Energy conservation for operation theatres by secondary return air system

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ABSTRACT
The traditional ways of maintaining cleanliness, temperature and moisture level in an operation theatre (OT) usually use larger HVAC system and keep both the heating and cooling functions operating at the same time. The temperature and moisture level are thus under control at the cost of tremendous amount of wasted energy. Incorporating the HVAC system with the newly developed secondary return air system, the system capacity can be reduced while the energy efficiency can be increased. In this paper, concepts of the secondary return air system are described. The influential factors of sensible heat ratio and air change rate of the secondary return air system on energy saving are discussed. An in-house computer program that was developed for calculating the systems' operation points and the performance of energy conservation were used in this research.

INDEX TERMS
HVAC system; Energy saving; Secondary return air; Operation theatre

INTRODUCTION
The HVAC system in the operation theatre (OT) can be characterized by high airflow volume, high cooling load and stringent temperature and moisture control. The temperature level is usually set at 21 ± 2°C, and the moisture level is maintained at 50 ± 5% RH in general.

The cleanliness in the OT is controlled by using high supply airflow volume. The practical recommendation ranges of supply airflow volume (in terms of air change per hour, ACH) for various classes of operating theatres are as follows. For Class 1 to Class 10 OTs, the ACH is in between 300 and 500 times per hour. For Class 100 OTs, the ACH is maintained at 100–300 times per hour. For Class 1000 OTs, the ACH is reduced to 50–80 times per hour. For the least clean Class 10 000 OTs, the ACH is further reduced to 25–40 times per hour. Table 1 shows the comparison of typical class 100 operating theatres.

Table 1 Airflow volume and air change rate for three typical class 100 OTs

	OT Area (m^2)	Supply air diffuser (m^2)	Airflow volume (m^3/h)	Air change rate (h^{-1})
Large OT	5.7 × 5.4	3.6 × 2.1	9526	103
Medium OT	5.1 × 4.8	3.6 × 2.1	9526	129
Small OT	4.5 × 4.2	3.6 × 2.1	9526	168

The cooling loads of operating theatres are higher than normal residential HVAC systems. The following empirical guidelines (Hu, 2002) can be used to determine the cooling load in OTs:

1. The walls of the OT are usually constructed by using metal sheets and plates. If an air conditioning system in the surrounding is not available, then the sensible heat transferred through the walls to the room can be reduced by about 40–80 W/m^2.

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2. The personnel working in an OT are standing all the time. The average heat generation per person is 70 W. The area occupied per person is 1.8–2.5 m^2 so the rate of heat generation is 30–40 W/m^2 per person.
3. The cooling load from lighting is about 15 W/m^2.
4. The electricity devices in OTs are different. A reasonable approximation is at about 70 W/m^2.
5. The latent heat that is generated from personnel and wet surfaces is about 75–95 W/m^2.
6. It can be seen from above data that the total sensible cooling load is 230–300 W/m^2, and latent cooling load is 100–120 $g/m^2/h$. The total heat ratio is about 0.6–0.75.

The above guidelines show that the supply airflow volume for HVAC systems used in an OT is 4–60 times higher than that in a residential HVAC system, and the cooling load is 2–3 times higher. If the relative cooling load is defined as the cooling load that one unit volume of supply air needs to handle, the HVAC system of an OT then has a small relative cooling load. This is why its temperature tolerance can be controlled at a small range.

The traditional way of maintaining the cleanliness, temperature and relative humidity usually is to make the HVAC system large in size, while both the heating and cooling coils often work at the same time. Tremendous amount of energy is wasted.

By designing the HVAC system with the concept of secondary return air, energy efficiency can be improved and the machine size can be reduced.

The concept of the secondary return air HVAC system is as follows. A portion of the return air bypasses the cooling coil and is sent back to the room. The bypassed return air still contains some energy and the energy is then used for dehumidification. The energy can then be saved because most of the time the cooling coil is working on dehumidification only. The efficiency of a cooling coil reaches the peak when the air temperature difference between the coil surfaces is maintained at 3–6. By using the secondary HAVC system, only part of the return air goes through the coil; therefore, the coil can be smaller and the temperature difference can be higher. However, the secondary return air system requires tedious calculation and a delicate control system. This study aims to develop a user-friendly program to simplify the design processes and to discuss factors that affect the system performance.

ENERGY CONSUMPTION: THE TRADITIONAL SYSTEM VERSUS SECONDARY RETURN AIR SYSTEM

The comparison of the two systems was conducted at a mock cardiac surgery OT. The cleanliness is Class 100 and the dimension is 7.2 m × 6.3 m × 2.85 m. The supply air area (6 m × 6 m) is covered with high efficiency particulate air (HEPA) filters and the average supply air velocity is 0.3 m/s. For the traditional OT HVAC system, the operation points are described by using the psychrometric chart in Figure 1.

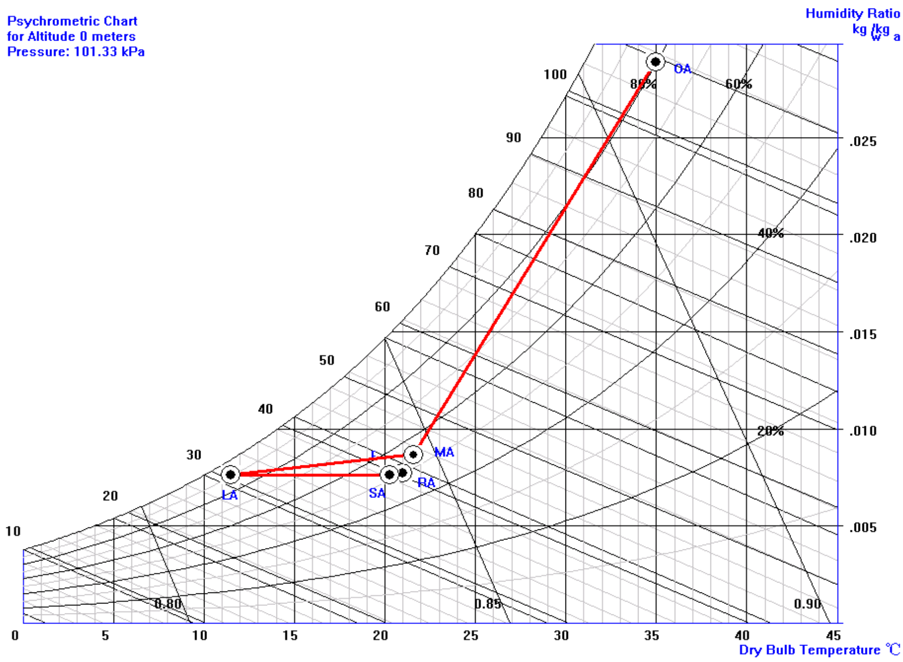

Figure 1 Airflow properties of the traditional return air system. [Note: OA = outdoor air, LA = air leaving coils, SA = supply air to the OT, RA = return air, MA = maxing air.]

The supply airflow volume is about 38 880 m^3/h (including 1700 m^3/h of fresh air). The cooling load is 167.5 kW and the re-heating load is 115.6 kW. The HVAC system designed thus has a difficult in maintaining precise temperature and moisture level. To do this, a lot of energy was wasted. It can be found that for the purpose of de-humidification the amount of heat transfer of the cooling process has to be larger than the room cooling load. The air is then conditioned to the dew point with respect to the room humidity. The purpose of re-heating is to maintain the temperature and moisture level in the room. So the heating and cooling functions were working at the same time, which increases the energy consumption.

By using the concept of secondary return air HVAC system (see Figure 2), the energy required for cooling reduced to 64.4 kW and the heating requirement is no longer necessary. The cleanliness, temperature and the moisture are all maintained within the tolerance range. The secondary return air volume is 31 811 m^3/h and the primary return air is 5369 m^3/h. Fresh air remains at 1700 m^3/h. The control system setup of the secondary air return system is shown in Figure 3.

It can be seen from the above description that the re-heating coil is not necessary by using secondary return air system. By referring to ASHRAE Handbook (2001), a computer program was developed to calculate the operating points and energy saving using VBA programming language. The output of the program is verified by the data discussed above. The energy saved is as follows. To the energy required for cooling, the saving is 167.54 – 64.4 = 103.17 kW (29.3 RT). By assuming the power consumption of the chiller is 0.9 kW/RT, the monthly saving is 29.3 RT × 0.9 kW/RT × 12 h/day × 24 days/month × 1.24 dollar /kWHR = 9417 NTD (new Taiwan Dollar). The energy saving in heating is 115.6 kW × 12 × 24 × 1.24 = 41 232 NTD. The total energy saving in one year is (9417 + 41 232) × 12 = 608 400 NTD. Note: 1 USD = 34 NTD.

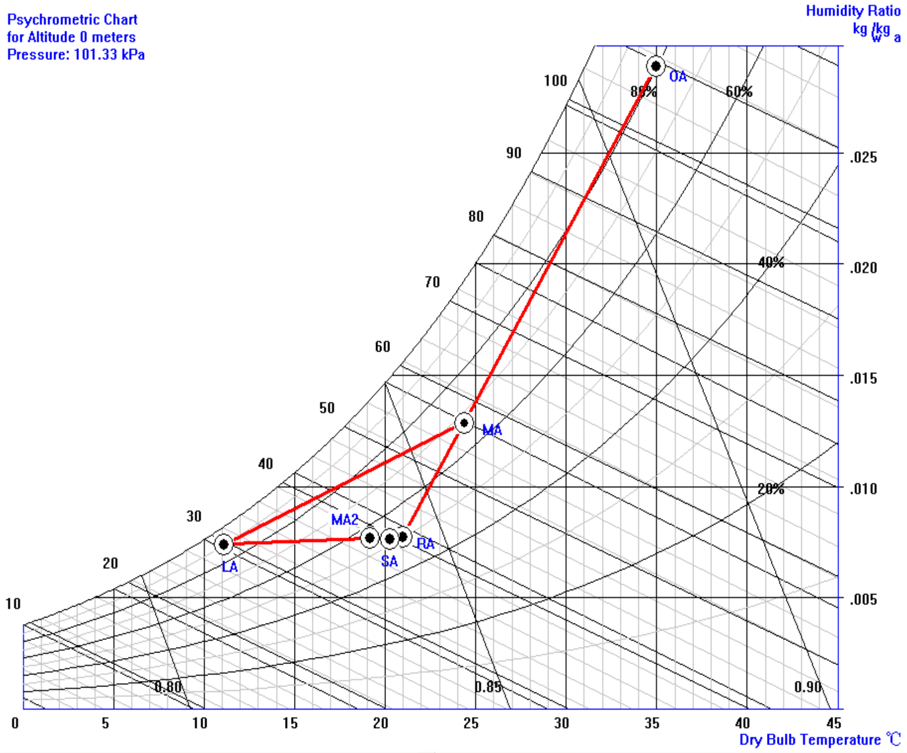

Figure 2 Airflow properties of the secondary return air system. [Note: OA = outdoor air, LA = air leaving coils, SA = supply air to the OT, RA = return air, MA = maxing air.]

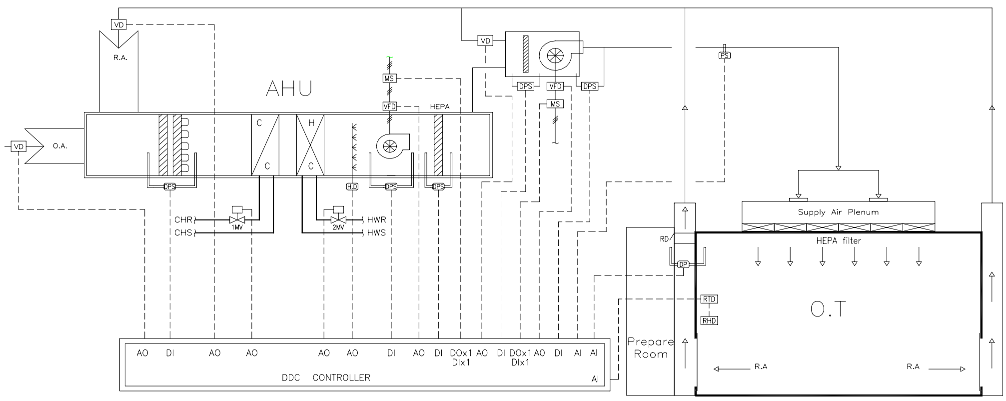

Figure 3 Control system of the secondary air return system.

DISCUSSION OF THE FACTORS THAT AFFECT THE ENERGY CONSUMPTION OF THE SECONDARY RETURN AIR SYSTEM

Based on the developed program, the following results regarding energy performances of the secondary return air system are discussed. When using the secondary return air system, the energy saving would be pronounced for both high air change rate and low sensible heat ratio cases, as shown in Figures 4 and 5. The energy saving is not significant at low air change rates when the sensible heat ratio is up to 0.8. However, these situations are seldom found in real OTs. When the air change rate is raised from 30 to 200 h^{-1}, the saving in energy is significant. In general, the most influential factors on the energy saving is air change rate and the physical size of the OT.

CONCLUSIONS

In this paper, the difference of energy performance between secondary return air system and the primary return air system was discussed. The influential factors of secondary return air system on energy saving were discussed. In general, the most influential factors on the energy saving is air change rate and the size of the OT. A computer software based on VBA has been developed successfully. This software is designed to analyse the operating parameter of the secondary return air system efficiently.

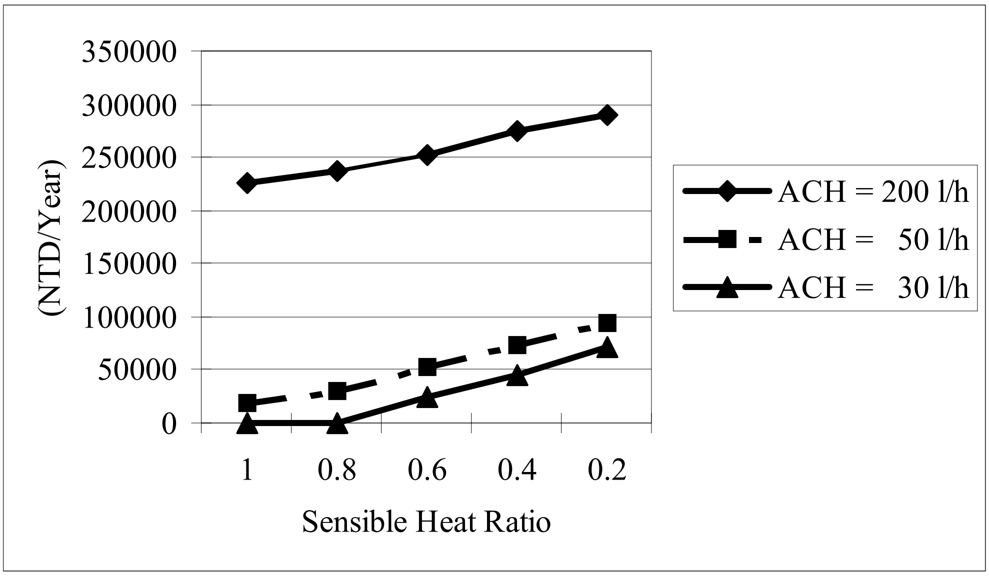

Figure 4 Money saved versus sensible heat ratio of the OT for different air change rate cases (OT dimension = 5.1 m × 4.8 m × 3 m; room conditions = 21°C and 50% RH).

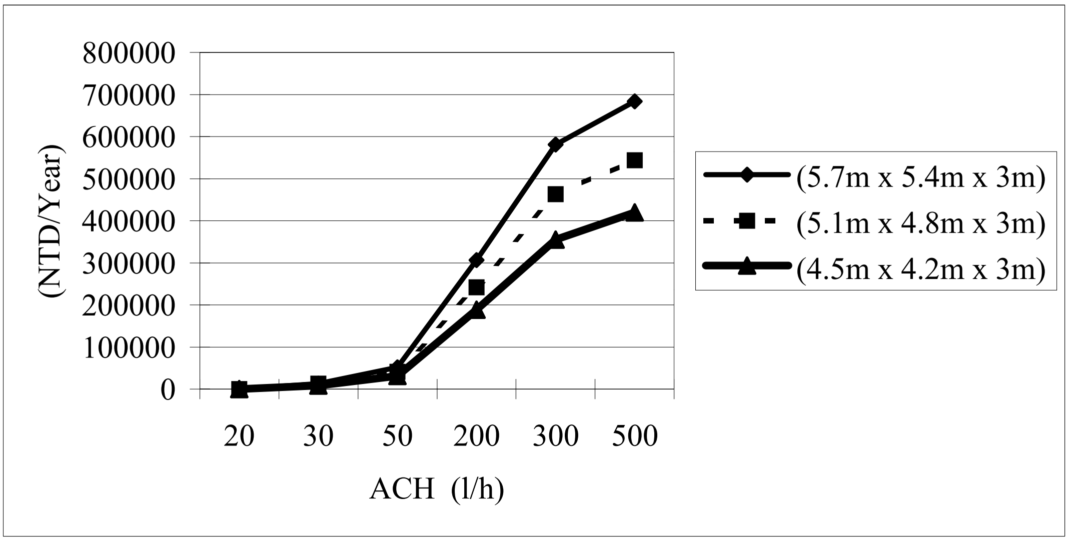

Figure 5 Money saved versus air change rate of the OT for different room sizes (room sensible ratio = 0.8, room conditions = 21°C and 50% RH).

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