

# Energy Aspects of HVAC System Configurations — Problem Definition and Test Cases

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*This paper reports on the energy implications of HVAC system configuration by analyzing the energy balance and psychrometrics of typical and innovative systems. Three criteria were shown to be significant: (1) the ability to minimize outside air load, (2) the ability to eliminate simultaneous cooling and heating and use mixing effectively, and (3) the availability of interzonal airflow. Configurations that meet these criteria would be able to deliver the desired indoor air quality with reduced energy consumption. The performance of ten two-zone system configurations, including single-duct, dual-duct, fan-coil-based variations, and other specialized systems in the literature, were analyzed for a number of operational conditions. The results confirmed that fan-coil-based configurations with interzonal airflow paths perform better than other configurations. The conclusion of this study may be used as a guideline for multi-zone system designs.*

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## INTRODUCTION

In the early stages of HVAC system design, the choice of configuration is one of the decisions that significantly affects the performance of the final system, involving design options such as zone setting, choice of equipment, arrangement of air circulation, and operation strategy. A “good” configuration enables the system to provide a high quality indoor environment with minimum cost and environmental impact. ASHRAE (2000) provides a list of typical HVAC systems, including single-zone and multi-zone systems; single-duct, dual-duct, and multi-deck systems; constant air volume (CAV) and variable air volume (VAV) systems; and packaged systems, such as fan-coil units and unitary systems. The performance of these systems, including energy efficiency and indoor air quality (IAQ) impacts, has been extensively studied. Novel HVAC system schemes have also been developed to deliver better IAQ with less thermal energy consumption (Cui et al. 2003; Song and Liu 2004). However, general criteria for “good” configuration designs, or the common aspects of configurations that affect the performance of the final system, have yet to be identified.

Apparently, the difficulty lies with the complexity of HVAC systems. The performance of an HVAC system depends on many factors, including characteristics of the building, ambient conditions and internal design requirements, system type, efficiency of the primary system and the air distribution system, control strategy, and individual components. One typical approach adopted by many researchers is to simulate the performance of the system as a whole or when integrated with a building model. By providing climatic and usage data, annual performance of

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the particular system can be analyzed. Since all factors are included in the analysis, the result is really only applicable to the design situation at hand. The drawback of this approach, then, is that the result is bound to the particular context of the building and climate. As a result, it is difficult to apply the findings to other circumstances.

On the other hand, a different approach is often adopted in studies of the performance of individual components. The component in question is isolated from the rest of the system and a parametric analysis is performed on the identified parameters to evaluate the impact of each factor. With this approach, not only can the performance in various circumstances be revealed, but it is also possible to identify the reason, or the main factors, driving the good or poor performance. In this paper, the research to identify the general criteria for the optimal HVAC system configuration is presented. In order to provide in-depth analysis without interference from other factors, the air-side secondary system is isolated from the primary system, the building, and any physical details of its components (idealistic models are used). For the same reason, the efficiency of air distribution is not included. A single objective for the optimal design is pursued, which is to minimize energy consumption for air conditioning while maintaining IAQ by fulfilling the minimum fresh air requirement.

Although HVAC systems vary in schematic design, operational strategy, working medium, and packaging, it is possible to represent the configuration of these systems as a set of basic psychrometric processes connected by airflows (Silverman et al. 1981). This paper studies various aspects of HVAC system configurations using psychrometric and energy balance analysis. Ten “typical” and “innovative” configurations are evaluated with a variety of test cases in heating, cooling, and free-cooling conditions. Configurations that have near-optimal performance are identified for two-zone systems. The contributions of the paper include establishing three general criteria for an optimal configuration as well as providing detailed analysis of the factors that have an impact on the performance of each configuration.

## SECONDARY HVAC SYSTEM CONFIGURATION

The secondary HVAC systems are the air-conditioning and distribution systems that have means to heat, cool, humidify, dehumidify, clean, and distribute air to the various conditioned spaces in a zone. Secondary HVAC systems are generally categorized into all-air systems, air-water systems, all-water systems, and unitary systems. In all-air systems, the energy is carried to the conditioned space solely by ducted air. In all-water systems, the energy is carried to the conditioned space by piped liquid, usually water. In this case, the energy transfer occurs directly with the air in the occupied space. Air-water systems, however, utilize a combination of air and water for transporting energy. Both fluids are distributed to each space to perform the cooling/heating function. Unitary systems are factory-matched primary and secondary components in single packages.

From the perspective of air-conditioning and distribution processes, the secondary HVAC systems can be described by their physical realization (single/dual/multi-duct), airflow control strategy (constant/variable volume), and strategies for handling multiple zones and the recirculation of exhaust air. Although a large variety of different systems has evolved to meet the diversity in building thermal loads, cost limitations, and expectations for thermal comfort, all existing systems and specialized components (such as fan-coil units) can be represented by a series of elementary air-handling processes. In this paper, the analysis is focused on the air-handling processes and how they are organized into different system configurations. As illustrated in Figures 1 through 9, ten configurations (including two symmetrical configurations in Figure 5) for a two-zone system were selected for analysis.

The configuration in Figure 1 represents a two-zone single-duct VAV system (SDUCT) (ASHRAE 2000). The small circles and boxes in the figure represent air mergers and splits,

respectively. Outside air is mixed with the return air and subsequently conditioned by the heating and cooling coils in the central plant before being supplied to each zone. The zones are equipped with reheat terminals, which also control the humidity ratio of the supplied air. It is worth noting the presence of a second recirculation path after the main air-handling unit (AHU) and before the terminals. Although the purpose of the bypass is to “maintain supply air quantity under minimum load conditions” (ASHRAE 2000), it also provides the possibility to utilize mixing to offset the requirement for reheating in some operational conditions. This will be further discussed in the section “Simultaneous Cooling and Heating.” Figure 2 represents a two-zone dual-duct VAV system (DDUCT) (ASHRAE 2000). Similarly, humidifiers are added

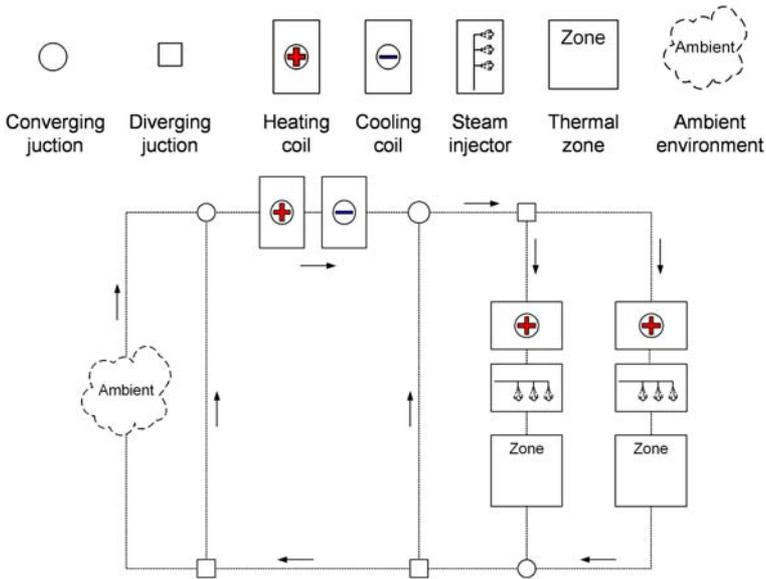


Figure 1. Single-duct system configuration (SDUCT).

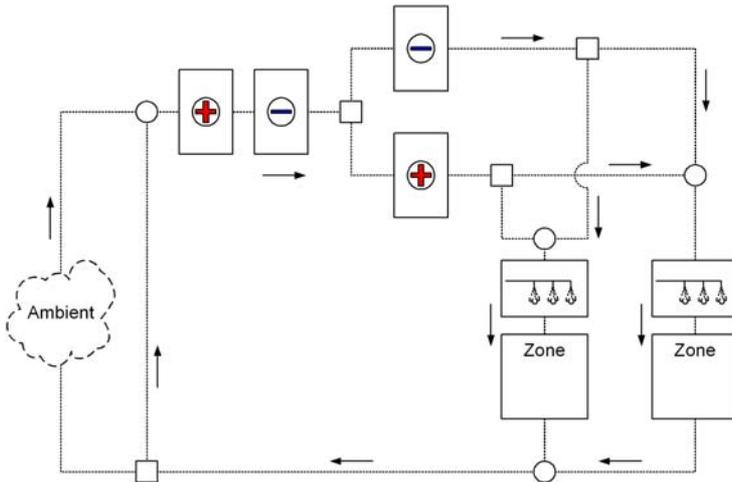
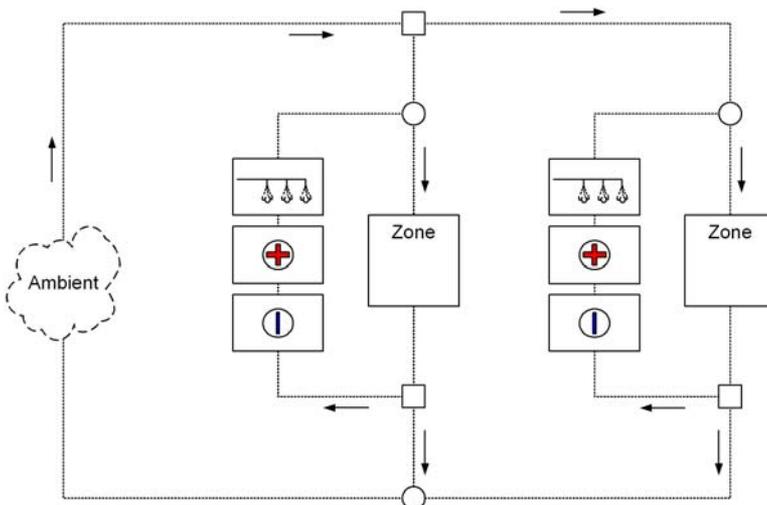


Figure 2. Dual-duct system configuration (DDUCT).

to provide humidity control. The single-duct and the dual-duct designs are the most common configurations of VAV systems. In this study, however, it is assumed that the setpoints for both cold and warm air decks can be reset to achieve optimum system performance under each design condition.

Figure 3 represents a fan-coil-based system configuration (FCOIL). Each zone is equipped with one fan-coil unit, comprising heating and cooling coils and a humidifier. Fresh air is supplied untreated to the zones. As there is no circulation between the two zones, it can be considered as two separate single-zone systems. The configurations in Figures 4–6 are derived from the fan-coil system, with recirculation paths inserted between zones. In Figure 4, the two zones are arranged in parallel as the extract air from both zones is mixed before being recirculated and/or discharged to the ambient (FCOILP). In Figure 5, the two zones are arranged sequentially, where extract air from the first zone is directed to the second before exhausting to the ambient. Part of the extract air from the second zone can be recirculated to the first zone. Depending on the location of the exhaust leg, the order of the zones can be arranged as zone 1 → zone 2 (FCOILS1) or zone 2 → zone 1 (FCOILS2). An extra air split junction is added in the configuration in Figure 6, which enables the extract air to be exhausted from either zone 1 or zone 2 by manipulating airflows in the system (FCOILX).

Specialized HVAC system configurations can be found in the literature. Song and Liu (2004) reported an integrated AHU system for large office buildings with distinct interior and exterior zones. The configuration of the patented office air-handling unit (OAHU) system is illustrated in Figure 7. The extract air from the interior zone may be channelled to the exterior zone before exhaust. Cui et al. (2003) reported an innovative AHU design for buildings that have both office and laboratory sections. The configuration of the laboratory air-handling unit (LAHU) system is illustrated in Figure 8. Since the exhaust air from the laboratory section of the building cannot be recirculated, a through path from the office section to the laboratory section is proposed in order to reduce the outside air load. In comparison, the traditional configuration for such applications comprises separated systems for office and laboratory, as represented in Figure 9 (LAHU0). The features of the configurations and their performance under a number of specific operational conditions are analyzed in this paper.



**Figure 3. Separated fan-coil system configuration (FCOIL).**

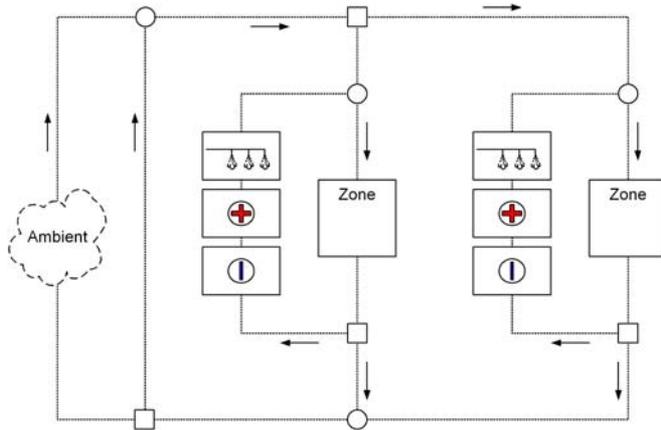


Figure 4. Fan-coil system—parallel arrangement (FCOILP).

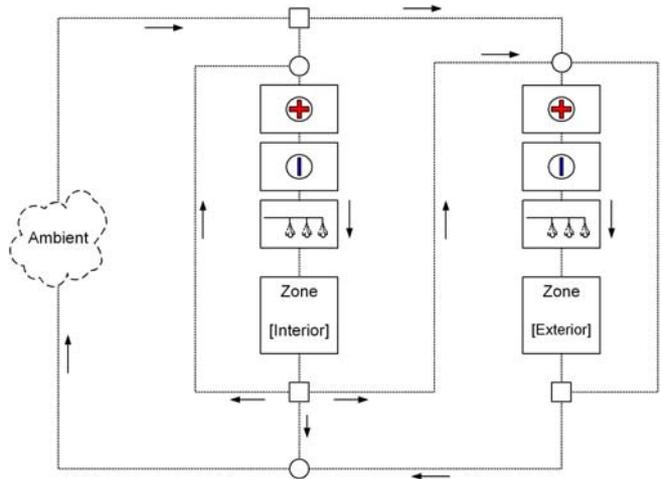


Figure 5. Fan-coil system—sequential arrangement (FCOILS).

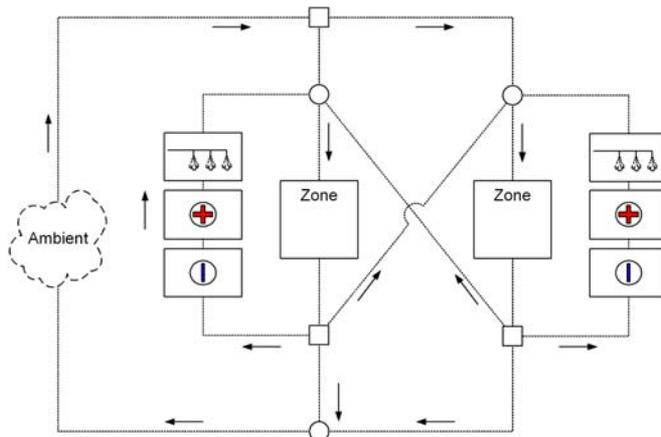


Figure 6. Fan-coil system with interzonal airflow (FCOILX).

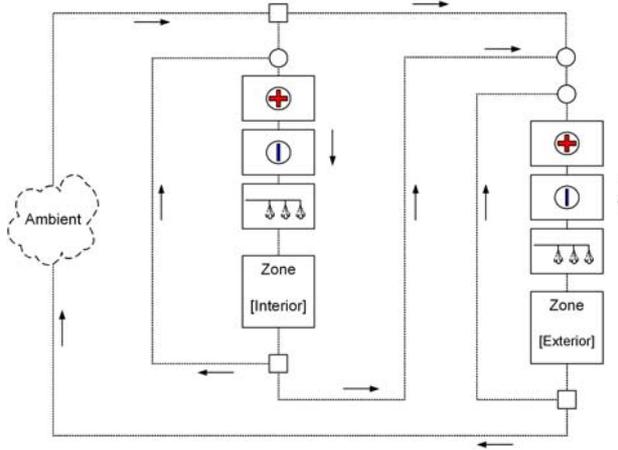


Figure 7. OAHU (Song and Liu 2004) system configuration (OAHU).

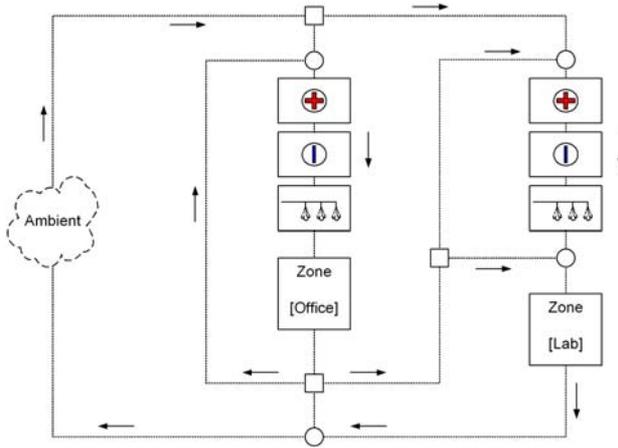


Figure 8. LAHU (Cui et al. 2003) system configuration (LAHU).

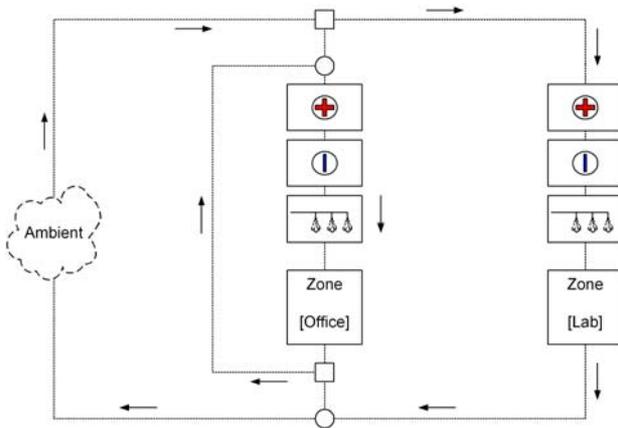


Figure 9. Separated office and laboratory systems configuration (LAHU0).

### ENERGY ANALYSIS

The energy balance of an HVAC system is shown in Figure 10. Three groups of thermal energy flows are identified, including the thermal load in the zones ( $\dot{Q}^{Load}$ ), the active heating/cooling input ( $\dot{Q}^{H/C}$ ), and the heat gain/loss due to air exchange with the ambient environment ( $\dot{Q}^{O/A}$ ). The objective of energy-efficient HVAC configuration design is to minimize the heating and cooling duties by reducing (1) outside air load, (2) zone load, and (3) the energy wasted during the air-conditioning process.

#### Outside Air Load

During temperate seasons, free cooling (when the enthalpy of outside air is lower than that inside) is an energy-efficient way of removing excess heat from a building. In winter and summer, however, fresh air supply is often a significant part of the system load. The outside air load is calculated by the enthalpy difference between ambient air and zone air, multiplied by mass flow rate. Therefore, it can be minimized by either reducing outside airflow, reducing the enthalpy difference, or both. The minimum outside airflow required to maintain IAQ of the conditioned zones is usually one of the control targets during summer/winter operations. For some multi-zone system configurations, however, the intake of outside air cannot be minimized in certain conditions due to the possible confliction between distribution of fresh air and the heating/cooling loads.

Figure 11 illustrates an example where the outside air intake cannot be minimized. The two zones in the single-duct configuration have the same minimum fresh air demand ( $\dot{m}_{min}^{O/A}$ ), whereas their sensible cooling loads are different ( $\dot{Q}_{s2} = 2 \times \dot{Q}_{s1}$ ). Without a reheating coil at the supply terminal to zone 1, the system satisfies the cooling demands in both zones by supplying zone 2 twice the air volume as that supplied to zone 1. Assuming that the minimum outside air mass flow rate  $\dot{m}_{min}^{O/A}$  is supplied to zone 1, the amount of outside air that zone 2 receives is, hence,  $2\dot{m}_{min}^{O/A}$ . In total,  $3\dot{m}_{min}^{O/A}$  is drawn from ambient, instead of  $2\dot{m}_{min}^{O/A}$ , resulting in 50% higher outside air load.

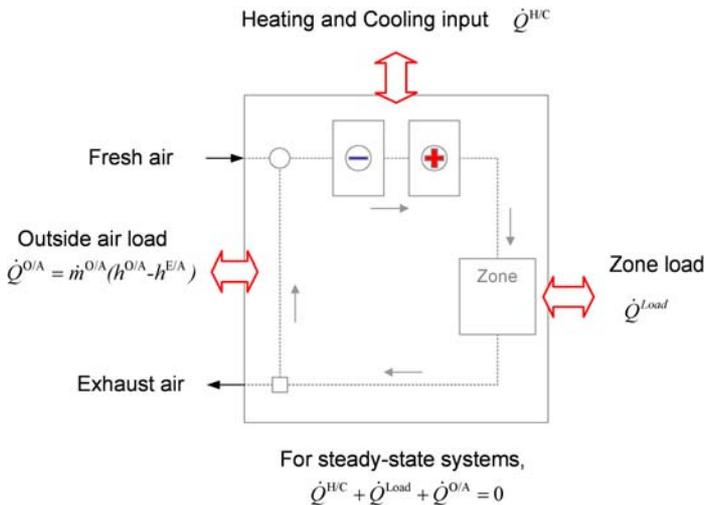


Figure 10. Energy balance of HVAC system.

The dual-duct systems (Figure 2) have a similar problem when the loads and outside air demands for each zone are out of proportion. Such limitation applies to all multi-zone systems where cooling/heating loads and outside air distribution are handled centrally. This also explains why some multi-zone systems suffer from poor indoor air quality during part-load operation. The outside air inlet openings of such systems are often fixed throughout the season, whereas the variations in zone loads change the distribution of outside air. As a result, the zones with lower load are prone to IAQ problems. Separate zone load handlers and dedicated fresh air circuits are preferred to avoid this problem.

The second approach to minimize outside air load is to reduce the enthalpy difference between the ambient air and the extract air. Imagine the two zones at different design indoor air conditions ( $T_1$  and  $T_2$ ) in Figure 12: the sensible load for outside air intake ( $\dot{Q}_s^{O/A}$ ) is calculated as  $\dot{Q}_s^{O/A} = \dot{m}^{O/A} C_p (T_{db}^{O/A} - T_2)$ , where  $\dot{m}^{O/A}$  and  $T_{db}^{O/A}$  are mass flow rate and dry-bulb temperature of the outside air. The two zones need to be arranged so that the design temperature of zone 2 is closer to the ambient temperature in order to reduce outside air load to the system.

**Interzone Airflow and Load Offset**

Each zone served by a multi-zone HVAC system can be considered a heat source or sink to the air within the system. Provided that at least two zones have opposite loads (heating vs. cool-

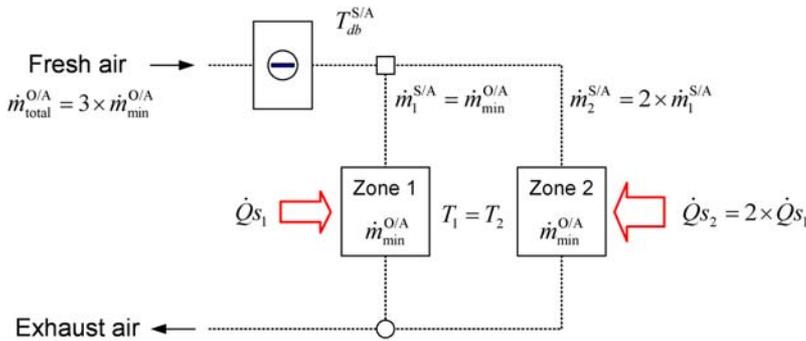


Figure 11. Fresh air requirements vs. cooling demands.

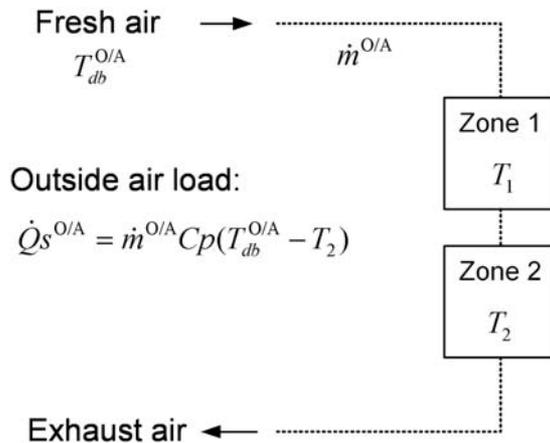
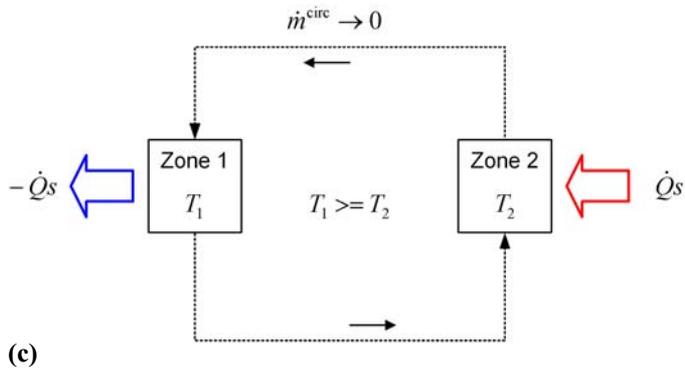
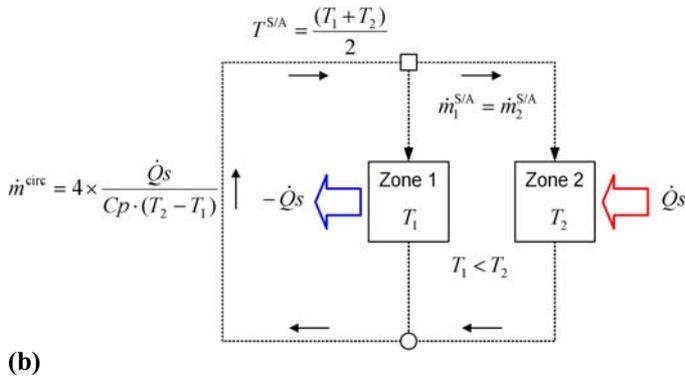
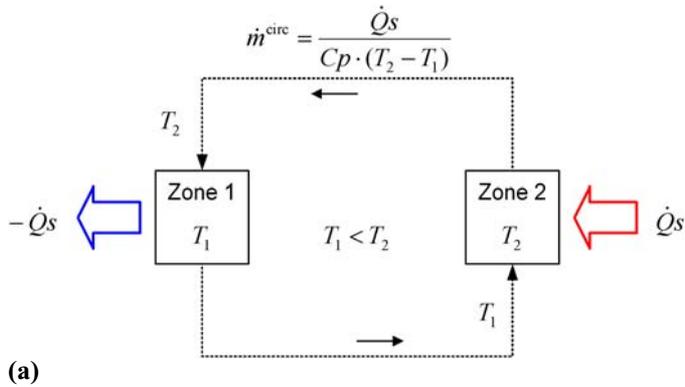


Figure 12. Minimizing enthalpy difference between fresh and extract air.

ing, or humidifying vs. dehumidifying), it may be possible to offset the zone loads. Interzonal load offset could be achieved by using a heat pump or simply circulating air between zones, though restrictions apply in the latter case.

Take the example in Figure 13, where the two zones are maintained at  $T_1$  and  $T_2$ , respectively. Assume that the sensible heat gain to zone 2 is  $\dot{Q}_s$ , whereas zone 1 has a sensible heat loss of  $-\dot{Q}_s$ . If  $T_1 < T_2$  (case A in Figure 13), the heating and cooling loads in both zones 1 and 2 can be offset by circulating air between them. The required circulation mass flow rate is calculated as



**Figure 13. Load reduction by interzonal airflow.**

$\dot{m}^{circ} = \frac{\dot{Q}_s}{C_p \cdot (T_2 - T_1)}$ . An economizer that recirculates the return air can serve for interzonal load offset too (case B in Figure 13). Since the return air temperature  $T^{S/A} = \frac{(T_1 + T_2)}{2}$ , the supplied air mass flow rate required to offset the load in each zone is doubled; therefore, the total mass flow rate of recirculation is  $\dot{m}^{circ} = 4 \times \frac{\dot{Q}_s}{C_p \cdot (T_2 - T_1)}$ , implying higher fan energy cost. If  $T_1 > T_2$ , as shown in case C, however, passing air from zone 1 to zone 2 would increase the cooling load to zone 2 and vice versa. As a result, interzonal circulation is not recommended in this case.

The above example demonstrates the situation for opposing sensible loads. If the latent load is considered, the benefit of interzonal circulation and the optimal mass flow rate is more difficult to determine. The capability of interzonal circulation, however, should be reserved in configuration design to exploit the potential of energy saving.

### Simultaneous Cooling and Heating

Several established multi-zone system configurations are designed with zonal reheat, e.g., the single-duct system in Figure 1. The terminal reheat coils are used to compensate for part-load conditions in the zones. With a CAV control strategy, the operation of the cooling coil is determined by the seasonal setpoint for the cold deck, whereas the reheating coils are employed to meet the variation of zone load. With VAV control, reheating is also required during part-load operation in order to maintain the minimum fresh air supply. The dual-duct system illustrated in Figure 2 may face the same problem during part-load operation. Mixing of actively cooled and heated air from the cold and warm decks in the system is also wasteful.

Simultaneous dehumidifying and humidifying can also happen with centralized multi-zone systems. Consider two zones requiring supply air at different humidity ratios: the central air-handling plant can only reset the supply deck according to the highest dehumidification requirement. The condition of the supply air to the zone with lower dehumidification demand has to be achieved with a terminal humidifier or mixing with humid air. The single-duct configuration is more prone to excess energy consumption in such circumstances.

Another reason simultaneous cooling and heating are often present is due to the limitations of the psychrometric processes of air conditioning. Consider the application of a cooling coil for dehumidification. The air is first cooled to below the required dew point to remove excess moisture content and then reheated to the desired temperature. Desiccant-based humidification involves simultaneous heating and cooling, too. Chemical dehumidification generally increases air temperature when reducing moisture content. Cooling is required to cool the treated air and the used desiccant has to be regenerated, usually by heating. Though it is arguable that free cooling, heat recovery, as well as renewable (solar) energy can be used in desiccant dehumidification to make the process energy efficient, this is outside the scope of this discussion.

In order to avoid simultaneous cooling and heating, the operating point of each component in the system has to be optimized, and mixing should be used when possible. The idea of mixing is to make best use of air at its present condition rather than actively process it to other conditions. Figure 14 shows the psychrometrics of the air-conditioning process in summer operation. In order to achieve the required supply air humidity ratio (B), it is necessary to overcool the air to B1'. Air at B1' must be reheated (B1' → B) before being supplied to the zone, which results in simultaneous cooling and heating in the system. The alternative process 2 uses mixing in place of reheating. The air from condition point A is further cooled to condition B2', which is in line with supply condition B and zone condition C. Then the air at B2' is mixed with C at a ratio of about 1:3 to achieve condition B. Providing the volume of supply air at condition B is the same as in process 1; the volume of air handled by the cooling coil is only one-fourth compared with



from B1. When the second bypass is present, process 2 starts with mixing outside air OA and return air Z to condition A2. The mixed air is then cooled to a lower dew point at B2. The second bypass is used to mix air at B2 with return air Z to the same humidity ratio (C2) as required for supply air. Though reheating is still required in this process, the amount of energy required is significantly reduced.

### Criteria for an “Optimum” Configuration

In summary, a conceptual optimal configuration should have the following features:

- separate air conditioning for each zone; therefore, outside airflow can be minimized
- arrangement of zones to minimize enthalpy difference between inlet and exhaust air
- presence of recirculation paths that allow interzonal load offset
- the ability to avoid simultaneous heating and cooling and simultaneous humidifying and dehumidifying in all circumstances

The performance of the ten secondary HVAC system configurations illustrated in Figures 1–9 is evaluated below with a set of deliberately diverse test design conditions. The configuration that is closest to the optimum is identified.

### EXPERIMENT DESIGN

In order to validate the conclusions of the conceptual analysis, an experiment was designed to evaluate the performance of the configurations. A number of diverse operational conditions have been specified as the test cases, and the energy efficiency of each configuration has been assessed. Since indoor air quality is one of the main issues in HVAC system design, the supply of outside air that meets the minimum fresh air requirement of each zone has to be guaranteed in each and every operational condition in which a configuration is tested. The operation of each configuration, which includes airflow rate in each branch and heating, cooling, or humidification duty of each component, is optimized. In this way, the minimum energy consumption that is required to satisfy the design requirements, such as zone temperature and humidity setpoints, can be analyzed. In these circumstances, the optimization of flow rates and component duties can be considered the function of an instantaneous optimal supervisory controller.

The methods for modeling, simulation, and optimization used in this study are similar to those described by Zhang (2005). In brief, the configurations are modeled as a set of ideal psychrometric processes, which are linked by airflows. The capacity of each air-handling component is defined as the required thermal input to the airflow, without consideration of the form of energy or the efficiency of the primary system. The total energy consumption of a configuration in each test case is the sum of the absolute value of the heating/cooling/humidification duties of the components in kilowatts. Since neither the efficiency of the primary system nor the energy use for air distribution is considered in this experiment, and also as airflow rates and component duties are optimally controlled, a carefully chosen set of test cases should reveal the relative overall performance of the test HVAC systems that is solely a result of their configuration design features.

The choice of test cases is based on the coverage of the diverse but feasible operational conditions that the HVAC systems may handle. According to the conceptual analysis, the requirement of outside air, the distribution of zonal loads, and the internal design conditions of different zones are the major challenges to the configuration design. Four sets of test cases were devised. The first set of test cases contains both summer and winter operational conditions. The outside air requirement to each zone is 0.064 kg/s, whereas the internal design conditions and the heating/cooling loads of the zones vary from case to case. In the second set, however, the minimum

outside air requirement for zone 1 is significantly lower than that for zone 2. Such a situation can happen when the two zones experience different occupation densities, such as one as an office and the other as a meeting room. Test case set 3 contains operational conditions in which a heating load is present in one zone with a cooling load in the other. In winter, these conditions often occur in perimeter/interior zones or east/west-facing zones where window solar gains vary significantly between morning and afternoon. These cases are to test the capacity of the HVAC configurations utilizing interzonal load offset. The outside air load is removed deliberately from the system; therefore, the effect of interzonal circulation can be isolated. The last set of operational conditions is assumed to occur during intermediate seasons when free cooling is available.

In order to quantify the outside air load to the system, summer weather data were taken from Shanghai, China (hot and humid). Montreal, Canada (cold and dry), was chosen for the winter weather data. Weather conditions representing intermediate seasons were selected from climatic data for London, UK. The ambient weather conditions are listed in Table 1. The test operational conditions, including internal design conditions, sensible and latent loads, and minimum fresh air requirements, are listed in Table 2.

**Table 1. Ambient Design Conditions**

Season	Weather data	$T_{db}$ (°C)	$T_{wb}$ (°C)	RH (%)	w (g/kg)	h (kJ/kg)
Summer	Shanghai, China	30.3	27.68	82	22.53	87.93
Winter	Montreal, Canada	-26.7	-26.7	100	0.3276	-25.9
Free cooling	London, UK	16.6	11.71	56	6.59	33.28

**Table 2. Internal Design Conditions**

Cases	Season	Zone 1					Zone 2					Note
		$T_{db}$ (°C)	RH (%)	$\dot{Q}_s$ (kW)	$\dot{Q}_l$ (kW)	$\dot{m}_{min}^{O/A}$ (kg/s)	$T_{db}$ (°C)	RH (%)	$\dot{Q}_s$ (kW)	$\dot{Q}_l$ (kW)	$\dot{m}_{min}^{O/A}$ (kg/s)	
Set 1	1.1 Summer	22.0	60	5.00			22.0	60	5.00			
	1.2 Winter	20.0	40	-2.50			20.0	40	-2.50			
	1.3 Summer	24.0	60	5.00			20.0	60	5.00	0.40	0.064	$T_{db1} > T_{db2}$
	1.4 Winter	22.0	40	-2.50			18.0	40	-2.50			$T_{db1} > T_{db2}$
	1.5 Summer	22.0	60	8.00			22.0	60	2.00			$\dot{Q}_{s1} > \dot{Q}_{s2}$
	1.6 Winter	20.0	40	2.50			20.0	40	-2.50			$\dot{Q}_{s1} = -\dot{Q}_{s2}$
Set 2	2.1 Summer	22.0	60	5.00			22.0	60	5.00			
	2.2 Winter	20.0	40	-2.50			20.0	40	-2.50			
	2.3 Summer	24.0	60	5.00			20.0	60	5.00			$T_{db1} > T_{db2}$
	2.4 Summer	20.0	60	5.00			24.0	60	5.00	0.60	0.096	$T_{db1} < T_{db2}$
	2.5 Summer	22.0	60	8.00	0.20	0.032	22.0	60	2.00			$\dot{Q}_{s1} > \dot{Q}_{s2}$
	2.6 Summer	22.0	60	2.00			22.0	60	8.00			$\dot{Q}_{s1} < \dot{Q}_{s2}$
	2.7 Winter	20.0	40	2.50			20.0	40	-2.50			$\dot{Q}_{s1} = -\dot{Q}_{s2}$
	2.8 Winter	20.0	40	-2.50			20.0	40	2.50			$-\dot{Q}_{s1} = \dot{Q}_{s2}$
Set 3	3.1 Winter	20.0					20.0					
	3.2 Winter	22.0	40	-2.50	0.0	0.0	18.0	40	2.50	0.0	0.0	$T_{db1} > T_{db2}$
	3.3 Winter	18.0					22.0					$T_{db1} < T_{db2}$
Set 4	4.1 Free cooling	22.0		5.00	0.4	0.064	22.0		5.00	0.4	0.064	
	4.2 Free cooling	24.0		5.00	0.2	0.032	20.0		5.00	0.6	0.096	$T_{db1} > T_{db2}$
	4.3 Free cooling	20.0	60	5.00	0.2	0.032	24.0	60	5.00	0.6	0.096	$T_{db1} < T_{db2}$
	4.4 Free cooling	22.0		8.00	0.2	0.032	22.0		2.00	0.6	0.096	$\dot{Q}_{s1} > \dot{Q}_{s2}$
	4.5 Free cooling	22.0		2.00	0.2	0.032	22.0		8.00	0.6	0.096	$\dot{Q}_{s1} < \dot{Q}_{s2}$

**RESULTS AND DISCUSSION**

The results of the evaluation of ten configurations from Figures 1–9 are summarized in Tables 3–8. The configurations can be divided into three groups. The first group includes the single-duct (SDUCT) and the dual-duct (DDUCT) configurations. These configurations represent typical centralized multi-zone systems, which handle air-conditioning demand at a central plant and then distribute air to different zones. According to the analysis presented in this paper, it is anticipated that these systems may be incapable of minimizing outside air load in certain operational conditions. The second group comprises a variety of fan-coil-based configurations in which the air-conditioning demand of each thermal zone is handled separated by a dedicated AHU. The patented OAHU configuration is included in this group. The last group of configurations is for laboratory-office air conditioning, where the extract air from the laboratory space cannot be recirculated. A typical system (LAHU0) is compared to the innovative design from the literature (LAHU), which utilizes the extract air from the office area to reduce the overall load.

Table 3 compares the actual outside air intake of the configurations in the first set of test cases. Numbers show the excess outside air drawn from ambient environment compared to the minimum fresh air demand. Since the outside air demands of the two zones are equal in these cases, most of the configurations can minimize the outside airflow rate, including the dual-duct configuration. The single-duct configuration, however, failed on the two summer conditions in which either zone design temperature or sensible load differed between zones. Similar situations in winter did not affect the outside airflow rate of SDUCT because of the presence of terminal reheating coils that effectively functioned as separated air handlers.

Table 4 summarizes the comparison of excess outside airflow in test case set 2. The ratio of minimum fresh air demand between zones 1 and 2 is 1:3. The imbalance of distribution of load and fresh air demand between the zones is the dominant reason that the single-duct and dual-duct systems drew significant excess outside air over that required, which consequently led to the high energy consumption figures in Table 6. Again, the impact was less significant in the

**Table 3. Comparison of Actual Outside Air Intake in Test Case Set 1**

Cases	Excess outside air intake (kg/s)									
	SDUCT	DDUCT	FCOIL	FCOILS1	FCOILS2	FCOILP	FCOILX	OAHU	LAHU	LAHU0
1.1 Summer	0	0	0	0	0	0	0	0	0.564	0.650
1.2 Winter	0	0	0	0	0	0	0	0	0.218	0.352
1.3 Summer $T_{ab1} > T_{ab2}$	0.075	0	0	0	0	0	0	0	0.564	0.628
1.4 Winter $T_{ab1} > T_{ab2}$	0	0	0	0	0	0	0	0	0.218	0.282
1.5 Summer $\dot{Q}_{s1} > \dot{Q}_{s2}$	0.088	0	0	0	0	0	0	0	0.149	0.219
1.6 Winter $\dot{Q}_{s1} = -\dot{Q}_{s2}$	0	0	0	0	0	0	0	0	0.218	0.282

**Table 4. Comparison of Actual Outside Air Intake in Test Case Set 2**

Cases	Excess outside air intake (kg/s)									
	SDUCT	DDUCT	FCOIL	FCOILS1	FCOILS2	FCOILP	FCOILX	OAHU	LAHU	LAHU0
2.1 Summer	0.061	0.011	0	0	0	0	0	0	0.564	0.596
2.2 Winter	0.003	0.002	0	0	0	0	0	0	0.218	0.250
2.3 Summer $T_{ab1} > T_{ab2}$	0.008	0.070	0	0	0	0	0	0	0.564	0.596
2.4 Summer $T_{ab1} < T_{ab2}$	0.156	0.039	0	0	0	0	0	0	0.564	0.608
2.5 Summer $\dot{Q}_{s1} > \dot{Q}_{s2}$	0.149	0.042	0	0	0	0	0	0	0.149	0.181
2.6 Summer $\dot{Q}_{s1} < \dot{Q}_{s2}$	0.011	0	0	0	0	0	0	0	0.979	1.011
2.7 Winter $\dot{Q}_{s1} = -\dot{Q}_{s2}$	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.218	0.270
2.8 Winter $-\dot{Q}_{s1} = \dot{Q}_{s2}$	0.001	0.001	0	0	0	0	0	0	0.218	0.250

winter season, due to the presence of terminal reheating coils in SDUCT and DDUCT configurations. For the summer condition of case 2.6, in which the sensible cooling load is  $\dot{Q}_{s1} : \dot{Q}_{s2} = 1 : 4$ , the imbalance was less significant. Therefore there was less excess outside air intake.

One interesting case is 2.7 (winter condition). All configurations (except the laboratory systems) agreed on drawing 0.02 kg/s excess outside air in order to minimize system duty. This was because zone 1 had a cooling load of 2.5 kW, which can be offset by free cooling with outside air. The quantity of outside air required, however, exceeds the minimum fresh air requirement for zone 1 (0.032 kg/s). As a result, excess outside air is taken. Contrarily, in case 2.8, where zone 2 had a 2.5 kW cooling load and required 0.096 kg/s of fresh air, no excess outside air is taken and the total system heating duty was lower than that for case 2.7 (see Table 6). If the design temperature of zone 2 were lower than that of zone 1 in case 2.7, it would be possible with some of the configurations to circulate air between the zones to offset the simultaneous cooling and heating loads. This is illustrated by the results of test case set 3.

Consider the energy consumption results (Table 5) and the first set of test cases. Here the minimum fresh air requirement to both zones is equal, and figures for the group of decentralized configurations are similar. The system duties for the single-duct and dual-duct systems are higher than for the fan-coil-based systems and the OAHU system in three of the six cases.

Since the loads, internal design conditions, and fresh air requirements of the two zones are the same in case 1.1, all configurations except the laboratory systems behave like a single-zone system and the difference in energy consumption is negligible. For winter conditions 1.2 and 1.6, the non-laboratory configurations showed similar performance as well. This is explained by the fact that both SDUCT and DDUCT are equipped with terminal reheating coils. For the winter condition in case 1.4, where the design temperature of each zone is different from the other, the arrangement of exhaust air showed an impact on the energy consumption. Since  $T_{db2} < T_{db1}$ , exhausting air from zone 2 reduces enthalpy difference between exhaust and outside air, therefore

**Table 5. Comparison of Energy Consumption in Test Case Set 1**

Cases	Total heating/cooling/humidification duty (kW)									
	SDUCT	DDUCT	FCOIL	FCOILS1	FCOILS2	FCOILP	FCOILX	OAHU	LAHU	LAHU0
1.1 Summer	10.79	10.78	10.75	10.75	10.75	10.75	10.75	10.76	33.97	41.09
1.2 Winter	10.87	10.97	10.97	10.96	10.97	10.98	10.96	10.96	20.98	32.04
1.3 Summer $T_{db1} > T_{db2}$	20.71	14.30	10.75	10.97	10.55	10.78	10.54	10.77	38.82	41.27
1.4 Winter $T_{db1} > T_{db2}$	11.00	10.99	10.98	10.70	11.23	10.97	10.70	10.70	20.59	26.63
1.5 Summer $\dot{Q}_{s1} > \dot{Q}_{s2}$	19.13	10.76	10.75	10.76	10.75	10.75	10.76	10.76	20.87	22.39
1.6 Winter $\dot{Q}_{s1} = -\dot{Q}_{s2}$	5.91	5.90	5.90	5.89	5.89	5.91	5.90	5.91	15.95	22.72

**Table 6. Comparison of Energy Consumption in Test Case Set 2**

Cases	Total heating/cooling/humidification duty (kW)									
	SDUCT	DDUCT	FCOIL	FCOILS1	FCOILS2	FCOILP	FCOILX	OAHU	LAHU	LAHU0
2.1 Summer	14.66	12.54	10.75	10.70	10.73	10.75	10.75	10.75	36.81	38.07
2.2 Winter	10.95	10.95	10.96	10.96	10.95	10.96	10.96	10.96	21.34	26.22
2.3 Summer $T_{db1} > T_{db2}$	19.47	16.14	10.86	14.50	10.47	13.62	10.47	10.87	37.14	41.16
2.4 Summer $T_{db1} < T_{db2}$	25.02	15.04	10.65	10.53	10.97	10.65	10.53	10.62	30.70	35.19
2.5 Summer $\dot{Q}_{s1} > \dot{Q}_{s2}$	23.04	14.78	10.68	10.68	10.68	10.68	10.68	10.69	18.40	21.90
2.6 Summer $\dot{Q}_{s1} < \dot{Q}_{s2}$	12.94	14.20	10.75	10.75	10.75	10.75	10.75	10.76	51.24	54.20
2.7 Winter $\dot{Q}_{s1} = -\dot{Q}_{s2}$	6.94	6.95	6.99	6.98	6.98	6.99	6.98	6.98	16.24	22.28
2.8 Winter $-\dot{Q}_{s1} = \dot{Q}_{s2}$	5.85	5.86	5.90	5.88	5.89	5.90	5.89	5.89	16.80	21.37

reducing the overall system duty. In the table, FCOILS1, FCOILX, and OAHU showed the lowest system duty figure, whereas FCOILS2, which exhausts air from zone 1, has the highest consumption figure. In contrast, the temperature setting in case 1.3 (summer condition) dictates air should be exhausted from zone 1. Predictably, FCOILS2 showed the best performance, as did FCOILX, which can be operated as either FCOILS1 or FCOILS2. Due to the particularity of OAHU, it allows air to exhaust either from zone 2 or from both zones. As a result, its performance was on par with FCOIL and FCOILP in case 1.3. The centralized systems (SDUCT and DDUCT) have high energy consumption due to increased outside air input and simultaneous cooling and reheating. The SDUCT configuration suffered from high outside air input, too, in case 1.5.

Since outside air load is a major part of the total load of the system, the energy consumption of the laboratory HVAC systems is much higher than that of other configurations. Comparing LAHU and LAHU0, however, the energy savings achieved by passing extract air from the office zone to the laboratory space is significant. This remains true in the second set of tests, although the amount of savings is smaller due to the fact that the amount of air that can be circulated from office to laboratory is limited by the outside air supply to the office zone.

In the second set of tests (Table 6), where the minimum fresh air demand for zone 1 is one-third of that for zone 2, the single-duct and dual-duct systems failed to minimize the outside air intake in most of the cases. This is because of the imbalance of heating/cooling load distribution versus the fresh air demand. Results from test cases 2.5 and 2.6 show most clearly how the distribution of load affects the heating/cooling duties of the centralized systems.

Similar to cases 1.3 and 1.4, cases 2.3 and 2.4 show the impact of the arrangement of the air exhaust point in the configurations. In case 2.3, the indoor air temperature of zone 1 is higher than that of zone 2; therefore, the enthalpy difference between ambient air and zone 1 extract air is smaller. As a result, air should exhaust from zone 1 in order to reduce outside air load. Likewise, the optimal arrangement of exhaust air location is from zone 2 in case 2.4. Simulation results confirmed these analyses. Either of the sequentially arranged fan-coil-based systems (FCOILS1 and FCOILS2) works best in one of the conditions but poorly in the other. Since fan-coil-based configuration with cross-zone air paths (FCOILX) may be operated as either FCOILS1 or FCOILS2, it has the best performance in both cases.

In the third set of test cases (Table 7), which comprises winter operational conditions with simultaneous heating and cooling loads to different zones (such as interior and exterior zones), the benefit of interzonal circulation is more prominent. In cases 3.1 and 3.2, where zone temperature settings prevent interzonal load offset, it was not possible for any of the configurations to reduce energy consumption. In case 3.3, on the other hand, the configurations equipped with recirculation paths between zones significantly reduced the system heating/cooling duty. These configurations include single-duct and dual-duct systems, parallel and sequentially arranged fan-coil-based configurations, and the fan-coil configuration with cross pass between zones. The OAHU system was unable to take advantage of interzonal circulation due to the lack of a circulation path from the exterior zone to the interior zone. In operation, the volume of airflow transported from the interior zone to the exterior zone equals the volume of fresh air supply to the

**Table 7. Comparison of Energy Consumption in Test Case Set 3**

Cases	Total heating/cooling/humidification duty (kW)									
	SDUCT	DDUCT	FCOIL	FCOILS1	FCOILS2	FCOILP	FCOILX	OAHU	LAHU	LAHU0
3.1 Winter	5.85	4.92	4.89	4.88	4.87	4.88	4.88	4.89	-	-
3.2 Winter $T_{db1} > T_{db2}$	8.82	6.33	4.89	4.89	4.89	4.89	4.89	4.89	-	-
3.3 Winter $T_{db1} < T_{db2}$	0	0.68	4.88	0	0	0	0	4.88	-	-



## CONCLUSION

As the result of this study, three major factors are identified as having significant impact on the performance of HVAC configurations. These factors are (1) the ability to minimize outside air load by minimizing both outside air mass flow rate and the enthalpy difference between the outside air and the exhaust air, (2) the ability to eliminate simultaneous cooling and heating, and (3) the availability of interzonal air circulation paths. Ten different configurations were tested with a number of design conditions. In the tests, the fan-coil-based configurations generally performed better than the single-duct and dual-duct systems. Innovative configuration designs from the literature demonstrated significant improvement over the traditional systems in terms of energy consumption. Overall, a fan-coil system with cross-air-paths between the two zones achieved the lowest energy consumption in all test cases. It is evident, however, that some configurations perform better than others in certain conditions; therefore, optimization is required to identify the optimum configuration design for any given application.

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## NOMENCLATURE

$C_p$	= specific heat of air, 1.0061 kJ/(kg·K) at 20°C	$\dot{Q}_l$	= zone latent load (kW)
$h$	= enthalpy of air (kJ/kg)	$\dot{Q}_s$	= zone sensible load (kW)
$\dot{m}^{O/A}$	= mass flow rate of outside air (kg/s)	$\dot{Q}_s^{O/A}$	= sensible outside air load (kW)
$\dot{m}_{min}^{O/A}$	= minimum required mass flow rate of outside air (kg/s)	RH	= relative humidity (%)
$\dot{m}^{circ}$	= mass flow rate of (re)circulated air (kg/s)	$T$	= zone design temperature (°C)
$\dot{Q}^{Load}$	= zone load (kW)	$T_{db}$	= dry-bulb temperature (°C)
$\dot{Q}^{H/C}$	= heating/cooling/humidification duty (kW)	$T_{wb}$	= wet-bulb temperature (°C)
$\dot{Q}^{O/A}$	= outside air load (kW)	$T_{db}^{O/A}$	= dry-bulb temperature of outside air (°C)
		$T^{S/A}$	= temperature of supply air (°C)
		$w$	= air humidity ratio (kg/kg)

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