Example #1 - Indoor Recreational Pool

Design Criteria - Case study

Commissioned:	January 1990
Location:	Toronto, Ontario
Room Size:	1500 ft^2
Pool Size:	700 ft^2 (nominal)

Design Conditions: - water 80°F [27°C]
- air 82°F [28°C], 55% relative humidity

Heating Load:	21 MBH (building fabric only)
Pool cover:	Yes

Preamble

The formula found in ASHRAE for calculation of the minimum air quantity required to remove evaporated water can be reduced to a single manageable digit if certain assumptions are made. This method may be used successfully as a “rule of thumb” method. For a more detailed calculation, see the guide for calculation of ventilation requirements based on moisture evaporation in this section.

The first assumption is that the design conditions for a pool equate to 0.04 lb [0.02 kg] of moisture evaporated from the surface per square foot of pool surface.

The second assumption is that in a worst case scenario, the required ventilation equates to 25 cfm per lb of water evaporated. Therefore:

0.04 lb/ft^2 x 25 cfm/lb = 1 cfm/ft^2 (at 40° F [4°C] ambient temperature and 55% RH inside)

The rule of thumb is therefore:

Pool ventilation requires **1 cfm/ft^2** of pool surface.

Minimum Ventilation Calculation

Pool size = 700 ft^2

700 ft^2 x 1 cfm/ft^2 = 700 cfm

Thermodynamic Load Calculation for Ventilation

The calculation of thermodynamic load for ventilation in simple terms is as follows:

$$Q = \frac{\text{Flow Rate (cfm)} \times \text{Run Time (hr./day)} \times 1.08 \times \text{Degree Days (°F)}}{\text{Btu (natural gas)} \times \text{\% Efficiency (heating device)}}$$

Where:

Flow Rate = 700	Btu = 35,335
Run Time = 24%	Efficiency = 0.9
Degree Days = 6827	

Therefore:

$$Q = \frac{700 \times 24 \times 1.08 \times 6827}{35,335 \times 0.9}$$

Q = **3895 m^3 (137 million Btu/year)**
(heat load for ventilation only)

Cost Savings Analysis with HRV700i

With natural gas at $0.22 per cubic meter, the ventilation heat load without heat recovery represents an annual cost of:
Cost = 0.22 $/m^3$ x 3895 m^3 = **$857.00** (based on 24 hr. operation)

The HRV700i heat recovery ventilator operating under a level of consistent high humidity will have a regain efficiency of 75%. Installation of this unit will reduce the annual cost to an amount equal to:
Cost = [3895 m^3 - (3895 m^3 x 75%)] x 0.22 $/m^3$ = **$214.00** (based on 24 hr. operation)

Equipment Selection

1) Hydronic boiler, 150 MBH (nom.) capacity (pool water, space and post core heating).
2) HRV700i, fixed plate Heat Recovery Ventilator, polypropylene core.
3) Hydronic copper/aluminum finned coil (sized to suit total calculated heat loss, with protective coating).
4) Controls: Pool Plus wall control (includes dehumidistat control), low voltage thermostat.
5) Ducting system: Coated spiral round, or sealed rectangular duct sized to suit airflow, (installed as a closed loop below the floor slab or above windows).
6) Air delivery: Linear slot diffusers, 15° inclined, aluminum (air to sweep glass surfaces).
7) Return air: Single grille, aluminum.
8) Defrost air: Via dedicated duct and grille (defrost air is not to be taken from any mechanical room).

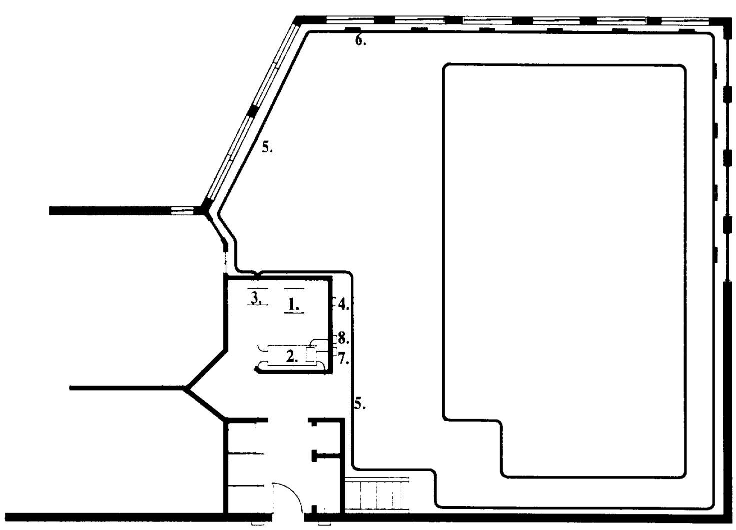
Operation

General

This application uses the supply fan of the HRV700i as the source of energy for both recirculation of the space heating and as a source of dilution ventilation for the purpose of moisture control. In the absence of either a heating or dehumidification call, the entire system is idle, with minimal energy usage.

Control Strategy

The room thermostat closes on a call for heat, (class 2 wiring), which operates a hydronic zone valve which is complete with end switch. The end switch initiates circulator pump action. The same room thermostat via a relay, starts the supply fan (only) of the HRV, to provide space heating.
Upon a call for dehumidification, with or without a call for heating, the dehumidistat closure causes the HRV to go to high speed ventilation with both fans on providing full flow dilution ventilation.



Example #2 - Theoretical Indoor Assembly Area

A Theoretical Application in New Orleans, LA

Preamble

The following example is offered as an illustration of the need for the application of different technology to the task of reducing cooling loads in a semi tropical location.
Indoor swimming pools are the exception rather than the norm in such climates, an indoor assembly area is used as an example.

Concept

Regardless of the precise application, energy use reduction in geographical locations with high wet bulb temperatures, using energy recovery ventilation equipment, does not reclaim energy, but diverts the unwanted portion of same back to atmosphere, thereby reducing the installed load of the comfort cooling system.

Ventilation, per se, has been assumed to be mandatory, and to consist of two components, infiltration/exfiltration and controlled mechanical. The incidence of improved building practices is steadily decreasing the presence of the former, while increasing the presence of the latter. This evolving trend allows pre-planning for the use of the ERV to reduce both the first cost of equipment and to more accurately predict the cost (fuel) operation of the equipment.

Calculations

1. Airflow Rates
Many attempts have been made to produce both estimated and mandated ventilation flow rates, intended for application in the broad, and sometimes bureaucratic sense, however circumstances alter cases. A typical assembly room can accommodate people pursuing a multitude of activities. There may be some who are smoking, some who are moving energetically, others sedentary. For the purpose of this discussion the selection of 25 cfm (cubic feet per minute) per person is illustrative, not definitive, as is an occupancy of 100 persons.
For the purpose of this example a flow rate of 2500 cfm is used.

2. Design Conditions (summer)
New Orleans, LA has a design dry bulb temperature of 92°F [33°C] and a design wet bulb of 78°F [25°C]. Or an enthalpy of 41.7 Btu/lb of dry air and associated moisture.
Inside design conditions approximate: 75° F [24°C] dry bulb, and 55% R.H. Or an enthalpy of 29.5 Btu/lb of dry air and associated moisture.
A difference of 12.5 Btu/lb etc.

3. Load at Design
The additional load upon the air conditioning of this building, attributable to controlled ventilation will be proportional to:
4.5 x 12.5 x 2500 or 140,625 Btu/hr or approximately 12 tons.

Load Seasonal
This may best be estimated by using the recorded bin records for St. Charles AP, LA which provides time @ temperature in hours.
Five bins are relevant, 95 - 100° F, 90 - 95° F, 85 - 90° F, 80 - 85° F, 75 - 80° F.
35 - 38°C, 32 - 35°C, 29 - 32°C, 27 - 29°C, 24 - 27°C.
The co-incident wet bulb temperature for these dry bulb conditions provide an enthalpy difference for each bin which may easily be tabulated into a seasonal total using the formula:

4.5 x cfm x De

Where: cfm = flow rate in cubic feet per minute
De = enthalpy difference in Btu/lb of introduced air
4.5 = mass flow rate of standard air per hour

Using this formula:
The first bin equates to: 4.5 x 2500 x 12.5 x 2 = 281,250 Btu/season

The second bin equates to:	4.5 x 2500 x 12.5 x 246	= 34,593,750 Btu/season
The third bin equates to:	4.5 x 2500 x 8 x 664	= 59,760,000 Btu/season
The fourth bin equates to:	4.5 x 2500 x 6.3 x 819	= 58,046,625 Btu/season
The fifth bin equates to:	4.5 x 2500 x 2.5 x 1572	= 44,212,500 Btu/season

Using a Seasonal Energy Efficiency Ration (SEER) of 10 Btu/Watt the total energy usage equals 19,689,413 Watts or 19,689 kW.

4. Operating Cost

Using an arbitrarily set price per Kilowatt hour of $0.075, the energy used represents an operating cost of $1,476.68. The building used for this example is assumed to operate during the hours of maximum enthalpy difference, no adjustment is made for hours of occupancy.

Mitigation of Operating Cost

This can be achieved via the application of an ERV unit or units, utilizing coated rotary wheel technology. Such a unit diverts ambient air borne moisture back to the atmosphere via the fan energy contained in the outgoing air stream of the ventilation unit. Simultaneously, there is some sensible temperature reduction of the fresh air entering the building. These units have been shown to provide combined latent/sensible diversion/regain levels of 70% on a seasonal basis. Thus, the previously calculated operating cost may be reduced to:

[1,476.68 - (1,476.68 X 70%)] = $443.00 **This is a savings of $1,033.68 per season.**

Conclusions

Unequivocally, some form of an Energy Saving device MUST form an essential part of every contemporary mechanical design package. The only question at issue is one of selection of the appropriate technology to suit the climatic conditions. It is worthy to note that while two extremes have been examined, a location with both low winter temperature and high summer dry bulb conditions may show an **annual** savings greater than either of those mentioned.

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