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ABSTRACT

General building energy simulation programs do not typically simulate interactions between HVAC systems and building envelopes. For this reason, a simulation tool which includes interactions between duct systems, building envelopes, and heating and cooling appliances in residential buildings has been developed. The simulation tool uses DOE-2 (to model space conditioning loads and zone temperatures), COMIS (an airflow network solver), and a model of duct leakage and conduction losses. Three augmentations to the simulation tool are presented herein, along with sample results. These include: 1) modeling of steady-state thermosiphon flows in ducts, 2) improved simplified modeling of duct thermal mass effects, and 3) modeling of multi-speed space conditioning equipment. Multi-speed air conditioners are shown to be more sensitive to duct efficiency than single-speed equipment, because their efficiency decreases with increasing cooling load. Thermosiphon flows through duct systems are estimated to be 5-16% of the total heating load, depending on the duct insulation level. The duct dynamics model indicates that duct thermal mass decreases the energy delivery efficiency of the distribution system by 1-6 percentage points.

NOMENCLATURE

A .... is a constant used in determining the duct time constant

Bi .... is the duct Biot number, hl/k

$E_{mass\text{-}lost}$ .... is the energy not delivered through the register because of cycling of the duct thermal mass [J]

$E_{delivered}$ .... is the energy delivered through the supply registers [J]

E_{duct} .... is the thermal storage of the duct system [J]

h .... is a convection coefficient [W/m^{2}K]

l .... is the thickness of the duct [m]

L .... is the length of the duct [m]

k .... is the thermal conductivity of the duct material

x .... is the distance from the duct inlet [m]

m(x) .... is the duct flow [m^3/s]

d .... is the diameter of the duct [m]

C_p .... is the specific heat of air [J/kg K]

U .... is the conductance of the duct wall [W/m^{2}K]

V .... is the bulk velocity in the duct [m/s]

α .... is the thermal diffusivity of the duct material [m^2/s]

ρ .... is the density of air [kg/m^3]

$\tau(x)$ .... is $T_{duct\text{-}air}(x) - T_{surroundings}$ [K]

τ_{duct} .... is the time constant of the duct material [s]

$\tau_{heat\text{-}exchanger}$ ... is the time constant of the furnace/ac heat exchanger [s]

$\tau_{transit}$ .... is the average time a slug of air is in the duct system [s]

τ_1 .... is the time constant of the duct system thermal mass while the fan and furnace/ac are both on [s]

τ_2 .... is the time constant of the duct system thermal mass during fan overrun [s]

τ_3 .... is the time constant of the duct system thermal mass while the fan is off [s]

INTRODUCTION

General-purpose building energy simulation programs are generally not designed to provide meaningful analyses of the detailed interactions between HVAC systems and building envelopes. For this reason, simulation tools designed to take into account the interactions between the duct system, the building envelope, and the heating and cooling appliances in residential buildings have been under development by several groups over the past decade (Jakob et al. 1986, Locklin et al. 1987, Modera and Jansky 1992, Parker et al. 1993). Over time, the level of detail of these simulation tools has evolved, both to allow for better understanding of unmodelled effects and/or interactions, and to help explain results obtained in the field. The simulation tool used by the authors is based upon the DOE-2 thermal simulation code (Birdsall et al. 1990), the COMIS airflow network code (Feustel and Raynor-Hoosen 1990), and a duct performance simulation model. The duct-system model, in addition to determining the combined impacts of leakage and conduction on duct performance, serves as the interface between COMIS and DOE-2 (Modera and Jansky 1992). The basic methodology is to use COMIS to compute air flows through the duct system and the building simultaneously, with and without the distribution fan and heating/cooling equipment in operation. These flows are then passed through DUCTSIM, which calculates the duct system efficiency and the fractional on-time of the system that are incorporated into the file that is passed to DOE-2. DOE-2 is then used to calculate loads in conditioned zones and temperatures in unconditioned zones. The process is iterated three times. This procedure was found to produce results within 1% of the asymptotic value.

This paper briefly describes three augmentations of the Modera and Jansky (1992) simulation code that were implemented over the past 2 years, as well as some sample simulation results that were obtained with the modified code.The three augmentations are: a simple model for the dynamic behavior of the duct system, a simple model for predicting thermosiphon-driven air flow through the duct system, and the addition of the capability to model the interactions between ductwork and two-speed conditioning equipment.

DUCT-SYSTEM THERMAL MASS

Theory

Over the past fifteen years, various researchers have addressed the issue of modelling the dynamic performance of residential duct systems. However, the techniques investigated were either not flexible enough to treat the different cases of interest to us (e.g. leaky duct systems) (Grot and Harrje 1981), or too computationally intensive for our purposes (Parker et al. 1993). The basic idea behind the model described herein was to provide a simplified technique for simulating the impacts of duct thermal mass on the energy delivered at the registers over complete cycles of the heating or cooling equipment. The philosophy was to calculate the steady-state solutions, including all leakage and conduction effects, and then to develop reasonable functional relationships to characterize the system-on and system-off endpoints. To accomplish this, the dynamic performance of the duct system is characterized by three time constants, one for the period in which the furnace/ac and the fan are on (τ_1), one for the period when only the fan is on (τ_2), and one for the period when both the fan and furnace/ac are off (τ_3). A schematic representation of this process is shown in Figure 1.

The time constants τ_1,τ_2, and τ_3, are determined for any given duct system from functional combinations of the three characteristic times for the HVAC system, τ_{duct}, $\tau_{transit}$, and $\tau_{heat\text{-}exchanger}$. The first of these characteristic times is related to the thermal mass of the duct system, and is based upon the solution for an infinite flat plate, using the temperature outside the duct as the temperature boundary condition. The second characteristic time is related to the transit time of a slug of air through the duct system, and the third is a time constant for the air leaving the furnace heat exchanger (Grot and Harrje 1981). The functional relationship between the three system time constants (τ_1,τ_2, and τ_3) and the three characteristic times (τ_{duct}, $\tau_{transit}$, and $\tau_{heat\text{-}exchanger}$) was obtained by fitting the model to a series of unsteady finite difference simulations for different duct lengths and material thicknesses. The first two time constants are defined in Equations 1-3, and the third time constant was obtained from empirical data from Grot and Harrje (1981), as well as from in-house field test data (Modera 1993). The value used for $\tau_{heat\text{-}exchanger}$ in the results presented is 120 seconds.

$$A \cdot \tan A = Bi = \frac{hl}{k} \tag{1}$$

$$\tau_{duct} = \frac{A^2 \cdot \alpha}{l^2} \tag{2}$$

$$\tau_{transit} = \frac{L}{V} \tag{3}$$

For every hour of simulation, the number of furnace/AC cycles is determined from the fractional on-time of the equipment using a parabolic thermostat model (Parken et al. 1985), and the length of each burner-on period and burner-off period is in turn determined from the number of cycles and the fractional on-time. Referring to Figure 1, the change in the energy stored in the duct system between the beginning of period 1 and the end of period 2 is partitioned between the registers and the duct surroundings according to their effective conductances during fan operation. Any energy remaining in the system when the fan turns off is partitioned according to the ratio of the effective conductances with the fan off, where the effective conductance to the room is based upon a simulated thermosiphon flow (see below).

Application

To examine the impact of duct mass on duct efficiency, a series of simulations of an attic-only duct system installed in an Atlanta ranch house were performed. The house insulation levels are U=0.44,0.30, and 0.19 w/m^{2}K (R-13, 19, and 30 °F·ft^2·h/Btu) for the walls, floor, and ceiling insulation, respectively. These insulation levels are typical of current building standards and practices. The house has a trunk and branch duct system with a surface area (duct surface area/floor area = 27%) which is typical for new construction. Simulations were performed for different duct-wall constructions during the first week in January. In all cases, the fan control was based upon a bonnet temperature of 46 °C to turn on the fan and 32 °C to turn off the fan.The results of these simulations are summarized for furnace heating in Table 1.

The results in Table 1 provide an indication of the importance of duct thermal mass to the energy performance of the system. The $E_{mass\text{-}lost}$ column represents the net energy lost to the duct surroundings as a result of the thermal mass of the duct system for the first week in January. As expected, the sheet-metal ductwork has larger thermal storage effects as compared to low-mass plastic flexducts. However, it may seem counterintuitive that the net energy lost increases as insulation is added to plastic flexduct, whereas it decreases as insulation is added to the sheet-metal duct. The principal reason for this is the composite nature of the insulated sheet-metal ducts. For the flexduct system, the quantity of energy stored in the duct during the on-cycle is increased as the thickness and mass of the insulation is increased, whereas the fraction of the stored energy recovered does not change appreciably. For the sheet metal duct system, as for the flexducts, the addition of insulation increases the quantity of energy stored during the on-cycle. However, since most of the thermal mass of the insulated sheet metal duct is in the metal (70-73% depending on metal gauge) and the metal is on the inside of the insulation, the percentage of the thermal energy stored in the metal that reaches the conditioned space via free convection during the off-cycle is significantly increased by insulation. More succinctly, the fractional energy recovery from sheet-metal ducts is increased significantly by insulation, whereas this is not the case for plastic flexducts.

The $E_{mass\text{-}lost}/E_{delivered}$ column in Table 1 represents the ratio of the net energy lost due to cycling of the duct-system thermal mass to the total energy delivered to the conditioned space. As such, this column also represents the percentage decrease in the duct-system energy delivery efficiency due to the fact that the duct system is not massless, where the delivery efficiency is defined as the fraction of the energy imparted to the duct system by the heating or cooling equipment that is delivered to the space at the registers:

$$\eta_{del} = \frac{\left(\sum_{1}^{n_{reg}} \dot{m}_i C_P \left(T_{reg\,(i)} - T_{room}\right)\right)}{\dot{m}_{fan} C_P \left(T_{sup-plen} - T_{ret-plen}\right)} \qquad (4)$$

THERMOSIPHON FLOWS IN DUCT SYSTEMS

Theory

In the simplified model described above to account for duct-system thermal storage, any flow through the duct system while the system fan is off was assumed to simply recapture some fraction of the energy left in the duct system when the fan was turned off. However, as reported by Modera and Jansky (1992), a stable circulation of air from the house to the ducts and back to the house can be set up by temperature differentials between the house and the zones containing the ductwork. This effect is referred as the thermosiphon loss mechanism. Modera and Jansky did not develop a model for predicting thermosiphon flows in a general manner. In order to study the implications of duct insulation level and geometry on thermosiphon air and energy flows, a simple model of this phenomenon was developed.

The thermosiphon model developed is based on iterative application of the COMIS air flow network used for simulating house and duct flows during fan operation, and the same analytic solution for the duct temperature profile used for the forced convection cases. The difference is that during fan operation the duct flows are essentially constant, whereas the thermosiphon flows are quite temperature dependent. As thermosiphon flows are not dominated by a fan, they are more problematic to simulate. The assumed (or initial) temperature profile within the ducts determines the flow direction and vice-versa, making the solutions at least bistable [1].

Because COMIS is a steady-state flow solver, the thermosiphon flows determined by the model presented are the steady-state flows. To simulate the temporal development of thermosiphon flows and the evolution of duct temperatures would require an air flow model that incorporated fluid inertia effects. The present procedure is to assume that the ducts are uniformly at indoor temperature for the first COMIS flow simulation. The impact of this initial guess is that the flow direction is essentially determined by the direction of the leakage flow. Even for a duct system that has only 1% of typical leakage area, the leakage flows will nudge the solution enough to determine the flow direction. Once COMIS provides a set of flows and directions, the heat transfer equations are used to calculate a new duct temperature profile. As the combined heat transfer and leakage algorithm does not know about flow direction, and the flow directions at any bifurcation are not intuitively obvious, an automated procedure for continuously checking flow direction and leakage direction had to be developed, and the correct solution for Equation 5 has to be chosen for each flow condition. Future work will investigate the influence of the cycling of the furnace or air conditioner by initiating the simulations with the temperature profile at the end of a furnace or air-conditioner on period.

$$\frac{d\tau(x)}{\tau(x)} + \frac{\dfrac{Q_{leak-in}}{L} + \dfrac{U\pi d}{\rho C_P}}{\dot{m}(x)}dx = 0 \qquad (5)$$

Two additional points need to be raised about the solution procedure. First, the heat transfer coefficients inside the duct for free convection are not recalculated based upon the flowrates generated by the COMIS model. Because of this, the model in its present form would not be valid for ducts without insulation, as the overall thermal resistance of uninsulated ducts is too sensitive to the inside convective heat transfer coefficient. As long as the primary resistance to heat transfer is insulation, using a constant coefficient is a reasonable assumption. The second point is that the friction coefficients are also not recalculated based upon the flow-

[1] Field studies performed by the primary author have shown that the same duct system can set up different flow patterns depending on the length of a furnace cycle, the outside temperature, and the insulation level of the system.

rate. However, this assumption is reasonable for flex duct, as it is a very "rough" pipe, and friction factors for rough pipes are relatively constant at the Reynolds numbers of interest. On the other hand, this assumption would have to be checked more carefully for smoother ducts (e.g. sheet-metal, plastic).

Application

In order to examine the impact of thermosiphon flows on the overall energy implications of a duct system, simulations were performed for four different levels of duct insulation. These simulations were for the first week of January in Atlanta, GA and used the same ranch house as for the thermal mass application. The only differences from the thermal mass results are that the return duct is located in the crawlspace and the ducts have 1% of the leakage used previously. The results of the simulations for furnace heating are summarized in Table 2.

Table 2 demonstrates that duct insulation can dramatically reduce thermosiphon energy losses, and also that these losses represent a non-negligible fraction of the house's heating load. It should be noted that the thermosiphon flow as a fraction of building load is dependent on the insulation level of the envelope. The envelope simulated is consistent with current building standards, as is an R-4 duct system.

To better understand how insulation impacts thermosiphon energy flows, the air flow rates and the temperature of the air entering the house are summarized in Table 3 for the most extreme hour of the week in Atlanta. The results indicate that: 1) the thermosiphon air flows are relatively large (12-16% of fan flow) and do not decrease dramatically with insulation, and 2) the major impact of insulation is to reduce the temperature drop of the air moving through the duct system.These results are not surprising, as the flow should scale with the square-root of the temperature difference between the house and the ducts.

INTERACTIONS BETWEEN TWO-SPEED HVAC EQUIPMENT AND DUCT SYSTEMS

Theory

As earlier work has indicated that HVAC-system heating or cooling capacity can have a significant impact on the efficiency of a duct system (Modera et al. 1992), the impact of connecting a duct system to variable-capacity equipment is a question of some importance. In addition, as variable-capacity equipment has a rather different part-load efficiency curve relative to single speed equipment (i.e., much higher efficiency at lower loads), the impact of the duct system on the efficiency of the equipment is also worth examining[2].

The expected impact of variable-speed equipment on the efficiency of the ductwork is also complicated by another issue. If the fan is slowed down in proportion to the capacity of the equipment so as to maintain a constant supply-plenum enthalpy (typical for present equipment on the market), the pressure in the ductwork will be lower, and therefore the flow through duct leaks will be lower (see discussion below). On the other hand, as the flow through the duct system is reduced, the transit time of a slug of air passing through the duct is increased, thereby increasing the impact of conduction through the duct walls. From a slightly different perspective, during low-capacity low-flow operation the conduction losses through the effectively oversized duct walls represents a larger fraction of the energy being supplied by the heating or cooling equipment.

To attempt to address these issues, the simulation tool was modified to allow simulation of the operation of a two-speed air conditioner. The modifications included automating the simulation of air flows and heat transfer in both modes of operation, as well as developing algorithms for simulating the control of the two-speed unit.The key control issue for two-speed air conditioners is how they are operated when the cooling load is between the low and high cooling capacities. Two control strategies were implemented. One extreme case, which is denoted here as HIGH-CAP CYCLING, forces the unit to operate at high capacity and cycle whenever the low capacity does not meet the cooling requirements. The opposite extreme, denoted here as LOW-CAP MAX, forces the unit to be on continuously whenever the cooling load is not met at low capacity, the high speed mode coming on at most once per hour.

Application

Simulations of a two-speed air conditioner and single-speed air conditioner connected to the same duct systems were performed for Sacramento, California. The house and duct system layout were the same as in the duct thermal storage calculations. Some characteristics of the two pieces of equipment simulated are summarized in Table 4. The two duct systems to which they were connected were: 1) a perfect duct system (airtight, no conduction losses), and 2) a plastic flexduct attic-only duct system with 0.7 m^2 K/W(R-4 °F·ft^2·h/Btu) insulation and typical duct leakage (1 cm^2 duct leakage area/m^2 house floor area). The results of summer simulations (June 1 - August 31) of these two systems are summarized in Table 5.

The results in Table 5 clearly indicate that duct-system inefficiencies have a disproportionate impact on the performance of two-speed air conditioning equipment. More specifically, it is clear that the fractional energy savings associated with going from the single-speed to the two-speed unit is approximately cut in half by typical duct-system inefficiencies. The two causes of these results are isolated in the last two rows of Table 5. The delivery efficiency row illustrates the fact that low-capacity operation results in lower delivery efficiencies, the effect being sharply more pronounced for LOW-CAP MAX control. The last row of Table 5 shows the part-load efficiency curve impact, which is

[2]Many manufacturers of air conditioners produce high SEER units using multiple or variable speed compressors and fans. Presumably, they "size" the low speed of their compressor to just meet the part-load capacity requirements for the SEER test. At the low capacity, the heat exchanger is oversized and has a higher coefficient of performance (COP). Consequently, a high SEER rating is obtained for a unit which is no more efficient at its high capacity than single speed units with lower SEER ratings

even larger than the capacity impact. This row presents the ratio of the average efficiency of the air-conditioner with the typical duct system to that with a perfect duct system. It clearly illustrates that the increased load due to the ducts increases the average efficiency of the single-speed compressor, and decreases the average efficiency of the two-speed compressor. This result simply reflects the fact that increased loads due to duct losses reduce the cycling of the single-speed equipment, thereby increasing its efficiency, whereas those same increased loads cause the two-speed equipment to operate more in its high-speed lower-efficiency mode.

DISCUSSION

One common thread in two of the systems-modelling issues discussed above is the need to distinguish between duct system thermal losses due to leakage and those due to conductive heat transfer. Although these losses are commonly lumped together, it is clear that the balance between the two is an important issue. This is particularly true with respect to the appropriate choice of low-capacity fan-speed for variable capacity equipment. Reducing the fan speed reduces the fraction of the fan flow that is lost by leakage, as the duct pressure scales with the fan flow squared, and the leakage roughly scales with the pressure to the 0.65 power. The fractional leakage therefore roughly scales with the fan flow to the 0.3 power. On the other hand, reducing the fan speed increases the residence time of the air in the duct while also increasing the temperature difference relative to the surroundings, which increases conduction losses. Thus, the choice of fan control strategy depends upon the ratio of leakage and conduction losses.

With respect to duct dynamics, the key issue is that of fan overrun. A significant quantity of energy can be extracted from the heat exchanger as well as the duct system by means of fan overrun. However, fan overrun also increases the effect of duct leakage flows. To illustrate the interaction between conduction and leakage, for an airtight uninsulated duct system, the best strategy would be to maximize fan overrun so as to quickly pull the stored heat out of the ducts into the house. Conversely, for a leaky well-insulated duct system the best strategy would be to reduce fan overrun and let more of the heat stored in the ducts be naturally convected into the house.

Another key point that merits discussion is the overall energy-consumption impact of the duct-system losses described in this paper. The two-speed compressor results were presented in terms of the overall impact on energy consumption, however the thermal mass and thermosiphon results were presented only in terms of their impacts on energy delivery efficiency and loads. In the two latter cases, the increased losses will impact the loads seen by the equipment, and therefore its efficiency. The direction of the equipment-efficiency impacts depends on the type of equipment (see two-speed compressor example above).

For each of the three model applications presented, the example duct system was located principally in the attic. Although this is a very popular location for ductwork, it is also about the worst possible location. It is worth noting that the energy consumption implications of all three of the effects examined would be almost completely mitigated if the ductwork was installed within the conditioned space. If the ductwork was installed in a basement with an uninsulated ceiling, approximately half of the losses are typically recovered, much less so if the basement ceiling is insulated. Ducts located in a crawlspace generally have smaller losses compared to attic ductwork, which is principally due to smaller temperature differentials. Crawlspace losses can also be mitigated by sealing the crawlspace and insulating its walls rather than its ceiling.

CONCLUSIONS

Although the applications of the three model improvements presented in this paper (improved simplified modelling of duct thermal mass effects, modelling of thermosiphon effects, and modelling of two-speed air conditioners) are by no means comprehensive, they do allow us to draw several conclusions about the effects under study. The primary conclusion to be drawn is that these effects, particularly the interaction between two-speed compressors and duct systems, as well as the thermosiphon-driven heat exchange, require fairly detailed systems modeling to provide us with confidence in our ability to understand overall HVAC system performance.

Concerning the impact of duct thermal mass, the results suggest that the mass effect is relatively small for plastic flexducts and insulated sheet metal, but that it needs to be accounted for when considering the effects of adding insulation. The losses due to mass effects are significantly smaller than those due to thermosiphon flows. Based on these results, moderately detailed modelling of duct-system dynamics (at least at the level presented in this paper) should generally be included in future examinations of residential equipment control strategies, particularly for uninsulated metal ductwork.

Based upon the limited thermosiphon results presented, we can safely conclude that the magnitude of the effect is sufficiently large that it should not be ignored when comparing the energy efficiency of alternative air distribution systems. The results presented also point out that further work is needed in order both to understand the multi-stable nature of the flow patterns developed, and to obtain closure between model predictions and experimental results from the field.

Of the simulation results presented, the most immediate implications arise from the interactions between air distribution systems and variable-capacity heating and cooling equipment. At the very least, it is clear that the energy savings potential of variable-capacity equipment is severely compromised by installing air distribution systems at present design and installation quality standards. In addition, present equipment efficiency yardsticks and test procedures, such as the Seasonal Energy Efficiency Ratio, do not appear to appropriately characterize the relative performance of variable-capacity equipment.This is of particular current importance, as both electric utility companies and the U.S. government are attempting to accelerate the deployment of energy efficient HVAC systems into the marketplace.

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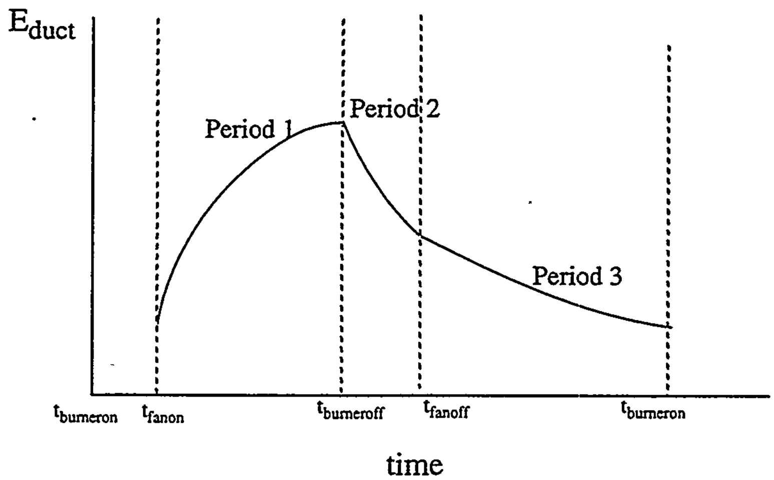

FIGURE 1. THE EVALUATION OF THE DUCT ENERGY AS A FUNCTION OF TIME FOR A FURNACE.

TABLE 1. IMPACT OF DUCT MASS ON ENERGY DELIVERED TO THE CONDITIONED SPACE.[a]

Duct Construction	*Energy Deficit (Heating)*	
	$E_{mass\text{-}lost}$ *[MJ]*	$E_{mass\text{-}lost}/E_{delivered}$
R-4 plastic flex duct	28	1.4%
R-8 plastic flex duct	41	2.0%
Sheet metal w/o insulation	123	6.1%
Sheet metal w/R-5 fiberglass insulation	44	2.2%

a. Results are from simulations of an attic-only trunk and branch duct system in Atlanta, GA for January 1-7.

TABLE 2. THERMOSIPHON ENERGY FLOWS FROM A ONE-WEEK SIMULATION OF AN ATTIC/CRAWLSPACE DUCT SYSTEM IN ATLANTA (JANUARY 1-7).

Quantity	*Nominal Insulation Level [m^2 K/W (ft^2 °F h/Btu)]*			
	R-0.35 (R-2)	*R-0.7 (R-4)*	*R-1.4 (R-8)*	*R-2.1 (R-12)*
Heating Load Due to Thermosiphon [MJ]	318	230	142	101
Percentage of Total Heating Load [%]	16%	12%	7%	5%

TABLE 3. MASS FLOWS AND ENTERING-AIR TEMPERATURES FOR THERMOSIPHON DURING THE MOST EXTREME HOUR OF JANUARY 1-7 IN ATLANTA.

Quantity	*Nominal Insulation Level [m^2 K/W (ft^2 °F h/Btu)]*			
	R-0.35 (R-2)	*R-0.7 (R-4)*	*R-1.4 (R-8)*	*R-2.1 (R-12)*
Mass Flow [kg/h]	322	291	263	249
Temperature [°C]	1.8	6.9	11.6	13.7

TABLE 4. CHARACTERISTICS OF SIMULATED SINGLE-SPEED AND TWO-SPEED AIR CONDITIONERS

A/C Equipment		*Cooling Capacity [kW (kBTU/h)]*	*Fan Flow [m^3/h (cfm)]*	*Fan power [W]*	*Steady-State COP[a] [-]*
Single Speed		10.5 (36)	2040 (1200)	550	2.9
Two Speed	(High Capacity)	10.5 (36)	2040 (1200)	550	2.9
	(Low Capacity)	5.3 (18)	1020 (600)	80	4.5

a. Part load curves for both systems are similar to those measured by Parken et. al. (1985).

TABLE 5. ENERGY CONSUMPTION AND EFFICIENCY RESULTS FOR SINGLE-SPEED AND TWO-SPEED AIR CONDITIONERS (JUNE 1 - AUGUST 31 IN SACRAMENTO RANCH HOUSE)

Parameter	*Single Speed*	*Two Speed (HIGH-CAP CYCLING)*	*Two Speed (LOW-CAP MAX)*	*Ratio of HIGH-CAP CYCLNG to Single Speed*	*Ratio of LOW-CAP MAX to Single Speed*
Nominal Seasonal Energy Efficiency Ratio [- (kBtu/kWh)]	3.5 (12)	4.7 (16)	4.7 (16)	0.75[a]	0.75[a]
Energy Consumption with Perfect Ducts[b] [kWh]	1228	866	755	0.71	0.60
Energy Consumption with Typical Ducts[c] [kWh]	1770	1548	1413	0.88	0.80
Typical Duct System Delivery Efficiency [-]	0.68	0.65	0.59	0.96	0.87
A/C Equipment Efficiency Ratio [-]	1.08	0.91	0.93	0.84	0.86

a. The inverse ratio, i.e. $SEER_{single\text{-}speed}/SEER_{two\text{-}speed}$, is given to allow comparison with the two rows immediately below.

b. Airtight ducts with no conduction losses.

c. Attic duct installation with R-4 insulation, supply leakage of 14% of fan flow, and return leakage of ~17% of fan flow.