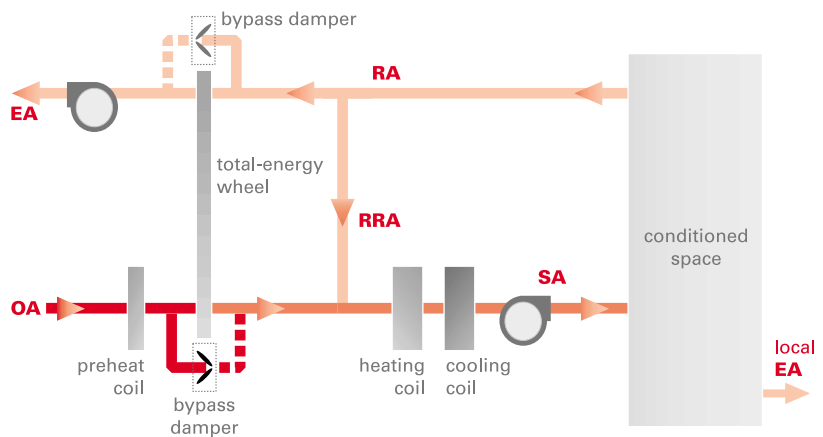


Applications Engineering Manual

Air-to-Air Energy Recovery in HVAC Systems





Air-to-Air Energy Recovery in HVAC Systems

**John Murphy, senior applications engineer
Brenda Bradley, information designer**



Preface

As a leading HVAC manufacturer, we believe that it is our responsibility to serve the building industry by regularly disseminating information gathered through laboratory research, testing programs, and practical experience. Trane publishes a variety of educational materials for this purpose. Applications engineering manuals such as this document can serve as comprehensive reference guides for professionals who design building comfort systems.

This manual focuses on **air-to-air energy recovery**. It describes concepts and mechanical implementation, and identifies system-level characteristics for effective operation and control. We encourage you to familiarize yourself with its contents and to review the appropriate sections when designing a comfort-system application that could benefit from airside energy recovery. ■

Trane, in proposing these system design and application concepts, assumes no responsibility for the performance or desirability of any resulting system design. Design of the HVAC system is the prerogative and responsibility of the engineering professional.

TRACE, Trane, and the Trane logo are registered trademarks, and System Analyzer and Traq are trademarks, of Trane, which is a division of American Standard Inc.

Contents

Introduction	1
Why Recover Energy?	2
Economics	2
Compliance with Codes	2
Applications for Air-to-Air Energy Recovery	5
Basic Concepts	5
Sensible versus Total Energy Recovery	5
Effectiveness Ratio	7
Balanced versus Unbalanced Airflows	7
Outdoor-Air (OA) Preconditioning	9
Sensible versus Total Energy Recovery	9
Effect on First Cost	13
Effect on Operating Costs	16
Other Considerations	18
Supply-Air (SA) Tempering	20
Supply-Air Tempering in Mixed-Air Systems	21
Supply-Air Tempering in Dedicated Outdoor-Air Systems	26
Effect on First Cost	32
Effect on Operating Costs	33
Other Considerations	37
Energy-Recovery Technologies	38
Coil Loop	38
Typical Performance	38
Capacity Control	40
Frost Prevention	41
Cross-Leakage	42
Coil Loop with Three Coils	42
Fixed-Plate Heat Exchanger	43
Typical Performance	43
Capacity Control	44
Frost Prevention	45
Cross-Leakage	46
Heat Pipe	47
Typical Performance	47
Capacity Control	49
Frost Prevention	50
Cross-Leakage	50
Rotary Heat Exchanger	51
Typical Performance	52
Capacity Control	53
Frost Prevention	55
Cross-Leakage	57

Contents

ARI Standard 1060	60
Standard Rating Conditions	61
Using EATR and OACF for System Design	62
System-Level Control of Recovered Energy	65
OA Preconditioning:	
Dedicated OA System for Neutral-Temperature Air	68
Dedicated OA Systems for Cold Air	71
CV-MA Systems for Modulated-Temperature Supply Air	74
VAV-MA Systems for Constant-Temperature Supply Air	77
SA Tempering:	
Dedicated OA System for Neutral-Temperature Air	80
CV-MA Systems for Modulated-Temperature Supply Air	82
Applications that Combine OA Preconditioning and SA Tempering	84
Dual-Function, Dedicated OA System for Neutral-Temperature Air	84
Alternatives for Tempering with Sensible-Energy Recovery	88
Tempering (Recooling) in an Active-Desiccant System	88
Suggestions for Effective Airside Energy Recovery	90
OA Preconditioning	90
Sensible- or Total-Energy Recovery?	90
Central or Local OA Preconditioning?	93
Minimizing Life-Cycle Cost	95
SA Tempering	98
Sensible- or Total-Energy Recovery?	98
Series or Parallel Arrangement?	98
Minimizing Life-Cycle Cost	100
Glossary	102
References	106
Index	108

Introduction

Energy consumption costs money, uses natural resources, increases air pollution, and contributes to global warming. All of us share responsibility for prudent energy use.

The mechanical heating, ventilating, and air-conditioning (HVAC) system typically accounts for much of the energy costs in a building. Heat gains and losses from various sources must be offset to create a comfortable, healthy indoor environment. Discarded heat from one location may be useful elsewhere in the building, so it is logical (often cost-effective, too) to recover energy that might otherwise be wasted... particularly in a time of rising energy costs.

This manual discusses various technologies for and applications of **air-to-air energy recovery** in HVAC systems. ■

Why Recover Energy?

The answer to the question posed by the title of this section is twofold, at least in the context of an HVAC system: **economics** and **compliance with local codes**.

Economics

The energy impact of increased ventilation rates is one of several issues addressed in the U.S. EPA study, *Energy Cost and IAQ Performance of Ventilation Systems and Controls* (EPA-402-S-01-001, January 2000). ■

Efforts to improve indoor air quality (IAQ) typically include bringing more outdoor air into the building ... a practice that increases monthly utility costs by increasing the amount of heating, cooling, humidifying, and dehumidifying performed by the HVAC system. A study conducted by the U.S. Environmental Protection Agency (EPA) quantifies the economic effect of increased ventilation rates, which were defined in the 1989 version of ASHRAE Standard 62, *Ventilation for Acceptable Air Quality*. Depending on the building type, system type, and climate, the EPA study found that HVAC energy costs increased by 2 percent to 67 percent, while required capacities for peak cooling increased by 15 percent to 40 percent.

These statistics underscore the importance of identifying opportunities to reduce consumption, particularly when the cost and availability of energy are of increasing concern. Correctly implemented, **energy recovery** offsets the operating-cost penalty that results from bringing more outdoor air into the building. Recovering enough energy may even allow the use of smaller mechanical heating and cooling equipment.

Compliance with Codes

For more information about Standard 90.1 and its effect on the design of HVAC systems, refer to Trane *Engineers Newsletter*, “90.1 Ways to Save Energy,” which is available at www.trane.com/commercial/library/newsletters.

ANSI/ASHRAE/IESNA Standard 90.1–2001 is available from ASHRAE’s online bookstore at www.ashrae.org. The companion user’s manual explains the intent and application of the standard; it also includes a complete set of compliance forms and worksheets. ■

Setting aside the logical and practical arguments for recovering energy from other areas of the building, local codes may actually mandate the practice.

In the United States, the primary standard related to energy consumption in commercial buildings is ANSI/ASHRAE/IESNA Standard 90.1–2001, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. Standard 90.1 sets **minimum** design requirements that encourage energy efficiency throughout the building. Affected systems include lighting, motors, HVAC, service water heating, and the building envelope.

Section 6.3.6.1 of that standard requires the use of exhaust-air energy recovery when an individual fan system meets both of the following conditions:

- Design supply airflow equals or exceeds 5,000 cfm (2.4 m³/s) and
- Minimum outdoor airflow equals or exceeds 70 percent of the design supply airflow

Why Recover Energy?

This requirement generally applies to two types of systems: dedicated (100-percent) outdoor-air systems and mixed-air systems that serve spaces with high minimum requirements for outdoor airflow (for example, densely occupied spaces). Of course, exhaust-air energy recovery may make economic sense for systems that deliver 40-percent or less outdoor airflow by permitting equipment downsizing and consuming less new energy.

Standard 90.1 mandates a minimum effectiveness of 50 percent when exhaust-air energy is required. (The inset on p. 4 lists the exceptions to this requirement; **effectiveness** is discussed on pp. 6–7.) As defined by the standard,

Fifty percent energy recovery effectiveness shall mean a change in the *enthalpy* of the outdoor air supply equal to 50% of the difference between the outdoor air and return air at design conditions.

The publications listed below discuss energy recovery from refrigerant in direct-expansion refrigeration systems and from condenser water in water-cooled chilled water systems. They also offer suggestions for system optimization:

- *Refrigerant Heat Recovery*, Trane applications engineering manual SYS-AM-5
- *Multiple-Chiller-System Design and Control*, Trane applications engineering manual SYS-APM001-EN
- *Principles of Centrifugal Chiller Heat Recovery Operation*, Trane applications engineering manual AM-REF-2
- *Chiller Heat Recovery*, ASHRAE application guide ISBN 883413-99-0 ■

Further, the exhaust-air energy-recovery system must be installed with bypass dampers or other controls that permit airside economizer operation when such economizers are prescribed by Section 6.3.1.1 of the standard.

Section 6.3.2 of Standard 90.1 also relates to the use of energy recovery; it limits coincident heating and cooling of the same air stream. According to Section 6.3.2.3, if space humidity is directly regulated — through the use of humidistats, for example — controls must prevent reheating, mixing of hot and cold air streams, or other means of simultaneously heating and cooling the same air. (The inset on p. 4 lists the exceptions to this requirement.) Exception E permits reheat if at least 75 percent of the energy used to reheat or recool the air is recovered on site.

Limiting the use of “new” energy for reheat will almost certainly prompt HVAC design professionals to recover heat from the cooling process when reheat is required, particularly in dehumidification applications. Potential sources of recoverable heat include:

- Condenser water in a water-cooled, chilled water system
- Hot refrigerant in a refrigeration system
- Another air stream or another location in the same air stream

The rest of this manual describes how air-to-air heat exchangers can be used to transfer energy within or between air streams. ■

Why Recover Energy?

ANSI/ASHRAE/IESNA Standard 90.1–2001 on Exhaust-Air Energy Recovery ...

6.3.6.1 Exhaust Air Energy Recovery.

Individual fan systems that have both a design supply air capacity of 5,000 cfm (2.4 m³/s) or greater and have a minimum outside air supply of 70% or greater of the design supply air quantity shall have an energy recovery system with at least 50% recovery effectiveness. Fifty-percent energy recovery effectiveness shall mean a change in the enthalpy of the outdoor air supply equal to 50% of the difference between the outdoor air and return air at design conditions. Provision shall be made to bypass or control the heat recovery system to permit air economizer operation as required by 6.3.1.1.

Exceptions to 6.3.6.1:

- a) Laboratory systems meeting 6.3.7.2.
- b) Systems serving spaces that are not cooled and that are heated to less than 60°F (16°C).
- c) Systems exhausting toxic, flammable, paint or corrosive fumes or dust.
- d) Commercial kitchen hoods (grease) classified as Type 1 by *NFPA 96*.
- e) Where more than 60% of the outdoor heating energy is provided from site-recovered or site solar energy.
- f) Heating systems in climates with less than 3,600 HDD65.
- g) Cooling systems in climates with a 2.5% cooling design wet-bulb temperature less than 65°F (18°C).

Note: During development of the standard, the basis for reporting ASHRAE weather data changed from the 2.5% condition to the 1% condition. Although the weather data published in Appendix D reflects that change, the reference in Section 6.3.6.1 was missed.

- h) Where the largest exhaust source is less than 75% of the design outdoor airflow.
- i) Systems requiring dehumidification that employ series-style energy recovery coils wrapped around the cooling coil.

... and on Dehumidification

6.3.2.3 Dehumidification. Where humidistatic controls are provided, such controls shall prevent reheating, mixing of hot and cold air streams, or other means of simultaneous heating and cooling of the same air stream.

Exceptions to 6.3.2.3:

- a) The system is capable of reducing supply air volume to 50% or less of the design airflow rate or the minimum rate specified in 6.1.3 of ASHRAE Standard 62, whichever is larger, before simultaneous heating and cooling takes place.
- b) The individual fan cooling unit has a design cooling capacity of 80,000 Btu/h (23 kW) or less and is capable of

unloading to 50% capacity before simultaneous heating and cooling takes place.

- c) The individual mechanical cooling unit has a design cooling capacity of 40,000 Btu/h (12 kW) or less. An individual mechanical cooling unit is a single system composed of a fan or fans and a cooling coil capable of providing mechanical cooling.
- d) Systems serving spaces where specific humidity levels are required to satisfy process needs, such as computer rooms, museums, surgical suites, and buildings with refrigerating systems, such as supermarkets, refrigerated warehouses, and ice arenas. This exception also applies to other applications for which fan volume controls in accordance with Exception (a) are proven to be impractical to the enforcement agency.
- e) At least 75% of the energy for reheating or for providing warm air in mixing systems is provided from a *site-recovered* (including condenser heat) or *site solar energy* source.
- f) Systems where the heat added to the air stream is the result of the use of a desiccant system and 75% of the heat added by the desiccant system is removed by a heat exchanger, either before or after the desiccant system with energy recovery. ■

Applications for Air-to-Air Energy Recovery

Air-to-air energy recovery describes the transfer of sensible heat, or sensible plus latent heat, between two or more air streams, or between two locations within the same air stream. When applied in HVAC systems, air-to-air energy-recovery devices perform one of two principal functions:

- Precondition outdoor air brought into the building for ventilation
- Temper, or reheat, supply air for independent control of space conditions (dry-bulb temperature, humidity)

By including two separate, air-to-air energy-recovery devices, one for each function, an HVAC system can simultaneously provide preconditioning and tempering. (See “Applications that Combine OA Preconditioning and SA Tempering,” pp. 84–88.)

Basic Concepts

Let’s briefly review three fundamentals of energy recovery before discussing its application in the design of comfort systems: sensible versus total energy recovery, effectiveness ratio, and balanced versus unbalanced airflow.

Sensible versus Total Energy Recovery

Sensible-energy recovery refers to the transfer of sensible heat only. Adding sensible heat raises the dry-bulb temperature of an air stream; removing sensible heat lowers the dry-bulb temperature. Removing sensible heat may cause moisture to condense from the warmer air stream, but that moisture never transfers to the colder air stream.

Common examples of sensible-energy-recovery devices include coil loops, fixed-plate heat exchangers, heat pipes, and sensible-energy, rotary heat exchangers (also known as sensible-energy wheels).

Total-energy recovery describes the transfer of both sensible and latent heat. Latent heat is the energy stored in the water vapor of a moist air stream. Adding latent heat raises the dew-point temperature (moisture content) of an air stream; removing latent heat lowers the dew-point temperature.

Common examples of total-energy recovery devices include total-energy rotary heat exchangers (also known as total-energy wheels) and fixed-membrane heat exchangers.

(“Energy-Recovery Technologies,” pp. 38–64, further describes the sensible- and total-energy-recovery devices identified above.)

Applications for Air-to-Air Energy Recovery

Fundamental Equations for Air-to-Air Energy Recovery

Historically, manufacturers of air-to-air energy-recovery equipment used various methods to calculate the effectiveness of their devices. ARI Standard 1060–2000, *Rating Air-to-Air Energy Recovery Ventilation Equipment*, normalizes the calculation of effectiveness for rotary heat exchangers, heat pipes, and fixed-plate heat exchangers by defining the “effectiveness ratio” equation shown at right.

The only way to assure equitable comparisons of airside energy-recovery equipment is to require certification of performance in accordance with ARI 1060.

For more information about the standard, see “ARI Standard 1060,” pp. 60–64.

Note: ARI Standard 1060 does not apply to coil loops. ARI Standard 410–1991, Forced-Circulation Air-Cooling and Air-Heating Coils, defines the rating requirements for individual coils. To calculate coil-loop effectiveness, use the certified coil performance in the “effectiveness ratio” equation.

Effectiveness ratio. The following equation compares the amount of energy that is *actually* transferred with the maximum amount that *could* be transferred between the two air streams:

$$\varepsilon = \frac{\dot{m}_s \times (x_1 - x_2)}{\dot{m}_{min} \times (x_1 - x_3)}$$

where,

ε = sensible, latent, or total effectiveness

$\dot{m}_s$ = mass flow rate, lb/hr (kg/s), through *supply* side of heat-exchange device

$\dot{m}_e$ = mass flow rate, lb/hr (kg/s), through *exhaust* side of heat-exchange device

$\dot{m}_{min}$ = smaller mass flow rate (supply, $\dot{m}_s$; or exhaust, $\dot{m}_e$)

x_1 = entering supply air: dry bulb, °F (°C); humidity ratio, lb (water)/lb (dry air) (kg/kg); or enthalpy, Btu/lb (kJ/kg)

x_2 = leaving supply air: dry bulb, humidity ratio, or enthalpy

x_3 = entering exhaust air: dry bulb, humidity ratio, or enthalpy

x_4 = leaving exhaust air: dry bulb, humidity ratio, or enthalpy

Sensible-heat transfer is determined by the sensible effectiveness of the device, the minimum airflow rate, and the difference between the entering dry-bulb temperatures of the two air streams:

$$Q_s = \varepsilon \times \rho \times C_p \times \frac{60min}{1hr} \times V_{min} \times (T_1 - T_3)$$

$$(Q_s = \varepsilon \times \rho \times C_p \times V_{min} \times [T_1 - T_3])$$

where,

Q_s = sensible heat actually transferred, Btu/hr (kW)

ε = sensible effectiveness of device

ρ = density of air, lb/ft³ (kg/m³)

C_p = specific heat of air, Btu/lb • °F (kJ/kg • °K)

V_{min} = smaller airflow (supply or exhaust side), cfm (m³/s)

T_1 = entering supply-air dry bulb, °F (°C)

T_3 = entering exhaust-air dry bulb, °F (°C)

Note: At “standard” conditions, the product of density, specific heat, and the time conversion equals 1.085 (1.21). The minutes-per-hour factor is only needed to calculate Q_s in I-P units.

Total-heat transfer is determined by the total effectiveness of the device, the minimum airflow rate, and the difference between the entering enthalpies of the two air streams:

$$Q_T = \varepsilon \times \rho \times \frac{60min}{1hr} \times V_{min} \times (h_1 - h_3)$$

$$(Q_T = \varepsilon \times \rho \times V_{min} \times [h_1 - h_3])$$

where,

Q_T = total (sensible plus latent) heat actually transferred, Btu/hr (kW)

ε = total effectiveness of device

ρ = density of air, lb/ft³ (kg/m³)

V_{min} = smaller airflow (supply or exhaust side), cfm (m³/s)

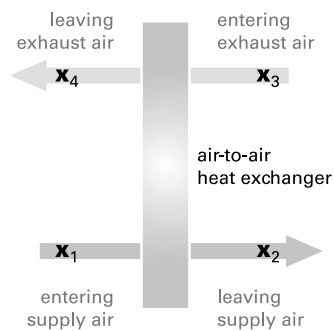
h_1 = entering supply-air enthalpy, Btu/lb (kJ/kg)

h_3 = entering exhaust-air enthalpy, Btu/lb (kJ/kg)

Note: At “standard” conditions, the product of density and the time conversion equals 4.5 (1.2). The minutes-per-hour factor is only needed to calculate Q_T in I-P units. ■

Applications for Air-to-Air Energy Recovery

Figure 1. Air conditions at an energy-recovery device



Effectiveness Ratio

Effectiveness, ϵ , quantifies the performance of the heat-exchange device when it is used for energy recovery. Mathematically, the effectiveness ratio represents the amount of energy (sensible, latent, or total) that the device actually transfers divided by the maximum energy transfer possible between two air streams; see Figure 1.

For a given heat exchanger geometry, effectiveness is determined at test conditions using various airflow rates. ANSI/ASHRAE Standard 84–1991, *Method of Testing Air-to-Air Heat Exchangers*, provides the equation (shown in the inset on p. 6) that describes the effectiveness of an air-to-air energy-recovery device. The equation can be used to determine sensible, latent, or total effectiveness based on measurements obtained through laboratory tests. Note that the minimum mass flow rate is the smaller of the mass flow rates through either side of the heat exchanger.

The amount of sensible heat transferred depends on the sensible effectiveness of the device, the minimum airflow rate, and the difference between the dry-bulb air temperatures entering the supply and exhaust sides of the heat exchanger. The heat-transfer equations on p. 6 express the mathematical relationships between these variables.

Similarly, the amount of total energy transferred depends on the total effectiveness of the device, the minimum airflow rate, and the difference between the enthalpies of the air streams entering the supply and exhaust sides of the device. Again, minimum airflow is the smaller of the two airflows, whether supply or exhaust.

Balanced versus Unbalanced Airflows

Airflows are considered “unbalanced” when the rate of flow through one side of the heat exchanger differs from the rate of flow through the other side. Laboratory tests suggest that unbalanced airflows increase the effectiveness of the device. In actual operation, however, the device transfers less heat overall because the amount of heat transferred depends on the smaller of the two airflows. This irony is best explained using an example.

Figure 2 (p. 8) illustrates the relationship between airflows and the effectiveness of a generic sensible-energy-recovery device. The horizontal axis represents exhaust-side airflow divided by supply-side airflow. A value of 1.0 indicates

Applications for Air-to-Air Energy Recovery

Figure 2. Example of effectiveness for a sensible-energy-recovery device

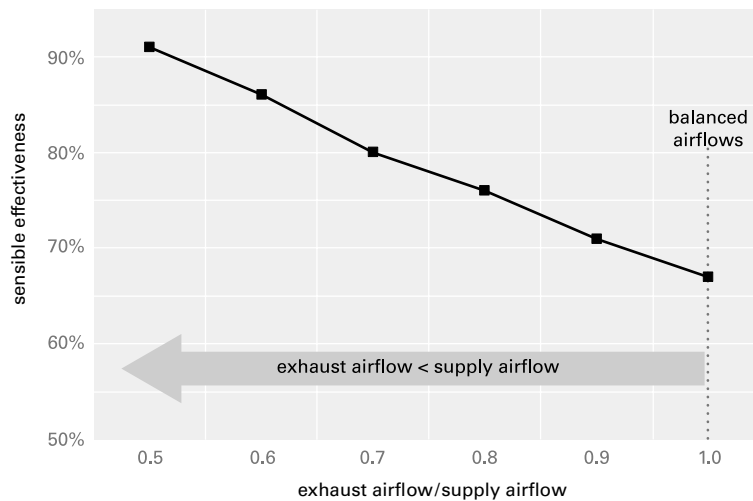
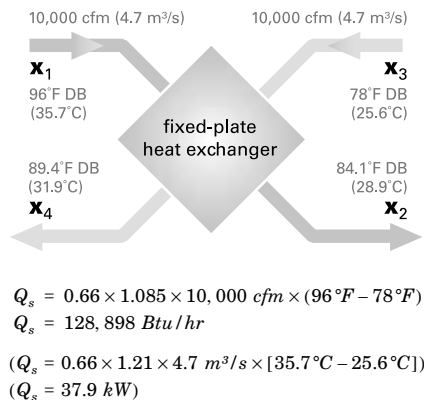


Figure 3. Balanced (equal) airflows



balanced airflows; a value less than 1.0 indicates that the exhaust airflow is less than the supply airflow.

Figure 3 shows how the sensible-energy-recovery device performs when operated with balanced exhaust and supply airflows, that is, with 10,000 cfm (4.7 m³/s) of air flowing through each side of the device. Sensible effectiveness at this balanced condition is 66 percent (Figure 2). Using that value to solve the sensible-heat-transfer equation indicates that the device will transfer 128,898 Btu/hr (37.9 kW) of sensible heat from the supply air stream to the exhaust air stream at standard air conditions.

Figure 4. Unbalanced (unequal) airflows

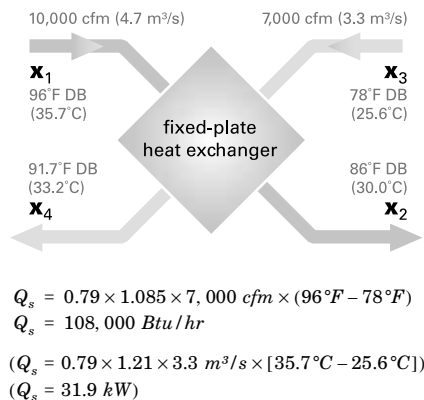


Figure 4 demonstrates how unbalanced airflows affect the performance of the same energy-recovery device. In this case, the supply airflow is 10,000 cfm (4.7 m³/s), but the exhaust airflow is only 7,000 cfm (3.3 m³/s). Although the imbalance increases the sensible effectiveness of the device to 79 percent, it now transfers 16 percent less sensible heat from the supply air stream to the exhaust air stream — that is, only 108,000 Btu/hr (31.9 kW).

Balanced airflow is difficult to achieve in actual installations: exfiltration and local exhausts usually result in a system-level exhaust airflow that is less than the outdoor airflow brought into the building. Despite this challenge, equalizing outdoor and exhaust airflows not only affords the greatest reductions in cooling and heating capacities but also increases energy savings.

Note: In most applications, an exhaust airflow that is at least 70 percent of the outdoor airflow yields an attractive payback.

Applications for Air-to-Air Energy Recovery

Outdoor-Air Preconditioning

Buildings are ventilated with outdoor air to prevent the buildup of contaminants that are generated indoors. In most climates, this outdoor air imposes a sizable load on the HVAC equipment.

Outdoor-air preconditioning is the process of cooling, dehumidifying, heating, and/or humidifying the outdoor air entering the system. Using air-to-air energy recovery (sometimes called “exhaust-air energy recovery”) to perform this function can significantly reduce the load on HVAC equipment as well as the energy consumed by that equipment. Energy recovery may even be required to comply with ASHRAE Standard 90.1–2001 (Section 6.3.6) and other energy codes.

Figure 5. OA-preconditioning arrangements

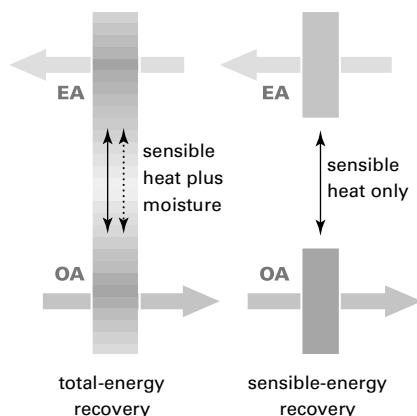


Figure 5 represents air-to-air energy-recovery arrangements that precondition outdoor air (OA) by exchanging energy with the exhaust air (EA) stream. During the cooling season, when it is hot and humid outside, the *total-energy-recovery* device pre-cools and “pre-dries” (dehumidifies) the outdoor air by transferring both sensible heat and moisture to the exhaust air stream. During the heating season, when outdoor conditions are cold and dry, the same arrangement preheats and prehumidifies the outdoor air by removing both sensible heat and moisture from the exhaust air stream and releasing it into the supply air stream.

Substituting a *sensible-energy* recovery device limits energy transfer to sensible heat; moisture (latent heat) is not recovered. As the examples that follow demonstrate, either type of recovered energy — sensible or total — can effectively precondition outdoor air. First cost and potential energy savings typically determine which type of recovery best suits a particular application.

Sensible versus Total Energy Recovery

A Scenario to Aid Comparison

An energy-recovery device installed in a central air handler preconditions 10,000 cfm (4.7 m³/s) of outdoor air. Local exhausts for the rest rooms remove air from the building, as does the exfiltration caused by maintaining an indoor pressure that is slightly positive relative to outdoors. With only 7,000 cfm (3.3 m³/s) exhausted centrally, unbalanced airflows exist at the energy-recovery device. Table 1, p. 10, summarizes the design airflows and thermal conditions for this scenario.

The rest of this section compares the energy-recovery performance of this system in two climates — Jacksonville, Florida (predominantly cooling) and Minneapolis, Minnesota (predominantly heating) — and with different energy-recovery devices: a coil loop that recovers only sensible energy and a total-energy wheel that recovers both sensible and latent energy.

Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

Table 1. Design parameters for examples of OA preconditioning (Figures 6–9)

		Cooling (ASHRAE 0.4%)	Heating (ASHRAE 99.6%)
Location ¹	Jacksonville, Fla.	96°F DB (35.7°C) 76°F WB (24.5°C)	31°F DB (-0.8°C)
	Minneapolis, Minn.	91°F DB (32.8°C) 73°F WB (22.7°C)	-16°F DB (-26.5°C)
Return air from the conditioned space		78°F DB (25.6°C) 63.5°F WB (17.5°C)	70°F DB (21.1°C) 54°F WB (12.2°C)
Airflows	Outdoor	10,000 cfm (4.7 m³/s)	
	Central exhaust	7,000 cfm (3.3 m³/s)	
	Local exhaust (rest rooms)	2,500 cfm (1.2 m³/s)	
Exfiltration through building envelope		500 cfm (0.2 m³/s)	

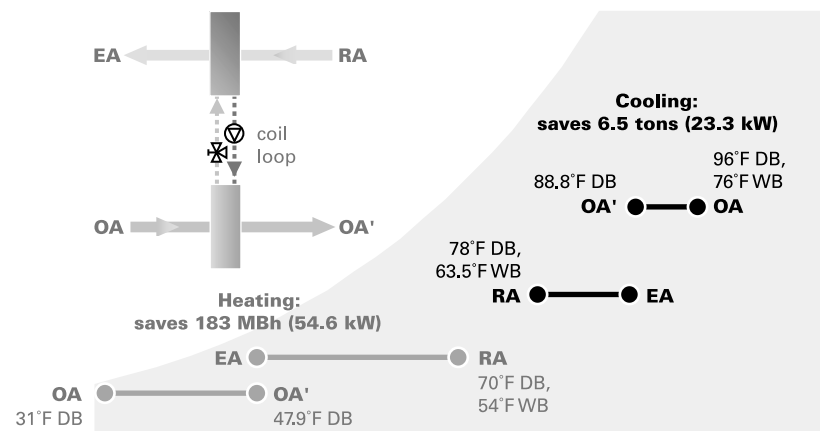
¹ Cooling design conditions for these locations represent the ASHRAE 0.4% design condition for sensible cooling, that is, the peak dry-bulb temperature and mean coincident wet-bulb temperature. Heating design conditions represent the ASHRAE 99.6% design condition.

OA Preconditioning with Recovered Sensible Energy

For this comparison, the sensible-energy recovery device in Figure 6 represents a coil loop with an effectiveness of 57 percent. During the cooling season, the coil loop preconditions the outdoor air stream by transferring sensible heat to the exhaust air stream. During the heating season, it warms the entering outdoor air with sensible heat recovered from the exhaust air.

Let's first look at the design conditions for Jacksonville, Florida, where cooling predominates. The ASHRAE 0.4-percent condition for sensible cooling design (that is, peak dry-bulb temperature and mean coincident wet-bulb temperature) is 96°F dry bulb and 76°F wet bulb (35.7°C DB, 24.5°C WB). At this condition,

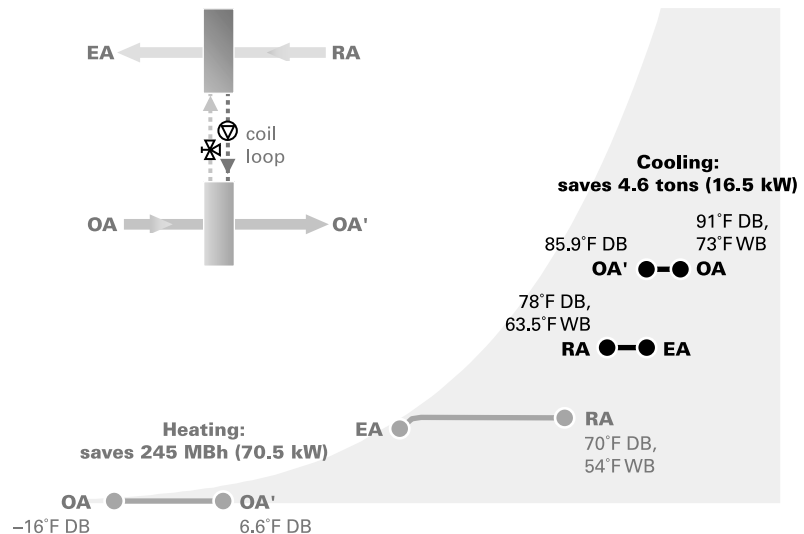
Figure 6. OA preconditioning with sensible-energy recovery in Jacksonville, Florida



Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

Figure 7. OA preconditioning with sensible-energy recovery in Minneapolis, Minnesota



Cooling-season savings in Jacksonville, Fla.:

$$Q_s = 1.085 \times 10,000 \text{ cfm} \times (96^\circ\text{F} - 88.8^\circ\text{F})$$

$$Q_s = 78,120 \text{ Btu/hr} = 6.5 \text{ tons}$$

$$(Q_s = 1.21 \times 4.7 \text{ m}^3/\text{s} \times [35.7^\circ\text{C} - 31.6^\circ\text{C}])$$

$$(Q_s = 23.3 \text{ kW})$$

Heating-season savings in Jacksonville, Fla.:

$$Q_s = 1.085 \times 10,000 \text{ cfm} \times (47.9^\circ\text{F} - 31^\circ\text{F})$$

$$Q_s = 183,365 \text{ Btu/hr} = 183 \text{ MBh}$$

$$(Q_s = 1.21 \times 4.7 \text{ m}^3/\text{s} \times [8.8^\circ\text{C} - (-0.8^\circ\text{C})])$$

$$(Q_s = 54.6 \text{ kW})$$

Cooling-season savings in Minneapolis, Minn.:

$$Q_s = 1.085 \times 10,000 \text{ cfm} \times (91^\circ\text{F} - 85.9^\circ\text{F})$$

$$Q_s = 55,335 \text{ Btu/hr} = 4.6 \text{ tons}$$

$$(Q_s = 1.21 \times 4.7 \text{ m}^3/\text{s} \times [32.8^\circ\text{C} - 29.9^\circ\text{C}])$$

$$(Q_s = 16.5 \text{ kW})$$

Heating-season savings in Minneapolis, Minn.:

$$Q_s = 1.085 \times 10,000 \text{ cfm} \times (6.6^\circ\text{F} - [-16^\circ\text{F}])$$

$$Q_s = 245,210 \text{ Btu/hr} = 245 \text{ MBh}$$

$$(Q_s = 1.21 \times 4.7 \text{ m}^3/\text{s} \times [-14.1^\circ\text{C} - (-26.5^\circ\text{C})])$$

$$(Q_s = 70.5 \text{ kW})$$

based on data from a coil selection program, the coil loop precools the outdoor air to 88.8°F DB (31.6°C). It also reduces the required cooling capacity by 6.5 tons (23.3 kW), which means lower operating costs and permits downsizing of the cooling plant.

Figure 6 also shows the winter performance of the coil loop. The ASHRAE 99.6-percent condition for heating design in Jacksonville is 31°F DB (-0.8°C). This time the coil loop transfers sensible heat from the exhaust air stream to the outdoor air stream, warming it to 47.9°F (8.8°C). The recovered energy reduces the design heating capacity by 183 MBh (54.6 kW) and, in turn, reduces operating costs and permits downsizing of the heating plant.

How well does the same coil loop recover sensible energy in a climate where heating predominates? Figure 7 depicts the performance of our example HVAC system in Minneapolis, Minnesota. At the sensible-cooling-design condition of 91°F DB and 73°F WB (32.8°C DB, 22.7°C WB), the coil loop precools the outdoor air to 85.9°F DB (29.9°C) and reduces the required cooling capacity by 4.6 tons (16.5 kW). At the heating design condition of -16°F (-26.5°C), recovered energy preheats the outdoor air to 6.6°F (-14.1°C), which saves 245 MBh (70.5 kW).

Notice that the air that passes through the exhaust side of the coil loop reaches the saturation curve (Figure 7), indicating that moisture from the exhaust air stream is condensing on the coil surface. The small amount of latent heat removed from the exhaust air contributes to the total quantity of heat that transfers to the outdoor air stream; however, only sensible (not latent) heat is transferred. Condensation on the exhaust side of the energy-recovery device can pose a problem in cold climates. (See "Frost Prevention," pp. 18–19.)

Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

OA Preconditioning with Recovered Total Energy

Figure 8 and Figure 9 illustrate the effect of recovering *total* energy in the scenario described on p. 9. For this example, the energy-recovery device is a total-energy wheel with an effectiveness of 82 percent, given unbalanced airflows of 10,000 cfm (4.7 m³/s) on the supply side of the wheel and 3,000 cfm (3.3 m³/s) on the exhaust side. During the cooling season, the wheel transfers sensible and latent heat (moisture) from the outdoor air stream to the exhaust air stream, precooling and dehumidifying the incoming outdoor air. Conversely, the wheel warms and humidifies the outdoor air stream during the heating season, using sensible heat and moisture recovered from the exhaust air stream.

Cooling-season savings in Jacksonville, Fla.:

$$Q_T = 4.5 \times 10,000 \text{ cfm} \times (39.3 - 33.3 \text{ Btu/lb})$$

$$Q_T = 270,000 \text{ Btu/hr} = 22.5 \text{ tons}$$

$$(Q_T = 1.2 \times 4.7 \text{ m}^3/\text{s} \times [91.4 - 77.5 \text{ kJ/kg}])$$

$$(Q_T = 78.4 \text{ kW})$$

Heating-season savings in Jacksonville, Fla.:

$$Q_s = 1.085 \times 10,000 \text{ cfm} \times (53.3^\circ\text{F} - 31^\circ\text{F})$$

$$Q_s = 241,955 \text{ Btu/hr} = 242 \text{ MBh}$$

$$(Q_s = 1.21 \times 4.7 \text{ m}^3/\text{s} \times [11.8^\circ\text{C} - (-0.8^\circ\text{C})])$$

$$(Q_s = 71.7 \text{ kW})$$

$$Q_L = 0.7 \times 10,000 \text{ cfm} \times (29.3 - 20.2 \text{ grains/lb})$$

$$Q_L = 63,700 \text{ Btu/hr} = 63.7 \text{ MBh}$$

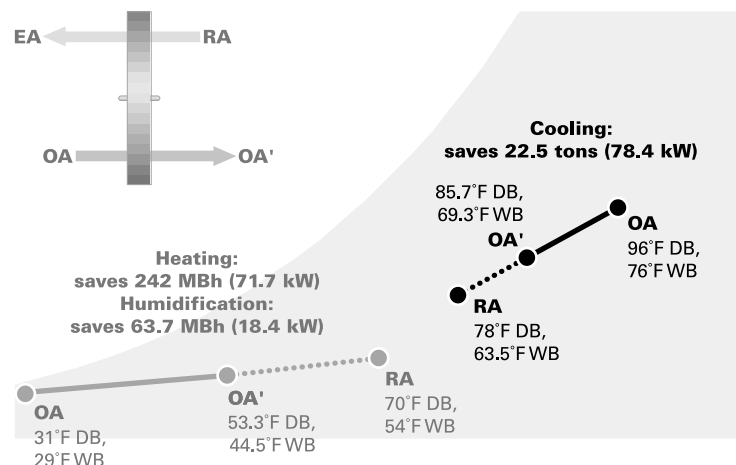
$$(Q_L = 3,010 \times 4.7 \text{ m}^3/\text{s} \times [0.0042 - 0.0029 \text{ kg/kg}])$$

$$(Q_L = 18.4 \text{ kW})$$

The selection program for the wheel indicates that, at Jacksonville's cooling design condition (Table 1, p. 10), the wheel will cool and dehumidify the outdoor air to 85.7°F DB and 69.3°F WB (29.8°C DB, 20.7°C WB). See Figure 8. Transferring moisture as well as sensible heat lowers operating costs and reduces the required cooling capacity — this time by 22.5 tons (78.4 kW).

At Jacksonville's heating design condition, the sensible energy recovered from the exhaust air stream raises the dry-bulb temperature of the entering outdoor air to 53.3°F DB (11.8°C). Preheating the outdoor air in this fashion reduces the required heating capacity by 242 MBh (71.7 kW). The total-energy wheel also humidifies the entering outdoor air with moisture recovered from the exhaust air stream, increasing the moisture content from 20.2 grains/lb to 29.3 grains/lb (0.0029 kg/kg to 0.0042 kg/kg). If the HVAC system includes a mechanical humidifier, the capacity required at design can be reduced by 63.7 MBh (18.4 kW).

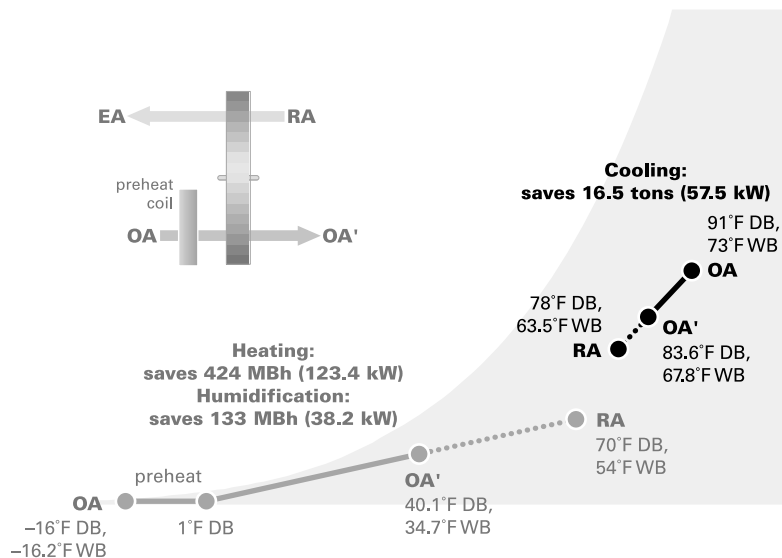
Figure 8. OA preconditioning with total energy recovery in Jacksonville, Florida



Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

Figure 9. OA preconditioning with total energy recovery in Minneapolis, Minnesota



Cooling-season savings in Minneapolis, Minn.:

$$Q_T = 4.5 \times 10,000 \text{ cfm} \times (36.5 - 32.1 \text{ Btu/lb})$$

$$Q_T = 198,000 \text{ Btu/hr} = 16.5 \text{ tons}$$

$$(Q_T = 1.2 \times 4.7 \text{ m}^3/\text{s} \times [84.9 - 74.7 \text{ kJ/kg}])$$

$(Q_T = 57.5 \text{ kW})$

Heating-season savings in Minneapolis, Minn.:

$$Q_s = 1.085 \times 10,000 \text{ cfm} \times (40.1^\circ\text{F} - 1^\circ\text{F})$$

$$Q_s = 424,235 \text{ Btu/hr} = 424 \text{ MBh}$$

$$(Q_s = 1.21 \times 4.7 \text{ m}^3/\text{s} \times [4.5^\circ\text{C} - (-17.2^\circ\text{C})])$$

$$(Q_s = 123.4 \text{ kW})$$

$$Q_L = 0.7 \times 10,000 \text{ cfm} \times (21.0 - 2.0 \text{ grains/lb})$$

$$Q_L = 133,000 \text{ Btu/hr} = 133.0 \text{ MBh}$$

$$(Q_L = 3,010 \times 4.7 \text{ m}^3/\text{s} \times [0.003 - 0.0003 \text{ kg/kg}])$$

$(Q_L = 38.2 \text{ kW})$

Figure 9 illustrates how the same total-energy wheel performs in Minneapolis at design conditions. During the summer, the wheel cools and dehumidifies the outdoor air to 83.6°F DB, 67.8°F WB (28.7°C DB, 19.9°C WB), which lowers the required cooling capacity by 16.5 tons (57.5 kW). During the winter, the wheel heats the outdoor air to 40.1°F (4.5°C) DB, which saves 424 MBh (123.4 kW) in sensible heating capacity. The wheel also humidifies the entering outdoor air, raising its moisture content from 2 grains/lb to 21 grains/lb (0.0003 kg/kg to 0.003 kg/kg). If the HVAC system includes mechanical humidification, then operating the total-energy wheel will reduce the required humidification capacity by 133 MBh (38.2 kW).

Note: We added a preheat coil upstream of the wheel for our Minneapolis example. The preheat coil warms the entering outdoor air to about 1°F (-17.2°C) to prevent frost from forming on the exhaust-side surface of the wheel. See "Frost Prevention," pp. 18–19 and pp. 55–56, for more information.

Effect on First Cost

The previous examples demonstrate the effects of preconditioning outdoor air with energy recovered from the exhaust air stream. Although the benefits are appealing, they are not without cost. Depending on size and complexity, adding an energy-recovery device can significantly increase the initial investment in the air handler.

Additional exhaust ductwork. Except for coil loops, air-to-air energy-recovery devices require adjacent air streams. Routing most of the exhaust air

Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

to a central location so that the path is adjacent to that of the entering outdoor air may require more ductwork (and space) than an application without energy recovery. Even if a coil loop is used, minimizing the distance between the air streams will not only reduce piping- and glycol-related costs but also permit a smaller pump and expansion tank.

Increased fan motor sizes. Adding an air-to-air energy-recovery device to precondition outdoor air increases the static pressure drop in both the outdoor- and exhaust-air paths, requiring larger fan motors. It may even create the need for a powered, central exhaust fan where one was not required before. The configuration of the fans will determine the extent to which their motor sizes — and energy consumption — will increase.

In a mixed-air system, the impact on the supply fan depends on the pressure drop through the outdoor-air path versus that of the return-air path. If the static pressure drop through the return-air path is greater, the supply fan need not overcome the entire static pressure drop created by the energy-recovery device. Adding a third (and even a fourth) fan could make it unnecessary to increase the sizes of the supply and exhaust fan motors. The energy costs of this approach would be nearly the same as for a two-fan system, although the installation and maintenance costs would almost certainly be higher.

Heating- and cooling-plant downsizing. Whether the climate is predominantly cooling (Jacksonville, Florida) or predominantly heating (Minneapolis, Minnesota), the first-cost savings from even a modest reduction in HVAC capacity can at least partially subsidize the investment in the energy-recovery device, associated ductwork, and larger fan motors.

The decision to use an air-to-air energy-recovery device for outdoor-air preconditioning requires careful consideration...and not for solely economic reasons. Downsizing the HVAC equipment, particularly in packaged direct-expansion (DX) applications, avoids other system- and building-related problems that can result from overcapacity. Providing too much available cooling can cause compressors to short-cycle and create swings in supply-air temperature that not only make occupants uncomfortable but also result in inadequate humidity control. Failure to maintain appropriate humidity levels can cause microbial growth to flourish inside the building, adversely affecting occupant health and accelerating the deterioration of structural components and furnishings.

Table 2 summarizes the plant-downsizing potential provided by the sensible- and total-energy recovery devices in the previous examples. Recall that unbalanced airflows through an energy-recovery device increase effectiveness but lessen the amount of heat transferred overall. Table 3 illustrates the additional first-cost savings made possible by routing the rest-room exhaust through the energy-recovery device — which reduces the airflow imbalance from 3,000 cfm to 500 cfm ($1.4 \text{ m}^3/\text{s}$ to $0.2 \text{ m}^3/\text{s}$) — rather than expelling it locally. The net savings in first cost usually makes it beneficial to route as much

Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

exhaust air to the energy-recovery device as possible, despite the extra expense incurred for rest-room-exhaust ductwork and exhaust-fan power.

The economics of using an air-to-air energy recovery device to precondition outdoor air often hinges on downsizing both plants, cooling and heating. Some engineers hesitate to reduce the capacity of the heating plant for installations in cold climates even though the energy-recovery device makes the extra capacity redundant. (The importance of redundancy depends on the specific application.) Their reluctance often reflects concern that the failure of the energy-recovery device, usually a total-energy wheel, will leave the system without sufficient capacity to heat the building. That concern is unwarranted because the total-energy wheel only reduces the heating load of the outdoor air brought into the building for ventilation. If a mechanical failure occurs, the system operator can choose to temporarily bring less outdoor air into the building until the wheel is repaired.

This observation raises another important point: *An energy-recovery device that preconditions outdoor air is no less critical than the boiler or the belt on a supply fan.* If the wheel breaks, repair it promptly.

Table 2. Capacity reductions attributable to OA preconditioning (unbalanced airflows)

Capacity Reduction Due to Recovered Energy ¹	Jacksonville, Florida		Minneapolis, Minnesota	
	Sensible	Total	Sensible	Total
Cooling plant	6.5 tons (23.3 kW)	22.5 tons (78.4 kW)	4.6 tons (16.5 kW)	16.5 tons (57.5 kW)
Heating plant	183.0 MBh (54.6 kW)	242.0 MBh (71.7 kW)	245.0 MBh (70.5 kW)	424.0 MBh (123.4 kW)
Humidification system	n/a	63.7 MBh (18.4 kW)	n/a	133.0 MBh (38.2 kW)

¹ Capacity reductions are based on the effectiveness of the energy-recovery device, which results from the unbalanced airflows of 10,000 cfm (4.7 m³/s) outdoor air and 7,000 cfm (3.3 m³/s) exhaust air.

Table 3. Capacity reductions attributable to OA preconditioning ("balanced" airflows)

Capacity Reduction Due to Recovered Energy ¹	Jacksonville, Florida		Minneapolis, Minnesota	
	Sensible	Total	Sensible	Total
Cooling plant	7.3 tons (26.3 kW)	25.1 tons (88.3 kW)	5.2 tons (18.3 kW)	19.9 tons (69.7 kW)
Heating plant	187.7 MBh (56.5 kW)	289.7 MBh (84.9 kW)	299.5 MBh (87.9 kW)	517.5 MBh (151.3 kW)
Humidification system	n/a	78.4 MBh (23.7 kW)	n/a	161.0 MBh (46.9 kW)

¹ Capacity reductions are based on the effectiveness of the energy-recovery device, which results from the nearly balanced airflows of 10,000 cfm (4.7 m³/s) outdoor air and 9,500 cfm (4.5 m³/s) exhaust air.



Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

Effect on Operating Costs

Less energy consumed for heating and cooling. Typically, the “real” money in using air-to-air energy recovery for outdoor-air preconditioning accrues from the first-cost “subsidy” rather than lower operating costs. Still, if the first-cost subsidy from downsizing the cooling and heating plants does not entirely offset the initial investment in the energy-recovery device, operating cost savings may quickly make up the difference.

Energy simulation software makes it easier to predict and evaluate how various energy-recovery devices will perform in a particular application. Such an analysis can predict the length of the payback period based on the building location, type of building, type of HVAC system, system control modes, quantity of outdoor air, airside economizer operation, and local utility rates. Interaction between different system operating modes and the airside economizer often requires the use of a whole-building economic analysis (performed with the help of DOE-2, TRACE™, System Analyzer™, or similar software).

More energy consumed by fans. Whenever air passes through the air-to-air energy recovery device, the static pressure drop of the system increases and the fans, in turn, consume more energy. The magnitude of the pressure drop through the device and the configuration of the fans determine how much additional energy is consumed. The operating cost savings provided by recovered energy must exceed the increased cost of operating the fans in order to justify the cost of the energy-recovery device. A year-round, project-specific analysis must be performed to determine the overall effect on the energy consumption of the HVAC system.

Proper airside economizer control. In many climates, the operation of a mixed-air system’s airside economizer can provide the benefits of “free” cooling for much of the year. While the economizer operates, air-to-air energy recovery offers no additional benefit. In fact, unless it is turned off, the energy-recovery device actually increases the cooling load by adding heat to the supply air stream. Achieving maximum energy savings and minimizing the payback period for energy recovery therefore depends on proper economizer operation as well as on proper integration of the energy-recovery device into the system control modes. (For a detailed discussion of these control modes, see “System-Level Control of Recovered Energy,” pp. 65–89.)

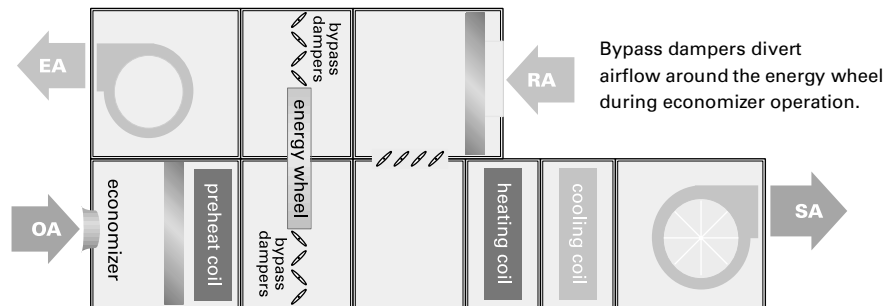
When the HVAC system includes an airside economizer, size the air-to-air energy-recovery device to precondition only the minimum outdoor airflow required for ventilation... not the maximum airflow expected during economizer operation. (Avoiding oversizing also minimizes the first cost of the device.)

To accommodate economizer operation when the energy-recovery device is idle, add bypass dampers (Figure 10) to reduce the airside pressure drop and

Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

Figure 10. Central air handler with economizer and energy recovery



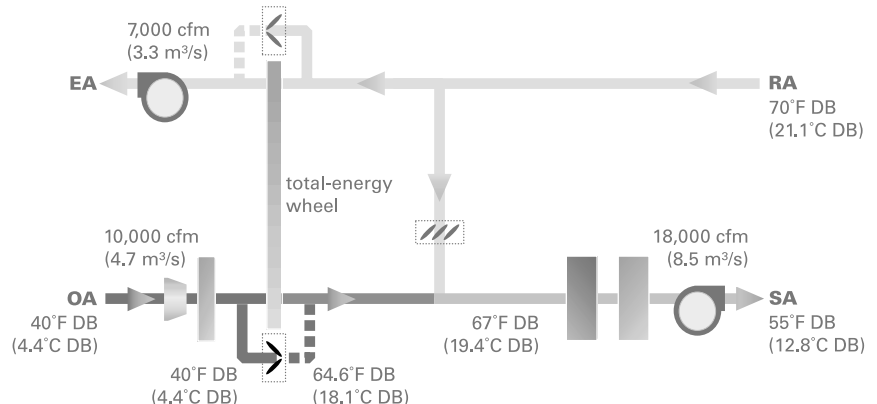
minimize the energy consumption of the fans. Alternatively, provide two separate paths for outdoor air: one for “ventilation” air and the other for “economizer” air.

Note: Section 6.3.6.1 of ASHRAE Standard 90.1–2001 requires bypass, or some other control, to permit airside economizer operation when an economizer is required by Section 6.3.1.1.

Need for capacity control. Although it is often overlooked, capacity control is a key aspect of designing an HVAC system that uses air-to-air energy recovery to precondition outdoor air. Regardless of whether sensible or total energy is recovered, modulating the capacity of the energy-recovery device prevents it from overheating the supply air when weather conditions are cold but not extreme. Consider the following example.

Assume that a total-energy wheel preconditions outdoor air in a mixed-air, variable-air-volume (VAV) system that supplies air at a constant 55°F DB (12.8°C); see Figure 11. When it is 40°F DB (4.4°C) outside, the system delivers a

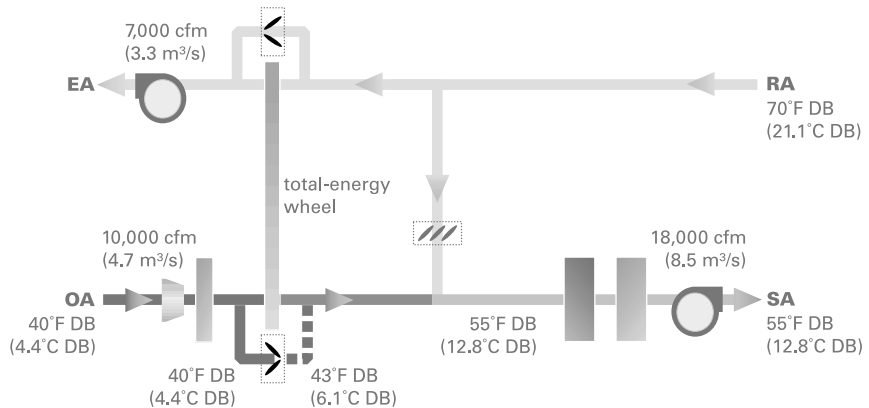
Figure 11. Unmodulated energy-recovery capacity for OA preconditioning



Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

Figure 12. Modulated energy-recovery capacity for OA preconditioning



supply airflow of 18,000 cfm (8.5 m³/s). With the wheel turned off, the resulting mixed-air condition would be 53.3°F (11.8°C). Activating the wheel transfers sensible heat (and moisture) from the warm exhaust air to the cool entering outdoor air, avoiding the use of mechanical heating. However, if allowed to operate at full capacity, the wheel would heat the entering outdoor air to 64.6°F (18.1°C) and, after mixing with return air from the space, produce a mixed-air condition of 67°F (19.4°C). Mechanical or economizer cooling is then necessary to obtain the supply-air target of 55°F (12.8°C).

Regulating the capacity of the wheel — in this case, by modulating the exhaust-side dampers (Figure 12) — can prevent overheating. Operating the wheel at less than full capacity allows it to warm the entering outdoor air to only 43°F (6.1°C), which yields 55°F (12.8°C) mixed air without the help of mechanical heating or cooling.

Note: Capacity control for energy-recovery devices is important during heating operation, but is seldom required during the cooling mode. Specific control methods are discussed in "Energy-Recovery Technologies," pp. 38–59.

Other Considerations

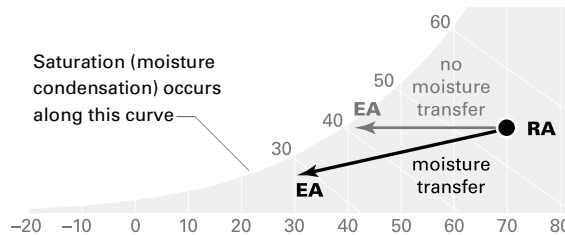
Although it is important to optimize life-cycle costs, other issues must be considered to effectively precondition outdoor air using air-to-air energy recovery.

Frost prevention. Any air-to-air energy-recovery device that preconditions outdoor air is subject to frost buildup during very cold weather. That's because the moisture that condenses on the exhaust-side of the device (recall Figure 7, p. 11) freezes, eventually blocking airflow, if the surface temperature of the device falls below 32°F (0°C). Device effectiveness, entering conditions, and

Precondition Outdoor Air

Applications for Air-to-Air Energy Recovery

Figure 13. Psychrometric Comparison of Frost Thresholds for Energy-Recovery Devices



airflows determine the outdoor temperature at which frost forms; but the threshold is much lower for total-energy-recovery devices than for devices that recover only sensible energy. Figure 13 illustrates why...

When exhaust air passes through a total-energy-recovery device, the transfer of moisture to the outdoor air stream (psychrometrically) prevents the leaving exhaust-air condition from reaching the saturation curve. Sensible-energy-recovery devices do not remove moisture from the exhaust air stream, so the leaving exhaust-air condition reaches saturation (and frosts) at a warmer outdoor condition.

Note: Although the latent heat of condensation is slight, it contributes to the amount of heat that a sensible-energy-recovery device transfers to the outdoor air stream.

Frost can be avoided in one of two ways:

- **Reduce the heat-transfer capacity of the energy-recovery device.** This practice raises the surface temperature of the device, but limits the extent to which the heating plant can be downsized.
- **Preheat the outdoor (or exhaust) air before it enters the energy-recovery device.** Permitting full-capacity energy recovery at the coldest outdoor conditions maximizes the potential downsizing of the heating plant.

"Air-to-Air Energy Recovery Technologies" (pp. 38–59) discusses specific frost prevention methods for various air-to-air energy-recovery devices.

Proximity of outdoor-air intake to building exhaust. Local codes often impose a minimum separation distance between the outdoor-air intake and the building exhaust. In applications that route the exhaust air to an indoor air handler for energy recovery, the outdoor- and exhaust-air ductwork can usually be arranged to provide the minimum separation distance. Meeting this requirement poses a greater design challenge in applications with outdoor air handlers.

Note: A proposed addendum to ASHRAE Standard 62 includes minimum distance requirements for the location of an outdoor-air intake. For example,

Applications for Air-to-Air Energy Recovery

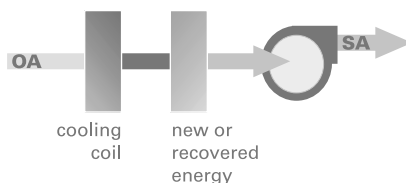
the standard suggests positioning the outdoor-air intake at least 1 ft (0.3 m) above the ground or roof, and at least 5 ft (1.5 m) from any source of general (Class 1 or 2) building exhaust.

Acceptable cross-leakage. Many types of air-to-air energy-recovery devices permit some degree of cross-leakage, although the amount of leakage is usually less than 5 percent (even for wheels) in most applications. (“Energy-Recovery Technologies,” pp. 38–59, discusses specific methods of reducing cross-leakage for various types of air-to-air energy-recovery devices.)

Cross-leakage between the exhaust- and supply-air streams is seldom problematic in mixed-air systems: most of the air that returns from the space is recirculated as supply air anyway. Local codes that permit the recirculation of exhaust air from rest rooms usually require sufficient dilution with fresh outdoor air.

Note: Another proposed addendum to ASHRAE Standard 62, if adopted, will prescribe acceptable amounts of cross-leakage for different classes of air. As an example, the addendum would limit recirculated exhaust air from rest rooms to not more than 10 percent of supply airflow.

Figure 14. Supply-air (SA) tempering



For a more detailed discussion of the use of supply-air tempering to control space humidity, refer to the *Engineers Newsletter* titled “Dehumidify with Constant-Volume Systems” (ENEWS-29/4). You can find it in the newsletters section of www.trane.com/commercial.

A forthcoming Trane applications engineering manual, *Dehumidification in HVAC Systems* (SYS-APM004-EN), will also address this subject. Publication is scheduled for June 2002. Check the Trane bookstore at www.trane.com/bookstore for availability. ■

Supply-Air Tempering

Supply-air tempering adds sensible heat to the air downstream of the cooling coil so that the dry-bulb temperature and humidity of the conditioned space can be controlled independently.

In the simple schematic depicted in Figure 14, the cooling coil dehumidifies the supply air, lowering the dew point sufficiently to maintain the desired space humidity. If the dehumidified supply air is so cold that it would overcool the space, then it must be reheated (or “tempered”) before delivery to prevent the dry-bulb temperature in the space from falling below the setpoint.

Only systems that directly control the space humidity or supply-air dew point require supply-air tempering. Common applications include:

- Constant-volume, mixed-air systems that directly control space humidity
- Dedicated (100-percent) outdoor-air systems that deliver dry (low dew-point) air at a “space-neutral” dry-bulb temperature

Supply-air tempering can be accomplished using either new energy — electricity, hot water, steam or gas, for example — or recovered energy from another part of the system. Moisture transfer is seldom desirable when tempering supply air, so energy recovery is typically limited to sensible heat.

Temper Supply Air

Applications for Air-to-Air Energy Recovery

Note: To comply with ASHRAE Standard 90.1–2001 (Section 6.3.2) and related energy codes, site-recovered energy may be required for applications that include supply-air tempering. Standard 90.1 (see p. 4) limits the use of reheat in a previously cooled air stream unless at least 75 percent of the reheat energy is recovered.

The rest of this section examines the effect of using air-to-air heat exchangers to temper the supply air in a constant-volume mixed-air system and in a dedicated outdoor-air system.

Supply-Air Tempering in Mixed-Air Systems

Table 4. Design parameters for examples of SA tempering in a mixed-air system (Figures 15–20)

Cooling (ASHRAE 0.4%)		
Outdoor conditions ¹	Sensible	96°F DB (35.7°C) 76°F WB (24.5°C)
	Latent	84°F DB (28.8°C) 76°F DP (24.6°C)
Indoor setpoints	Dry-bulb temperature	74°F DB (23.3°C)
	Relative humidity (max limit)	60% ²
Airflows	Supply	1,500 cfm (0.7 m³/s)
	Outdoor	450 cfm (0.2 m³/s)

¹ Cooling design conditions represent the peak dry-bulb temperature and mean coincident wet-bulb temperature for sensible cooling, and the peak dew point and mean coincident dry-bulb temperature for latent cooling, in Jacksonville, Florida.

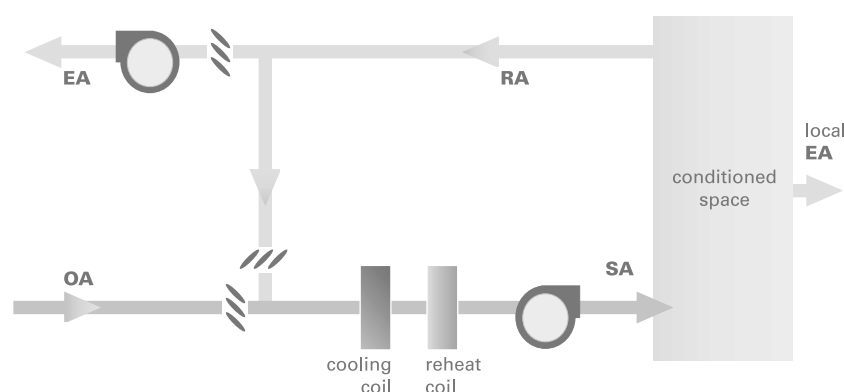
² The maximum limit for indoor humidity control in this example is 60 percent, which is also the upper limit recommended by ASHRAE Standard 62–1999.

To demonstrate the potential savings of tempering supply air with recovered energy, let's examine the mixed-air HVAC system designed for a year-round school in Jacksonville, Florida. See Figure 15. The system consists of a constant-volume air handler that delivers 1,500 cfm (0.7 m³/s) of supply air to a single classroom. Outdoor air for ventilation accounts for 450 cfm (0.2 m³/s) of the supply air.

At the ASHRAE 0.4-percent “sensible-design” condition — that is, peak dry-bulb temperature of 96°F and mean coincident wet-bulb temperature of 76°F (35.7°C DB, 24.5°C WB) — the system delivers 55.7°F (13.2°C) supply air to offset the space sensible load. Table 4 summarizes the thermal conditions and design airflows for this scenario.

At part-load conditions, the constant-volume system responds to the decreasing space sensible load by supplying warmer air to the space. When the relative humidity in the space exceeds the 60-percent limit, modulation of the cooling coil's capacity maintains the space relative humidity below the maximum limit, while modulation of the reheat coil's capacity maintains the dry-bulb temperature at setpoint.

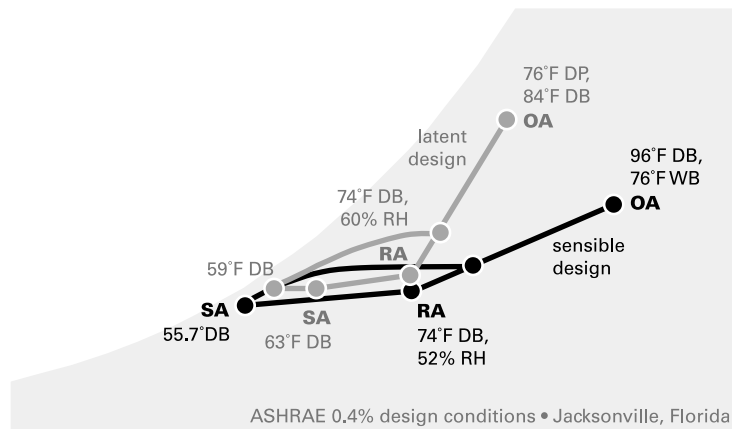
Figure 15. SA tempering in a constant-volume, mixed-air HVAC system



Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 16. Mixed-air HVAC System with SA tempering



Note: A common alternative to modulating the capacity of the cooling coil is to operate it at maximum capacity when dehumidifying. This control strategy quickly lowers space humidity, and then returns the system to the normal cooling mode.

Figure 16 plots the psychrometric conditions of the outdoor air, supply air, and return air at the ASHRAE 0.4-percent design conditions. For Jacksonville, the “latent-design” condition is a peak dew point of 76°F and a mean coincident dry-bulb temperature of 84°F (24.6°C DP, 28.8°C DB). The sensible load in the space is lower at this condition than it is at the sensible-design condition. Therefore, the temperature of the supply air must be warmer — 63°F (17.2°C) in this example — to match the sensible cooling capacity with the sensible load in the space and avoid overcooling. To accomplish this, the cooling coil first dehumidifies the supply air to 59°F (15°C) in order to maintain a relative humidity of 60 percent in the space. The reheat coil then raises the dry-bulb temperature of the supply air to 63°F (17.2°C). Without the use of energy recovery, tempering the supply air requires 6,510 Btu/hr (1.86 kW) of new energy.

Tempering energy needed at latent design:

$$Q_s = 1.085 \times 1,500 \text{ cfm} \times (63^\circ\text{F} - 59^\circ\text{F})$$

$$Q_s = 6,510 \text{ Btu/hr}$$

$$(Q_s = 1.21 \times 0.7 \text{ m}^3/\text{s} \times [17.2^\circ\text{C} - 15^\circ\text{C}])$$

$$(Q_s = 1.86 \text{ kW})$$

Air-to-air recovery for supply-air tempering can be accomplished using either of two configurations: *series* or *parallel*. The next sections compare their respective performance.

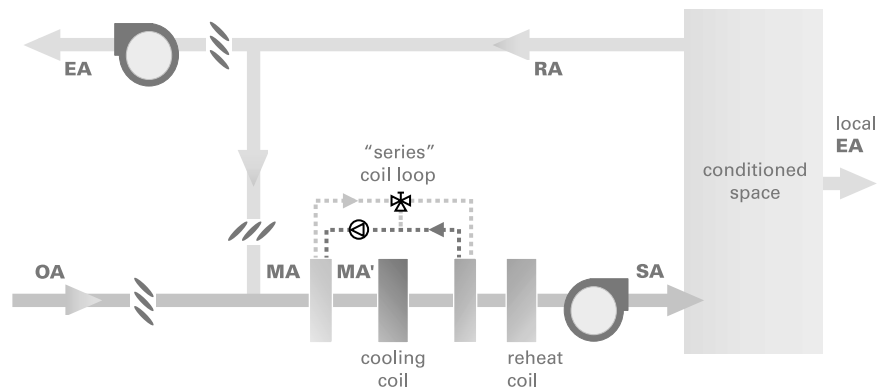
Series Configuration

Figure 17 shows an air-to-air, sensible-energy-recovery device applied in a *series*, or “wraparound,” configuration within a constant-volume, mixed-air system. (Although the example depicts a coil loop, any sensible-energy-recovery device could be used.) The device removes heat from the air upstream of the cooling coil and releases it in the supply air downstream of the coil.

Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 17. SA tempering: "Series" transfer in a constant-volume, mixed-air system

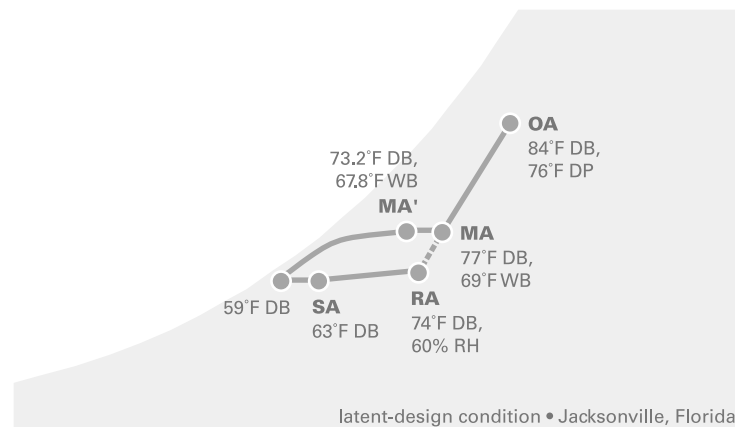


Although it is described as energy "recovery," this arrangement does not "recover" heat; instead, it transfers heat from one location to another within the same air stream.

At the part-load, latent-design condition for our example classroom, mixed air enters the upstream coil of the coil loop at 77°F DB and 69°F WB (25°C DB, 20.6°C WB). The coil loop removes sensible heat from the mixed air, cooling it to 73.2°F DB (22.9°C DB), and then transfers that heat to the other side of the coil loop, downstream of the cooling coil. As shown in Figure 18, the cooling coil must dehumidify the air to 59°F (15°C) in order to keep the relative humidity in the space below the 60-percent limit. The downstream coil of the coil loop then warms the supply air to the desired dry-bulb temperature of 63°F (17.2°C).

Using air-to-air energy recovery not only saves the 6,510 Btu/hr (1.86 kW) of heating energy needed for tempering, but also reduces the load on the cooling

Figure 18. SA tempering: "Series" transfer in a constant-volume, mixed-air system



Temper Supply Air

Applications for Air-to-Air Energy Recovery

Cooling-energy savings at latent design:

$$Q_T = 4.5 \times 1,500 \text{ cfm} \times (33.1 - 32.2 \text{ Btu/lb})$$

$$Q_T = 6,075 \text{ Btu/hr}$$

$$(Q_T = 1.2 \times 0.7 \text{ m}^3/\text{s} \times [77.0 - 74.9 \text{ kJ/kg}])$$

$$(Q_T = 1.76 \text{ kW})$$

coil by precooling the mixed air before it enters the cooling coil. Reducing the enthalpy of the mixed air to 32.2 Btu/lb (74.9 kJ/kg) saves 6,075 Btu/hr (1.76 kW) of cooling energy.

Notice that in a mixed-air system, the use of an air-to-air energy-recovery device in a series configuration does not reduce the installed capacity of the cooling plant. At the sensible-design condition, the 55.7°F (13.2°C) air that is supplied to the space does not require tempering. Without heat transfer, no precooling occurs. Therefore, the cooling coil and cooling plant must be sized to handle the entire cooling load at the sensible-design condition.

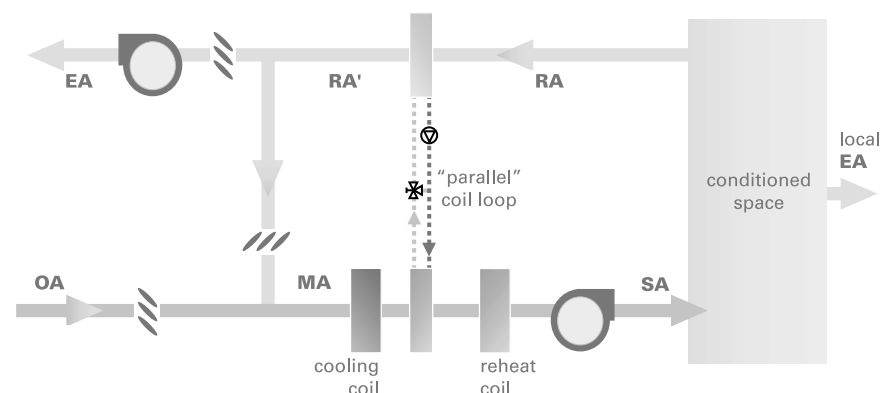
Parallel Configuration

Figure 19 shows the same constant-volume, mixed-air system ... but this time the air-to-air, sensible-energy-recovery device is applied in a parallel configuration. (Again, although this example depicts a coil loop, any sensible-energy-recovery device can be used.) The device recovers sensible heat from the return air stream and releases it in the supply air downstream of the cooling coil, raising its temperature.

At the part-load, latent-design condition, air returns from the space and enters the return-side coil of the coil loop at 74°F DB and 64.5°F WB (23.3°C DB, 18.1°C WB). The coil loop removes sensible heat from the return air stream, cooling it to 70.1°F DB (21.2°C DB), and then transfers this heat to the supply-side coil of the coil loop, downstream of the cooling coil. See Figure 20. At this condition, the air leaving the cooling coil must be 59°F (15°C) to maintain the relative humidity in the space below the 60-percent limit. The supply-side coil of the coil loop then heats the supply air to the desired dry-bulb temperature of 63°F (17.2°C).

Like the series configuration, the parallel configuration also reduces the load on the cooling coil while saving the 6,510 Btu/hr (1.86 kW) of heating energy

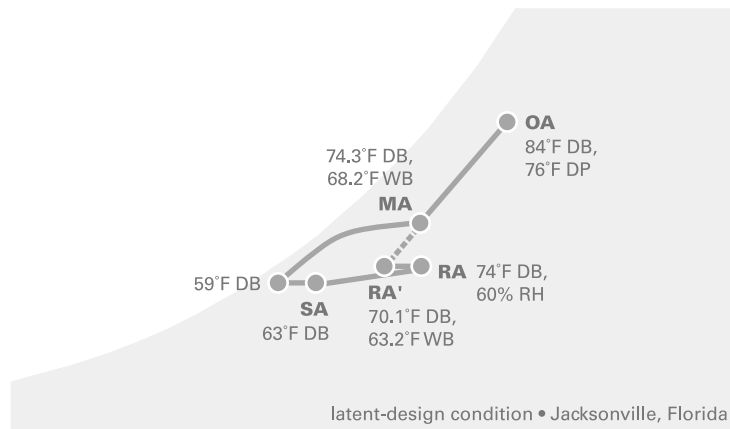
Figure 19. SA tempering: "Parallel" recovery in a constant-volume, mixed-air system



Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 20. SA tempering: "Parallel" recovery in a constant-volume, mixed-air system



Cooling-energy savings at latent design:

$$Q_T = 4.5 \times 1,500 \text{ cfm} \times (33.1 - 32.5 \text{ Btu/lb})$$

$$Q_T = 4,050 \text{ Btu/hr}$$

$$(Q_T = 1.2 \times 0.7 \text{ m}^3/\text{s} \times [77.0 - 75.6 \text{ kJ/kg}])$$

$$(Q_T = 1.18 \text{ kW})$$

needed for tempering. By removing sensible heat from the return air, the coil loop effectively lowers the enthalpy of the mixed air from 33.1 Btu/lb to 32.5 Btu/lb (77.0 kJ/kg to 75.6 kJ/kg). The resulting savings is 4,050 Btu/hr (1.18 kW) of cooling energy.

Applying an air-to-air energy recovery device in a parallel (rather than series) arrangement within a mixed-air system still does not reduce the installed capacity of the cooling plant. The supply air does not require tempering at the sensible-design condition, so no precooling occurs. Both the cooling coil and the cooling plant must therefore be sized to handle the entire cooling load at the sensible-design condition.

Series versus Parallel

In mixed-air systems, both series and parallel configurations reduce the heating energy required for tempering. Although the series configuration saves more cooling energy than the parallel configuration, neither permits downsizing of the cooling or heating plant.

With the series configuration, outdoor conditions largely determine the condition of the mixed air that enters the energy-recovery device. As a result, the amount of heat available for transfer varies, but can be larger than the constant quantity of heat that the parallel configuration recovers from the return air stream. In other words, the series configuration benefits when the outdoor air is warmer than the return air; the parallel configuration benefits when the outdoor air is cooler than the return air.

Three criteria ultimately determine which energy-recovery arrangement is the right choice for a given project...



Temper Supply Air

Applications for Air-to-Air Energy Recovery

- First cost
- Savings in cooling and heating energy
- Additional fan energy to overcome the static pressure drop of the energy-recovery device

Supply-Air Tempering in Dedicated Outdoor-Air Systems

Dedicated outdoor-air systems should dehumidify the outdoor air to a low dew point, typically 45°F to 55°F DP (7.2°C to 12.8°C), before tempering the air to a “neutral” dry-bulb temperature. This allows the dedicated outdoor-air system to offset the space latent loads as well as the ventilation load. Dehumidifying to a low dew point provides two benefits:

- Smaller space loads, which permits downsizing of the constant-volume terminal units
- Virtually no condensation in the space (The cooling coil in each terminal operates dry because it handles only the sensible load.)

A forthcoming Trane applications engineering manual, *Dehumidification in HVAC Systems* (SYS-APM004-EN), will also address dedicated outdoor-air systems in detail. Publication is scheduled for June 2002. Check the Trane bookstore at www.trane.com/bookstore for availability. ■

Dedicated outdoor-air systems that are designed to deliver dry (low dew-point), “neutral-temperature,” conditioned air also need tempering and can benefit by using air-to-air sensible-energy recovery to provide it. Systems of this type typically deliver the conditioned outdoor air to ceiling plenums (when used with fan-coils, for example) or directly to occupied spaces (as in water-source heat pump applications, for example). Sensible-energy recovery may decrease the first cost and operating costs of the system when compared to using new energy for tempering. It may also enable compliance with energy standards and local codes.

To demonstrate the potential for energy savings, let’s revisit the example classroom in Jacksonville, Florida. This time, a dedicated outdoor-air handler (or makeup-air unit) delivers conditioned outdoor air, CA, directly to several classrooms at a “neutral,” dry condition of 71°F DB, 53°F DP (21.7°C DB, 11.7°C DP). It conditions 450 cfm (0.2 m³/s) of outdoor air for each classroom. The capacity of the cooling coil modulates to maintain the conditioned-air dew point at setpoint; similarly, the capacity of the reheat coil modulates to maintain the conditioned-air dry-bulb temperature at setpoint. Separate, constant-volume terminals, one in each classroom, handle the space load. See Figure 21.

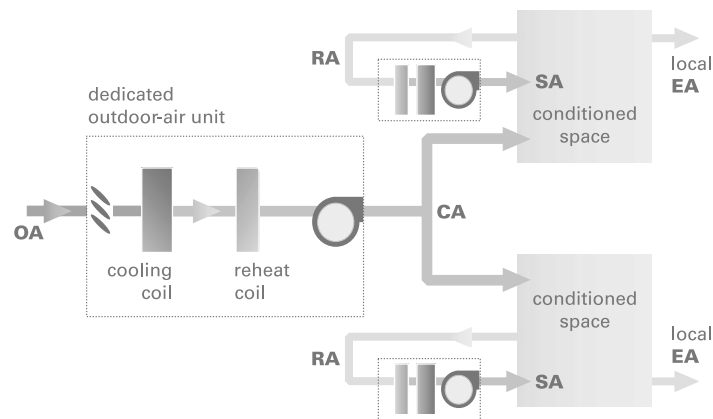
Note: An alternative to maintaining a fixed dew point is to reset the conditioned-air dew point to maintain the humidity in the “worst-case” space below the maximum limit. In other words, the capacity of the cooling coil modulates to make the conditioned outdoor air dry enough to keep the most humid room below 60-percent relative humidity.

For Jacksonville, the ASHRAE 0.4-percent peak-enthalpy design condition—that is, 42.4 Btu/lb (98.6 kJ/kg)—represents a peak wet-bulb temperature of 79°F (26.2°C WB) and a mean coincident dry-bulb temperature of 91°F (32.6°C DB). In a dedicated outdoor-air system that maintains a constant conditioned-air dew point, this is the condition at which the cooling-coil load is the highest because the enthalpy of the outdoor air is the highest.

Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 21. SA tempering in a dedicated outdoor-air system



Energy required for OA dehumidification:

$$Q_T = 4.5 \times 450 \text{ cfm} \times (42.4 - 22.1 \text{ Btu/lb})$$

$$Q_T = 41,108 \text{ Btu/hr}$$

$$(Q_T = 1.2 \times 0.2 \text{ m}^3/\text{s} \times [98.6 - 51.4 \text{ kJ/kg}])$$

$$(Q_T = 11.33 \text{ kW})$$

Energy required for tempering:

$$Q_s = 1.085 \times 450 \text{ cfm} \times (71.0^\circ\text{F} - 53.5^\circ\text{F})$$

$$Q_s = 8,544 \text{ Btu/hr}$$

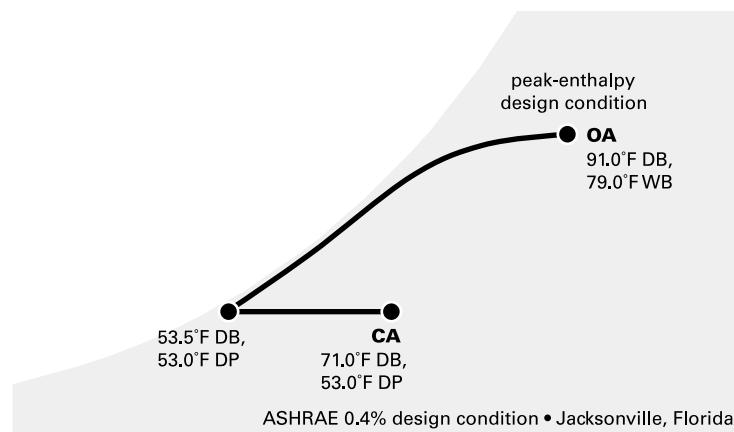
$$(Q_s = 1.21 \times 0.2 \text{ m}^3/\text{s} \times [21.7^\circ\text{C} - 11.9^\circ\text{C}])$$

$$(Q_s = 2.37 \text{ kW})$$

The cooling coil dehumidifies the outdoor air to 53.5°F DB, 53°F DP (11.9°C DB, 11.7°C DP), which corresponds to an enthalpy of 22.1 Btu/lb (51.4 kJ/kg). See Figure 22. The reheat coil then warms this air to the “neutral” dry-bulb temperature of 71°F (21.7°C). Without the use of energy recovery, the system would consume 41,108 Btu/hr (11.33 kW) of new energy for dehumidification and 8,544 Btu/hr (2.37 kW) for tempering.

Now, let’s examine the impact of using an air-to-air sensible-energy-recovery device to provide the heat for tempering. As for constant-volume, mixed-air systems, sensible-energy recovery can be applied in either of two configurations to temper the conditioned air in a dedicated outdoor-air system: *series* and *parallel*.

Figure 22. Dedicated outdoor-air system with SA tempering



Applications for Air-to-Air Energy Recovery

Series Configuration

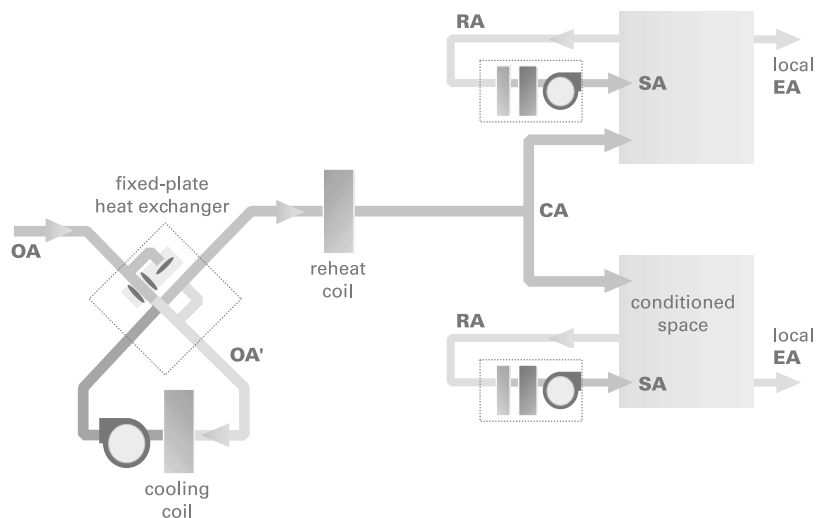
Figure 23 shows an air-to-air, sensible-energy-recovery device that is applied in a series, or “wraparound,” configuration within a dedicated outdoor-air system. (Although the example depicts a fixed-plate heat exchanger, any sensible-energy-recovery device could be used.) The device removes sensible heat from the air upstream of the cooling coil and releases it in the air downstream of that coil. Rather than “recover” heat, the series arrangement transfers heat from one location to another within the same air stream.

At the peak-enthalpy condition for our example classroom, the upstream side of the fixed-plate heat exchanger removes heat from the entering outdoor air, cooling it to 78.4°F DB, 74.6°F WB (25.8°C DB, 23.7°C WB). Because the air passing through the heat exchanger becomes saturated (Figure 24), that device removes both sensible heat and moisture from the outdoor air stream. The latent heat of condensation that is generated as moisture condenses from the air adds to the overall amount of heat transferred; however, only sensible heat—not moisture—moves to the other side of the heat exchanger.

The cooling coil dehumidifies the air to a dew point of 53°F (11.7°C) and all of the heat (sensible and latent) removed from the outdoor air upstream of the cooling coil is transferred (as sensible heat only) to temper the air downstream of that coil. The resulting dry-bulb temperature of the supply air is 71°F (21.7°C).

Using the air-to-air energy-recovery device as described here not only saves the 8,544 Btu/hr (2.37 kW) of heating energy needed for tempering, but also decreases the load on the cooling coil by precooling the outdoor air upstream

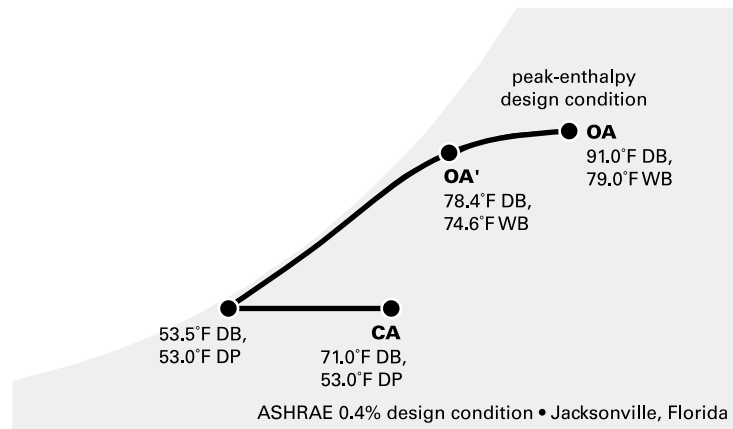
Figure 23. SA tempering: “Series” transfer in a dedicated outdoor-air system



Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 24. SA tempering: “Series” transfer in a dedicated outdoor-air system



Cooling-energy savings:

$$Q_T = 4.5 \times 450 \text{ cfm} \times (42.4 - 38.2 \text{ Btu/lb})$$

$$Q_T = 8,544 \text{ Btu/hr}$$

$$(Q_T = 1.2 \times 0.2 \text{ m}^3/\text{s} \times [98.6 - 88.9 \text{ kJ/kg}])$$

$$(Q_T = 2.33 \text{ kW})$$

of the coil. Reducing the enthalpy of the outdoor air to 38.2 Btu/lb (88.9 kJ/kg) saves 8,544 Btu/hr (2.33 kW) of cooling energy.

Unlike its application in a mixed-air system, series transfer of sensible energy in a dedicated outdoor-air system enables a reduction in the installed capacity of the cooling plant. By precooling the outdoor air, the energy-recovery device lessens the load on the cooling coil — in this case, by 8,544 Btu/hr (2.33 kW). Of course, delivering the air at a “neutral” condition means that tempering is required whenever the cooling coil is dehumidifying, even at design conditions.

When it is cold outside and the dedicated outdoor-air system must heat the entering outdoor air, the cooling coil is turned off. At such times, a series-configured, air-to-air energy-recovery device provides no benefit.

Parallel Configuration

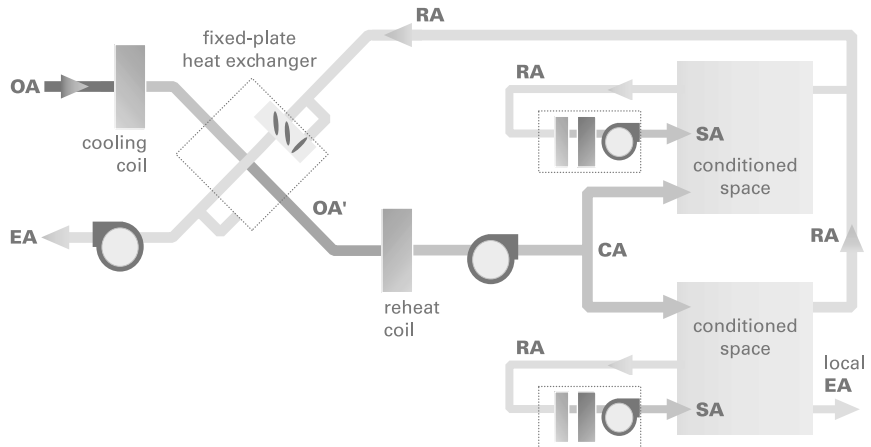
Figure 25, p. 30, shows the same dedicated outdoor-air system ... but this time the air-to-air sensible-energy-recovery device is applied in a parallel configuration. (Again, although a fixed-plate heat exchanger is shown in this example, any sensible-energy-recovery device can be used.) The device recovers sensible heat from the exhaust air stream and releases it downstream of the cooling coil to temper the conditioned outdoor air.

At the peak-enthalpy condition for this same classroom example, the outdoor air is cooled and dehumidified to 53°F DP (11.7°C DP) by the cooling coil. The fixed-plate heat exchanger recovers heat from the exhaust air stream and uses it to temper the air downstream of the cooling coil, warming it to 66.9°F DB (19.4°C DB). See Figure 26 on p. 30. The supplemental reheat coil further warms the conditioned air to the desired 71°F DB (21.7°C DB).

Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 25. SA tempering: “Parallel” recovery in a dedicated outdoor-air system



Reheat-energy savings:

$$Q_s = 1.085 \times 450 \text{ cfm} \times (66.9^\circ\text{F} - 53.5^\circ\text{F})$$

$$Q_s = 6,543 \text{ Btu/hr}$$

$$(Q_s = 1.21 \times 0.2 \text{ m}^3/\text{s} \times [19.4^\circ\text{C} - 11.9^\circ\text{C}])$$

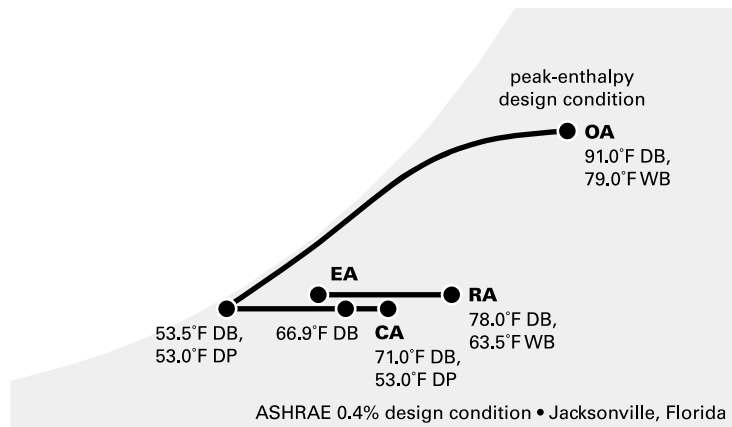
$$(Q_s = 1.82 \text{ kW})$$

The air-to-air energy recovery device in the parallel configuration only saves 6,543 Btu/hr (1.82 kW) of heating energy needed for tempering at this condition. This is because the temperature of the air entering the warm side of the heat exchanger is cooler (78°F versus 91°F [25.6°C versus 32.6°C]) than in the series configuration.

Also, unlike the series configuration, the parallel configuration neither reduces the load on the cooling coil, nor permits a smaller cooling plant. Why? Because the heat that is recovered from the exhaust air stream was going to be rejected from the building anyway. Cooling the exhaust air stream does not reduce the load on the cooling coil.

The parallel configuration differs from the series configuration in another important way. When it is cold outside and the dedicated outdoor-air system

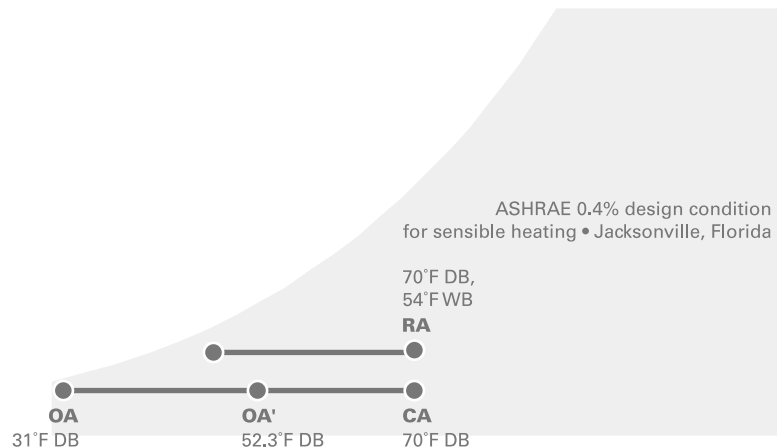
Figure 26. SA tempering: “Parallel” recovery in a dedicated outdoor-air system



Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 27. SA tempering: “Parallel” recovery in a dedicated outdoor-air system



Heating-energy savings:

$$Q_s = 1.085 \times 450 \text{ cfm} \times (52.3^\circ\text{F} - 31^\circ\text{F})$$

$$Q_s = 10,400 \text{ Btu/hr}$$

$$(Q_s = 1.21 \times 0.2 \text{ m}^3/\text{s} \times [11.3^\circ\text{C} - (-0.8^\circ\text{C})])$$

$$(Q_s = 2.93 \text{ kW})$$

must heat the entering outdoor air, the parallel configuration lets the energy-recovery device use heat from the exhaust air stream to warm the entering outdoor air. To demonstrate, consider the winter design condition, which is 31°F DB (−0.8°C DB), for the example classroom in Jacksonville. The heat exchanger removes sensible heat from the 70°F (21.1°C) exhaust air and uses it to warm the entering outdoor air to 52.3°F (11.3°C); see Figure 27. The main heating coil further heats the air to the desired conditioned-air temperature. In this scenario, the parallel configuration saves 10,400 Btu/hr (2.93 kW) of heating energy, cutting operating costs and allowing the heating plant to be downsized.

Applying an air-to-air energy-recovery device in a parallel configuration lets it temper supply air *and* precondition entering outdoor air, although it cannot perform both functions simultaneously. The device tempers the conditioned outdoor air whenever dehumidification is required and preconditions the outdoor air during the heating season.

Note: In cold climates, some method of frost prevention may be required. Also, if a chilled-water cooling coil is used, some method to protect that coil from freezing may be required because the energy-recovery device adds heat downstream of the cooling coil.

Series versus Parallel

In dedicated outdoor-air systems that deliver neutral-temperature air, both series and parallel configurations reduce the new energy that is required for tempering. When it is hot outside, the series configuration can transfer more heat than the parallel arrangement. However, because it is less dependent on outside conditions—and because the return-air condition is relatively stable—the parallel configuration can recover a nearly constant amount heat.



Temper Supply Air

Applications for Air-to-Air Energy Recovery

Because it warms the ventilation (outdoor) air, a dedicated OA system that delivers neutral air requires more cooling-plant capacity than a comparable system that supplies cold air. Although series-configured energy recovery in a neutral conditioned-air system permits the cooling plant to be downsized with respect to a system with no heat recovery, the cooling plant will still be larger than for a dedicated outdoor-air system that supplies cold air. ■

The series configuration saves cooling energy, even allowing the cooling plant to be downsized. By contrast, the parallel configuration can be used in the heating mode, which saves heating energy and allows the heating plant to be downsized.

The parallel configuration requires that most of the building exhaust be routed back to a central location, while the series configuration does not require this additional expense. This additional expense drives most applications to use the series configuration, unless the system includes an additional energy-recovery device to perform outdoor-air preconditioning. An example of this “dual-function” energy-recovery system is discussed on pp. 84–88.

As in mixed-air applications, three criteria ultimately determine which energy-recovery arrangement — series or parallel — is the right choice for a given project:

- First cost
- Savings in cooling and heating energy
- Additional fan energy to overcome the static pressure drop of the energy-recovery device

Note: One innovative design combines both configurations using a coil loop with three coils. When the cooling coil is active, the coil loop operates in the series configuration to save both cooling energy and reheat energy. When the cooling coil is off and the heating coil is active, the coil loop switches and operates in the parallel configuration to save heating energy. For further discussion of this concept, see pp. 42–43.

Effect on First Cost

The previous examples demonstrate the effects of tempering supply air with energy recovered from the exhaust air stream or from upstream of the cooling coil. Energy savings must be balanced with the cost of adding the energy-recovery device to the air-handling system. Depending on the size and complexity of the system, the initial investment may be significantly higher.

Heating and cooling-plant downsizing. Table 5 summarizes the plant-capacity reductions obtained in the preceding examples of supply-air tempering. When used to temper supply air in a mixed-air system, air-to-air energy recovery does not reduce the capacity of either the cooling plant or the heating plant.

Using air-to-air energy recovery may permit a smaller cooling or heating plant in dedicated outdoor-air systems that deliver neutral air. If applied in the series configuration, the cooling plant can be downsized because the energy-recovery device precools the air upstream of the cooling coil whenever the cooling coil is dehumidifying — even at design conditions. If the air-to-air energy-recovery

Temper Supply Air

Applications for Air-to-Air Energy Recovery

Table 5. Capacity reductions attributable to supply-air tempering

Reduction in Mechanical-Plant Capacity		Energy-Recovery Configuration	
		Series	Parallel
Mixed-air system	Cooling	—	—
	Heating	—	—
Dedicated OA system (neutral air)	Cooling	8,544 Btu/hr (2.33 kW)	—
	Heating	—	10,400 Btu/hr (2.93 kW)

device is applied in a parallel configuration, the heating plant can be downsized. During the heating mode, the energy-recovery device recovers sensible heat from the exhaust air stream and uses it to warm the entering outdoor air.

Of course, when investigating the economic benefit of air-to-air energy recovery for tempering supply air, you must also consider its impact on the first cost of other system components.

Increased fan motor sizes. As noted earlier in this chapter, adding an air-to-air energy-recovery device to the air stream increases the static pressure loss in the air-distribution system. The configuration of the energy-recovery device and the fans will determine the extent of the pressure loss and the resulting fan-power requirements. Providing the additional power necessary may require larger fan motors.

Additional exhaust ductwork. If applied in a parallel configuration to temper conditioned outdoor air, most energy-recovery devices used in dedicated outdoor-air systems require adjacent outdoor- and exhaust-air streams. This prerequisite means that most of the exhaust air must be routed back to a central location, and, therefore, typically requires more ductwork than the same system without energy recovery.

This generalization does not apply to coil loops. A coil loop can recover heat from one or several exhaust air streams, none of which is adjacent to each other nor to the outdoor air stream. Of course, when the two air streams are far apart, the coil loop requires more piping and glycol as well as a larger pump and expansion tank.

Effect on Operating Costs

Less energy consumed for heating and cooling. Using air-to-air sensible-energy recovery can reduce energy consumption associated with supply-air tempering. Given the complex interaction between system control modes and the humidity in the space, simulation software is often required to estimate the performance of various energy-recovery devices. Such an analysis can predict the length of the payback period based on the building location, type of

Temper Supply Air

Applications for Air-to-Air Energy Recovery

building, type of HVAC system, system control modes, quantity of outdoor air, airside economizer operation, and local utility rates.

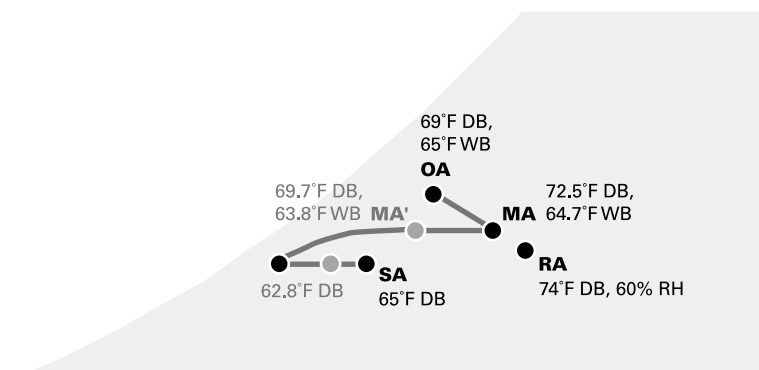
Need for supplemental heat. Applications that temper supply air using recovered energy may require a supplemental heating coil downstream of the energy-recovery device to achieve the desired dry-bulb temperature at all conditions. The example in Figure 25 (p. 30) illustrates one such application: a dedicated outdoor-air system with a parallel-configured energy-recovery device. The same example system, using the series configuration, would probably also need a supplemental heating coil to maintain the desired dry-bulb temperature at less severe outdoor conditions such as a mild, rainy day.

Let's revisit another example — this time, the mixed-air system that applies a coil loop in a series configuration (Figure 17, p. 23). As depicted in Figure 28, the condition of the outdoor air is 69°F DB, 65°F WB (20.6°C DB, 18.3°C WB). To satisfy the sensible load in the space *and* maintain the relative humidity below 60 percent, the supply air must be delivered at a dew point of 57.5°F (14.2°C). Furthermore, the dry-bulb temperature can be no colder than 65°F (18.3°C) to avoid overcooling the space.

Operating at full capacity, the coil loop transfers heat from the mixed air upstream of the cooling coil to the air downstream of the coil, warming it to 62.8°F DB (17.1°C DB). Despite operating at full capacity, the coil loop cannot transfer enough heat to achieve the desired supply-air temperature of 65°F DB (18.3°C DB). To make up the shortfall, a supplemental heating coil is required downstream of the coil loop. Most likely, this coil will also operate during the heating mode when a heating load exists and the coil loop is turned off.

Recall that in mixed-air systems, the amount of heat that can be transferred from the return air stream is more constant for parallel configurations of energy recovery than it is for series configurations. Recall, too, that the parallel configuration is less dependent on ambient conditions. When the outdoor air is warmer than the return air, the series configuration has more heat to transfer

Figure 28. Mixed-air system with “series” heat transfer and supplemental heat



Temper Supply Air

Applications for Air-to-Air Energy Recovery

than the parallel configuration. When the outdoor air is cooler than the return air, the series configuration has less heat to transfer than the parallel configuration.

The relationship between ambient conditions, energy-recovery configuration, and available heat for transfer also exists in dedicated outdoor-air systems. A device in the series configuration can transfer more heat when it is hot outside — but less heat when the outdoor air is cooler than the exhaust air — than a parallel-configured device. The parallel configuration is less dependent on outdoor conditions because the temperature of the exhaust air stream remains relatively constant.

More energy consumed by fans. As noted earlier, the static pressure drop of the system will increase — causing the fans to consume more energy — whenever air passes through the air-to-air energy-recovery device. The amount of additional energy consumed depends on the static pressure drop through the device. The operating cost savings provided by recovered energy must exceed the increased cost of operating the fans in order to justify the operational benefit of the energy-recovery device. A year-round, project-specific analysis must be performed to determine the effect on the energy consumption of the entire HVAC system.

Bypass dampers may be used to reduce the airside pressure loss when the energy-recovery device is idle, thus reducing fan energy consumption. They are particularly valuable when the energy-recovery device is applied in a series configuration. Whenever the cooling coil is turned off, the bypass dampers can save substantial fan energy.

Need for capacity control. An air-to-air energy-recovery device used for supply-air tempering generally requires some method for modulating the capacity of the device. Modulation prevents the device from overheating the supply air during mild weather.

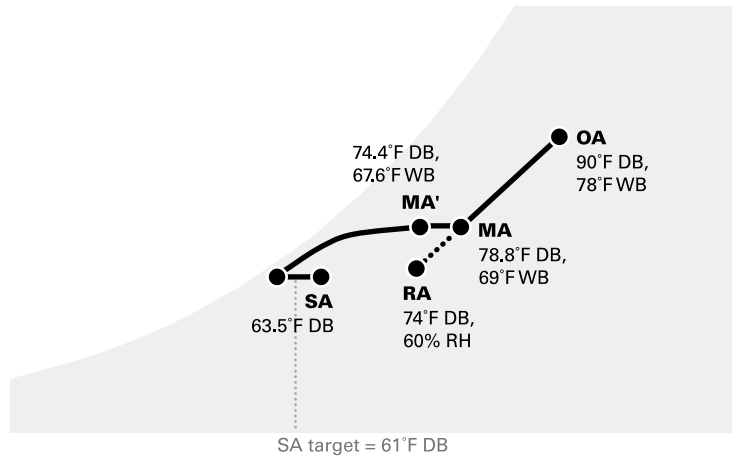
To illustrate, consider the mixed-air system represented psychrometrically in Figure 29 on p. 36. The condition of the outdoor air is 90°F DB, 78°C WB (32.2°C DB, 25.6°C WB). To satisfy the sensible load without overcooling the space, the supply air must be delivered at a dry-bulb temperature of 61°F (16.1°C). Operating at full capacity, the “series” coil loop cools the mixed air to 74.4°F DB (23.6°C DB). The heat removed from the mixed air upstream of the cooling coil is transferred to the air downstream of the coil, warming the supply air to 63.5°F DB (17.5°C DB). Letting the coil loop operate at full capacity (run “wild”) overheats the supply air, causing the space to be warmer than desired. Modulating the capacity of the coil loop avoids this situation.

Note: Tempering is not needed when the relative humidity in the space is lower than the maximum limit. To prevent heat transfer, the coil-loop pump should be turned off.

Temper Supply Air

Applications for Air-to-Air Energy Recovery

Figure 29. Unmodulated energy-recovery capacity for SA tempering (mixed-air system)



A mixed-air system that tempers supply air to control space humidity generally requires very little reheat when it is warm outside and lots of reheat when the weather is cool and damp (that is, at part load). The capacity of the energy-recovery device, whether arranged in a series or parallel configuration, must be modulated to avoid transferring too much heat.

In a dedicated outdoor-air system that delivers “neutral” air at constant dew-point and dry-bulb temperatures, the condition of the air leaving the cooling coil will be the same whenever the coil operates. Because this dehumidified outdoor air is then reheated to the same dry-bulb temperature, the required amount of tempering (reheat) is also constant whenever the energy-recovery device operates. In the *parallel* configuration, the source of heat to be recovered (the exhaust air stream) is relatively consistent, and capacity control for the energy-recovery device may not be required. In the *series* configuration, however, the source of heat to be transferred (the outdoor air stream) definitely fluctuates. Therefore, some method of controlling the capacity of the energy-recovery device will probably be needed to avoid overheating the conditioned outdoor air at certain operating conditions.

The soon-to-be-published Trane application manual, *Dehumidification in HVAC Systems* (SYS-APM004-EN), discusses dedicated OA systems in detail, including resetting leaving air conditions to optimize system energy consumption. Publication is scheduled for June 2002. Check the Trane bookstore at www.trane.com/bookstore for availability. ■

Further, if the setpoints (either dew point, dry bulb, or both) of the air leaving the dedicated outdoor-air unit are automatically reset based on actual space conditions, a method of capacity control is required to optimize the energy use of the HVAC system. Capacity control is needed whether the energy-recovery device is arranged in the parallel or series configuration.

“Energy-Recovery Technologies,” pp. 38–59, discusses specific capacity control methods for various air-to-air energy-recovery devices.

Temper Supply Air

Applications for Air-to-Air Energy Recovery

Other Considerations

Several other issues must be addressed to optimize the life-cycle cost of a mixed-air or dedicated outdoor-air system that tempers supply air using air-to-air energy recovery.

Frost prevention. Although frost prevention is typically required for outdoor-air preconditioning applications, it is seldom necessary in supply-air tempering applications.

As noted earlier, when a parallel-configured energy-recovery device is applied in a dedicated outdoor-air system, frost prevention may be required in cold climates if the energy-recovery device is also used to precondition the outdoor air during the heating season.

"Air-to-Air Energy Recovery Technologies" (pp. 38–59) discusses specific frost prevention methods for various air-to-air energy-recovery devices.

Acceptable cross-leakage. Many types of air-to-air energy-recovery devices permit some degree of cross-leakage. This is seldom of concern in mixed-air systems because most of the air that returns from the space is recirculated as supply air. Cross-leakage is also a non-issue when the series configuration is used because both sides of the energy-recovery device are located in the same air stream.

Only dedicated outdoor-air systems that apply air-to-air energy recovery in a parallel configuration require special attention. Leakage between the two air streams, however slight, may transfer exhaust-air contaminants to the conditioned outdoor air stream. ("Energy-Recovery Technologies," pp. 38–59, discusses specific methods of reducing cross-leakage for various types of air-to-air energy-recovery devices.) ■