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Impact of Ventilator Parameters on System Energy Consumption

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Synopsis

The operation and performance of forced-air ventilation systems with the aid of a dynamic modeling and simulation computer program are presented. The functions and features of GEMS (Generalized Engineering Modeling and Simulation), a dynamic modeling and simulation software tool, are briefly described. Using GEMS, the effects of different ventilation airflow rates and sensible and moisture efficiencies on the thermal comfort environment within the conditioned space were analyzed. The impact of these ventilator parameters on the indoor air humidity levels and annual energy requirements for two cities—one cooling dominated and one heating dominated—was studied. Simulation results from the cooling-dominated region are presented.

Keywords: *air quality, comfort, consumption, control, energy, residential, total energy, ventilation, mechanical ventilation, thermal comfort*

Introduction

The advent of "tighter, less leaky" residential construction has resulted in poor indoor air quality (IAQ) within conditioned spaces. One of the major reasons for deteriorating IAQ is the reduced level of infiltration/exfiltration for ventilating the space. Increasingly, forced-air mechanical ventilation systems such as heat or energy recovery ventilators (HRV or ERV) are being used to improve the IAQ by introducing preconditioned "fresh" outdoor air into an occupied space. Although these systems aid in reducing irritants and pollutants such as odors, volatile organic compounds (VOCs), and the like, they impose additional sensible and latent loads on the building's primary heating, ventilation, and air conditioning (HVAC) system. The objective of this study was to use dynamic simulation models to analyze the impact of residential mechanical ventilation system parameters on the thermal comfort conditions within a conditioned space and on the annual heating and cooling energy requirements. The GEMS program was used to study the effect of ventilator airflow rates and sensible and moisture removal efficiencies (η_s and η_m, respectively) on the indoor air moisture levels and annual energy requirements for two U.S. cities: Miami, Florida, a cooling-dominated climate, and Minneapolis/St. Paul, Minnesota, a significant heating climate with substantial cooling loads. Due to space limitations, only the results for Miami are presented.

First, a brief description of GEMS and the simulation models is presented. Next, the mechanical ventilator parameters and technical approach used in this study are discussed. Then the simulation results for Miami are presented. Finally, the analysis effort is summarized and conclusions are drawn.

Generalized Engineering Modeling and Simulation

GEMS is a structured, generalized modeling and simulation software that enables development and analysis of control algorithms and systems and prediction of thermal performance and energy use in buildings [*1*]. The basis for this tool is the state-space technique, which casts differential and algebraic equations describing the system into a vector-matrix form. In doing so, several powerful tools such as linear algebra, modern control theory, and vector-matrix numerical methods can be used to study the system. In modeling even a moderately detailed system, the state-space form presents an enormous bookkeeping and coupling problem. Moreover, each time the system is modified, the coupling among the equations necessitates a derivation of all the state-space vector-matrix equations. Herein lies the power of GEMS, which contains the tools to automatically generate the state-space representation of the entire system from user-input descriptions of the basic subsystem block dynamics and the interconnections of these blocks. Detailed and simplified building models can be generated directly from plans by using a library of construction element models (walls, floors, etc.). A closed-loop system simulation can be generated by interconnecting the building model with other component models selected from a library of HVAC equipment, controls, internal loads and schedules, and weather data.

Over the years, GEMS has been used extensively in the development and analysis of control algorithms for a large number of existing and future products. With the proliferation of inexpensive microprocessors, control algorithms developed with the aid of GEMS have been directly embedded into microprocessor-based controllers. Furthermore, GEMS has also been used in emulators (closed-loop or real-time simulations) for testing and analyzing product prototypes.

Simulation Models

Layout of the simulated house, construction, and HVAC system was based on an actual two-story single-family house with full basement located in Eden Prairie, Minnesota, which has been used in the past for field testing prototype controllers. The house is divided into four thermal zones and has a single HVAC system consisting of a central forced-air heating (furnace) and cooling (air conditioner (a/c)) plant whose operation is controlled by a proportional-plus-integral thermo-stat located within one of the conditioned zones. Time-scheduled internal heat and moisture gains, such as from occupants, appliances, household activities (e.g., cooking, showers, laundry), are included in the model. The only paths for air to flow between the inside and the outside of the house are the furnace stack, the make-up duct, and cracks in the building's structure. Stack flow is calculated as a function of the stack temperature, and make-up air is considered to be a constant value whenever the central fan is "on" and zero when the fan is "off". In the simulated system, the make-up flow equals the nominal stack flow of 84 cubic feet per minute (cfm) (39.65 liters per second (L/s)), and the central fan is "on" only when required. Infiltration was modeled as a function of both wind speed and indoor-to-outdoor temperature difference [*2*]. The baseline infiltration rate, using blower door testing, was determined to be 0.281 air changes per hour (ach) in the heating mode. The average annual infiltration rate was estimated to be 0.33 ach. No mechanical ventilation is designated as the reference or baseline case.

A simple steady-state model of a mechanical ventilator with constant η_s and η_m and fixed airflow rates was used in this study. This single model was used for analyzing three different ventilation systems: (a) direct mechanical ventilation to introduce unconditioned outside air (η_s = 0% and η_m = 0%), (b) a HRV (η_s > 0% and η_m = 0%), and (c) an ERV (η_s > 0% and η_m > 0%). The ventilation systems were assumed to be "on" all the time, supplying outside ("fresh") air into the supply side of the central duct distribution system. Simulations were performed with ventilation airflow rates of 100 and 200 cfm (47.2 and 94.4 L/s) with corresponding fan power consumption of 82 W and 164 W.

Technical Approach

Using the previously described GEMS model of a two-story single-family house with full basement and its associated HVAC components and system, "typical" day and seasonal (heating and cooling) simulations were conducted with "typical" year weather data for Minneapolis, Minnesota (significant heating climate with substantial cooling loads). A slab-on-grade version of this Minnesota house, reflecting Florida building construction practices (no basements), was also developed for simulations in a cooling-dominated hot and humid climate such as Miami, Florida. Assuming a year-round cooling mode of operation, "typical" year weather data for Miami was used for performing "typical" day and annual simulations.

Several simulations were conducted with different combinations of the following ventilator parameters for studying their impact on energy consumption and indoor conditions:

- Ventilator airflow rates of 100 cfm and 200 cfm (47.2 and 94.4 L/s) with fan power consumption of 82 W and 164 W, respectively;
- ηm = 0%, 30%, 50%, and 80%;
- η_s = 0%, and 70%.

Note that η_s = η_m = 0% reflects direct introduction of outside air into the conditioned space without any heat and/or moisture recovery for preconditioning this airstream. Also, η_s > 0% and η_m = 0% represents an HRV, and η_s > 0% and η_m > 0% implies an ERV.

The thermostat (dry-bulb temperature) set points were assumed to be constant:

- Minneapolis: 70°F (21.1°C) for heating, 76°F (24.4°C) for cooling;
- Miami: no heating, 72°F (22.2°C) for cooling.

Simulation Results

As previously mentioned, "typical" day and seasonal/annual simulations were performed with "typical" year weather data for Miami, Florida, and Minneapolis/St. Paul, Minnesota, with appropriate two-story single-family house and HVAC system simulation models representative of construction practices in the two geographic areas. Results for Miami, assumed to have a year-round cooling mode of operation (no heating), will be presented; however, data for Minneapolis/St. Paul, which has seasonal heating and seasonal cooling requirements, will not be presented due to space limitations.

"Typical" day simulations for Miami consisted of outdoor conditions that were (a) mild and humid (day 121), (b) hot and humid (day 169), and (c) cold and humid (day 304). Scenarios considered included a baseline case of no mechanical ventilation, systems with direct mechanical ventilation introducing unconditioned outside air into the conditioned space, and recovery ventilators with different sensible and moisture removal efficiencies and two airflow rates.

Dry-bulb temperatures of the outdoor air and within the conditioned space are shown in Figures 1, 2, and 3 for days 121, 169, and 304, respectively, for the baseline and different ventilation schemes. In almost all cases, the dry-bulb temperature within the conditioned space is maintained close to the set point by the conventional thermostat, which is basically a dry-bulb temperature controller. The "sawtooth" variation in the space air temperature is due to the air conditioner (a/c) cycling on and off. Two exceptions are noted for the hot and humid day in Figure 2:

1. Between 12:00 and 22:00 when supplying 200 cfm (94.4 L/s) of unconditioned outside air for ventilation, and

2. For some ventilation schemes between 17:00 and 20:00.

For these ventilation conditions, the air dry-bulb temperature does not oscillate and increases slightly, indicating that the a/c is locked on due to the cooling demand exceeding equipment capacity. Since the cooling demand for the baseline case of no ventilation does not exceed equipment capacity, the a/c cycles on and off throughout the day as implied by the sawtooth pattern. This indicates that any form of mechanical ventilation, with or without preconditioning the outside air, will increase the sensible cooling load on the house/HVAC system.

Dew-point temperatures of the outdoor air and within the conditioned space are shown in Figures 4, 5, and 6 for days 121, 169, and 304, respectively, for the baseline and different ventilation schemes. The bottommost curve in these figures is for the baseline case of no mechanical ventilation, and the topmost curve shows the outdoor dew-point temperature. It is obvious that any form of mechanical ventilation tends to raise the dew-point temperature within the conditioned space relative to the baseline. Again, the sawtooth variation in the temperature is due to the a/c cycling on and off—decreasing when the a/c is on and extracting moisture from the space air and increasing when the a/c is off due to internal gains and moisture brought in by the ventilation system. Ventilation airflow rates of 200 cfm (94.4 L/s) introduce more water, and heat, into the space than 100 cfm (47.2 L/s) and hence have higher space dew-point temperatures. Also, direct mechanical ventilation systems ($\eta_s = \eta_m = 0\%$) that introduce unconditioned outside

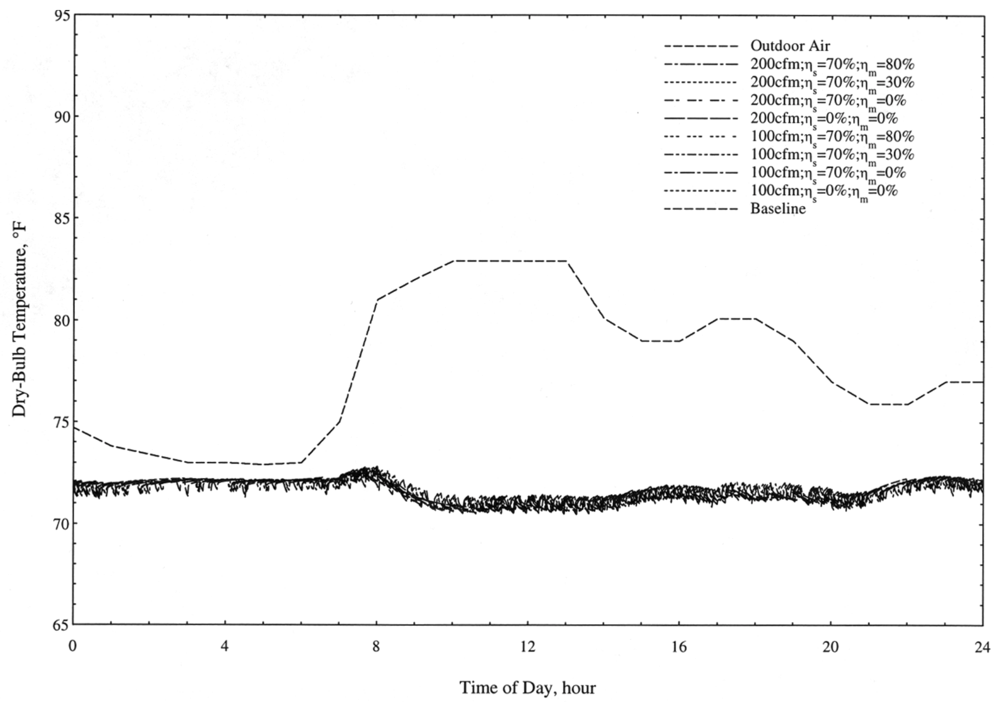

Figure 1. Dry-bulb temperatures on a mild and humid Miami day (Day 121; Zone 2; T_{dbsp} = 72ºF)

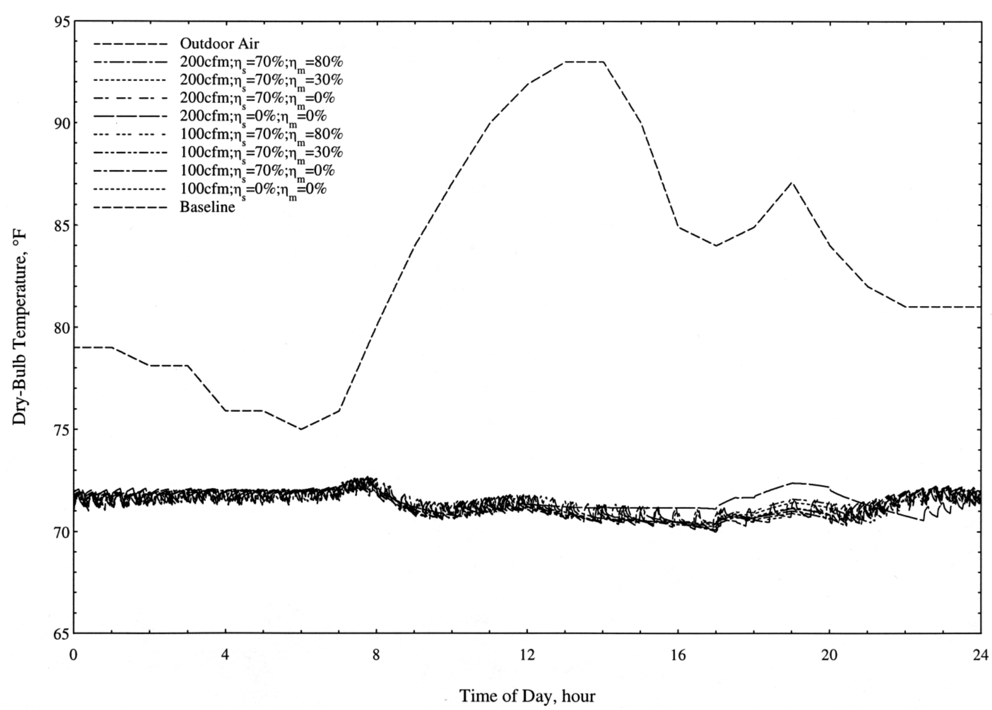

Figure 2. Dry-bulb temperatures on a hot and humid Miami day (Day 169; Zone 2; T_{dbsp} = 72ºF)

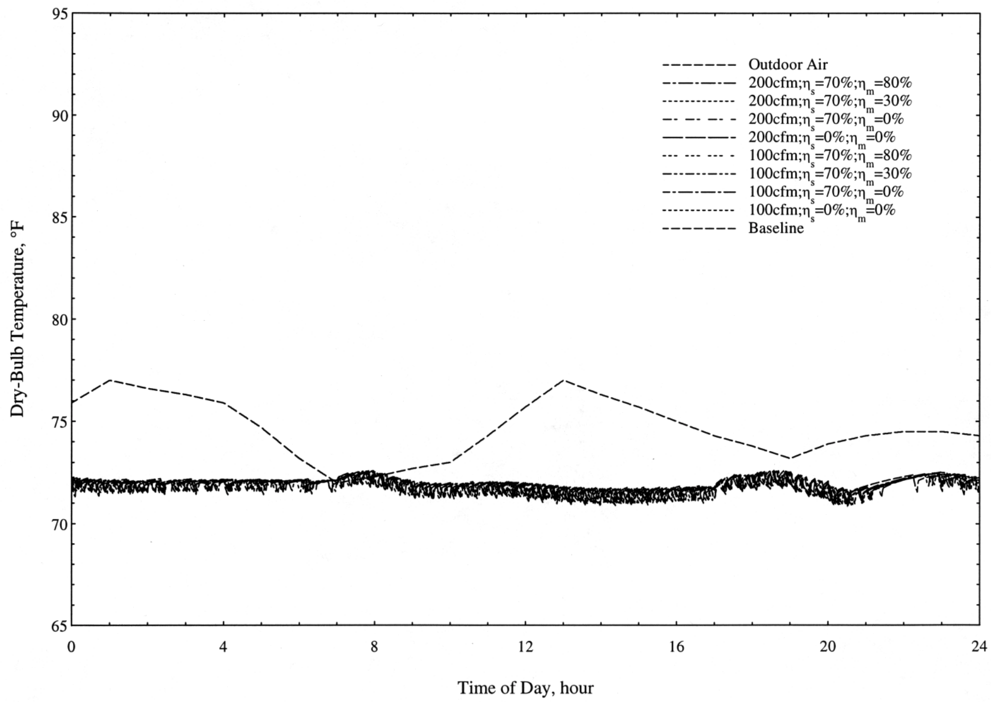

Figure 3. Dry-bulb temperatures on a cold and humid Miami day (Day 304; Zone 2; T_{dbsp} = 72ºF)

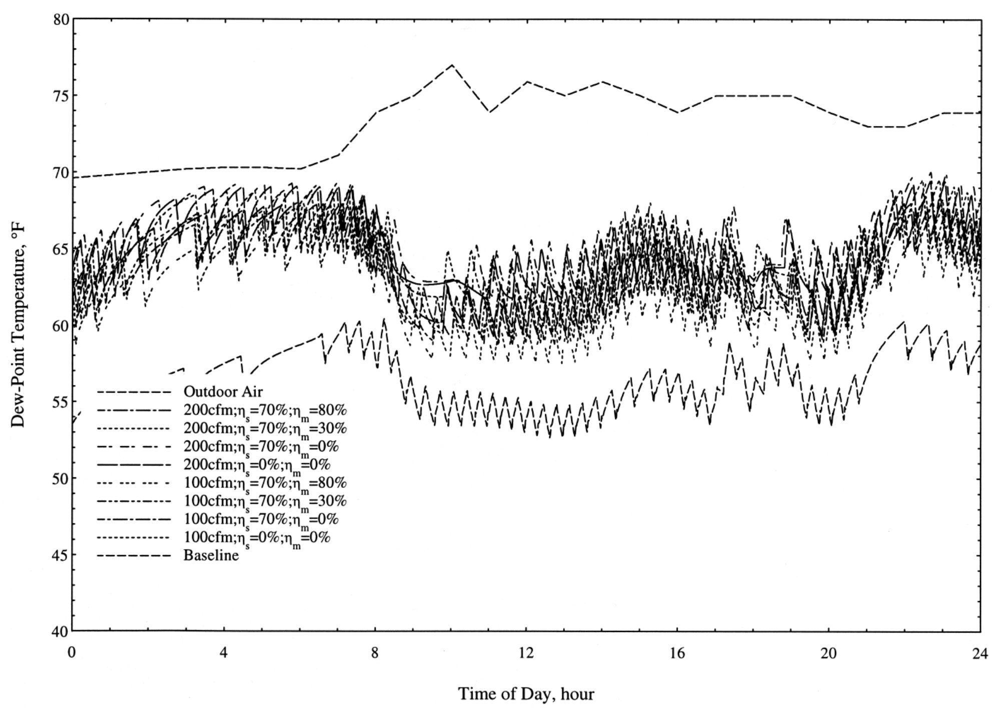

Figure 4. Dew-point temperatures on a mild and humid Miami day (Day 121; Zone 2; T_{dbsp} = 72ºF)

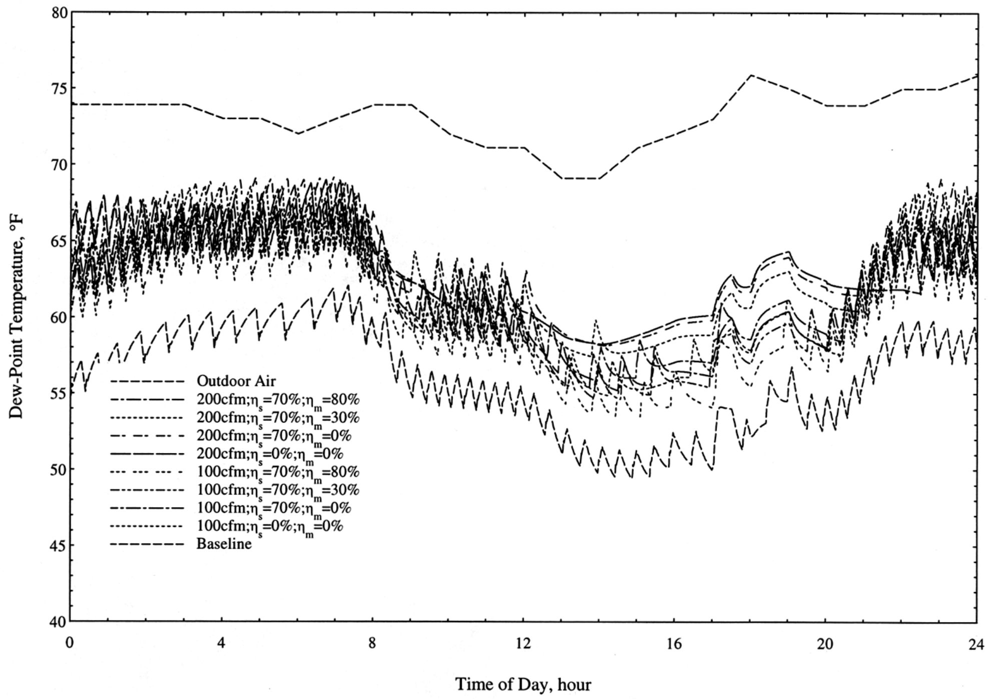

Figure 5. Dew-point temperatures on a hot and humid Miami day (Day 169; Zone 2; T_{dbsp} = 72ºF)

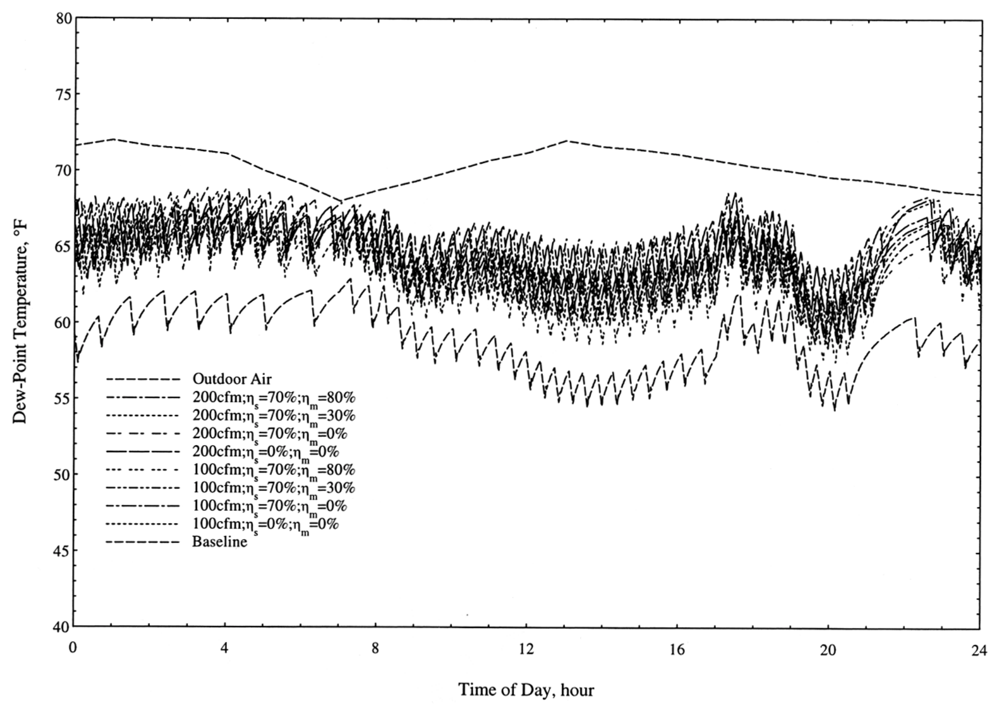

Figure 6. Dew-point temperatures on a cold and humid Miami day (Day 304; Zone 2; T_{dbsp} = 72ºF)

air into the space cause the space dew-point temperature to be greater than systems (i.e., ERVs) with moisture recovery ($\eta_m > 0\%$). Additionally, systems with higher η_m have lower indoor dew-point temperatures than those with low η_m. Figure 5 shows that between 13:00 and 19:00 hours for the 200 cfm case, when the a/c equipment does not have sufficient capacity to satisfy the sensible load and is locked on (Figure 2), the space dew-point temperature continues to increase despite the removal of water by the a/c coil. This implies that the a/c does not have sufficient latent removal capacity either. The slight dip in the dew-point temperatures just before 18:00 is due to reduction in the prescheduled internal moisture gain within the conditioned space.

Annual energy consumption and cost for operating the ventilation system and the cooling equipment for the baseline and various ventilation strategies are listed in Table 1 for the Miami house. As expected, all mechanical ventilation systems increase the cost of operating the cooling system. Direct mechanical ventilation systems (lower first cost) are the most expensive to operate because they introduce un-conditioned outside air into the space ($\eta_s = \eta_m = 0\%$). The annual energy, and operating cost, with HRVs ($\eta_s = 70\%$ and $\eta_m = 0\%$) is somewhat less than that for direct mechanical ventilation because the outdoor air is sensibly cooled by the exhaust air stream. The simulations indicate that the annual energy, and operating cost, with ERVs ($\eta_s = 70\%$ and $\eta_m > 0\%$) is somewhat less than that for HRVs because the moisture removal process from the exhaust air further cools down the incoming outside air stream. Obviously, higher ventilator airflow rates lead to higher energy consumption and cost. It should also be noted that any recovery ventilation system (HRV or ERV) will have some first cost and annual maintenance cost associated with it. The direct mechanical ventilation system will be least expensive to install with negligible maintenance requirements.

Annually-averaged indoor and outdoor conditions for the Miami, Florida, house are listed in Table 2 for the baseline and ventilation strategies. The average dry-bulb temperatures within the conditioned space are not affected because the thermostat controlling the air conditioning equipment is a dry-bulb temperature controller. As discussed previously, any mechanical ventilation will add moisture to the conditioned space whenever the outdoor dew-point temperature is higher than that indoors, and hence raise the house relative humidity. Since the air inside the house is diluted by "fresh" outdoor air, mechanical ventilation reduces carbon dioxide levels in the space. The overall ventilation rate in the house obviously increases when using a mechanical ventilator. The data in Table 2 indicate that the predictive mean vote (PMV) in one of the zones of the simulated house, Zone 2, might improve marginally when using mechanical ventilation.

Summary and Conclusions

Assuming year-round cooling requirements for Miami, Florida, "typical" day and annual simulations were conducted for a single-family two-story slab-on-grade house with different mechanical ventilation parameters such as airflow rates and sensible and moisture removal efficiencies (η_s and η_m). No mechanical ventilation is designated as the reference or baseline case. Ventilation strategies analyzed included (a) direct introduction of unconditioned outside air into the house ($\eta_s = \eta_m = 0\%$), (b) heat recovery ventilator (HRV, $\eta_s = 70\%$, $\eta_m = 0\%$), and (c) energy recovery ventilator (ERV, $\eta_s = 70\%$, $\eta_m = 30\%$, 50%, and 80%).

Table 1. Annual Energy Consumption and Cost for Miami, Florida (72ºF dry-bulb temperature set point)

System Configuration	Power Consumption (kwh)				Increased Operating Cost ($/year @$0.08/kwh)	No. of Cooling Plant Cycles
	Ventilator	Compressor	Blower	Total		
Baseline	0	7,733	2,195	9,982	–	14,800
100 cfm $\eta_s = \eta_m = 0\%$	718	9,727	2,718	13,160	259	13,210
100 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	718	9,456	2,643	12,820	231	13,850
100 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	718	9,283	2,599	12,600	214	14,020
100 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	718	8,966	2,518	12,200	182	14,310
200cfm $\eta_s = \eta_m = 0\%$	1,436	11,220	3,112	15,770	467	11,250
200 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	1,436	10,710	2,971	15,120	415	12,640
200 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	1,436	10,450	2,906	14,800	390	12,970
200 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	1,436	9,968	2,783	14,190	341	13,530

Table 2. Annually Averaged Conditions for the Miami, Florida, House (72ºF dry-bulb temperature set point)

System Configuration	Average Dry-Bulb Temperature (°F)				Average Relative Humidity (%)				Zone 2 PMV
	Outdoor	Zone 2	Zone 3	Zone 4	Outdoor	Zone 2	Zone 3	Zone 4	
Baseline	75.4	71.3	72.8	70.5	74.1	53.9	52.0	54.8	–0.1986
100 cfm $\eta_s = \eta_m = 0\%$	75.4	71.2	72.7	70.4	74.1	63.9	61.4	65.2	–0.1464
100 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	75.4	71.3	72.7	70.4	74.1	64.2	61.7	65.5	–0.1386
100 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	75.4	71.3	72.7	70.4	74.1	63.2	60.8	64.5	–0.1448
100 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	75.4	71.3	72.7	70.4	74.1	61.2	58.9	62.5	–0.1566
200cfm $\eta_s = \eta_m = 0\%$	75.4	71.3	72.7	70.5	74.1	67.5	64.8	68.9	–0.1194
200 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	75.4	71.3	72.8	70.5	74.1	68.1	65.4	69.6	–0.1109
200 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	75.4	71.3	72.8	70.5	74.1	67.0	64.3	68.5	–0.1191
200 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	75.4	71.3	72.7	70.4	74.1	64.7	62.2	66.1	–0.1351

System Configuration	Carbon Dioxide Concentration (ppm)				Ventilation Rate (ach)
	Outdoor	Zone 2	Zone 3	Zone 4	
Baseline	400.0	642.7	645.1	657.4	0.244
100 cfm $\eta_s = \eta_m = 0\%$	400.0	504.0	506.9	515.2	0.499
100 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	400.0	504.0	506.9	515.3	0.499
100 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	400.0	504.1	507.0	515.4	0.499
100 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	400.0	504.2	507.1	515.6	0.499
200cfm $\eta_s = \eta_m = 0\%$	400.0	465.0	467.9	474.6	0.754
200 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	400.0	465.0	468.0	474.7	0.753
200 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	400.0	465.0	468.1	474.8	0.753
200 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	400.0	465.1	468.2	474.9	0.753

"Typical" day simulations for Miami consisted of outdoor conditions that were (a) mild and humid (day 121), (b) hot and humid (day 169), and (c) cold and humid (day 304). These results indicate that

- With a conventional thermostat (dry-bulb temperature controller) operating the HVAC system, the space dry-bulb temperature is maintained near the specified set point;
- Relative to the baseline case, any form of mechanical ventilation (i.e., with or without recovery) will increase the dew-point temperature within the conditioned space:
  - For a given η_m, indoor moisture content increases with ventilation airflow rate;
  - For a given ventilation airflow rate, indoor moisture content decreases with increasing η_m.

The annual simulations for Miami show that

- With any form of mechanical ventilation
  - Whole-house ventilation rate increases with "fresh" (outside) airflow rate;
  - Carbon dioxide concentration levels within the occupied conditioned space decrease with increasing ventilation airflow rates;
  - Moisture content levels (or dew-point temperatures) within the conditioned space are higher than for the baseline case of no mechanical ventilation;
  - Total power consumption and energy cost are higher than the baseline case and increase with ventilation airflow rate;
  - Total number of cooling plant operations are reduced because "free cooling" is provided whenever the outdoor dry-bulb temperature is less than the thermostat setpoint temperature;
  - Total number of air conditioning equipment cycles decrease with increasing ventilation airflow rates;
- Ventilation airflow rates and η_m do not affect the dry-bulb temperature within the conditioned space;
- In a warm and humid climate, such as that of Miami, Florida, it is more advantageous to use an ERV than an HRV because the indoor moisture content is lower; however, there is no significant advantage in annual energy/operating cost.

In today's "tighter, less leaky" residential construction, the primary purpose of any form of mechanical ventilation is to improve the indoor air quality. Consequently, simulations have shown that the indoor moisture content levels are higher than those for the baseline case of no mechanical ventilation. Elevated levels of indoor moisture content can result in fungal and bacterial growth, leading to more adverse health problems and damage to property. One potential solution is to lower the thermostat (dry-bulb temperature) set point; however, this results in increased energy cost, a "chilly" space, and low temperatures even when not necessary. Another potential solution might be to use an ERV with the lowest possible η_s and the highest possible

η_m. The optimal ventilation airflow rate (low vs. high) needs to be investigated. Alternatively, a robust temperature and humidity controller [*3,4*] can be used for maintaining desired levels of dry-bulb and dew-point temperatures within the conditioned space without installing any additional equipment such as recovery ventilators (HRV, ERV), desiccant systems, or heat pipe units. Such controllers are beneficially attractive in that there is no capital and/or annual maintenance cost for additional equipment.

Given the current state of the art, a recommended solution is to use an energy recovery ventilator with the lowest possible ventilation airflow rate and the highest possible sensible and moisture removal efficiencies.

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