Laboratory Building Energy Analysis

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INTRODUCTION

Because laboratories use a high amount of energy—often more than five times as much per square foot as an office building—it is important to find cost-effective ways to reduce their energy use and costs. This energy analysis was performed to evaluate selected energy efficiency measures for a generic laboratory building. Using a computer model, the analysis compared results for a base case laboratory with results for laboratories in four different climates—those of Atlanta, Denver, Minneapolis, and Seattle.

The analysis focused on efficiency strategies designed to reduce the considerable amount of energy used in ventilating, cooling, and heating laboratory buildings. The impacts of humidity controls and plug load assumptions on energy loads were also considered. Results are presented and discussed in this report.

Enermodal Engineering, Inc., performed the analysis, along with staff in the U.S. Department of Energy (DOE) National Renewable Energy Laboratory (NREL). This study was conducted in support of "Laboratories for the 21st Century," a joint program of the U.S. Environmental Protection Agency (EPA) and DOE through the DOE Federal Energy Management Program in the Office of Energy Efficiency and Renewable Energy. "Labs 21" encourages the design, construction, and operation of safe, sustainable, high-performance laboratories.

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EXECUTIVE SUMMARY

This study analyzes the effects of energy efficiency measures in a simplified laboratory model in the climates of Minneapolis, Denver, Seattle and Atlanta. The analysis establishes a base case to which certain energy efficiency strategies are applied, as well as changes in humidity controls and plug load assumptions. The analysis compares energy use and costs, chilled water and hot water system sizing, and life-cycle costs.

The laboratory model is a four-story, 100,000-square-foot (sf) building with 70% of its area devoted to laboratories. The window-to-wall ratio is 0.25 and the windows are distributed equally around the building. The insulation levels and window energy performance values meet the ASHRAE 90.1-99 building energy standard, as does the lighting power density. The equipment power density for the building is 9 watts per square foot (W/sf) under design conditions. The cooling setpoint is 74°F and the heating setpoint is 72°F. A constant-volume reheat system serves the building with a maximum relative humidity of 60% and a minimum relative humidity of 30%. Outside air ventilation is supplied at a minimum of 2 cubic feet per minute per square foot (cfm/sf) by premium-efficiency fans. The building has a central plant with water-cooled chillers and hot water boilers. The chillers are rated at 0.5 kilowatt (kW)/ton and the boilers are 80% efficient. All pumps are high efficiency and have variable-speed drives.

Measured and predicted energy use data from laboratory case studies were used to tune the simulation models. The simulation models are within 5% of the electricity use measured in the Labs 21 case studies (2001). However, gas usage in the simulations is comparatively high, and the case studies do not provide sufficient detail to explain the differences. We estimated the space heating load from the outside air ventilation load and found good agreement with simulation results when we excluded humidity controls. We concluded that the difference in energy use is attributable to humidity controls, weather, internal loads, operating hours, and the high density of laboratory space.

The table below shows the simulation results for building energy use. Seattle has the mildest climate and the lowest annual energy cost; Atlanta has the lowest total energy use. Electricity rates of $0.03 per kilowatt-hour (kWh), $7/kW on-peak, and $4/kW off-peak were used. On-peak hours are 8 a.m. to 10 p.m., Monday through Friday. Gas rates are $0.60/therm.

Building Energy Use

	Peak Demand (W/sf)	Electricity (kWh/sf/yr)	Gas (kBtu/sf/yr)	Total (kBtu/sf/yr)	Annual Electricity Cost ($/sf)	Annual Gas Cost ($/sf)	Total Energy Cost ($/sf)
Minneapolis	20	84	861	1125	$4.30	$5.20	$9.50
Denver	16	80	792	1043	$4.00	$4.80	$8.80
Seattle	15	77	431	673	$3.90	$2.60	$6.50
Atlanta	20	92	362	652	$4.70	$2.20	$6.90

Energy efficiency strategies included reducing the air flow during unoccupied periods, a variable-air-volume (VAV) system; lower static pressure drop in the air distribution system; energy recovery by enthalpy wheels, heat pipes, and run-around loops; evaporative cooling; and more accurate accounting for plug loads. Other strategies, such as reducing lighting loads and solar heat gain, were not addressed. Furthermore, we did not quantify the impact of high-efficiency equipment such as chillers, boilers, fans, pumps, and motors.

Results show that the most efficient measures are the same for all climates with the exception of Denver's, where evaporative cooling is also cost-effective. Predicted energy savings differ from climate-to-climate. Based on the simulation results, we conclude the following:

- Using a VAV system (e.g., VAV fume hoods) rather than a constant-volume system has the potential to reduce fan energy and energy for space cooling and heating. Energy cost savings average $1/sf in all four climates.
- Some form of energy recovery should always be considered. Because of the sensible and latent energy recovery achieved with an enthalpy wheel, it is the most efficient of the energy recovery alternatives considered here.
- The increase in fan energy from energy recovery ventilation systems is not offset by the reduction in space cooling. However, the lower heating energy use more than compensates for the increase in fan energy.
- Energy recovery can potentially reduce the size of the heating and cooling equipment, and a VAV system has the potential to reduce the size of the heating system. The first-cost savings can cover a large portion of the cost of the energy efficiency strategy.
- Because of the high ventilation requirements in laboratory buildings, the air distribution system should be optimized to minimize pressure drop through the system and reduce energy use.
- Humidity control is energy-intensive and should be carefully integrated into the control strategies to minimize reheat and subcooling.
- Plug loads and internal gains from plug loads should be accurately assessed in order to design the mechanical system and determine power requirements. Significant increases in first costs and operating costs result from assuming too high a design load.

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CHAPTER 1. BUILDING ENERGY SIMULATION BASE CASE

Staff in the National Renewable Energy Laboratory Federal Energy Management Program (FEMP) contracted with Enermodal Engineering, Inc., to analyze energy efficiency measures for the Laboratories for the 21st Century ("Labs 21") program. The purpose of the study was to identify cost-effective energy efficiency measures for a generic laboratory in different climates. Task 1 is presented in this chapter; it describes the base case simulation model and building energy simulation results.

1.1 Energy Simulation Program

The building energy analysis was performed using the DOE-2.2 building energy simulation program. DOE-2.2 is an hourly simulation program that was developed by James J. Hirsch and Associates and Lawrence Berkeley National Laboratory (PC DOE-2.2-041h, 2002). Version 41 of DOE-2.2 includes models for energy recovery ventilation, improved chiller part-load operation, and water-side economizers.

1.2 Climates Selected

Four climates associated with four metropolitan areas were selected: Minneapolis, Denver, Seattle, and Atlanta. Seattle has the mildest climate conditions, Minneapolis the coldest, and Atlanta the warmest. Denver has a dry climate; Atlanta and Minneapolis are humid in summer. Table 1.1 shows winter and summer design temperatures and heating degree days (HDD) and cooling degree days (CDD) from the weather tape.

Table 1.1 Design Conditions

Design Conditions	Minneapolis	Denver	Seattle	Atlanta
Winter design temperature (°F)	-11	3	28	23
Summer design temperature (°F)	88/77	90/59	81/64	91/74
HDD65 (F-days)	8002	6113	4867	3089
CDD65 (F-days)	634	566	127	1611

For the simulations, TMY2 hourly weather data are used for all climates except Denver's. No TMY2 data were available for Denver, so TMY ("typical meteorological year") data were used. Altitude is not included in the TMY tape for Denver, so it was input to the DOE-2 file.

1.3 Utility Rates

Laboratory buildings are generally high-demand buildings with greater than 10 W/sf of peak demand. Electricity rate structures vary around the country for such high-demand buildings (>500 kW peak demand). Typically, there is an energy charge ($/kWh) and a peak demand charge ($/kW), and charge rates may vary with the time of day, time of year, amount of energy used, or all three.

For this study, we assumed a constant energy charge of $0.03/kWh, plus an on-peak demand charge of $7/kW and an off-peak demand charge of $4/kW. On-peak hours are 8 a.m. to 10 p.m., Monday through Friday. A fixed monthly charge of $15 was also included. For natural gas, a rate of $0.60/therm was assumed with a fixed monthly charge of $15.

1.4 Simulation Model

1.4.1 Shell

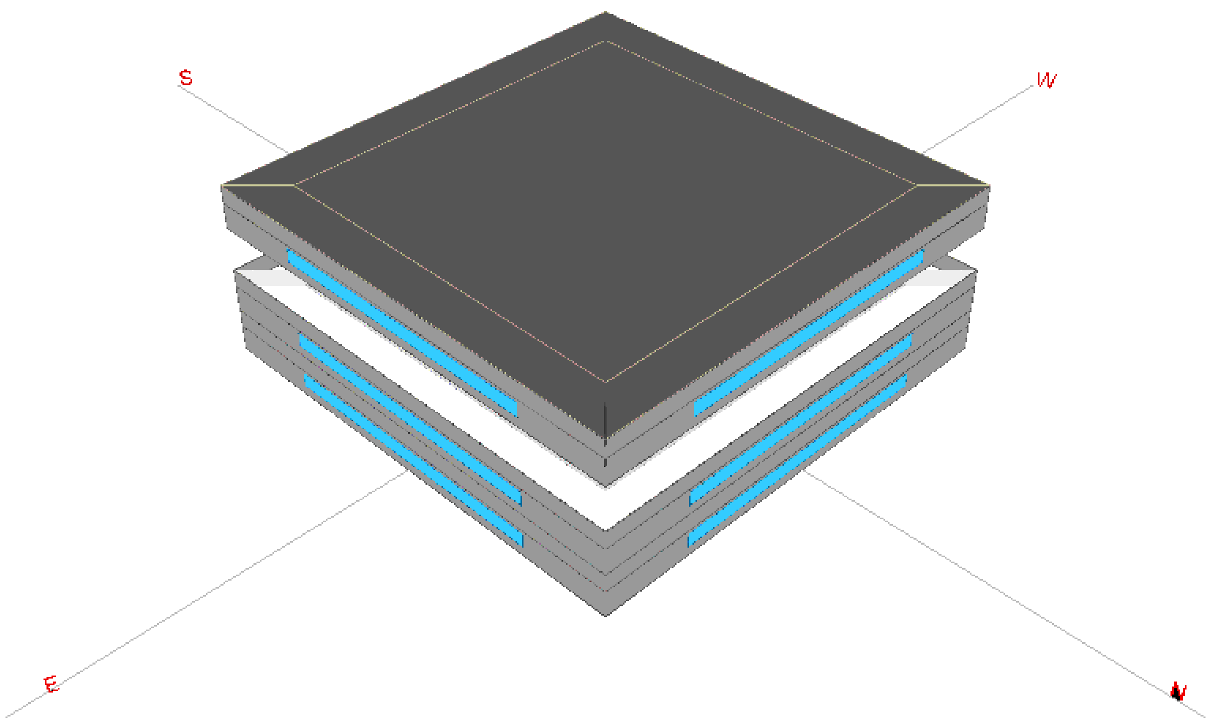


Figure 1.1a EQUEST rendering of building.

The model laboratory building has four stories above grade (Figure 1.1a). It has a total of 100,000 square feet (sf) with 25,000 sf on each floor. The floor-to-ceiling height is 9 feet and the floor-to-floor height is 15 feet. The building has a window-to-wall ratio of 0.25, and windows are distributed equally around the building. Table 1.2 provides details on the building shell, such as wall and window areas, and insulation levels. The insulation levels and window performance values are based on ASHRAE 90.1-99 prescriptive requirements for Minneapolis and Denver. The only difference in the standard requirements for Seattle and Atlanta is R-13 insulation in the walls. To simplify the simulation model, the higher wall insulation level was used for all locations.

Table 1.2 Building Details

Building Component	Model Assumption
Floor Area (sf)	100,000
Number of Stories	4
Floor-to-floor Height (ft)	15
Floor-to-ceiling Height (ft)	9
Net Wall Area (sf)	
North	9487
East	9487
South	9487
West	9487
Window Area (sf)	
North	593
East	593
South	593
West	593
Window-to-Wall Ratio	0.25
Window Shading	None
Wall Construction	
Insulation	R-13+R-3.8 c.i.
Total U-Value	0.084
Roof Construction	Built-up
Insulation	R-15 c.i.
Total U-Value	0.063
Slab	8” Concrete
Total R-Value	No Insulation
Window U-Factor (Btu/hr-ft^2-F)	0.57
Window Solar Heat Gain Coefficient	0.39 (all) 0.49 (north)

1.4.2 Internal Loads and Lighting

Rather than assume that 100% of the area in the building is laboratory space, we included ancillary spaces to make the model more realistic. The building space is thus divided into 70% laboratory area, 20% corridor, 5% restrooms, and 5% mechanical and electrical (ME) rooms. It is occupied between 8 a.m. and 10 p.m., and occupancy varies with time of day, as do the equipment and lighting schedules (Figures 1.2a, 1.2b, and 1.2c).

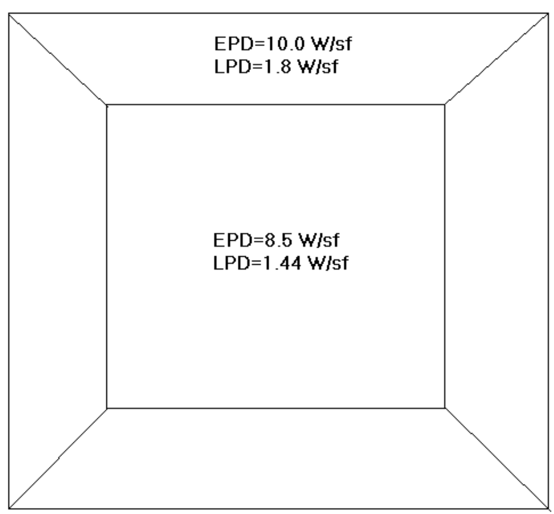


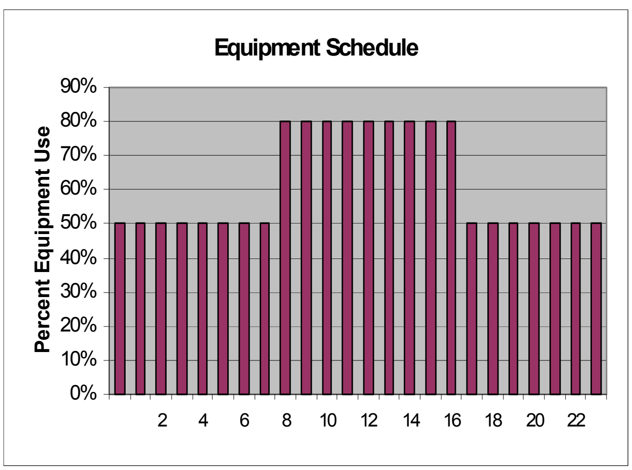
Figure 1.1b Perimeter and core zones in the building and their associated equipment power densities (EPD) and lighting power densities (LPD).

The equipment power density (i.e., plug loads) and lighting power density vary with space type (Table 1.3). We assumed that 81% of the perimeter is laboratory space and 19% consists of support spaces. In the core, we assumed that 67% is laboratory space and 33% is support spaces. The average equipment power density for the support spaces is 1.25 W/sf. Performing an area-weighted calculation results in 10 W/sf of equipment plug loads in the perimeter zones and 8.5 W/sf in core zones (Figure 1.1b). The area-weighted average equipment load for the building is 8.8 W/sf; adjusted for the equipment schedule, it is 7 W/sf.

Table 1.3 Internal Loads

Internal Load	% Area in Building	Model Assumption
Equipment Power Density (W/sf)		
Laboratories	70%	12 W/sf
Corridor and Lobby	20%	1.25 W/sf
Restrooms	5%	0.5 W/sf
ME Rooms	5%	2 W/sf
Equipment Schedule		1-8: 50%, 9-17: 80%, 18-24: 50%
Number of Occupants (sf/per)		275
Occupancy Schedule		1-8: 5%, 9-10: 20%, 11-12: 95%, 13-14: 50%, 15-18: 95%, 19: 30%, 20-22: 20%, 23-24: 5%
Lighting Power Density (W/sf)		
Laboratories	70%	1.8 W/sf
Corridor and Lobby	20%	0.7 W/sf
Restrooms	5%	1.0 W/sf
ME Rooms	5%	1.0 W/sf
Lighting Schedule		1-7: 10%, 8: 50%, 9-18: 90%, 19-22: 50%, 23-24: 10%
Ballast Type		Electronic
Ballast Power Factor		0.9

The lighting power density assumptions are based on ASHRAE 90.1-99 prescriptive requirements. The perimeter zones have a lighting power density of 1.8 W/sf, and the core zones are 1.44 W/sf (Figure 1.1b). The lighting power density for the building under design conditions is 1.5 W/sf; adjusted for the lighting schedule, it is 1.4 W/sf.



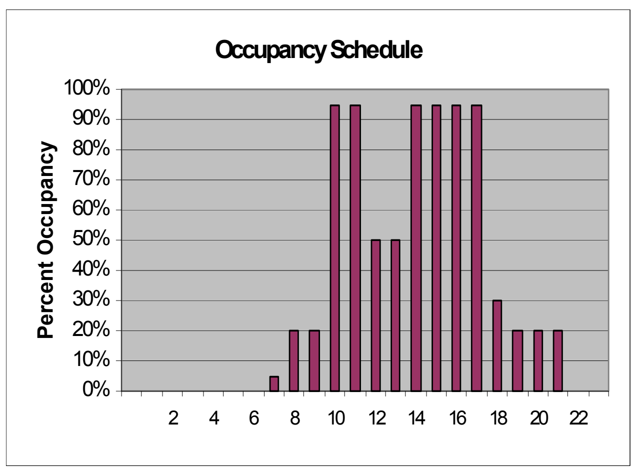


Figure 1.2a and b Equipment and occupancy schedules. The time represents the hour before the number (e.g., 8 a.m. represents 7 a.m. to 8 a.m.).

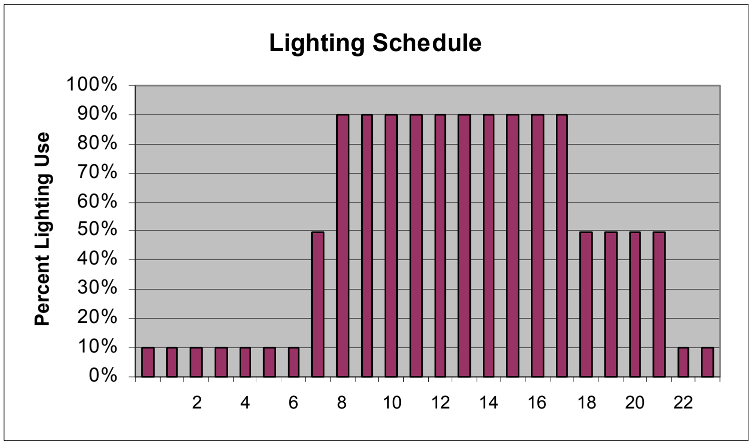


Figure 1.2c Lighting schedule. The time represents the hour before the number.

1.4.3 Mechanical System

For the mechanical system, the assumptions (see Table 1.4) are nearly identical for each climate except for the sizes of the heating, ventilation, and air-conditioning (HVAC) equipment. The HVAC system is a constant-volume reheat system. The building has four perimeter zones and a core zone on each floor (Figure 1.1b).

The room temperature is maintained at 72°F for heating and at 74°F for cooling. The relative humidity is controlled at a minimum of 30% and a maximum of 60%. Supply air is 100% outside air at a minimum of 2 cfm/sf in all spaces, although in an actual building the support spaces would be handled separately. The supply air temperature is reset based on the outside air temperature and the zone calling for the maximum heating or cooling. The design air temperature leaving the main cooling and heating coils is 55°F. As the outside air temperature drops below 80°F, the supply air temperature is adjusted up and is set as high as 65°F when the outside air is 60°F or lower.

There are three supply air fans and a manifold exhaust system. The perimeter zones are served by one fan, as are the first- and second-floor core zones and the third- and fourth-floor core zones. The minimum design air flow of 2 cfm/sf is based on having a fume hood every 450 sf. The fume hoods are assumed to have an average sash height of 18 inches and are 6 feet wide. A face velocity of 100 feet per minute (fpm) is maintained. The average exhaust is 900 cfm, which translates into an exhaust rate of 2 cfm/sf. We allowed DOE-2 to size the fans and found that the internal gains are the determining factor in calculating the design flow rates, not the exhaust requirements. The perimeter zones have higher internal gains than the core zones; therefore, the design flow rates are higher in the perimeter zones (Table 1.5). Note that Denver's high altitude results in higher flow rates than those of the other climates.

The fans are specified to meet ASHRAE 90.1-99 requirements of 0.8 W/cfm for supply fans, and they are assumed to be vane axial with premium motors. The supply fans experience 5.2 inches (water gauge) (in. w.g.) of total static pressure and have a fan-plus-motor efficiency of 76% (the fan is 80% efficient and the motor is 95% efficient).

The total static pressure in laboratories is often 6 in. w.g. or more, which makes it difficult to meet ASHRAE 90.1-99 fan requirements. The exhaust fans experience 2 in. w.g. of total static pressure and also have a total efficiency of 76%. The exhaust fans use 0.3 W/cfm.

The chillers are high-efficiency (0.5 kW/ton), centrifugal water-cooled chillers. The DOE-2.2 part-load efficiency curve was used to predict performance. High-efficiency chillers should be specified to optimize performance within the range in which the chiller most frequently operates. The chilled water is supplied at 44°F, and outside air reset controls are assumed. The supply temperature of the chilled water is set up as the outside air temperature drops below 80°F. The chilled water temperature is set at 55°F when the outside air is 60°F or below. The primary and secondary pumps have variable-frequency drives (VFDs) and high-efficiency motors. The cooling tower has two cells and two-speed fans and is sized by DOE-2.

Standard gas-fired boilers with an efficiency of 80% are assumed for water heating. Although high-efficiency boilers are not included as one of the efficiency measures, they are worth considering. All pumps have VFDs and high-efficiency motors.

Table 1.4 Mechanical System

Mechanical System	**Model Assumptions**
System	Constant-volume w/ hot water reheat
Supply Air Handling Units	3 CV units; 2 cfm/sf minimum, 76% fan+motor efficiency
Exhaust Air Handling Units	3 CV units (manifold exhaust), 76% fan+motor efficiency
Supply Static Pressure	5.2 in. w.g.
Exhaust Static Pressure	2 in. w.g.
OA Ventilation Rate	100% (minimum 2 cfm/sf)
Humidification	30% winter; 60% summer
Cooling Thermostat Setpoint	74°F
Chillers	Centrifugal chillers (water cooled)
Chilled Water Supply Temperature	44°F
Chiller Tonnage and Number	200 (2)
Chiller kW/ton	0.5 kW/ton
Chiller Part Load Performance	DOE-2.2 Default
Primary Chilled Water Pumps	VFD (2)
Secondary Chilled Water Pumps	VFD (2)
Cooling Towers	1 – open, 1 cell, 2 speed
Condenser Water Pumps	VFD (2)
Economizer Cycle	Air-side
Heating Thermostat Setpoint	72°F
Heating System	Hot water boilers (2) supplying main air handlers
Hot Water Supply Temperature	135°F
Supply Air Temperature Leaving Main Hot Water Coil	55°F
Boiler Efficiency	80%
Primary Hot Water Pumps	VFD (2)
Secondary Hot Water Pumps	VFD (2)

The sizes of the chillers, pumps, cooling tower, and boilers depend on the climate. Design conditions were assumed with 100% equipment loads, lighting use and occupancy. DOE-2.2 design-day calculations were included to determine the size of the HVAC equipment. However, DOE-2.2 does not adequately size the chillers and boilers, because peak conditions occur in Minneapolis and Atlanta when there is a significant latent load at off-design load conditions (i.e., lower dry bulb and higher wet bulb temperatures). Multiple runs were performed to determine the chiller and boiler sizes that minimized the number of hours in which the zones are underheated and undercooled to fewer than 20 hours. The design cooling and heating capacities for each zone were then fixed, and DOE-2 sized the loops, associated pumps, and cooling tower. Table 1.5 lists the sizes of the equipment used for each climate.

Table 1.5 HVAC Equipment Sizes

Equipment	Minneapolis	Denver	Seattle	Atlanta
Supply and Exhaust Fans	Core: 2 cfm/sf Perimeter: 2.7 cfm/sf	Core: 2.4 cfm/sf Perimeter: 3.2 cfm/sf	Core: 2 cfm/sf Perimeter: 2.6 cfm/sf	Core: 2 cfm/sf Perimeter: 2.7 cfm/sf
Chillers	650 tons (2)	400 tons (2)	350 tons (2)	650 tons (2)
Chiller Primary Pumps	1560 gpm [(a)] (2), 24 ft	960 gpm (2), 24 ft	840 gpm (2), 20Ft	1560 gpm (2), 24 ft
Chiller Secondary Pumps (2)	2600 gpm (total), 50 ft	1600 gpm (total), 50 ft	1350 gpm (total), 50 ft	2615 gpm (total), 50 ft
Condenser Water Pumps (2)	3560 gpm (total), 74 ft	2190 gpm (total), 74 ft	1920 gpm (total), 74 ft	3560 gpm (total), 74 ft
Cooling Tower (2 cells w/ 2-speed fans)	1500 tons	920 tons	800 tons	1500 tons
Boilers	450 hp (2)	400 hp (2)	250 hp (2)	250 hp (2)
Hot Water Primary Pumps	1500 gpm (2), 6 ft	1340 gpm, 6 ft (2)	840 gpm (2), 6 ft	840 gpm (2), 6 ft
Hot Water Secondary Pumps (2)	3580 gpm (total), 38 ft	3200 gpm (total), 38 ft	2035 gpm (total), 38 ft	2140 gpm (total), 38 ft

[(a)]gpm = gallons per minute.

1.5 DOE-2.2 SIMULATION RESULTS

1.5.1 Tuning the Models

The laboratory building was simulated in each of the climates using the DOE-2.2 building energy simulation program. Table 1.6 gives the peak electricity demand and energy intensities for the building in each climate. For reference, office buildings typically operate at less than 10 W/sf peak demand and use less than 100,000 British thermal units (100 kBtu)/sf. Laboratories have been known to consume 5-10 times that much energy.

Table 1.6 Energy Intensities

	Peak Demand (W/sf)	Electricity (kWh/sf/yr)	Gas (kBtu/sf/yr)	Total (kBtu/sf/yr)
Minneapolis	20	84	861	1125
Denver	16	80	792	1043
Seattle	15	77	431	673
Atlanta	20	92	362	652

FEMP has prepared a number of case studies on laboratories around the country. Table 1.7 presents measured data from two of these case studies for comparison with the model developed for this analysis. Both laboratories in the case studies employed significant energy efficiency measures. The predicted energy savings for the Process and Environmental Technology Laboratory are 40%, and the electricity energy savings for the Fred Hutchinson laboratory are 33%.

Table 1.7 Energy Use in Case Studies

Case Study	Annual Electricity Use (kWh/sf/yr)	Annual Gas Use (kBtu/sf/yr)	Total Site Energy (kBtu/sf/yr)	Notes
Process and Environmental Technology Laboratory, Albuquerque, NM	43	122	257 (428 before efficiency measures)	Original study predicted a total of 595 kBtu/sf/yr
Fred Hutchinson Cancer Research Center, Seattle, WA	49 (73 before efficiency measures)	181	335	Limited information on gas use

We compared the results of the simulation model with the case studies to provide us with a basic level of confidence in the model. The simulation model appears to be in good agreement with the case studies in terms of electricity use. The simulation model for Seattle predicts electricity usage of 77 kWh/sf/yr, which is within 5% of the estimate for the Fred Hutchinson Center without the efficiency measures.

The gas usage is more difficult to compare. The case studies report less than 200 kBtu/sf/yr of measured gas use with the efficiency measures. The case study for the laboratory in Albuquerque predicted total energy use at 428 kBtu/sf/yr without the efficiency measures. Estimating the electricity use at 75 kWh/sf/yr leaves 192 kBtu/sf/yr of gas use. Albuquerque's winters are more severe than Seattle's and less severe than Denver's. However, it is unclear from the case studies what fraction of gas usage is for loads other than space heating, and whether or not there are humidity controls.

To understand the simulation model results, we compared the space heating energy use with a calculation of the energy needed to heat the air over the year using heating degree days (base 65). Because this is effectively a 24-hour facility, the heating degree days provide a good benchmark for estimating space heating loads from the ventilation requirements. Under winter conditions in the middle of the night, the balance point temperature is 65°F, assuming internal loads of 4 W/sf. The flow rate through the fans has been determined under design conditions; it is 2 cfm/sf in the core zones and 2.7 cfm/sf to 3.2 cfm/sf (Table 1.5) in the perimeter zones.

The space heating load is calculated using the following equation:

$$Q = 1.08 \text{ FLOW } (\Delta T) \,, \qquad \text{(Equation 1.1)}$$

where Q is the heat loss, FLOW is the ventilation rate in cfm, and ΔT is the temperature difference in °F. The factor 1.08 represents the density and specific heat capacity of air at standard pressure and temperature multiplied by 60 to convert units. In Denver, standard air conditions are corrected for high altitude by dividing by 1.21. To calculate the annual heating load, substitute heating degree days for the temperature difference in Equation 1.1.

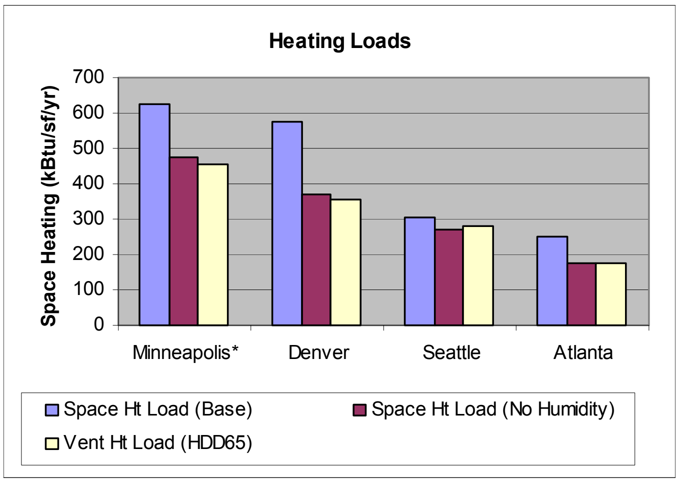


Figure 1.3 Space heating loads in Denver and Seattle.

Figure 1.3 compares the space heating load predicted by DOE-2 with the ventilation heating load calculated using heating degree days. The space heating load from DOE-2 is shown with and without humidity controls. The simulation results for the models without humidity control are within 5% of the space heating load calculated from heating degree days, base 65. Clearly, humidification of the air in Minneapolis and Denver has a significant influence on energy use.

Another issue we investigated is the DOE-2.2 equipment part-load efficiency curve. Figure 1.40 shows the part-load efficiency curves used in DOE-2 for chillers and boilers. The energy-input ratio (EIR) is the inverse of the coefficient of performance (COP) and is the ratio of the energy input to the cooling output (unitless). The DOE-2.2 curve for the chiller is compared with actual data from an energy-efficiency chiller and with the cubic equation used to fit the actual data. The EIR has been normalized by the EIR at full-load (0.5 kW/ton). Actual chiller performance can be characterized in version 41 of DOE-2.2 more precisely; this feature should be used in modeling actual buildings.

The curve for the boiler is compared with a one-to-one curve that represents a boiler with no change in efficiency at part-load operation. The heating-input ratio (HIR) is the ratio of the energy input to the heating output; it has been normalized in the curve with respect to the full-load efficiency (80%). For the boilers, part-load efficiency drops off significantly at a partial load of less than 30%. The simulation results show that the average annual efficiency for space heating in all the models is 68% in Atlanta, 71% in Seattle, and 72% in Denver and Minneapolis. From this, we concluded that the DOE-2.2 curves are reasonable for this study.

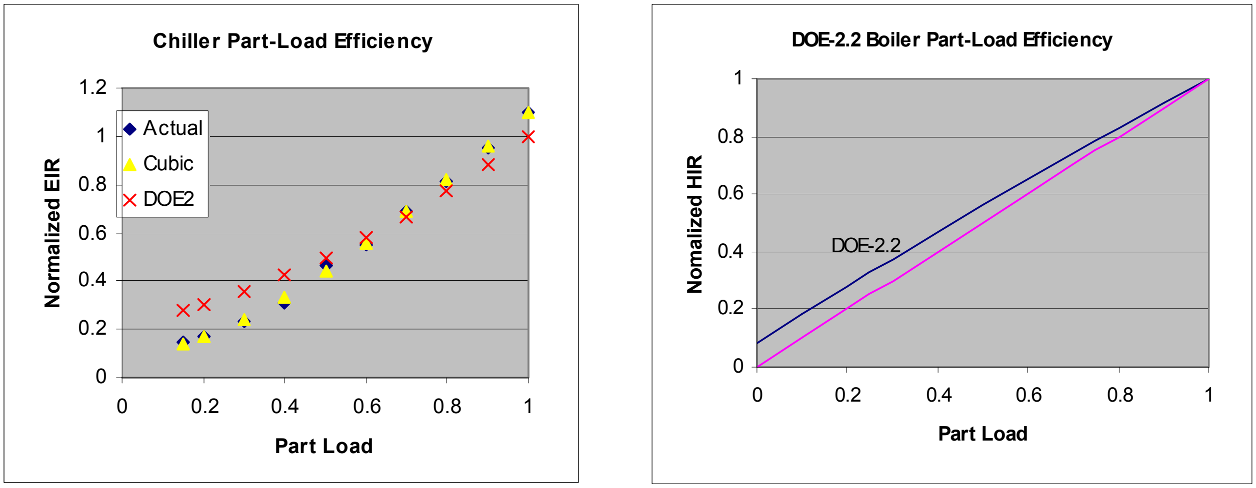


Figure 1.4 Part-load efficiency curves for chillers and boilers.

The models are predicting acceptable results. The comparison between the model heating loads and those calculated using heating degree days shows good agreement, assuming no humidity controls. The models predict much higher spacing heating energy use than the case studies do; this can be attributed to humidity controls, the high percentage of laboratory space in this building (70%), constant-volume fans, and design assumptions.

1.5.2 Annual Energy Use

Although equipment loads dominate energy use, in general, energy end uses and monthly energy use vary with climate. A breakdown of electricity end uses in Seattle is shown in Figure 1.5. The equipment accounts for more than 50% of electricity use in all four climates. Lights average 5% to 6% of the electricity use, fans are in the 25% to 28% range, pumps are 2% to 4%, and space cooling varies from 4% in Seattle to 17% in Atlanta (Figure 1.6). Although it is important to characterize the equipment loads

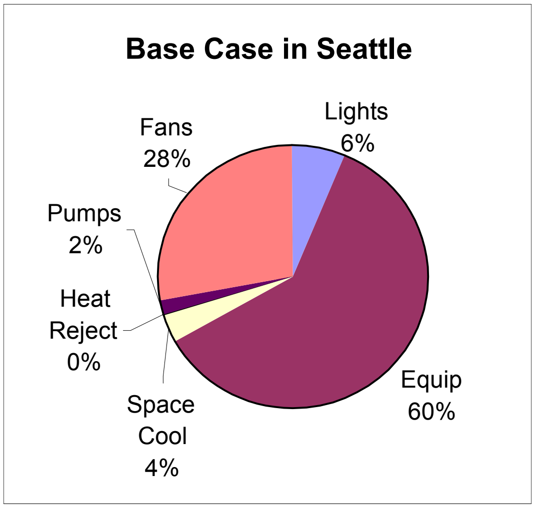


Figure 1.5 Electricity end uses in Seattle.

accurately for sizing equipment, addressing the energy efficiency of this specialized equipment is difficult, and it does not necessarily apply from one building to the next. Although other end uses appear to account for a small percentage of total energy use, they are nevertheless energy-intensive.

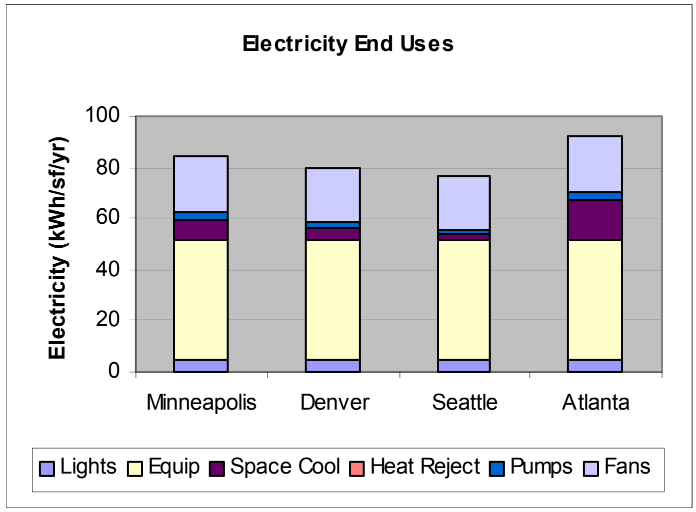


Figure 1.6 Electricity end uses in all climates.

Figure 1.7 excludes the equipment load from electricity use to demonstrate the significance of space cooling and fan electricity use. In all four climates, annual fan electricity use averages 21.7 kWh/sf. Annual space cooling electricity use varies; it is 2.8 kWh/sf in Seattle, 5 kWh/sf in Denver, 8.2 kWh/sf in Minneapolis, and 15.6 kWh/sf in Atlanta.

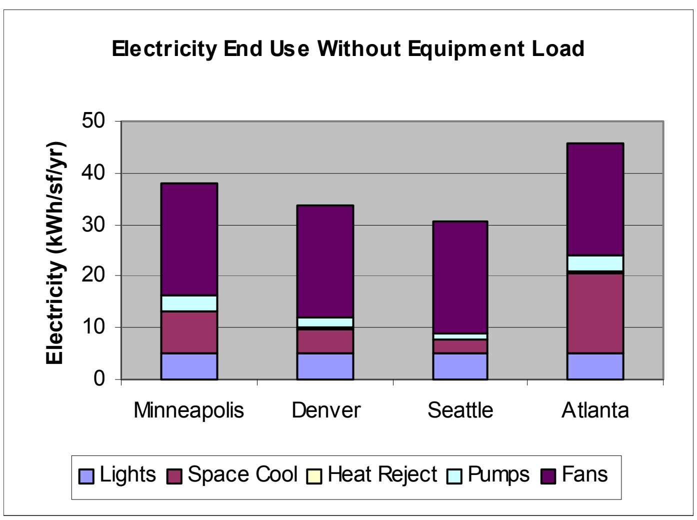


Figure 1.7 Electricity end uses excluding equipment load.

Another significant end use is space heating. The space heating energy includes energy used to heat the building and maintain specified humidity levels. For a benchmark, note that total energy use in office buildings is usually less than 100 kBtu/sf/yr. Space-heating energy use alone for this 24-hour laboratory with a constant ventilation rate is 4 to 8 times the energy use of an office building.

Figure 1.8 presents space heating energy use with humidity controls (base case) and without humidity controls. The model for Denver, which has the driest climate, shows the greatest humidification load. The results show that the minimum humidity levels in all four climates dip below 30% relative humidity (RH) for a significant number of hours. Humidity levels are in the 20%-29% RH range for 2600 hours in Atlanta and 6500 hours in Denver. We infer that the majority of these hours are above 28% RH.

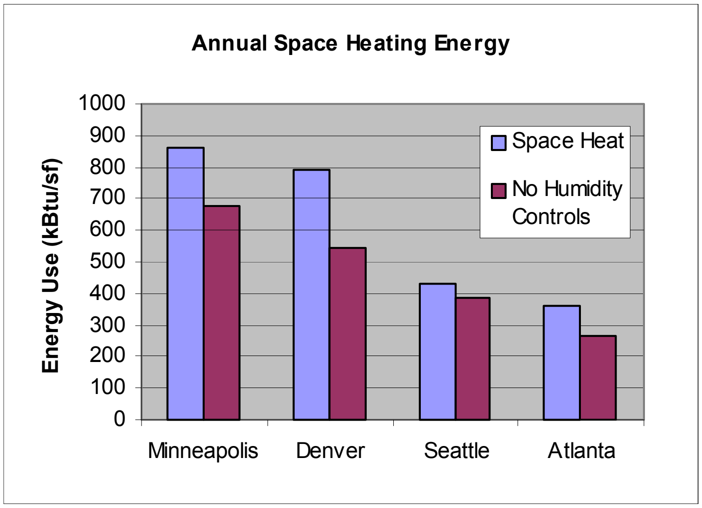


Figure 1.8 Annual space heating energy use with and without minimum humidity control.

1.5.3 Monthly Energy Use

Laboratory buildings have such high internal loads from equipment that monthly electricity use varies by less than 15% over the year in milder climates like those of Seattle and Denver (Figure 1.9). However, electricity use is 50% higher in the summer, however, in hotter climates like Atlanta's. There is much greater variation in gas usage from month to month in all four climates (Figure 1.10). The heating load in the summer months is the result of subcooling and reheat for dehumidification.

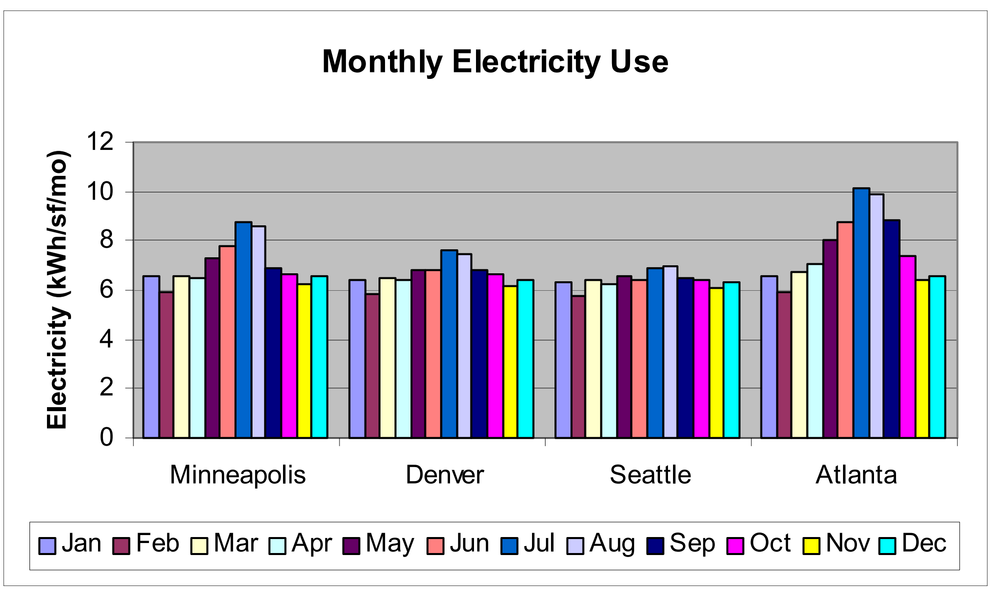


Figure 1.9 Monthly electricity use in each of the climates.

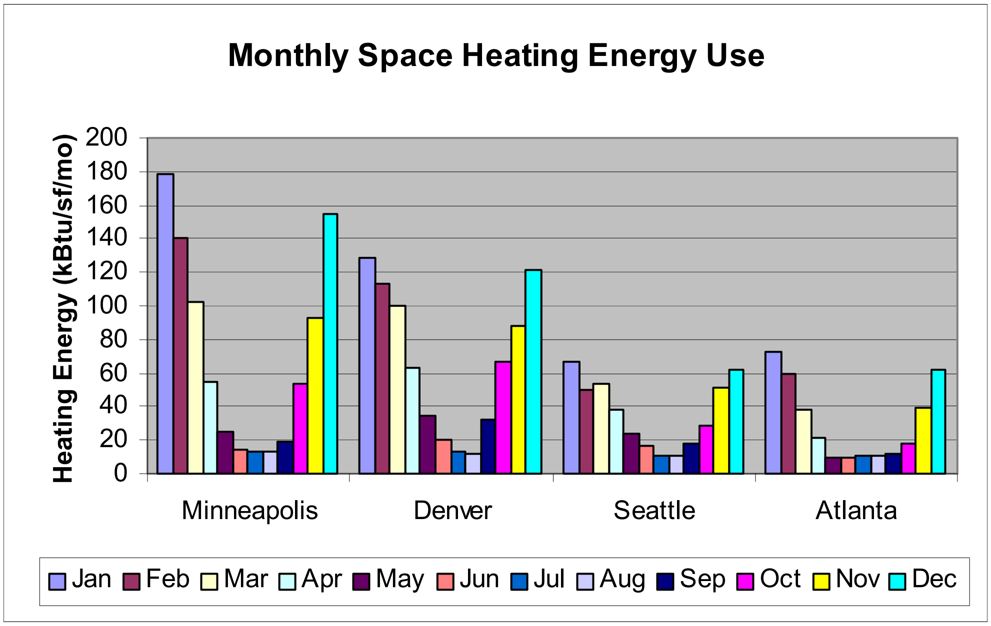


Figure 1.10 Monthly space-heating energy use.

1.5.4 Peak Electricity Demand

Peak electricity demand is two to three times greater than that generally found in commercial buildings (Figure 1.11). The gross equipment load itself is 7.2 W/sf, the fans contribute 2.5 W/sf, and lighting adds another 1.4 W/sf. The difference in peak demand between the climates is attributable to the difference in space cooling.

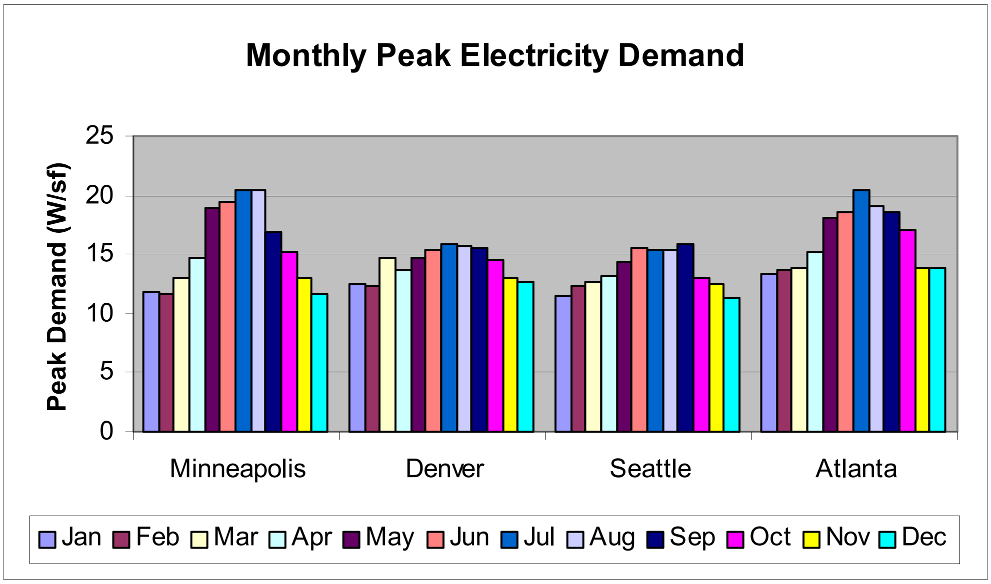
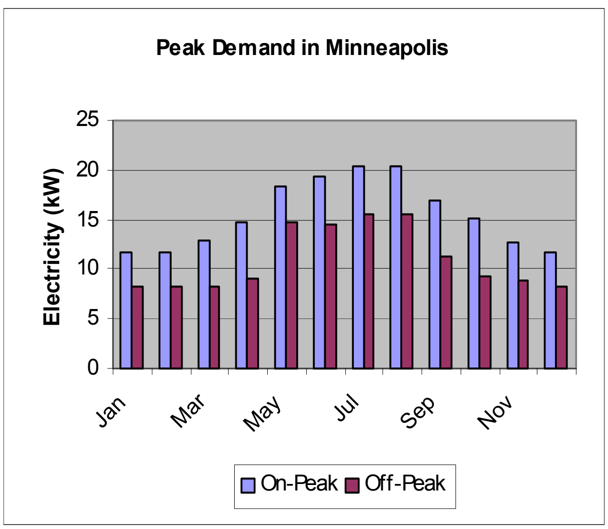
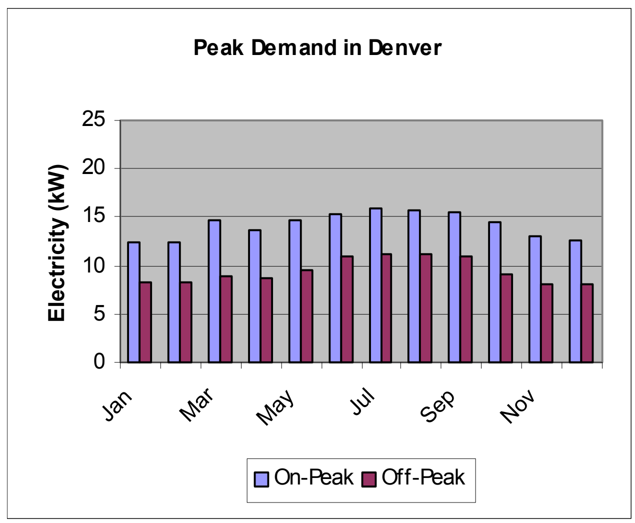


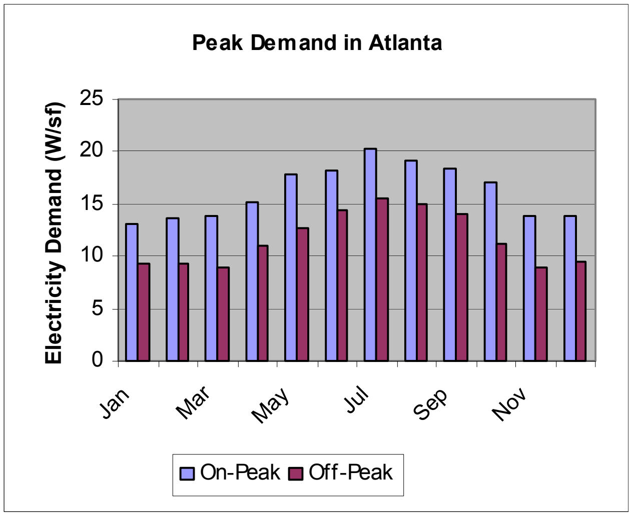
Figure 1.11 Monthly peak electricity demand for all climates.

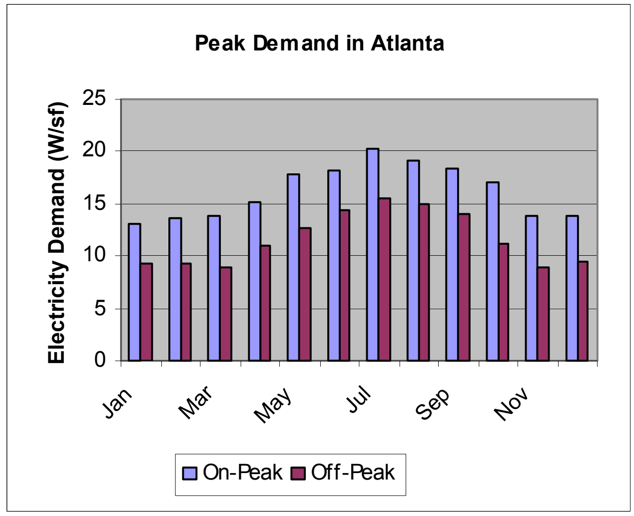
Another important consideration is on-peak and off-peak demand. For the electricity rates, the on-peak time is 8 a.m. to 10 p.m., Monday through Friday. Figure 1.12a, b, c, and d show the on-peak and off-peak electricity demand for each month.





Figures 1.12a and b On-peak and off-peak electricity demand for Minneapolis and Denver.





Figures 1.12c and d On-peak and off-peak electricity demand for Seattle and Atlanta

1.5.5 Supply Air

The supply air is a minimum of 2 cfm/sf of outside air for the base case. The design cooling load in the perimeter zones requires 2.6 to 2.7 cfm/sf of supply air; 2 cfm/sf is sufficient in the core zones, except in Denver (Table 1.5). The ventilation requirements in Denver are higher because of its elevation.

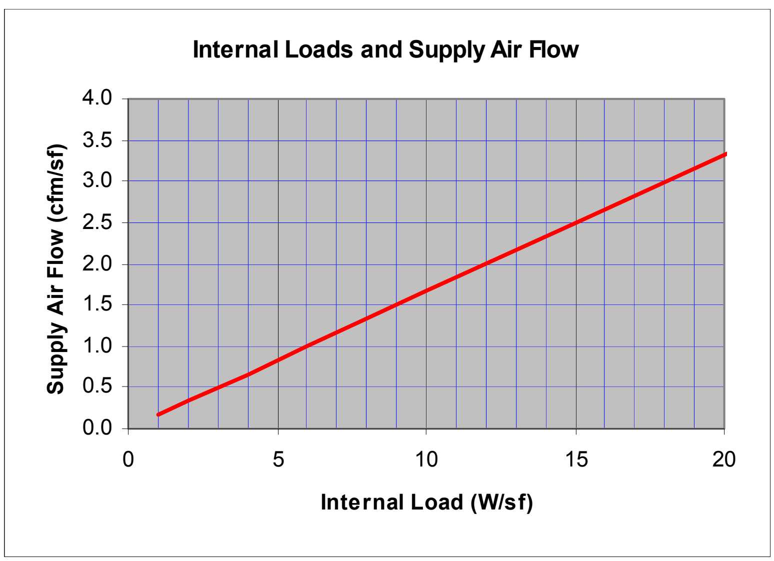


Figure 1.13 Supply air flow at 55°F to meet internal loads.

Figure 1.13 gives the supply air flow (cfm/sf) at 55°F required to offset internal loads. The internal loads in the base building average 10 W/sf during occupied periods, although the design loads are in the 15 W/sf range in the perimeter zones.

1.5.6 Sensible and Latent Loads

The base case model includes humidity controls, so latent cooling and latent heating loads have a greater impact on energy use than they do in a building without such controls. The latent cooling load refers to the energy content of the moisture in the supply air that exceeds the maximum humidity requirement. The latent heating load is the energy content of the moisture needed to meet the minimum humidity requirement.

DOE-2 reports the sensible heat ratio, which is the ratio of the sensible energy load to the total energy load (sensible plus latent). The lower the sensible heat ratio is, the more moisture there is in the air, and the higher the relative humidity is. In Denver, the sensible heat ratio is 1.0 (i.e., no latent cooling load) at the cooling peak in each month. In Seattle, the sensible heat ratio does not drop below 0.7 during the cooling peaks. In Minneapolis, the sensible heat ratio averages 0.42 at the cooling peak from July through September. In Atlanta, the sensible heat ratio drops as low as 0.308 in July; it is around 0.5 during other summer months. To dehumidify the air in Minneapolis and Atlanta, the supply air is subcooled to remove moisture from the air and reheated to bring the air temperature back up to the minimum 55°F supply air temperature.

In dry months, heating energy is required to evaporate moisture into the air to meet minimum humidity requirements, i.e., latent heating energy. Atlanta requires latent heating energy from October through March, whereas Minnesota and Seattle require it from October through May. Denver has year-round latent heating requirements, because its climate is so dry. Figure 1.8 compares space heating requirements with and without humidity controls. To add moisture to the air, the hot water loop serves a pan heat exchanger through which moisture is evaporated into the air.

1.5.7 Energy Costs

A rule of thumb for commercial buildings is that energy costs average $1/sf/yr. For laboratories, however, the cost is $5 to $10/sf/yr. The simulation models reflect this, with electricity costs averaging $4/sf/yr. Gas costs range from $2/sf/yr in Seattle to $5/sf/yr in Minneapolis (Figure 1.14). Based on the assumed utility rates, the cost for electricity averages $0.05/kWh; that for gas is $0.60/therm. The demand charges for electricity are 59% of the total electricity charges. The structure of the utility rates has a big impact on energy costs and varies from utility to utility.

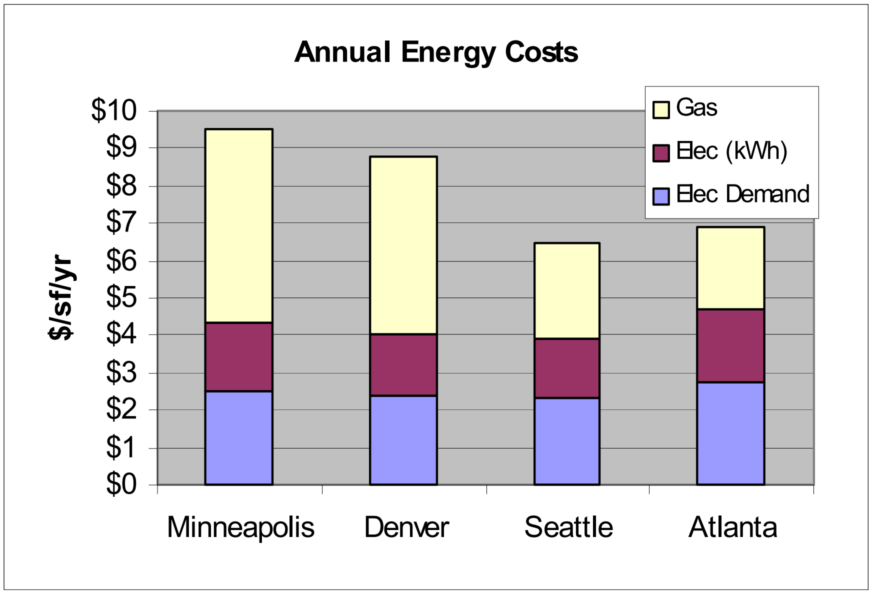


Figure 1.14 Annual electricity and gas costs for all climates.

CHAPTER 2. ENERGY EFFICIENCY STRATEGIES

Table 2.1 presents the energy efficiency strategies we considered. These strategies focus on reductions in fan energy use, energy recovery opportunities, and evaporative cooling. The effect of tighter humidity controls with and without an enthalpy wheel is also evaluated. In addition, we investigated the impact of equipment power density assumptions (plug loads) on mechanical system sizing.

Table 2.1 Energy Efficiency Strategies

Measure	Base	Run 1	Run 2
Ventilation Rates	Minimum setting: 2 cfm/sf (24 hrs/day)	Minimum setting: 2 cfm/sf occupied, 1 cfm/sf unoccupied (24 hrs/day)	Variable air volume with 1 cfm/sf minimum setting
Static Pressure Drop	5.2 in. supply, 2 in. exhaust	4 in. supply, 1.5 in. exhaust at 2 cfm/sf	3 in. supply, 1 in. exhaust at 2 cfm/sf
Energy Recovery: Enthalpy Wheel	None	0.75 sensible effectiveness; 0.75 latent effectiveness at 0.8 in. w.g. pressure drop	Same as previous with VAV system
Heat Pipes	None	0.48 effectiveness at 1 in. pressure drop	
Run-Around Loop	None	0.6 effectiveness at 1 in. pressure drop	
Chiller Energy Recovery	None	0.8 fraction of heat recovered from condenser water	
Evaporative Cooling	None	Direct evap. w/ 0.8 effectiveness and 0.1 in. w.g. pressure drop	Water-side economizer with 0.8 effectiveness
Humidification	30% RH Min/ 60% RH Max	40% RH Min/ 50% RH Max (Also ran 20% RH Min/ 60% RH Max)	40% RH Min/ 50% RH Max w/ Enthalpy Wheel
Plug Loads in Lab Space	12 W/sf	8 W/sf	4 W/sf

Table 2.2 reiterates energy use statistics for the base case building. The energy efficiency measures are compared with the base case for each climate. We conclude by looking at the most efficient strategies for each climate.

Table 2.2 Base Case Building Energy Use

	Peak Demand (W/sf)	Electricity Use (kWh/sf/yr)	Gas Use (kBtu/sf/yr)	Total Energy Use (kBtu/sf/yr)	Annual Electricity Cost ($/sf)	Annual Gas Cost ($/sf)	Annual Energy Cost ($/sf)
Minneapolis	20	84	861	1125	$4.30	$5.20	$9.50
Denver	16	80	792	1043	$4.00	$4.80	$8.80
Seattle	15	77	431	673	$3.90	$2.60	$6.60
Atlanta	20	92	362	652	$4.70	$2.20	$6.90

2.1 Ventilation Rates

The base case building has a constant-volume air system. The design flow rates are shown in Table 2.3 for the core and perimeter areas. Under peak cooling conditions (not design conditions), the cooling load is 10 W/sf and requires 1.6 cfm/sf of 55°F air to cool. During unoccupied periods, the cooling load is less than 6 W/sf and requires only 1 cfm/sf of 55°F air to cool (Figure 1.13). The difference between the design condition of more than 2 cfm/sf and a minimum of 1 cfm/sf provides opportunities to reduce energy use for fans, space cooling, and space heating.

Table 2.3 Design Supply and Exhaust Flow Rates

	Core (cfm/sf)	Perimeter (cfm/sf)
Minneapolis	2.0	2.7
Denver	2.4	3.2
Seattle	2.0	2.6
Atlanta	2.0	2.7

A simple control strategy is to reduce the minimum supply flow from 2 cfm/sf to 1 cfm/sf during unoccupied periods (CFM21). A more efficient approach is a VAV system with reheat to reduce the supply air flow in response to varying loads and varying ventilation requirements, as would occur with variable-volume fume hoods. The VAV system is modeled with a minimum outdoor air setting of 1 cfm/sf and an increase in static pressure drop of 0.5 in. w.g. to account for the losses associated with the variable-speed drive. DOE-2.2 calculates the load in each zone and determines the necessary supply air flow. The fans have variable-speed drives, and DOE-2 calculates the fan energy with respect to the lower flow rate and lower static pressure drop.

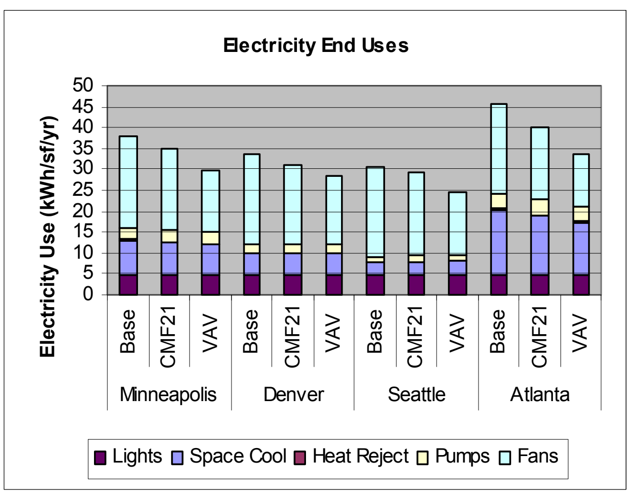
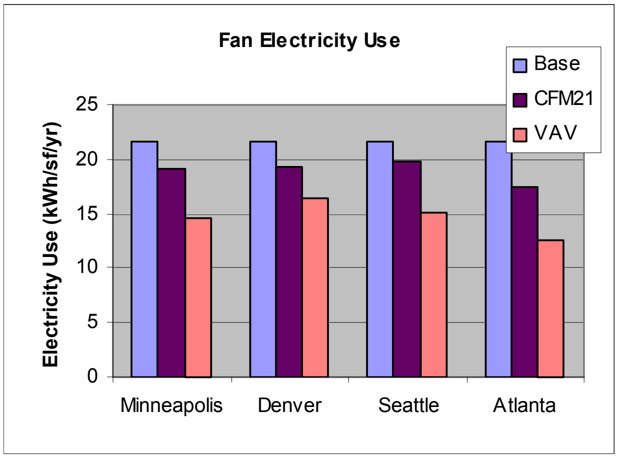


Figure 2.1 Electricity end uses for the three cases.

Figure 2.1 shows the electricity end uses for the base case, the flow setback case (CFM21), and the VAV case, excluding equipment plug loads. Flow setback reduces the annual electricity use by 3 kWh/sf in Minneapolis, 2 kWh/sf in Denver, 1 kWh/sf in Seattle, and 5 kWh/sf in Atlanta. The VAV system reduces annual electricity use by 8 kWh/sf in Minneapolis, 5 kWh/sf in Denver, 6 kWh/sf in Seattle, and 12 kWh/sf in Atlanta.

Both strategies reduce electricity use for fans. Figure 2.2a shows fan electricity use, and Figure 2.2b shows savings in comparison to the base case fan electricity use, 21.7 kWh/sf. The supply air flow setback strategy reduces the fan electricity use by 3 kWh/sf/yr in Minneapolis, 2 kWh/sf/yr in Denver and Seattle, and

4 kWh/sf/yr in Atlanta. The VAV system reduces fan electricity use by 7 kWh/sf/yr in Minneapolis, 5 kWh/sf/yr in Denver, 7 kWh/sf/yr in Seattle, and 9 kWh/sf/yr in Atlanta.



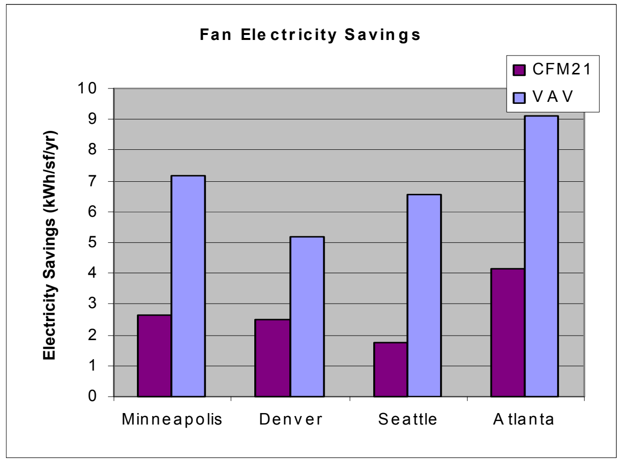


Figure 2.2a and b Annual fan electricity use (a) and savings (b) resulting from reducing supply air flow with setback controls (CFM21) and VAV.

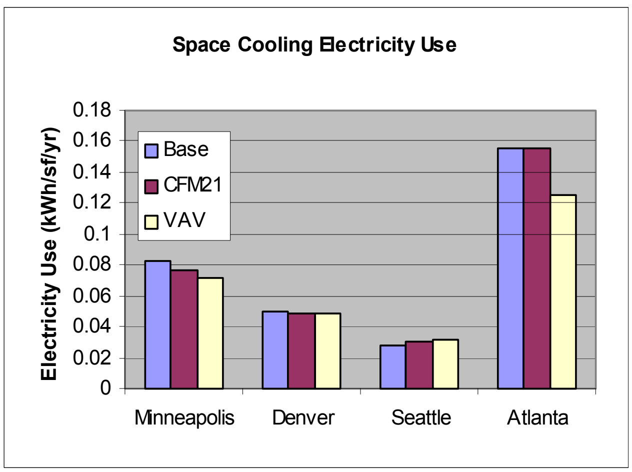
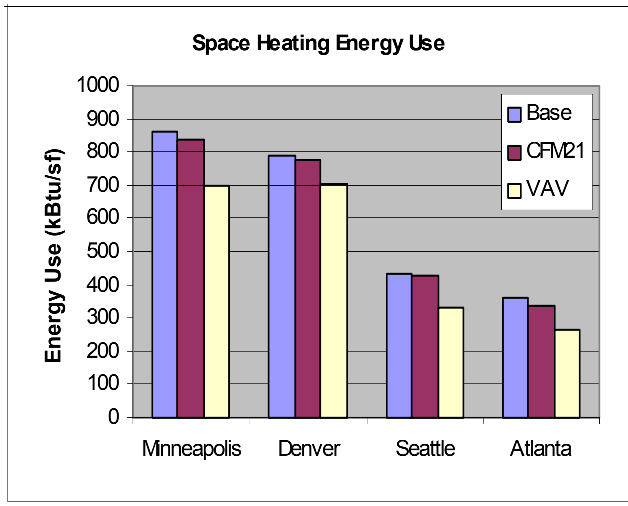


Figure 2.3 Annual electricity use for space cooling resulting from reducing supply air flow with setback controls (CFM21) and (VAV).

The effect of flow setback on annual space cooling savings is negligible (Figure 2.3). The most significant cooling savings occur in Atlanta, where the VAV system reduces cooling electricity use by 3.1 kWh/sf. This is a reduction of 20% in cooling electricity use, although it is just one-fourth of the savings resulting from reducing fan energy. In Seattle, space cooling is adversely affected as a result of running the chillers at lower part loads.

Peak electricity demand does not change with the supply air flow setback. It drops by 1 W/sf with VAV in Minneapolis and Denver and by 2 W/sf in Atlanta. In Seattle, the peak reduction is 0.5 W/sf.

Figures 2.4a and b show the space heating energy use and savings for each climate. The supply air flow setback reduces space heating energy use by 2% in Minneapolis and Denver, 1% in Seattle, and 6% in Atlanta. The VAV system reduces space heating energy use by 19% in Minneapolis, 11% in Denver, 23% in Seattle, and 28% in Atlanta.



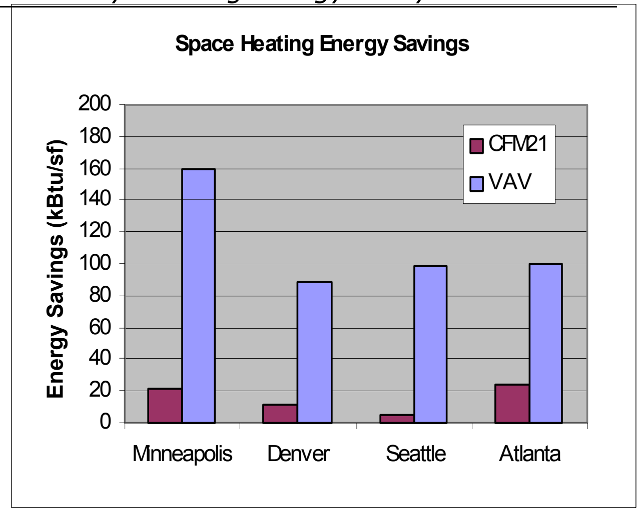


Figure 2.4a and b Space heating energy use and annual energy savings resulting from reducing space heating energy with flow reduction strategies.

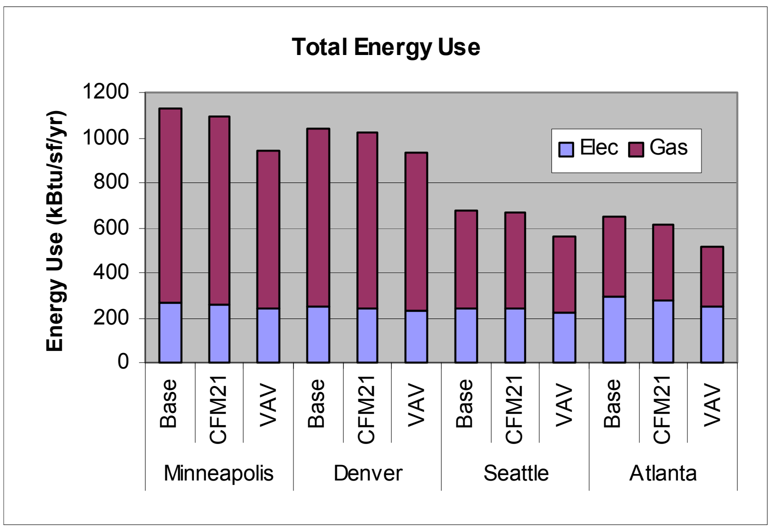
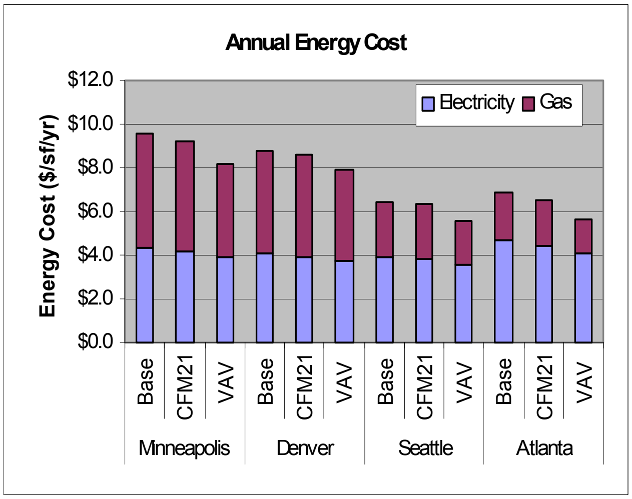


Figure 2.5 Total energy use for the base case and flow reduction strategies.

In terms of total energy use, the electricity energy use is comparable to the gas energy use in Atlanta, whereas in Minneapolis, gas energy use is nearly 4 times the electricity energy use (Figure 2.5). However, the cost per Btu of electricity is nearly 3 times the cost per Btu of gas in this analysis, which translates into 3 times the savings for every Btu of electricity conserved. The impact of the flow reduction strategies on total energy use is not as great as expected. The model assumes high internal gains from plug loads, which limits the opportunity to reduce air flow.

The base energy costs range between $6.50/sf in Seattle to $9.50/sf in Minneapolis. With the flow setback controls, the electricity savings are $0.20/sf in Minneapolis, $0.10/sf in Denver, $0.10/sf in Seattle and $0.20/sf in Atlanta. The electricity cost savings from the VAV system are $0.40/sf in Minneapolis, $0.30/sf in Denver and Seattle, and $0.60/sf in Atlanta (Figure 2.6a and b).

The gas savings from the flow setback are $0.10/sf in Minneapolis, $0.10/sf in Denver, $0.03/sf in Seattle, and $0.10/sf in Atlanta. The VAV system saves $1/sf in Minneapolis, $0.50/sf in Denver, $0.60/sf in Seattle, and $0.60/sf in Atlanta (Figure 2.6a and b).



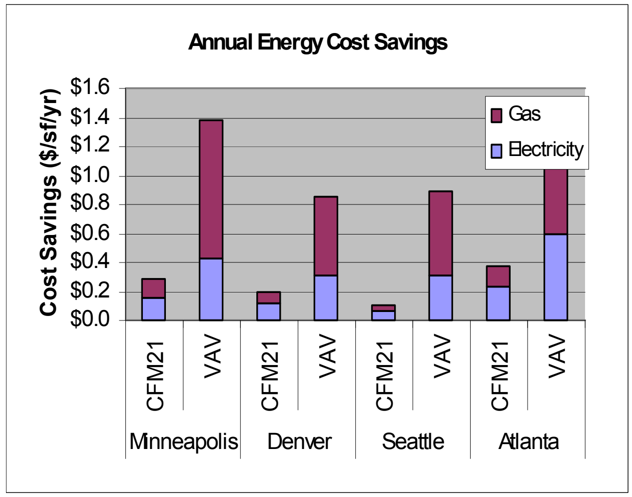


Figure 2.6a and b Annual energy costs and savings for the base case and flow reduction strategies.

The VAV system saves $0.20/sf to $0.30/sf in electricity costs and $0.40/sf to $0.90/sf in gas costs over the flow setback controls. On average, cost savings from the flow setback are 3% of the base case costs, and savings from the VAV system are 14% of the base case costs.

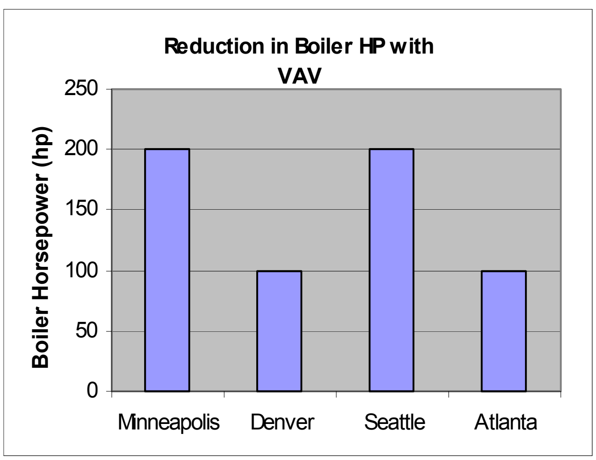


Figure 2.7 Potential reduction in boiler size with VAV system.

Another advantage of the VAV system is a potential opportunity to reduce the size of the heating system. The heating system would be designed to meet the heating load using a ventilation requirement of 2 cfm/sf, rather than the design flow determined under cooling design conditions. The potential reduction in boiler size varies from 100 to 300 hp in these four climates (Figure 2.7) and results in cost savings of $25,000 to $50,000 at $250/hp.

2.2 Fan Static Pressure Drop

Fan energy use is calculated from the following equation:

Energy = Static Pressure Drop *Flow * 0.746 / η / 6354 , (Equation 2.1)

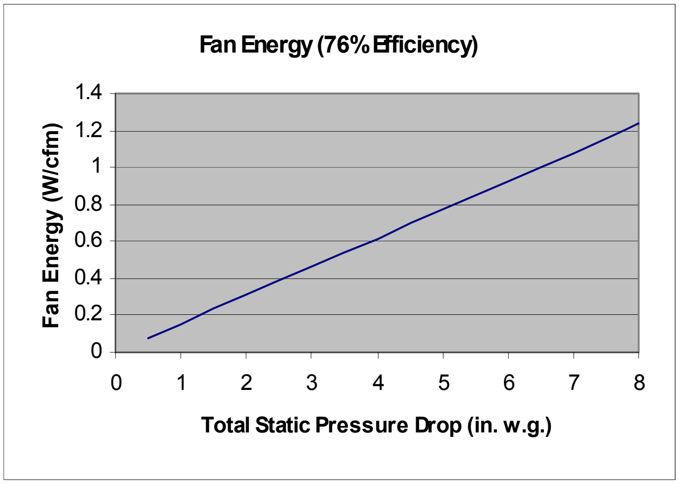


Figure 2.8 Fan energy with respect to static pressure.

where Static Pressure Drop is the pressure drop associated with coils, filters, and ducts; Flow is the air flow rate in cfm; the factor 0.746 converts horsepower to watts; η is the combined efficiency of the motor and fan; and the factor 6354 converts units to horsepower. The result is energy use (Energy) in W/cfm. For the base case, the fans are constant-volume and have a total efficiency of 76%. The motor efficiency is 95%. The supply fans see a total static pressure drop of 5.2 in. w.g. and the exhaust fans see a total static pressure drop of 2 in. w.g. Figure 2.8 shows the fan energy use in W/cfm with respect to static pressure drop. Reducing the static pressure drop reduces fan energy use. The static pressure drop can be reduced through the design and use of larger ducts and coils and filters with a lower pressure drop.

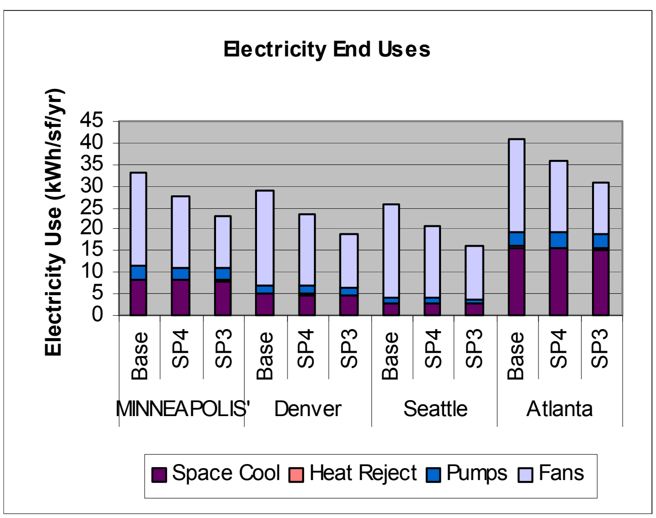


Figure 2.9 Electricity end uses for the base case, supply static pressure of 4 in. w.g. and exhaust static pressure of 1.5 in. w.g (SP4), and supply static pressure of 3 in. w.g. and exhaust static pressure of 1.0 in. w.g. (SP3).

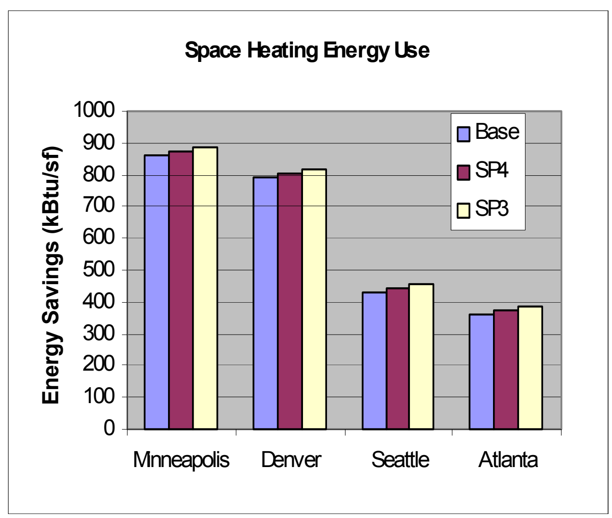


Figure 2.10 Space heating energy use for the base case, supply static pressure of 4 in. w.g. and exhaust static pressure of 1.5 in. w.g (SP4); supply static pressure of 3 in. w.g. and exhaust static pressure of 1 in. w.g. (SP3).

To demonstrate the potential electricity savings resulting from reducing the static pressure drop, we simulated two cases: (1) supply static pressure drop of 4 in. w.g. and exhaust of 1.5 in. w.g., and (2) supply static pressure drop of 3 in. w.g. and exhaust of 1.0 in. w.g. The input to DOE-2.2 is the design static pressure drop, fan and motor efficiency, and the fan curve. The reduction to 4 in. w.g. and 1.5 in. w.g. saves 5 kWh/sf/yr, and the reduction to 3 in. w.g. and 1 in. w.g. saves 10 kWh/sf/yr (Figure 2.9). There is an increase in space heating energy use because less fan energy is being added to the air stream (Figure 2.10). The peak electricity demand is reduced by 0.7 W/sf in the first case and by 1.3 W/sf in the second case. The fan system is constant-volume, so the fan energy use is constant for the base case.

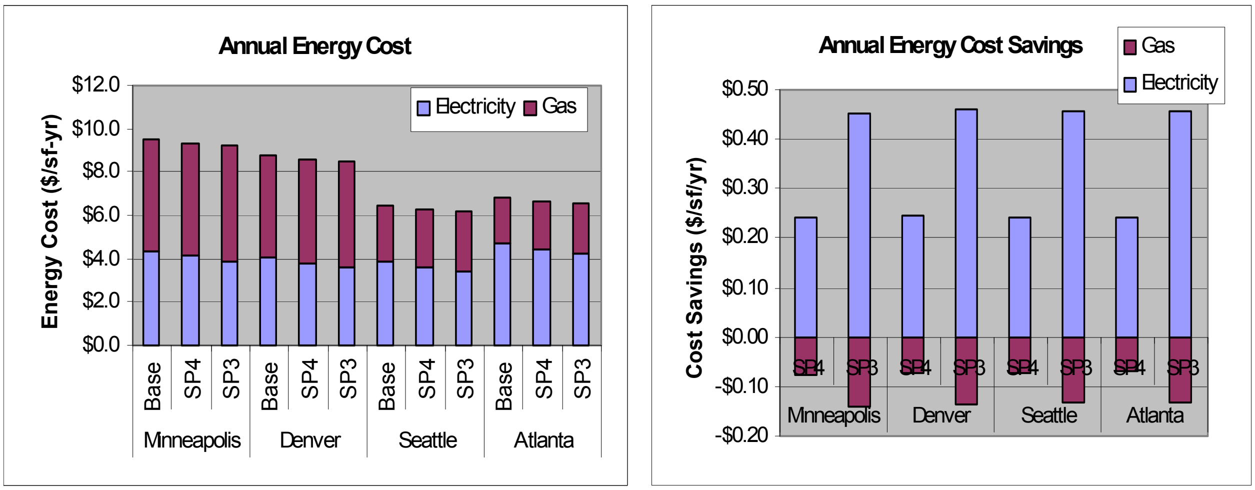


Figure 2.11a and b Annual energy cost and cost savings from reducing supply static pressure from 5.2 in. w.g. (base) to 4 in. w.g. (SP4) and 3 in. w.g. (SP3).

The annual net cost savings are $0.17/sf in the first case and $0.32/sf in the second case (Figure 2.11a and b). The increase in gas costs is $0.08/sf with the 4 in. w.g. static pressure (SP4) and $0.13/sf with the 3 in. static pressure (SP3).

With a VAV system, the flow rate and static pressure drop are reduced. As shown in the previous section, the result is lower electricity and gas usage. The annual cost savings with a VAV system are $1.40/sf in Minneapolis, $0.80/sf in Denver, $0.90/sf in Seattle, and $1.20/sf in Atlanta.

2.3 Energy Recovery

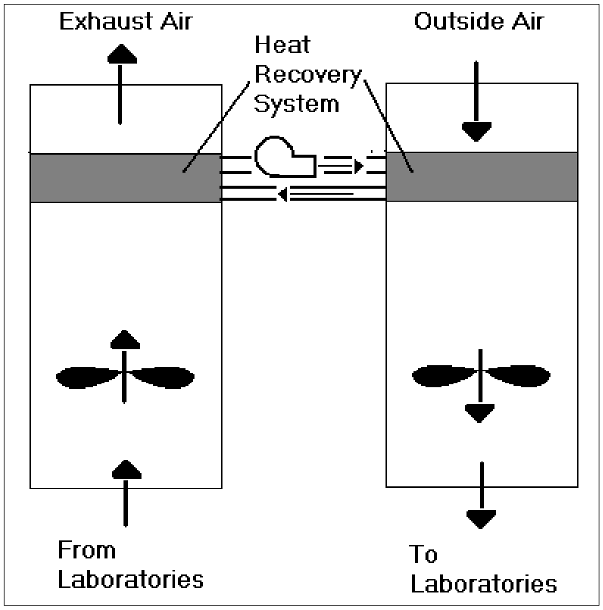


Figure 2.12 Example of run-around loop energy recovery system.

Energy recovery is often considered for laboratories because of the high outside air ventilation rates. There are many options available for air-to-air energy recovery; they are summarized in the ASHRAE HVAC Systems and Equipment Handbook (2000). for this analysis, we considered enthalpy wheels, which recover sensible and latent energy, and heat pipes and run-around loops (Figure 2.12), which recover sensible energy only. Energy recovery from the chiller to the hot water loop is also included.

For the performance of the enthalpy wheels and heat pipes, we referred to the ARI Air-to-Air Recovery Ventilation Equipment Certified Products Directory (March 2002) and Des Champs and Semco technical representatives. For the run-around loop, we used the ASHRAE HVAC Systems and Equipment Handbook (2000). The effectiveness of the air-to-air recovery devices is defined as the ratio of the actual energy recovered to the theoretical energy that could be recovered. Sensible energy is the energy associated with a temperature difference.

The sensible effectiveness is proportional to the ratio of the difference between the dry bulb temperature of the outside air and supply air to the difference between the dry bulb temperature of the exhaust air and the outside air. Latent energy is the energy of the moisture, and in this case the moisture in the air. The latent effectiveness is proportional to the ratio of the difference between the humidity ratio of the outside air and the supply air to the difference between the humidity ratio of the exhaust air and the outside air.

The DOE-2.2 model of energy recovery ventilators (ERV) has recently been added and is still being tested. DOE-2.2 (version h) is zeroing out the humidity ratio for the air leaving sensible heat recovery devices, but the model predicts it correctly for devices with sensible and latent heat recovery. In order to model heat pipes and run-around loops with humidity controls, we used the “enthalpy-hx” option in DOE-2.2 and set the latent effectiveness to 0.05.

In addition, the design calculations ignore the presence of the ERV, and the pumping energy use predictions are wrong. The program predicts higher energy use for the pumps with the ERV than without them. To account for the potential to downsize the cooling and heating equipment, we resized the equipment manually and ensured that space requirements would still be met. In sizing the heating and cooling equipment for the base case and energy efficiency measures, chillers were sized in 50-ton increments and boilers were sized in 50-hp increments. We recognize that, in some projects, the heating and cooling equipment will not be downsized, so the savings presented here will be greater than they would be if the equipment had not been downsized. We did not adjust the pumping energy use because it is no more than 4% of the total electricity use in all cases. This “oversight” makes the results a little more conservative.

For the three ERV cases, the air temperature leaving the ERV and entering the supply air plenum is controlled to meet the required supply air temperature. Humidification or dehumidification may still occur. The ERV runs during heating and cooling modes, and it has an outside air bypass to maintain the required supply air temperature and avoid condensation and frost on the heat exchanger.

Because of the humidity controls, we also tested fixing the temperature of the air leaving the ERV and entering the supply air plenum. In heating mode, the ERV may deliver the air at 55°F, but the air must often be humidified. Humidification may add 1°F to 2°F to the air temperature and then cooling may be required. Fixing the air temperature in heating mode to 53°F reduced annual energy costs by about 1%. Although they are not significant with respect to the simulation results, the runs demonstrated the importance of fine-tuning the mechanical system controls.

We have not evaluated energy recovery from or to a process loop supplying hot water, steam, or chilled water. Depending on the process temperature requirements and the operation of the loop with respect to cooling and heating the building, there can be cost-effective opportunities to exchange energy between the process loop(s) and the hot water and chilled water systems.

2.3.1 Enthalpy Wheels

Enthalpy wheels transfer sensible and latent energy between the exhaust air and the incoming outside air. For this analysis, we assumed 0.75 effectiveness for the sensible energy recovery, 0.75 effectiveness for the latent energy recovery, and 0.7 in. w.g increase in the static pressure on the supply and exhaust. The wheel is a counter-flow heat exchanger. It runs when the difference in enthalpy between the outside and exhaust air is 3 Btu/lb-°F or greater, except in Denver, and the wheel runs more efficiently when controlled by a minimum temperature difference of 5°F between the outside air and exhaust air. A bypass damper is specified on outside air to compensate for overheating and overcooling and to control against condensation and frost.

To avoid contamination of the supply air stream, the wheel is flushed with supply air that is deflected by a damper in the purging section of the rotor. The damper is located on the supply air outlet side at the point where the rotor passes from the exhaust air flow path to the supply air flow path. The purge section utilizes the pressure difference between the outside air and exhaust air streams. The model assumes 7% purge by volume of the supply air. The purge volume is in addition to the specified supply and exhaust flows.

The potential benefit of the enthalpy wheel is a reduction in energy use by the chillers, pumps, and cooling tower on the cooling side. On the heating side, boilers and associated pumps also use less energy. The increase in static pressure is significant with the large wheels, and so is the additional fan energy. There is also the possibility of downsizing the chillers, boilers, cooling tower, and pumps. The first-cost savings resulting from downsizing can pay for the enthalpy wheel(s).

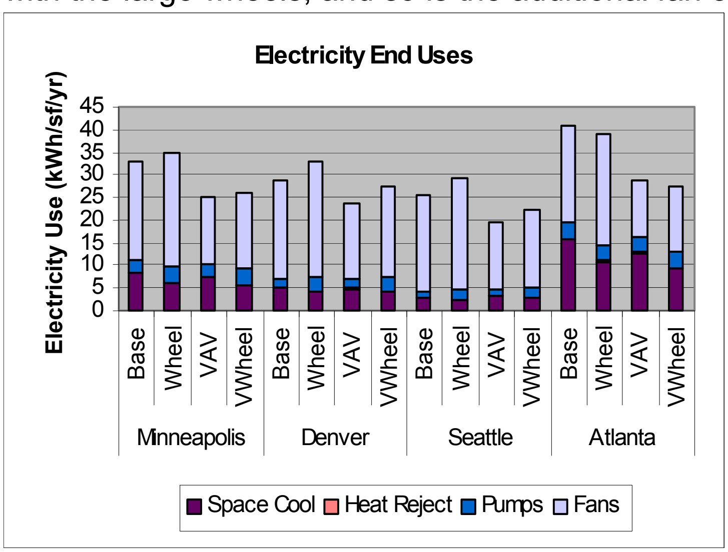


Figure 2.13 Electricity end uses for the base case and for the base case with an enthalpy wheel (Wheel) and the variable-air-volume system without (VAV) and with an enthalpy wheel (VWheel). Lighting and plug loads are not included.

The enthalpy wheel was run with the constant-volume reheat system (Wheel) and the variable-air-volume system (VWheel). Figure 2.13 shows the electricity end uses for the base case, the constant-volume system with an enthalpy wheel (Wheel), the variable-air-volume system (VAV), and VAV with an enthalpy wheel (VWheel), excluding lighting and plug loads. The lighting and plug loads are the same for all four cases. Electricity use actually increases with the enthalpy wheel in comparison to the system without the enthalpy wheel.

Figure 2.14 shows the fan and space cooling electricity "savings." With the constant-volume system and the enthalpy wheel, net electricity use increases by 2 kWh/sf in Minneapolis, 4 kWh/sf/yr in Denver, and 3 kWh/sf/yr in Seattle. Electricity use decreases by 2 kWh/sf/yr in Atlanta. With the VAV system, electricity use decreases by 8 kWh/sf in Minneapolis, 5 kWh/sf/yr in Denver, 6 kWh/sf/yr in Seattle, and 12 kWh/sf/yr in Atlanta. The VAV system with the enthalpy wheel results in an increase in electricity use with respect to the VAV system of 1 kWh/sf/yr in Minneapolis, 4 kWh/sf/yr in Denver, and 3 kWh/sf/yr in Seattle. In Atlanta, electricity use decreases by 1 kWh/sf/yr.

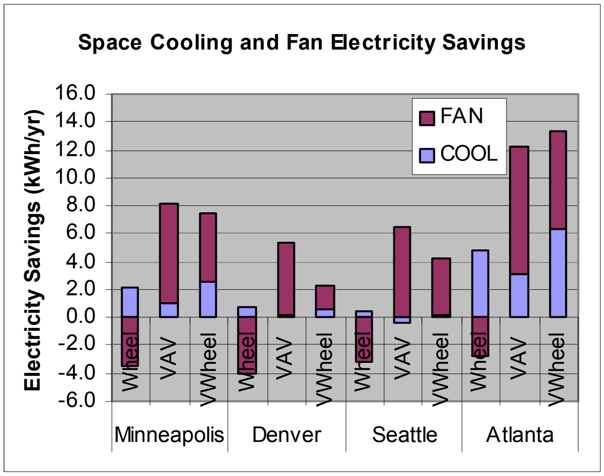


Figure 2.14 Space cooling and fan electricity savings for an enthalpy wheel with a constant-volume reheat system (Wheel), and a variable-air-volume system without (VAV) and with a wheel (VWheel).

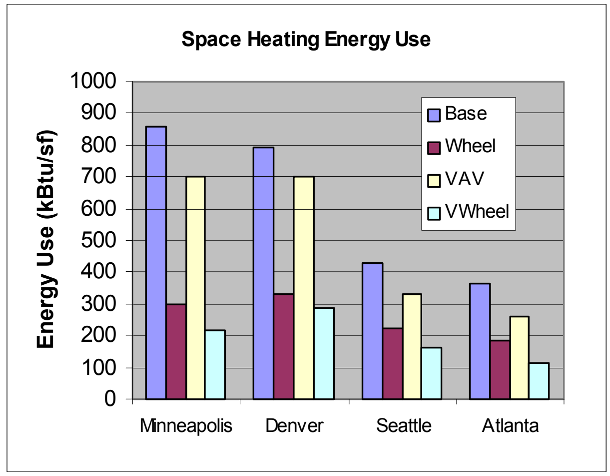
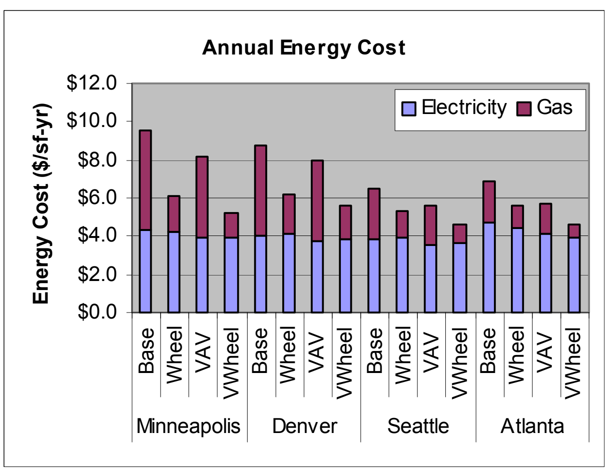


Figure 2.15 Space heating energy use for the base case, enthalpy wheel with a constant-volume reheat system (Wheel), and a variable-air-volume system without (VAV) and with a wheel (VWheel).



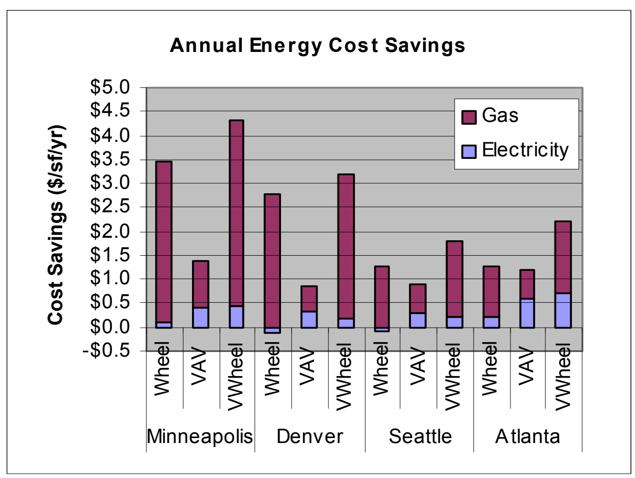


Figure 2.16a and b Annual energy costs (a) and cost savings (b) for enthalpy wheel with a constant-volume reheat system (Wheel) and a variable-air-volume system without (VAV) and with a wheel (VWheel).

The greatest amount of savings result from the reductions in space heating. Figure 2.15 shows the space heating energy use for the base case and the two cases with an enthalpy wheel. The constant-air-volume system with the enthalpy wheel reduces space heating energy use by 50% or more in all four climates. The VAV system with the enthalpy wheel reduces heating energy use by an additional 10% in Minneapolis, 5% in Denver, 13% in Seattle, and 20% in Atlanta.

A rule of thumb is that enthalpy wheels save more than $1/sf/yr; our results reflect this (Figure 2.16a and b). The majority of the cost savings result from reduced gas usage. Electricity costs decrease in Minneapolis ($0.10/sf/yr) and Atlanta ($0.20/sf/yr) with the constant-volume system and enthalpy wheel. In Denver and Seattle, electricity costs increase by $0.10/sf. With the constant-volume system, the total cost savings are $3.50/sf in Minneapolis, $2.60/sf in Denver, $1.20/sf in Seattle, and $1.30/sf in Atlanta.

As compared with the base case, the VAV system with the enthalpy wheel decreases electricity costs by $0.50/sf/yr in Minneapolis, $0.20/sf/yr in Denver, $0.20/sf/yr in Seattle, and $0.70/sf/yr in Atlanta. The associated energy cost savings from the reduction in space heating energy use are $3.90/sf/yr in Minneapolis, $3/sf/yr in Denver, $1.60/sf/yr in Seattle, and $1.50/sf/yr in Atlanta. Total cost savings are $4.30/sf in Minneapolis, $3.20/sf in Denver, $1.80/sf in Seattle, and $2.20/sf in Atlanta. These cost savings are $3/sf/yr more than the VAV system without the enthalpy wheel in Minneapolis, and $2.70/sf/yr in Denver. In Seattle and Atlanta, the costs savings are greater by $0.90/sf/yr and $1/sf/yr, respectively, than for the VAV system without the enthalpy wheel.

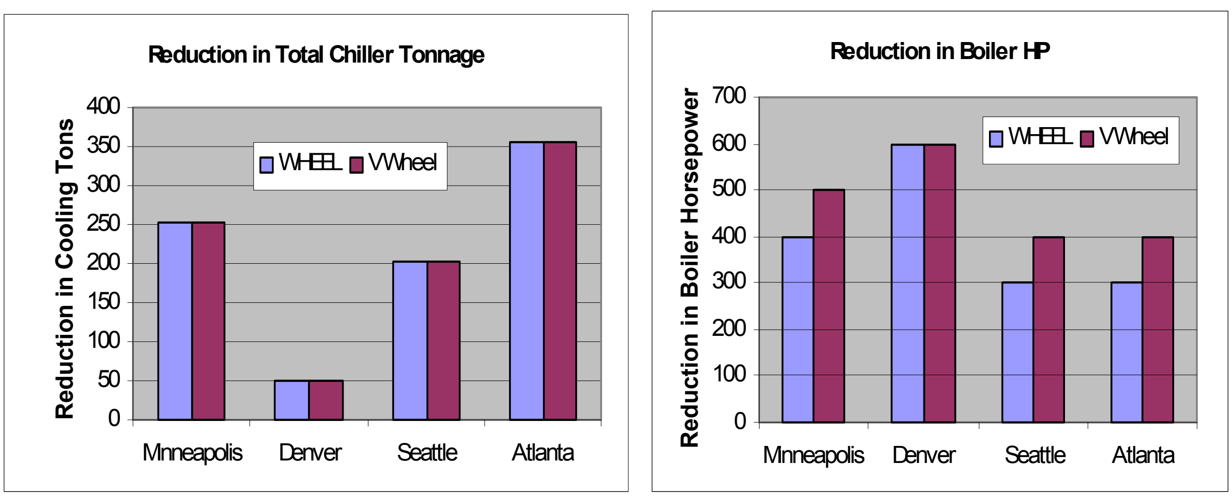
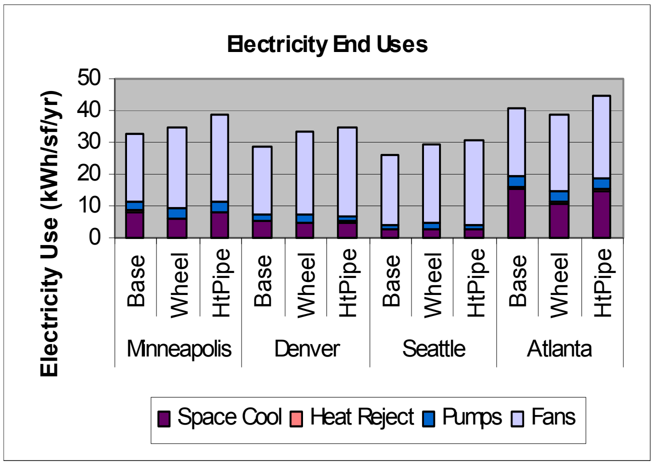


Figure 2.17a and b Reduction in chiller tonnage (a) and boiler horsepower (b) with an enthalpy wheel and base system (Wheel) and an enthalpy wheel and a variable-air-volume system (VAV Wheel).

As for downsizing equipment, the first-cost savings can offset the cost of the enthalpy wheel. Figures 2.17a and b show the reduction in total chiller and boiler capacity. There are also opportunities to reduce the size of the primary and secondary chiller and boiler pumps, as well as the cooling tower. The potential cost savings are $1000/ton on the cooling side and $250/hp on the heating side. The potential first-cost savings are $2.50/sf in Minneapolis, $2/sf in Seattle, $0.50/sf in Denver, and $3.50/sf in Atlanta.

2.3.2 Heat Pipes

Heat pipes transfer sensible energy between the exhaust air stream and the incoming outside air. If exhaust air is cooled below its dewpoint, condensation occurs and results in some latent heat transfer. We assumed 0.48 sensible effectiveness and 1 in. w.g increase in the static pressure on the supply and exhaust. Heat pipes are counter-flow heat exchangers like enthalpy wheels. The heat pipe operates when the difference in temperature is a minimum of 5°F between the outside air and exhaust air. A bypass damper is specified on the outside air to compensate for overheating and overcooling and to control against condensation and frost.



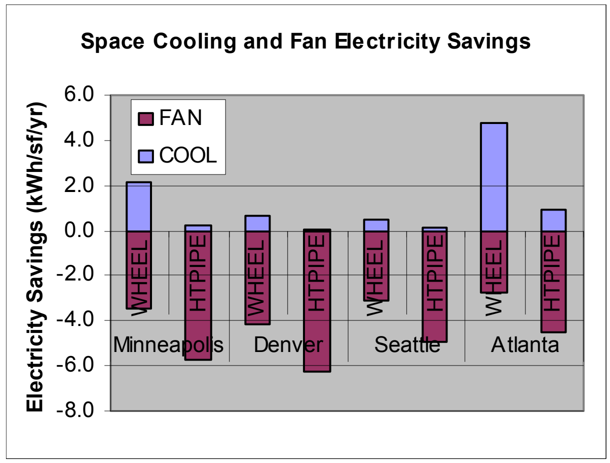


Figure 2.18a and b Electricity end uses (a) excluding lighting and plug loads for the base case and for the base case with an enthalpy wheel (Wheel) and a heat pipe (HtPipe); space cooling and fan electricity savings (b) for the enthalpy wheel and heat pipes compared with those of the base case.

The potential benefits of the heat pipe are similar to those of the enthalpy wheel, except there is no latent energy recovery. There is the possibility of downsizing the chillers, boilers, cooling tower, and pumps. First-cost savings from downsizing can offset the cost of the heat pipe. Heat pipes reportedly have lower maintenance costs because there are no moving parts; when one heat pipe fails in a bank, the remaining ones continue to function. Heat pipes also do not pose problems of cross-contamination between the supply and exhaust streams as do enthalpy wheels; however, supply and exhaust streams must still be next to one another. The downside is the increase in fan energy from the higher static pressure drop associated with a large bank of heat pipes.

The heat pipe was run with the constant-volume reheat system. For comparison, the same system with an enthalpy wheel is shown in the graphs. Figure 2.18a shows the electricity end uses, excluding lighting and plug loads for all three cases, and Figure 2.18b shows the fan and space cooling electricity savings for the two energy recovery runs, compared with the base case. As shown, electricity savings are greater with the enthalpy wheel in all climates. The pressure drop across the heat pipe is greater than that with the enthalpy wheel, so the fan energy is always greater with the heat pipe. Again, the greatest savings come from a reduction in space heating. Figure 2.19 shows space heating energy savings compared with the base case building. The enthalpy wheel reduces space heating energy use by more than twice as much in the heating-dominated climates of Minneapolis and Denver.

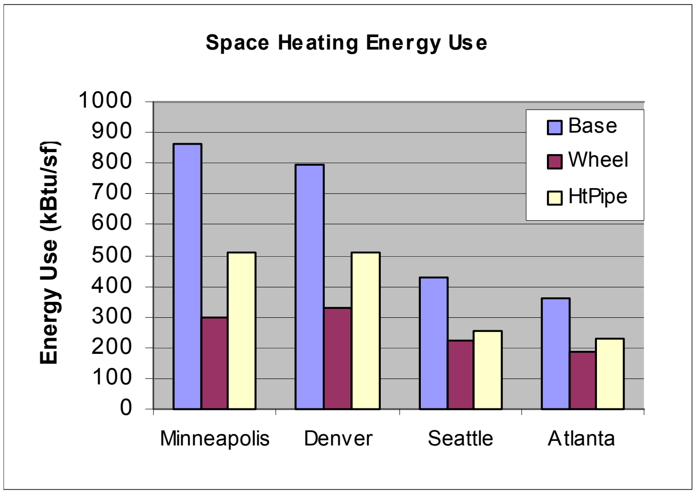
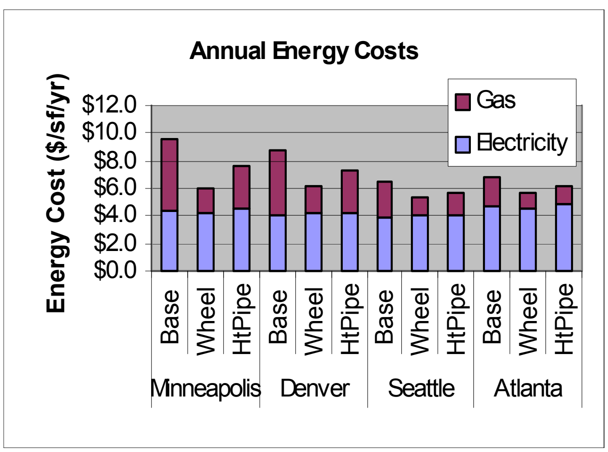


Figure 2.19 Space heating energy use for the base case and the base case with an enthalpy wheel (Wheel) and a heat pipe (HtPipe).

Figure 2.20a shows annual energy costs for the three cases, and 2.20b shows energy cost savings from the heat pipe and enthalpy wheel. The cost savings with the heat pipe are $1.90/sf in Minneapolis, $1.50/sf in Denver, $0.90/sf in Seattle, and $0.70/sf in Atlanta. The savings are greater in colder climates than in warmer climates, as anticipated.



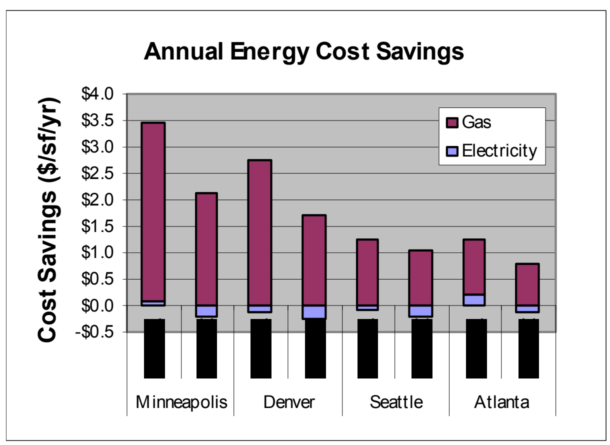
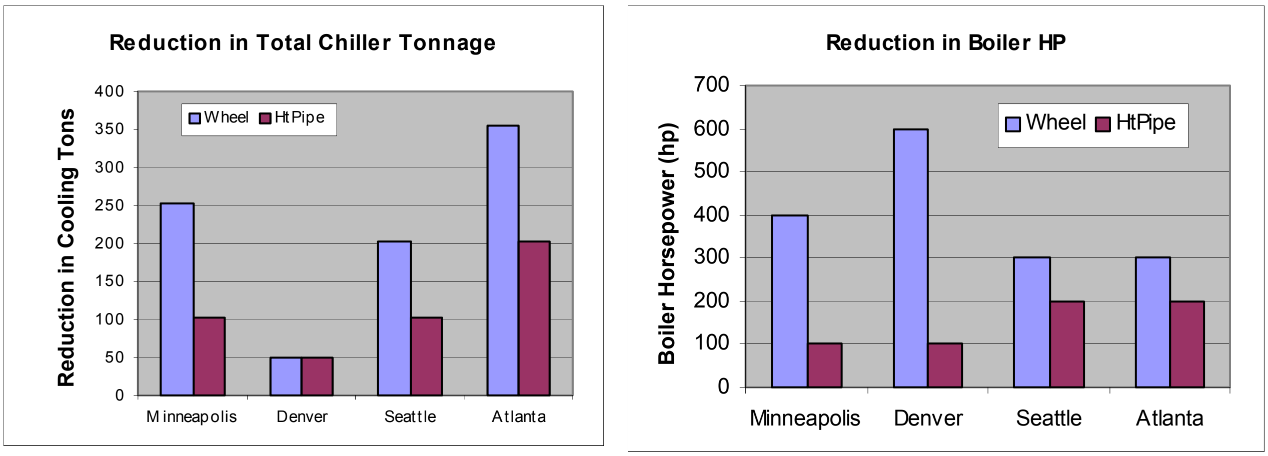


Figure 2.20a and b Annual energy costs (a) for the base case and the base case with an enthalpy wheel (Wheel) and heat pipes (HtPipe); energy cost savings (b) for the enthalpy wheel (Wheel) and heat pipe (HtPipe) compared with those of the base case.

As for downsizing equipment, Figure 2.21a and b show the reduction in total chiller and boiler capacity. There are also opportunities to reduce the size of the primary and secondary chiller and boiler pumps as well as the cooling tower and condenser water pumps. The potential cost savings are $1000/ton for downsizing the cooling side. The enthalpy wheel offers the greatest savings.

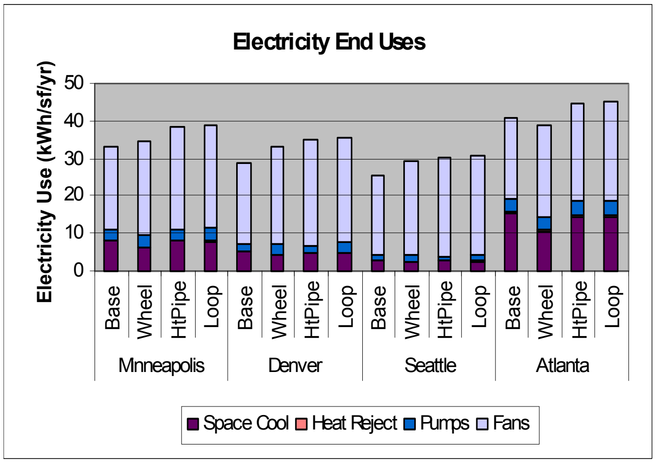
On the heating side, the enthalpy wheel is also more effective than the heat pipe. There are opportunities to downsize the boiler, although the potential first-cost savings are much less in Minneapolis and Denver with the heat pipe. This can be attributed to the lower effectiveness of the heat pipes and the lack of latent energy recovery.



Figures 2.21a and b Reduction in chiller tonnage (a) and boiler horsepower (b) for enthalpy wheel (Wheel) and heat pipe (HtPipe) with a constant-volume reheat system.

2.3.3 Run-Around Loop

A run-around loop circulates a fluid between the exhaust air stream and the supply air to recover energy. The advantage to a run-around loop is that the supply and exhaust do not need to be located in the same place. The simulations assume a sensible effectiveness of 0.6 with a pressure drop of 1 in. w.g. across the heat exchangers. For the pumping energy, 0.05 W/cfm is assumed. The loop operates when the difference in temperature is a minimum of 5°F between the outside air and exhaust air.



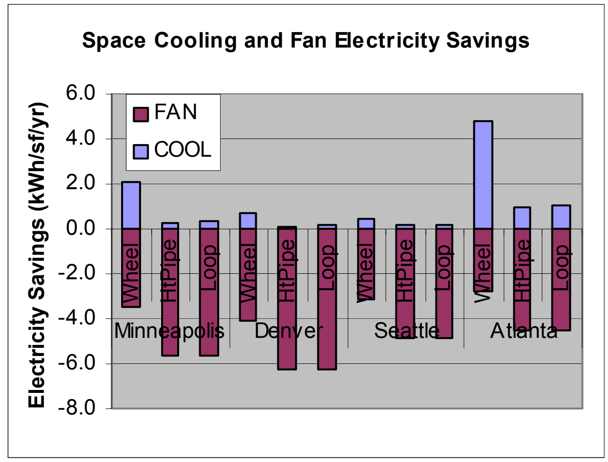


Figure 2.22a and b Electricity end uses (a), excluding lighting and plug loads, for the base case and for the base case with an enthalpy wheel (Wheel), heat pipe (HtPipe), and run-around loop (Loop); space cooling and fan electricity savings (b) for the enthalpy wheel, heat pipe, and run-around loop compared with those of the base case.

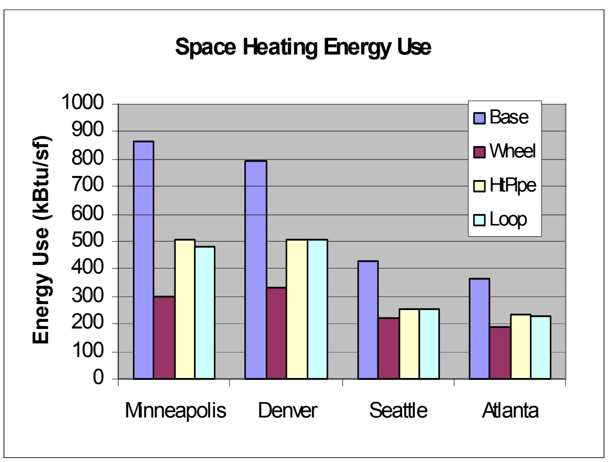


Figure 2.23 Space heating energy savings for the base case, enthalpy wheel (Wheel), heat pipe (HtPipe) and run-around loop (Loop) with a constant-volume reheat system.

Figure 2.22a shows the electricity end uses for the base case with a constant-volume reheat system, and for the base case with an enthalpy wheel (Wheel), heat pipe (HtPipe), and a run-around loop (Loop). Figure 2.22b shows electricity savings for space cooling and fans for the enthalpy wheel, heat pipe, and run-around loop as compared with those of the base case. All three cases assume a constant-volume reheat system. Except for the enthalpy wheel in Atlanta, total electricity use increases over that of the base case for the enthalpy wheel, heat pipe, and run-around loop. The DOE-2.2 energy recovery ventilation model still predicts the pumping energy incorrectly; however, the reduction in pumping energy would not offset the increase in fan energy.

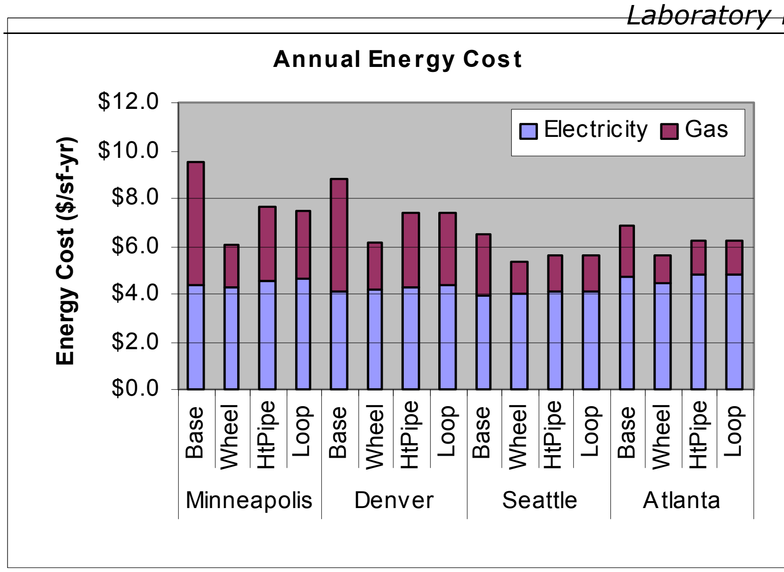
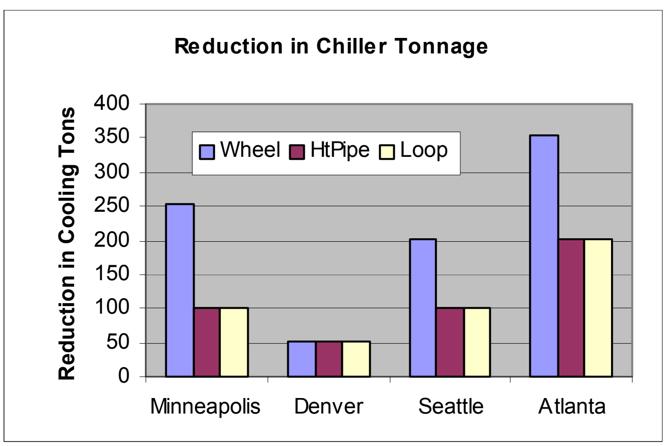


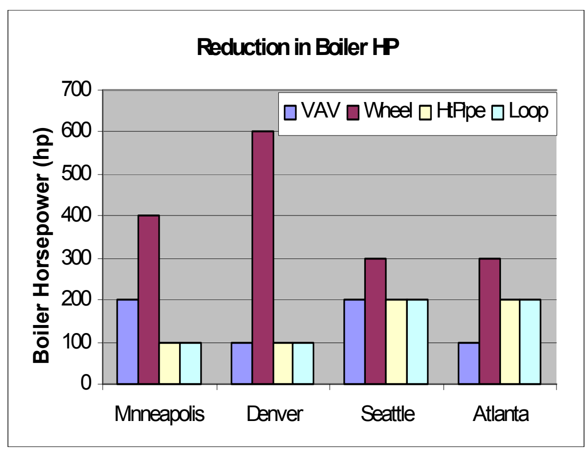
Figure 2.24 Annual energy cost for the base case, enthalpy wheel (Wheel), heat pipe (HtPipe), and run-around loop (Loop) with a constant-volume reheat system.

Figure 2.23 gives the space heating energy use for the four cases. The wheel has a sensible effectiveness and latent effectiveness of 0.75, the heat pipe has a sensible effectiveness of 0.48, and the run-around loop has a sensible effectiveness of 0.6. The heating energy savings associated with the enthalpy wheel are much greater than with the heat pipe and run-around loop because of the significant latent energy loads.

Electricity use increases with the run-around loop, and gas use decreases, with respect to the base case. This is consistent with the other energy recovery ventilators. In terms of costs, the net annual savings are $2/sf in Minneapolis, $1.50/sf in Denver, $0.90/sf in Seattle, and $0.70/sf in Atlanta (Figure 2.24).

There is also potential for downsizing with the use of a run-around loop. Figure 2.25a and b show the reduction in chiller and boiler capacity with the enthalpy wheel, heat pipe, and run-around loop. The downsizing potential is greater with the enthalpy wheel, although potential first-cost savings with the run-around loop are still significant.





Figures 2.25a and b Reduction in chiller tonnage (a) and boiler horsepower (b) for enthalpy wheel (Wheel) and run-around loop (Loop) with base case constant-volume reheat system. Boiler reduction with VAV is also shown.

2.3.4 Energy Recovery Chiller

We also modeled energy recovery from the condenser water to the hot water loop for space heating. The major difference between the chiller with energy recovery (CWER) and the other energy recovery approaches discussed so far is that this is not an air-side approach. The potential energy savings correspond to the amount of time during which there are coincident hot water and chilled water loads. The limitation is the number of cooling hours. The simulations show that there are space heating loads every hour of the year in all climates, whereas the cooling loads occur 4970 hrs/yr in Atlanta, 2830 hrs/yr in Minneapolis, 2780 hrs/yr in Denver, and 1700 hrs/yr in Seattle.

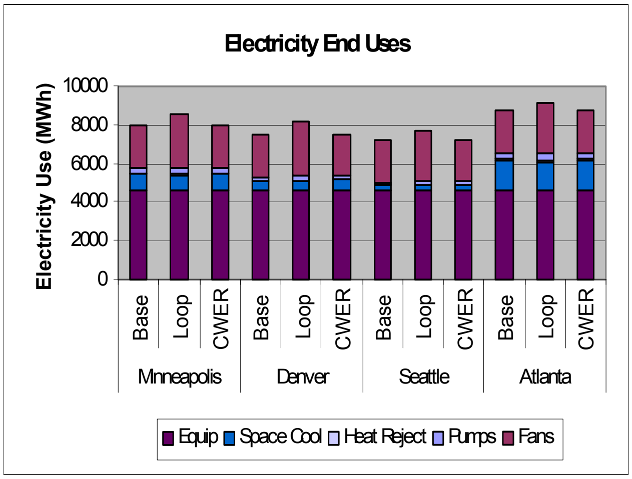


Figure 2.26 Electricity end uses for the base case and base case with a run-around loop (Loop) and an energy recovery chiller (CWER).

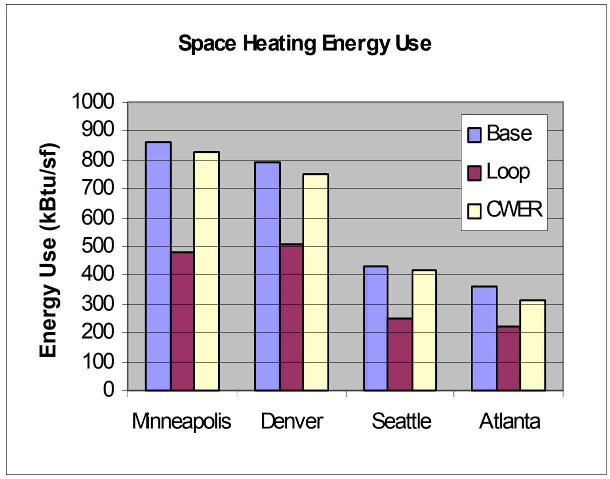


Figure 2.27 Space heating energy use for the base case and base case with a run-around loop (Loop) and an energy recovery chiller (CWER).

DOE-2.2 assumes the chiller has a "double-bundle" condenser, with the second condenser attached to the hot water loop. Any heat not rejected to the hot water loop is rejected to the main condenser water loop. We assumed that 80% of the design condenser heat is available for energy recovery. With this configuration, the energy recovery temperature must be greater than the return hot water loop temperature in order to recover useful energy. So, the design temperature of the hot water loop was lowered from 180°F to 95°F. The lower hot water temperature will require larger pipe sizes and more pumping. Alternatives to this configuration are running the condenser water through a water-source heat pump to heat the water, or using the recovered energy to heat water for a process load with a closer temperature match. Neither of those two approaches is modeled here.

The simulations for the energy recovery chiller predict that the electricity use is almost the same as that of the base case. There is more pumping energy and chiller energy use, but less cooling tower (i.e., heat rejection) energy use (Figure 2.26). On the heating side, space heating is reduced by 4% in Minneapolis, 6% in Denver, 3% in

Seattle, and 13% in Atlanta (Figure 2.27). Atlanta's climate has more simultaneous heating and cooling than the other climates have.

The total energy costs and energy cost savings for the three cases are shown in Figure 2.28a and b. The cost savings are $0.20/sf in Minneapolis, $0.30/sf in Denver, $0.10/sf in Seattle, and $0.30/sf in Atlanta.

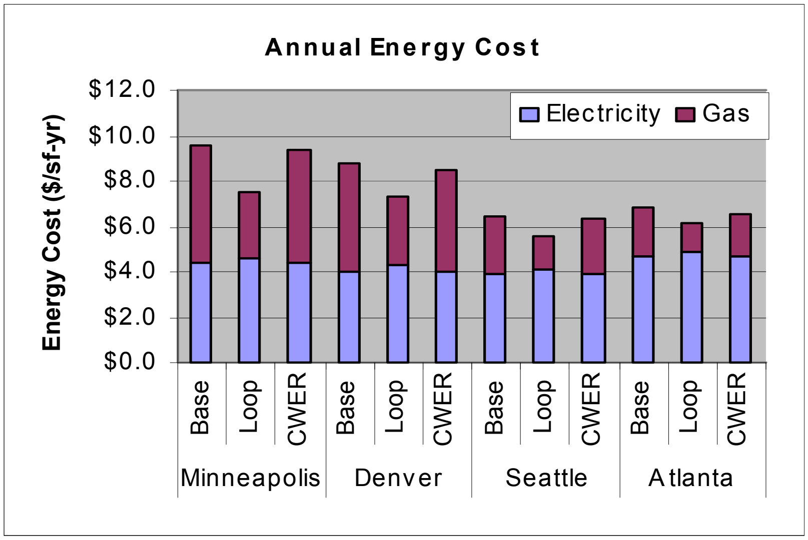


Figure 2.28a Annual energy costs for the base case and the base case with a run-around loop (Loop) and an energy recovery chiller (CWER).

The potential for downsizing heating and cooling equipment depends on design assumptions as to whether or not there are simultaneous loads under design conditions. The simulations show a simultaneous heating load and peak cooling condition in Denver, and indicate that the cooling tower could be downsized by 50 tons. In the other climates, the difference in cooling tower sizes is less than 15 tons.

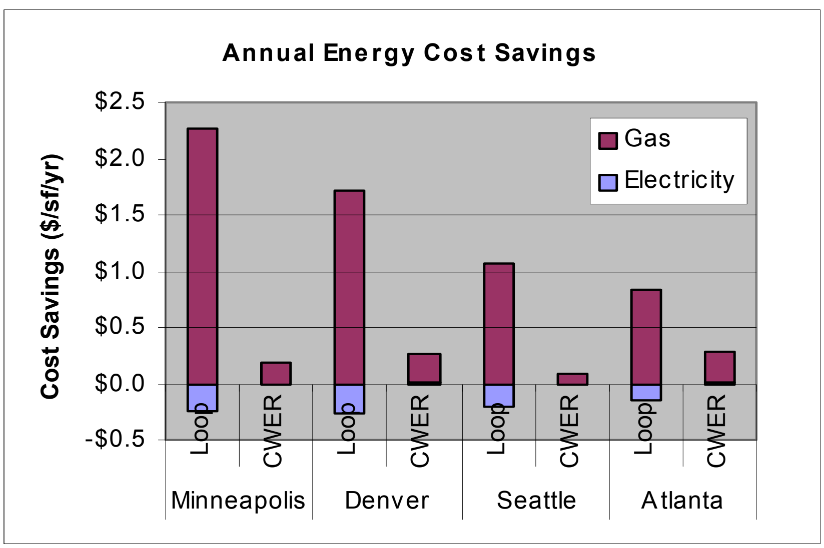


Figure 2.28b Annual energy cost savings for the base case with a run-around loop (Loop) and an energy recovery chiller (CWER).

2.4 Evaporative Cooling

Evaporative cooling cools air through direct contact with water or indirect cooling with water. Direct evaporative cooling is suitable in dry climates, such as Denver's, which benefit from added humidity. Indirect evaporative cooling may function as a water-side economizer or evaporative precooling stage. With a water-side economizer, the chilled water loop is coupled to the cooling tower through a heat exchanger, and so-called "free cooling" of the chilled water is achieved. There is still significant pumping and cooling tower energy use, although it is more efficient than running the chillers. With evaporative precooling, evaporatively cooled chilled water circulates through a heat exchanger in the supply air duct. The drawback to this approach is the increase in pressure drop for the supply fan. Note that evaporative cooling strategies reduce space cooling energy use, which in the base case is only 4% of electricity use in Seattle, 6% in Denver, 10% in Minneapolis, and 17% in Atlanta.

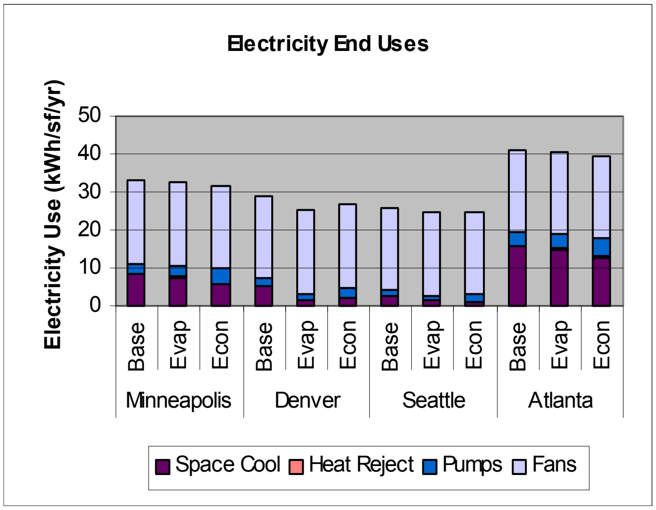


Figure 2.29 Electricity end uses for the base case, the base case with direct evaporative cooling stage (Evap), and the base case with a water-side economizer (Econ).

Figure 2.29 presents electricity end uses for the base case with a constant-volume reheat system, the base case with a direct evaporative cooling stage (Evap), and the base case with a water-side economizer (Econ). Figure 2.30 shows annual energy costs for each of the cases.

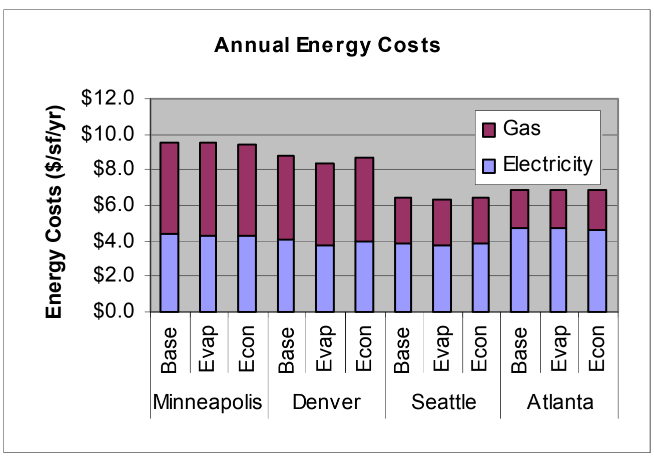


Figure 2.30 Annual energy costs for the base case, the base case with direct evaporative cooling stage (Evap), and the base case with a water-side economizer (Econ).

With direct evaporative cooling, supply air flows through a wet media or spray (i.e., atomization) that cools and adds moisture to the air. The effectiveness of the direct evaporative cooling stage is 0.8 with a pressure drop of 0.1 in. w.g. across the atomization evaporative cooler. (The pressure drop across a wetted media is 0.5 in. w.g.) The results show that direct evaporative cooling is suitable for drier climates, such as Denver's, where simultaneous space cooling and humidification of the air are required. In Minneapolis and Atlanta, energy cost savings are negligible. In Denver, the energy cost savings are $0.44/sf. In Seattle, the savings are $0.09/sf.

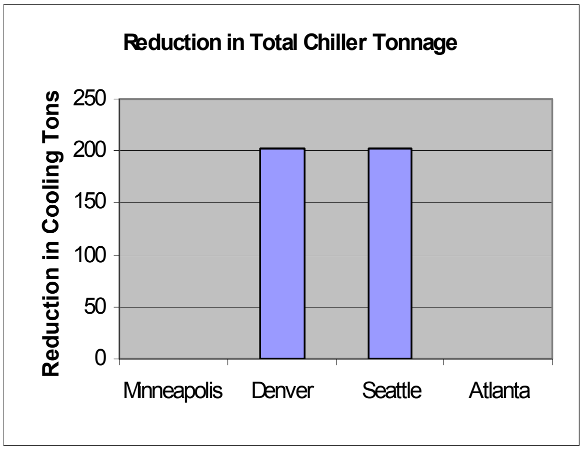


Figure 2.31 Reduction in chiller size with direct evaporative cooling stage.

Additional benefits with evaporative cooling include a reduction in peak demand and the potential to reduce the size of the chiller. In Denver, peak demand drops by 2 W/sf in mid-summer with direct evaporative cooling. This is equivalent to having the lights off 100% of the time. Figure 2.31 shows the potential reduction in chiller capacity resulting from the use of direct evaporative cooling. In Denver and Seattle, first-cost savings from downsizing the chillers are on the order of $2/sf.

The water-side economizer has a heat-exchanger effectiveness of 0.8. The simulations show that the water-side economizer is more effective than the direct evaporative cooling at reducing cooling energy use in all climates except Denver's (Figure 2.30). Annual energy cost savings with a water-side economizer are $0.80/sf in Minneapolis, $0.12/sf in Denver, $0.04/sf in Seattle, and $0.05/sf in Atlanta. The water-side economizer also reduces the number of hours the chiller operates at low part loads. In Minneapolis, Denver, and Seattle, the lead chiller runs at 20% or less part load over 70% of the total hours the chiller operates in the base case. The water-side economizer reduces these hours by 66% in Minneapolis, 85% in Denver, 63% in Seattle, and 39% in Atlanta. However, the DOE-2.2 water-side economizer model does not allow an economizer and chiller to operate together, so the utility of the economizer is underestimated. In addition, DOE-2.2 does not predict the potential reduction in chiller capacity obtained with a water-side economizer.

We ran additional evaporative cooling options for Denver. With direct evaporative cooling and a water-side economizer, DOE-2.2 predicts savings with the economizer that equal those with direct evaporative cooling plus half the savings predicted for the economizer alone. We also ran the evaporative cooling with a run-around loop. Adding the direct evaporative cooling stage to the case with the run-around loop saves an additional $0.30/sf.

2.5 Humidity Controls

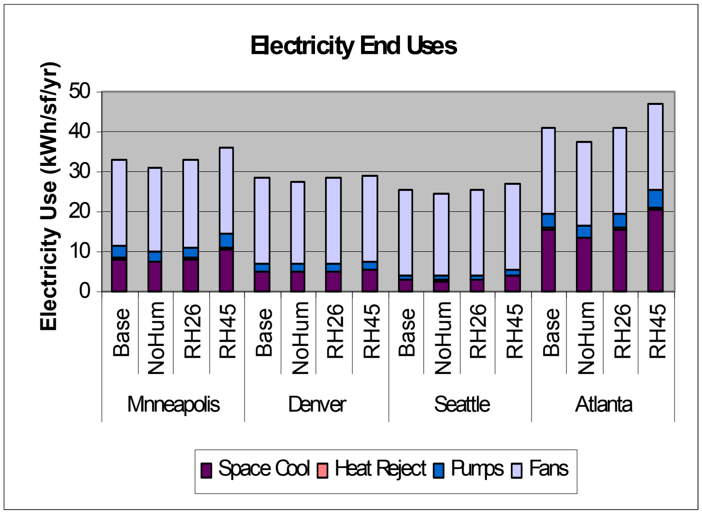


Figure 2.32 Electricity end uses for the base case and cases with no humidity controls (NoHum), minimum of 20% and maximum of 60% (RH26), and minimum of 40% and maximum of 50% (RH45) relative humidity.

Laboratory buildings often require tight control over temperature and humidity levels. The base simulations assume a minimum relative humidity of 30% and a maximum relative humidity of 60%, and the simulations show that this level of humidity control is energy-intensive. We considered a range of humidity control levels to assess the associated energy use. Figure 2.32 compares electricity use, excluding lighting and equipment end uses, for the base case with no humidity controls (NoHum), one with a minimum setting of 20% and a maximum setting of 60% (RH26), and one with a minimum setting of 40% and a maximum setting of 50% (RH45). Figure 2.33 compares space heating energy use for the same runs, and Figure 2.34 gives annual energy costs. Table 2.4 reports the number of hours the relative humidity is less than 30% and greater than 60% for the building without humidity controls.

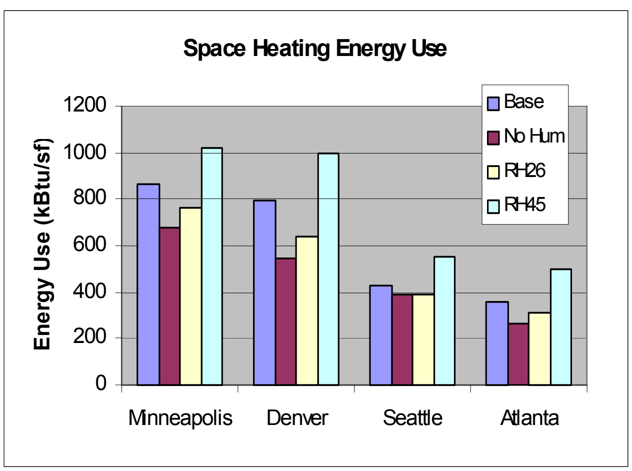


Figure 2.33 Space heating energy use for the base case and cases with no humidity controls (NoHum), minimum of 20% and maximum of 60% (RH26), and minimum of 40% and maximum of 50% (RH45) relative humidity.

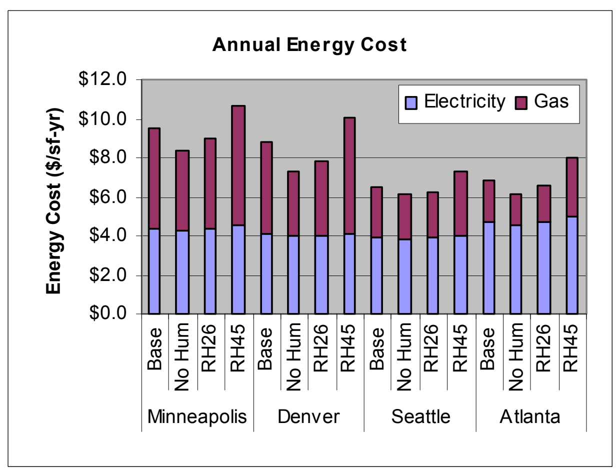


Figure 2.34 Annual energy costs for the base case, no humidity controls (NoHum), minimum RH of 20% and maximum RH of 60% (RH26), and minimum RH of 40% and maximum RH of 50% (RH45).

Of the four cities, Minneapolis and Atlanta have the most humid hours, Denver has the most dry hours, and Seattle is the most temperate. As expected, the case without humidity controls is the least energy-intensive. The case with 20% minimum relative humidity and 60% relative humidity settings has almost no impact on energy use in Seattle, and a small impact in the other climates. Minneapolis, because of its extreme climate, is the most energy-intensive.

Table 2.4 Number of Hours Relative Humidity (RH) is Greater Than 60% RH or Less Than 30% RH

Climate	Hours >60% RH	Hours <30% RH and >20% RH	Hours < 20% RH
Minneapolis	620	1300	3490
Denver	10	1840	4690
Seattle	5	2520	730
Atlanta	1740	1120	1550

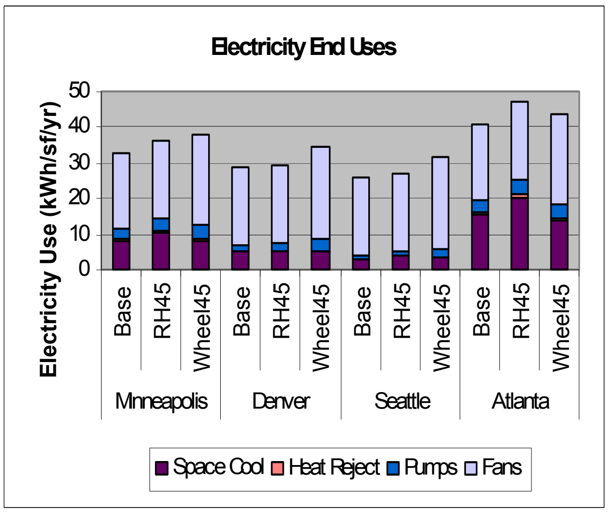


Figure 2.35 Electricity end uses for the base case, relative humidity controls of 40% and 50% (RH45), and enthalpy wheel with tighter humidity controls (Wheel45).

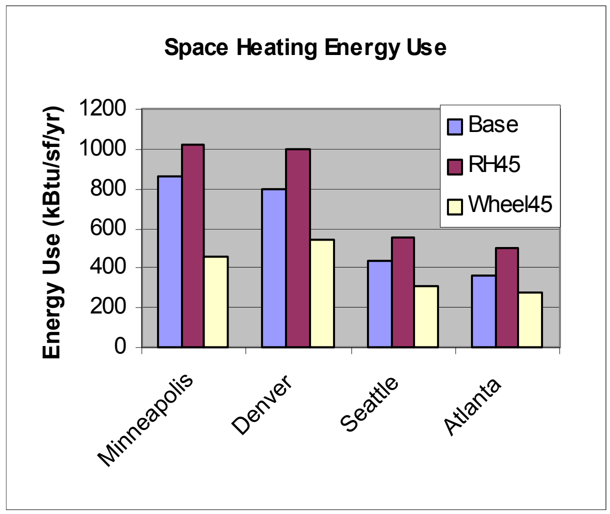


Figure 2.36 Space heating energy use for the base case, relative humidity controls of 40% and 50% (RH45), and enthalpy wheel with tighter humidity controls (Wheel45).

In addition, an enthalpy wheel was modeled with tighter humidity controls. Figure 2.35 shows electricity end uses, excluding lighting and equipment, for the base case, the case with tighter humidity controls (RH45), and the enthalpy wheel with tighter humidity controls (Wheel45). Figure 2.36 shows space heating energy use. As expected, electricity use and gas use increase with tighter humidity controls in all three cases. The only climate in which the enthalpy wheel lowers electricity use is Atlanta's.

Figure 2.37 presents annual energy costs for each case. The tighter humidity controls increase energy costs by $1.10/sf in Minneapolis, $1.30/sf in Denver, $0.80/sf in Seattle, and $1.20/sf in Atlanta. The enthalpy wheel saves $3.50/sf in Minneapolis,

$2.60/sf in Denver, $1.30/sf in Seattle, and $1.60/sf in Atlanta, in comparison to costs for the case with tighter humidity controls.

Figure 2.38a and b show the potential reduction in chiller and boiler capacity with the enthalpy wheel and tighter humidity controls. The reduction in chiller size is the same as with the enthalpy wheel applied to the base case. However, the reduction in boiler size is less because of the increase in humidification needs.

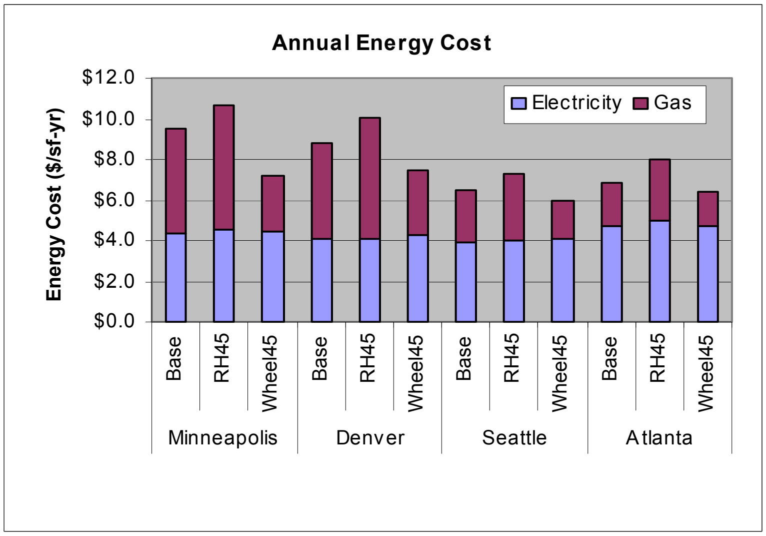
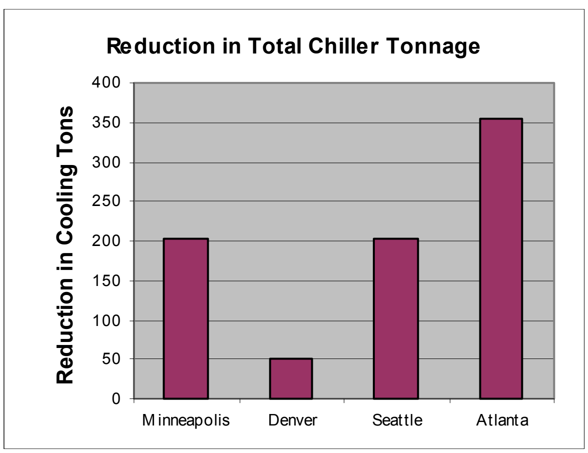
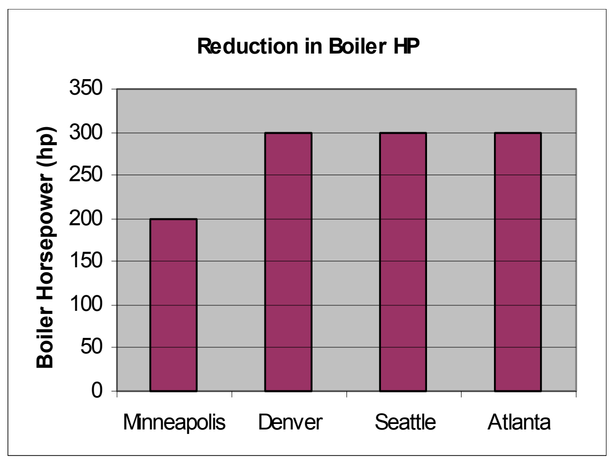


Figure 2.37 Annual energy costs for the base case, case with humidity controls of 40% and 50% (RH45), and case with tighter humidity controls and an enthalpy wheel (Wheel45).





Figures 2.38a and b Reduction in chiller tonnage and boiler horsepower for enthalpy wheel (Wheel45) with constant-volume reheat system and tighter humidity controls.

2.6 Plug Loads and Sizing Equipment

Internal gains from equipment drive the sizing of the mechanical equipment serving laboratories. Generally, the equipment power density, or plug load, is based on historic design values with an expectation of future increases. However, as more and more laboratories employ submetering to track energy loads, building managers are finding that plug loads remain significantly lower than the design assumption.

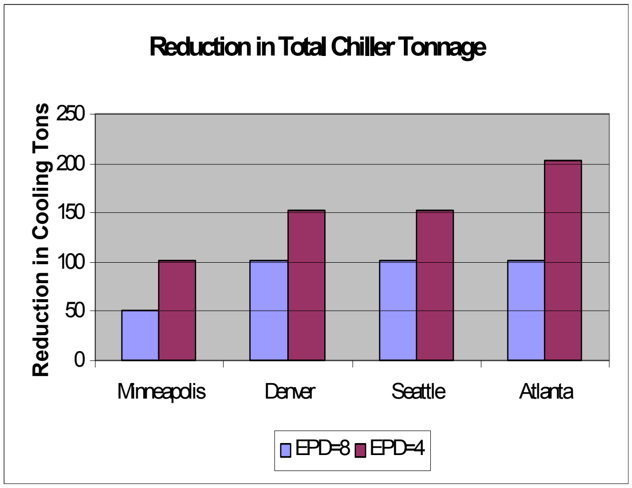


Figure 2.39 Results for potential size reduction in total chiller capacity.

To demonstrate the influence of the design assumption for plug loads on the sizing of mechanical equipment, we have modeled three different plug load assumptions for the laboratory spaces: (1)12 W/sf (base case), (2) 8 W/sf, and (3) 4 W/sf. Area-weighting these values with the other spaces gives 8.8 W/sf in the base case, 6 W/sf for the second case, and 3.2 W/sf for the third case. Note that the second case is modeled using 7 W/sf in the perimeter zones and 5.8 W/sf in the core zones. The third case uses 3.5 W/sf in the perimeter and 3.1 W/sf in the core zones. Recall that the base case uses 10 W/sf in the perimeter and 8.5 W/sf in the core zones.

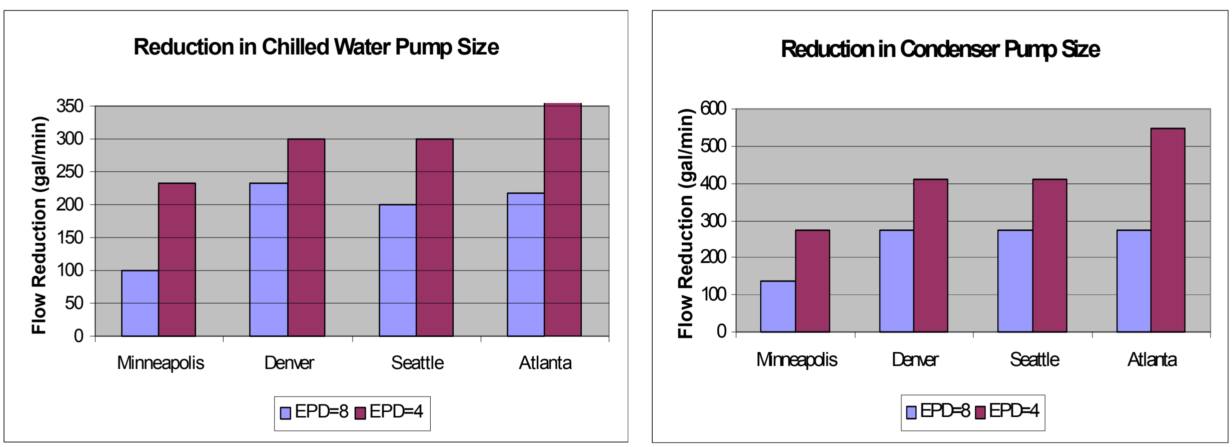


Figure 2.40a and b Results for potential size reduction in chilled water and condenser water pumps.

The simulation results show a total savings of 50 tons in Minneapolis and a savings of 100 tons in the other climates, for an equipment power density of 8 W/sf (EPD=8) in the laboratory spaces. For an equipment power density of 4 W/sf in the laboratory spaces (EPD=4), the chiller reduction in Minneapolis is 100 tons, the reduction in Denver and Seattle is 150 tons, and the reduction in Atlanta is 200 tons. The simulation models assume two chillers, and the savings represent the total reduction in cooling tonnage (Figure 2.39). The capacity of the cooling tower drops by about 10% more than that of the chillers.

The lower cooling load from the reduced internal gains from equipment also results in smaller pumps. Figure 2.40a shows the potential reduction in pump sizes in terms of gallons per minute. The simulation model assumes that there are two primary chilled water pumps and two secondary chilled water pumps. The figure shows the potential size reduction for each pump. A similar but slightly greater reduction is seen for the condenser pumps (Figure 2.40b).

Recall that a minimum outside air flow rate of 2 cfm/sf has been set for all of the models. This is not sufficient, however, to meet the design cooling load with an equipment power density of 12 W/sf in the lab spaces. The simulations show that 2.7 cfm/sf is required in the perimeter zones and 2.0 cfm/sf is required in the core. For an equipment power density of 8 W/sf in the lab spaces, the perimeter requires 2.2 cfm/sf under design cooling conditions. For an equipment power density of 4 W/sf in the lab spaces, the perimeter requires 2 cfm/sf under design cooling conditions. So, there are opportunities to downsize fans as well as the chilled water system. Table 2.5 gives the reduction in cfm for the perimeter fan.

Table 2.5 Reduction in Fan Flow for Perimeter Fan

	Flow Reduction
EPD=8 W/sf	0.5 cfm/sf
EPD=4 W/sf	0.7 cfm/sf

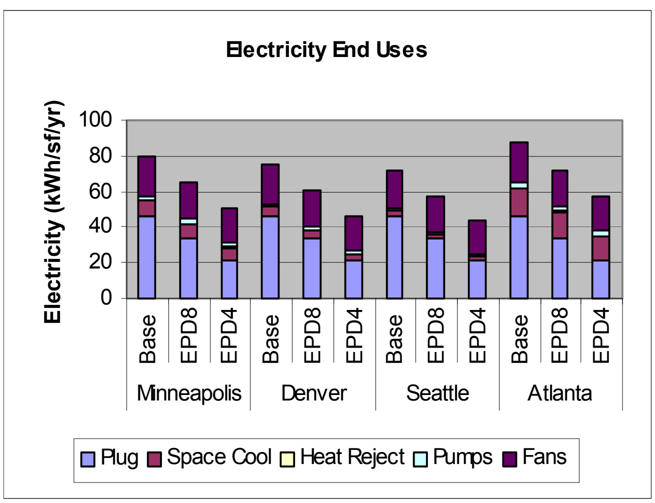


Figure 2.41 Electricity end uses for the base case and lower plug loads of 8 W/sf (EPD8) and 4 W/sf (EPD4).

The base case model assumes constant-volume fans, so the reduction in air flow with lower equipment power densities causes a reduction in the size of boilers and associated pumps. We recognize the trend to use VFDs on fans, so we have not included the potential downsizing of the heating equipment.

In terms of annual energy use, the electricity use for plug loads drops by 12 kWh/sf/yr with a plug load of 8 W/sf and by 25 kWh/sf/yr with a plug load of 4 W/sf (Figure 2.41), regardless of climate. The lower plug loads result in less electricity use for space cooling, fans, heat rejection (cooling towers), and pumps. On the space heating side, associated energy use decreases in Minneapolis and Denver and increases in Seattle and Atlanta (Figure 2.42). In Minneapolis and Denver, the heating energy use is lower because of the lower humidification load in winter. Although space heating energy use is higher in Seattle and Atlanta, it is less expensive to heat with natural gas than with electricity (Figure 2.43). Annual energy costs are $8.70/sf in Minneapolis, $7.90/sf in Denver, $5.80/sf in Seattle, and $6.20/sf in Atlanta, assuming 8 W/sf in the laboratories (6 W/sf average for the building). An assumption of 6 W/sf for the building is more in line with measured plug loads in the Labs 21 case studies.

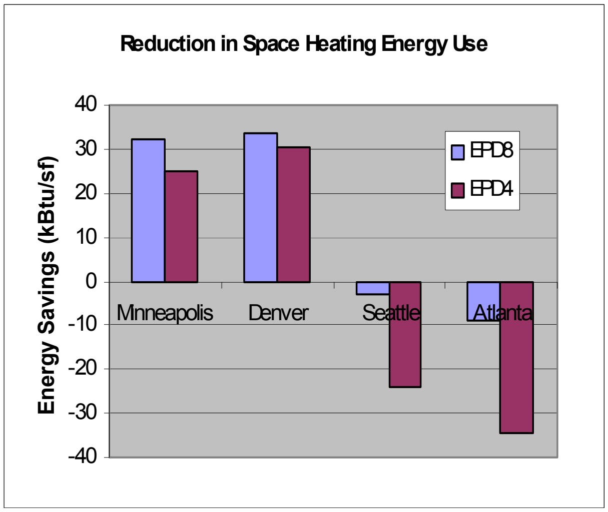


Figure 2.42 Space heating energy savings for lower plug loads of 8 W/sf (EPD8) and 4 W/sf (EPD4).

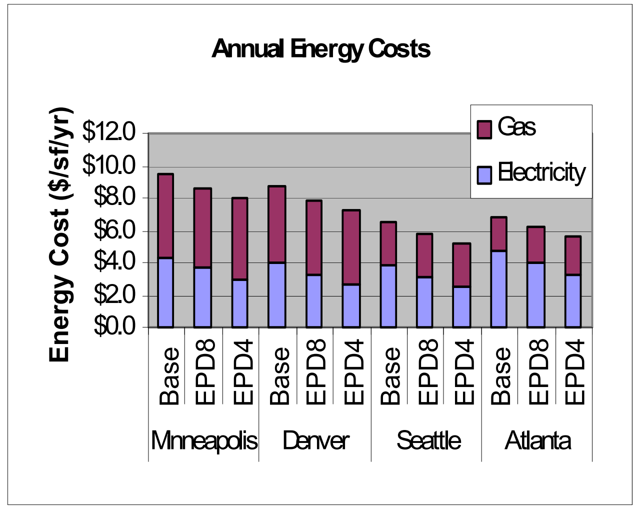


Figure 2.43 Annual energy cost savings for lower plug loads of 8 W/sf (EPD8) and 4 W/sf (EPD4).

2.7 Summary of the Simulation Results

The primary objective of this work is to assess the impact of energy efficiency strategies on the energy used in laboratory buildings. The DOE-2.2 building energy simulation program was employed to evaluate a range of energy efficiency strategies. This study focuses on ventilation and space heating loads, which are the most energy-intensive of the end uses in laboratory buildings.

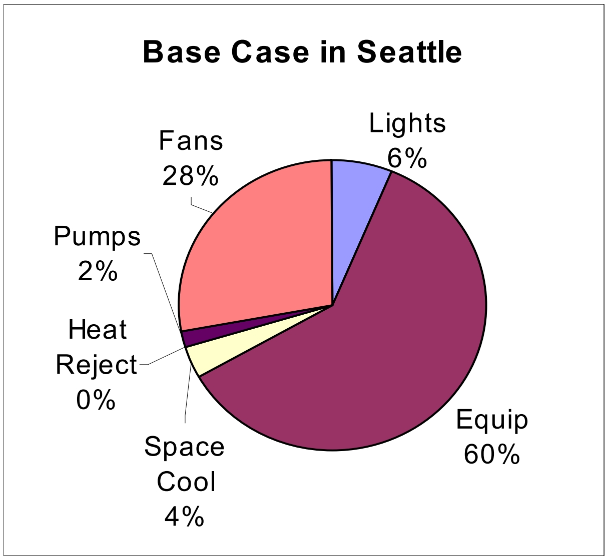
Although predicted energy savings differ from climate to climate, the most efficient measures are the same for all four climates. On the basis of the simulation results, we conclude the following:

- Using a variable-air-volume system rather than a constant-volume system reduces the energy use of fans and space cooling and heating equipment by a minimum of 10%.
- The VAV system reduces peak demand by 2 W/sf and annual electricity use by 12 kWh/sf in Atlanta. Reductions for Minneapolis and Denver are 1 W/sf in peak demand and 8 kWh/sf (Minneapolis) and 5 kWh/sf (Denver) in annual electricity use. Peak demand savings in Seattle are less than 0.5 W/sf, and annual

electricity savings are 6 kWh/sf. The resulting savings in gas usage vary significantly from climate to climate.

- Some form of energy recovery should always be considered. Because of the sensible and latent energy recovery achieved with enthalpy wheels, they are the most efficient of the energy recovery alternatives considered here. DOE-2.2 predicts 1-2 W/sf savings in peak demand and little-to-no electricity savings. Gas usage savings are significant, however, and vary from climate to climate.
- Energy recovery has the potential to reduce the size of heating and cooling equipment, and first-cost savings will cover a large portion of the cost of energy recovery equipment. In laboratories, a concern about the failure of the energy recovery unit may keep the design team from downsizing chillers and boilers. However, because of the redundancy often designed into a laboratory's mechanical equipment, the reserved chiller and boiler could serve as backups to the energy recovery unit.
- Because of the high ventilation requirements of laboratory buildings, the air distribution system should be optimized to minimize pressure drop through the system and reduce energy use.
- Humidity control is energy-intensive and should be carefully integrated into control strategies to minimize reheat and subcooling.
- Plug loads and internal gains from plug loads should be accurately assessed to design the mechanical system and determine power requirements. A significant increase in costs results from employing a design load that is too high.

Simulations were also done that combine these efficiency measures into a single run. The "advanced" runs include the VAV system with a run-around loop or enthalpy wheel, a supply static pressure drop of 4 in. w.g., and an exhaust static pressure drop of 1.5 in. w.g. We chose to model static pressure drop at 4 in. w.g. because of the challenges a designer faces in planning laboratory buildings and filtering air. An additional pressure drop from the energy recovery device is added to the supply and exhaust.



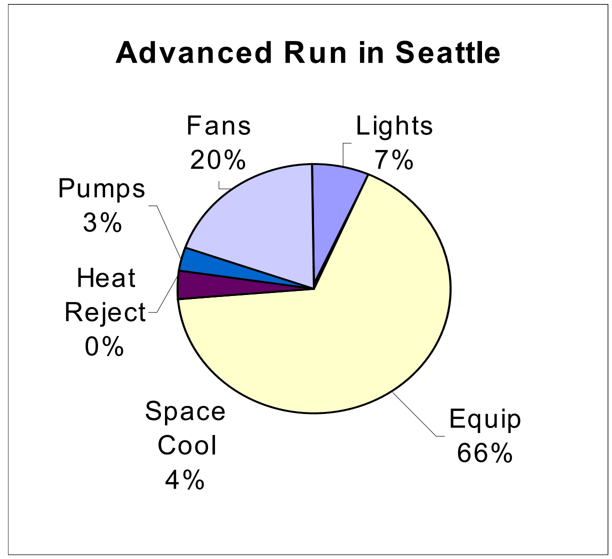


Figure 2.44a and b Percent breakdown of electricity end uses for the base case (a) and advanced run with an enthalpy wheel (b) in Seattle.

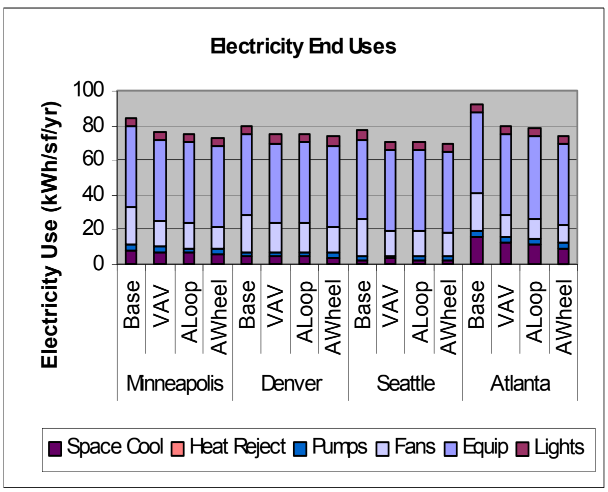


Figure 2.45 Electricity end uses for the base case, VAV system (VAV), and the advanced case with a run-around loop (Aloop) and an enthalpy wheel (AWheel).

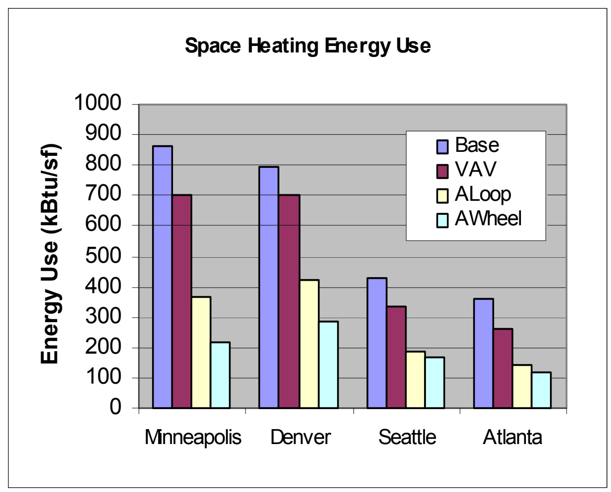


Figure 2.46 Space heating energy use for the base case, VAV system (VAV), and the advanced case with a run-around loop (Aloop) and an enthalpy wheel (AWheel).

Figures 2.44a and b compare the percent electricity end uses for the base case and the advanced run with an enthalpy wheel in Seattle. The relative importance of the fans has decreased by 8%, although the associated energy use is still significant. Figure 2.45 shows the electricity end uses for the base case, the VAV system, and the advanced runs (ALoop and AWheel). Electricity use for all end uses are lower in the advanced runs than in the other two runs, except for lights and pumping energy, which is inaccurate as modeled by DOE-2.2.

Space heating energy use is one-half to three-fourths of total energy use in all climates for the base case. This is attributable to the use of a constant-volume system, humidity requirements, and space heating requirements at night for this 24-hour, 7-days-a-week laboratory. Figure 2.46 shows space heating energy use for the base case, the VAV system, and the advanced runs (ALoop and AWheel). In the advanced cases, space heating energy use decreases by 57% in Minneapolis, 47% in Denver, 56% in Seattle, and 60% in Atlanta. The space heating savings for Denver are less than those for the other climates because of Denver's high humidification requirements.

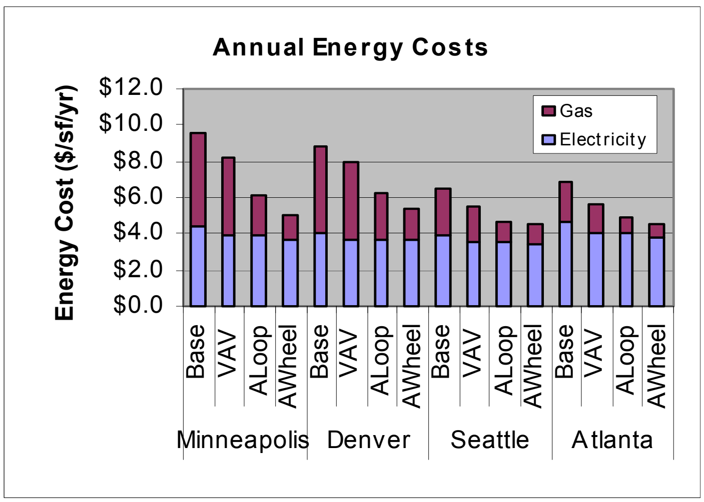


Figure 2.47 Annual energy costs for the base case, VAV system (VAV), and advanced case with a run-around loop (Aloop) and an enthalpy wheel (AWheel).

Figure 2.47 presents annual energy costs for the four cases. The VAV system alone saves close to $1/sf in energy costs in all climates. Annual energy cost savings for the advanced case and the enthalpy wheel range from $2/sf in Seattle to $4.50/sf in Minneapolis. Savings for the advanced case and the run-around loop are less than those for the advanced case with the enthalpy wheel by $1/sf in Minneapolis, $0.90/sf in Denver, $0.20/sf in Seattle, and $0.40/sf in Atlanta.

Figure 2.48 shows annual energy cost savings in terms of percent reduction with respect to the base case. The percent savings are calculated to include all energy end uses (% Savings) and per the LEED method (% LEED Savings). LEED stands for Leadership in Energy and Environmental Design; it is a system for rating the relative energy and environmental performance of a commercial building. One of the LEED rating system credits awards points for energy cost savings in comparison to costs for a building that complies with ASHRAE 90.1-99. The energy cost savings exclude equipment plug loads from the cost calculation, although they are included in building energy simulations. ASHRAE 90.1-99 does not clearly define a code-compliant laboratory building, so we have assumed the constant-volume air system to be code-compliant. Savings of 60% or greater earn the maximum 10 points for this credit. The advanced case with the enthalpy wheel achieves percent LEED savings of 53% in Denver, 44% in Seattle and Atlanta, and 63% in Minneapolis. The

VAV system would earn the minimum 2 points for 20% savings in Minneapolis, Seattle, and Atlanta. Savings in Denver are lower at 13%.

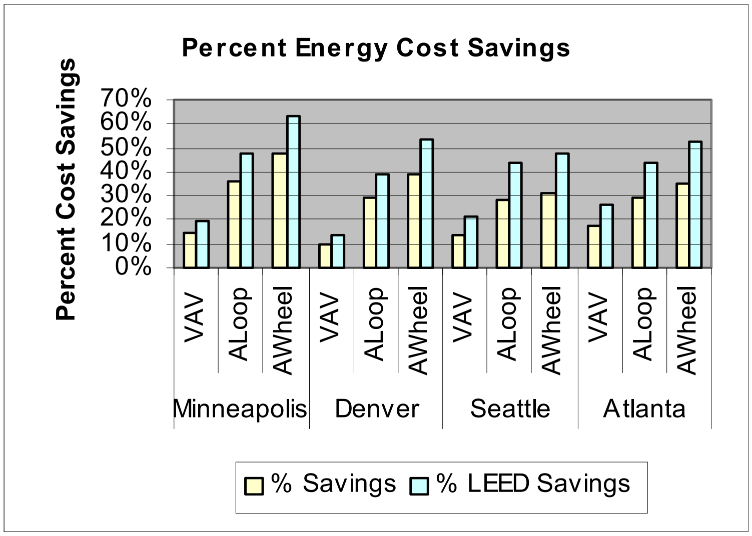


Figure 2.48 Percent energy cost and LEED savings relative to the base case for the VAV system and the advanced case with a run-around loop (Aloop) and an enthalpy wheel (AWheel).

The design implications of employing these measures and design assumptions have the potential to reduce mechanical and power system sizes. This includes fans, pumps, chillers, boilers, and power supplied to the facility. A VAV system, energy recovery, and lower static pressure do not change design air flow requirements.

Figure 2.49 shows the potential savings associated with downsizing chillers in each climate. The simulations predict a savings of 200 tons in Denver and Seattle, 400 tons in Minneapolis and 500 tons in Atlanta for the advanced case

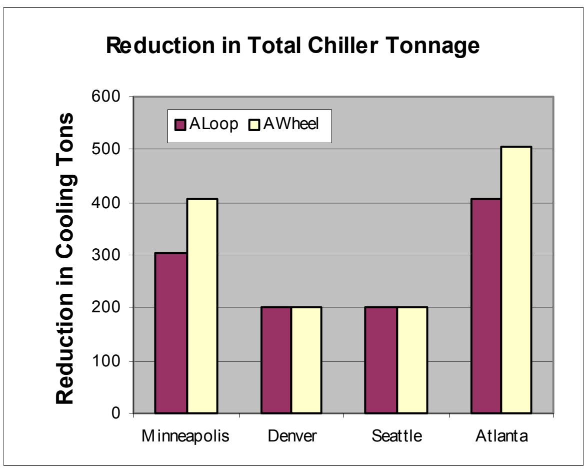


Figure 2.49 Reduction in chiller capacity for the advanced case with a run-around loop (Aloop) and an enthalpy wheel (AWheel).

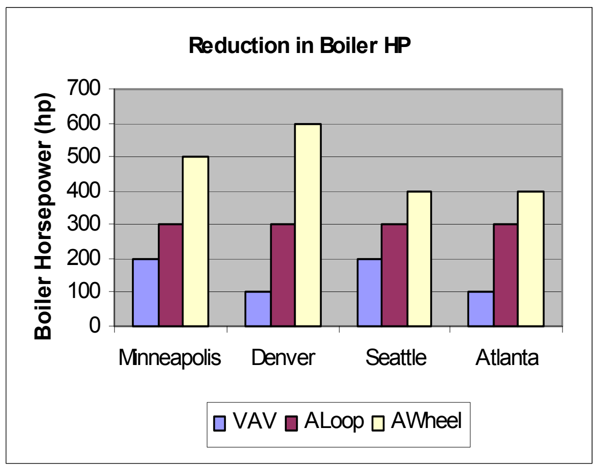


Figure 2.50 Reduction in boiler capacity for the advanced case with a run-around loop (Aloop) and an enthalpy wheel (AWheel).

with the enthalpy wheel in comparison to the base case. The greatest savings occur in climates with the largest cooling loads. In the more humid climates of Minneapolis and Atlanta, the enthalpy wheel could reduce the chiller capacity by 100 tons more than the run-around loop would. Figure 2.50 shows the potential reduction in boiler size. The greatest savings are possible in Denver with the enthalpy wheel, because the high humidification requirement is coupled with space heating needs. Potential savings in Minneapolis are also significant, at 500 hp.

The results of the simulation runs for each climate are summarized in the sections that follow. The most efficient measures are the same for all climates and are highlighted in gray, although predicted energy savings differ from climate to climate. The results are presented per square foot of gross area in the building, i.e., 100,000 sf. The energy use numbers include lighting and plug loads.

Note that a difference in annual electricity use of 1 kWh/sf is equivalent to approximately $5,000 for the whole building, and this is not an insignificant amount. Along the same lines, a difference in peak demand of 1 W/sf is about as much as a facility can achieve by turning more than 50% of the lights off during times of peak use. The results also include the percent reduction in energy costs as a total and per the LEED method relative to the base case.

2.7.1 Minneapolis

Minneapolis has the coldest climate of the four cities in the study. It has humid summers and dry winters, which translate into dehumidification and humidification requirements. Humidification accounts for 190 kBtu/sf of gas usage in the base case, or 22% of the space heating energy use. A VAV system reduces all the mechanical energy end uses, including dehumidification and humidification needs. The simulation results predict that the VAV system alone reduces peak demand by 1 W/sf, annual electricity use by 8 kWh/sf, and annual gas usage by 160 kBtu/sf. Electricity costs drop by $0.50/sf and gas costs drop by $1/sf (Tables 2.6a and 2.6b).

Table 2.6a Summary of DOE-2.2 Energy Simulation Results for Minneapolis

	Peak Demand (W/sf)	Electricity Use (kWh/sf/yr)	Gas Use (kBtu/sf/yr)	Total Energy Use (kBtu/sf/yr)
Base Case	20	84	861	1125
Flow Setback (CFM21)	20	81	839	1094
VAV	19	76	701	940
Supply Static Pressure of 4 in. w.g. (SP4)	20	79	874	1121
Supply Static Pressure of 3 in. w.g. (SP3)	19	74	885	1118
Enthalpy Wheel (Wheel)	17	86	300	570
Enthalpy Wheel w/ VAV (VWheel)	17	77	216	458
Heat Pipe (HtPipe)	21	90	508	789
Run-Around Loop (Loop)	21	90	482	765
Chiller Energy Recovery (CWER)	20	85	829	1094
Direct Evap. Cooling (Evap)	21	84	865	1128
Water-side Economizer (Econ)	20	83	861	1122
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	20	84	764	1028
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	20	87	1022	1296
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	18	89	458	738
Lab Plug Loads 8 W/sf(EPD8)	18	70	829	1048
Lab Plug Loads 4 W/sf(EPD4)	16	56	836	1011
Advanced w/Run-Around Loop (ALoop)	18	75	367	602
Advanced w/Enthalpy Wheel (AWheel)	16	73	214	444

Table 2.6b Summary of Energy Costs for Minneapolis

	Annual Electricity Cost ($/sf)	Annual Gas Cost ($/sf)	Annual Energy Cost ($/sf)	% Reduction in Energy Costs	% Reduction per LEED
Base Case	$4.40	$5.20	$9.50		
Flow Setback (CFM21)	$4.20	$5.00	$9.20	3%	4%
VAV	$3.90	$4.20	$8.10	14%	19%
Supply Static Pressure of 4 in. w.g. (SP4)	$4.10	$5.20	$9.40	2%	2%
Supply Static Pressure of 3 in. w.g. (SP3)	$3.90	$5.30	$9.20	3%	4%
Enthalpy Wheel (Wheel)	$4.30	$1.80	$6.10	36%	48%
Enthalpy Wheel w/ VAV (VWheel)	$3.90	$1.30	$5.20	45%	60%
Heat Pipe (HtPipe)	$4.60	$3.00	$7.60	20%	26%
Run-Around Loop (Loop)	$4.60	$2.90	$7.50	21%	28%
Chiller Energy Recovery (CWER)	$4.40	$5.00	$9.30	2%	2%
Direct Evap. Cooling (Evap)	$4.30	$5.20	$9.50	0%	0%
Water-side Economizer (Econ)	$4.30	$5.20	$9.50	1%	1%
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$4.40	$4.60	$8.90	6%	8%
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$4.50	$6.10	$10.60	-12%	-16%
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$4.40	$2.80	$7.20	24%	32%
Lab Plug Loads 8 W/sf(EPD8)	$3.70	$5.00	$8.60	9%	
Lab Plug Loads 4 W/sf(EPD4)	$3.00	$5.00	$8.00	16%	
Advanced w/Run-Around Loop (ALoop)	$3.90	$2.20	$6.10	36%	48%
Advanced w/Enthalpy Wheel (AWheel)	$3.70	$1.30	$5.00	48%	63%

Adding an enthalpy wheel to the VAV system reduces peak demand another 2 W/sf, but it increases electricity use by 1 kWh/sf. The electricity costs are equal at $3.90/sf. The enthalpy wheel reduces gas use by an additional 485 kBtu/sf, which equates to a savings of $2.90/sf in gas costs. The VAV system alone saves $1.40/sf, and adding an enthalpy wheel saves another $2.90/sf in annual energy costs.

An enthalpy wheel is not suitable for all applications, but one of the other energy recovery methods may be appropriate. Heat pipes and run-around loops increase peak demand by 1 W/sf, electricity use by 14 kWh/sf, and annual electricity costs by $0.20/sf because of the increase in fan energy compared with that of the constant-volume base case. Heat pipes reduce gas use by 353 kBtu/sf and gas costs by $2.20/sf. Run-around loops reduce gas use by 379 kBtu/sf and gas costs by $2.30/sf. We also simulated the run-around loop with the advanced case (i.e., VAV and lower static pressure drop). It saves an additional $2/sf over the VAV system alone.

2.7.2 Denver

Denver has a relatively cold and very dry climate, and humidification requirements account for 250 kBtu/sf, or 32%, of the space heating energy use. Denver has 1000 more hours of outside-air RH below 20% than Minneapolis does. This translates into a significant humidification load, which shows up as a space heating load in DOE-2.2. The latent recovery from the enthalpy wheel offsets the humidification load significantly (Table 2.7a).

The simulations predict that a VAV system alone reduces peak demand by 1 W/sf, annual electricity use by 5 kWh/sf, and annual gas use by 91 kBtu/sf. Electricity costs drop by $0.30/sf, and gas costs drop by $0.60/sf.

Table 2.7a Summary of DOE-2.2 Energy Simulation Results for Denver

	Peak Demand (W/sf)	Electricity Use (kWh/sf/yr)	Gas Use (kBtu/sf/yr)	Total Energy Use (kBtu/sf/yr)
Base Case	16	80	792	1043
Flow Setback (CFM21)	16	78	780	1023
VAV	15	75	703	938
Supply Static Pressure of 4 in. w.g. (SP4)	15	75	804	1038
Supply Static Pressure of 3 in. w.g. (SP3)	15	70	815	1034
Enthalpy Wheel (Wheel)	15	84	332	596
Enthalpy Wheel w/ VAV (VWheel)	15	79	288	535
Heat Pipe (HtPipe)	17	86	508	779
Run-Around Loop (Loop)	17	87	507	779
Chiller Energy Recovery (CWER)	16	80	748	999
Direct Evap. Cooling (Evap)	14	76	766	1006
Water-side Economizer (Econ)	16	78	792	1037
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	16	80	635	886
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	16	80	999	1251
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	17	86	537	807
Lab Plug Loads 8 W/sf(EPD8)	13	65	758	964
Lab Plug Loads 4 W/sf(EPD4)	11	51	761	922
Advanced w/Run-Around Loop (ALoop)	15	75	421	655
Advanced w/Enthalpy Wheel (AWheel)	14	73	284	515
Advanced w/Run-Around Loop and Direct Evap. Cool	14	72	396	622

Table 2.7b Summary of Energy Costs for Denver

	Annual Electricity Cost ($/sf)	Annual Gas Cost ($/sf)	Annual Energy Cost ($/sf)	% Reduction in Energy Costs	% Reduction per LEED
Base Case	$4.00	$4.80	$8.80		
Flow Setback (CFM21)	$3.90	$4.70	$8.60	2%	3%
VAV	$3.70	$4.20	$8.00	10%	13%
Supply Static Pressure of 4 in. w.g. (SP4)	$3.80	$4.80	$8.60	2%	3%
Supply Static Pressure of 3 in. w.g. (SP3)	$3.60	$4.90	$8.50	4%	5%
Enthalpy Wheel (Wheel)	$4.20	$2.00	$6.20	30%	41%
Enthalpy Wheel w/ VAV (VWheel)	$3.90	$1.70	$5.60	36%	49%
Heat Pipe (HtPipe)	$4.30	$3.10	$7.30	17%	23%
Run-Around Loop (Loop)	$4.30	$3.00	$7.40	16%	22%
Chiller Energy Recovery (CWER)	$4.00	$4.50	$8.50	3%	4%
Direct Evap. Cooling (Evap)	$3.80	$4.60	$8.40	5%	7%
Water-side Economizer (Econ)	$3.90	$4.80	$8.70	1%	2%
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$4.00	$3.80	$7.90	11%	15%
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$4.10	$6.00	$10.10	-14%	-19%
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$4.30	$3.20	$7.50	15%	20%
Lab Plug Loads 8 W/sf(EPD8)	$3.30	$4.60	$7.90	10%	
Lab Plug Loads 4 W/sf(EPD4)	$2.60	$4.60	$7.20	18%	
Advanced w/Run-Around Loop (ALoop)	$3.70	$2.50	$6.30	29%	39%
Advanced w/Enthalpy Wheel (AWheel)	$3.60	$1.70	$5.40	39%	53%
Advanced w/Run-Around Loop and Direct Evap Cool	$3.60	$2.40	$6.00	32%	43%

Adding an enthalpy wheel to the VAV system reduces peak demand by another 1 W/sf and decreases electricity use by an additional 2 kWh/sf. The associated electricity costs decrease by $0.10/sf from the VAV case. The enthalpy wheel reduces gas use by an additional 419 kBtu/sf and saves $2.50/sf in gas costs. The VAV system saves $0.80/sf, and adding an enthalpy wheel saves another $2.60/sf in annual energy costs.

Space heating savings with the other energy recovery methods are significant. On one hand, heat pipes and run-around loops increase peak demand by 1 W/sf, increase electricity usage by 6 kWh/sf, and increase annual electricity costs by $0.30/sf because of fan energy increases in comparison to that of the constant-volume base case. On the other hand, heat pipes and run-around loops reduce gas usage by 284 kBtu/sf and gas costs by $1.80/sf. And the run-around loop with the advanced case saves an additional $1.70/sf over the VAV system. Adding direct evaporative cooling to this case saves an additional $0.30/sf.

2.7.3 Seattle

Seattle has the mildest climate of the four cities. Humidification is only 10%, or 45 kBtu/sf, of gas use in the base case. A VAV system reduces all mechanical energy end uses, including dehumidification and humidification needs. The simulation results predict that the VAV system alone reduces annual electricity use by 6 kWh/sf and annual gas use by 100 kBtu/sf. Electricity costs drop by $0.30/sf, and gas costs drop by $0.60/sf.

Table 2.8a Summary of DOE-2.2 Energy Simulation Results for Seattle

	Peak Demand (W/sf)	Electricity Use (kWh/sf/yr)	Gas Use (kBtu/sf/yr)	Total Energy Use (kBtu/sf/yr)
Base Case	15	77	431	673
Flow Setback (CFM21)	15	76	426	663
VAV	15	71	333	556
Supply Static Pressure of 4 in. w.g. (SP4)	15	72	443	668
Supply Static Pressure of 3 in. w.g. (SP3)	14	67	453	664
Enthalpy Wheel (Wheel)	15	80	221	473
Enthalpy Wheel w/ VAV (VWheel)	15	74	164	396
Heat Pipe (HtPipe)	16	82	254	510
Run-Around Loop (Loop)	16	82	252	510
Chiller Energy Recovery (CWER)	15	77	417	659
Direct Evap. Cooling (Evap)	14	76	432	670
Water-side Economizer (Econ)	16	76	431	670
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	15	77	392	633
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	16	78	550	795
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	15	83	309	568
Lab Plug Loads 8 W/sf(EPD8)	13	63	434	631
Lab Plug Loads 4 W/sf(EPD4)	10	49	455	608
Advanced w/Run-Around Loop (ALoop)	15	70	188	409
Advanced w/Enthalpy Wheel (AWheel)	14	70	166	385

Table 2.8b Summary of Energy Costs for Seattle

	Annual Electricity Cost ($/sf)	Annual Gas Cost ($/sf)	Annual Energy Cost ($/sf)	% Reduction in Energy Costs	% Reduction per LEED
Base Case	$3.90	$2.60	$6.50		
Flow Setback (CFM21)	$3.80	$2.60	$6.40	2%	2%
VAV	$3.60	$2.00	$5.60	14%	22%
Supply Static Pressure of 4 in. w.g. (SP4)	$3.60	$2.70	$6.30	3%	4%
Supply Static Pressure of 3 in. w.g. (SP3)	$3.40	$2.70	$6.10	5%	8%
Enthalpy Wheel (Wheel)	$4.00	$1.30	$5.30	18%	28%
Enthalpy Wheel w/ VAV (VWheel)	$3.70	$1.00	$4.70	28%	44%
Heat Pipe (HtPipe)	$4.10	$1.50	$5.60	13%	21%
Run-Around Loop (Loop)	$4.10	$1.50	$5.60	13%	21%
Chiller Energy Recovery (CWER)	$3.90	$2.50	$6.40	1%	2%
Direct Evap. Cooling (Evap)	$3.80	$2.60	$6.40	1%	2%
Water-side Economizer (Econ)	$3.80	$2.60	$6.40	1%	1%
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$3.90	$2.40	$6.20	4%	6%
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$4.00	$3.30	$7.30	-12%	-20%
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$4.10	$1.90	$5.90	8%	13%
Lab Plug Loads 8 W/sf(EPD8)	$3.20	$2.60	$5.80	11%	
Lab Plug Loads 4 W/sf(EPD4)	$2.50	$2.70	$5.20	19%	
Advanced w/Run-Around Loop (ALoop)	$3.50	$1.10	$4.70	28%	44%
Advanced w/Enthalpy Wheel (AWheel)	$3.50	$1.00	$4.50	31%	48%

Adding an enthalpy wheel to the VAV system increases electricity use by 3 kWh/sf and electricity costs by $0.10/sf. The enthalpy wheel reduces gas use by an additional 169 kBtu/sf for $1/sf in gas costs. In summary, the VAV system saves $0.90/sf and adding an enthalpy wheel saves another $0.90/sf in annual energy costs.

The other energy recovery methods have the same end results. Heat pipes and run-around loops increase electricity use by 5 kWh/sf, and annual electricity costs by $0.20/sf because of the increase in fan energy compared with that of the constant-volume base case. They both reduce gas use by 177 kBtu/sf and gas costs by $1.10/sf. This is the only climate in which an enthalpy wheel has a small advantage over other recovery methods, because there is no need for humidity control.

2.7.4 Atlanta

Atlanta has the warmest climate of the four cities; it requires both dehumidification and humidification. Humidification accounts for 96 kBtu/sf of gas use in the base case, or 26% of the space heating energy use. A VAV system reduces all mechanical energy end uses, including dehumidification and humidification needs. The simulation results predict that the VAV system alone reduces peak demand by 2 W/sf, annual electricity use by 8 kWh/sf, and annual gas use by 100 kBtu/sf. Electricity costs decrease by $0.60/sf, and gas costs drop by $0.60/sf.

Table 2.9a Summary of DOE-2.2 Energy Simulation Results for Atlanta

	Peak Demand (W/sf)	Electricity Use (kWh/sf/yr)	Gas Use (kBtu/sf/yr)	Total Energy Use (kBtu/sf/yr)
Base Case	20	92	362	652
Flow Setback (CFM21)	20	87	339	611
VAV	18	80	263	514
Supply Static Pressure of 4 in. w.g. (SP4)	20	87	374	647
Supply Static Pressure of 3 in. w.g. (SP3)	19	82	384	642
Enthalpy Wheel (Wheel)	17	90	187	470
Enthalpy Wheel w/ VAV (VWheel)	16	79	117	364
Heat Pipe (HtPipe)	20	96	231	532
Run-Around Loop (Loop)	20	96	225	527
Chiller Energy Recovery (CWER)	20	92	314	604
Direct Evap. Cooling (Evap)	21	92	365	654
Water-side Economizer (Econ)	20	91	363	647
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	20	92	315	604
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	21	98	500	809
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	18	95	278	575
Lab Plug Loads 8 W/sf(EPD8)	18	77	371	614
Lab Plug Loads 4 W/sf(EPD4)	15	63	397	593
Advanced w/Run-Around Loop (ALoop)	17	78	144	390
Advanced w/Enthalpy Wheel (AWheel)	16	74	116	350

Table 2.9b Summary of Energy Costs for Atlanta

	Annual Electricity Cost ($/sf)	Annual Gas Cost ($/sf)	Annual Energy Cost ($/sf)	% Reduction in Energy Costs	% Reduction per LEED
Base Case	$4.70	$2.20	$6.90		
Flow Setback (CFM21)	$4.50	$2.00	$6.50	5%	8%
VAV	$4.10	$1.60	$5.70	17%	26%
Supply Static Pressure of 4 in. w.g. (SP4)	$4.40	$2.20	$6.70	3%	4%
Supply Static Pressure of 3 in. w.g. (SP3)	$4.20	$2.30	$6.50	5%	7%
Enthalpy Wheel (Wheel)	$4.50	$1.10	$5.60	18%	28%
Enthalpy Wheel w/ VAV (VWheel)	$4.00	$0.70	$4.70	32%	48%
Heat Pipe (HtPipe)	$4.80	$1.40	$6.20	9%	14%
Run-Around Loop (Loop)	$4.80	$1.30	$6.20	10%	15%
Chiller Energy Recovery (CWER)	$4.70	$1.90	$6.60	4%	6%
Direct Evap. Cooling (Evap)	$4.70	$2.20	$6.90	0%	0%
Water-side Economizer (Econ)	$4.60	$2.20	$6.80	1%	1%
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$4.70	$1.90	$6.60	4%	6%
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$5.00	$3.00	$8.00	-17%	-25%
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$4.70	$1.70	$6.40	7%	10%
Lab Plug Loads 8 W/sf(EPD8)	$4.00	$2.20	$6.20	10%	
Lab Plug Loads 4 W/sf(EPD4)	$3.20	$2.40	$5.60	18%	
Advanced w/Run-Around Loop (ALoop)	$4.00	$0.90	$4.90	29%	44%
Advanced w/Enthalpy Wheel (AWheel)	$3.80	$0.70	$4.50	35%	53%

Adding an enthalpy wheel to the VAV system reduces peak demand another 2 W/sf and electricity use by 1 kWh/sf. Electricity costs decrease by $0.10/sf from the VAV case. The enthalpy wheel reduces gas use by an additional 146 kBtu/sf, which equates to a savings of $1/sf in gas costs. The VAV system saves $1.20/sf, and adding an enthalpy wheel saves another $1/sf in annual energy costs.

Heat pipes and run-around loops increase electricity use by 4 kWh/sf and annual electricity costs by $0.10/sf because of the increase in fan energy compared with that of the constant-volume base case. Heat pipes reduce gas use by 131 kBtu/sf and gas costs by $0.80/sf. Run-around loops reduce gas use by 137 kBtu/sf and gas costs by $0.90/sf.

CHAPTER 3. COST ANALYSIS

Generally speaking, the cost-effectiveness of an energy efficiency measure determines whether or not it will be employed on a project. We estimated the cost-effectiveness of the various strategies using the following assumptions:

- 10-year life cycle
- 10% annual discount rate
- $1000/ton first-cost savings for cooling system reductions (e.g., chilled water system)
- $250/hp first-cost savings on heating system reductions (e.g., boiler)
- First-cost and operation and maintenance (O&M) costs for measures that were the same for each climate

Table 3.1 lists first costs and O&M costs for the energy efficiency strategies. The first costs are given in dollars per cubic foot per meter and converted to dollars per square foot for each location. The majority of the strategies relate to reducing ventilation requirements, so costs depend on the amount of air that is moved. The conversion between air flow and building area is done for each location because of the higher air flow rates in Denver.

The first cost for the VAV system is based on data from the E Source Commercial Space Cooling and Air Handling Technology Atlas (1997). The first cost for reducing static pressure drop can vary significantly, depending on the specifications for coils, filters, and other conditioning components, causing pressure drop in the air distribution system. Besant and Simonson (2000) give average costs for energy recovery ventilators and downsizing chilled water systems. The first cost of the boiler reduction strategy is estimated from Means Mechanical Cost Data (1999) and does not include the savings for downsizing pumps and piping. The upgrade cost for advanced cases equals the sum of the costs of the individual measures.

The O&M costs (ASHRAE Handbook of HVAC Systems and Equipment) are given in $/sf/yr. The O&M costs are less dependent on the total amount of air flow and will vary more with location and the availability of skilled labor.

Table 3.1 First-Cost and O&M Costs for Energy Efficiency Strategies

	First Cost ($/cfm)	Operation and Maintenance Cost ($/sf/yr)
Base Case		
Flow Setback (CFM21)	$1.00	$0
VAV	$1.50	$0.03
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$0.30	$0
Supply Static Pressure Drop of 3 in. w.g. (SP3)	$0.50	$0
Enthalpy Wheel (Wheel)	$2.50	$0.15
Enthalpy Wheel w/ VAV (VWheel)	$4.00	$0.17
Heat Pipe (HtPipe)	$2.50	$0.05
Run-Around Loop (Loop)	$2.00	$0.10
Chiller Energy Recovery (CWER)	$1.00	$0.05
Direct Evap. Cooling (Evap)	$1.30	$0.15
Water-side Economizer (Econ)	$1.00	$0.02
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$0	$0
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$0	$0
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$2.50	$0.20
Lab Plug Loads 8 W/sf(EPD8)	$0	$0
Lab Plug Loads 4 W/sf(EPD4)	$0	$0
Advanced w/Run-Around Loop (ALoop)	$3.80	$0.18
Advanced w/Enthalpy Wheel (AWheel)	$4.30	$0.20

The cost-effectiveness of the measures is climate-dependent. Tables 3.2, 3.3, 3.4, and 3.5 show the present value of the strategies in each climate. The present value is the value of the investment today assuming 10 years of energy cost savings and O&M costs. The present value includes the first cost of the strategy and the first-cost savings associated with downsizing the equipment. The last column in the tables gives the present value. A positive value represents a cost-effective strategy; a negative value represents a strategy that will increase the overall cost of a project.

In all climates, tighter humidity controls increase operating costs, and this strategy has a negative return on investment. In addition, chiller energy recovery, direct evaporative cooling, and water-side economizer have negative present values in Minneapolis and Seattle. Direct evaporative cooling is cost-effective in Denver, and chiller heat recovery is cost-effective in Atlanta. These strategies reduce space cooling energy use, which in the base case is only 4% of electricity use in Seattle, 6% in Denver, 10% in Minneapolis, and 17% in Atlanta. So, potential energy cost savings are relatively small. Associated first-cost savings are not significant enough to offset the first cost of the strategy.

The most cost-effective strategies are the variable-air-volume system (i.e., VAV fume hoods) and the enthalpy wheel. The present value of the VAV system is $5.40/sf in

Minneapolis, $1.20/sf in Denver, $2.50/sf in Seattle, and $4/sf in Atlanta. The present value for the VAV system in Denver is the lowest, because a higher first cost is associated with the higher air flow rate there, and the potential reduction in the boiler is the smallest.

The advanced case with VAV, an enthalpy wheel and a lower static pressure drop increases the present value of the VAV system by $17/sf in Minneapolis, $10.70/sf in Denver, $2/sf in Seattle, and $5/sf in Atlanta. The increase in present value is the smallest in Seattle, because Seattle has the lowest humidification and dehumidification loads.

Improving the VAV system by adding a run-around loop and reducing the static pressure drop across fans increases the present value of the VAV system by $9.90 in Minneapolis, $6.80 in Denver, $1.80 in Seattle, and $3.50 in Atlanta. The difference in performance between a run-around loop and an enthalpy wheel is small in Seattle because of the relatively small humidification and dehumidification loads.

We recognize that reducing plug loads does not actually result in energy cost savings. However, accurately assessing plug loads can reduce the first costs of the HVAC equipment as well as those of the power system serving the building.

Table 3.2 Cost-Effectiveness of Energy Efficiency Strategies in Minneapolis

	First Cost ($/sf)	Cooling Down-size (tons)	Cooling Savings ($/sf)	Heating Downsize (hp)	Heating Savings ($/sf)	Energy Cost Savings ($/yr)	Present Value ($/sf)
Base Case							
Flow Setback (CFM21)	$2.20	0	$0	0	$0	$0.30	($0.50)
VAV	$3.30	0	$0	200	$0.50	$1.40	$5.40
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$0.70	0	$0	0	$0	$0.20	$0.30
Supply Static Pressure Drop of 3 in. w.g. (SP3)	$1.10	0	$0	0	$0	$0.30	$0.80
Enthalpy Wheel (Wheel)	$5.60	250	$2.50	400	$1.00	$3.50	$18.30
Enthalpy Wheel w/ VAV (VWheel)	$8.90	250	$2.50	500	$1.30	$4.30	$20.40
Heat Pipe (HtPipe)	$5.60	100	$1.00	100	$0.30	$1.90	$7.10
Run-Around Loop (Loop)	$4.50	100	$1.00	100	$0.30	$2.00	$8.70
Chiller Energy Recovery (CWER)	$2.20	50	$0.50	100	$0.30	$0.20	($0.70)
Direct Evap. Cooling (Evap)	$2.90	0	$0	0	$0	($0)	($3.80)
Water-side Economizer (Econ)	$2.20	0	$0	0	$0	$0.10	($1.90)
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$0	0	$0	0	$0	$0.60	$3.60
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$0	0	$0	0	$0	($1.10)	($6.90)
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$5.60	200	$2.00	200	$0.50	$2.30	$10.00
Lab Plug Loads 8 W/sf(EPD8)	$0	50	$0.50	0	$0	$0.90	
Lab Plug Loads 4 W/sf(EPD4)	$0	100	$1.00	0	$0	$1.50	
Advanced w/Run-Around Loop (ALoop)	$8.50	300	$3.00	300	$0.80	$3.50	$15.50
Advanced w/Enthalpy Wheel (AWheel)	$9.60	400	$4.10	500	$1.30	$4.50	$22.40

Table 3.3 Cost-Effectiveness of Energy Efficiency Strategies in Denver

	First Cost ($/sf)	Cooling Down-size (tons)	Cooling Savings ($/sf)	Heating Downsize (hp)	Heating Savings ($/sf)	Energy Cost Savings ($/yr)	Present Value ($/sf)
Base Case							
Flow Setback (CFM21)	$2.70	0	$0	0	$0	$0.20	($1.50)
VAV	$4.00	0	$0	100	$0.30	$0.80	$1.20
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$0.80	0	$0	0	$0	$0.20	$0.20
Supply Static Pressure Drop of 3 in. w.g. (SP3)	$1.30	0	$0	0	$0	$0.30	$0.60
Enthalpy Wheel (Wheel)	$6.70	50	$0.50	600	$1.50	$2.60	$10.50
Enthalpy Wheel w/ VAV (Vwheel)	$10.80	50	$0.50	600	$1.50	$3.20	$9.80
Heat Pipe (HtPipe)	$6.70	50	$0.50	100	$0.30	$1.50	$2.70
Run-Around Loop (Loop)	$5.40	50	$0.50	100	$1.50	$1.40	$3.60
Chiller Energy Recovery (CWER)	$2.70	0	$0	100	$0.30	$0.30	($1.10)
Direct Evap. Cooling (Evap)	$3.50	200	$2.00	0	$0	$0.40	$0.30
Water-side Economizer (Econ)	$2.70	0	$0	0	$0	$0.10	($2.20)
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$0	0	$0	0	$0	$0.90	$5.80
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$0	0	$0	0	$0	($1.30)	($7.70)
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$6.70	50	$0.50	300	$0.80	$1.30	$1.30
Lab Plug Loads 8 W/sf(EPD8)	$0	100	$1.00	0	$0	$0.90	
Lab Plug Loads 4 W/sf(EPD4)	$0	150	$1.50	0	$0	$1.60	
Advanced w/Run-Around Loop (ALoop)	$10.30	200	$2.00	300	$0.70	$2.50	$7.00
Advanced w/Enthalpy Wheel (AWheel)	$11.60	200	$2.00	600	$1.50	$3.40	$11.90

Table 3.4 Cost-Effectiveness of Energy Efficiency Strategies in Seattle

	First Cost ($/sf)	Cooling Down-size (tons)	Cooling Savings ($/sf)	Heating Downsize (hp)	Heating Savings ($/sf)	Energy Cost Savings ($/yr)	Present Value ($/sf)
Base Case							
Flow Setback (CFM21)	$2.20	0	$0	0	$0	$0.10	($1.60)
VAV	$3.30	0	$0	200	$0.50	$0.90	$2.50
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$0.70	0	$0	0	$0	$0.20	$0.40
Supply Static Pressure Drop of 3 in. w.g. (SP3)	$1.10	0	$0	0	$0	$0.30	$0.90
Enthalpy Wheel (Wheel)	$5.50	200	$2.00	300	$0.80	$1.20	$3.40
Enthalpy Wheel w/ VAV (VWheel)	$8.90	200	$2.00	400	$1.00	$1.80	$4.20
Heat Pipe (HtPipe)	$5.50	100	$1.00	200	$0.50	$0.90	$1.00
Run-Around Loop (Loop)	$4.40	100	$1.00	200	$0.50	$0.90	$1.70
Chiller Energy Recovery (CWER)	$2.20	50	$0.50	200	$0.50	$0.10	($1.00)
Direct Evap. Cooling (Evap)	$2.90	200	$2.00	0	$0	$0.10	($1.20)
Water-side Economizer (Econ)	$2.20	0	$0	0	$0	$0	($2.10)
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$0	0	$0	0	$0	$0.20	$1.40
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$0	0	$0	0	$0	($0.80)	($5.00)
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$5.50	200	$2.00	300	$0.80	$0.50	($0.80)
Lab Plug Loads 8 W/sf(EPD8)	$0	100	$1.00	0	$0	$0.70	
Lab Plug Loads 4 W/sf(EPD4)	$0	150	$1.50	0	$0	$1.20	
Advanced w/Run-Around Loop (ALoop)	$8.40	200	$2.00	300	$0.80	$1.80	$4.30
Advanced w/Enthalpy Wheel (AWheel)	$9.50	200	$2.00	400	$1.00	$2.00	$4.50

Table 3.5 Cost-Effectiveness of Energy Efficiency Strategies in Atlanta

	First Cost ($/sf)	Cooling Down-size (tons)	Cooling Savings ($/sf)	Heating Downsize (hp)	Heating Savings ($/sf)	Energy Cost Savings ($/yr)	Present Value ($/sf)
Base Case							
Flow Setback (CFM21)	$2.20	0	$0	0	$0	$0.40	$0.10
VAV	$3.30	0	$0	100	$0.30	$1.20	$4.00
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$0.70	0	$0	0	$0	$0.20	$0.40
Supply Static Pressure Drop of 3 in. w.g. (SP3)	$1.10	0	$0	0	$0	$0.30	$0.90
Enthalpy Wheel (Wheel)	$5.60	350	$3.50	300	$0.80	$1.30	$5.60
Enthalpy Wheel w/ VAV (VWheel)	$8.90	350	$3.50	400	$1.00	$2.20	$8.10
Heat Pipe (HtPipe)	$5.60	200	$2.00	200	$0.50	$0.60	$0.70
Run-Around Loop (Loop)	$4.40	200	$2.00	200	$0.50	$0.70	$1.60
Chiller Energy Recovery (CWER)	$2.20	150	$1.50	100	$0.30	$0.30	$1.00
Direct Evap. Cooling (Evap)	$2.90	0	$0	0	$0	($0)	($3.90)
Water-side Economizer (Econ)	$2.20	0	$0	0	$0	$0.10	($2.00)
Humidity Controls: Max 60%RH, Min 20%RH (RH26)	$0	0	$0	0	$0	$0.30	$1.80
Humidity Controls: Max 50%RH, Min 40%RH (RH45)	$0	0	$0	0	$0	($1.10)	($7.10)
Humidity Controls: Max 50%RH, Min 40%RH w/ Enthalpy Wheel (RH45 Wheel)	$5.60	350	$3.50	300	$0.80	$0.50	$0.30
Lab Plug Loads 8 W/sf(EPD8)	$0	100	$1.00	0	$0	$0.70	
Lab Plug Loads 4 W/sf(EPD4)	$0	200	$2.00	0	$0	$1.20	
Advanced w/Run-Around Loop (ALoop)	$8.40	400	$4.10	300	$0.80	$2.00	$7.50
Advanced w/Enthalpy Wheel (AWheel)	$9.60	500	$5.10	400	$1.00	$2.40	$10.00

CHAPTER 4. CONCLUSIONS

The primary objective of this work was to assess the impact of energy efficiency strategies on energy use and costs in laboratory buildings. The strategies included reducing energy use for ventilation, cooling, and heating. We also looked at the impact of humidity controls and plug load assumptions on energy use. The DOE-2.2 building energy simulation program was used to evaluate the strategies, and limitations to the program have been noted.

Other strategies, such as reducing lighting loads and solar heat gain, were not addressed. The energy savings associated with those strategies may be significant, as they are in most office buildings; however, in laboratory buildings, those savings are overshadowed by savings obtained by using efficiency measures that impact ventilation and space heating. Furthermore, we did not include the impact of high-efficiency equipment such as chillers, boilers, fans, pumps, and motors. Such strategies—which include high-efficiency lighting and premium-efficiency equipment—should not be overlooked.

On average, office buildings use 100 kBtu/sf/yr, and laboratory buildings use 5 to 10 times as much energy as office buildings (Table 4.1). Their high ventilation loads and equipment plug loads result in high energy intensities. Because of the high amount of energy use, the economics of employing energy efficiency strategies are very attractive.

Table 4.1 Total Energy Use (kBtu/sf/yr)

	Minneapolis	Denver	Seattle	Atlanta
Base Case	1125	1043	673	652
VAV	940	938	556	514
Supply Static Pressure Drop of 4 in. w.g. (SP4)	1121	1038	668	647
Enthalpy Wheel	570	596	473	470
Run-Around Loop	765	779	510	527
Chiller Heat Recovery	1094	999	659	604
Direct Evap. Cooling (Evap)	1128	1006	670	654
Plug Loads 8 W/sf	1048	964	631	614
Advanced w/Run-Around Loop	602	655	409	390
Advanced w/Enthalpy Wheel	444	515	385	350

The base case model for the laboratory complies with the ASHRAE 90.10 energy efficiency standard. The base case building has a constant-volume air system with humidity controls set to a minimum RH of 30% and a maximum RH of 60%. The energy use associated with humidity control depends on the climate, but in general it is significant. Denver, with its dry climate, experiences the greatest increase in energy use and costs because of high humidification loads.

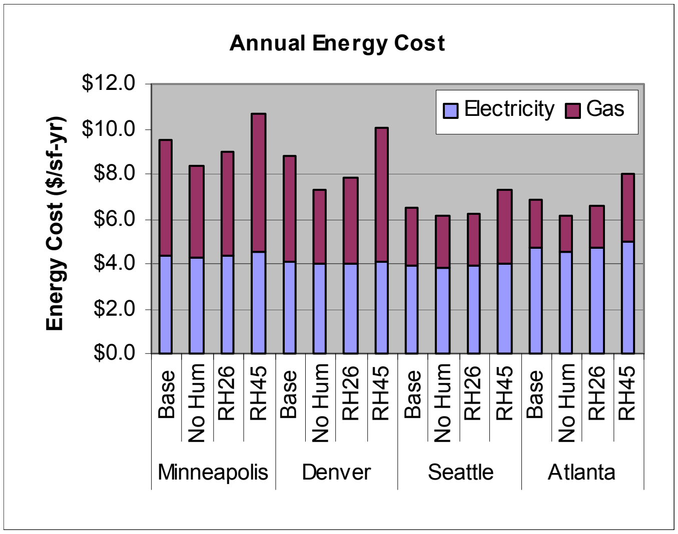


Figure 4.1 Annual energy costs for the base case, no humidity controls (No Hum), a minimum setting of 20% RH and a maximum of 60% (RH26), and a minimum setting of 40% and a maximum of 50% (RH45).

Regardless of climate, the results show that a variable-air-volume system (i.e., VAV fume hoods) reduces total energy costs by an average of $1/sf and has a positive present value in all climates (Table 4.2). In all climates except Denver's, a VAV system would earn 2 points under the LEED "Optimization of Energy Performance" credit.

Table 4.2 Summary of Results for VAV System

	Minneapolis	Denver	Seattle	Atlanta
Electricity Savings (kWh/sf/yr)	8	5	6	12
Gas Savings (kBtu/sf/yr)	160	89	98	100
Cost Savings ($/sf/yr)	$1.40	$0.80	$0.90	$1.20
First Cost ($/sf)	$3.30	$4.00	$3.30	$3.30
Present Value ($/sf)	$5.40	$1.20	$2.50	$4.00
Percent LEED Energy Savings (%)	19%	13%	22%	26%

Enthalpy wheels are the most efficient of the energy efficiency strategies considered in this study. Table 4.3 summarizes the results for the advanced case, which has a VAV system with an enthalpy wheel and reduced static pressure drop of 4 in. w.g. The savings are smallest in Seattle, where humidification and dehumidification loads are the smallest. Latent energy recovery with enthalpy wheels has a significant advantage over other energy recovery ventilators; however, run-around loops are often more practical because of the isolation of the supply and exhaust air streams from one another (Table 4.4). Savings associated with a heat pipe are comparable to those obtained with a run-around loop.

Table 4.3 Summary of Results for Advanced Case with Enthalpy Wheel

	Minneapolis	Denver	Seattle	Atlanta
Electricity Savings (kWh/sf/yr)	11	7	7	18
Gas Savings (kBtu/sf/yr)	647	507	265	246
Cost Savings ($/sf/yr)	$4.50	$3.40	$2.00	$2.40
First Cost ($/sf)	$9.60	$11.60	$9.50	$9.60
Present Value ($/sf)	$22.40	$11.90	$4.50	$10.00
Percent LEED Energy Savings (%)	63%	53%	48%	53%

Table 4.4 Summary of Results for Advanced Case with Run-Around loop

	Minneapolis	Denver	Seattle	Atlanta
Electricity Savings (kWh/sf/yr)	9	5	7	14
Gas Savings (kBtu/sf/yr)	494	371	243	218
Cost Savings ($/sf/yr)	$3.50	$2.50	$1.80	$2.00
First Cost ($/sf)	$8.50	$10.30	$8.40	$8.40
Present Value ($/sf)	$15.50	$7.00	$4.30	$7.50
Percent LEED Energy Savings (%)	48%	39%	44%	44%

4.1 Peak Electricity Demand

The peak demand is highest for the two most severe climates, those of Minneapolis and Atlanta. The reduction in peak demand resulting from the different strategies ranges from 0 W/sf to 4 W/sf, and the greatest reductions occur in Minneapolis and Atlanta with the advanced case and enthalpy wheel (Table 4.5). The peak-demand costs used in this study are relatively low at $7/kW; potential cost savings are much greater in locations with high demand charges.

Table 4.5 Peak Demand (W/sf)

	Minneapolis	Denver	Seattle	Atlanta
Base Case	20	16	15	20
VAV	19	15	15	18
Supply Static Pressure Drop of 4 in. w.g. (SP4)	20	15	15	20
Enthalpy Wheel	17	15	15	17
Run-Around Loop	21	17	16	20
Chiller Heat Recovery	20	16	15	20
Direct Evap. Cooling (Evap)	21	14	14	21
Plug Loads 8 W/sf	18	13	13	18
Advanced w/Run-Around Loop	18	15	15	17
Advanced w/Enthalpy Wheel	16	14	14	16

4.2 Electricity and Gas Use

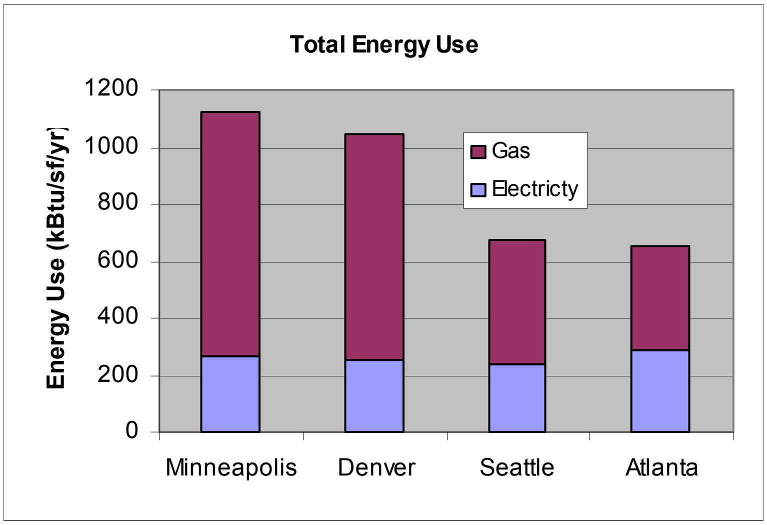


Figure 4.2 Electricity and gas use for the base case.

In terms of total energy use, gas use accounts for 76% of the total in Minneapolis, 75% in Denver, 64% in Seattle, and 55% in Atlanta (see Figure 4.2).

Gas usage makes up the dominant portion of total energy use. However, note that the value of each unit of electricity saved is nearly three times that of gas.

The electricity use predicted for the base model is fairly representative of that measured in existing laboratory buildings (see the Labs 21 case studies). The highest electricity use occurs in Atlanta; this reflects the high cooling and dehumidification loads (Table 4.6). Figure 4.3 shows electricity usage for each end use. Equipment plug loads account for more than 50% of total electricity use, followed by fans and space cooling.

Table 4.6 Electricity Use (kWh/sf/yr)

	Minneapolis	Denver	Seattle	Atlanta
Base Case	84	80	77	92
VAV	76	75	71	80
Supply Static Pressure Drop of 4 in. w.g. (SP4)	79	75	72	87
Enthalpy Wheel	86	84	80	90
Run-Around Loop	90	87	82	96
Chiller Heat Recovery	85	80	77	92
Direct Evap. Cooling (Evap)	84	76	76	92
Plug Loads 8 W/sf	70	65	63	77
Advanced w/Run-Around Loop	75	75	70	78
Advanced w/Enthalpy Wheel	73	73	70	74

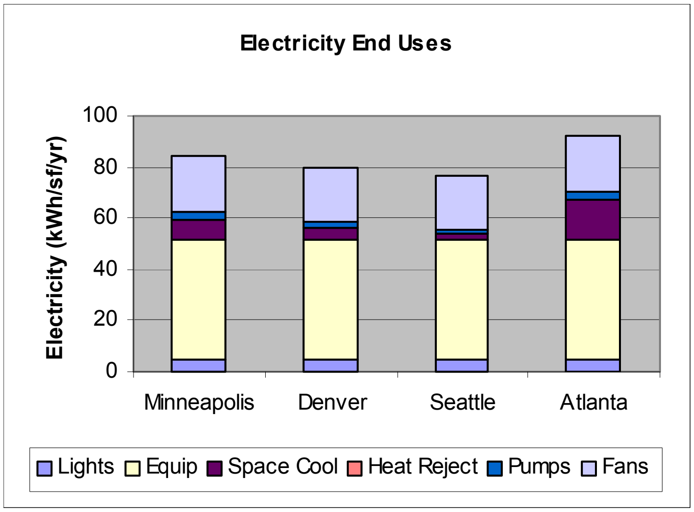


Figure 4.3 Electricity end uses for the base case.

Electricity savings resulting from using the energy efficiency strategies are 20% or less of total electricity use (Table 4.7). Energy recovery ventilation strategies actually increase electricity use because of the increase in the static pressure drop across the supply and exhaust fans. Electricity use for fans is 2-7 times greater than electricity use for space cooling in the base case. The reduction in cooling energy use associated with ERVs does not offset the increase in fan energy.

Table 4.7 Percent Electricity Savings (%)

	Minneapolis	Denver	Seattle	Atlanta
Base Case				
VAV	10%	7%	8%	13%
Supply Static Pressure Drop of 4 in. w.g. (SP4)	6%	7%	7%	6%
Enthalpy Wheel	-2%	-5%	-4%	2%
Run-Around Loop	-7%	-9%	-7%	-4%
Chiller Heat Recovery	0%	0%	0%	0%
Direct Evap. Cooling (Evap)	0%	4%	2%	0%
Plug Loads 8 W/sf	17%	18%	19%	16%
Advanced w/Run-Around Loop	11%	6%	9%	15%
Advanced w/Enthalpy Wheel	13%	8%	9%	19%

The temperature of the supply air to the building remains between 55°F and 60°F throughout the year, even in Minneapolis, because of high internal gains from equipment. However, because the supply air is 100% outside air, there is still a considerable heating load, especially in the colder climates. Space heating energy use also includes humidification and dehumidification loads. The simulations show that humidification loads in Minneapolis and Denver incur significant gas use (Figure 4.1). The gas savings obtained with a VAV system range from 11% in Denver to 28% in

Atlanta. The ERVs reduce heating energy use by more than 35% in all climates. The greatest amount of savings are achieved with an enthalpy wheel, which provides latent and sensible energy recovery.

Table 4.8 Gas Use (kBtu/sf/yr)

	Minneapolis	Denver	Seattle	Atlanta
Base Case	861	792	431	362
VAV	701	703	333	263
Supply Static Pressure Drop of 4 in. w.g. (SP4)	874	804	443	374
Enthalpy Wheel	300	332	221	187
Run-Around Loop	482	507	252	225
Chiller Heat Recovery	829	748	417	314
Direct Evap. Cooling (Evap)	865	766	432	365
Plug Loads 8 W/sf	829	758	434	371
Advanced w/Run-Around Loop	367	421	188	144
Advanced w/Enthalpy Wheel	214	284	166	116

Table 4.9 Percent Gas Savings (%)

	Minneapolis	Denver	Seattle	Atlanta
Base Case				
VAV	19%	11%	23%	28%
Supply Static Pressure Drop of 4 in. w.g. (SP4)	-1%	-2%	-3%	-3%
Enthalpy Wheel	65%	58%	49%	48%
Run-Around Loop	44%	36%	42%	38%
Chiller Heat Recovery	4%	6%	3%	13%
Direct Evap. Cooling (Evap)	0%	3%	0%	-1%
Plug Loads 8 W/sf	4%	4%	-1%	-2%
Advanced w/Run-Around Loop	57%	47%	56%	60%
Advanced w/Enthalpy Wheel	75%	64%	61%	68%

4.3 Energy Costs

Annual energy costs for the building are 5 to 10 times greater than those of a typical office building. The range in costs is climate-dependent because of the 100% outside air requirement associated with laboratories. Costs are given in $/sf of net building area. The costs can be converted to a cost-per-volume rate of air delivered, $/cfm, by dividing by 2.2 cfm/sf in Minneapolis, Seattle, and Atlanta. In Denver, the cost per square foot is divided by 2.7 cfm/sf. For the base case, the cost is $4.30/cfm in Minneapolis, $3.30/cfm in Denver, $2.90/cfm in Seattle, and $3.10/cfm in Atlanta.

The VAV system reduces energy costs by an average of $1/sf in all climates; savings associated with the ERV are more climate-dependent. Cost savings associated with an

ERV are greatest in Minneapolis and Denver, which have the highest heating loads. The advanced case with the enthalpy wheel saves 40% to 50% in annual energy costs.

Table 4.10 Annual Energy Cost ($/sf/yr)

	Minneapolis	Denver	Seattle	Atlanta
Base Case	$9.50	$8.80	$6.50	$6.90
VAV	$8.10	$8.00	$5.60	$5.70
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$9.40	$8.60	$6.30	$6.70
Enthalpy Wheel	$6.10	$6.20	$5.30	$5.60
Run-Around Loop	$7.50	$7.40	$5.60	$6.20
Chiller Heat Recovery	$9.30	$8.50	$6.40	$6.60
Direct Evap. Cooling (Evap)	$9.50	$8.40	$6.40	$6.90
Plug Loads 8 W/sf	$8.60	$7.90	$5.80	$6.20
Advanced w/Run-Around Loop	$6.10	$6.30	$4.70	$4.90
Advanced w/Enthalpy Wheel	$5.00	$5.40	$4.50	$4.50

Table 4.11 Annual Energy Cost Savings ($/sf/yr)

	Minneapolis	Denver	Seattle	Atlanta
Base Case				
VAV	$1.40	$0.80	$0.90	$1.20
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$0.20	$0.20	$0.20	$0.20
Enthalpy Wheel	$3.50	$2.60	$1.20	$1.30
Run-Around Loop	$2.00	$1.40	$0.90	$0.70
Chiller Heat Recovery	$0.20	$0.30	$0.10	$0.30
Direct Evap. Cooling (Evap)	$0	$0.40	$0.10	$0
Plug Loads 8 W/sf	$0.90	$0.90	$0.70	$0.70
Advanced w/Run-Around Loop	$3.50	$2.50	$1.80	$2.00
Advanced w/Enthalpy Wheel	$4.50	$3.40	$2.00	$2.40

The LEED ratings for Optimization of Energy Performance are given in Table 4.12. The percent energy cost savings calculation per the LEED system excludes the cost of the electricity used to operate equipment in the spaces (i.e., it does not include mechanical equipment). The percent savings associated with the VAV system are 13% in Denver, and about 20% or more for the other three cities. The enthalpy wheel alone saves more than 40% in Minneapolis and Denver and 28% in Seattle and Atlanta. The advanced cases achieve savings of 40% or more, and they qualify for 6 points or more under the LEED rating system. The percent savings associated with the direct evaporative stage in Denver are significant at 7%. LEED does not allow for the reduction of plug loads under this credit, so no percent savings are associated with that measure.

Table 4.12 LEED Energy Cost Savings (%)

	Minneapolis	Denver	Seattle	Atlanta
Base Case	0%	0%	0%	0%
VAV	19%	13%	22%	26%
Supply Static Pressure of 4 in. w.g. (SP4)	2%	3%	4%	4%
Enthalpy Wheel	48%	41%	28%	28%
Run-Around Loop	28%	22%	21%	15%
Chiller Heat Recovery	2%	4%	2%	6%
Direct Evap. Cooling (Evap)	0%	7%	2%	0%
Plug Loads 8 W/sf				
Advanced w/Run-Around Loop	48%	39%	44%	44%
Advanced w/Enthalpy Wheel	63%	53%	48%	53%

4.4 Downsizing HVAC Equipment

Using energy recovery strategies and a lower design plug load assumption provides an opportunity to downsize the chilled water and hot water plants. The VAV system also requires a smaller heating plant, because the required air flow under design conditions in winter is lower than under design conditions in summer. This assumes that internal gains drive air flow requirements under design conditions in summer rather than laboratory requirements for fresh air.

Table 4.13 lists potential chiller savings for the various efficiency strategies. The associated cost savings are $1000/ton. This includes the money saved by downsizing the entire chilled water system. As expected, savings are greatest in climates with the highest cooling loads.

Table 4.13 Chiller Savings (tons)

	Minneapolis	Denver	Seattle	Atlanta
Base Case				
VAV	0	0	0	0
Supply Static Pressure of 4 in. w.g. (SP4)	0	0	0	0
Enthalpy Wheel	250	50	200	350
Run-Around Loop	100	50	100	200
Chiller Heat Recovery	50	0	50	150
Direct Evap. Cooling (Evap)	0	200	200	0
Plug Loads 8 W/sf	50	100	100	100
Advanced w/Run-Around Loop	300	200	200	400
Advanced w/Enthalpy Wheel	400	200	200	500

For water heating, cost savings of $250/hp are assumed. This does not include the money saved by downsizing the entire chilled water system. The potential reduction in boiler size with the enthalpy wheel saves almost $1/sf in Minneapolis. Adding chiller savings of $2.50/sf results in first-cost savings of $3.50/sf in Minneapolis. The first cost of the enthalpy wheel is $5.60/sf ($2.50/cfm), so cost savings from downsizing the equipment offset 60% of the cost of the wheel.

Table 4.14 Boiler Savings (hp)

	Minneapolis	Denver	Seattle	Atlanta
Base Case				
VAV	200	100	200	100
Supply Static Pressure of 4 in. w.g. (SP4)	0	0	0	0
Enthalpy Wheel	400	600	300	300
Run-Around Loop	100	100	200	200
Chiller Heat Recovery	100	100	200	100
Direct Evap. Cooling (Evap)	0	0	0	0
Plug Loads 8 W/sf	0	0	0	0
Advanced w/Run-Around Loop	300	300	300	300
Advanced w/Enthalpy Wheel	500	600	400	400

Design assumptions for plug loads may also drive the design air flow rate. Accurately assessing power requirements for laboratory and office equipment will allow optimal sizing of the chilled water and air distribution systems. The first-cost implications are considerable, as are the implications for part-load operation of chillers and boilers.

4.5 Economics

A life-cycle-cost analysis shows that the VAV system, the reduction in static pressure drop, and the energy recovery ventilation strategies are all cost-effective (Table 4.15). The analysis assumes a 10-year life and a 10% discount rate.

The VAV system has the highest present value in Minneapolis and Atlanta, which have the highest cooling loads and are the most humid. The present value of the VAV system in Denver is lower than that of the other climates because higher first costs are associated with the higher air flow rates and lower energy cost savings.

Energy recovery ventilation strategies have greatest value in the heating-dominated climates of Minneapolis and Denver. In general, these strategies have the lowest present value in Seattle, where potential energy cost savings are lowest.

Table 4.15 Present Value ($/sf)

	Minneapolis	Denver	Seattle	Atlanta
Base Case	$0	$0	$0	$0
VAV	$5.40	$1.20	$2.50	$4.0
Supply Static Pressure Drop of 4 in. w.g. (SP4)	$0.30	$0.20	$0.40	$0.40
Enthalpy Wheel	$18.30	$10.50	$3.40	$5.60
Run-Around Loop	$8.70	$3.60	$1.70	$1.60
Chiller Heat Recovery	-$0.70	-$1.10	-$1.00	$1.00
Direct Evap. Cooling (Evap)	-$3.80	$0.30	-$1.20	-$3.90
Plug Loads 8 W/sf				
Advanced w/Run-Around Loop	$15.50	$7.00	$4.30	$7.50
Advanced w/Enthalpy Wheel	$22.40	$11.90	$4.50	$10.00

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