Adding More Fan Power Can Be a Good Thing

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As the HVAC industry strives to reduce commercial building energy use, fan power often comes up as a focal point to optimize building performance. However, the goal of minimizing fan energy can be contrary to other methods used to reduce building energy. One of these goals is to reduce heating and cooling loads introduced by ventilation air. A building's heating and cooling load can be lowered by recovering energy to and from the building exhaust air with an energy recovery exchanger (*Figure 1*). This energy exchange comes with an energy cost, static pressure loss across the exchanger that can add to the power required both by the fans supplying conditioned air to the space and the fans exhausting air. The added fan power can be significant and may lead one to ponder: is the cooling or heating recovered really worth it?

How Much Recovery vs. Pressure Loss is Typical?

Figure 2 shows examples of ratios of static pressure loss per air path compared to energy recovery effectiveness for sensible-only crossflow plate exchangers and total (sensible and latent) recovery wheels. This data is a sample of 5,000 cfm (2360 L/s) energy recovery exchangers located in equipment with cooling coils sized for a face velocity of 500 fpm (2.54 m/s). The data is typical of what one would find available for commercial equipment. The cooling coils are what dictate the size of the air-handling unit, and the exchangers are selected to fit within this size constraint. As with most heat transfer equipment, one could increase the size of the air-handling equipment to get a lower pressure drop vs. effectiveness ratio; the data presented is meant to be the worst-case scenario. The amount of exhaust air available also will affect how much energy is recovered. The chart is based on an exhaust air to ventilation air ratio of 0.90 (exhaust air is 10% less than the ventilation air to pressurize the building).

Effectiveness is a term used to quantify an energy recovery components' ability to recover energy and is defined by ASHRAE Standard 84-2013, *Method of Testing Air-To-Air Heat/Energy Exchangers,* as "the actual energy transfer (sensible, latent, or total) divided by the product of the minimum energy capacity rate and the maximum difference in temperature, humidity ratio, or enthalpy."

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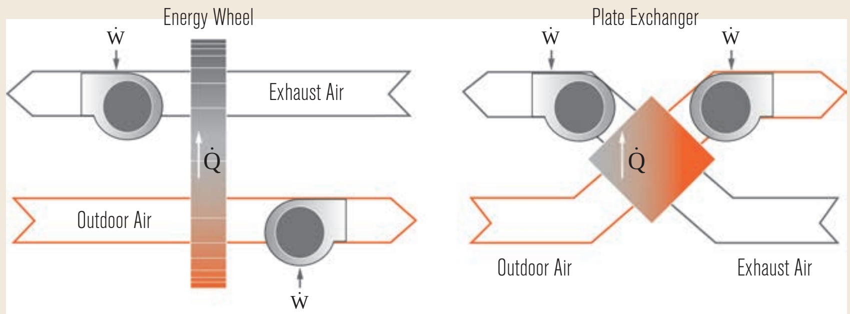
An easier way to understand it is in equation form.

$$\text{Effectiveness, } \varepsilon = \frac{\text{Heat Rate}}{\text{Thermodynamic Max. Heat Rate}}$$

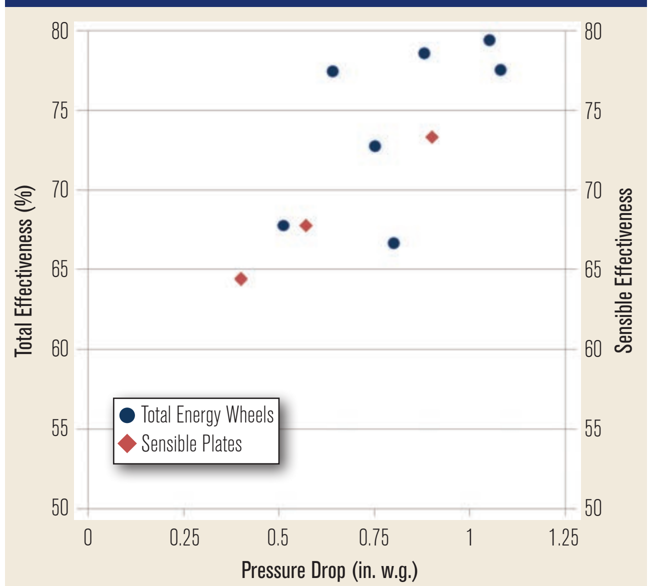
Therefore, if the ratio of exhaust air to outdoor air is 0.90, the theoretical best exchanger of 100% effectiveness would reduce the ventilation heating or cooling load by 90%.

FIGURE 1 Energy exchanged between airstreams for more work by the fans.



The data is somewhat scattered, and it is not all inclusive, but it does include data from multiple manufacturers of exchangers. Although the pressure drop vs. the effectiveness is a function of design trade-offs made by each exchanger manufacturer, one can see there is a relationship of effectiveness and pressure drop for a fixed air velocity. In general, a higher effectiveness will result in higher pressure loss (*Figure 2*).

FIGURE 2 Sample effectiveness vs. pressure drop.



How Much Energy Recovery Is Required?

The prescriptive path of ASHRAE/IES Standard 90.1-2010 in Section 6.5.6.1, requires the cooling load and/or heating load added by the outdoor air to be reduced by 50% using energy recovery. However, the standard does not use clear language, it states: "Energy recovery systems required by this section shall have at least 50% energy 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 enthalpies at design conditions."

What can be confusing is that it redefines the effectiveness term as the amount the outdoor air cooling and/or heating load is reduced vs. defining the effectiveness of the exchanger. *Figure 3* redraws the effectiveness chart with respect to the amount that the outdoor air cooling and heating load is reduced. These typical exchangers meet the requirement although using a sensible plate exchanger would require a heating-only application or that it be used in an arid climate where at design cooling the outdoor air cooling load is mostly sensible heat, rather than the 95°F/78°F (35°C/26°C) high latent heat condition shown.

What Is an Acceptable Amount of Pressure Loss?

The Standard 90.1-2010 indirectly specifies the acceptable pressure drop per airstream versus the amount of energy recovered. The fan power limitation in the prescriptive path of the standard (Section 6.5.3.1) gives a fan power credit when energy recovery is used. This is not a limit on the pressure drop for the energy recovery device. An energy recovery component could exceed this credit, but if following the prescriptive path, one may have to reduce pressure loss somewhere else in the system to make up for an exchanger that exceeds this credit.

For an energy recovery device, other than a coil run-around loop, the pressure drop credit is defined as the following: "(2.2 × Energy Recovery Effectiveness) – 0.5 in w.c. for each airstream." The effectiveness term here is the same as defined by Section 6.5.6 of Standard 90.1-2010, and is based on the change in outdoor air enthalpy, outdoor air heating, and/or cooling load as plotted on a psychrometric chart in *Figure 4*. *Figure 5* plots the fan power credit limit for energy recovery vs. the amount of energy recovered. The prescribed energy recovery requirement by Standard 90.1 is achievable within the fan power credit given and for the amount of energy recovery prescribed. But is it worth the fan power added? An additional metric must be used to answer this question.

Recovery Efficiency Ratio

A metric to measure the efficiency of an exhaust air energy recovery exchanger is recovery efficiency ratio (RER). RER as defined in ASHRAE Standard 84-2013 is the "ratio of energy recovered divided by the energy expended in the recovery process." This ratio can be represented in terms of Btu/h·W for cooling and W/W for heating. These are the familiar units that are used to state cooling and heating efficiency.

Looking at the performance this way can give a comparable number to see if, at a specific operating point, it is more efficient to recover the energy to condition the outdoor air or use the mechanical heating and cooling equipment. Most of the additional energy required when adding an energy recovery system is fan energy, primarily from static pressure loss. Energy recovery wheels also will have a fractional horsepower motor used to rotate the wheel. Most of the motors used are less than 0.5 hp (0.4 kW). Fixed plate heat exchangers have an airtight seal between the two airstreams. However, since energy wheels rotate, they will not have an airtight seal and will leak air to the lower pressure side. This may require extra air to be handled by the fans and extra fan power.

The relationship between energy recovery and pressure loss specified in the fan power limitation credit in ASHRAE Standard 90.1 is a realistic relationship that one can expect for a wheel or an air-to-air plate heat exchanger. This relationship between energy recovered and pressure drop defined in Standard 90.1 can be used to calculate an RER for summer cooling and winter heating, given certain conditions.

Figure 6 shows the exchangers in the same sample set as in the earlier examples for a design cooling day with 95°F/78°F (35°C/26°C) outdoor air dry-bulb and wet-bulb temperature and an exhaust air condition of 75°F (24°C) dry-bulb temperature with a relative humidity of 55%.

FIGURE 3 Ventilation cooling load reduction at design.

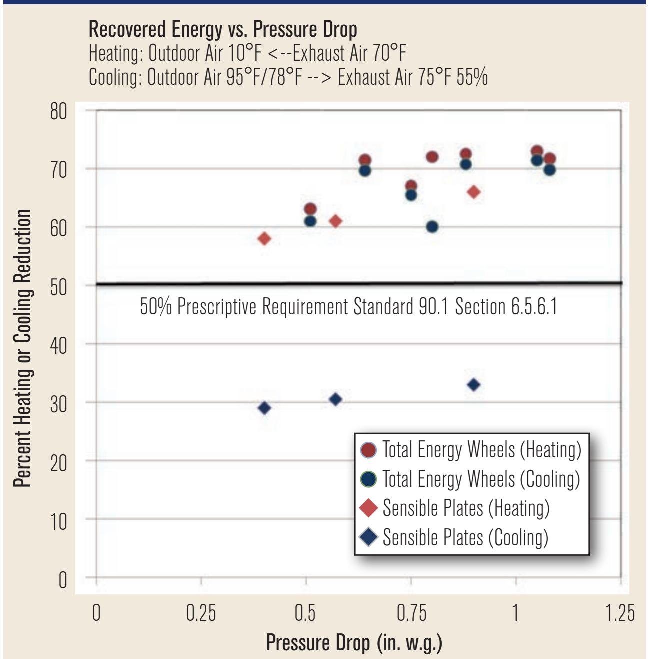


FIGURE 4 Psychrometric plot of ventilation loads.

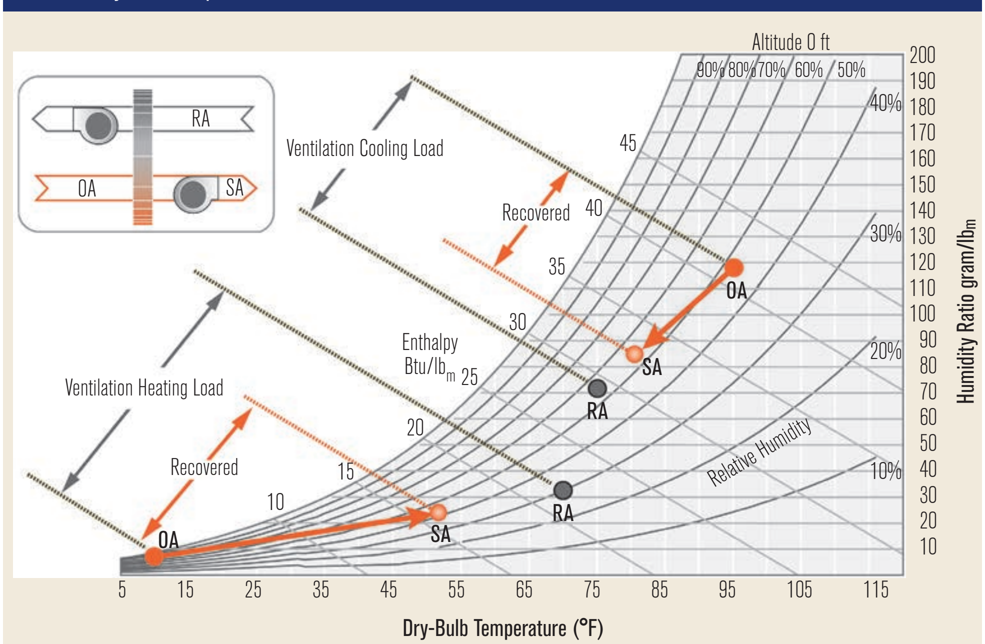


FIGURE 5 Reduction of ventilation peak loads.

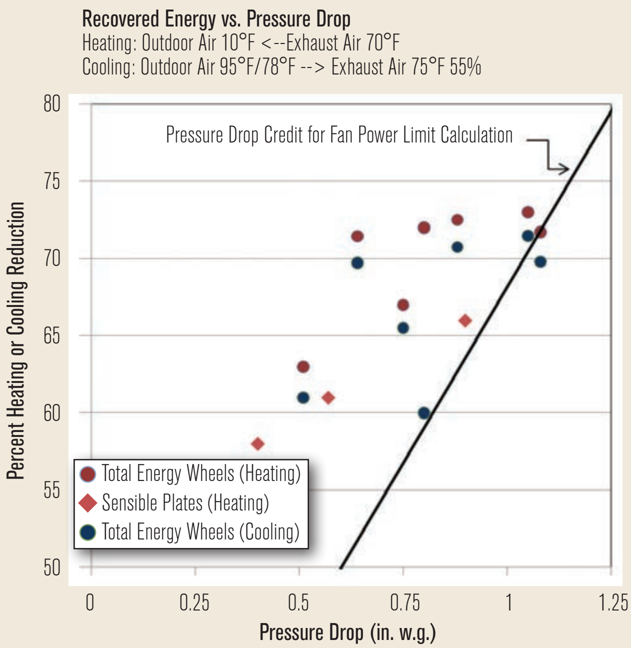
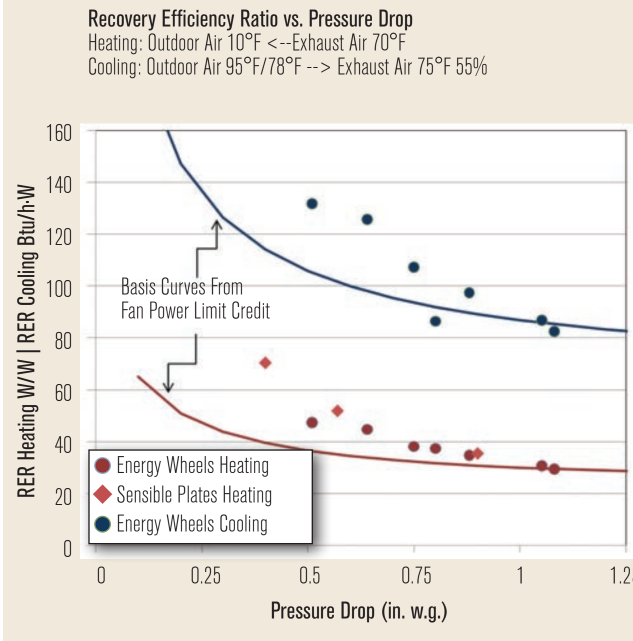


FIGURE 6 Design day recovery efficiency.

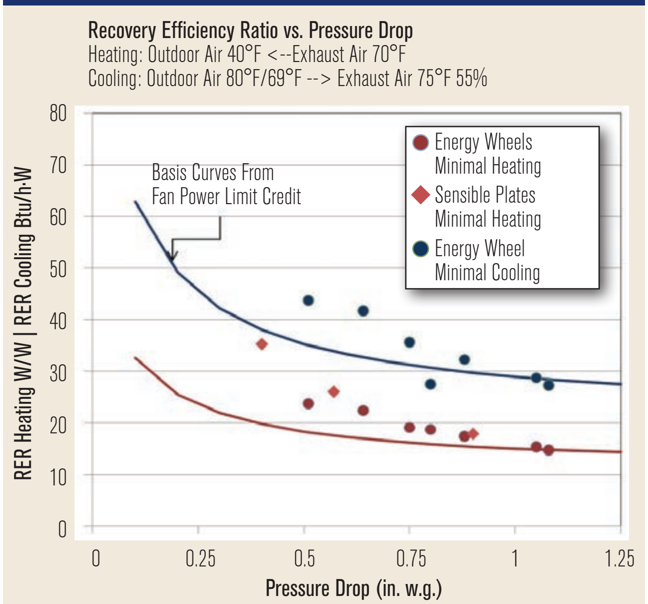


The curves represent boundaries below which an energy-recovery exchanger would add more fan power than credited in Standard 90.1-2010. The exhaust air will cool and dehumidify the outdoor air at the cost of added fan power. The added power to recover the energy in this example is assumed to be a 0.33 hp (0.25 kW) motor with 10% air leakage using a 60% overall efficient fan to overcome the pressure loss. These processes can be seen on the psychrometric plot in *Figure 4*. This chart shows that it will be far more efficient at design conditions to use energy recovery than use traditional cooling or heating methods

At the design cooling day, the RER range is 80 to 130 Btu/h·W (23 to 38 Wh/W). This is approximately eight times more efficient than typical air-cooled vapor compression systems and approximately four times more efficient than typical water-cooled vapor compression equipment at this design condition. Compared with gas-fired heating equipment that has a COP <1.0 or heating or electric heat at best with a COP of 1.0, a heating design day RER of 30 to 50 W/W will be more than 30 to 50 times more efficient!

What about part-load heating and cooling conditions? *Figure 7* plots RER at 80°F/69°F (27°C/21°C) outdoor condition during cooling and 40°F (4°C) during heating. Recovering energy is still more efficient than vapor compression during this part-load cooling condition and 15 to 20 times more efficient during heating. At

FIGURE 7 Part-load day recovery efficiency.



this condition, it is possible the only heat needed for the building may be recovered, showing why in regions where there is a heating and cooling season, most of the operating savings will be in heating.

Which Exchanger to Use?

RER is a good metric for comparing how efficiently the recovery device conditions ventilation air, rather than other methods of heating or cooling. However, when comparing different energy recovery exchangers, both effectiveness and RER need to be considered. An exchanger with a higher effectiveness may have a lower RER, but the higher effectiveness results in less work to be done by the less-efficient mechanical heating or cooling system. Which exchanger will result in higher overall system efficiency is a balance between effectiveness and RER, and depends on the efficiency of the mechanical conditioning system. AHRI Guideline V, *Calculating the Efficiency of Energy Recovery Ventilation and Its Effect on Efficiency and Sizing of Building HVAC Systems,* is a good resource that includes several examples on calculating this combined efficiency.

Energy Recovery Exchanger Placement vs. Performance

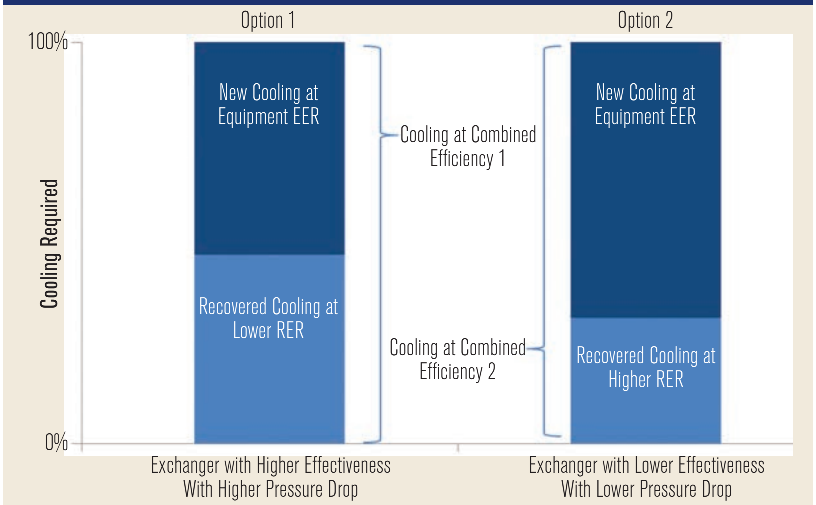
The application of the device within the system also will affect the fan power added with energy recovery. *Figure 6* will give a good approximation of what to expect for a 100% outdoor air system. However, in some cases, adding exhaust air recovery may not add supply fan power at design and may actually lower it. The air-handling equipment arrangement can make a difference. This is especially true for mixed air systems, those that use air-handling equipment to cool and heat both the ventilation air and recirculated air to condition the building.

For example, let's look at a variable air volume air handler with ducted return air. This example system (shown as System A in the tables) at design has a 1.25 in. w.g. (311 Pa) external static pressure on the return path, with a supply airflow of 10,000 cfm (4719 L/s) and outdoor airflow of 5,000 cfm (2360 L/s) and exhaust airflow of 4,500 cfm (2124 L/s). The alternative proposed system with energy recovery (System B) adds an energy wheel with a 0.90 in. w.g. (224 Pa) supply pressure drop and 0.80 in. w.g. (199 Pa) exhaust pressure drop having an exchanger effectiveness of 80% that reduces the outdoor air enthalpy 71% with respect to the exhaust.

The prescriptive fan power limit in Standard 90.1 allows for an additional 2.44 bhp (1.82 kW) to the allowed system design fan power limit (see sidebar, "Example Fan Power Limitation"). When considering this system one may divide this credit between the exhaust and supply fans. More often many assume that including recovery will add the 0.90 in. w.g. (224 Pa) to both the exhaust and supply fan total static pressure, and use this as the basis for the building energy model for System B. This assumption will result in 3.8 bhp (2.83 kW) more at design, raising the system total brake horsepower from 11.9 bhp to 15.7 bhp (8.8 kW to 11.7 kW). The accuracy of either of these assumptions is dependent on how the recovery exchanger is applied in the system.

Energy recovery is well adapted to systems that condition mixed air. For systems with ducted return air, the increase in fan energy due to energy recovery is often less than assumed. Why? The majority of mixed air systems have outdoor intake hoods, louvers, eliminators, dampers, and outdoor air filters all sized for air economizing.

FIGURE 8 Combined system efficiency comparison.



Economizing airflows are typically two to three times higher than the required minimum ventilation rate. Only the minimum ventilation rate will be required for outdoor airflow during hours of energy recovery, the static pressure drop on the outdoor air intake sized for economizing will be low. Therefore, the return air path static pressure is often higher than the outdoor air path with the energy recovery device, so adding energy recovery to this type of system often does not increase the required supply fan power.

Adding energy recovery allows the cooling coil to be downsized, which can reduce the rows or fin spacing

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Example Fan Power Limitation

ASHRAE 90.1-2010 Section 6.5.3.1.1

For Example VAV System A

Design Power Limit (bhp) = $CFM_S \times 0.0013 + A$

A = Sum of $(PD \times CFM_D/4{,}131)$

Ducted Fully Ducted Return Credit of 0.5 in.

Max bhp= 10,000 × 0.0013. (0.5 × 9,500)/ 4,131 = 14.15 bhp

Additional Credit for Energy Recovery

Energy Recovery Device Pressure Credit per Airstream

PD = (2.2 × Energy Recovery Effectiveness) – 0.5 in. w.c.

PD = (2.2 × 0.71) – 0.5 = 1.06 in.

Power Limit Add in bhp

A = sum of $(PD \times CFM_D/4{,}131)$

$A = (1.06 \times 5{,}000/4{,}131 + 1.06 \times 4{,}500/4{,}131) = 2.44$ bhp

CFM_S = the maximum design supply airflow rate to conditioned spaces served by the system.

TABLE 1 Example VAV systems. Static pressure summary.

	VAV Air Handler – Supply Fan Static Pressure (in. w.g)			
SYSTEM	**A**	**B**	**C**	**D**
Outdoor Air Intake	0.04	0.04	0.04	+0.10
Outdoor Air Filter		0.05	0.05	
Energy Wheel		0.9	0.9	
Return Duct Static	1.25	1.25	1.25	1.25
Return Damper	0.05	0.05	0.05	0.05
Filter	0.55	0.55	0.55	0.55
Heating Coil	0.10	0.10	0.10	0.10
Cooling Coil	0.70	0.55	0.55	0.55
Discharge Loss	0.25	0.25	0.25	0.25
Supply Duct Static	1.50	1.50	1.50	1.50
Supply Fan TSP	4.40	4.25	3.39	4.25
	VAV Air Handler – Exhaust Fan Static Pressure (in. w.g.)			
SYSTEM	**A**	**B**	**C**	**D**
Return Duct Static	1.25	1.25	1.25	–
Filter		0.55	0.55	
Energy Wheel		0.8	0.8	
Exhaust Outlet	0.21	0.21	0.21	
Exhaust Fan TSP	1.46	2.81	2.81	–

of the coil. Therefore, the supply air total static pressure at design might actually decrease slightly when energy recovery is added. For the example, total static pressure seen by the supply fan drops from 4.4 in. w.g. (1095 Pa) (System A) to 4.25 in. w.g. (1057 Pa) (System B). The total static pressure seen by the exhaust fan increases (1.46 in. w.g. [363 Pa] for System A to 2.8 in. w.g. [696 Pa] for System B) more than the recovery device pressure drop due to the filter which is also added in the exhaust fan path. The net result at design is that total system bhp increases from 11.9 bhp (8.9 kW) to only 13.3 (9.9 kW) (much less than 15.7 bhp [11.7 kW] when the exchanger static pressure drop is assumed to be added to the fans).

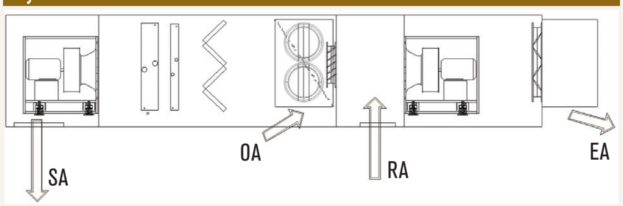
The assumption that energy recovery will add significant static pressure to the supply fan leads some to consider using a return fan system (System C) to off load some of this static pressure. This may remove the return air duct static pressure and exhaust filter static loss from the supply air fan, however the outdoor air path will have the heat exchanger in its path, therefore, it will only reduce 0.5 in. w.g. (124 Pa) of static from the supply fan. This fan placement more than doubles the power required to handle the exhaust air, so the net result is 4.4 bhp (3.3 kW) added for the example. This will also have a negative impact on airflow control for the system. Energy recovery adds 0.8 in. w.g. (199 Pa) of static pressure to each air path downstream of the return fan and this will cause over 2 in. w.g. (498 Pa) of pressure difference across the return damper. A small change in damper position will change return flow greatly.

Another approach is to keep the same basic air handler (as in System A) and add a separate energy recovery ventilator (ERV) to the system (System D). With this arrangement, the ERV's outdoor air and exhaust air paths are typically not oversized to enable air economizing. This can add significant fan energy to the system during hours of energy recovery, especially to the exhaust path where a smaller abrupt discharge and a less efficient fan are often used. This system uses additional 5.8 bhp (4.3 kW) at design, more than three times the added fan power versus incorporating into the recovery exchanger into the air handler unit. These two units are considered one "fan system" when ducted together, and the prescribed fan power limitation in the Standard 90.1 is exceeded.

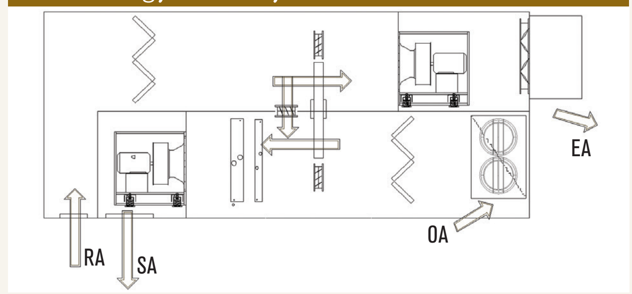
The exhaust and supply fans for all these selections range between 63% to 66% efficient. For the add-on ERV unit in the example (System D), the fan efficiency and internal static pressures are unknown as the ERV brake

FIGURE 9 AND TABLE 2 Example VAV Systems. Unit diagrams and fan power comparison.

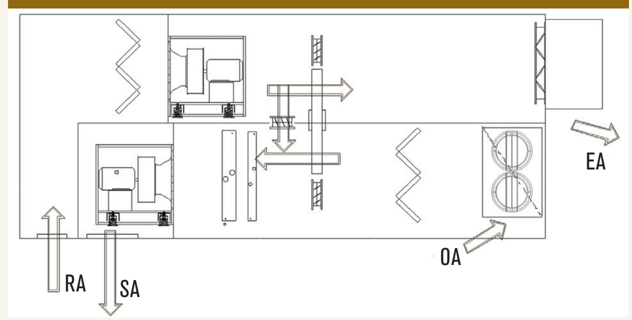
System A: VAV Air Handler with Exhaust Fan



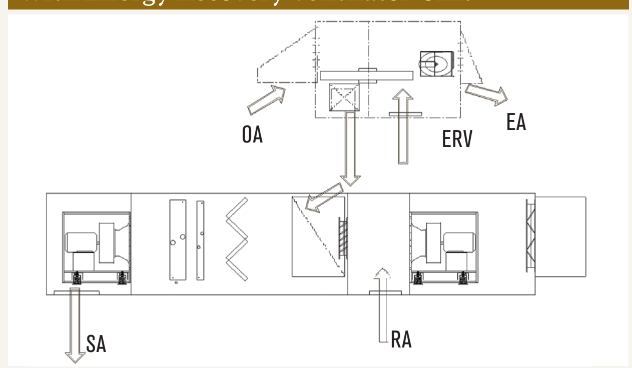
System B: VAV Air Handler with Exhaust Fan With Energy Recovery



System C: VAV Air Handler with Return Fan With Energy Recovery



System D: VAV Air Handler with Exhaust Fan With Energy Recovery Ventilator Unit



	SYSTEM A	SYSTEM B	SYSTEM C	SYSTEM D
Design Supply Fan Airflow (cfm)	10,000	10,000	10,000	10,000
Supply Fan TSP (in. w.g.)	4.4	4.25	3.4	4.25
Supply Fan Static Efficiency	66%	66%	66%	66%
Supply Fan Power (bhp)	10.41	10.11	8.05	10.11
Design Exhaust/Return Fan Airflow (cfm)	4,500	4,500	9,500	0
Exhaust/Return Fan TSP (in. w.g.)	1.46	2.8	2.8	N/A
Exhaust/Return Fan Static Efficiency	68%	63%	62%	N/A
Exhaust/Return Fan Power (bhp)	1.52	3.18	6.8	N/A
ERV Ventilation Fan (bhp)	N/A	N/A	N/A	2.8
ERV Exhaust Fan (bhp)	N/A	N/A	N/A	4.5
System Total Fan Power(bhp)	**11.93**	**13.29**	**14.85**	**17.41**

Design Notes:

- SA=Supply Air; OA=Outdoor Air; EA=Exhaust Air; RA=Return Air (Exhaust+Recirculated); TSP=Total Static Pressure
- At Design OA=5,000 cfm; EA=4,500 cfm; SA=10,000 cfm
- System D exhaust fan is off on design and used for economizing relief only
- Outdoor Air intakes and exhaust air outlets sized for economizing

horsepower is cataloged against external static pressure. For all examples, the RER values are significantly higher than mechanical heating or cooling efficiencies. For mixed air systems, there will be a large variance in efficiency, depending on how it is applied. However, in most cases adding energy recovery to a mixed air unit can be even more efficient than shown in *Figures 6* and *7* particularly when it is added integral to the air handler.

In the example case (System B), energy recovery is at 120 Btu/h·W (35 W/Wh) at design cooling and 48 W/W at design heating. These results fall in line with the current prescriptive energy recovery requirements in Standard 90.1 that list energy recovery for mixed air along with 100% outdoor air systems as being economically beneficial. Another point to consider when applying energy recovery integral to the system, is to make sure the building modeling software models it this way. The easiest way to model the fan energy is to treat energy recovery as an add-on ERV. Adding separate fan power in the exhaust and outdoor airstreams, as shown in the design day point example, can be inaccurate and over calculate the power required.

One thing is clear, If the energy recovery device is on when it is supposed to be on, off when it supposed to be off, and bypassed during economizing, it should reduce the overall building energy usage (see March 2012 *ASHRAE Journal* article, "Total Energy Wheel Control in a Dedicated OA System" by John Murphy for a good control overview).

Conclusion

Energy recovery will add fan energy and efficiently use this fan energy to save more cooling and heating energy. However, care should be taken with building simulation with respect to fan energy. Energy recovery is not as simple as adding static loss to the ventilation and exhaust air paths—the system layout matters. Layout will affect the required fan power and, therefore, the system efficiency, but regardless of arrangement, if conditioned exhaust air is present, exhaust air recovery will be multiple times more efficient than using the building mechanical heating or cooling source. ■

TABLE 3 Example VAV Systems. RER comparison.

		VAV	VAV WITH ENERGY WHEEL	VAV WITH RETURN FAN + ENERGY WHEEL	VAV + ERV
SYSTEM		A	B	C	D
Design System bhp	bhp	11.9	13.5	14.9	17.7
Standard 90.1 Fan Power Limitation	bhp	14.2	16.5	16.5	16.5
Added Fan Power for Energy Recovery	bhp		1.6	3.0	5.8
System Input Power Added At Design	W		1,565	2,747	5,202
Recovered Design Cooling	Btu/h		188,000	188,000	188,000
Cooling Recovery Efficiency Ratio, RER	Btu/h·W		120.1	68.4	36.1
Recovered Design Heating	Btu/h		256,000	256,000	256,000
Heating Recovery Efficiency Ratio, RER	W/W		48	27	14