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Original Research

Design and analytical investigation on air-to-air cross flow heat exchanger of an industrial heat recovery ventilation system

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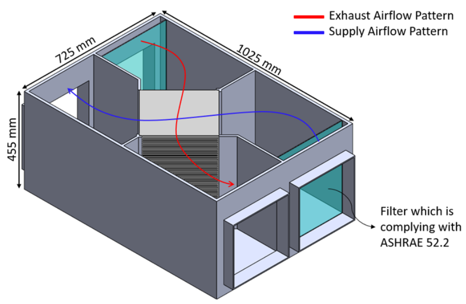

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Abstract: Energy consumption based on the building heating, ventilation, and air-conditioning (HVAC) systems is sharply rising daily. At this point, heat recovery ventilation systems save energy while contributing to indoor air quality and thermal comfort. This work mainly focuses on designing the air-to-air cross-flow heat exchanger system and duct lines. First, the ducts, bends, vents, and heat recovery ventilation unit are placed around the selected domain complying with the ASHRAE 55 and 62.1 standards. The requirements on temperature levels of the cold and hot streams, air flow rates, number of vents, velocity at the supply air vent outlets are considered. Then, calculations are conducted for the cross-flow air-to-air heat exchanger to determine the number of layers, heat transfer surface areas, flow regime, and heat transfer rate. Thermal calculations of the recuperator system are initially performed by effectiveness-number of transfer unit (ϵ-NTU) method as the outlet temperatures are not known at the beginning of the design. In addition, the findings are compared and validated via logarithmic mean temperature difference methodology. The results show that the fresh air temperature can increase from 5 °C to 13.32 °C when the exhaust air temperature is at 26 °C in winter. Furthermore, the heat transfer rate of air-to-air heat exchanger system is analytically calculated as 1806.7 W and 1807.5 W via ϵ-NTU and logarithmic mean temperature difference (LMTD) methods, respectively.

Keywords: Heat recovery ventilation, indoor air quality, recuperator, heat exchanger.

Introduction

Due to population expansion and economic growth, energy consumption is rising daily. International Energy Agency (IEA) reports that almost 27% of total energy sector emissions and 30% of global energy consumption are accounted for buildings [1]. At this point, heating, ventilating, air conditioning, and refrigeration (HVAC-R) units are the dominant systems consuming about 40–60% of the total building-based energy [2,3]. Furthermore, mechanical ventilation systems account for more than 26% of HVAC-R-based energy consumption [4]. From a global perspective, the energy consumption of ventilation systems results in much carbon emissions that significantly impacts climate change [5].

Heat recovery ventilation (HRV) systems contribute to reducing the energy consumption of HVAC-R units via heat exchange among the cold and hot air streams [5–7]. HRV units contain air-to-air heat exchanger(s), fan systems, filters, and controllers [8]. Unmixed hot and cold air streams sweep the heat transfer surfaces of the recuperator and exchange thermal energy from more energetic stream to a less energetic one [9, 10]. The most of the literature on heat recovery in buildings concentrates on energy efficiency and primary energy use. Depending on the heating system and weather conditions, HRV can generally provide significant energy savings. Carlsson et al. [11] reported that the integration of in-suite HRV and enclosure retrofit reduces the total heating energy load about 78%, that corresponds to an 83% decrease in greenhouse gas emissions. Dodoo et al. [12] investigated the trade-off between lowering the need for space heating and increasing the energy used for ventilation. The findings indicated that HRV systems might save more energy than non-electric heated buildings. Likewise, Wahlström et al. [13] reported heat savings of 25–42 kWh/m² and an annual increase in energy usage of 2–8 kWh/m² in a case study of 14 Swedish buildings with HRV systems. Hamid et al. [7] combined the models and field data. The results showed that HRV systems within the heritage buildings provide about 39 kWh/m², saving overall energy consumption. Evola et al. [14] focused on the effects of HRV on Italian structures' energy demand. The research ended up with primary energy savings between 20% and 90% compared to mechanical exhaust and natural ventilation systems, respectively. On the other hand, an investigation by Logue et al. [15] showed that adding an HRV system to existing US homes slightly increases total energy consumption due to fans. Fouih et al. [16] analysed the performance of HRV systems for low-energy demand buildings in France and detected them energy efficient for residential buildings, in contrast to commercial buildings.

Heat exchangers within the HRV systems mainly reveal the thermal efficiency and pressure drop of the HRV unit. Lu et al. [17] theoretically and experimentally worked on plastic film plate heat exchanger for heat recovery ventilation. The results of the cross-flow system indicated that the effectiveness of the heat recovery system varies from 0.65 to 0.85 under different flow rates. On the other hand, Fernandez-Seara et al. [18] conducted an experimental analysis of an air-to-air heat recovery unit with a fixed plate heat exchanger made of a suitable polymer for balanced ventilation systems in residential buildings. The findings showed that relative humidity dropped from 95% to about 34% with the proposed HRV system. Furthermore, the heat transfer rate and thermal efficiency of the steady-state operation were measured as 672 W and 80%, respectively.

In this work, we design a fixed plate air-to-air heat exchanger for heat recovery-aided ventilation of a 200 m² R&D office domain occupied by 10 researchers. The main goal and motivation of the research is to satisfy the indoor air quality and personal comfort requirements of the selected ventilation domain via the proposed HRV unit and ventilation duct network. In parallel, total pressure losses experienced by the ducts, vents, HRV unit, and heat exchanger channels are investigated to reduce the total energy consumption. First, outdoor air intake and zone minimum primary airflow are calculated according to the ASHRAE 62.1 standard to satisfy indoor air quality conditions. Furthermore, duct network, vents (complies with the ASHRAE 55), and HRV position are carefully designed to satisfy less energy consumption during the ventilation. Then, fixed plate cross flow heat exchanger and other parts of the HRV unit have been integrated to the ventilation duct network. Heat transfer rates and outlet temperatures of the cold and hot streams are calculated with ϵ-NTU method. In addition, the results are double-checked and confirmed via the LMTD method.

Method and Model

This section presents the main methodology, calculation steps, and design processes of the proposed heat recovery ventilation network. The subsections contain the calculation procedures for the indoor air quality requirements, supply and exhaust flow rates, heat recovery ventilation network, and air-to-air heat exchanger system.

Ventilation requirements and flow regime

This work determines the required ventilation levels and indoor air quality conditions according to ASHRAE 62.1 [19] standard. The case is selected as a 200 m² ventilation domain with 10 occupants. In a ventilation system, the outdoor air intake flow (V_{ot}) should be calculated for every specific domain to ensure enough oxygen and that the air is not contaminated. At this point, breathing zone outdoor airflow (V_{bz}) must be determined as follows:

$$V_{bz} = R_p \ P_z + R_a \ A_z \tag{1}$$

where A_z and P_z are zone floor area and zone population, respectively. R_p refers to the outdoor airflow rate per person, and R_a is the outdoor airflow rate per area determined from ASHRAE 62.1 Table 6-1 for the office space scenario. Provided the zone outdoor airflow (V_{oz}) to the ventilation zone can be calculated via:

$$V_{oz} = \frac{V_{bz}}{E_z} \tag{2}$$

where E_z is the zone air distribution effectiveness taken from ASHRAE 62.1 standard. For single-zone ventilation systems with one supply air stream, outdoor air intake flow (V_{ot}) shall be calculated as:

$$V_{\mathbf{ot}} = V_{oz} \tag{3}$$

Then, the minimum primary airflow ($V_{pz\text{-}min}$) is calculated via outdoor air intake and a safety factor to satisfy indoor air quality for the zone.

$$V_{pz\text{-}min} = V_{oz} \ 1.5 \tag{4}$$

To avoid infiltration in the building, supply flow rate assumed to be 33% higher than the exhaust flow rate corresponding to study by Ali et al. [20]. Note that the exhaust and supply airflow rates have been calculated as 459 m³/h (0.13 m³/s) and 612 m³/h (0.17 m³/s), respectively. The average velocity and mass flow rate are calculated for 300 mm × 300 mm duct cross-section.

$$\dot{V} = V_{avg} \ A_c \tag{5}$$

$$\dot{m} = \rho.V_{avg} \ A_c \tag{6}$$

Here, A_c is the cross-sectional area of duct, and ρ is the density of air at specific temperature. $\dot{V}$ and $\dot{m}$ denote the volumetric and mass flow rates, respectively. For non-circular tubes or ducts, hydraulic diameter (D_{hyd}) approach is used to transform them into equivalent diameter pipes. The hydraulic diameter of the ventilation ducts can be calculated with following equation:

$$D_{hyd} = \frac{4A_c}{P} \tag{7}$$

where *P* is the wetted perimeter which is the perimeter swept by the cold and hot air streams. On the other hand, Reynolds number (Re) shall be *calculated* with the following equation to check the flow regime within the duct and heat recovery ventilation units [21, 22]:

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$$\mathrm{Re} = \frac{\text{Inertia forces}}{\text{Viscous forces}} = \frac{\rho\, V_{avg}\, D_{hyd}}{\mu} = \frac{V_{avg}\, D_{hyd}}{\nu} \qquad (8)$$

where ν and μ represent kinematic and dynamic viscosity, respectively. As the cold and hot air streams experience duct flow, the critical Reynolds number of the laminar regime is determined as 2000 under internal flow conditions.

Thermal investigations on heat recovery system

Energy consumption of thermal-fluid systems under laminar flow regime is comparatively lower than the turbulent flow systems as the turbulence effects and vortices increase the pressure losses [23]. Therefore, we aimed to satisfy indoor air quality and thermal performance criteria under laminar flow regime ($Re_{max} \leq 2000$). The flow is assumed as under steady-state conditions for each flow stream. Nusselt number can be calculated by following equation to determine dimensionless heat transfer:

$$\mathrm{Nu} = \frac{q_{convection}}{q_{conduction}} = \frac{h\, D_{hyd}}{k_f} \qquad (9)$$

Where h is the convective heat transfer coefficient, K_f and refers to the fluid thermal conductivity. Nusselt number is the ratio of convective and conductive heat transfer rates. As we design laminar flow in ducts and fixed plate heat exchanger systems, the Nusselt number is taken from Table 1.

Table 1. Nusselt number and friction factor for fully developed laminar flow in tubes.

	Nusselt Number		Friction Factor
a/b	**T_s = Const.**	**$\dot{q}_s$ = Const.**	**f**
1	2.98	3.61	56.92/Re
2	3.39	4.12	62.20/Re
…	…	…	…
∞	7.54	8.24	96.00/Re

where a is long, and b is the short side of the spacing. A membrane-fixed plate heat exchanger exchanges waste heat between the cold and hot air streams. Thermal resistances (R_{cond} and R_{conv}) and overall heat transfer coefficient (U) are calculated via the following equations [24,25]:

$$R_{cond} = \frac{L}{kA} \qquad (10)$$

$$\frac{1}{U\, A_s} = \frac{1}{h_i\, A_i} + \frac{1}{h_o\, A_o} + R_{cond} \qquad (11)$$

As the wall thickness of the heat exchanger plates is small (L = 1 mm) and the thermal conductivity (k) of the membrane is high, the relation can be written as:

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o} \qquad (12)$$

In heat exchanger analysis, heat capacity represents the required heat to alter a given matter's temperature by 1°C. As we have two different fluids, two heat capacity levels of and are calculated (Eq. (13)) at estimated film temperatures, and the procedure was double-checked at the end of the thermal calculations. Note that relation among the cold and hot stream heat capacities is called the heat capacity ratio which is given in Eq. (14) [24, 25]:

$$C_{hot} = \dot{m}_{hot}\, c_{p,\,hot} \ \text{ and } \ C_{cold} = \dot{m}_{cold}\, c_{p,\,cold} \qquad (13)$$

$$c = \frac{C_{min}}{C_{max}} \qquad (14)$$

First, the ϵ-NTU method is applied for the investigated HRV heat exchanger component because of the unknown outlet temperature levels of the cold and hot streams. NTU is calculated by combining the overall heat transfer coefficient and minimum heat capacity (Eq. (15)). The particular reason for C_{min} usage instead of C_{max} is that the system intends to work for fluid with low heat capacity. The relation between NTU and effectiveness can be determined for unmixed cross-flow heat exchanger as [24–27]:

$$\mathrm{NTU} = \frac{U\, A_s}{C_{min}} \qquad (15)$$

$$\epsilon = 1 - \exp\left\{\frac{\mathrm{NTU}^{0.22}}{c}\ \left[\exp(-c\ \mathrm{NTU}^{0.78}) - 1\right]\right\} \qquad (16)$$

The effectiveness value is then used to calculate the actual heat transfer rate, which is the amount of transferred heat per unit of time between the fluids. On the other hand, the relationship between maximum heat transfer rate and effectiveness can be expressed as:

$$\dot{Q}_{max} = C_{min}\ (T_{hot,\,in} - T_{cold,\,in}) \qquad (17)$$

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}} \qquad (18)$$

Next, the outlet temperature can be calculated via the actual transferred heat and the heat capacities. The cold and hot stream outlet temperatures are given as:

$$T_{cold,\,out} = T_{cold,\,in} + \frac{\dot{Q}}{C_{cold}} \qquad (19)$$

$$T_{hot,\,out} = T_{hot,\,in} - \frac{\dot{Q}}{C_{hot}} \qquad (20)$$

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We also use logarithmic mean temperature difference (LMTD) method to validate our thermal results. This strategy needs outlet temperatures of cold and hot fluids to derive actual heat transfer rate. As the fluids experience cross-flow within the heat exchanger, the temperature differences and heat transfer rate are calculated as follows [24,25]:

$$\Delta T_1 = T_{hot,\,in} - T_{cold,\,in} \tag{21}$$

$$\Delta T_2 = T_{hot,\,out} - T_{cold,\,out} \tag{22}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \tag{23}$$

$$\dot{Q} = \boldsymbol{f}\ U\ A_s\ \Delta T_{lm} \tag{24}$$

Where f is correction factor which is measure of deviation of the ΔT_{lm} for heat exchangers [24].

Results and Discussion

Based on the ventilation procedure and thermal investigation, we aimed to design a heat exchanger and HRV device for our case. We utilized Eq. (1) to determine the breathing zone, calculated as 306 m^3/h for 10 occupants in the office domain. According to ASHRAE 62.1 Standard, zone air distribution effectiveness equals one when the cold stream is supplied from the ceiling of the selected domain. In this case, zone outdoor airflow and single zone system airflow are identical (Eqs. (2) and (3)). Next, zone minimum primary airflow is calculated via Eq. (4), and the value is 459 m^3/h for a 1.5 safety factor. We assumed that the supply air flow for the selected space is one-third greater than the exhaust airflow; therefore, the exhaust airflow and supply airflow have been calculated as 459 m^3/h (0.13 m^3/s) and 612 m^3/h (0.17 m^3/s), respectively.

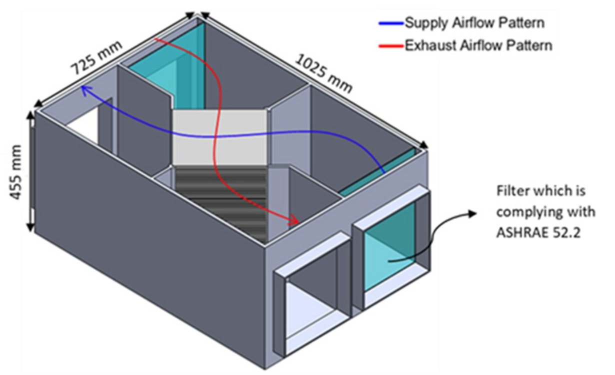

Figure 1. Designed HRV device dimensions and flow patterns.

The main dimensions of the HRV device are presented in Figure 1 with supply and exhaust air stream patterns. Furthermore, proposed crossflow heat exchanger dimensions are shown in Figure 2. The maximum Reynolds number within the heat exchanger channels is aimed at 2000 to ensure a laminar flow regime conditions as the energy consumption of thermal-fluid systems under laminar flow regime is comparatively lower than the turbulent flow systems due to the turbulence effects and vortices. Hydraulic diameter and average velocity level of each flow channel are calculated as 0.0079 m and 3.5 m/s (Re ≈ 2000) for supply air stream in crossflow heat exchanger spacing. By dividing the supply airflow rate by the calculated velocity, we obtained the frontal space area as 0.048 m^2. Then, dividing the frontal space area by the cross-sectional area of one air gap (4 mm x 290 mm) means that the required number of spacing is about 41.79 ≈ 42. According to the determination of the spacing number, exhaust air stream velocity in each heat exchanger space can be calculated as 2.67 m/s (Re ≈ 1345).

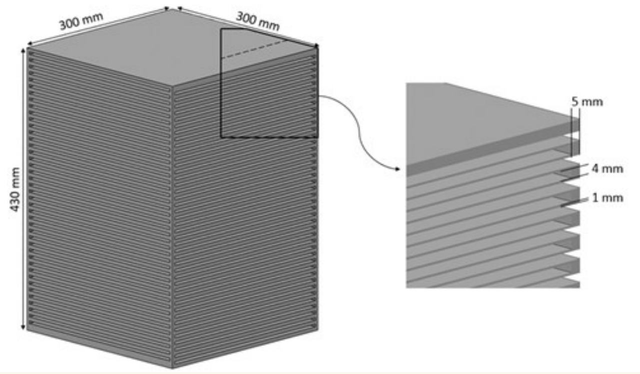

Figure 2. Fixed plate heat exchanger design and main dimensions.

We assumed constant heat flux in the heat exchanger. For this reason, Nusselt Number is 8.24, according to Table 1. Then, thermal investigation continuous with heat transfer coefficient and determined for supply (h_i) and exhaust (h_o) air as; 25.04 W/(m^2K) and 26.7 W/(m^2K), respectively, corresponding to 5 °C and 26 °C air properties at 1 atm pressure. The membrane material is a critical issue for cost and heat transfer rate. R_{cond} is calculated for the most commonly used materials, as shown in Table 2.

Table 2. Commonly used material's properties.

	k (W/mK)	R_{cond} (K/W)	R_{conv} (m^2K/W)	ρ (kg/m^3)
Aluminum	237	4.85×10^{-5}		2705
Copper	398	2.88×10^{-5}	7.7×10^{-2}	8944
HDPE	0.5	2.3×10^{-2}		970

The comparison indicates that using copper and aluminium will effectively enhance the heat transfer rate instead of high density polyethylene (HDPE). In Table 2, the thermal resistance of the aluminium and copper heat exchanger channels are very close due to the high thermal conductivity of these materials. We continued with aluminium because aluminium is lighter, more flexible, and twice cheaper than copper. Note that the thermal conduction resistance of aluminium is comparatively small (≈ 100 times) compared to the convective resistance. Therefore, conductive resistance has been neglected in the overall heat transfer coefficient calculations. According to this strategy, the overall heat transfer coefficient of the designed crossflow heat exchanger channel is about 12.92 W/(m²K) (Eq. (12)). Before moving on to the ϵ-NTU method, we need heat transfer area and heat capacities. The total heat transfer area of the 42 gaps crossflow heat exchanger system is 14.4 m². Furthermore, heat capacities of the supply and exhaust air streams are 217 W/K (C_{max}) and 151 W/K (C_{min}), respectively.

NTU and effectiveness for cross-flow heat exchangers with unmixed fluids are calculated as 1.23 and 0.568, respectively. Next, the effectiveness value determines the maximum and actual heat transfer rates. Results indicate that maximum heat transfer rate is 3181 W while the actual heat transfer rate is 1806.7 W. Finally, the supply and exhaust stream outlet temperatures are calculated as 13.32 °C and 14.01 °C, respectively. Figure 3 presents the trend in temperature distribution within the HRV device. Once the outlet temperature of the supply and exhaust air streams are calculated via the ϵ-NTU method, we conducted a validation study to double-check our thermal results. At this point, the LMTD method is utilized to compare the actual heat transfer rate obtained by the -NTU method. The LMTD method results show that the actual heat transfer rate is 1807.5 W for the calculated correction factor equal to 0.905. Actual heat transfer rate results represent that the discrepancy between the LMTD and ϵ-NTU methods is less than 0.044%.

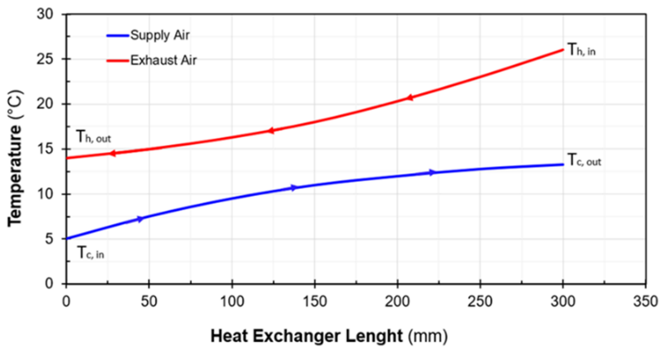

Figure 3. Temperature variation within the HRV heat exchanger system.

Conclusion

In this study, the thermal performance of an industrial cross-flow fixed aluminum plate heat exchanger has been analytically investigated for a 200 m² ventilation domain with 10 occupants. Indoor air quality requirements, ventilation rates and personal comfort conditions have been addressed complying with ASHRAE 62.1 and 55 standards. The thermal performance of the investigated HRV unit crossflow heat exchanger system is calculated via ϵ-NTU and LMTD methods to double-check the results. We determined the design parameters of the fixed plate heat exchanger according to the thermal and hydraulic needs. We aimed to reduce the total pressure drop as much as possible while satisfying high-performance conditions. The main conclusions are given as follows:

- 459 m³/h flow rate is the threshold for both supply and exhaust air streams, yet we increased supply air flow rate by 33% (up to 612 m³/h) to prevent infiltration risks.
- The laminar flow regime (Re ≤ 2000) is aimed within the HRV unit heat exchanger spacings to minimize pressure losses and energy consumption. The total number of fixed plates within the crossflow heat exchanger system a laminar flow regime and enhanced heat transfer is calculated as 42.
- The actual heat transfer rate is calculated as 1806.7 W with the help of the ϵ-NTU method. At this point, outlet temperatures of the supply and exhaust air streams are obtained as 13.32 °C and 14.01 °C, respectively.
- The results have been double-checked with the LMTD method to ensure the accuracy of the findings. LMTD results indicate that the actual heat transfer rate of the investigated HRV system is 1807.5 W. That corresponds only a 0.044% discrepancy between the ϵ-NTU and LMTD methods.

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Authors Contribution

E. Tas: Conceptualization, Writing; S. Gungor: Investigation, Writing - review and editing. All authors have approved the final version of the manuscript.

Conflicts of Interest

The authors have declare no conflicts of interest

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