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ENERGY ANALYSISF FOR WORKSHOPS WITH FLOOR-SUPPLY DISPLACEMENT VENTILATION UNDER THE U.S. CLIMATES

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ABSTRACT

Many studies have shown that floor-supply displacement ventilation systems are better than mixing ventilation systems. The benefits include indoor air quality, thermal comfort, and reduced energy use. The energy benefits depend on the climate conditions. This research compared the energy use of a floor-supply displacement ventilation system in a large industrial workshop with that of a mixing ventilation system for five U.S. climate regions. It was found that the energy use and the system performance vary with the locations. The displacement ventilation system may use more fan and boiler energy but less chiller energy than the mixing ventilation system. The total energy used is slightly less with displacement ventilation, although the ventilation rate was increased in order to handle the high cooling loads found in U.S. buildings. Thus, the displacement ventilation system can save some energy in cooling mode. However, displacement ventilation system has a lower capacity of dehumidification. This system alone, thus, is not suggested for use in humid regions.

KEYWORDS

Floor-supply displacement ventilation, Energy performance, Energy cost, Climates

INTRODUCTION

Energy Information Administration (EIA) [1] reported that the energy related to ventilation, space cooling and heating accounts 37.9 % of the total energy in the buildings studied. Therefore, using a properly designed ventilation system would be very significant for building energy saving. The most common and traditional ventilation system in the U.S. buildings is ceiling-supply mixing ventilation system. However, the applications of floor-supply displacement ventilation are growing significantly in the U.S. in the recent decade. More and more building owners, HVAC system designers, architects and building services consultants are more willing to consider the use of floor-supply displacement ventilation system in their new buildings. It is estimated that around 12% of new office building have raised floors nowadays and around 40% of these buildings installed floor-supply displacement ventilation [2]. Modern office buildings are frequently found to be changing the space configuration and usages. Computers and devices for information technology are almost a default facility in an office. Therefore, the use of raise floor has the advantage of easier cable management and greater flexibility [3, 4]. Floor-supply displacement ventilation system has additional benefits which attracted people consideration. It improved indoor air quality [4-7], occupant comfort, productivity and health [3, 4, 6, 8-11]. Moreover, it also reduced energy use [2, 4, 6, 10], life-cycle building costs and floor-to-floor height in new construction [4, 10].

The displacement ventilation was claimed to have a significant energy saving when compared with mixing ventilation. For example, due to the temperature stratification in a room, a higher return air temperature is normally expected even with the same air temperature in the occupied zone. Shute [12] estimated that the cooling load can be reduced by as much as 15% compared to conventional well-mixing ventilation systems. Displacement ventilation systems use warmer supply air (about 16-19 ^{o}C). Thus, the water in the cooling coil of the air handling system does not have to be chilled to the temperature as low as the conventional one. With a warmer evaporating temperature in the refrigeration cycle, the chiller efficiency is increased. Besides, the cooling demand at the coil is reduced because of the cooling load reduction explained above. Loudermilk [13], Akimoto et al. [14], Bielli [15] and Wilson [16] carried out investigations with similar conclusions. According to Matsunawa et al. [17], the efficiency of the refrigerating machine may improve by around 5%.

The supply air temperature of mixing ventilations is normally 13-14 °C while that of displacement ventilations is 16-18.5 °C. The higher temperature of supply air also means that a longer free cooling season. The economizer can be operated using 100% outdoor air when the outdoor air temperature is in the range of 13-18.5 °C for displacement ventilation. Akimoto et al [14] studied the energy saving of a building in Tokyo, Japan. In that study, the floor-supply displacement system with free cooling requires 34% less energy than the conventional mixing system with free cooling. The degree of saving by free cooling depends on the characteristics of the weather. Fig. 1 shows the number of hours of outdoor air temperature which is within the operating range of the economizer at five different weather zones in the U.S. It was discovered that Seattle and New Orleans have longer periods of time that allowed the use of free cooling.

Although many studies have been carried out in the separated aspects of energy of floor-supply displacement ventilation system, very few of them are systematic and even fewer are for U.S. buildings under different weather conditions. In addition, little comparison of whole building energy performance is made between the floor-supply displacement ventilation and traditional mixing ventilation systems. Building ventilation designers need complete and systematic information to understand the energy performance of this system for different types of buildings in different climate regions. This forms the objective of the study reported in this paper.

RESEARCH METHOD

Energy simulation methods range from manual methods to detailed computer simulation methods. The manual methods, such as the degree-day and bin methods [18], are still widely used in practical designs, although they are not accurate. The degree-day method uses only one value of temperature, while the bin method [18] calculates the energy over several intervals (bins) of temperature. However, the detailed methods often calculate the energy on an hour-by-hour basis. Although the manual methods are simple, they could not, for example, be used for the comparison of energy consumption by displacement ventilation with that by mixing ventilation. The detailed computer simulation can do the comparison. Therefore, the present investigation used a detailed computer simulation method by using EnergyPlus Program [19].

EnergyPlus calculates integrated solution of building loads, system loads, and energy used by plants. The solution is based on the following heat balance equation referred to as the predictor-correction method and assumes that the room air is uniform:

$$C_z \frac{dT_z}{dt} = \sum_{i=1}^{N_{sl}} \dot{Q}_i + \sum_{i=1}^{N_{surfaces}} h_i A_i \left(T_{si} - T_z\right) + \sum_{i=1}^{N_{zones}} \dot{m}_i c_p \left(T_{zi} - T_z\right) + \dot{m}_{\inf} c_p \left(T_{\inf} - T_z\right) + \dot{Q}_{sys} \tag{1}$$

where:

$\sum_{i=1}^{N_{sl}} \dot{Q}_i$ = sum of internal convective loads from people, computers, etc.

$\sum_{i=1}^{N_{surfaces}} h_i A_i \left(T_{si} - T_z\right)$ = convective heat transfer from zone surfaces

$\sum_{i=1}^{N_{zones}} \dot{m}_i c_p \left(T_{zi} - T_z\right)$ = heat transfer due to interzone air mixing

$\dot{m}_{\inf} c_p \left(T_{\inf} - T_z\right)$ = heat transfer due to infiltration

$\dot{Q}_{sys} = \dot{m}_{sys} c_p \left(T_s - T_z\right)$ = heat transfer from ventilation system

$C_z \frac{dT_z}{dt}$ = rate of energy storage in air

The predictor-corrector scheme first predicts the system energy required to balance Equation (1) with the room temperature equal to the set-point temperature at steady state (i.e. neglecting the rate of energy storage in air). Then, with that quantity as a demand, the system is simulated to determine its actual supply capability at the time of the simulation. This will include a plant simulation if necessary. The actual supply capability is used in Equation (1) to correct the resulting room temperature.

The non-uniform room air temperature distribution makes displacement ventilation systems more difficult to model and design than mixing ventilation systems. In Equation (1), the room air temperature is assumed as uniform. It may be closer to the situation of traditional mixing ventilation systems; however, it is not true for displacement ventilation systems due to the temperature stratification [9]. The present study used the model developed by Carrilho da Graca [20] to account for the temperature gradient in the room air with floor-supply displacement ventilation. The model divided a room vertically into three well separated regions – each region characterized by a single temperature as shown in Fig. 2:

- A floor level temperature (T_{FLOOR}). The floor region is 0.2 meters deep and T_{FLOOR} represents the temperature at the mid-point of the region.
- An occupied sub-zone temperature (T_{OC}), representing the temperature in the region between the floor layer and the upper, mixed layer.
- An upper node representing the mixed-layer/outflow temperature (T_{MX}) essential for overall energy budget calculations and for modeling comfort effects of the upper layer temperature.

BUILDING AND AIR HANDLING SYSTEM DESCRIPTION

Although a lot of studies had been already done for office buildings, little energy analysis has been performed for industrial workshops with floor-supply displacement ventilation. This investigation studied an industrial workshop that is a large space with a lot of equipment and workers inside, as shown in Table 1. The table shows also the thermal properties of the building enclosure.

The energy performance of the workshop with a floor-supply displacement ventilation system is compared with that with a complete mixing ventilation system. Figs. 3 and 4 show the schematic of the air handling system for the displacement and mixing ventilation, respectively. The two air handling systems shared a lot of similarities. Both systems used a heat exchanger to recover the energy from the exhaust air to pre-cool and pre-heat the outdoor air in the cooling and heating season respectively. They used steam humidifier for humidification, and used the cooling coil for dehumidification. Free cooling was used in the shoulder seasons as well.

However, some parameters in these air handling systems were different. From the study of Hu et al. [21], Loudermilk [13] and Webster et al. [22], they pointed out that floor-supply displacement ventilation systems allow the chiller to operate at higher efficiencies (5-15%) due to the higher supply air temperature which allow the water temperature to be higher as well. Therefore, the authors had assumed that the coefficient of performance (COP) for the chiller for mixing ventilation is 2.9 and that for displacement ventilation is 3.1. Besides, the supply air temperatures of these two systems were also set at different values as show in Table 1.

In order to give an overall evaluation of the system performance, this investigation used the weather data from five different climatic regions for the energy analysis. The five climatic regions represent the most typical weathers in the continental U.S.: Seattle, WA (maritime), Portland, ME (cold), Phoenix, AZ (hot and dry), New Orleans, LA (hot and humid) and Nashville, TN (moderate).

RESULTS

Comparison of Energy Uses

In this section, the energy use by the under-floor displacement ventilation is compared with that by mixing ventilation for the workshop building.

Fig. 5 shows the monthly energy consumption of a typical workshop in Nashville, TN and New Orleans, LA. The displacement ventilation system used more fan energy than the mixing ventilation system. Although the exhaust air temperature with the displacement ventilation system was higher than that with the mixing ventilation system, the temperature difference between the exhaust and supply air was smaller. Typically, the exhaust air temperature for the displacement system is 29^{o}C in summer. Therefore, the temperature difference between the exhaust and supply air is typically 10 K. The temperature difference between the exhaust and supply air with the mixing system (12 K) is higher than that of displacement ventilation system. To offset the same amount of cooling load, a larger amount of supply air is needed with displacement ventilation. In the winter, the floor-supply displacement system is used to supply heated air from the floor level. The heating demand of the displacement ventilation is increased due to the temperature stratification in the room air. Therefore, higher flow rate of supply air is demanded to maintain the room temperature set point. The fan energy consumed by the displacement ventilation system during winter was also higher. The trend of the energy consumption is the same for office buildings with traditional displacement ventilation [21].

The energy consumption profiles were similar not only for Nashville, TN and New Orleans, LA but also for all the five locations as shown in Fig. 6. The magnitudes of the energy consumption were different in those locations but the trends still kept the same. The energy consumed by the chiller in the displacement ventilation system was much less than that in the mixing ventilation system. Since the supply temperature was higher in the displacement ventilation system than that in the mixing ventilation system, the chiller in the displacement system does not need to cool the air as much as in the mixing system. On the other hand, a higher supply air temperature would allow the displacement system to use free cooling for a longer period during the shoulder seasons. Also, the COP value was slightly higher with displacement ventilation (3.1) than with mixing ventilation (2.9).

Nevertheless, Webster et al. [22] mentioned that floor-supply displacement ventilation system in humid climates might need secondary equipment for dehumidification (such as desiccant equipment). It was because warm supply air temperature at the cooling coil not allowing to give sufficient dehumidification. From the results of the indoor humidity distributions for these two systems, Fig. 7 shows clearly the high humidity problem of displacement ventilation in the summer. In some periods, the relative humidity of the workshop with displacement ventilation can be as high as 80%.

Remediation of this high humidity situation is only carried out in the simulation in Nashville. Desiccant equipment was not added to our system with displacement ventilation because it would increase significantly initial equipment cost. Our study used a coil temperature for displacement ventilation that was sub-cooled to be the same as that of mixing ventilation (13^{o}C). Then the supply air was reheated back to 19^{o}C to avoid draft in the workshop. With such arrangement in the air handling process, Fig. 8 shows that the relative humidity in the workshop with both ventilation systems is nearly identical. However, as shown in Fig. 9, this sub-cool option for the displacement ventilation system would cause extra energy consumption, which was even higher than a traditional mixing ventilation system. This was because the efficiency of chiller was reduced by lowering the coil temperature and it makes no difference between using displacement ventilation and mixing ventilation. The small temperature difference between the supply and return will cause more fan energy. The temperature stratification would need more energy for heating. Therefore, using displacement ventilation system with sub-cool is not a good suggestion for dehumidifying the air under humid climates. Of course the floor-supply displacement ventilation will still provide better indoor air quality than the mixing ventilation. The application of floor-supply displacement ventilation for humid climate becomes a trade-off between more energy consumption and better air quality.

Comparison of Energy Costs

Since the system performance and the cost of different energy vary according to locations and climate, comparing the overall energy cost of the workshop will be a fair way in evaluating the benefits of using floor-supply displacement ventilation in a building. In our study, we assume that the fans and chillers used electricity and the boilers used natural gas in the five locations. As shown Table 2, the ratio of the cost of electricity to natural gas is different at the five locations. The data was taken from the Energy Information Administration (www.eia.doe.gov) of the U.S. Department of Energy.

Fig. 10 presents the normalized total energy cost in the five locations. Note that the results for displacement ventilation are for the system without sub-cooling in the cooling coil for humidity control. The total energy cost of displacement ventilation in Nashville was normalized as 1, and then the differences in different locations can be compared with this reference city. Fig.1 shows that New Orleans and Seattle have the longest free cooling period and therefore, in the result of the total energy cost, these two locations have the lowest energy cost among these five locations when the floor-supply displacement ventilation is used. The energy saving by using displacement ventilation is significant in cooling operation. In addition, for the locations with a high electricity cost, this saving is more evident. Moreover, the cost of natural gas per thermal unit is lower than the electricity in these five locations as it should be. The additional cost for natural gas consumed by displacement ventilation during heating operation can be easily compensated by the electricity saving in cooling operation. Therefore, in this study, displacement ventilation system has a lower annual energy cost than mixing ventilation system in all the locations except Seattle. Since the ratio of the cost of electricity to gas is nearly 2:1 in Seattle that is mush lower than that in Portland (7:1), the saving in electricity cost cannot compensate the extra cost in gas for the displacement ventilation system. In short, using floor-supply displacement ventilation cannot save the energy cost in the location with high price in gas.

CONCLUSIONS

The present investigation studied the energy consumption of the floor-supply displacement and mixing ventilation systems in a large industrial workshop at five typical weather regions in the continental U.S.: Seattle, WA (maritime), Portland, ME (cold), Phoenix, AZ (hot and dry), New Orleans, LA (hot and humid) and Nashville, TN (moderate). The investigation accounted for the most important characteristics of the displacement ventilation system, such as air temperature stratification and higher supply air temperature. Both factors contribute to the reduction of the energy consumption by the air handling system using displacement ventilation because of the improved COP of chiller and long free cooling period by using outside air.

The study showed that the floor-supply displacement ventilation system may use more fan and boiler energy but less chiller energy than the mixing ventilation system. The total energy used is slightly less with displacement ventilation. The displacement ventilation system can reduce energy use only for cooling but uses more energy for heating. However, displacement ventilation system without additional dehumidification measures could lead to a high indoor humidity in summer in regions with humid climate (such as New Orleans and Nashville). This is because the displacement ventilation system does not have the capacity to dehumidify the outdoor air at a higher coil temperature than the mixing ventilation system. If the coil temperature is the same for the two systems, the indoor humidity is similar. Then the displacement ventilation system will use more energy than the mixing ventilation system, but a better air quality.

Our analysis on the over energy cost suggested that the displacement ventilation should be employed in most U.S. climate regions. The energy saving by using displacement ventilation is significant in cooling operation. The saving is especially evident for locations with a high electricity cost and low natural gas cost. Only in Seattle, when the gas price in gas is high, the displacement ventilation has a higher energy cost because displacement ventilation consume more energy in winter heating due to its indoor air temperature stratification.

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NOMENCLATURE

A_i = area of the zone surfaces

c_p = specific heat capacity of air

C_z = capacitance of energy storage in air

h_i = convective heat coefficient of the zone surfaces

$\dot{m}_i$ = mass flow rate due to inter-zone air mixing

$\dot{m}_{\mathrm{inf}}$ = mass flow-rate of the air infiltration

$\dot{m}_{sys}$ = mass flow-rate of the ventilation system

$\dot{Q}_i$ = internal convective loads from people, computers, etc.

$\dot{Q}_{sys}$ = air system output

T_{inf} = temperature of infiltrated air

T_s = temperature of the supplied air

T_{si} = temperature of the zone surfaces

T_z = air temperature of the zone

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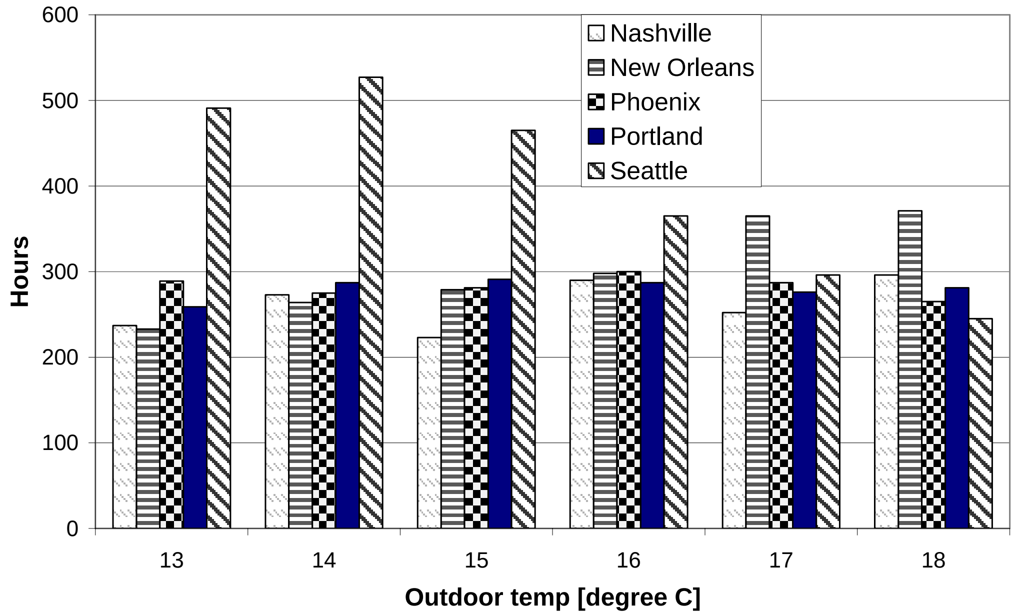


Fig. 1 Hourly outdoor temperature distribution at the five cities in the United States.

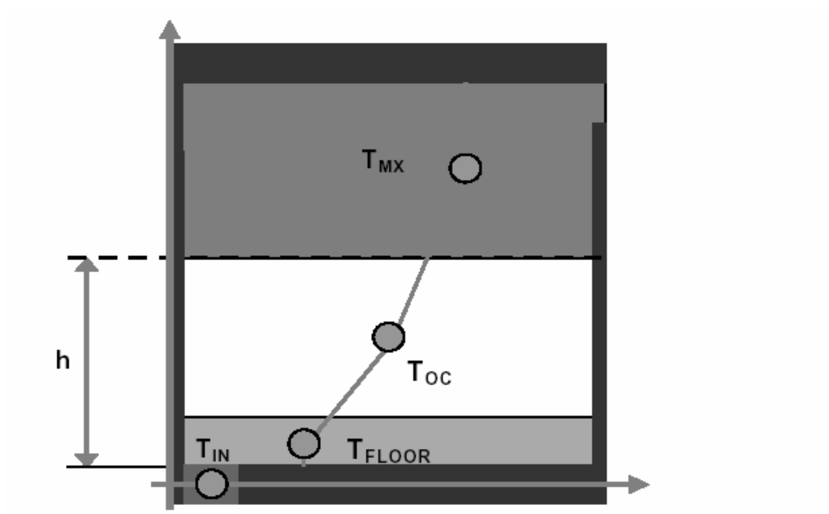


Fig. 2 Schematic representation of the three temperature points and temperature gradients in a room with floor-supply displacement ventilation

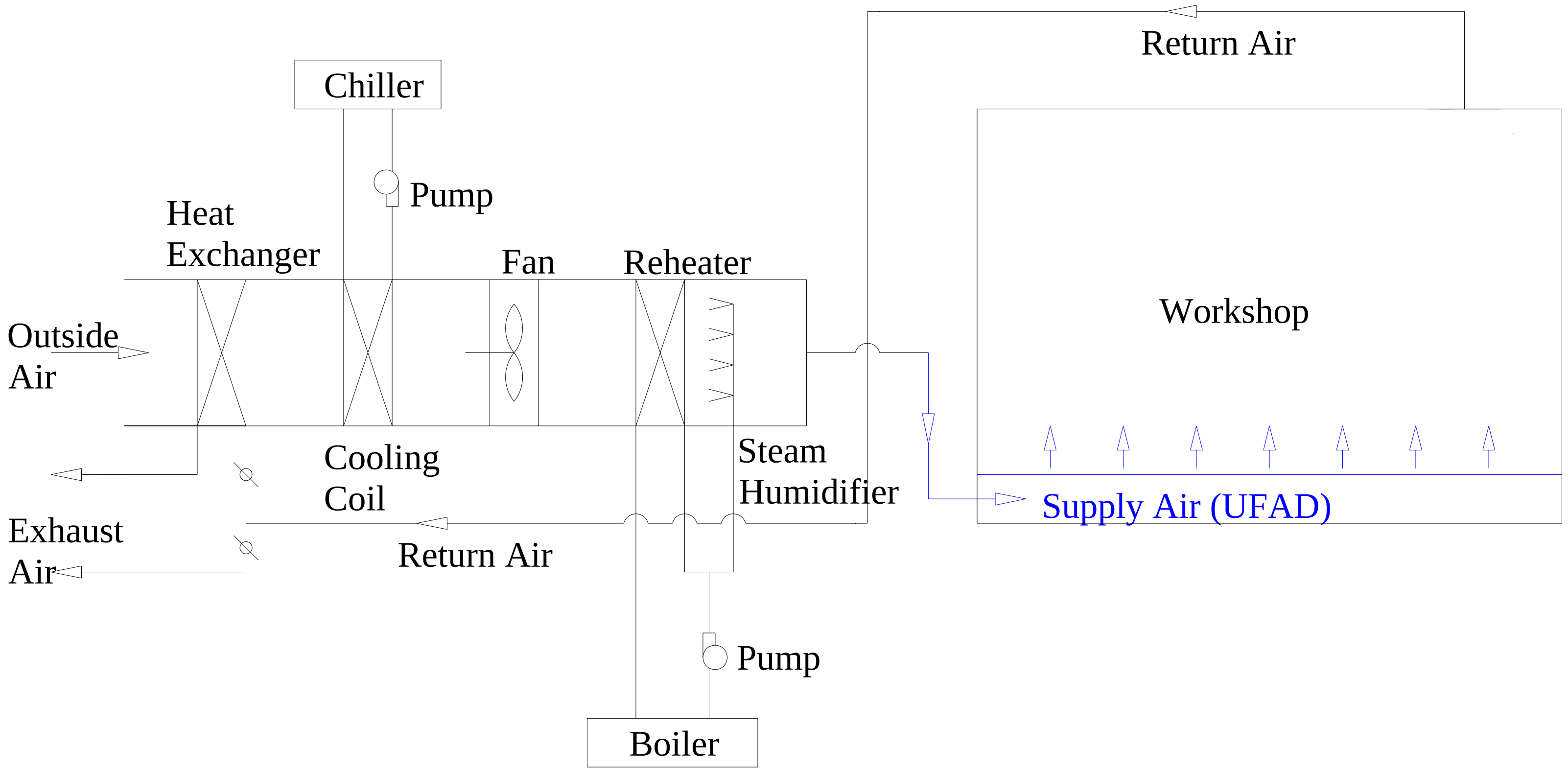


Fig. 3 Air handling system for the workshop with displacement ventilation

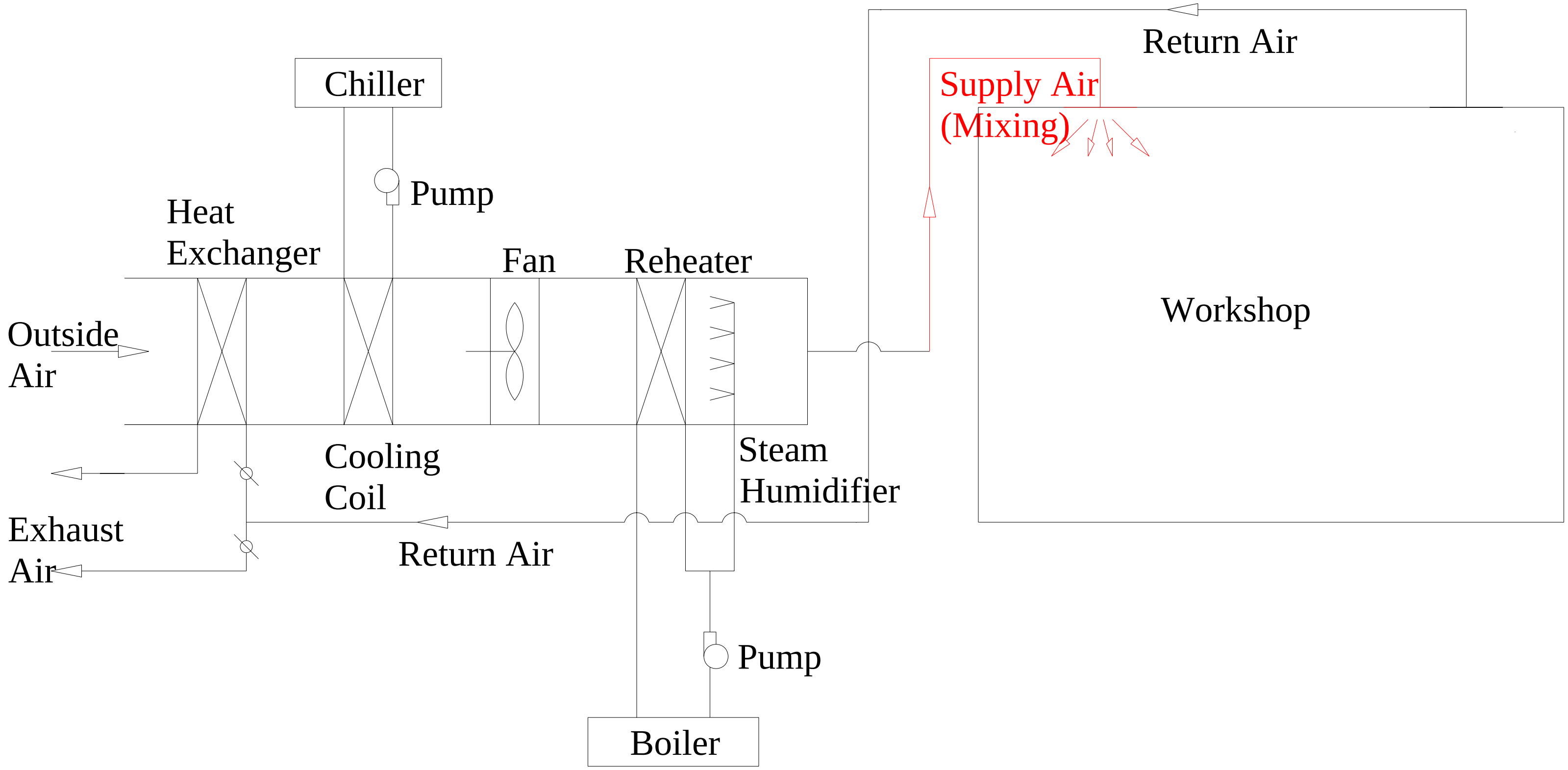
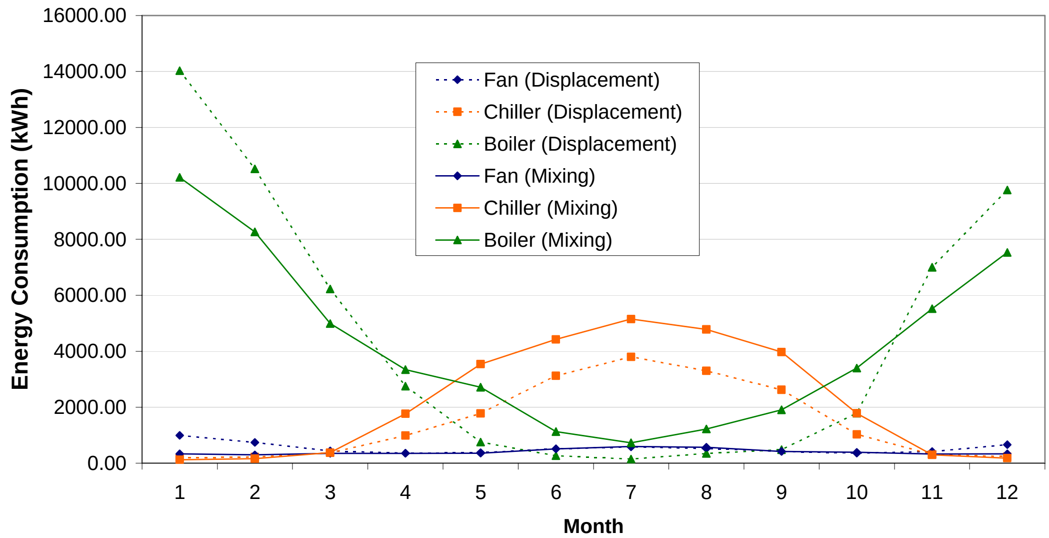
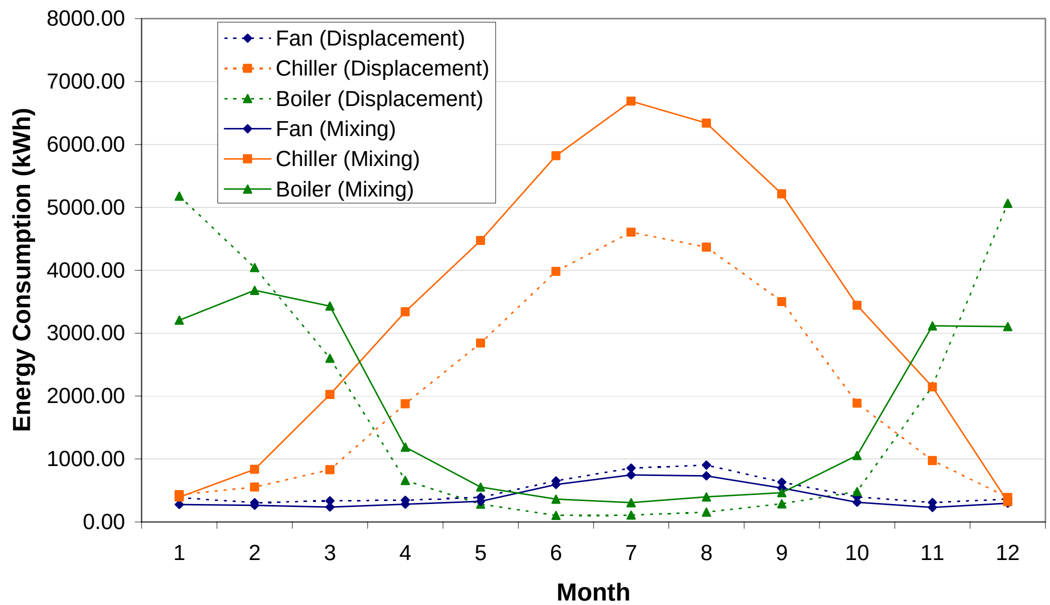


Fig. 4 Air handling system for the workshop with mixing ventilation



(a) Nashville, TN



(b) New Orleans, LA

Fig. 5 The monthly energy consumption of the workshop

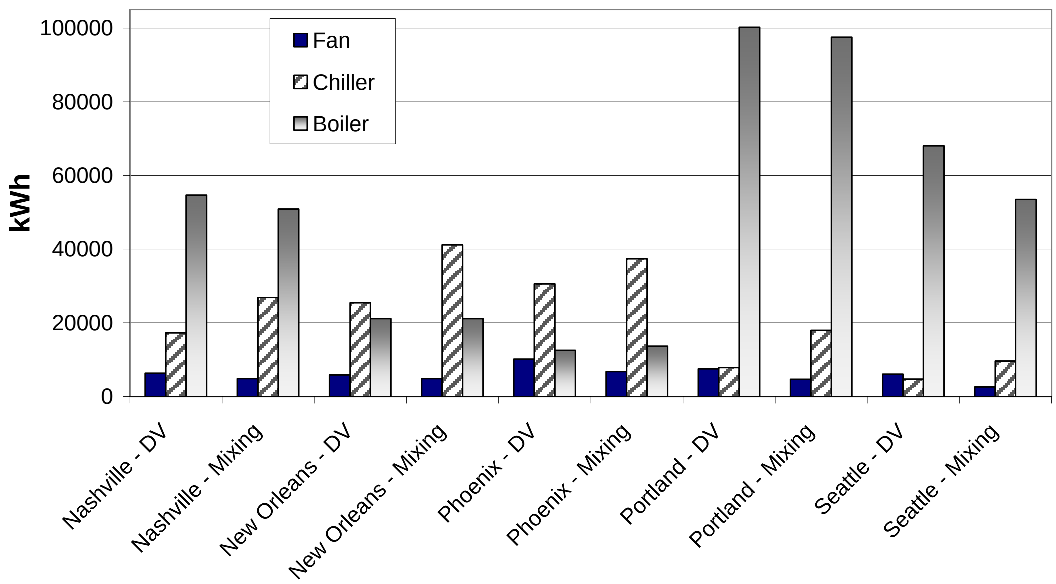


Fig. 6 Comparison of annual energy consumption of the workshop with floor-supply displacement ventilation with that with mixing ventilation in the five locations

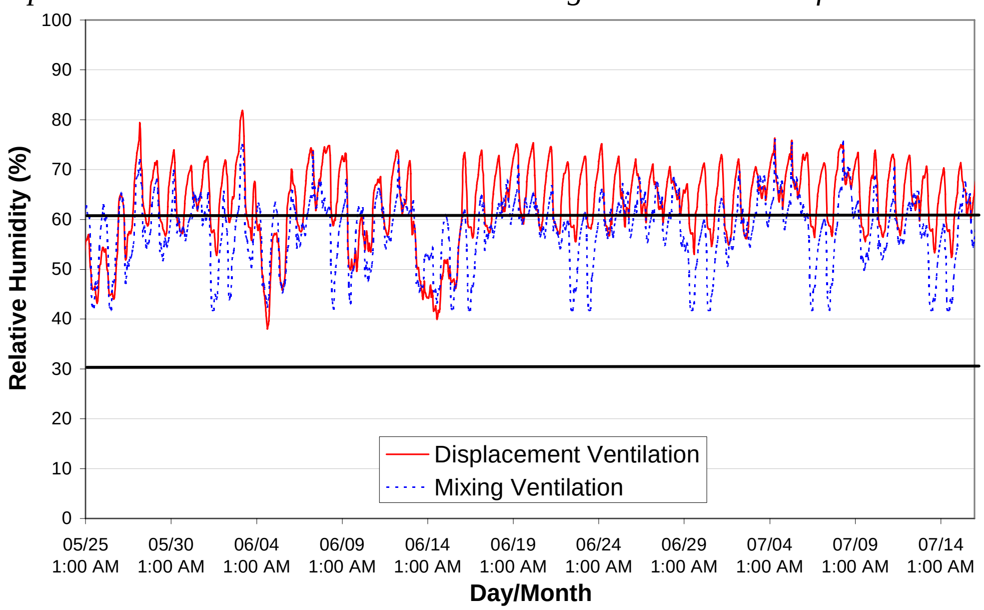


Fig. 7 Comparison of the relative humidity in the workshop with floor-supply displacement ventilation at higher coil temperature with that with mixing ventilation at lower coil temperature in Nashville, TN

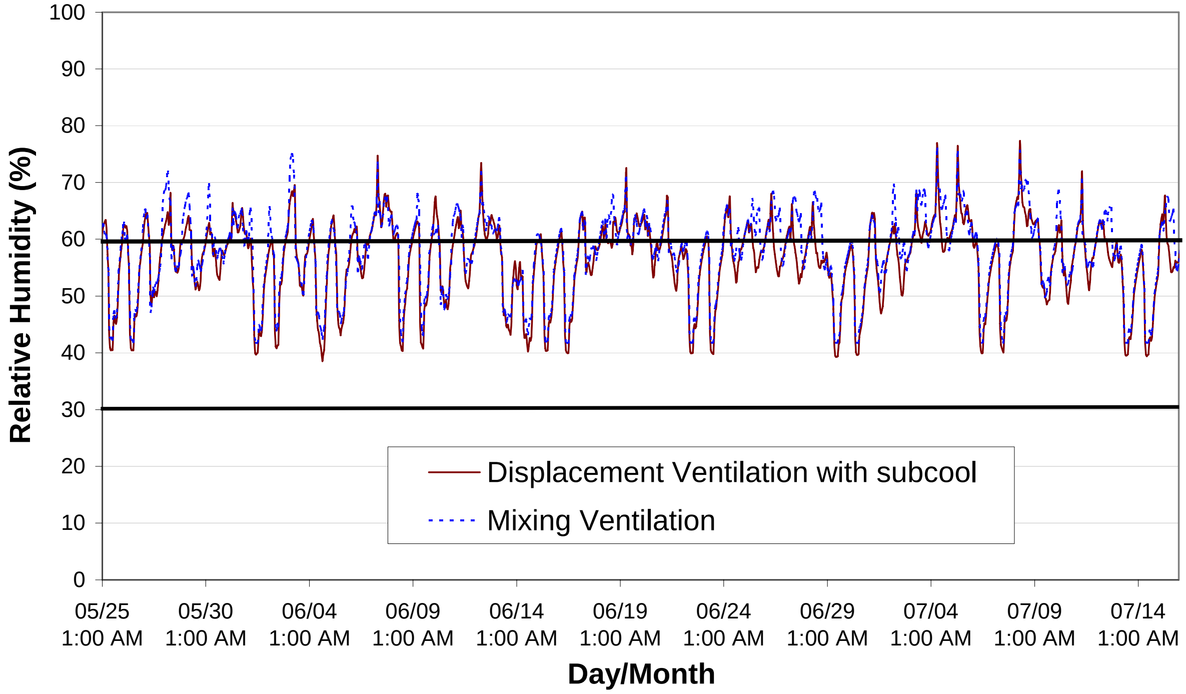


Fig. 8 Comparison of relative humidity in the workshop ventilated with the two ventilation systems with the same coil temperature in Nashville, TN.

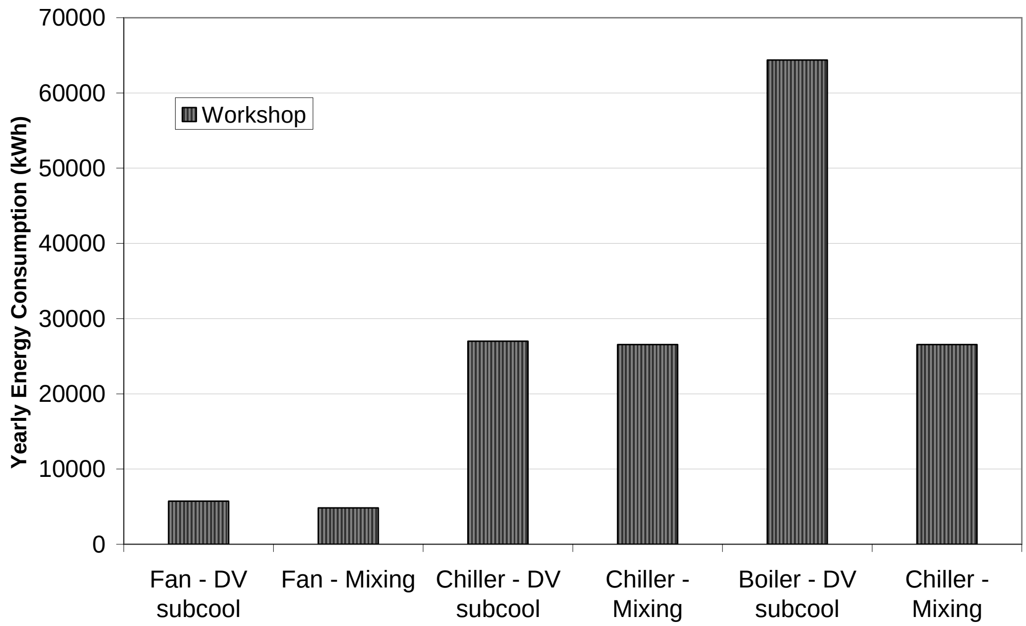


Fig. 9 Comparison of the annual energy consumption of the workshop with the two ventilation systems with the same cooling coil temperature

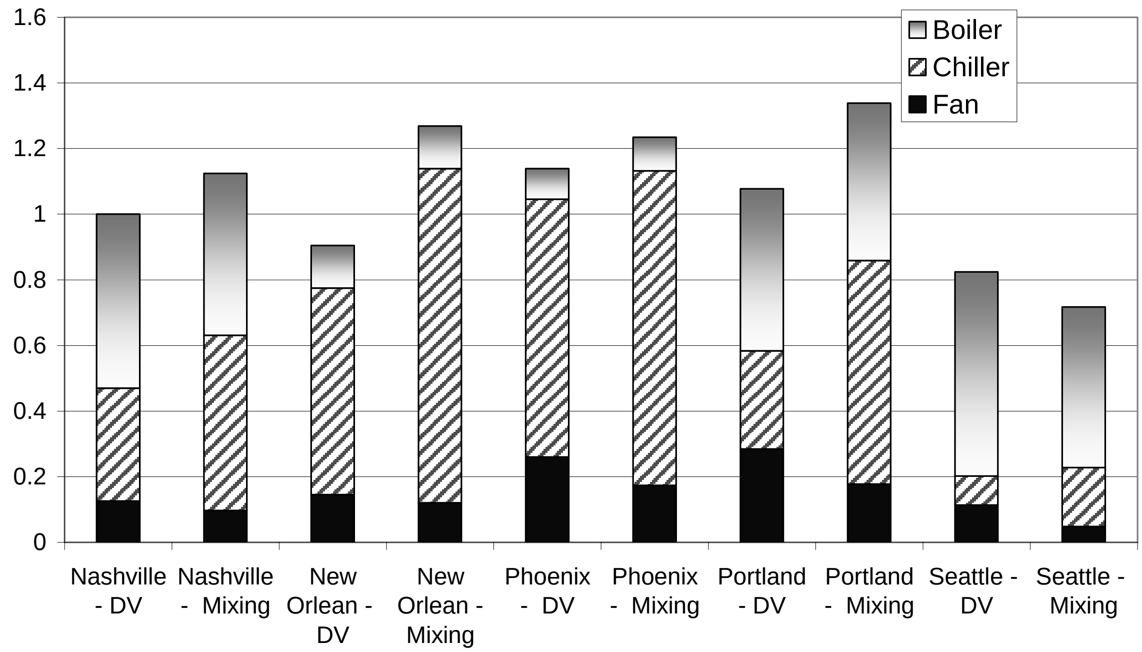


Fig. 10 The normalized total energy cost at the five locations

Table 1 The characteristics and thermal conditions of the workshop

<table>
<tr><td colspan="3">Space Type</td><td>Workshop</td></tr>
<tr><td colspan="2" rowspan="3">Space Size</td><td>Length (m)</td><td>26.2</td></tr>
<tr><td>Width (m)</td><td>21.0</td></tr>
<tr><td>Height (m)</td><td>4.5</td></tr>
<tr><td rowspan="4">Exterior Envelope</td><td rowspan="2">Wall U-Value (W/m^2-K)</td><td>Seattle & Portland</td><td>0.72</td></tr>
<tr><td>Phoenix, New Orleans, & Nashville</td><td>0.96</td></tr>
<tr><td rowspan="2">Glazing</td><td colspan="2">Double Glazing: U = 4.6 W/m^2-K</td></tr>
<tr><td colspan="2">50% of Exterior Wall Area is Glazed</td></tr>
<tr><td rowspan="3">Internal Load (W)</td><td rowspan="3">Sensible</td><td>Equipment</td><td>3362</td></tr>
<tr><td>Lights</td><td>5502</td></tr>
<tr><td>People</td><td>112(105)=11760</td></tr>
<tr><td colspan="2" rowspan="2">Room Temperature Set Point (°C)</td><td>Cooling</td><td>25</td></tr>
<tr><td>Heating</td><td>23</td></tr>
<tr><td rowspan="4">Supply Air Temperature (°C)</td><td rowspan="2">DV</td><td>Cooling</td><td>19</td></tr>
<tr><td>Heating</td><td>32</td></tr>
<tr><td rowspan="2">Mixed</td><td>Cooling</td><td>13</td></tr>
<tr><td>Heating</td><td>32</td></tr>
<tr><td rowspan="4">Supply Air Humidity Ratio ($g_{moist\ air}$ / $kg_{dry\ air}$)</td><td rowspan="2">DV</td><td>Cooling</td><td>11.5</td></tr>
<tr><td>Heating</td><td>9.5</td></tr>
<tr><td rowspan="2">Mixed</td><td>Cooling</td><td>8.5</td></tr>
<tr><td>Heating</td><td>9.5</td></tr>
</table>

Table 2. The ratio of the cost of electricity to gas at the five locations

		Energy cost ratio (per BTU)	
State	**City**	**Electricity**	**:Natural Gas**
TN	Nashville	16.52	: 8.05
LA	New Orleans	20.54	: 5.09
AZ	Phoenix	21.3	: 6.19
ME	Portland	31.46	: 4.09
WA	Seattle	15.46	: 7.59