	0.37 (A3)	3.6/3.1	−0.1
A-I: 16	Awning	wood	2 locks	0.43	0.41	0.37 (A3)	3.5/3.1	−0.1
A-I: 17	Hor sliding	aluminum	4 moving sashes	0.63	0.55 (A3)		5.3/4.6	−0.1
A-I: 18	Hor Sliding	PVC	1 moving sash	3.23	2.59	1.89 (A1)	22.6/15.9	−1.1
A-I: 19	Hor Sliding	PVC	1 moving sash	1.17	1.06	1.01 (A2)	9.1/8.5	−0.1
A-I: 20	Vert Sliding	wood/PVC clad	2 tilt-in sashes	0.62	0.58	0.46 (A3)	4.9/3.9	−0.2
A-I: 21	Vert Sliding	PVC	1 moving sash	1.62	1.93	2.85	20.0/23.9	1.1
A-I: 22	Vert Sliding	PVC	1 tilt-in sash	9.19	5.83	3.69	53.6/31.0	−3.1
A-I: 23	Vert Sliding	PVC	1 moving sash	0.74	0.58	0.54 (A3)	5.1/4.5	−0.1
A-I: 24	Hor Sliding	PVC	1 moving sash	1.11	1.02	0.57 (A2)	8.7/6.2	−0.4

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TABLE 1 (Continued)
Summary of Air Leakage Test Results for 35 Windows

Sample	Type	Material	Hardware	Measured Infiltration per Unit Crack Length ($m^3/h{\cdot}m$) at Indoor Air 20°C and ΔP = 75 Pa for Outdoor Air (°C)			Infiltration Heat Loss/yr (MJ/m)	
				−30	0	20	New/Old	Change in ER
A-I: 25	Casement	PVC	mp locking (3)	0.72	0.51	0.46 (A3)	4.6/3.9	−0.2
			1 snubber					
A-I: 26	Casement	f: Wood/PVC clad	2 locks	2.66	1.02	0.57 (A2)	10.9/4.8	−0.7
		s: PVC						
A-I: 27	Casement	Wood/Aum clad	2 locks	1.66	1.63	1.56 (A2)	13.7/13.1	−0.2
A-I: 28	Casement	PVC	mp locking (3)	0.90	0.41	0.34 (A3)	4.1/2.9	−0.2
			2 snubbers					
A-I: 29	Casement	PVC	2 locks	0.37	0.08	0.05 (A3)	1.1/0.4	−0.1
			1 snubber					
A-I: 30	Casement	PVC	2 locks	0.22	0.11	0.06 (A3)	1.1/0.5	−0.1
			1 snubber					
				for Outdoor Air (°C)				
				−30	0	20		
NRC: 1	Casement	aluminum one of two units operable		0.63	0.61	0.58 (A2)	5.1/4.9	−0.1
NRC: 2	Casement	fiberglass		0.17	0.18	0.18 (A3)	1.5/1.5	0.0
NRC: 3	Casement	wood		0.48	0.48	0.40 (A3)	3.9/3.4	−0.1
NRC: 4	VertSlider	aluminum		6.05	6.34	6.03	52.1/50.6	−0.3
NRC: 5	Vert Slider	PVC		1.92	1.80	1.30 (A2)	15.5/10.9	−0.6

- With double-hung windows, contraction of the sash with respect to the frame sometimes resulted in loss of contact between weather strips and the adjacent members. Windows with tilt-in sash had particularly bad performance at low-temperatures.
- With respect to the horizontal sliders, tilt-turn, and awning windows tested, little differential movement was noticed.

The test results were even more revealing:

- Virtually all windows demonstrated increased air leakage at low temperatures. For 75 pascal ΔP, the worst had five times the air leakage at −30°C as at 20°C.
- Increases at low temperature depended moderately on window type. Casement increases seemed to be greater than other types.
- Variation in infiltration rate for a given product cannot be linked to the type of frame and sash material (PVC, aluminum, wood, etc.) but rather to overall product design. Changes in weather stripping and hardware design and location can radically change performance.

The Canadian window performance standard CAN/CSA-A440-M90 provides for rating of operable windows in three categories when tested at the specified 20°C and ΔP = 75 Pa:

A1: less than 2.79 ($m^3/h{\cdot}m$)

A2: less than 1.65 ($m^3/h{\cdot}m$)

A3: less than 0.55 ($m^3/h{\cdot}m$)

Most windows were very good by this standard, falling into the best category (A3). A few were rated A2, and two PVC sliders were rated A1. Although the standard does not apply to tests at other than 20°C, it is useful to consider the same categories at lower temperatures. Almost all windows increased air leakage at cold temperatures, some substantially, but nearly half remain in the same category even at −30°C. Others moved down a category, and two products would not meet the minimum level.

The ASTM E-1424-91 standard proved effective to evaluate operable window performance at low temperatures. To minimize costs of testing windows at many temperatures, energy performance standards could be changed to call for two tests, one at 20°C as required now and another at a winter design temperature (e.g., 2.5% January temperature), or, alter-

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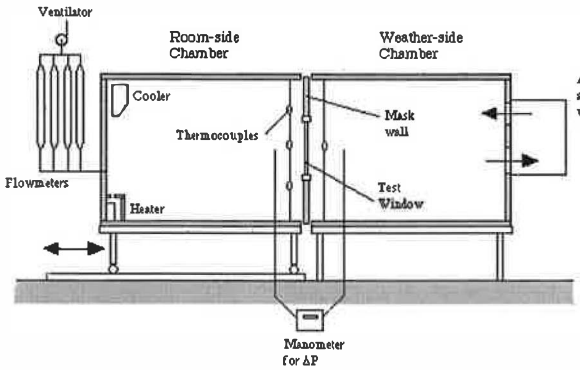

Figure 1 *Test facility in Montreal for determining window air leakage at cold temperatures.*

natively, just a single test but at some more representative heating season temperature such as −5°C.

IMPACT ON ANNUAL ENERGY RATING

Both the CSA and NFRC annual energy rating systems for windows include terms for air leakage losses. Both currently use estimates based on measurements made with ASTM E283, in other words, at 20°C and no ΔT across the unit.

In CSA A440.2, the window ER number is calculated from

$$ER = 72.2\ SHGC - 21.9\ U_w - 0.54\ (L_{75}/Aw) \qquad (1)$$

where

ER = window energy rating, W/m^2;

SHGC = window solar heat gain coefficient, dimensionless;

U_w = overall window U-factor, W/m^2·°C;

L_{75} = window air leakage rate at 75 Pa, m^3/h;

Aw = total window area, m^2.

The last term, the air-leakage term, uses a measured quantity, L_{75}, determined according to CSA A440 (ASTM E283 test). The coefficient in this term, 0.54, as well as coefficients for other terms in the ER equation, were determined by a process of modeling a standard two-story house in many Canadian and northern U.S. locations over a typical heating season. Until now, for lack of better data and because this term is small compared to the other two terms in the ER equation, air leakage data measured only at 20°C have been used.

A different approach is taken to determining a fenestration heating rating (FHR) in NFRC 900, but contributions come from solar gain, transmission loss, and air leakage as in CSA A440.2, with the latter evaluated again in a simplified manner.

It is not sufficient to obtain air leakage at some design temperature, for instance −25°C, and use this in the air leakage term, since during the heating season, climatic conditions are probably only at this temperature for a few hours, if at all. Rather, the combined effects of wind and temperature (stack effect) on pressure together with variations in outside temperature, during all of the hours of the heating season, should be evaluated.

Data obtained from measurements of typical window characteristics for air leakage under a range of pressure and temperature differentials could be used in hour-by-hour energy simulations to evaluate the impact on rating numbers. Windows with specific low-temperature air-leakage characteristics could be modeled in a particular climate, over a complete heating season, to determine the effect on energy consumption.

Such a study was undertaken for CANMET as part of a CSA study (NRCan 1996). To simplify calculations, binned weather data were used with the ENERPASS energy simulation computer program to model a typical house in a number of Canadian cities. The house characteristics were the same as, and the process similar to, those used to develop the ER equation, which is the basis for the current CSA standard for window annual energy performance.

Five windows from the commercial testing laboratory study were evaluated. These were selected from among casement and slider types representative of average and high air-leakage rates at 20°C and 75 Pa. Measured data taken at −20°C, 0°C, and 20°C were used to produce profiles as shown in Figure 2.

For each window, its air leakage characteristics as a function of outdoor temperature and pressure difference (due to wind and stack effects) for a specific location are calculated for the bin hours during the heating season. In Figure 3, results are shown for one example (Montreal weather and window A-I: 18), which includes comparison with infiltration estimation based on 20°C measurements only. Note that roughness in the curve is simply a result of temperatures being recorded in bins and the finite number of hours recorded. Finally, infiltration rates are summed for the hours at each temperature difference to determine annual infiltration heat loss. The impact of the air infiltration at other temperatures was approximated by using an average outside air temperature of −5°C and an average pressure difference of 3.4 Pa during

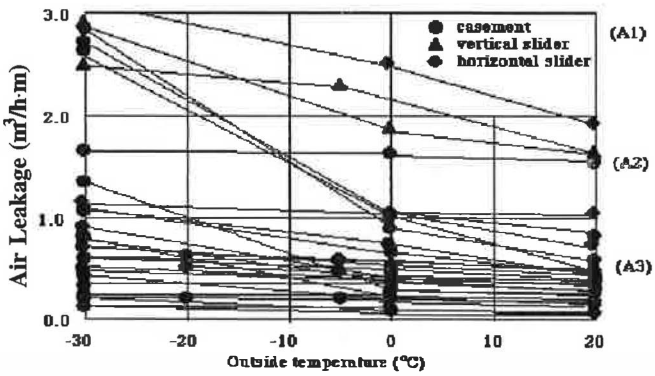

Figure 2 *Measured window air infiltration at low temperatures (m^3/h·m).*

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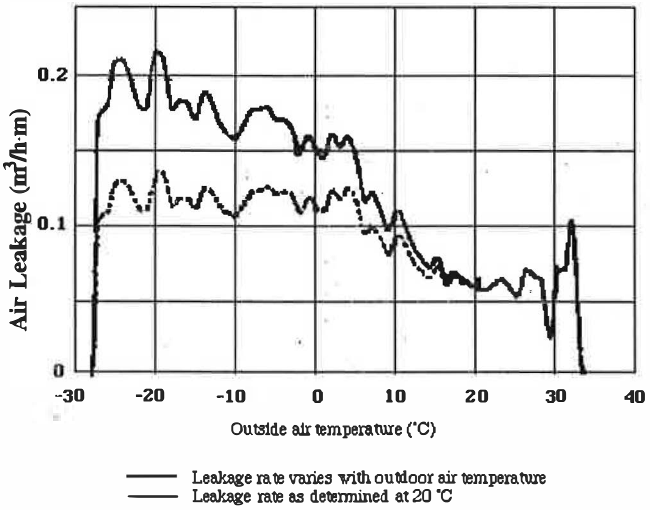

Figure 3 *Window air infiltration as a function of outside air temperature ($m^3/h{\cdot}m$) (window A-I: 18 in Montreal).*

4693 heating hours to calculate a new annual infiltration rate. Air leakage based on crack length is converted to total window values by using the crack length for each window type as defined by the specified sizes of the A440.2 standard. Comparison with infiltration rates based on 20°C measurement is shown in the second to last column of Table 1.

Finally, the impact on the ER number is calculated by substitution into the ER equation.The impact is listed for each window in the last column of Table 1. Note that, with the exception of A-I: 22 and A-I: 23, all windows decrease ER number by less than 1.0. Many decrease by as little as 0.1. This amount is not significant when comparing products. The exceptions were one window that actually slightly decreased leakage at lower temperatures and the other a tilt-in vertical slider that did not meet minimum CSA air-leakage requirements at any temperature.

Annual energy ratings currently do not take into account performance changes as a result of use or aging. In the same way, investigations described in this paper have not considered changes in the window product as a result of temperature cycling, as would occur from day to night in winter or season to season. Previous studies (NRCan 1991; NRCan 1993) covered pressure and motion cycling, and it is expected that temperature cycling would have no greater impact.

CONCLUSIONS

The results of this investigation fully support the use of the existing window ER system, and additional low-temperature tests are not warranted. Thirty-five samples of windows representative of a broad range of operable types, frame/sash materials, design, and hardware were tested for air leakage at the usual temperature of 20°C and at various temperatures down to −30°C. The results may be summarized as follows:

1. The ASTM E-1424-91 standard proved effective to evaluate operable window performance at low temperatures. It could be used to get more information about window air leakage characteristics for annual energy performance estimation.
2. Two-thirds of the windows met the highest level (A3) of the CSA A440 standard for air leakage. Of those that did not, most fell into the next category, while three did not even meet minimum requirements. Worst were vertical sliders, PVC, and aluminum.
3. At lower temperatures, most windows exhibited increased air leakage, although many only very slightly. At the lowest temperatures, nearly half remained in the same category, while others increased more dramatically, and four did not meet minimum CSA levels. Changes seem to be affected by window design more than materials, and there seems no justification to claims that one window material is better than others at low temperatures.
4. An estimate of the impact of increased air leakage at low temperatures on annual energy performance was carried out. For the vast majority of windows, the ER number was reduced by less than one. Anomalies for two windows that also were very poor performers at normal temperatures could be explained by design details.

The air leakage term in the ER calculation is normally small but needs to be retained, however, in order to account for poor performance of the few very leaky windows.

Durability issues relating to energy performance were not investigated, but other studies have suggested similar increases in air leakage as a result of pressure and motion cycling.

Further work may be warranted to investigate design details that affect air leakage and changes with temperature so that products can be improved. Nevertheless, most products in today's marketplace are excellent, and air leakage does not appear to be reducing energy efficiency significantly at any temperature. Specifiers and consumers, however, would be well advised to demand tested products to avoid getting the odd "lemon" in the marketplace.

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REFERENCES

ASTM E1424. 1991. Standard test method for determining the rate of air leakage through exterior windows, curtain walls, and doors under specified pressure and temperature differences across the specimen.

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CSA. 1993. CSA A440.2-93, Energy performance evaluation of windows and sliding glass doors, Toronto: Canadian Standards Association.

CSA. 1996. Additional Air-Ins data distributed to CSA A440.2 sub-committee, Canadian Standards Association, January.

Elmahdy, A.H. 1995. Air leakage characteristics of windows subjected to simultaneous temperature and pressure differentials. In *Proc. Window Innovations '95, Toronto,* June, pp. 146-163.

Henry, R.P. 1995. Canada's energy rating systems for windows and doors now implemented in a National Energy Code for Houses. *Proceedings, Window Innovations '95, Toronto,* June, pp. 231-241.

Kehrli, D.W. 1985. Window air leakage performance as a function of differential temperatures and accelerated environmental aging. In *Thermal performance of exterior envelopes of building III*, pp. 872-890. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

NRCan. 1991. A study of long-term performance of operating and fixed windows subjected to pressure cycling. CANMET Report, Natural Resources Canada, December.

NRCan. 1993. Long-term performance of operating windows subjected to motion cycling. CANMET Report, Natural Resources Canada, March.

NRCan. 1995. Air Infiltration rate of windows under temperature and pressure differentials. CANMET Report, Natural Resources Canada, October.

NRCan. 1996. CANMET Contract 23240-95-1210 with Enermodal Engineering, presentation to CSA A440.2 sub-committee, March.