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MODELING AND SIMULATION OF BUILDING ECONOMIZER AND ENERGY RECOVERY SYSTEMS FOR OPTIMUM PERFORMANCE

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ABSTRACT

This paper describes how building economizer and energy/enthalpy recovery system control and operation can be modelled and simulated to evaluate and optimize their performance. A improved method to determine optimum temperature setpoints and outdoor air fractions is discussed and demonstrated. Through the simulation, the new strategy based on total system energy use is shown to be able to save up to 11% of energy under certain conditions and to be most useful for dry climates.

INTRODUCTION

Buildings are constructed to serve a purpose, namely to provide a comfortable, safe, healthy and productive environment for human endeavours. They require the investment of scarce resources and substantial time and effort to design, build and operate, so efficiency and cost effectiveness are always considerations. In particular, energy efficient performance, and the means to achieve it, is of critical interest to building designers and owners. A tension exists, however, between building energy efficiency, thermal performance and occupant health and comfort, whereby the means employed to minimize energy usage may negatively impact thermal and air quality within the building's conditioned space. This is a constant challenge faced by building designers and operators, and the focus of many design codes, standards and tools. In addition to requiring building envelope components to reduce or control unwanted heat transfer, building codes impose minimum requirements for ventilation air, sometimes known as fresh air, to control the buildup of contaminants in the indoor air. Ventilation standards are based on a combination of factors, usually including building use, size and occupancy, and inadequate ventilation has been linked to problems with moisture, allergens and generally unhealthy conditions.

It is within this context that building economizer cycles and energy recovery systems have evolved. The introduction of ventilation air from the outside may impact heating or cooling loads since the outdoor air needs to be conditioned to maintain the indoor comfort conditions. This means that in extreme climate conditions (winter, summer) additional energy may be required to heat or cool the ventilation air as compared to a sealed building. An economizer cycle utilizes outdoor air when appropriate to provide "free cooling", the rough equivalent of opening the windows on a nice day rather than using mechanical cooling. The former study has demonstrated that the energy savings associated with economizer could be very significant (Brandemuehl et al., 1999). The energy saving of the robust control strategy on cooling coil energy consumption which was evaluated by over one year's comparison tests on two air-handling units in a building was 88.47% in contrast to the energy consumption using the conventional control (damper at fixed position) in Hong Kong (Wang et al., 2004). Energy recovery systems exploit the "free heating or cooling" provided by exchanging energy between the exhaust and outdoor airstreams. In both of these systems, some control logic must be implemented to determine how and when the systems should be operated to obtain the most favourable outcome.

This paper examines the current methods used for controlling building economizer and energy recovery systems, and related methods for modeling and simulating their performance. It identifies some inconsistencies in approaches and suggests some improved models and simulation methods to more accurately evaluate energy performance and increase the energy savings potential from the use of these technologies.

SIMULATION

There are several important aspects related to the control, modeling and simulation of building HVAC economizer and energy recovery systems. The discussion will commence with economizer control, and then extend to the analysis of energy recovery systems. The actual implementation of the economizer control, for example, can be done in different ways depending on the installed sensors and actuators. The basic strategies (ASHRAE Standard 90.1-2010) employed include the following conditions, under which the outdoor air intake damper would be opened beyond the minimum required for ventilation:

- fixed dry bulb-outdoor air dry bulb temperature less than a set value
- differential dry bulb- outdoor air dry bulb temperature less than return air temperature
- fixed enthalpy- outdoor air enthalpy less than a set value
- differential enthalpy- outdoor air enthalpy less than return air enthalpy

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• combination- some combination of the above

• optimal- a combined strategy that maximizes energy savings

The differential modes require at least two sensors, and the enthalpy-based strategies require the measurement of relative humidity or some other moisture content indicator, while the combination strategies require multiple temperature and enthalpy sensors. Lacking these sensors limits the control strategies that can be implemented. Even when the sensors are installed, sensor errors and other measurement problems can seriously compromise performance, particularly for enthalpy measurements, which are notoriously unreliable (Zhou et al., 2008). Several studies have illustrated the potential degradation of economizer energy performance due to sensor uncertainty (Seem et al., 2010). That said, this paper will not explore that issue any further, but will focus on the theoretical basis for economizer control assuming properly functioning and reasonably accurate sensors.

A schematic representation of optimum economizer control logic based on (Taylor, 2010) is shown in Figure 1 in the form of a psychrometric chart, which is formatted to show a wide range of air temperature and moisture conditions. Note that lines of constant dry bulb temperature are nearly vertical on a psychrometric chart, increasing from left to right on the horizontal axis, while moisture content increases on the vertical axis, and lines of constant enthalpy slant diagonally downward from left to right. Two air condition points define the boundaries for optimum economizer control, one being the return air temperature T_r (which is assumed to be equal to the zone air temperature) and the supply air temperature T_s (which is assumed to be saturated). If the condition of the air entering the cooling coil is above the moisture content defined by the supply air temperature intersection with the saturation curve, condensation will occur on the coil (wet coil) while if the moisture content is below the line, the condensation will not occur (dry coil). Thus, it will be advantageous to use 100% outdoor air if the outdoor air condition lies in region A, based on lower enthalpy for the wet coil, or region B, based on lower dry bulb temperature for the dry coil. In regions D and E, the use of outdoor air is not advantageous, while in region F, some fraction of outdoor air may be beneficial.

This figure, while useful, misses several subtle points. First, the supply air condition in most cases will not be completely saturated, so will lie to the right of the saturation curve on a line connecting the condition of the air entering the coil and the coil apparatus dew point. This will shift the location of the room air condition, which in turn will change the condition of the air entering the cooling coil. This effect is described more fully below. Second, outdoor air conditions in region C, being warmer but dryer than the return air, may provide reduced cooling loads with partial economizer operation. This can be a substantial benefit in certain climate regions. A method for optimal control in this region will be outlined below. Third, the optimal control logic in Figure 1 is based on minimizing cooling coil loads, but the energy is actually expended to operate a chiller and pump to provide chilled water to the coil, and a fan to provide conditioned air to the space. The energy inputs to these components depend on a number of factors, including chilled water supply temperature, part load ratio and condenser entering temperature. Optimizing economizer operation to minimize energy input by adjusting setpoints without diminishing comfort will be demonstrated.

A typical air handling unit (AHU) has hydronic heating and cooling coils, a supply fan, and ducts, dampers and control valves. For the sake of simplicity, the system is assumed to control the air temperature within a single zone, although that assumption can readily be extended to a multi-zone case without loss of applicability. A source of hot and chilled water is further assumed to provide the heating and cooling functions to the room air by way of the respective heating and cooling coils. Outdoor air can be drawn in to the system through the outdoor air damper, and similarly exhausted through the exhaust air damper. These two dampers work in conjunction with the return air damper to admit the desired quantity of outdoor air to meet ventilation, pressurization and economizer requirements for the building zone. The two basic modes of fan operation are constant air volume (CAV) or variable air volume (VAV), with the latter being preferred for its energy saving potential.

In either CAV or VAV operation, the zone heating and cooling loads are met by supplying conditioned air to the zone such that the product of the mass flow rate of the supply air, the specific heat of air and the temperature change of the air from supply (T_s) to return (T_r) are equal to the zone thermal load:

$$\dot{q}_z = \dot{m} c_p (T_r - T_s) \quad (1)$$

The difference between the CAV and VAV modes is that as loads vary, supply air temperature is varied by the control system for CAV, and airflow rate is varied for VAV. Thus, supply air temperature is held at a setpoint (although not necessarily constant) for VAV, while being varied for CAV. Zone air temperature is also held at a setpoint, although the setpoint may be adjusted under certain conditions. Since the economizer cycle is primarily a cooling related function, it is useful to represent the processes involved in conditioning the zone air on a psychrometric chart, as shown in figure 2. The AHU does not have direct control of zone relative humidity

(RH_z), so the zone air condition can lie anywhere on the room air dry bulb temperature line shown in figure 3 as T_r. The outdoor air condition can be located literally anywhere on the chart, but for this example has been selected to represent a hot and humid condition typical of summer day (T_o, RH_o). Any mixture of return air and outdoor air must lie on the line connecting the two points, the mixed air line, in proportion to the relative amounts of each. For the outdoor air condition shown in this figure, using the minimum amount of outdoor air would result in the lowest cooling loads. For example purposes, a mixed air temperature (T_m) has been marked on the figure.

In order for the mixed air to be cooled and dehumidified to the desired supply air temperature setpoint (T_s), the cooling coil must have chilled water flowing through it so that its temperature is somewhat colder than the desired supply air temperature. The temperature of the cooling coil is called the apparatus dew point temperature (T_d), because water vapour will condense from the mixed air if its dew point temperature exceeds T_d, which would be true for this example. The important point, and the reason for this example, is to emphasize that the condition of the discharge air leaving the cooling coil must lie on the line connecting T_m and T_d, known as the condition line, but generally not all the way to T_d for real cooling coils. Physically this is because the coil is finite in size and not all of the air passing through it will be cooled uniformly, a phenomena termed bypass factor, which can range from 10 to 20% or higher. The control system will modulate the chilled water flow through the coil to maintain the dry bulb temperature of the air leaving the cooling coil at the setpoint, T_s, and any dehumidification will be a by-product of the sensible cooling of the mixed air. The moisture content of the air leaving the cooling coil is determined by the intersection of the condition line with the supply air temperature line. Finally, the supply air is delivered to the zone, absorbing heat (sensible load) and moisture (latent load), and under equilibrium conditions, maintaining the zone temperature at the setpoint. Unless 100% outdoor air is being used, the complete cycle must form a closed loop as shown in the figure.

The forgoing discussion sets the stage for modeling economizer control. The critical and frequently overlooked point is that at any point in time, there are specific relationships between the zone sensible loads q_S and latent loads q_L, zone air conditions, outdoor air conditions, outdoor air fraction, apparatus dew point temperature and supply air temperature. Thus, it is not valid to pick any arbitrary collection of values to determine whether the economizer should be used and with what outdoor air fraction, or to compute the coil loads. In fact, the room condition (dry bulb temperature and RH) for any combination of room loads will vary with outdoor air fraction, which will in turn influence the cooling coil loads and supply air moisture content. This is shown in figure 2 by the difference between the heavy dashed lines for 100% recirculated air and the heavy solid lines for 50% outside air. For this particular outdoor air condition, this does not affect the optimum outdoor fraction, which would be the minimum allowable considering ventilation requirements, but does impact coil loads, energy consumption and room comfort conditions. There are other outdoor air conditions, however, for which these effects do have an impact on optimum outdoor air fraction, as will be discussed in the following.

The simulation objective is to obtain the optimized setpoints with the optimization criteria of minimizing the total system energy usage for cooling. This requires models of the performance characteristics of the each component, the chiller, chilled water pump and supply fan. For this analysis, polynomial fits were used with generic coefficients, with the important variables being chilled water supply temperature, coil loads, chilled water flow rate, outdoor air fraction, supply airflow rate, supply air temperature and room temperature. For the simulation software, EES(Engineering Equation Solver) was selected because of its built –in high-accuracy thermodynamic and heat transfer parameters and it is ideally suited for solving design problems in which the effects of one or more parameters must be determined. Given any set of room sensible and latent loads, along with the outdoor air temperature and humidity, the optimum values for the temperature setpoints and controlled variables can be determined from the thermodynamic relationships previously described. In particular, figure 3 shows the economizer control logic for the hot and dry outdoor air condition. It can be shown (see Figure 4) that the minimum coil load will occur if the mixed air fraction is such that moisture content of the air is equal to the moisture content of saturated air, in other words, if the cooling coil is at the borderline between wet and dry conditions. Under these conditions, the coil only removes the sensible heat load, and the drier outdoor air precisely balances the room latent load. The condition line is horizontal in this case, and the increase in sensible cooling load due to the introduction of outdoor air is more than compensated by the reduction in latent load.

A further implication of this analysis is that since mechanical dehumidification is not required, we are free to adjust the cooling coil apparatus dew point temperature, provided we can maintain acceptable room comfort conditions. This effect can be more clearly seen in the next example, as shown in figure 5. As can be seen in this figure, for the cool dry outdoor air condition, 100% outdoor air fraction can be used and minimizes the coil load. However, since only sensible cooling is required, the apparatus dewpoint temperature T_d need only be cold enough to cool the outdoor air to the desired supply air temperature, and the resulting room air condition is will be at a relatively low humidity. This allows T_d to be adjusted to minimize energy input while still meeting the room load. T_d is primarily a function of chilled water supply temperature, which has a big influence on chiller input power. Changing chilled water temperature will affect water flow rate and pump power to meet the same cooling load, but fan power will not change for a constant room load. Thus, the chiller and pump characteristics can be used to determine the optimal Td and outdoor air fraction to minimize total energy input to the system when using the economizer cycle.

Similar logic applies to energy recovery systems, which when analyzed as a sub-system, function the same as 100% outdoor air systems (Mumma, 2001). The objective for their control should be to condition the incoming air as close to the supply condition as possible. Achieving this requires activating energy recovery only when useful, or modulating the energy transfer to avoid excessive thermal loads, as shown in figure 6. This figure shows three outdoor air conditions, cool and dry (T_1), cool and humid (T_2), and warm and dry (T_3). The bold dashed lines represent the range of air states that might be output from the energy recovery system for the outdoor air and return air conditions indicated. These are not precise limits, as the amount of sensible and latent heat transfer will vary with the specific type of equipment employed, but serve to illustrate the concept. For the cool humid condition, operating the energy recovery system will increase cooling coil load since the outdoor air enthalpy is less than the return air. For the cool dry case, energy recovery should only be used to warm the outdoor air to the supply air temperature Ts (indicated by the open circle), which will require some capacity control functionality to be utilized. For the warm and dry case, similar to economizer control, energy recovery should be used to bring the exiting air condition to the point where the cooling coil will be dry (indicated by the open circle).

DISCUSSION AND RESULTS

For the purposes of demonstration, the foregoing analysis was implemented for a typical HVAC system using a series of simulations. The simulations assumed constant room sensible and latent loads for a typical cooling condition (Philadelphia, PA, summer design day) for two outdoor air conditions (cool and dry, and warm and dry). Two different chillers were simulated for both VAV and CAV operation. The goal of the simulations was to demonstrate how to determine the optimum setpoints for chilled water temperature and outdoor air fraction for the various combinations of equipment, operating strategy and outdoor condition (see Table 1). The methods could be applied for other conditions or hourly simulations as required for annual energy analysis or design purposes. Table 2 shows the simulation results for the VAV and CAV systems, respectively. For the cool and dry condition, Td of 9 °C was optimum for the cent. chiller, and 11 °C for the recip. chiller, both at 100% outside air. For the warm dry condition, the optimum outdoor air fraction was 68%. Figure 7 shows how total energy input varied with chilled water supply temperature, for these specific loads. These results would likely vary for other part load ratios and outdoor air conditions. Table 3 is the comparison of energy consumption in the hot&dry condition (30 °C and 20% relative humidity) between using economizer with established strategy (using 100% outdoor air) and economizer with the improved control strategy presented in this paper (using 68% outdoor air). The red number is the energy consumption value for optimized condition. According to the simulation results, new approach will save 8.17% and 5.40% energy for VAV system with centrifugal chiller and reciprocating chiller respectively. For CAV system, the energy saving will be even more obvious, 11.15% for centrifugal chiller and 6.75% for reciprocating chiller. With regard to the actual building, the presented control strategy has considerable energy saving potential.

CONCLUSIONS

A method was presented and demonstrated for determining the optimum operating strategy for economizer or energy recovery systems. The method enables the determination of the best outdoor air fraction and apparatus dewpoint temperature to minimize total HVAC system energy input for a specific combination of equipment characteristics and indoor and outdoor air conditions. The method differs from the more typical approaches which are based on either a comparison of return air and outdoor air dry bulb temperatures or enthalpies, or the related concept of minimizing cooling coil thermal load. The difference between this method and the more typical approaches lies in the emphasis on minimizing energy input to the energy consuming system components, including chiller, pumps and fans, by selecting the optimum operating setpoints, which depend both on equipment performance and the relative conditions of the indoor and outdoor air. The use of the energy-based control strategy instead of the coil load based strategy will have an impact for a wide range of climate conditions, but will be most noticeable for less humid locations. Furthermore, the use of partial outdoor air fractions to maintain a dry coil expands the useful application of economizer cycle and energy recovery for warm, dry outdoor air conditions.

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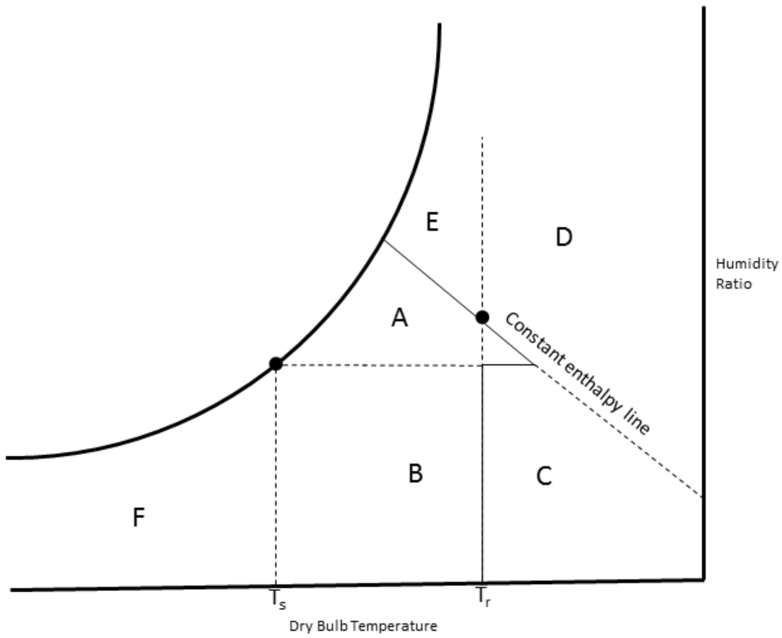

Figure 1 *Typical logic for economizer control*
Region A: 100% outdoor air based on lower enthalpy for the wet coil
Region B: 100% outdoor air based on lower dry bulb temperature for the dry coil
Region C, D and E: set to minimum for outdoor air
Region F: some fraction of outdoor air may be beneficial

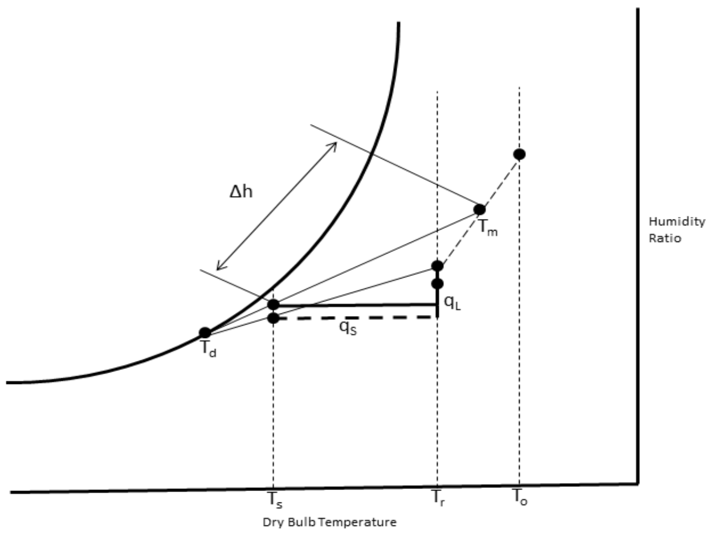

Figure 2 *Temperature and energy balance relationships for mixed outdoor and return air conditions showing shift in room conditions*

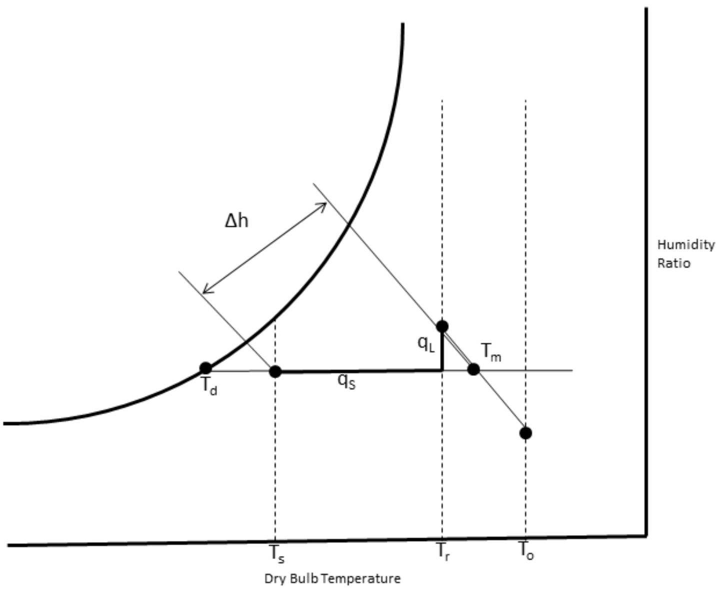

Figure 3 *Optimum outdoor air fraction for warm, dry outdoor air condition*

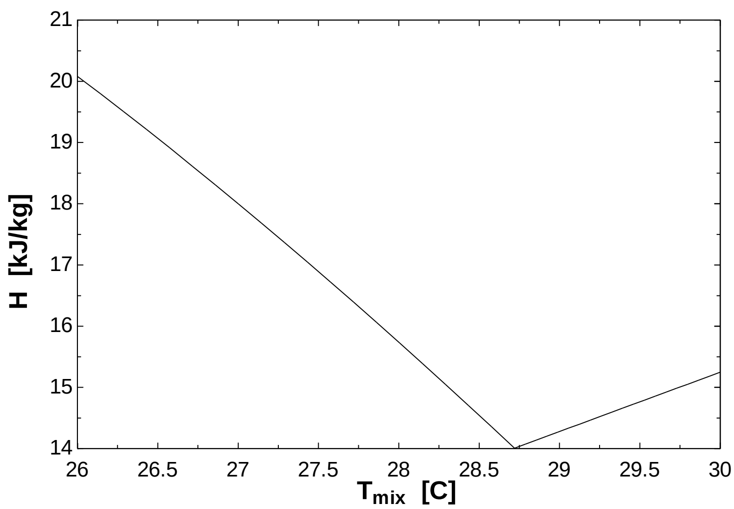

Figure 4 *Simulation result shows the mixed air point in Figure 3 is the optimum mix air point. ΔH=enthalpy of mix air-enthalpy of supply air*

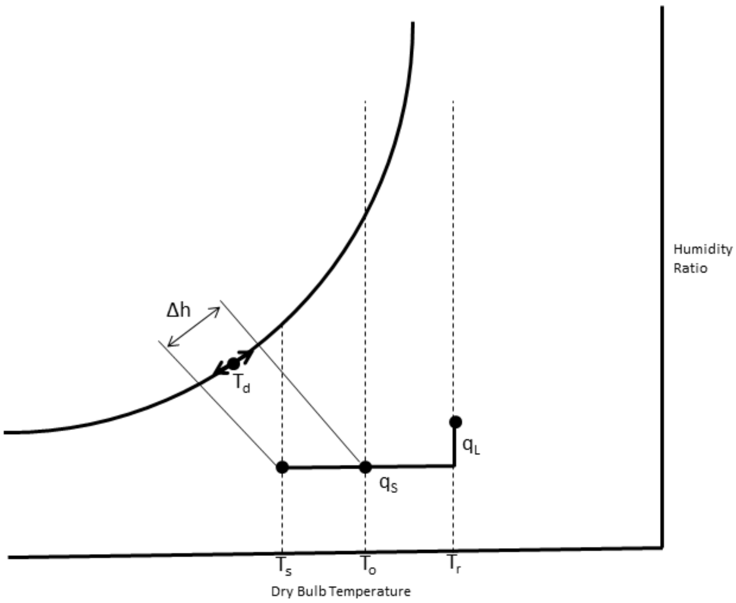

Figure 5 *Allowable variation in apparatus dewpoint temperature for the cool, dry outdoor air condition*

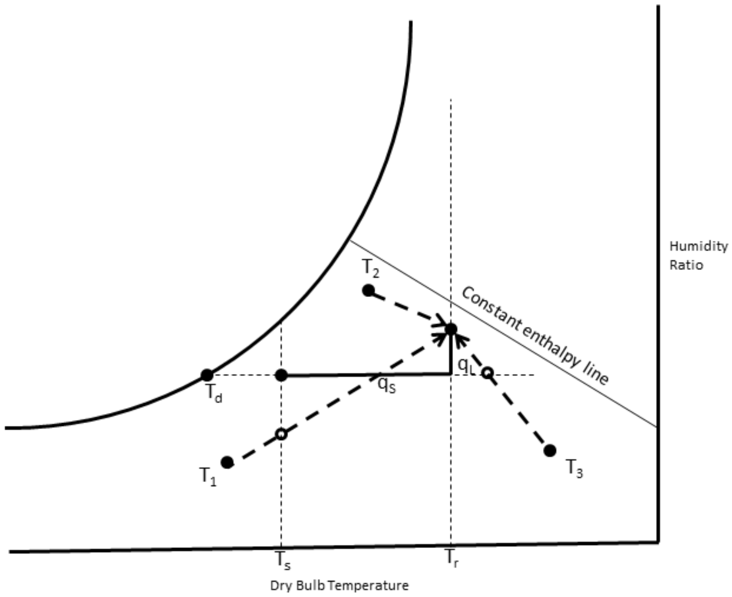

Figure 6 *Operating logic considerations for energy recovery control*
T_1-cool and dry T_2-cool and humid T_3-warm and dry

Table 1 *Reference characteristics of equipment (lcw-leaving chilled water, ecf-entering condenser fluid, chw-chilled water, cdw-condenser water)*

Components	Selected parameters values	
Chillers	25000 Capacity (W)	2.75 COP
	6.67 T_lcw (℃)	29.4 T_ecf (℃)
	0.002 V_chw (m^3/s)	0.0023 V_cdw (m^3/s)
Variable Volume Fan	2.15 Rated Flow rate (m^3/s)	1837 Rated Power (W)
	600 Pressure Rise (Pa)	0.7 Fan Efficiency
Variable Speed Pump	0.0012 Rated Flow rate (m^3/s)	500 Rated Power (W)
	300 Pump head (KPa)	0.66 Pump Efficiency

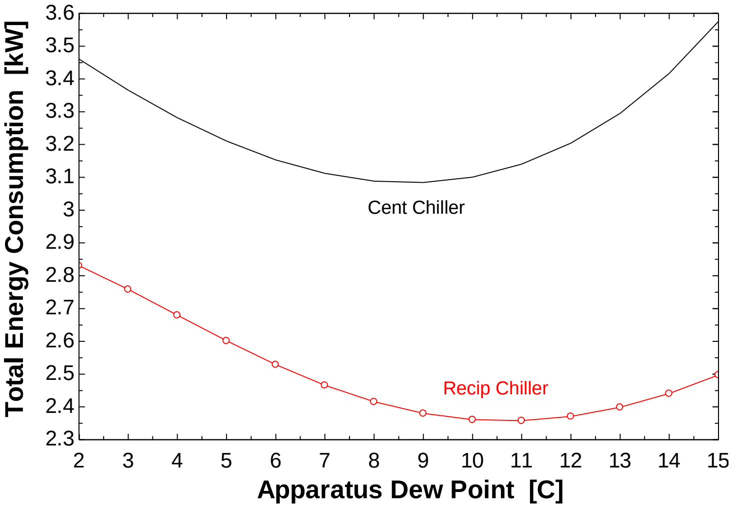

Figure 7 *The variation in total system energy input with chilled water temperature for both chillers for VAV system in cool & dry condition*

Table 2 *Energy consumption comparison of VAV & CAV system using centrifugal and reciprocating chiller*
Room temperature is 26 ℃ and supply air temperature is 15 ℃. CAV system constant air flow rate is 2 m^3/s.

Chiller Type	Outdoor T(°C) &RH	Optimum Outdoor air Fraction &T_mix (°C)	T_d (°C)	VAV System				CAV System			
				Fan Power (W)	Pump Power (W)	Chiller Power (W)	Total Power (KW)	Fan Power (W)	Pump Power (W)	Chiller Power (W)	Total Power (KW)
Cent Chiller	18&30%	100% 18	9	792.7	33.44	2258	3.084	1605	88.46	2984	4.678
	30&20%	68% 28.72	10	985.9	304.4	6940	8.231	1605	335	7745	9.685
Recip Chiller	18&30%	100% 18	11	792.7	42.99	1793	2.358	1605	113.7	3022	4.741
	30&20%	68% 28.72	10	985.9	304.4	7724	9.014	1605	335	8276	10.22

Table 3 *Energy consumption comparison between the established control approach (100% outdoor air) and the improved method (68% outdoor air)*
Room temperature is 26 ℃ and supply air temperature is 15 ℃. CAV system constant air flow rate is 2 m^3/s.

Chiller Type	Outdoor T(°C) &RH	Outdoor air Fraction &T_mix (°C)	T_d (°C)	VAV System				CAV System			
				Fan Power (W)	Pump Power (W)	Chiller Power (W)	Total Power (KW)	Fan Power (W)	Pump Power (W)	Chiller Power (W)	Total Power (KW)
Cent Chiller	30&20%	*100%& 30	10	985.9	331.4	7646	8.963	1605	376.2	8915	10.9
		68%& 28.72		985.9	304.4	6940	8.231	1605	335	7745	9.685
Recip Chiller		*100%& 30		985.9	331.4	8211	9.529	1605	376.2	8977	10.96
		68%& 28.72		985.9	304.4	7724	9.014	1605	335	8276	10.22

**conventional control strategy*