Energetic-exergetic analysis of an air handling unit to reduce energy consumption by a novel creative idea

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Abstract

Purpose – This study aims to simulate the flow and heat transfer through an air handling unit to reduce its energy consumption by a novel creative idea of using an air-to-air heat exchanger.

Design/methodology/approach – To do this, both first and second laws of thermodynamics energy and exergy balance equations were solved numerically by an appropriate developed computer code.

Findings – Using the air-to-air heat exchanger in dry conditions decreases the cooling coil load by 0.9 per cent, whereas the reduction for humid conditions is 27 per cent. Similarly, using air-to-air heat exchanger leads to an increase in the first law of efficiency in dry and humid conditions by 0.9 per cent and 36.8 per cent, respectively.

Originality/value – The second law of efficiency increases by 1.55 per cent and 2.77 per cent in dry and humid conditions, respectively. In other words, the effect of using an air-to-air heat exchanger in humid conditions is more than that in dry conditions.

Keywords Efficiency, Exergy analysis, Irreversibility, AHU, Air-to-air heat exchanger

Paper type Research paper

Nomenclature

Symbols

C_p = specific heat of humid air $\left(\frac{kJ}{kg.K}\right)$;

h = enthalpy $\left(\frac{kJ}{kg}\right)$;

H_{fg} = latent heat of vaporization (kJ/kg);

$\dot{m}$ = mass flow rate $\left(\frac{kg}{s}\right)$;

P = pressure (kPa)

Q = heat transfer rate (kW);

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RR = return ratio $\left(RR = \frac{\textit{recirculated air mass flow rate}}{\textit{supply air mass flow rate}}\right)$;
R_v = universal vapor constant $\left(461\ \frac{J}{mol.K}\right)$;
S = entropy $\left[\frac{kJ}{kg.K}\right]$;
T = temperature (K);
v = specific volume $\left(\frac{m^3}{kg}\right)$; and
X = exergy $\left(\frac{kJ}{kg}\right)$.

Greek letters
ε = air-to-air heat recovery effectiveness $\left(\frac{kg}{m.s}\right)$;
φ = relative humidity; and
ω = humidity ratio $\left(\frac{kg_v}{kg_a}\right)$.

Subscripts
a = air;
c = cold;
cc = cooling coil;
con = condensation;
h = reheater, hot;
l = latent;
o = ambient condition;
r = room condition;
s = sensible;
sat = saturation;
tot = total;
v = vapor; and
w = water.

Superscripts
Des = destroyed.

1. Introduction

Building energy consumption can reach up to 40 per cent of global energy demand (Geng *et al.*, 2018). Heating, ventilation and air conditioning (HVAC) consumes more than 50 per cent of the required energy for buildings' maintenance and operation (Geng *et al.*, 2018). Heat or energy recovery is an effective technique to tackle the issue of high energy consumption in a building. Energy recovery is defined as a device (enthalpy air-to-air heat exchanger) that transfers latent and sensible energy from one airstream to another. However, in heat-recovery device (sensible air-to-air heat exchanger), only sensible energy is exchanged (Zeng *et al.*, 2017). In sensible or enthalpy air-to-air heat exchanger, depending on the circumstances, the coldness or hotness from the exhaust air is recovered to the supply air. It is highly promising that the enthalpy air-to-air heat exchanger can reuse 60-95 per cent of the waste energy (Liu *et al.*, 2019) .The efficiency of the enthalpy air-to-air heat exchanger is influenced by system arrangements and operation circumstances. In general, the total and sensible effectiveness of typical enthalpy air-to-air heat exchanger can vary from 40 per cent to 80 per cent and 50 per cent to 80 per cent, respectively (O'Connor *et al.*, 2016). For a comprehensive detail of air-to-air heat recovery formulation, readers are referred to Zeng *et al.*, 2017; Deshko *et al.*, 2016; and Lowrey and Sun, 2018. The experimental and parametric study of an air-to-air heat recovery unit was conducted in 2011 by Fernández-Seara *et al.*, 2011. They reached the conclusion that the heat-transfer rate in plate heat exchanger would be increased almost linearly by decreasing the inlet fresh air temperature. It is found that the plate heat exchanger thermal efficiency is not influenced by the fresh air temperature from 5°C to 15°C. However, for air temperatures greater than 15°C, the increase in temperature leads to a slight decrease in the efficiency. They revealed that the effect of inlet air relative humidity on the thermal efficiency was lower than 10 per cent. Experiments on the performance of an enthalpy air-to-air heat exchanger were conducted in 2010 by Nasif *et al.*, 2010. Many temperature and moisture content measurements have been made with the purpose of calculating the sensible and total effectiveness. It is found that in hot and humid conditions, the energy consumption of an air conditioner, coupled with the enthalpy air-to-air heat exchanger, is lower, up to 8 per cent less than the conventional one. Calorimetric method has been proposed to evaluate the performance of a decentralized system (Coydon *et al.*, 2015). This method has been applied for three locations (moderate, cold and warm) to evaluate the seasonal energy efficiency. In Coydon *et al.*, 2015, the authors found that the heat recovery in cold regions is more advantageous. Calculations on seasonal energy efficiency affirmed that the percentage of seasonal energy efficiency for both devices in all regions is greater than 50 per cent. A novel heat recovery system, which comprised a heat exchanger, blower and ducts, has been constructed by Cuce *et al.*, 2016. The proposed system acts under roof application with the purpose of preheating fresh air using stale air. Many experiments in winter have been carried out to investigate the heat-recovery efficiency. The result of experiments has shown that the best heat-recovery efficiency has reached up to 89 per cent. Many attempts have been made (O'Connor *et al.*, 2014) to use the rotary thermal wheel in a wind tower with the purpose of decreasing the energy demand. Results showed that the ventilation rate of the proposed wind tower with rotary wheel was above the required criteria. The temperature of the supply air in the wind tower with rotary wheel heat recovery is 2°C higher than the conventional one. They concluded that the heating energy consumption has been reduced. The experimental study of using air-to-air heat exchanger in a specific HVAC system has been conducted by Delfani *et al.* (2010). In all experiments, the cooling coil load and air volume flow rate were constant. The results show that the advantage of using an air-to-air heat exchanger is higher in regions with higher difference between dry-bulb and dew point temperature. In other words, the higher difference between dry-bulb and dew point temperature leads to higher latent load quotas. They found that the humidity ratio of the room supply air for a cooling coil combined with an air-to-air heat exchanger has been reduced. Delfani *et al.* (2012) experimentally studied several types of air-conditioning to investigate the effect of heat recovery on the amount of energy consumption. The results reveal that using an air-to-air heat exchanger in hot and humid regions would decrease energy consumption up to 32 per cent.

Exergy analysis is another technique that can highlight energy inefficiencies through the process and offers advantageous information to designers for improving energy efficiency and reducing the required energy consumption in the building. For comprehensive detail of the exergy concept in buildings, readers are referred to Sangi and Müller, 2018; Kerdan *et al.*, 2017; and Sangi and Müller, 2018. Exergy is defined as the maximum useful work (via a reversible process) that is obtained from a system with respect to a specified environment. The first and second laws of thermodynamics deal with the quantity and quality of energy. The second law is concerned with the degradation of work potential of the energy and is an appropriate metric to analyze and optimize the HVAC process. Several studies, for example, building envelope (Açıkkalp *et al.*, 2014; Buyak *et al.*, 2017), heat pumps (Chen *et al.*, 2019; Akbulut *et al.*, 2016; Khalid *et al.*, 2016, cooling system Meggers *et al.*, 2017, boilers Usón *et al.*, 2019, renewable integration Shahsavar and Rajabi, 2018, energy storage systems (Ezan *et al.*, 2010; Ezan *et al.*, 2010; Rezaei *et al.*, 2013) and HVAC (Balta *et al.*, 2010; Sakulpipatsin *et al.*, 2010) have been conducted on building exergy analysis to improve thermal performance. In Razmara *et al.*, 2015, the exergy analysis of a building HVAC has been done based on model predictive control (MPC) technique. By this method, the energy demand and irreversibility decreases up to 36 per cent and 22 per cent, respectively, in comparison with traditional on–off controller. Martinaitis *et al.*, 2016 suggested a new algorithm for the second law analysis of a ventilation heat recovery exchanger. They used coenthalpy concept as the direct potential of the exergy flow to calculate the irreversibility through the heat recovery exchanger. The proposed algorithm can be used at variable reference surroundings. Wei and Zmeureanu, 2009, developed the exergy equation for a variable air volume system operating in a large office building and asserted that the exergy efficiency was quite low (2-3 per cent). The energy and exergy formulation for three developed heating and cooling systems in a residential building was tested by Khalid *et al.*, 2015. Among the proposed layout, the system that is operated by natural gas and vapor absorption chiller has the highest energy efficiency (27.5 per cent). The lowest energy efficiency (19.9 per cent) is referred to the photovoltaic (PV) and solar thermal operated with vapor compression chiller. But the exergy efficiency (3.9 per cent) for this system is found to be the highest. The PV and solar thermal operated system equipped with heat pump has the lowest exergy efficiency (1.2 per cent) among all the proposed systems. The exergy and energy performance of a new integrated system were numerically studied by Ghosh and Dincer, 2015. The proposed integrated system comprises various psychometric processes such as cooling with dehumidification, space cooling and heating, evaporative cooling and heating with humidification. It is found that the largest exergy destruction took place through the space heating process with approximately 31.2 per cent of total irreversibility. It should be mentioned that the first and second law efficiencies of the integrated system were 18.6 per cent and 33.31 per cent, respectively. The energetic and exergetic analysis of three conventional air-conditionings have been developed by Caliskan *et al.*, 2011. Moreover, authors proposed a novel air-conditioning based on the Maisotsenko cycle. Many calculations have been done to investigate the effect of dead state temperature on the exergy efficiency. At dead state temperature greater than 23°C, the exergy efficiency of the proposed system is higher than the three conventional ones. The maximum exergy efficiency of the proposed air-conditioning has been reached up to 60.329 per cent at the dead state of 50°C. The energetic and exergetic analysis of a novel air cooler based on Maisotsenko cycle were formulated by Caliskan *et al.*, 2012. Different dead state temperatures were defined to investigate its effect on the exergetic parameters variation. They found that the increase in dead state temperature from 0°C to 27.78°C would decrease the exergy input rates. They affirmed that the exergy loss was inversely proportional to the temperature of the dead state. It is found that with an increase in dead state temperature, the exergy loss decreases. The results of exergy show that the dead state temperature would effectively affect the irreversibility and entropy generation rate. The increase in dead state temperature leads to increase in irreversibility and entropy generation. The exergy analysis of two distinct humidification processes has been studied by Ghazikhani *et al.*, 2016. The distribution of exergy loss in the constant enthalpy humidification has been compared with the constant temperature humidification. Calculations show that the power input for constant temperature process is

12 per cent more than the enthalpy one. The results of exergy analysis proved that the constant temperature humidification has more irreversibility than the constant enthalpy humidification. They reached the conclusion that the constant enthalpy humidification has more advantages than the constant temperature humidification because of low irreversibility and energy consumption.

In this paper, the exergy analysis of an air handling unit (AHU) combined with the air-to-air heat exchanger has been studied. The energy and exergy balance equations have been developed for air-to-air heat exchanger, mixing box, cooling coil, reheater and room space to evaluate the psychometric properties and exergy destroyed through the various component. Numerical computations have been performed base on a developed program in EES to describe the energy and exergy analysis. In this study, first and second law efficiencies were defined to determine the performance of the AHU. The effect of using air-to-air heat recovery with respect to return air quota was investigated. Moreover, the extent influence of the air-to-air heat exchanger on the first and second law efficiencies and irreversibility with respect to the ambient humidity was discussed.

2. Description of the system

The thermodynamic processes involved in the AHU should be investigated prior to assessing the effect of using air-to-air heat exchanger. According to Figure 1, the input air (point 0) first enters the air-to-air heat exchanger so that heat and mass can be transferred between the exhaust and input air. The air exiting from the air-to-air heat exchanger with the condition indicated at point (1) is then mixed with the return air, producing the air with condition of point (3). Passing over the cooling coil, the mixed air undergoes cooling and dehumidification processes.

The temperature of the air (point 4) increases as it enters the heating coil so that the required properties are met before entering the room. On the other hand, sensible and latent heats also transfer to the room from the surrounding environment. The properties of the air in the room space are indicated at Point (2). Two paths are followed by the air with the conditions of Point (2) as it leaves the room: a fraction of it, which is determined by parameter RR, returns to the AHU to be mixed with the air at Point (1):

$$RR = \frac{\dot{m}_{2^*}}{\dot{m}_3} \tag{1}$$

The remaining portion enters the air-to-air exchanger and leaves it after the heat and mass transfer processes are carried out.

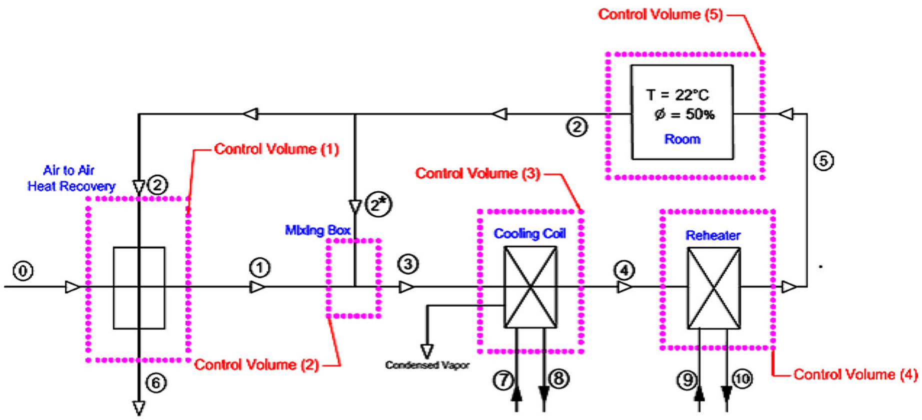

Figure 1.
AHU in cooling mode

3. Air handling unit energy and exergy analysis

Based on the first and second laws of thermodynamics, energy and exergy balance equations must be extracted. The following assumptions are considered:

- The atmospheric humid air is treated as an ideal gas mixture.
- Water vapor enthalpy is equal to the saturated vapor enthalpy at the same temperature.
- The pressure loss in various components is disregarded.
- Cooling coil bypass factor is zero.
- The temperature of the control volume (5) is equal to the ambient temperature.

The mass, energy and exergy balance equations for each control volume of the AHU according to Figure 1 can be written as follows:

Control volume (1): air-to-air heat exchanger

Dry air mass balance $$\dot{m}_0 = \dot{m}_1 \,,\ \dot{m}_2 = \dot{m}_6 \tag{2a}$$

Water mass balance $$\dot{m}_{0w} + \dot{m}_{2w} = \dot{m}_{1w} + \dot{m}_{6w} \tag{2b}$$

Energy balance $$\dot{m}_0 h_0 + \dot{m}_2 h_2 = \dot{m}_1 h_1 + \dot{m}_6 h_6 \tag{2c}$$

Exergy balance $$\dot{m}_0 X_0 + \dot{m}_2 X_2 - (\dot{m}_1 X_1 + \dot{m}_6 X_6) - X_{des,air\ to\ air} = 0 \tag{2d}$$

Control volume (2): mixing box

Dry air mass balance $$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 \tag{3a}$$

Water mass balance $$\dot{m}_{1w} + \dot{m}_{2w} = \dot{m}_{3w} \tag{3b}$$

Energy balance $$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \tag{3c}$$

Exergy balance $$\dot{m}_1 X_1 + \dot{m}_2 X_2 - \dot{m}_3 X_3 - X_{des,mix} = 0 \tag{3d}$$

Control volume (3): cooling coil

Dry air mass balance $$\dot{m}_3 = \dot{m}_4 \tag{4a}$$

Water mass balance $$\dot{m}_{3w} = \dot{m}_{4w} + \dot{m}_{cond}$$

$$\dot{m}_7 = \dot{m}_8 \tag{4b}$$

Energy balance
$$\dot{m}_3 h_3 + \dot{m}_7 h_7 = \dot{m}_4 h_4 + \dot{m}_8 h_8 + \dot{m}_{cond} h_{cond} \tag{4c}$$

Exergy balance
$$\dot{m}_3 X_3 + \dot{m}_7 X_7 - \dot{m}_4 X_4 - \dot{m}_8 X_8 - \dot{m}_{cond} X_{cond} - X_{des,cc} = 0 \tag{4d}$$

Control volume (4): reheater

Dry air mass balance
$$\dot{m}_4 = \dot{m}_5 \tag{5a}$$

Water mass balance
$$\dot{m}_{4w} = \dot{m}_{5w} \tag{5b}$$

Energy balance
$$\dot{m}_4 h_4 + \dot{m}_9 h_9 = \dot{m}_5 h_5 + \dot{m}_{10} h_{10} \tag{5c}$$

Exergy balance
$$\dot{m}_4 X_4 + \dot{m}_9 X_9 - \dot{m}_5 X_5 - \dot{m}_{10} X_{10} - X_{des,h} = 0 \tag{5d}$$

Control volume (5): room

Dry air mass balance
$$\dot{m}_5 = \dot{m}_2 \tag{6a}$$

Energy balance
$$\dot{m}_5 h_5 + Q_s + Q_l = \dot{m}_2 h_2 \tag{6b}$$

Exergy balance
$$\dot{m}_5 X_5 - \dot{m}_2 X_2 - X_{des,r} = 0 \tag{6c}$$

The amount of sensible, latent and total heat along with the properties of the humid air at points (2) and (5) are determined from the following relations:

$$Q_s = \dot{m}_5 \times C_p \times (T_2 - T_5) \tag{7}$$

$$Q_L = \dot{m}_5 \times h_{fg} \times (\omega_2 - \omega_5) \tag{8}$$

$$\mathrm{Q_T} = \mathrm{Q_s} + \mathrm{Q_L} = \dot{m}_5 \times (\mathrm{h_2} - \mathrm{h_5}) \tag{9}$$

The following relations can be used to calculate the properties of air at the outlet of the air-to-air heat exchanger Handbook, 1996:

$$\dot{m}_{min} = \min\left(\mathrm{c_0}\dot{m}_0,\ \mathrm{c_2}\dot{m}_2(1 - RR)\right) \tag{10}$$

$$Q_{s,a} = \varepsilon_s \dot{m}_{min} \times (h_A - h_2) \tag{11}$$

$$Q_{t,a} = \varepsilon_t \dot{m}_{min} \times (h_0 - h_2) \tag{12}$$

Where h_A is obtained from the psychometric chart, given the conditions [$T = T_O$, $\omega = \omega_2$]. Considering the above equations, the properties of the air at the air-to-air outlet is obtained as:

$$T_1 = T_0 - \frac{Q_{s,a}}{C_{p0}\dot{m}_0} \tag{13}$$

$$h_1 = h_0 - \frac{Q_{t,a}}{\dot{m}_0} \tag{14}$$

$$T_6 = T_2 + \frac{Q_{s,a}}{C_{p2}\dot{m}_2(1 - RR)} \tag{15}$$

$$h_6 = h_2 + \frac{Q_{t,a}}{\dot{m}_2(1 - RR)} \tag{16}$$

The enthalpy and humidity ratio at point (3) are obtained from equation (17) with respect to parameter RR:

$$RR = \frac{h_1 - h_3}{h_1 - h_2} = \frac{\omega_1 - \omega_3}{\omega_1 - \omega_2} \tag{17}$$

The exergy of the humid air and the produced water are required to perform an exergy analysis. These parameters are obtained using the following relations Bejan, 2016.

$$X_{humid\,air} = (C_{p,a} + \omega C_{p,v})\left(T - T_0 - T_o ln\left[\frac{T}{T_o}\right]\right) + (1 + 1.608\omega)R_a T_o ln\frac{P}{P_0}$$
$$+R_a T_o\left[(1 + 1.608\omega)ln\frac{1 + 1.608\omega}{1 + 1.608\omega_o} + 1.608\omega ln\frac{\omega}{\omega_0}\right] \tag{18}$$

$$X_{\text{condensed water}} = h_f - h_{f0} - T_0(s_f - s_{f0}) - R_v T_0 \ln(\varphi_0) + v_f(P - P_{sat}) \tag{19}$$

Where h_{f0} and s_{f0} denote the enthalpy and entropy of water at ambient temperature, and φ_O is the ambient relative humidity.

According to equations (20) and (21) the first and second laws of thermodynamic efficiencies can be calculated as follow:

$$\eta_I = \frac{Desired\ output}{Required\ input} = \frac{Q_s + Q_l}{Q_{cc} + Q_h} \tag{20}$$

$$\eta_{\mathrm{II}} = 1 - \frac{destroyed\ exergy}{input\ exergy} \tag{21}$$

4. Results

The governing equations of the AHU were investigated in the previous section. The temperature, humidity ratio, relative humidity and enthalpy can be calculated by simultaneously solving energy equations. Then, the exergy balance for each control volume can be written to obtain the exergy loss. Input parameters given in Table I were used to perform the exergy and energy calculations for the AHU shown in Figure 1.

The energy analysis results are demonstrated in Figure 2.

As shown in Figure 2, the energy input (451.815 kW) and energy output (451.815 kW) are equal. The cooling coil load was obtained from the energy difference between the inlet and outlet of chilled water, the value of which was 22.622 kW. Therefore, the first law of efficiency was calculated to be 73.7 per cent by applying equation (20). The respective exergy analysis is also shown in Figure 3, according to which, the output and input exergies are not equal, meaning that the AHU under study is not reversible (ideal). The exergy difference between the input and output is an indication of the irreversibility. The lower this difference, the closer the cycle is to its ideal state. As shown in Figure 3, this exergy

Value	Parameter	Value	Parameter
$Q_s = 12$ kW	Sensible heat to tde room	$T_r = 22°C$	Room temperature
$Q_l = 5$ kW	Latent heat to tde room	$\varphi_r = 50\%$	Room relative humidity
$\dot{m}_c = 1.5\ \frac{kg}{s}$	Cooling water mass flow rate	$T_O = 35°C$	Ambient temperature
$T_c = 5°C$	Cooling water temperature	$\varphi_O = 40\%$	Ambient relative humidity
$\dot{m}_h = 1.5\ \frac{kg}{s}$	Heating water mass flow rate	$RR = 0.5$	Return air ratio
$T_h = 60°C$	Heating water temperature	$\varepsilon_s = 0.7$, $\varepsilon_t = 0.5$	Air-to-air heat recovery efficiency

Table I. Input parameters to analyze tde AHU

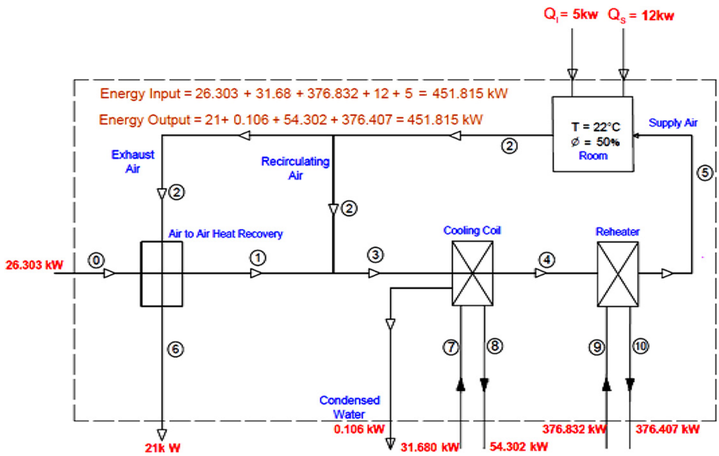

Figure 2. Energy balance for AHU with recovery

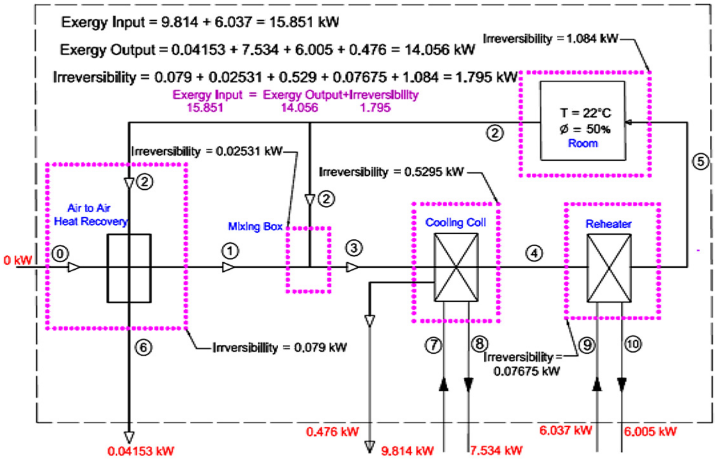

Figure 3.
Exergy balance for AHU with heat recovery

difference is 1.795 kW, and the highest exergy loss occurs during the room process (a total loss of 60 per cent).

The second-highest exergy loss is associated with the cooling coil (roughly 30 per cent). The heat transfer process at the high temperature difference between the input air and the water inside the heat exchanger pipes causes a considerable amount of exergy loss. The second law of efficiency for the AHU according to Figure 3 was obtained as 88.69 per cent based on equation (21).

To assess the effect of air-to-air heat exchanger, the results of energy and exergy analyses for an AHU without recovery were found and presented in Table II. Using air-to-air heat exchanger decreased the cooling coil load from 28,195 W to 22,620 W (19.7 per cent decrease).

Calculations also indicate that using air-to-air heat exchanger also reduced the temperature of the air at the inlet of cooling coli (point 3 in Figure 1), as well as the air mass flow rate (Table II). On the other hand, the cooling coil load is dependent on both parameters. The cooling coil load decreases as the air mass flow rate and inlet air temperature decrease. The heating load of the reheater decreased from 569 W to 425 W in case the air-to-air heat exchanger was used. The reason for the slight decrease in the reheater heating load can be attributed to the insignificant decrease in the mass flow rate of the passing air through the reheater. Overall, the sum of the cooling and heating loads decreased by 19.8 per cent in case an air-to-air heat exchanger was used in the AHU, hence improving the first law efficiency from 59.1 per cent to 73.7 per cent (24.7 per cent increase). According to Table II, by using the air-to-air heat exchanger, the exergy loss in the cooling coil decreased because of its decreased load. As a result, the exergy loss in an AHU equipped with air-to-air (1,793 W) heat exchanger was 13.63 per cent lower than that with no heat exchanger (2,076 W). Therefore, the second law of efficiency improved by 2 per cent by using the air-to-air heat exchanger.

The effect of using an air-to-air heat exchanger was significantly more obvious in the absence of return air, in which case the cooling load changed from 41,711 W to 28,258 W (32.2 per cent decrease). On the other hand, the reheater heating load also decreased by 54.7 per cent in case an air-to-air heat exchanger was used. Overall, the sum of the cooling and heating loads decreased by 33 per cent for the AHU in case an air-to-air heat exchanger was

Parameters	AHU with 100% fresh air RR = 0 No recovery (basic AHU)	RR = 0 $\varepsilon_s = 0.7$ $\varepsilon_t = 0.5$	AHU with return air No recovery RR = 0.5	RR = 0.5 $\varepsilon_s = 0.7$ $\varepsilon_t = 0.5$
Energy analysis				
$\dot{m}_{supply}\left[\frac{kg}{s}\right]$	0.82	0.7481	0.7455	0.7308
$Q_s\,[W]$	12,000	12,000	12,000	12,000
$Q_l\,[W]$	5,000	5,000	5,000	5,000
$Q_{cc}\,[W]$	41,711	28,258	28,195	22,620
$Q_h\,[W]$	1,315	595	569	425
η_I	0.3951	0.5892	0.591	0.737
Exergy analysis				
$Ex^{des}_{air\,to\,air}\,[W]$	–	161.8	–	79
$Ex^{des}_{mix}\,[W]$	–	–	96.58	25.31
$Ex^{des}_{cc}\,[W]$	1,447	725.7	803.3	529.5
$Ex^{des}_{H}\,[W]$	230.8	106.9	102.3	76.75
$Ex^{des}_{room}\,[W]$	1,036	1,072	1,074	1,082
$Ex^{des}_{tot}\,[W]$	2,715	2,066	2,076	1,793
η_{II}	0.8287	0.8697	0.869	0.8869

Table II. Comparison between the parameters affected by using air-to-air heat exchanger

used in the absence of return air, in which case the first law of efficiency improved from 39.5 per cent to 58.9 per cent, with a 49.1 per cent increase. According to Table II, exergy loss decreased from 2,715 W to 2,066 W by using an air-to-air heat exchanger. In other words, the exergy loss decreased by 23.9 per cent, allowing the second law efficiency to increase by roughly 4.9 per cent.

According to Figure 4, the highest exergy loss in the basic AHU, (i.e. an AHU with no return air and no air-to-air heat exchanger) occurs in the cooling coil, which accounts for 53 per cent of the total irreversibility. However, in case return air and air-to-air heat exchanger are used, the cooling coil load significantly decreases, consequently decreasing the exergy loss in this part. In the case of an AHU equipped with air-to-air heat exchanger (with return air or without return air), the highest exergy loss occurs in the room process (from points 5 to 2 in Figure 1) because of irreversible heat transfer between room space and ambient.

The variations in supply air mass flow rate ($\dot{m}_5$) and the cooling coil load with respect to RR [equation (1)] are demonstrated in Figure 5.
As shown, the supply air mass flow rate decreases as the fraction of the return air (RR) is increased. The cooling coil load is influenced by the supply air mass flow. As the supply air mass flow decreases, the cooling coil load decreases. Therefore, Figure 5 shows that the cooling coil load decreases by a decrease in the air mass flow rate. According to equation (20), the first law of thermodynamic efficiency is expected to increase as the cooling coil load (Q_{cc}) decreases. In other words, the increase in recirculated ratio (RR) leads to increase in the first law of efficiency as shown in Figure 6.

The cooling coil load would effectively affect the exergy efficiency. The lower the cooling coil load, the lower the irreversibility in the cooling coil control volume. Therefore, the increase in the RR leads to reduction in cooling coil load, which consequently decreases the irreversibility. According to equation (21), the second law of thermodynamic efficiency is expected to increase as the irreversibility decreases.

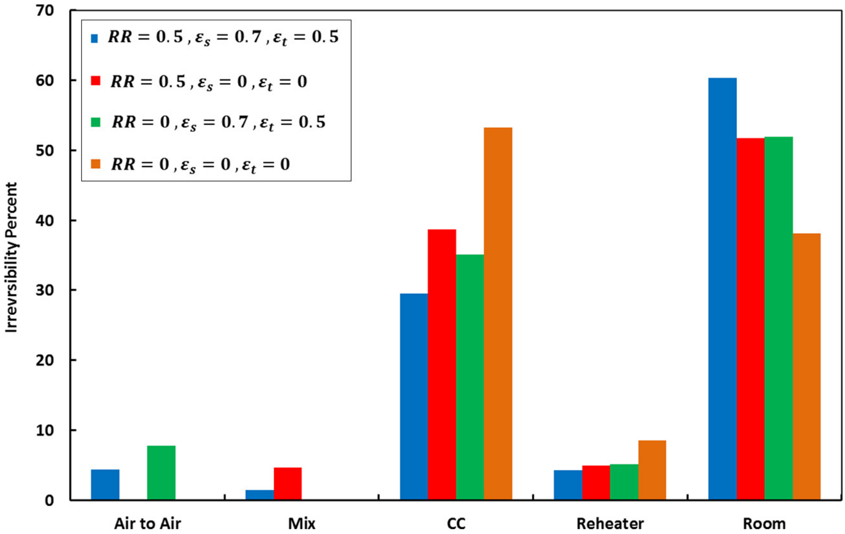

Figure 4.
Exergy loss analysis for different parts of the AHU

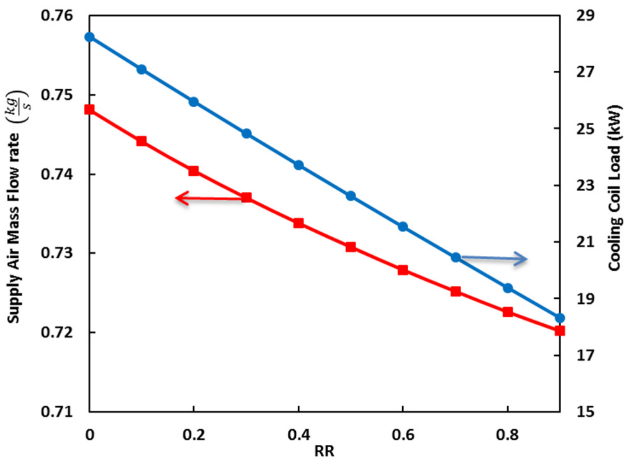

Figure 5.
Variations in the supply air mass flow rate and cooling coil load with respect to RR parameter

To investigate the effect of using an air-to-air heat exchanger in dry and humid environments, the relative humidity of the input air was changed. The variations in the cooling coil load and efficiency with respect to the ambient relative humidity are shown in Figure 7.

In case the amount of heat transferred to the room is considered constant ($Q_s, Q_l = cte$), the cooling coil load increases by increasing the relative humidity of the input air, as a larger amount of vapor is condensed over the cooling coil. Considering the high amount of vapor latent heat, more power is consumed in the cooling coil as the amount of condensed vapor is increased. Consequently, according to Figure 7, the cooling coil load would be increased almost linearly by increasing the inlet fresh relative humidity. The presence of air-to-air slightly modifies the humidity of the inlet air. Therefore, the cooling coil load rate of change for the case with heat recovery is less. According to equation (20), the first efficiency is

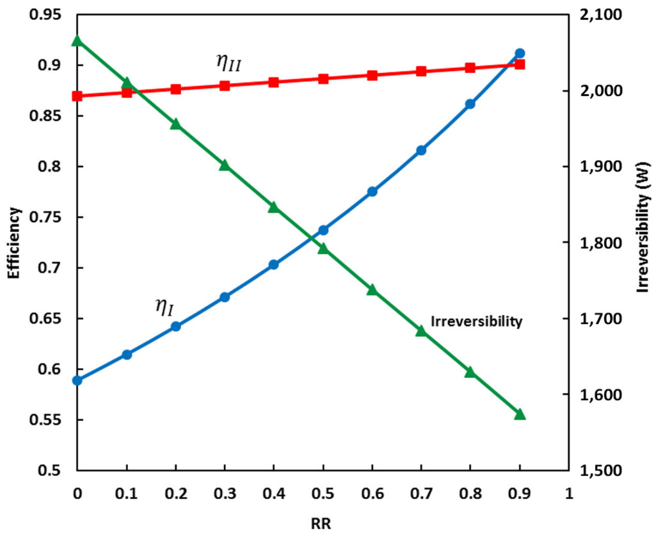

Figure 6. Variations in the exergy loss along with the first and second law efficiencies with respect to RR parameter

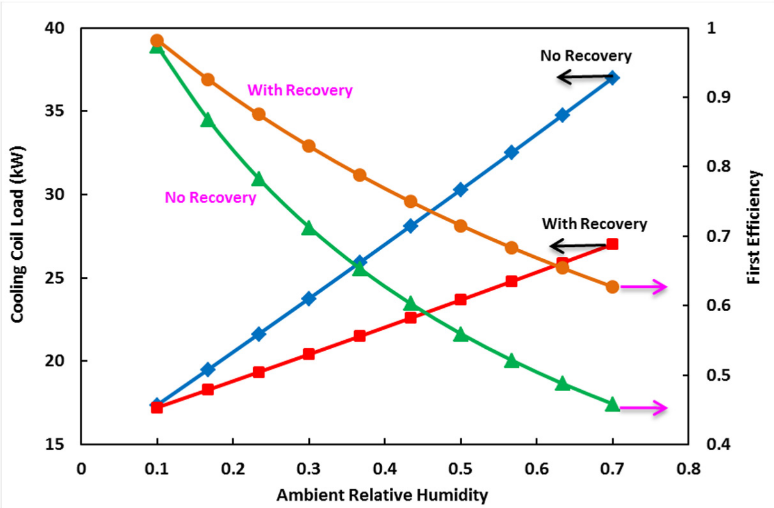

Figure 7. Variations in the cooling coil load and first law efficiency with respect to the ambient humidity

influenced by the cooling coil load. As the cooling coil increases, the first efficiency decreases. Therefore, Figure 7 shows that the first efficiency decreases with an increase in the ambient relative humidity.

According to Table III, In the absence of an air-to-air heat exchanger, increasing the relative humidity from 10 per cent to 70 per cent increases the cooling coil load (from 17.379 to 37 kW) and decreases the first law of efficiency (from 0.9736 to 0.4582) by 113 per cent and 53 per cent, respectively. In the presence of air-to-air heat exchanger, increasing the relative humidity from 10 per cent to 70 per cent increases the cooling coil load (from 17.218 to 27.022 kW) and decreases the efficiency (from 0.9826 to 0.627) by 56.8 per cent and 36.2 per cent, respectively. In other words, it is concluded that the air-to-air heat exchanger moderates the variations.

Variations in the irreversibility and the second law of efficiency with respect to relative humidity are demonstrated in Figure 8. As shown in Figure 7, increasing the relative humidity leads to increase in the cooling coil load, which in turn increases the irreversibility as indicated in Figure 8. The increase in irreversibility causes the system to further deviate from its ideal behavior, which in turn reduces the second law efficiency as shown in Figure 8.

The effect of using air-to-air heat exchanger in dry and humid conditions is compared in Table III. As shown, using heat exchanger in the humid conditions has a greater effect. For instance, using the air-to-air heat exchanger in dry conditions decreased the coil power from 17.379 kW to 17.218 kW (0.92 per cent reduction), whereas this reduction was 27 per cent for the humid conditions (from 37.007 kW to 27.022 kW). Similarly, by using air-to-air heat exchanger, the first law efficiency in the dry and humid conditions was increased by 0.9 per cent and 36.8 per cent, respectively. The second law efficiency was increased by 1.55 per cent and 2.77 per cent by using the air-to-air heat exchanger in dry and humid conditions, respectively. As pointed out, the effect of using air-to-air heat exchanger in a humid condition is greater than that of a dry one.

5. Conclusion

Consumption of energy in buildings accounts for 40 per cent of the total energy consumption in the world. Air-conditioning is one of the energy-intensive systems in buildings, for which some

Table III. The effect of using air-to-air heat exchanger in dry and humid air

Ambient relative humidity	Cooling coil load (kW) NR	WR	First efficiency NR	WR	Second efficiency NR	WR
0.1	17.379	17.218	0.9736	0.9826	0.9078	0.9219
0.7	37.007	27.022	0.4582	0.627	0.8331	0.8562

Notes: WR = AHU with recovery, NR = no recovery in AHU

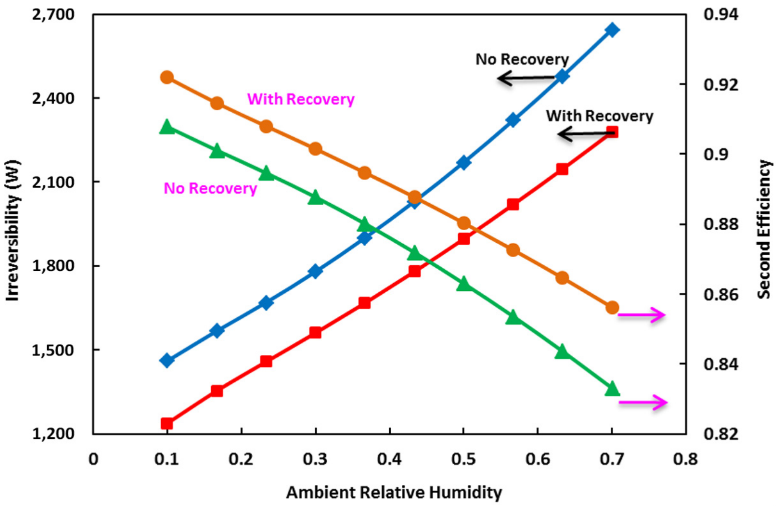

Figure 8. Variations of irreversibility and second law efficiency with respect to the ambient humidity

techniques must be devised to reduce the required energy. For instance, the consumed energy can be reduced by using air-to-air heat exchangers. Generally, two types of AHU are available: 1) air-conditioning system in which the mass flow rate is not given and should be determined through thermodynamic analysis and 2) air-conditioning system in which the flow rate is determined through the amount of fresh air required by the residents, as well as the times of air changes per hour. In other words, the system flow rate is pre-determined and independent of the thermodynamic analysis of the AHU. In this study, the first type of AHU was analyzed using the first and second laws of thermodynamics. First, the thermodynamic equations were extracted based on the first and second laws of thermodynamics, and then the equations were solved in EES software. The results are as follows:

- Using air-to-air heat exchangers decreases the air mass flow rate and inlet air temperature to cooling coil, simultaneously. On the other hand, the cooling coil load is dependent on both parameters. The cooling coil load decreases as the air mass flow rate and inlet air temperature decrease.
- The effect of using air-to-air heat exchanger in AHU with 100 per cent fresh air is more than the AHU with return air. In other words, the effect of using air-to-air heat exchanger is reduced as the return air quota is increased. By using air-to-air heat exchanger in an AHU with 100 per cent fresh air, the cooling coil load and the irreversibility decrease by 32.2 per cent and 23.9 per cent, respectively, which in turn increase the first and second laws of efficiencies by 49.1 per cent and 4.9 per cent, respectively. By using an air-to-air heat exchanger in an AHU with 50 per cent return air, the cooling coil load and the irreversibility decrease by 19.7 per cent and 13.6 per cent, which in turn increase the first and second laws of efficiencies by 24.7 per cent and 2 per cent, respectively.
- For constant sensible and latent heat transfer to the room (Q_s,Q_l = *cte*), as the inlet air relative humidity increases, the cooling coil load increases because of more condensation. Therefore, the first law of efficiency decreases. However, by using an air-to-air heat exchanger, the increase in the cooling coil load and decrease in the first law of efficiency are alleviated. In the absence of an air-to-air heat exchanger, increasing the relative humidity from 10 to 70 per cent increases the cooling coil load and decreases the first law of efficiency by 113 per cent and 53 per cent, respectively. In the presence of air-to-air heat exchanger, increasing the relative humidity from 10 to 70 per cent increases the cooling coil load and decreases the first efficiency by 56.8 per cent and 36.2 per cent, respectively.
- The effect of using air-to-air heat exchanger is influenced obviously by the ambient relative humidity. Using the heat exchanger in dry conditions decreases the cooling coil load by 0.9 per cent, while the reduction is 27 per cent for humid conditions. Similarly, using air-to-air heat exchanger leads to an increase in the first law of efficiency in dry and humid conditions by 0.9 per cent and 36.8 per cent, respectively. Also the second law of efficiency increases by 1.55 per cent and 2.77 per cent in dry and humid conditions, respectively. In other words, the effect of using air-to-air heat exchanger in humid conditions is more than that in dry conditions.

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