Optimal Air, Energy Recovery, Geothermal

3 Simple Approaches To Energy Efficiency

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Many building owners are choosing sustainable design because the economics make more sense now. For example, rising energy costs are reducing the payback period for capital improvements that improve energy performance. However, in contrast to the wealth of promotional information for why sustainable building practices should be used, surprisingly little application information is available on sustainable design.

This article focuses on one aspect of sustainable design—changing traditional design parameters to improve the energy performance of an HVAC system. It provides several examples of design parameters to improve HVAC system efficiency with solid economic payback, or in some cases, immediate payback.

To investigate the amount of savings that can occur from making changes to an HVAC system, one must start with a benchmark system to provide a performance baseline against which the alternative system can be measured. For the examples provided in this article, ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, Appendix G, Performance Rating Method, was used to provide the benchmark HVAC system based on building size. These benchmarks are summarized in *Table 1* for nonresidential spaces.

The energy rates[1] used in all three analyses are summarized in *Table 2*. To perform the energy analysis, a private energy analysis tool using reduced weather data gives a ballpark idea of energy savings. Appendix G details specific energy analysis program requirements including 8,760 hour weather data. Appendix G also is being used as the benchmark for the Energy and Atmosphere Credit 1 in Leadership in Energy and Environmental Design® (LEED)-NC v2.2.

Example 1: Optimal Air Systems

To understand the synergies in the system, *Figure 1* shows average annual energy consumption of a building in Chicago using VAV with reheat with water-cooled chillers. The pie chart will change depending on the location of the building and the HVAC system used. Interestingly, fan energy is the greatest energy consumer in the mechanical system because the fans are used the entire

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time the building is occupied. From this you could draw a conclusion that money spent for high-efficiency chillers might be better spent on reducing fan energy expenses to generate greater returns.

Pursuing these airside savings can challenge the most sacred HVAC design principle—supplying 55°F (13°C) air to the space. Cooling supply air to 55°F (13°C) generally provides the required humidity ratio to maintain space conditions at 75°F (24°C) and 50% relative humidity. However, cooling supply air below 55°F (13°C) offers the potential for significant capital or energy cost savings in many applications. Chapter 26 of the *2005 ASHRAE Handbook—Fundamentals* addresses insulation to avoid condensation based on relative humidity levels in the space.

As the supply air temperature is reduced, the supply air volume decreases proportionally. That is, a 10% increase in supply air ΔT (space setpoint minus the supply air temperature) will result in a 10% drop in required supply air volume. Any resulting energy savings from the lower airflow can be taken as a reduction in annual energy consumed or as capital cost savings:

- Sizing the air handler and ductwork as if the system is providing 55°F (13°C) supply air results in energy savings during the life of the equipment. As the ΔT of the system increases, the amount of airflow required to provide the same cooling effect is reduced. This lowers the static pressure and allows smaller fan motors to be installed. The decreased fan size also reduces fan noise, resulting in a quieter system. This approach is ideal for existing buildings where the cost to remove and replace existing ductwork may be prohibitive.
- Reducing the duct, air handler and fan motor sizes to match the lower supply air volume can result in capital cost savings and, in the case of indoor air-handling units, provide more leasable space. This approach is ideal for new buildings where there are no preexisting conditions affecting the design (i.e., ductwork).

The practice of reducing supply air temperature is common in grocery stores to improve humidity control, and it is gaining popularity in comfort cooling applications. The key design parameter is to identify the optimal air or balance point. This is the lowest supply air temperature that can be used without increasing the annual operating cost of the building. While it is typically 48°F to 52°F (9°C to 11°C), every building is different and annual energy analysis is required to determine this point. The amount of time and effort on finding the balance point depends on the complexity of the project.

The penalty for achieving these reductions is that the colder supply air temperature requires more refrigeration work and reduces the number of hours in a year where economizer operation can be used. For example, lowering the supply air setpoint from 55°F to 50°F (13°C to 10°C) removes the opportunity to cool the building with outdoor air when the ambient dry bulb is between 55°F and 50°F (13°C and 10°C). In Chicago, 601 hours of the year are between 50°F and 55°F (10°C and 13°C), or roughly 7% of the hours when a HVAC system may be operating. With integrated economizers, some cooling effect can be gained, but supplemental mechanical cooling is required.

In addition, the primary system must be capable of providing the low supply air temperature. This is not an issue with chiller and air-handling systems, but it does require that packaged rooftop and other unitary systems offer some flexibility in DX coil and refrigeration component selections to avoid oversizing the unit to provide the lower supply air temperature.

Table 3 shows the effect of lowering the supply air temperature to 50°F (10°C) for a one-story, 100,000 ft^2 (9290 m^2) retail building. To illustrate the effects of changing the design supply temperature from the traditional 55°F to 50°F (13°C to 10°C), a VAV packaged rooftop system that meets the minimum requirements of Standard 90.1-2004 was selected as the baseline system from Appendix G (Table 1). In this comparison, the duct size was maintained to keep capital costs flat and investigate energy savings of the lower airflow rate and reduced static pressure. Three locations throughout the U.S. were chosen to illustrate the results in different climates. The 50°F (10°C) supply air temperature was selected for illustration purposes only and may or may not represent the optimal air or balance point for each application.

Note that the building energy cost savings ranged from 4.8% to 6.1%. These are significant energy cost savings with little or no impact on design work or capital costs. With a little more effort, the system could be further optimized to deliver even greater energy cost savings:

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- The lower supply air temperature also lowers the relative humidity in the space. This should allow the room setpoint to be raised while still maintaining acceptable conditions as detailed in ANSI/ASHRAE Standard 55-2004, *Thermal Environmental Conditions for Human Occupancy.*
- The airside can be specified with a blow-through (fan before coil) vs. a draw-through (fan after coil) configuration to eliminate the additional airflow required to compensate for fan heat in the airstream (typically a 2°F to 3°F [1°C to 1.5°C] temperature rise).
- Specifying more efficient fans and compressors may raise capital costs, but the payback could be very favorable.

Each of these strategies can be justified or cast aside using the same energy analysis shown in this example.

Example 2: Recovering Energy from Condenser Water

A huge amount of heat energy is sent to the atmosphere through the cooling towers. In addition to every Btu collected in the building, approximately 25% more heat energy from compressor and accessory work is sent to the tower. Typically, chillers send 95°F (35°C) water to the cooling tower. By raising this design parameter to between 105°F (40°C) and 140°F (60°C) through one of two condenser heat recovery options, the increased chiller lift penalizes the compressor efficiency of the cooling system but benefits the heating system. Overall, the total HVAC system may perform more economically and show a solid economic payback for the additional capital cost.[2]

Condenser heat recovery can be used for building reheat or to preheat domestic hot water. Standard 90.1-2004 has minimum requirements for heat recovery:

6.5.6.2 Heat Recovery for Service Water Heating*—Condenser heat recovery systems shall be installed for heating or preheating of service hot water provided all of the following are true:*

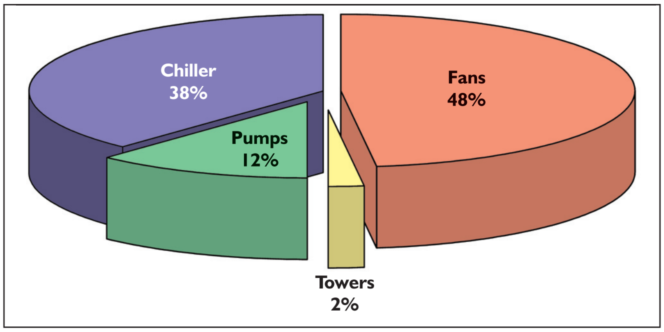


Figure 1: Average office building annual energy consumption.

a. The facility operates 24 hours day.

b. The total installed heat rejection capacity of the water-cooled system exceeds 6,000,000 Btu/h (1700 kW) of heat of rejection.

c. The design service water heating load exceeds 1,000,000 Btu/h (300 kW).

The required heat recovery system shall have the capacity to provide the smaller of:

a. 60% of the peak heat rejection load at design conditions, or

b. Preheat of the peak service hot water draw to 85°F (29°C).

Exceptions to 6.5.6.2

a. Facilities that employ condenser heat recovery for space heating with a heat recovery design exceeding 30% of the peak water-cooled condenser load at design conditions.

b. Facilities that provide 60% of their service water heating from site solar or site recovered energy, or from other sources.

In addition, Standard 90.1 exempts the requirement for economizers, 6.5.1 (d) if the HVAC system includes a condenser heat recovery system that is required by 6.5.6.2. Standard 90.1 also allows simultaneous heating and cooling for comfort or dehumidification purposes if 75% of the reheat energy is site-recovered, such as condenser heat recovery

Building Type*		Fossil Fuel & Purchased Heat	Electric And Other
Number of Floors	Area		
3 Or Less	<75,000 ft^2	Packaged Rooftop Constant Volume	Packaged Rooftop Heat Pump Constant Volume
4 or 5	<75,000 ft^2	Packaged Rooftop VAV with Reheat	Packaged Rooftop VAV With Parallel Fan Powered Boxes
5 or Less	75,000 ft^2 to 150,000 ft^2		
More Than 5	Any	VAV With Reheat & Chillers	VAV With Parallel Fan Powered Boxes And Chillers
* Nonresidential buildings only are listed here.			

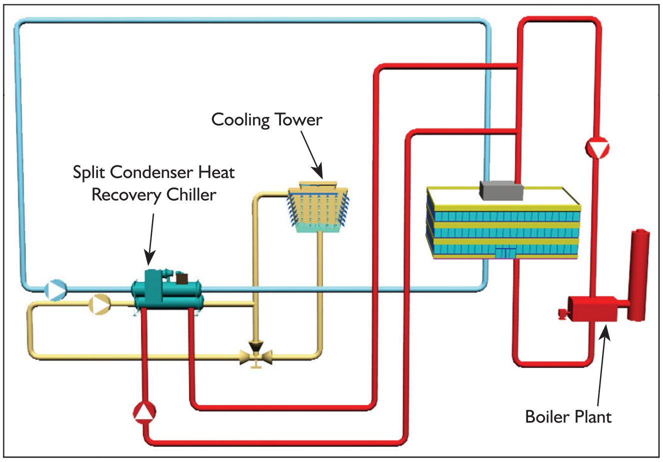
Table 1: Baseline system from Standard 90.1-2004's Appendix G.

Location	Electricity ($/kWh)	Natural Gas $/Therm	Natural Gas $/GJ
Chicago	$0.0754	$1.11	$10.52
Miami	$0.0761	$1.33	$12.61
Philadelphia	$0.0851	$1.20	$11.37
Minneapolis	$0.0631	$1.01	$9.57
Helena	$0.0742	$1.04	$9.86
Denver	$0.0689	$0.92	$8.72

Table 2: Average energy prices used in comparisons.

	Baseline Energy Cost	Performance System Energy Cost	Percent Energy Savings
Chicago	$307,983	$289,259	6.1%
Miami	$228,482	$217,394	4.9%
Philadelphia	$261,108	$248,544	4.8%

Table 3 Energy cost comparison: 50°F and 55°F SAT.



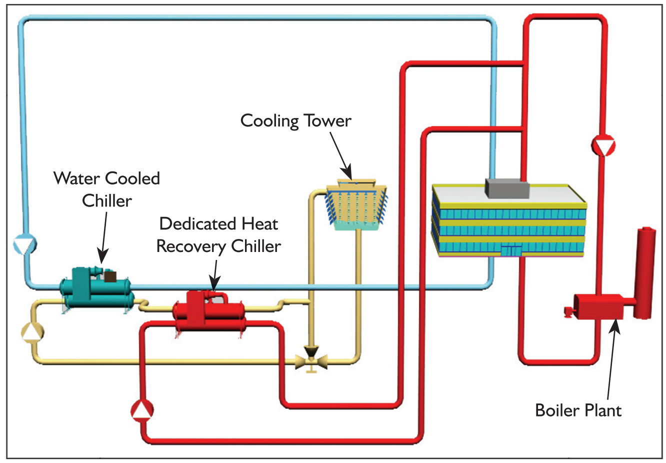


Figure 2 (left): Split condenser heat recovery chiller system. Figure 3 (right): Cascaded system with dedicated heat recovery chiller.

systems. The two main building types likely to meet the condenser recovery requirement are health-care and hospitality facilities. Condenser heat recovery can be performed using a heat recovery chiller, or a cascaded system with a dedicated heat recovery chiller.

A split condenser heat recovery chiller, shown in *Figure 2*, changes the typical design parameters of chiller operation on the condenser side from 85°F/95°F (39°C/35°C) to 95°F/105°F (35°C/40°C). This increases the chiller lift during heat recovery mode and results in a penalty on chiller efficiency. The chiller either operates in cooling tower mode, rejecting condenser heat outdoors, or in heat recovery mode, where the condenser produces up to 105°F (40°C) water for building use. This system is limited by the lift capabilities of the chiller. At 105°F (40°C), reheat coils may have to be resized, increasing capital costs and pressure drop compared to a coil sized for a design reheat temperature of 140°F (60°C).

Cascading the system and having a dedicated heat recovery chiller, as shown in *Figure 3,* allows the source chiller to run at typical comfort cooling conditions. Condenser water is sent to a cooling tower, or through the evaporator of the dedicated heat recovery chiller. The dedicated heat recovery chiller has similar lift requirements as a chiller at standard conditions, so it is able to generate up to 140°F (60°C) water for building use. An advantage of this system is that the dedicated heat recovery chiller can be bypassed in non-heat recovery mode. A disadvantage is the added capital cost of another chiller.

Recovered Energy for Building Heat

Constant volume with reheat and four-pipe fan coil systems are good candidates for condenser energy recovery because they share a common trait of simultaneous heating and cooling. VAV systems may be a good candidate, depending upon the minimum turndown on the VAV boxes. If the minimum turndown is 50% or more, as is the case with many health-care applications, it may make sense.

Figure 4 shows a typical annual heating and cooling load profile for a constant volume with reheat system commonly used in health-care facilities. This figure clearly demonstrates the Golden Rule of energy recovery "You must have an energy source at the same time you have an energy need!"

The ideal size for the energy recovery equipment is the point where the instantaneous heat source meets the instantaneous heating requirement. This is where the heating and cooling lines cross in *Figure 4.* Energy can be recovered in the shaded area under both curves.

The only way to know for certain if condenser energy recovery for heating makes sense is to run an annual energy analysis. For this example, a three-story, 480,000 ft^2 (4400 m^2) acute care hospital with a 1,600 ton (455 kW) chiller plant and a 20,000 kBtu/h (2000 MJ/h) boiler plant is considered. The HVAC system is a combination of VAV and constant volume with reheat and chillers providing 400,000 cfm (190 000 L/s) of supply air and 192,000 cfm (90 600 L/s) of ventilation air.

The baseline system is comprised of two 800 ton (225 kW) centrifugal chillers. The first energy recovery option includes two 800 ton (225 kW) centrifugal chillers and a 460 ton (130 kW) dedicated heat recovery chiller (4.4 COP). The dedicated heat recovery chiller is designed to provide 140°F (60°C) hot water. Additional design requirements for this system include increasing the source chiller head by 20 ft (6 m) to accommodate the added pressure drop of the energy recovery chiller. A hot water tertiary pump has also been included.

The second heat recovery option includes a 1,200 ton (340 kW) centrifugal chiller and a 375 ton (100 kW) split condenser heat recovery chiller (0.73 kW/ton, 4.80 COP) optimized for the heat recovery load and supplying 105°F (40°C) hot water. A design consideration for this system includes adding deeper heating coils to account for the lower grade heat from the split condenser heat recovery chiller. Static pressure has been increased by 0.20 in. (50 Pa) to account for the deeper coils. In addition, a hot water tertiary pump has been added.

As shown in *Table 4,* both energy recovery options saved between 8% to 10.7% on building energy. Further analysis could be done to achieve more energy savings such as using variable flow, changing condenser water temperature, changing chilled water temperature range, or changing other chiller system design parameters.

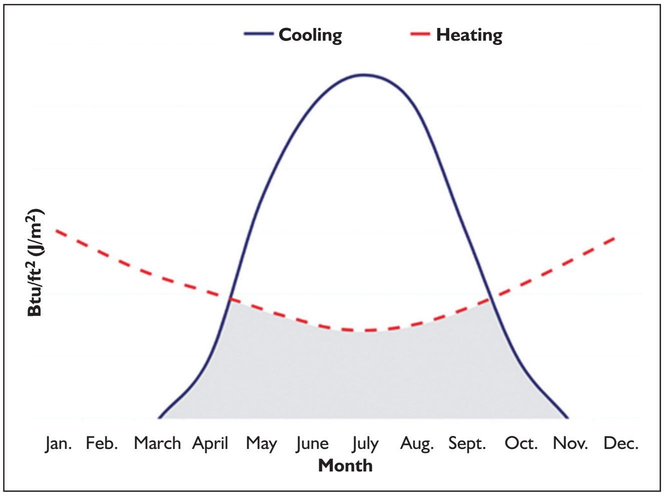


Figure 4: Optimum heating and cooling load profile for condenser heat recovery systems.

Results	Baseline	Dedicated Heat Recovery	Split Condenser Heat Recovery Chiller
Chicago	$1,630,568	$1,494,932	$1,499,952
Miami	$1,853,206	$1,654,514	$1,657,945
Philadelphia	$1,752,934	$1,601,086	$1,603,705
Percent Energy Savings	**Baseline**	**Dedicated Heat Recovery**	**Split Condenser Heat Recovery Chiller**
Chicago	N/A	8.3%	8.0%
Miami	N/A	10.7%	10.5%
Philadelphia	N/A	8.7%	8.5%

Table 4: Building energy comparison: base chiller system vs. energy recovery options.

Location	Baseline Energy Cost	Geothermal Energy Cost	Percent Savings
Philadelphia	$137,503	$118,429	13.9%
Minneapolis	$103,151	$90,260	12.5%
Helena	$112,064	$103,842	7.3%
Denver	$101,320	$95,737	5.5%
Chicago	$117,617	$106,021	9.9%

Table 5: Comparison of baseline VAV rooftop system and geothermal WSHP systems in a five-story 100,000 ft² (9290 m²) office building.

Example 3: Geothermal Heat Pump Systems

The final example relies on selecting an HVAC system that is more efficient to begin with and optimizing it to achieve capital cost savings and additional energy savings. Geothermal water source heat pump systems are some of the most energy efficient systems for applications such as offices, schools, medical facilities, dormitories, condominiums, and recreational facilities.

For this example, the baseline building is a five-story, 100,000 ft^2 (9290 m^2) office building with standard office building hours. Referring back to Appendix G (Table 1), the baseline HVAC system is a packaged VAV rooftop system that meets the minimum requirements of Standard 90.1-2004. With the exception of the HVAC system change, all other design parameters such as lighting and building orientation were kept the same to provide an equal comparison of the systems. The results for five different cities are shown in *Table 5*. Energy cost savings range from 5.5% to 13.9%.

What About Costs?

The high efficiency of a geothermal system often is perceived to come at a cost premium, primarily because the costs associated with the ground loop are perceived to be high. As with many new technologies or systems, the cost premium can depend on the experience of the local market.

Table 6 shows the calculated simple payback for the geothermal system, which ranges from 1.3 to 13.3 years depending on the utility and maintenance cost savings, and the capital cost premium. Some assumptions are made in this calculation. First, it is assumed that the geothermal systems cost less to maintain than standard VAV rooftop systems. The maintenance cost savings vs. the rooftop VAV system was given a range from \$0.02/ft^2 to \$0.06/ft^2 (\$0.20/m^2 to \$0.60/m^2).

Second, the capital cost premium of the geothermal system was given a range from \$0.50/ft^2 (\$5/m^2) to \$1.50/ft^2 (15/m^2). It is often difficult to develop accurate information on the costs of different system types compared to equipment types. For this reason, a range was given for both maintenance savings and capital cost premium. As you can see from *Table 6*, the cost premium for a geothermal system is easily overcome by cumulative energy savings in some climates and applications. Note that simple payback only looks at first year cost savings. Rising costs for natural gas and electricity in many areas of the United States can reduce the payback.

Optimizing Geothermal Systems

Three design parameters were taken into consideration to optimize the geothermal system: the effect of raising the loop temperature on operating and capital costs, using variable frequency drives (VFDs) on the pumps and using energy recovery ventilators (ERV) for the makeup air.

Loop Temperature vs. Operating Costs

The loop temperature of a geothermal system affects its efficiency and capital cost. A smaller geothermal loop will run at higher temperatures, decreasing the cooling efficiency and increasing the heating efficiency of the water source heat pumps. However, a smaller loop results in significant capital cost savings. *Table 7* compares the energy use and capital cost of geothermal systems with loops designed for water entering the heat pumps at cooling design conditions of 95°F (35°C) and 85°F (29°C). The cost of the loop is assumed to be $10 per linear foot ($30 per linear meter). From this chart you can see that the energy savings for the larger loop (85°F/29°C) is very small and does not justify the additional capital cost. Alternatively, one could say that the insignificant energy penalty associated with the smaller loop (95°F/35°C) justifies the capital cost savings.

Adding Variable Frequency Drives (VFDs) to the Pumps

The pumps serving geothermal systems are small, but they provide constant flow and run continuously. These small pumps can use a significant amount of energy over the course of a year. Adding VFDs to the system to provide variable flow can reduce this energy consumption considerably. In recent years, VFD costs have been reduced, so that the payback (in energy cost savings) is worth the capital cost premium for installing a VFD. Sometimes the cost of a VFD is almost the same as the cost of a pump starter. *Table 8* shows the payback for using a VFD in this geothermal system, considering three different capital cost premiums for VFDs. The payback ranges from less than one year to just over a year depending on the location and energy cost savings.

Energy Recovery of Ventilation Air

In many parts of the United States, makeup air must be conditioned before it enters the building. Water source heat pumps generally are not suitable for handling ventilation loads because the compressors cycle on and off. In the off cycle, dehumidification does not occur and humid air can enter the space. A standard makeup air unit and an energy recovery ventilation (ERV) unit were modeled to supply ventilation air for the geothermal system. Using a range from \$0.50/cfm (\$1 per L/s) to \$1.50/cfm (\$3 per L/s) capital cost premium for an ERV system, *Table 9* shows some of the climates that are most favorable for an ERV system. In general, climates that can recover both heating and cooling energy achieve the greatest energy savings.

Geothermal water source heat pump systems are ideal for achieving high efficiency that pays back year after year in energy cost savings for building owners. While the installed cost of the system can be higher than more conventional systems, the payback is often favorable for achieving lower life-cycle costs.

Conclusion

The examples presented in this article can be implemented today in existing buildings, as in Example 1, or new construction designs, as in Examples 2 and 3, to improve the efficiency of HVAC systems and reduce building energy costs with solid economic payback.

Although whole building energy costs were modeled, only the energy cost differences of changing the HVAC systems were considered. The effects of changing the lighting, envelope, and miscellaneous electric loads in the building were not taken into account in order to focus on the HVAC system. All of these factors should be considered in the scope of high-performance building design.

References

1. Department of Energy Information Administration, www.eia.doe.gov.

2. *2004 ASHRAE Handbook—HVAC Systems and Equipment,* Chapter 8, Applied Heat Pump and Heat Recovery Systems.

Location	Annual Utility Cost Savings	Maintenance Cost Premium (\$/ft², \$/m²)	Capital Cost Premium (\$/ft², \$/m²) Payback (Years) \$0.50 (\$5)	\$1.00 (\$10)	\$1.50 (\$15)
Philadelphia	\$19,074	\$0.02 (\$0.20)	1.5	3.0	4.4
		\$0.04 (\$0.40)	1.4	2.8	4.2
		\$0.06 (\$0.60)	1.3	2.6	4.0
Minneapolis	\$12,891	\$0.02 (\$0.20)	2.1	4.3	6.4
		\$0.04 (\$0.40)	2.0	3.9	5.9
		\$0.06 (\$0.60)	1.8	3.6	5.5
Helena	\$8,222	\$0.02 (\$0.20)	3.2	6.4	9.6
		\$0.04 (\$0.40)	2.8	5.6	8.5
		\$0.06 (\$0.60)	2.5	5.1	7.6
Denver	\$5,583	\$0.02 (\$0.20)	4.4	8.8	13.3
		\$0.04 (\$0.40)	3.8	7.5	11.3
		\$0.06 (\$0.60)	3.3	6.5	9.8
Chicago	\$11,596	\$0.02 (\$0.20)	2.3	4.7	7.0
		\$0.04 (\$0.40)	2.1	4.3	6.4
		\$0.06 (\$0.60)	2.0	3.9	5.9

Table 6: Operating, maintenance and capital cost comparison and payback.

Location	Energy Cost 95°F (35°C) Loop	Energy Cost 85°F (29°C) Loop	Capital Cost Difference	Energy Savings	Payback (Years)
Philadelphia	\$118,429	\$117,249	\$116,800	\$1,180	99
Minneapolis	\$90,260	\$89,492	\$75,800	\$768	99
Helena	\$103,842	\$102,978	\$66,100	\$864	77
Denver	\$95,737	\$94,754	\$92,400	\$983	94
Chicago	\$106,021	\$105,031	\$95,400	\$990	96

Table 7: Energy and cost comparison of 95°F and 85°F loops.

Location	Utility Cost Savings	Capital Cost Premium Payback (Years) \$6,000	\$8,000	\$10,000
Philadelphia	\$12,406	0.48	0.64	0.81
Minneapolis	\$8,581	0.70	0.93	1.17
Helena	\$9,868	0.61	0.81	1.01
Denver	\$9,597	0.63	0.83	1.04
Chicago	\$10,612	0.57	0.75	0.94

Table 8: Constant flow vs. variable flow using VFDs.

Location	Utility Cost Savings	Capital Cost Premium (\$/cfm, \$/L/s) Payback (Years) \$0.50 (\$1.00)	\$1.00 (\$2.00)	\$1.50 (\$3.00)
Philadelphia	\$4,869	1.85	3.70	5.55
Minneapolis	\$7,274	1.24	2.47	3.71
Helena	\$5,615	1.60	3.21	4.81
Denver	\$2,768	3.25	6.50	9.75
Chicago	\$5,919	1.52	3.04	4.56

Table 9: Payback results using energy recovery ventilation units.