HVAC System Design Employing Certified Air-to-Air Energy Recovery Equipment

Robert W. Besant and Carey J. Simonson

Robert W. Besant
Professor Emeritus,
Dept. of Mechanical Engineering,
University of Saskatchewan,
57 Campus Drive,
Saskatoon, SK, S7N 5A9.

Carey J. Simonson
Assistant Professor,
Dept. of Mechanical Engineering,
University of Saskatchewan,
57 Campus Drive,
Saskatoon, SK, S7N 5A9.
Carey.Simonson@usask.ca

Introduction

Air-to-air heat and moisture exchangers have been available for HVAC ventilation design for many years. Despite energy cost shocks and environmental concerns, they have yet to gain full acceptance in HVAC design. This is perhaps due to:

(1) the added complexity of incorporating energy exchangers in a good design, and
(2) a lack of guaranteed performance factors.

In this paper these knowledge deficiencies are discussed in detail by presenting:

(i) all the air-to-air exchanger performance factors for heat exchangers and heat and moisture exchangers
(ii) presenting a simple step-by-step HVAC design procedure
(iii) illustrating this design methodology in a simple example and
(iv) demonstrating the importance of certified equipment.

Air-to-Air Exchanger Performance

Each air-to-air exchanger has several performance factors that should be determined under tightly controlled test conditions [1,2] in a fully qualified laboratory if errors are to be minimized [3,4,5]. These include: effectiveness for sensible energy ($?_s$), moisture transfer or latent energy ($?_m$), and enthalpy or total energy ($?_t$), pressure drop through both the supply ($?p_s$) and exhaust ($?p_e$) air flow sides, exhaust air transfer ratio (EATR), outside air correction factor (OACF) and recovery efficiency ratio (RER). All these performance factors are discussed in more detail in the previously listed references. Typical values for each of these performance factors for various types of air-to-air heat exchangers and heat and moisture exchangers are presented in Tables 1 and 2.

It should not be concluded from these tables that each of these performance factors is precisely known when it is stated by a manufacturer. The facts are that the effectivenesses of devices such as rotary energy wheels vary with operating conditions so that summer performance factors will differ from winter ones. ARI has developed a certification program [1] to ensure equipment performance, but even ARI certified units will have uncertainty in the effectiveness values at any operating condition that can range over ± 5% because it is not possible to provide more accurate data even when testing to standard test requirements. Compared to other test standards, the new ASHRAE Std 84-91 [1] test conditions are quite complex and manufacturers usually don't have the requisite test facilities and cannot do the required online balances and uncertainty analysis during testing; therefore, for uncertified units, this range of uncertainty will most likely be much larger, perhaps ± 10 to 30% of the listed value.

The designer should not assume that it will be easy to hold a supplier accountable after the equipment is installed. Generally it is very difficult and costly to try to get accurate data from tests on installed equipment – so it is rarely done.

A designer of an HVAC system needs to know all the expected performance factors of each unit or device used in the system for all operating conditions over a typical year of operation. This information should be provided by each manufacturer.

HVAC System Design

Well-designed HVAC systems often integrate one or more air-to-air exchangers into processing the supply air. An example of such a system is shown in *Figures 1 and 2* using energy (*e*) and heat (*h*) exchangers where the only useful source of waste airflow and energy is assumed to be the space return and exhaust air. Return air (*a*) recirculation is included in both figures because it is often necessary to achieve comfortable supply air temperatures and at flow rates that result in good mixing in each space.

In *Figures 1 and 2,* the exchange of heat and moisture in (*e*) and heat in (*h*) are assumed to be controllable with flow directions that depend on the potential difference for temperature and water vapor pressure. The utility input power to fans (*F*), cooling (*C*) or heating (*H*) equipment are controlled to meet the required supply air delivery mass flow rate ($\dot{m}$), temperature and humidity at Point 7. Dampers control the required supply air ventilation mass flow rate ($\dot{m}_1$) and the fraction of recirculated air (*a*).

The HVAC system design process is sequential and iterative starting with the building requirements and the summer and winter design conditions for a particular location. This design sequence is shown in the "HVAC System Design Process," and it includes the design objectives of minimum life-cycle and first costs. The tradeoff or weighting between the life-cycle and first costs is a decision that should be made by the building owner.

HVAC System Design Process

1. List all building requirements for the HVAC system and constraints including each utility cost. Design a process for the supply air that fully exploits the waste energy flows from the exhaust air and other sources.
2. Estimate the properties at each station and the peak heat and moisture transfer rates at the summer and winter design conditions using typical performance factors for the air-to-air exchangers as in Tables 1 and 2. From these results, estimate the remaining utility peak energy rates.
3. Estimate the utility annual energy loads and costs using an estimate of part load and operating condition time duration.
4. Estimate the first or installed cost of all components from known data.

Iterate on Steps 2, 3 and 4 using the above estimates and manufacturer's quoted prices, certified equipment performance data, utility energy rate data, and perhaps simulation studies to meet the design objectives of a weighted minimum life-cycle and first costs.

Design Example [5]

The system design process is best illustrated using a simple example. For a particular new building in a specified location, we will estimate the first design iteration, size the air-to-air exchangers, calculate the expected savings in cooling and heating equipment capacity, and calculate the expected annual energy savings as outlined in the first three steps of The Design Process. Costing of all the installed components and computing the optimal design for the weighted minimum first cost and life-cycle cost are not included due to the need to use more data.

Analysis for the System in Figure 1

1. Building location: Chicago, IL, USA

(a) Outdoor design conditions:

summer	t_1 (1%) = 34°C	t_1 wb = 23°C
winter	t_1 (99%) = -21°C	t_1 wb = -21°C

(b) Supply air conditions:

mass flow	$\dot{m}_7$ = 2 kg/s	$\dot{m}_1$ = 1 kg/s
summer	t_7 = 15°C, f_7 = 0.60, where f is the relative humidity	
winter	15 = t_7 = 23°C	f = 0.15

(c) Return air conditions:

mass flow rate	$\dot{m}_8$ = 2 kg/s	$\dot{m}_{12}$ = 1 kg/s
summer	t_8 = 25°C f_8 = 0.55	
winter	t_8 = 23°C f_8 = 0.20	

(d) Space heat load (supplied by baseboard heaters) equals two times the ventilation air heat load without heat recovery, but with a break-even temperature for the building equal to t_1 = 10°C (i.e., space heating is required only when t_1 = 10°C).

(e) Selection of the air-to-air exchanger performance factors for energy exchange from *Table 1*. High, but realistic, effectivenesses are selected to maximize the potential energy savings (i.e. $e_{s,e} = e_{m,e} = e_{s,h}$ = 0.75 where s = sensible, m = moisture, e = energy wheel, e = heat exchanger).

2. (a) Calculate the sensible and latent heat exchange rates for the supply air using the definitions for energy exchange effectiveness:

$$\dot{q}_{s,e} = e_{s,e}\dot{m}_1 C_p (t_{11} - t_1) \quad (1)$$

= energy wheel sensible heat rate

$$\dot{q}_{m,e} = e_{m,e}\dot{m}_1 h_{fg} (W_{11} - W_1) \quad (2)$$

= energy wheel moisture or latent heat rate

$$\dot{q}_{s,h} = \varepsilon_{s,h}\dot{m}_5 C_p (t_9 - t_5) \quad (3)$$

= heat wheel sensible heat rate

(b) During summer design conditions, the critical air temperature at the cooling coil outlet, t_5, is determined by the space supply air temperature and humidity.

t_5 (t_7 = 15°C, f_7 = 0.60) = 10.8°C

(c) Determination of properties in winter and summer: the remaining temperatures and humidity ratios at each station in *Figure 1* are determined using energy balance equations. The results for the winter and summer design conditions are in *Figures 1 and 2*. It is assumed that t_6 = 15°C in summer implying that the heat exchanger (h) will be controlled so this temperature is maintained. In winter, t_6 is allowed to increase but remain in the range 15 = t_6 = 23°C. Fan power input was assumed to have a negligible effect on the temperature of the air.

(d) Determination of energy rates in winter and summer.
From the equations in (a) we get the winter and summer peak heating and cooling rates shown in *Table 3.*

(e) The remaining auxiliary heating and cooling for the design conditions are implied in *Table 3.* The last column gives the reduction in installed capacities for the design example. In winter the boiler capacity can be reduced by 41% compared to the base case with no air-to-air exchangers. In summer the chiller capacity can be reduced by 38% and the boiler can be eliminated.

3. The annual energy loads are calculated using the integral of the time duration of each operating condition during the entire year. That is, without air-to-air exchange, the annual heating energy for ventilation air is given by:

$$q_{\text{vent}} = \int_0^{8760 hours} \dot{q}_{vent}(t_1)dt \qquad (4)$$

where

$$\dot{q}_{vent} = \dot{m}_1 Cp(t_7 - t_1) \text{ for } t_7 > t_1 \qquad (5)$$

Besant and Simonson [4] show how this can be readily done once the typical year outside air temperatures are rearranged so that t_1 increases monotonically from its lowest to its highest value as a function of time duration.

Table 4 presents the results of such calculations for both temperature and enthalpy for each of the ventilation air, the space heating and cooling needs and the totals. It shows that the total heating energy can be reduced by 92% in winter and 100% in summer for a total reduction of yearly heating energy of 93%. The total chiller cooling energy can be reduced by 21%. *Table 4* shows, similar to *Table 3*, that the air-to-air exchangers are able to eliminate the need to heat ventilation air in the winter and reheat the ventilation air in the summer.

The net extra fan power required with total fan efficiencies of 50% as a result of introducing exchangers (e) and (h) and eliminating the auxiliary heater (H) are about 180 W for the supply air motor and 540 W for the exhaust. Without part load by-pass for the exchangers, this extra power is essentially constant for all the operating hours, giving an annual energy use of about 6,300 kWh for the specified flow rates and assuming a pressure drop of 200 Pa for each airstream across each exchanger. The auxiliary power to rotate the energy and heat wheels would be about 200 W and over the year the annual energy use would be 1,750 kWh. The ratios of all the energy saved over the year divided by the extra energy input for the fans and auxiliary motors gives a seasonal recovery efficiency ratio (SRER) of 22.

These significant reductions in energy use can be related to annual operating cost savings using typical energy cost data. If we assume net total heating energy costs provided to the building are \$0.05/kWh and the cooling energy costs are \$0.15/kWh, then the total annual energy costs will be \$8,380 and \$570 for the boiler without and with the air-to-air exchangers. For the chiller these corresponding costs will be \$21,300 and \$16,800.

Effect of Degraded Performance Factors

The question the HVAC designer needs to consider when specifying any piece of equipment is – how accurate or reliable is each performance factor? If a device under-performs – will the overall HVAC system still meet comfort conditions and what will it do to the operating energy use and costs? The designer should know what the possible range of performance factors are for all the HVAC equipment including boilers, chillers, etc. In the case of air-to-air exchangers, the designer should be aware that exchangers certified by ARI likely have an uncertainty of ±5% in effectiveness, while uncertified exchangers may have an uncertainty as high as ±10% to ±30%.

To answer these two questions for the air-to-air exchangers, two cases are considered. First, we assume that the effectiveness values are degraded by 5% of the value listed (i.e. e = 71%) as it could be for certified exchangers. Second, we assume that these values are degraded by 20% (i.e. e = 60%) which could be the case for uncertified exchangers, where the uncertainties in the test results have not been carefully analyzed.

The results of these new cases are presented in Table 3, for the peak heating and cooling rates, and Table 4 for the annual energy use. Comparing the new or revised peak energy rates and annual energy use with the case of the design assumed effectiveness of 0.75 or 75% reveals some interesting observations. The certified air-to-air exchangers (e = .71) reduces the benefits of the air-to-air exchangers slightly while the uncertified exchangers (e = .60) causes a significant reduction in the benefits and may result in the need to put a heating coil in the supply duct.

Conclusions

Several types of air-to-air heat and energy exchangers are available. The performance factors for these devices are documented and can be accurately determined within a specified uncertainty range which will differ for certified and uncertified exchangers.

The example system in Chicago shows large reductions in the required capacity of the boiler (41%) and chiller (38%) and the elimination of any auxiliary heating in the supply air duct. The resulting annual energy savings of 92% for the boiler and 21% for the chiller lead to a significant operating cost savings. These numbers change only slightly when certified performance factors are degraded slightly (i.e. e = .71 from e = .75) but significantly when uncertified exchangers are considered (i.e. e = .60 from e = .75).

References

1. ANSI/ASHRAE Standard 84-1991, (Revised in 2003) *Method of Testing Air-to-Air Heat Exchangers*.
2. Air Conditioning and Refrigeration Institute, 2001. ARI Standard 1060-2001, *Rating Air-To-Air Energy Recovery Ventilation Equipment.*
3. Simonson, C.J., D.L. Ciepliski and R.W. Besant. 1999. "Determining the performance of energy wheels: Part I and Part II". *ASHRAE Transactions* 105(1):174-.
4. Besant, R.W. and C.J. Simonson. 2000. "Air-to-air energy recovery." *ASHRAE Journal* 42(5):31-42.
5. Besant, R.W. and C.J. Simonson. 2003. "Air-to-Air Exchangers". *ASHRAE Journal* 45(4):2-9.

Table 1 Comparison of Air to Air Heat Exchangers

	Fixed Plate	**Rotary Heat Wheel**	**Heat Pipe**	**Runaround Coil Loop**
$e_s(m_s = m_e)$	.5 to .8	.5 to .85	.45 to .70	.40 to .65
$\Delta P_s = \Delta P_e (P_a)$	100 to 1000	100 to 300	150 to 500	150 to 500
EATR (%)	0 to 5	1 to 10	0 to 1	0
OACF	0.97 to 1.06	1 to 1.2	0.99 to 1.01	1.0
Face velocity, m/s	1 to 5	2 to 5	2 to 4	1.5 to 3
Temperature range	-60 to 800ºC (-76 to 1470ºF)	-55 to 800C (-67 to 1470ºF)	-40 to 40C (-40 to 105ºF)	-45 to 500C (-49 to 930ºF)
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coils only Complete system
Advantages	No moving parts Low pressure drop Easily cleaned	Compact large sizes Low pressure drop Easily cleaned	No moving parts except maybe an active control tilt Fan location not critical High allowable pressure differences	Exhaust airstream can be separated from supply air Fan location not critical High allowable pressure differences
Limitations	Some pressure deformation of plates for some designs	Some EATR without purge	Effectiveness limited by pressure drop and cost	High effectiveness often requires an accurate simulation model
Heat rate control schemes	Bypass dampers and ducting for supply air	Bypass dampers and wheel speed control	Tilt angle down to 10% of maximum heat rate or bypass	Bypass valve or pump speed control

Table 2 Comparison of Air-to-Air Heat and Moisture Exchangers

	Rotary Dehumidifier	**Energy Wheel**	**Permeable Plate**
e_s (m_s = m_s)		0.5 to 0.85	0.5 to 0.8
e_m or e_l		0.5 to 0.85	0.3 to 0.8
e_t		0.5 to 0.85	0.4 to 0.8
COP / RER	0.4 to 1	40 to 100	20 to 80
$\Delta P_s = \Delta P_e (P_a)$	100 to 300	100 to 300	100 to 300
EATR (%)	0 to 1	1 to 10	1 to 5%
OACF	1 to 1.02	1 to 1.1	0.97 to 1.06
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system
Advantages	Compact large sizes Low pressure drop Ability to use waste heat	Compact large sizes Low pressure drop Availability on all ventilation system platforms	Compact large sizes Low pressure drop No moving parts
Limitations	Requires hot air source supply air Requires sensible energy cooling	Supply air requires further cooling or heating in some applications	Supply air requires further cooling or heating in some applications
Heat and moisture rate control	Bypass dampers and wheel speed control	Bypass dampers and wheel speed control	Bypass dampers

Table 3 **Peak Energy Rates**

	Loads Without Exchanger (q)			**Air-to-Air Exchanger Savings (q_{AE})**				**Ratios**
	Space Heating or Cooling	Ventilation Air	Total Boiler/ Chiller	e	Space Heating or Cooling	Ventilation Air	Total	$\frac{q_{AE}}{q}$
Winter $\dot{q}_{Boiler}$ (kW)	62.0	36.0	98.0	.75 .71 .60	4.0 3.3 1.2	36.0 36.0 36.0	40.0 39.3 37.2	.41 .40 .38
Summer $\dot{q}_{Chiller}$ (kW)	22.8	50.0	72.8	.75 .71 .60	4.2 4.2 4.2	23.7 22.3 19.9	27.9 26.5 24.1	.38 .36 .33
Summer $\dot{q}_{Boiler}$ (kW)	4.2	4.2	8.4	.75 .71 .60	4.2 4.2 4.2	4.2 4.2 4.2	8.4 8.4 8.4	1.0 1.0 1.0

Table 4 **Energy Use**

	Energy Use Without Exchangers			**Air-to-Air Exchanger Savings**				**Ratios**
	Space Heating or Cooling	Ventilation Air	Total Boiler/ Chiller	e	Space Heating or Cooling	Ventilation Air	Total	$\frac{q_{AE}}{q}$
Winter q_{Boiler} (kWh)	74000	60000	134000	.75 .71 .60	63000 61000 55000	60000 60000 60000	123000 121000 115000	.92 .90 .86
Summer $q_{Chiller}$ (kWh)	65000	81000	146000	.75 .71 .60	14000 14000 14000	16000 15000 13000	30000 29000 27000	.21 .20 .18
Summer q_{Boiler} (kWh)	14000	14000	28000	.75 .71 .60	14000 14000 14000	14000 14000 14000	28000 28000 28000	1.0 1.0 1.0

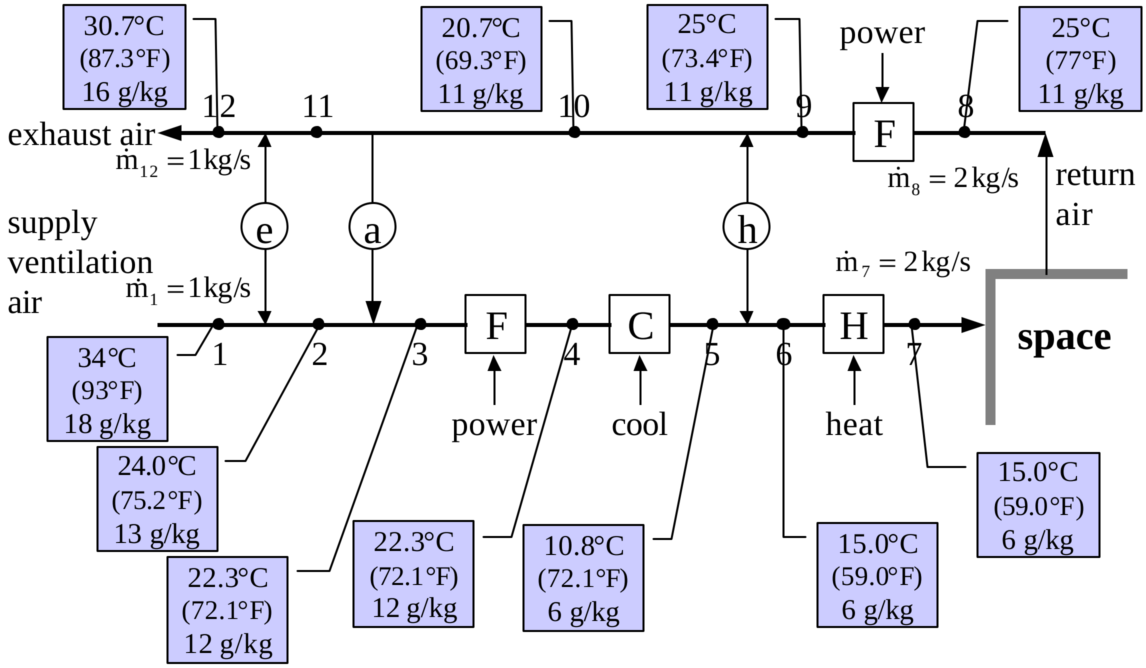

Figure 1. HVAC system integration showing air, heat and moisture exchange and the resulting temperatures and humidities at summer design conditions.

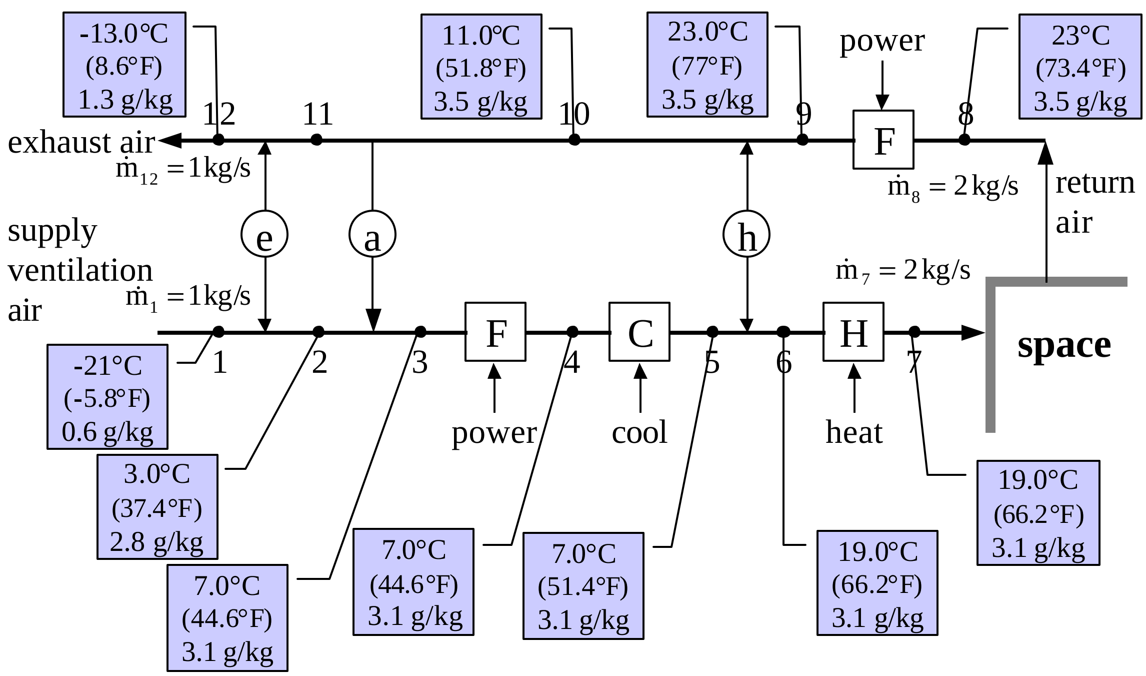

Figure 2. HVAC system integration showing air, heat and moisture exchange and the resulting temperatures and humidities at winter design conditions.