Multi-Functional, High-Performance Run Around Energy Recovery Systems in Cold Climate Zones

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ABSTRACT

While the thermodynamics of a run-around energy recovery system are relatively simple, it is critical that high-performance systems operate at optimum performance under varying operating parameters. With several variable input parameters (outside air/supply air/return air temperatures; air volumes; fluid volumes & temperatures), controlling and optimizing a system requires a numerical simulation based controller ('performance mapping') that allows variable amounts of heat transfer fluid to be circulated throughout the system.

In multi-functional systems, additional heating and/or cooling is introduced into the fluid circuit, either to increase the heating/cooling capability of the energy recovery system from waste heat/cooling sources, or to control the supply air temperature to the building to eliminate separate heating/cooling coils in the supply air handlers. These features add yet another level of complexity to the controller function.

INTRODUCTION

HVAC systems are among the greatest energy consumers of large buildings – in particular laboratory buildings and hospitals with 100% outside supply air. High performance run-around energy recovery systems (RAERS) with advanced control software are operating at 70%-90% net effectiveness (based on annual energy consumption for heating and cooling), taking into account the additional electricity needed for fluid pumps and added fan power due to the additional air pressure drop of the air-to-fluid heat exchangers in the air handlers.

1 THE BASIC RUN-AROUND ENERGY RECOVERY SYSTEM

1.1 System Components

A basic run-around energy recovery system consists of fluid-to-air heat exchangers ('coils') in the supply air handler and the exhaust air handler, and a pump to transport the fluid through the coils and interconnecting piping. In heating mode ('winter operation'), the exhaust air is cooled while the fluid in the exhaust air coil is being heated up, and conversely, the supply air is heated while the fluid in the supply air coil is cooled. Figure 1 shows the system components and the temperature diagram of the system: the outside air is heated from 0^0C to 15^0C (32F to 59F) while the exhaust air is cooled from 22^0C to 7^0C (71.6F to 44.6F) and the fluid circulates in between 4^0C and 19^0C (39.2F and 66.2F).

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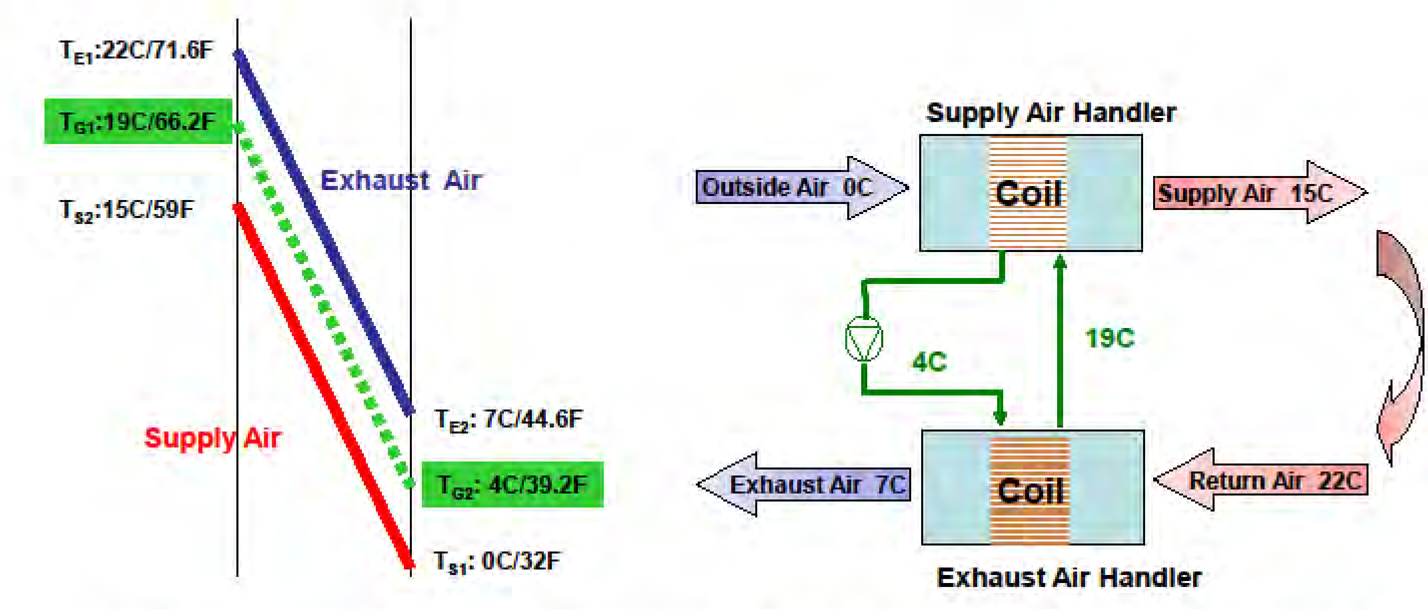

Figure 1: Temperature diagram and components of a basic run-around energy recovery system

1.2 Critical Thermodynamic Fundamentals

A run around energy recovery system is governed by the energy transfer equation:

$$\mathbf{Q = k * A * \Theta_m} \qquad \mathbf{(1)}$$

Q: exchanged energy duty [W]
k: heat transfer coefficient [W/(K*m²)]
A: heat exchanger surface area [m²]
Θ_m: logarithmic mean temperature difference [K]

While the heat exchanger surface area in an installed system is fixed, the log mean temperature difference Θ_m and the heat transfer coefficient k will vary with changing operating parameters and need to be optimized by the control software of a system.

Log mean temperature difference: The log mean temperature difference relates to the temperature differences between air and fluid at each end of an heat exchanger. With the nomenclature of the exhaust air coil in Figure 1, assuming counter-current between air and fluid, it is:

$$\mathbf{\Theta_m = [(T_{E1} - T_{G2}) - (T_{E2} - T_{G1})] \,/\, \ln [(T_{E1} - T_{G2}) \,/\, T_{E2} - T_{G1})] = (dT_1\text{-}dT_2) \,/\, \ln (dT_1/dT_2)} \qquad (2)$$

While for typical coils with more than about 12-14 rows, the assumption of counter-current flow is allowable, the formula for a mix counter/cross-current flow coil can only be approximated (see Reference Bes T.). An "engineering approach" to calculating Θ_m for mixed counter/cross current coils is to introduce an empirical correction factor:

$$\Theta_m \text{ [counter/cross-current]} = \Theta_m \text{ [counter current]} * \text{RFAK} \quad (3)$$

RFAK: correction factor for counter/cross current flow; 0 < RFAK < 1

This correction factor will vary for each coil design, coil geometry and fluid velocity; hence, a vast empirical database is necessary to determine this factor with sufficient accuracy.

Also, in case dT_1 equal to dT_2 in Formula 2 (as is the case for the system of Figure 1), it can be shown that Θ_m equals dT_1 (and dT_2); hence for the exhaust air coil of Figure 1: $\Theta_m = 3$.

Heat transfer coefficient: The main heat transfer processes involved are the convective heat transfers from air to the coil fins and from the fluid to the tube walls, and the conductive heat transfer in the fins & tubes of the heat exchanger. The individual heat transfer coefficients of these processes add up to the overall heat transfer coefficient of the system. Critical in the convective heat transfer is that the fluid (or gas) flow is turbulent, in particular the fluid flow in the coil tubes, as the heat transfer coefficient deteriorates rapidly with partial laminar and complete laminar flow. Figure 2 illustrates the dependence of the heat transfer coefficient of the Reynolds number, a measure for turbulence in liquid flows in a pipe.

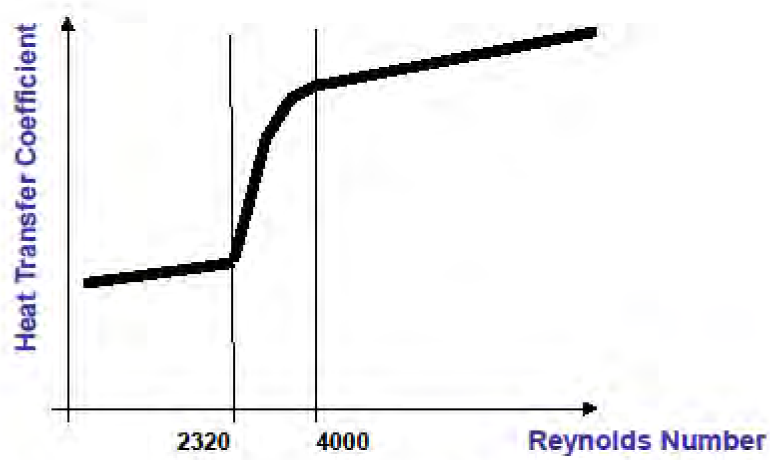

Figure 2: Dependence of heat transfer coefficient and Reynolds number

Furthermore, viscosity and thermal conductivity of a typical heat transfer fluid (20%-50% Ethylene/Propylene Glycol) vary depending on temperature; this variance need be taken into account as well.

Efficiency: The energy (or heat) recovery efficiency is usually expressed as the temperature transfer efficiency at a specific operating or design point of a system; with the nomenclature of Figure 1, this would be:

$$\mu_T = (T_{E1} - T_{E2}) / (T_{E1} - T_{S1}) \quad (4)$$

Or analogous, the energy transfer efficiency and enthalpy transfer efficiency can be determined. The shortcomings of this definition of efficiency when comparing different energy recovery technologies or in doing a financial analysis are obvious:

- Over the course of a year, a system will operate in a wide spectrum of operating parameters.
- The annual climate pattern of the geographic location of an installation is an important factor.

- Energy consumed by an energy recovery system (e.g. fan power due to additional air pressure drop in the air handlers, or the fluid pump power for a run-around system) should be taken into account.

To design and evaluate an energy recovery system, these factors need be taken into account.

1.3 Optimizing System Performance

The parameters governing system performance according to Formula 1 are the heat transfer coefficient (k), the heat exchanger surface area (A) and the log mean temperature difference (Θ_m). While the heat exchanger surface area in an installed system is fixed the heat transfer coefficient and the log mean temperature difference will vary depending on the operating parameters and requires continuous optimization. Figure 3 shows as an example a system at design air flow (100% air flow rate) and 100% fluid flow rate, the same system at 40% air flow and 100% fluid flow rate (pump without VFD), and lastly the same system at 40% air flow and reduced fluid flow to optimize k * Θ_m at these operating conditions. The temperatures in this example have been experimentally measured as well as predicted by a system controller.

Operating parameters:

Air Volume	[m3/h]	10,000	4,000	4,000
Fluid Volume	[m3/h]	4,000	4,000	1,250
μ_T		67%	58%	72%
Energy recovered [Wh/m3 air]		4.8	4.2	5.1

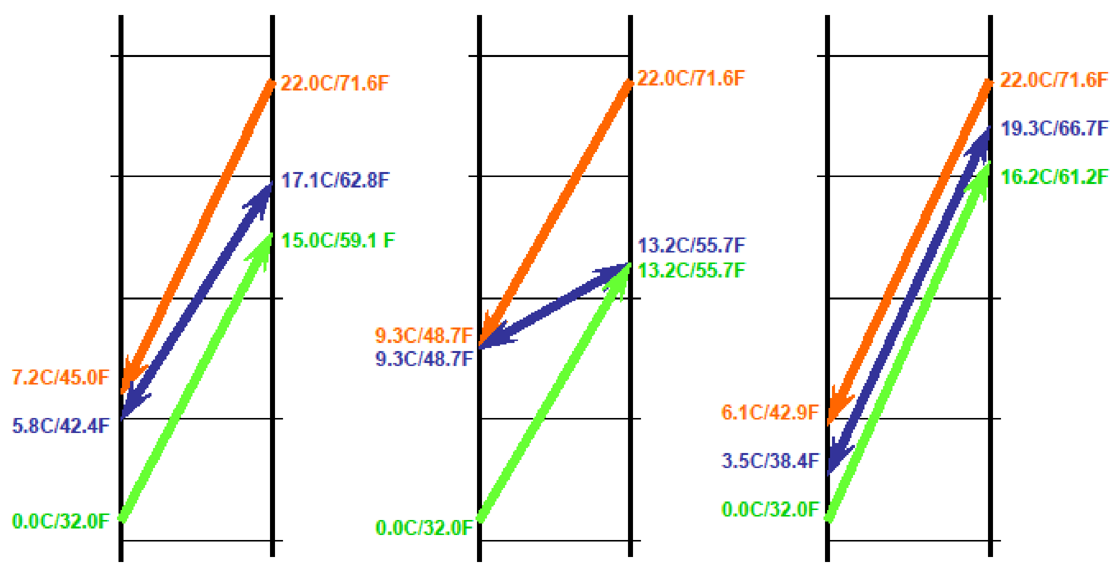

Figure 3: Optimizing k * Θ_m

It has been shown for cross-flow flat-plate heat exchangers that this optimization occurs when the capacitance rates of air and fluid are equal [see Ref. Fan et.al.]. Under ideal conditions, this is applicable to finned tube heat exchangers as well, however, parameters such as coil geometries and Reynolds numbers will deviate the optimal fluid flow rate from equal capacitance.

1.4 Exhaust Frost Control

Of particular concern in cold climates is the potential for frost build-up in the exhaust coil: if the exhaust air is cooled close to 0C and below the dew point, the precipitating moisture will freeze in the exhaust coil. Solutions to prevent this are either preheating the supply air to avoid low fluid temperatures being returned to the exhaust coil, or to control the fluid temperature to the exhaust coil through a bypass. Extensive experience with a specific coil design shows that if the fluid temperature to the exhaust coil is maintained above a temperature of about -2C/28.4F, frosting will not occur (this temperature will vary between different coil designs). In the system of Figure 4, 49% of the fluid flow of the exhaust air coil is diverted through the bypass to elevate the fluid temperature returning from the supply air coil of -21.4C/-6.5F to -2C/28.4F. However, opening this bypass will at the same time decrease the fluid flow through the supply air coil and if not designed properly, turbulent fluid flow will not be maintained, thus reducing the heat transfer coefficient in the supply air coil drastically and causing the system to operate in an unstable condition.

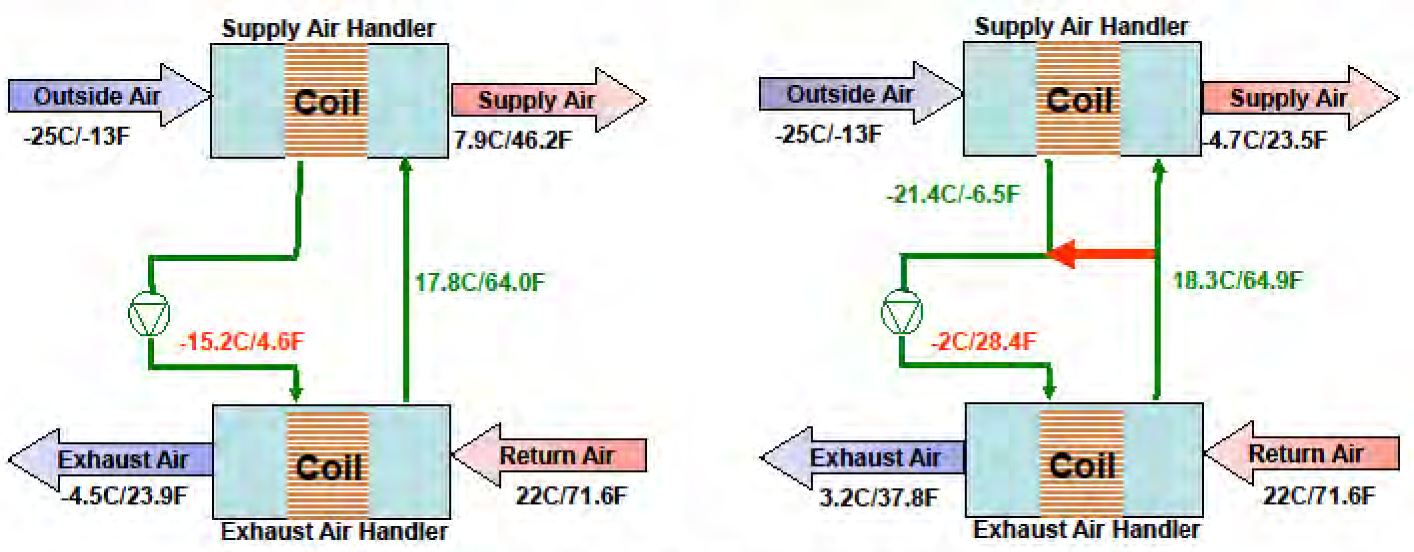

Figure 4: System without and with Frost Control Bypass

2 MULTI-FUNCTIONAL RUN-AROUND ENERGY RECOVERY SYSTEMS

2.1 Combining Several Air Handlers in one Energy Recovery System

The advantage of combining several air handlers in one energy recovery system becomes obvious when supply and/or exhaust air temperatures and/or volumes are different: Figure 5 shows an arrangement of two supply air handlers and two exhaust air handlers. While the supply air set temperature in supply air handler #1 is 12C and the return air temperature of exhaust air handler #1 is 28C, the supply air set temperature in air handler #2 is 18.3C and the return air temperature in exhaust air handler #2 is 22C. If these air handler-pairs are equipped with individual energy recovery systems, the total heat recovered at the depicted operating point, with individually optimized fluid flow rates, is 343.6 kW because not all the heat available for recovery in exhaust #1 can be introduced in supply #1 due to the low supply air set temperature, while a total of 387.4 kW can be recovered in a combined energy recovery system, an increase in recovery efficiency of 13%.

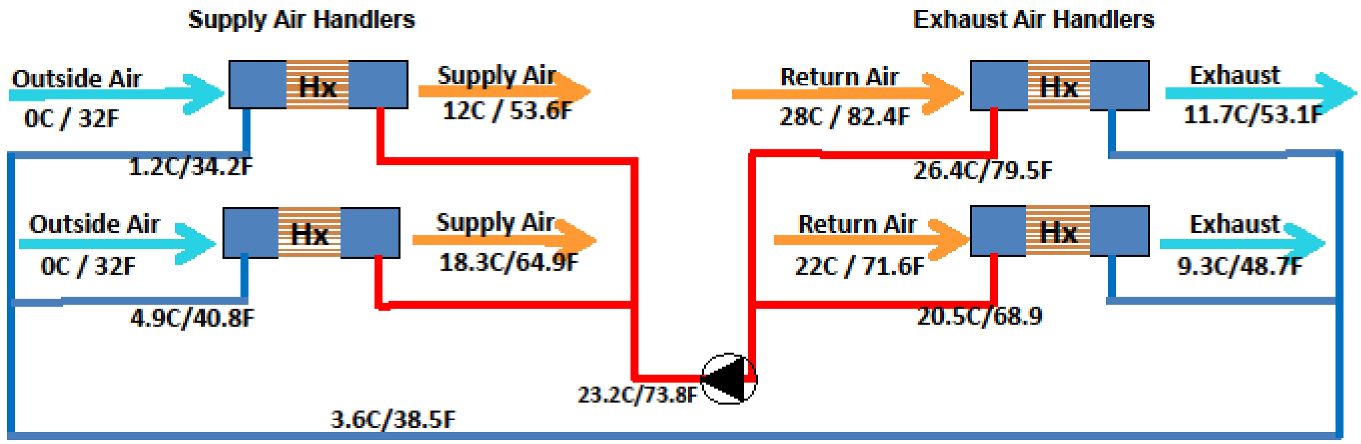

Figure 5: Combining several Air Handlers in one Energy Recovery System

Combined System

	Air Volume	Air Temperature		Fluid Vol.	Fluid Temperature		Power
		In	Out		In	Out	
	m3/h	C	C	m3/h	C	C	kW
Supply 1	40000	0.0	12.0	6648	23.2	1.2	153.4
Supply2	40000	0.0	18.3	12200	23.2	4.9	234.0
Exhaust 1	40000	28.0	11.7	9272	3.6	26.4	221.8
Exhaust 2	40000	22.0	9.3	9272	3.6	20.6	164.4
Individual Systems							
Supply 1	40000	0.0	12.0	5404	27.7	0.8	153.4
Exhaust 1	40000	28.0	16.3	5404	0.8	27.6	153.2
Supply2	40000	0.0	14.9	12200	18.9	4.1	190.2
Exhaust 2	40000	22.0	7.3	12200	4.1	18.8	189.4

Table 1: Operating parameters combined energy recovery system vs. individual systems

2.2 Adding Heat to the Fluid Circuit

The main purpose of adding heat by means of a plate-and-frame heat exchanger to the fluid circuit of an energy recovery system is to eliminate the heating coil and thus minimize the air pressure drop and fan power in an air handler. The pitfall in adding heat to the fluid circuit in an 'uncontrolled' (constant fluid flow rate) system is that with the elevated fluid temperature entering the supply air coil, the leaving fluid temperature, also elevated, can exceed the exhaust air temperature, thus heating the exhaust air rather than cooling it. Reducing the fluid flow rate at elevated fluid temperatures entering the supply air coil will decrease the fluid leaving temperature (i.e. will increase the leaving-to-entering temperature difference of the fluid), eliminating the risk of heating the exhaust air. Yet again, stable operating conditions will only be achieved if turbulent fluid flow in the coils is maintained under reduced fluid flow rates.

A well controlled system will yield 1-3% less annual heat recovery by adding heat to the fluid circuit due to slightly elevated fluid temperatures leaving the supply air coil at certain operating conditions, a trade-off to be considered in eliminating the heating coil; however, the exhaust air will not be heated under any operating parameters while the system is in heating mode. Alternatively, if the goal is to achieve equal annual heat recovery with adding heat as without adding heat, designing the coils with additional heat transfer surface (i.e. adding rows) would be a possibility, however, this would also increase pump and fan power - factors that need be taken into consideration in designing an energy recovery system.

The 1-3% reduction in annual yield has been observed and measured in many installed systems in cold climate zones of Switzerland and Germany, and is has been simulated in different North American climates such as Calgary AB, Edmonton AB, Chicago IL, Denver CO.

2.3 Adding Cold to the Fluid Circuit

Similarly to adding heat, the main purpose of adding cold by means of a plate-and-frame heat exchanger to the fluid circuit of an energy recovery system is to eliminate the cooling coil. The effect of reducing air pressure drop and fan power is even more pronounced as cooling coils usually create higher air pressure drops than heating coils, and in an annual energy consideration, in cold climates, a cooling coil will be less than 1000 hours in operation, adding the air pressure drop during the entire 8760 hours of fan operation (provided there is no air bypass around the cooling coil in the air handler). The pitfall in adding cold to the fluid circuit is even more pronounced than in adding heat, as the approach temperatures air – fluid in cooling mode typically are smaller than in heating mode, increasing the likelihood of cooling rather than heating the exhaust air.

On the other hand, the reduction of annual yield in cooling mode is of minor importance because of the few annual operating hours a system can actually recover cold from the exhaust air, and because the fluid circuit will bypass the exhaust air coils during times where the exhaust air temperature is higher than the outside air temperature.

3 CONTROL STRATEGY

A multi-functional energy recovery system cannot be controlled by a traditional 'sequence of operation'. The control input parameters are:

- Air volumes in all air handlers
- Outside air temperature, exhaust air temperatures (optional: exhaust air humidity), supply air set temperatures
- Heat/cold addition options

All of these parameters can and will vary over time. The control strategy is to optimize energy recovery, meaning to optimize the product of heat transfer coefficient and logarithmic mean temperature differences of Formula 1 in the supply air and exhaust air heat exchangers. The result of a numerical simulation of the energy recovery system on the basis of operating performance maps of all relevant elements in the system (heat exchangers, pumps, valves) is the optimal fluid flow rate through each coil bank in the system. Based on these fluid flow rates, the pump speed and control valves will be set accordingly.

The most challenging elements in developing such a numerical simulation are the operating performance maps of the heat exchangers (coils): the variables of these performance maps are air and fluid volumes and temperature differences, and the heat transfer coefficient.

CONCLUSION

Combined energy recovery / heating / cooling coils in supply air handlers can be very beneficial to the annual energy recovery balance in cold climates, as the air pressure drop across coils and hence fan power is minimized.

However, combining these functions in one coil requires a sophisticated control concept with variable fluid flows optimizing the log mean temperature difference in both the supply air and exhaust air coils. Variable fluid flow only yields the desired results if turbulent fluid flow is maintained in the coils at the critical operating points.

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