BNL-66610
Informal Report

**HOW TO HEAT AND COOL A HOME WITH 400 CFM SUPPLY AIR
AND KEEP THE DUCTS IN THE CONDITIONED SPACE**

J.W. ANDREWS

MAY 1999

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Prepared for:
Office of Building Technologies
State and Community Programs
U.S. Department of Energy
Washington, DC 20585

Under Contract No. DE-AC02-98CH10886

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ABSTRACT

A design strategy is presented that can enable a typical new home to be heated, cooled, and ventilated with less than 400 cfm of delivered air. The strategy has three major elements. First, peak cooling loads are minimized by using good available technologies for the envelope, with emphasis on minimizing heat gains through the windows. Second, the envelope is designed to have very low natural air leakage rates, such that all the ventilation air can be drawn in at one point and passed over the cooling coil before it is mixed with the house air. This permits a significant portion of the cooling load to be met at an air flow rate of ~200 cubic feet per minute (cfm) per ton, compared with the typical 400 cfm per ton in standard air-conditioning systems. Third, by reducing the amount of supply air needed to meet the envelope loads, the required size of ductwork is reduced, making it easier to locate the ducts within the conditioned space. This reduces duct loads to zero, completing the three-part energy conserving strategy.

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INTRODUCTION

Much has been written recently about the significance of energy losses in residential ductwork. A consensus has developed that for typical duct systems in attics or crawl spaces, 25% to 40% of the heat or cooling energy provided to the duct system by the equipment is lost.

An obvious strategy to eliminate these losses is to design the system such that all the ductwork is inside the conditioned space. If this is done, any losses are to spaces where the heat or cooling is useful, as long as these losses are not so great that the system cannot be balanced. However, builders have been reluctant to choose this option because most ductwork is ugly and hard to hide. It would be easier to incorporate ducts within the living space if they were smaller than is usually the case today. This report discusses an integrated approach to minimizing the size of ductwork in new housing.

In pursuing this objective, it should be noted that the size of the ducts is influenced by the air flow (cfm) requirement, and this in turn is driven by the cooling load, even in northern climates. A typical house, whether in Florida or New York, tends to be equipped with a 3-ton air conditioner, and at 400 cfm/ton, this requires a 1200 cfm air flow rate. On the heating side, even in a 5000 or 6000 heating degree-day climate, the peak heating load of a reasonably well constructed modern home is < 30,000 Btu/h. A furnace can supply this much heat within the constraint of 400 cfm air flow and 68 °F temperature rise. Of course, typical furnaces have much higher output rates than 30,000 Btu/h. This is in part because of a historical tendency to oversize furnaces and in part to coordinate with the cfm rates required for cooling. But the fact remains that the need for ~1200 cfm in conventional installations is determined by the cooling load, not the heating load. (The situation is somewhat more complicated with heat pumps in cold climates, but the above argument is valid for furnaces in all but the coldest climates and for heat pumps in moderate to warm climates. These cover the vast majority of installations today.)

Therefore, a promising way to reduce the required size of the ductwork is to reduce the peak cooling load. In new housing, all elements of the building envelope contribute to the peak load, but a very important component is the design, placement, and shading of windows.

A second and complementary approach, which is featured in this article, is to design the house for a very low natural infiltration rate, 0.15 air change per hour (ACH) or less, and then provide ventilation air via the duct system. The ventilation air, which in conventional housing would come in through all the cracks in the envelope, at rates determined by wind and temperature gradients, is now pulled in at one location. The ventilation air is passed over the cooling coil before it is mixed with the house air. This permits a large amount of cooling to be done per cfm of air flow. Instead of the usual 400 cfm per ton, cooling can be done on this ventilation air at ~200 cfm per ton. Admittedly, the amount of such ventilation air is limited by the required ventilation rate, which we will take here to be 0.35 ACH, but still it can provide a significant reduction in the air flow rate required for cooling, which when combined with other steps to reduce the peak cooling

load, can reduce the required air flow to cool and heat a typical new house to 400 cfm even in a hot humid climate.¹

We will now show the calculations required to demonstrate the feasibility of heating and cooling a typical new house with 400 cfm of supply air. Two locations, one in Florida and one in the northeastern U.S., will be considered. Manual J, published by the Air-Conditioning Contractors of America (ACCA 1986), will be used for the peak-load computations. This will lend credibility to the calculations by ensuring that only available state-of-the-art components of the building envelope are specified. This article expands somewhat on an earlier publication by the author (Andrews 1992).

DESIGN STRATEGY FOR ORLANDO, FLORIDA

An ACCA Manual J (ACCA-J) calculation begins with the selection of a city for which the loads are to be determined. Table 1 of ACCA-J provides the following data for Orlando, Florida:

- Cooling: 2 1/2% design dry bulb 93 °F; coincident wet bulb 76 °F; moderate daily range; grains difference 44 (50% RH, 75°F indoors).
- Heating: 97 1/2% design dry bulb 38 °F.

The house is specified as a typical size and construction for Florida: 35 ft X 55 ft (1925 ft²) single-story slab-on-grade. The conditioned volume, based on 8-ft ceilings, is 15 400 ft³. The envelope components and their references in the ACCA-J cooling load tables are as follows:

- Opaque walls: Either: 1) R-30 Wood Frame (12N in ACCA-J Table 4); or 2) 4" Brick + 8" Masonry Block + R-19 (14H in ACCA-J).
- Windows: Double-Pane Heat Absorbing Glass with Low-e Coating (ACCA-J Table 3D). 16% of wall area, preferentially distributed on north and south sides of the house.
- Doors: Urethane Core Metal (11E in ACCA-J Table 4). Number of doors = 3.
- Ceiling: R-26 Batts, 2" X 8" Rafters (18E in ACCA-J Table 4).
- Floor: Concrete Slab on Grade (22 in ACCA-J Table 4).

The remaining components of the load are infiltration, internal gains, and duct gains (cooling) or losses (heating). For infiltration, we specify that the envelope be constructed sufficiently tight that a forced ventilation system, with air intake at a single point and outflow through cracks, will place the entire interior of the envelope under positive pressure with respect to outside, and thus reduce infiltration effectively to zero.

¹ We note in passing two other approaches that may have merit under some situations. One is to design for more air flow through a given cross section of duct than is customary. This permits the ducts to be downsized without reducing the cfm requirement. Another tactic that is commonly used with this approach is to design for lower-than-customary air temperatures leaving the cooling coil. These design options could be combined with the approach advocated above to further reduce the required duct cross sections, but we will not assume their use in anything that follows.

A forced ventilation rate of 0.35 ACH, as specified above, leads to a ventilation air flow rate of 90 cfm for the 1925 ft² house. If the natural and forced ventilation rates are added in quadrature, per ANSI/ASHRAE Standard 136-1993, p. 5, a house with a natural air infiltration rate of 0.10 to 0.15 ACH, when fitted with forced ventilation at 0.35 ACH, will have a total effective air change rate of 0.36 to 0.38 ACH. This means that the role of infiltration through cracks is reduced to a negligible 0.01 to 0.03 ACH, which can be ignored, and all of the air coming into the house can be assumed to enter via the ventilation route.²

Alternatively, if the infiltration impacts of the ventilation system are treated as an excess of return duct leakage over supply duct leakage, then for the assumptions above, namely 0.35 ACH forced ventilation and 0.10 to 0.15 ACH natural infiltration in the absence of ventilation, the calculations in ASHRAE Standard 152P (ASHRAE 1999) yield a residual infiltration rate of exactly zero. This is, no doubt, an approximation, but it is consistent with negligible infiltration outside of the ventilation system.

Cooling Load in Orlando

The sensible heat gains for the above envelope components are calculated using ACCA-J. Table 1 shows these calculations. Only the whole-house values are shown; division into individual rooms, which would be needed for duct layout and design, will not be of concern here since we are only interested in seeing to what extent the air flow requirement for the whole house can be reduced. The heat transfer multipliers (HTMs) are based on a 20 °F temperature difference.

The sensible and latent gains due to ventilation are derived from the temperature and grains moisture difference between the inside and the outside under peak conditions:

- Sensible: $1.1 \times 90 \text{ cfm} \times (93 - 75) = 1780 \text{ Btu/h}$
- Latent: $0.68 \times 90 \text{ cfm} \times 44 \text{ grains difference} = 2690 \text{ Btu/h}$

Internal gains are taken to be the same as in the ACCA-J example problem. Sensible internal gains are six people @ 300 Btu/h + appliances @ 1200 Btu/h = 3000 Btu/h. Latent internal gains are those due to people, again six @ 230 Btu/h = 1380 Btu/h.

Ducts are assumed to be located entirely within the conditioned space. Duct gains are therefore set equal to zero.

The sensible cooling load is just under 11 000 Btu/h; this is quite low by normal standards but is achieved by reining in conductive heat gain through the walls, gains through the windows, and infiltration/ventilation gains.

²This ignores the influence of exhaust fans. With the house under a constant positive pressure, the need for such fans should be reduced. If exhaust fans must be used, they should be designed for intermittent operation on occupant command, e.g., don't hard-wire bathroom fan to the light switch.

Table 1. ACCA-J Load Calculations for Orlando

1 Name of Room			OPTIMIZED HOUSE			Entire House		
2 Running ft Exposed Wall			ORLANDO, FL			180		
3 Room Dimensions ft			See Note 1			35 X 55		
4 Ceiling Ht. Ft.						8		
TYPE OF EXPOSURE			Cons.	HTM		Area or	Btuh	
			No.	Htg.	Clg.	Length	Htg.	Clg.
5	Gross	a	12n or 14h			1440		
	Exposed	b				0		
	Walls &	c				0		
	Partitions	d				0		
6	Windows	a	5	12.6		232	2923	
	& Glass	b				0	0	
	Doors	c				0	0	
	Htg.	d				0	0	
7	Windows	North	3d		8	88		704
	& Glass	E&W	tinted		13	56		728
	Doors	South	heat abs.		9	88		792
	Clg.		glass			0		0
8	Other Doors		11	6.7	4.5	54	362	243
9	Net	a	12n or 14h	1.6	0.8	1154	1846	923
	Exposed	b				0	0	0
	Walls &	c				0	0	0
	Partitions	d				0	0	0
10	Ceilings	a	18	1.4	1.4	1925	2695	2695
11	Floors	a	22a	28.3	0	180	5094	0
12	Ventilation (Note 2)						3465	1980
13	Sub Total Btuh Loss = 6 + 8 + 9 + 10 + 11 + 12						16385	
14	Duct Btuh Loss		0	%			0	
15	Total Btuh Loss = 13 + 14						16385	
16	People @ 300 & Appliances 1200							3000
17	Sensible Btuh Gain = 7 + 8 + 9 + 10 + 11 + 12 + 16							11065
18	Duct Btuh Gain		0	%				0
19	Tot. Sens. Gain = 17 + 18							11065
Note 1: Summer Temp. Diff. = 20 F, Moderate Swing, 44 grains difference 50% RH; Winter Temp. Diff. = 35 F								
Note 2: Forced Ventilation @ 0.35 ACH : 90 cfm; positive pressure cancels nearly all infiltration.								

The latent cooling load is the sum of the latent ventilation load and the latent internal loads from the occupants, $2690 \text{ Btu/h} + 1380 \text{ Btu/h} = 4070 \text{ Btu/h}$. The total cooling load is then very nearly $15\,000 \text{ Btu/h}$, or 1.25 tons, which could be met by an air flow of 500 cfm at a nominal 400 cfm per ton.

The nominal air flow is rationalized as follows. Assume 75°F indoor air at 50% RH, or 66 grains water vapor per pound dry air. Assume 55°F air leaving the coil. This produces a sensible cooling rate of $1.1 \times 400 \text{ cfm} \times 20^\circ\text{F} = 8800 \text{ Btu/h}$. The latent cooling rate (per ton of total cooling) must then be $12\,000 - 8800 = 3200 \text{ Btu/h}$. This is consistent with a moisture reduction of 12 grains ($0.68 \times 400 \times 12 \approx 3200$), and when plotted on a psychrometric chart this leads to a 48°F coil temperature and a 25% bypass factor.

Applying the same coil temperature and bypass to the ventilation air entering at 93°F wb and 76°F db leads to air leaving the coil at 59°F db and 56.5°F wb, with a temperature difference between the two of $93 - 59 = 34^\circ\text{F}$ and a moisture difference of $108 - 64 = 48$ grains. Applying these differences to 90 cfm of ventilation air yields sensible and latent cooling rates on this air of:

- Sensible: $1.1 \times 90 \text{ cfm} \times 34^\circ\text{F} = 3370 \text{ Btu/h}$
- Latent: $0.68 \times 90 \text{ cfm} \times 48 \text{ gr} = 2940 \text{ Btu/h}$

The total cooling of $\sim 6300 \text{ Btu/h}$ delivered in 90 cfm of air works out to $\sim 170 \text{ cfm/ton}$ for this air.

This is an approximate treatment, and in the detail design of the equipment its specific characteristics will need to be considered. The results in any case, however, should be close to what we are discussing here.

The impact on the whole cooling job is as follows. Subtracting the 3370 Btu/h sensible cooling in the ventilation air stream from the overall sensible cooling load of $10\,870 \text{ Btu/h}$ yields a residual sensible load of 7500 Btu/h . Subtracting the 2940 Btu/h latent cooling in the ventilation air stream from the overall latent cooling load of 4070 Btu/h yields a residual latent load of 1130 Btu/h . The total load to be met other than via the ventilation air is then $7500 + 1130 = 8630 \text{ Btu/h}$, which at 400 cfm per ton will require 290 cfm.

The upshot, then, is that 90 cfm of ventilation air plus 290 cfm of house air passed over the coil will be sufficient to perform the cooling job at peak load. This meets the goal of conditioning this house with less than 400 cfm of air into the duct system.

Heating Load in Orlando

We have stated that heating is a slack variable in the calculation, i.e., that once we have provided sufficient air flow to do the cooling load, then heating the house will be no problem. Let us see whether this is so in our example case.

In Table 1 the heating-load calculations per ACCA-J are shown in the appropriate column, using a design winter temperature difference of 35 F. To be conservative, we have assumed that the slab has no edge insulation; this results in about one-third of the heating load being caused by slab losses. The total envelope loss at design conditions is 16 385 Btu/h, with the slab contributing ~5000 Btu/h and the ceiling, walls, windows, and ventilation each contributing ~2000 to ~3000 Btu/h. It should be emphasized that we are not recommending the omission of edge insulation; rather, we wish to show that the strategy recommended here will work whether or not edge insulation is provided.

If the furnace is selected to operate at 400 cfm with a delivered air temperature of 130 F, the output rate would be $1.1 \times 400 \times (130 - 70) = 26\,400$ Btu/h, which is 60% oversized relative to the load, which is slightly more oversizing than the ASHRAE recommended 40% but not outside reasonable bounds. A somewhat smaller furnace could be selected; that would just reduce the output temperature slightly. The minimum output temperature T_{out} capable of meeting the load can be obtained from the equation $16385 = 1.1 \times 400 \times (T_{out} - 70)$, which yields $T_{out} = 107$ °F. This is within the range of temperatures that a heat pump might be expected to deliver.

EQUIPMENT SELECTION

Detail design of the heating and cooling plant is beyond the scope of this paper, but it is appropriate here to sketch out the kind of equipment that might be used. A conceptual design of an air-handler unit incorporating the indoor coil of a two-speed heat pump is shown in Figure 1. (Alternatively, the element labeled "coil" could be a combination refrigerant-to-air/water-to-air coil mated to a two-speed air conditioner for cooling and a water heater for heating.)

The air-handler unit consists of a box with cutouts at one end to receive duct connections for return house air and ventilation air, and at the other end to receive a duct connection for supply air. The coil is centrally located in the box, as shown. A two-speed system fan is located downstream of the coil. The portion of the box upstream of the coil is partitioned into two sections as shown, with a V-shaped damper having two settings. On the damper's ventilation-only setting, the upper vane blocks off air flow from the return duct and the lower vane is tucked out of the way, to allow ventilation air to flow over the entire coil. On the damper's circulation+ventilation setting, the upper vane is in a horizontal position, allowing return air to flow to the coil, while the lower vane blocks off the opening between the upper and lower portions of the inlet side of the box, so that the ventilation air is sequestered from the return air until both streams have passed over the coil. As discussed above, this separation is necessary in the cooling to achieve a high degree of dehumidification of the moist outside air used for ventilation.

Figure 1. Conceptual Design of Air Handler

The system would have six operating modes covering heating, cooling, and ventilation, and standby:

- Cooling mode, circulation + ventilation: used when the cooling load is greater than what can be accommodated using ventilation air only.
- Cooling mode, ventilation only: used when the cooling load is greater than zero and less than the maximum that can be accommodated using ventilation air only.
- Heating mode, ventilation only: used when the heating load is greater than zero and less than the maximum that can be accommodated using ventilation air only.
- Heating mode, circulation + ventilation: used when the heating load is greater than what can be accommodated using ventilation air only.
- Ventilation without heat or cool: ventilation air is continuously introduced into the house, without heating or cooling. Used on spring and fall days when there is no call for heat or cooling, and the homeowner prefers to operate the house with closed windows.
- Standby mode: system fan is off. Used on spring and fall days when the homeowner prefers to have natural ventilation via open windows.

The system would switch among the first five modes automatically. The last mode would be selected manually when desired.

SYSTEM DESIGN IN A NORTHERN CLIMATE

It will be instructive to see how the above design might change when we move to a cooler climate, such as that in the metropolitan New York City area. Intuitively one would expect the cooling load to be significantly less than in Florida, and on an annual basis this certainly is true. However, as we will see, the peak cooling load in New York, although lower than in Florida, is not as much different from the more southerly location as one might expect. The peak heating load for a similarly constructed house is significantly greater, and we will compensate for this by paying more attention to the design of the foundation. Recall that in Orlando we assumed that the house was placed on an uninsulated slab. For New York, in line with common building practice, we should use either a basement or a crawl space. To simplify matters, and as a conservative assumption, we'll select the crawl space.

But first things first. The New York climate is again specified according to Table 1 of ACCA-J:

- Cooling: 2 1/2% design dry bulb 89 °F; coincident wet bulb 73 °F; moderate daily range; grains difference 33 (50% RH, 75°F indoors).
- Heating: 97 1/2% design dry bulb 15 °F.

The house is specified the same as the one in Florida, except that the slab is replaced by a crawl space with R-30 insulation (element 20e in ACCA-J Table 4 for cooling and Table 2 for heating). The HTM's are then selected using a design temperature difference of 15 °F in the cooling mode and 55 °F in the heating mode.

Cooling Load in New York

As in the previous case, the sensible heat gains for the envelope components listed in the Orlando discussion are calculated using the ACCA-J methodology. Table 2 shows these calculations. As before, only the whole-house values are shown. The heat transfer multipliers (HTMs) are based on a 15 °F temperature difference.

The sensible and latent gains due to ventilation are derived from the temperature and grains moisture difference between the inside and the outside under peak conditions:

- Sensible: $1.1 \times 90 \text{ cfm} \times (89 - 75) = 1390 \text{ Btu/h}$
- Latent: $0.68 \times 90 \text{ cfm} \times 33 \text{ grains difference} = 2020 \text{ Btu/h}$

Internal gains are taken to be the same as in the ACCA-J example problem, and hence are unchanged from the Orlando case. Sensible internal gains are six people @ 300 Btu/h + appliances @ 1200 Btu/h = 3000 Btu/h. Latent internal gains are those due to people, again six @ 230 Btu/h = 1380 Btu/h.

Also as before, ducts are assumed to be located entirely within the conditioned space. Duct gains are therefore set equal to zero.

The sensible cooling load is 10 340 Btu/h, which only 6% less than in Orlando. The design temperature difference is 25% less than in Orlando, but other factors help to maintain the design cooling load. Chief among these are 1) the window loads depend on solar gains, which do not scale with design temperature difference; 2) the internal gains do not change; and 3) we have "traded in" the uninsulated slab for an insulated crawl space, which reduces the heating load significantly but adds somewhat to the cooling load.

The latent cooling load is again the sum of the latent ventilation load and the latent internal loads from the occupants, $2020 \text{ Btu/h} + 1380 \text{ Btu/h} = 3400 \text{ Btu/h}$, which is 16% less than in Orlando.

The total cooling load is then very nearly 13 740 Btu/h, 8% less than in Orlando. It would appear that the same cooling plant that was used in Orlando would work well here, with the proviso that the required sensible heat ratio will be somewhat higher.

Table 2. ACCA- J Load Calculations for New York City

1 Name of Room			OPTIMIZED HOUSE			Entire House		
2 Running ft Exposed Wall			NEW YORK, NEW YORK			180		
3 Room Dimensions ft			See Note 1			35 X 55		
4 Ceiling Ht. Ft.			Directions Room Faces			8		
TYPE OF EXPOSURE			Cons.	HTM		Area or	Btuh	
			No.	Htg.	Clg.	Length	Htg.	Clg.
5	Gross	a	12n or 14h			1440		
	Exposed	b				0		
	Walls &	c				0		
	Partitions	d				0		
6	Windows	a	5	19.8		232	4594	
	& Glass	b				0	0	
	Doors	c				0	0	
	Htg.	d				0	0	
7	Windows	North	3d		7	88		616
	& Glass	E&W	tinted		12	56		672
	Doors	South	heat abs.		8	88		704
	Clg.		glass			0		0
8	Other Doors		11	10.5	3.5	54	567	189
9	Net	a	12n	1.8	0.6	1154	2077	692
	Exposed	b				0	0	0
	Walls &	c				0	0	0
	Partitions	d				0	0	0
10	Ceilings	a	18	2.2	1.2	1925	4235	2310
11	Floors	a	20	2	0.4	1925	3850	770
12	Ventilation (Note 2)						5450	1390
13	Sub Total Btuh Loss = 6 + 8 + 9 + 10 + 11 + 12						20773	
14	Duct Btuh Loss		0	%			0	
15	Total Btuh Loss = 13+14						20773	
16	People @ 300 & Appliances 1200							3000
17	Sensible Btuh Gain = 7 + 8+ 9+ 10+ 11+ 12+ 16							10343
18	Duct Btuh Gain		0	%				0
19	Total Sensible Gain = 17 + 18							10343
Note 1: Summer Temp. Diff. = 15 F, Moderate Swing, 33 grains difference 50% RH; Winter Temp. Diff. = 55 F								
Note 2: Forced Ventilation @ 0.35 ACH : 90 cfm; positive pressure cancels nearly all infiltration.								

Applying the same cooling coil data as before (coil temperature and bypass factor) to the ventilation air entering at 89 °F wb and 73 °F db leads to air leaving the coil at 58 °F db and 55.5 °F wb, with a temperature difference between the two of $89 - 58 = 31$ °F and a moisture difference of $97 - 62 = 35$ grains. Applying these differences to 90 cfm of ventilation air yields sensible and latent cooling rates on this air of:

- Sensible: $1.1 \times 90 \text{ cfm} \times 31 \text{ °F} = 3070 \text{ Btu/h}$
- Latent: $0.68 \times 90 \text{ cfm} \times 35 \text{ gr} = 2140 \text{ Btu/h}$

The total cooling of ~5200 Btu/h delivered in 90 cfm of air works out to ~210 cfm/ ton for this air.

The impact on the whole cooling job is as follows. Subtracting the 3070 Btu/h sensible from the overall sensible cooling load of 10 340 Btu/h yields a residual sensible load of 7270 Btu/h. Subtracting the 2140 Btu/h latent from the overall latent cooling load of 3400 Btu/h yields a residual latent load of 1260 Btu/h. The total load to be met other than via the ventilation air is then $7270 + 1260 = 8530 \text{ Btu/h}$, which at 400 cfm per ton will require just under 290 cfm.

The upshot, then, is the same as in Orlando: 90 cfm of ventilation air plus 290 cfm of house air passed over the coil will be sufficient to perform the cooling job. This again meets the goal of conditioning this house with less than 400 cfm of air into the duct system.

Heating Load in New York

We found that heating is indeed a slack variable in Orlando. Does it remain so in New York? In Table 2 the heating-load calculations per ACCA-J are shown in the appropriate column, using a design winter temperature difference of 55 F. Again we point out the substitution of the crawl space for the slab. The total envelope loss at design conditions is 20 770 Btu/h, with the foundation, ceiling, windows, and ventilation each contributing in the ~4000 to ~5500 Btu/h range, with walls adding somewhat less than this.

If the furnace is selected to operate at 400 cfm with a delivered air temperature of 130 F, as in the previous example, the output rate would be $1.1 \times 400 \times (130 - 70) = 26,400 \text{ Btu/h}$, which is 27% oversized relative to the load. A furnace sized to the ASHRAE-recommended 1.4 times the heating load would need to have an output of 29 000 Btu/h and a delivered air temperature of 136 °F, which is reasonable.

As in Orlando, the integrated system option for heating (based on an efficient hot water heater) could also be chosen.

Thus, the New York house, like the one in Orlando, can be heated and cooled with 400 cfm of supply air. Moreover, contrary to what intuition might lead one to believe, the sizing of the equipment would not be very different in the two locations.

THE DANISH HOUSE AT BROOKHAVEN NATIONAL LABORATORY

In 1985 a Danish factory-built house was given to Brookhaven National Laboratory by the Danish Housing Ministry as part of a cooperative research project. The house was erected on the laboratory site and monitored over part of a heating season. The approach to system integration advocated in this paper is based on experience gained with this house.

The house is an L-shaped, one-story 1500 ft² structure with an unfinished attic. The envelope characteristics are similar to those specified above, with the exception that the windows, while advanced when compared with the average existing U.S. housing stock, are not designed to the same standard of cooling-load reduction as the ones called for here. (The Danes, after all, live in a part of the world where air conditioning is not needed.) The natural air infiltration rate was measured to be 0.1 ACH. The normalized heat-loss rate of the house, after accounting for internal gains (Loss et al. 1986) was calculated to be 2.1 Btu/ft²-°F-day on the basis of measured data. For comparison, the above-specified house in New York would have a normalized heat-loss rate of 4.0 Btu/ft²-°F-day.³ It would thus appear that our specifications are not unrealistically stringent when compared with an energy-efficient production model house.

CONCLUSIONS

It has been shown that a new home of typical size (1925 ft²) can be heated and cooled, even under peak-load conditions, in either a northern or southern climate using 400 cfm of supply air. The design strategy has three main elements:

- Use efficient envelope components on the market today, as indicated by their inclusion in ACCA Manual J.
- Design the envelope to have a low natural air change rate (0.10 to 0.15 ACH) and combine this with forced ventilation at 0.35 ACH, with cooling of this ventilation air to take place before it is mixed with the house air.
- Take advantage of the reduced air volume requirements to place the ductwork--whose required cross section is now much reduced--within the conditioned space.

Although detailed design of the heating and cooling plant is beyond the scope of this article, a conceptual design for an indoor air handler unit is presented which, when mated to a two-speed heating/cooling unit, could provide the air-sourcing, flow, and heating/cooling capacity characteristics required for the system.

³This value is obtained by subtracting sensible internal gains of 3000 Btu/h from the ACCA-J load of 20 773 Btu/h, dividing the result by the 55 °F temperature difference, multiplying by 24 hours per day, and dividing that by the conditioned area of the house, 1925 ft².

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