Design and Optimization of Cross-corrugated Triangular Ducts Towards Reducing CO₂ Emission and Improving Energy Efficiency

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ABSTRACT

Reducing CO₂ emission can effectively control global warming. And energy conservation is the most scientific and systematic ways to reduce CO₂ emissions. The total heat exchangers with membrane core, which consist of cross-corrugated triangular ducts obviously were considered as the most efficient, energy-saving, environmentally friendly air-conditioning heat exchanger. Both simulation and experiment results indicated that, in the cross-corrugated triangular ducts, the flow intensification and the heat transfer efficiency could be significantly improved, which means the energy was saved and the emission of CO₂ was reduced. Further studies revealed that the different included angles affected the fluid flow and heat transfer parameters. To disclose the heat transfer enhancement mechanism in the channel, the field synergies had been investigated among velocity gradient, temperature gradient and pressure gradient, which have been validated coincident with the performances of the macroscopic heat transfer parameters. Moreover, the volume-averaged synergy angle took the minimum when the included angle of cross-corrugated triangular ducts took 90° with a Reynolds number of 2000, which was 1–2.3° less than the volume-averaged synergy angles of other included angles with the same Reynolds numbers. For the sake of illustrating the advantage of this channel, four flow channels were compared with the cross-corrugated triangular ducts. And the simulation results reveal that the cross-corrugated triangular ducts have the most outstanding field synergies effect among velocity gradient, temperature gradient and pressure gradient. The total heat exchanger with cross triangular corrugated channel composed of moisture permeable membrane can realize sensible heat and latent heat exchange, and the energy saving is up to 45%. It is supposed that the electricity consumption 76.2 billion kW·h of electricity yearly if this technology is applied in Guangdong, which means bring 1.4×10^5 tons of carbon emission reduction or 12% of the target of reducing carbon emissions in Guangdong.

Keywords: Cross-corrugated triangular ducts, Synergy angles, Total heat exchanger, Energy conservation, Carbon emission reduction

1 INTRODUCTION

The earth has entered a climate emergency. According to the data of the World Meteorological Organization in 2019, earth surface average temperatures were 1.1°C warmer than two hundred years ago. To avoid any further deterioration, the first response is to stabilize the cumulative emission of carbon dioxide in the atmosphere and control the greenhouse effect. On 2020 September 12th, the Chinese government announced that it would increase "National Determined Contributions" to strive for a peak in CO₂ emissions by 2030 and for carbon neutrality by 2060, and made it clear that non fossil energy will account for about 25% of primary energy consumption by 2030 and more than 85% by 2060. Three ways are regarded as the scientific and systematic

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ways to promote the carbon reduction schemes: energy conservation, energy substitution and de-energy. Energy conservation focuses on the conservation of energy resources by the use of management measures and technologies, so as to realize pollutant reduction. Energy substitution refers to the use of clean energy sources such as hydro, nuclear, wind, photovoltaic, geothermal or marine to replace fossil energy sources. Energy substitution includes two modes: incremental substitution and stock substitution. De-energy means to adjust the energy structure and decrease the high-hazard and high-pollution enterprises' impact on the environment, mainly through the measures including industrial transfer, shutdown and other ways (Li and Lou, 2019). Although coal-fired power has declined from 82% to 68% over the past decade, it remains the main source of electricity and has a stable generating capacity. In order to solve the problem of reducing emissions, China makes constant efforts to increase the proportion of water, nuclear, wind, and solar energy. The heat exchanger, which links with all kinds of energy, should improve energy efficiency firstly. And the heat transfer enhancement technology has developed to the fourth generation. Fig. 1 shows the composition of electricity consumption and the carbon emission reduction in recently years (National Bureau of Statistics of the People's Republic of China, 2020).

Enough outdoor fresh air is often necessary for Variable Air Volume System to obtain good indoor air quality. However, plenty of fresh air also brings forth massive fresh air load. For most regions, the reheat energy consumption of fresh air account for 19.0%–22.8% of the total energy consumption of fresh air. Most researchers identified the total heat exchanger as an efficient, energy-saving, environmentally friendly air-conditioning heat exchanger. Besides energy wheels and metal plate sensible heat exchangers, in recent years, membrane-based energy recovery ventilators have been applied for heat and moisture exchange between the fresh and the exhaust air. Based on this concept, heat and moisture in the fresh air are transferred to the exhaust air in summer, past a novel hydrophilic membrane core, thus reducing the cooling load. By going through a unique penetrability filter film, the fresh outdoor air is sent to the indoor room after filtration, purification and heat recovery process. In contrast, harmful indoor gases are sent outside after the same process. During this process, the indoor temperature is independent of fresh air. This kind of special penetrability filter film may be created a proper flow channel alone or with other materials. Owing to the innovation of novel membrane materials with outstanding vapor permeability and high water/air selectivity, heat and mass transfer mechanisms in the exchanger have drawn the same attention. And the cross-corrugated triangular flow-path is a new structure of the total heat exchanger (See in Fig. 2). Firstly, the heat transfer surface is folded into a triangular corrugated board. Then the boards are stacked together, which the neighboring board keeps vertical (Zhang, 2009). The cross-corrugated triangular heat exchanger is a simple structure and a moderate strength, as well as the internal flow is enhanced by the periodic expansion and contraction in the flowing channels, giving rise to means of the convective heat transfer (Scott and Labato, 2003). Due to the supporting role of multi-point between the neighboring corrugated board, its overall strength is much more stable. Even in the case of thin wall thickness of the heat

Fig. 1. Electricity consumption and quantity of carbon emission reduction in China.

Fig. 2. The structure of cross-corrugated triangular ducts.

transfer surface, it also maintains a relatively high mechanical strength and higher pressure capability. The structure greatly enhanced the heat and mass transfer and this efficiency improvement is attributed to the pattern of flow that undergoes abrupt turnaround, contraction and expansion. Scholars get used to confine themselves to the simulation on the cross-corrugated triangular flow channel, but little experimental data were reported (Stasiek *et al.*, 1996; Ciofalo *et al.*, 1998; Ergin and Ota, 2005; Liu *et al.*, 2013; Liang *et al.*, 2014; Niu *et al.*, 2002).

Zhang (2006) and Zhang *et al.* (2011) carried out some simulation and experimental study on the cross-corrugated triangular ducts. Besides, the macroscopic heat transfer parameters, such as fully developed local or cyclic mean friction factor (f_D), fully developed local or cyclic mean Nusselt numbers (Nu_D), heat transfer factor (j), performance evaluation criteria value of heat transfer (PEC), were calculated before (Chen *et al.*, 2013). However, the microscopic mechanism of heat transfer enhancement has not been found. Excellent heat transfer performance of the cross-corrugated triangular ducts can save energy, which corresponds to reduction carbon dioxide emission from thermal power plants.

Based on the field synergies, effects of the included angles on the fluid flow and heat transfer are validated by analyzing the contours of local synergy angles for different included angles. Further, the contours of the local synergy angles under different wall conditions are analyzed and compared through numerical simulation and preparation of custom functions UDF. Two kinds of average synergy angles are achieved and named as volume mean value and integral mean value. Because both of them follow a consistent pattern except that the latter is 5° bigger than the former, the volume mean value of the field-averaged synergy angle (β_{VOL}) are selected. The results were later verified by Chen (2016) that the cross-corrugated triangular ducts have the most excellent field synergies effect on velocity gradient, temperature gradient and pressure gradient (Chen, 2016).

The plate of heat exchanger uses specially processed paper or membrane on the diaphragm to exchange heat or heat and mass through the temperature difference between the two sides of the membrane and the partial pressure difference of water vapor. Energy-saving effect evaluation of total heat exchanger with the structure of cross-corrugated triangular ducts compare with sensible heat exchanger. Compared to the common sensible heat exchanger, an hour-by-hour simulation reveals that 29–52% of primary energy can be saved by the independent air dehumidification with heat recovery. Especially, 45% energy efficiency was achieved in June to October. It is supposed that the electricity consumption 76.2 billion kW·h of electricity yearly if this technology is applied in Guangdong, which means bring 1.4×10^5 tons of carbon emission reduction or 12% of the target of reducing carbon emissions in Guangdong. In a given calculation example, the coefficient of performance (COP), electricity expense and payback period for different air conditioning system are calculated and analyzed.

2 METHADODOLOGY

2.1 Mathematical Model

The general form of the mass continuity equation as shown below is valid for compressible and incompressible flows:

Continuity equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \quad (1)$$

Governing the momentum

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = \frac{\partial \sigma_{ij}}{\partial x_j} + B_i \quad (2)$$

$$\sigma_{ij} = -p\delta_{ij} + \mu \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] - \frac{2}{3} \mu \frac{\partial u_k}{\partial x_k} \delta_{ij} \quad (3)$$

when $i = j$, $\delta_{ij} = 1$; when $i \neq j$, $\delta_{ij} = 0$

$$\frac{\partial(\rho\phi)}{\partial t} + \frac{\partial}{\partial x_i} \left[\rho u_i \phi - \Gamma \frac{\partial \phi}{\partial x_i} \right] = S \quad (4)$$

Γ is generalized diffusion coefficient; S is generalized source term.

Energy equation

$$\frac{\partial(\rho H)}{\partial t} + \frac{\partial}{\partial x_i} \left[\rho u_i H - \lambda \frac{\partial T}{\partial x_i} \right] = \frac{\partial \rho}{\partial t} \quad (5)$$

2.2 Heat Transfer Macroscopic Parameters

During the modeling process, the included angles are selected 45° , 60° , 90° , 120° . The fully developed average friction coefficient (f_D) and the fully developed average Nusselt number (Nu_D) are obtained by analyzing the data of the numerical simulation and the experimental data respectively. Wherein each parameter is calculated as follows:

$$f_D = \frac{[(P_i - P_o)/L] D_h}{(1/2) \rho u_m^2} \quad (6)$$

$$Nu_D = \frac{h_m D_h}{\lambda} \quad (7)$$

$$h_m = \frac{G \rho c_p (T_o - T_i)}{A \Delta T} \quad (8)$$

$$\Delta T = \frac{(T_w - T_i) - (T_w - T_o)}{\ln[(T_w - T_i)/(T_w - T_o)]} \quad (9)$$

where D_h is hydrodynamic equivalent diameter of the flow channel; h_m is the average heat transfer coefficient.

And the heat transfer factor (j) and the performance evaluation criteria value (PEC) are calculated as follows:

$$j = \frac{Nu_D}{Re Pr^{1/3}} = St \cdot Pr^{2/3} \quad (10)$$

$$PEC = (Nu_D/Nu_c)/(f_D/f_c)^{1/3} \quad (11)$$

where PEC is the performance evaluation criteria, Nu_c and f_c is the Nu and f for circular straight tube at the same Re (Zhang *et al.*, 2000).

2.3 Experimental Setup

Two sets of experimental apparatus were designed to test the flow and heat transfer respectively (see in Figs. 3(B) and 3(C)). And an open circuit system was the shared section, which included five main components: a variable speed blower, a wind tunnel, an upstream section, a test section, and a down-stream section. The flow velocity can be adjusted from 0.13 m s^{-1} to 39 m s^{-1} so as to have different Re numbers from 100 to 30,000. But the experimental testing sections of them have significant differences. The flow testing section was made of acrylic-glass ducts and circumscribed corrugated channels with dozens of holes for the flow detector to plug in (see in Fig. 3(B)). IFA300 was the flow detector, which constant temperature hot wire anemometer, which can continuously measure the flow velocity and realize the optimal frequency response in real time. The length to height of test channel equal to 12 mm. The different sections were connected by couplings and the whole ducting work was enclosed with 5 mm thick glass fiber plate. The heat transfer experimental section was made of two copper plates with 10 mm deep triangular grooves and stacked vertically (see in Fig. 3(C)). The heat source was the heating wire, which twining the upper and lower surfaces. Dozens of point temperatures between the upstream and the down-stream sections can be collected by thermoelectric couples, and the experimental section was surrounded by 3 mm of polystyrene foam.

2.4 Simulation Solution Method

A mathematical model is established as followings. Boundary conditions are defined and a uniform heat transfer surface heat flux temperature and non-slip velocity wall conditions are assumed. At the inlet, velocity is set to uniform and parallel to the corrugation of the upper wall. The governing equations are solved by using finite volume method and pressure correction that employ control-volume based discretization techniques along with a pressure-correction algorithm. The Navier-Stokes equations (N-S equations) are solved by SIMPLEC solution. In contrast, the convective term in the energy equation is solved by first-order upwind implicit approximation, and the diffusive term is by second-order upwind diffusion central difference method with low Re turbulence model number k - ϵ turbulence model. The fluid in the study is selected as air.

Fig. 3 A

Fig. 3 B

Fig. 3 C

Fig. 3. Schematic diagram of experimental and test section. (A) shows the schematic diagram of the experimental apparatus, (B) shows the test section for flow experiment and (C) shows the test section for heat transfer experiment.

Fig. 4. Peripheral structure of geometry.

The grid is divided into an unstructured tetrahedral mesh (Fig. 4), and independency test has been done. Additional notes that the included angles of the grid diagram are selected as 60° , and other three included angles grid diagrams are similar to the diagram of 60° . Numerical simulation calculations were primarily carried out with three different grid densities, 330686, 551142, and 918568 mesh points. The channel fully developed periodic mean pressure drop and temperature change for the two fine grids are almost the same and 11.5% higher than that for the coarse grid. For the finest grids, 918568, the solution time is 165 minutes, which is hard to use practically. Based on the above experience, which establishes the grid independency, the final calculations are performed for the 551142 grids, and the results obtained in this paper refer to the grid geometry mentioned above. The boundary conditions are set as follows: the inlet air is perpendicular to inlet with 300 K temperature, and the heat transfer surface heat flux is 0.53 kW m^{-2} .

3 RESULTS AND DISCUSSION

3.1 Flow Experiment Results and Discussion

Energy spectrum represents the energy of various frequency components. Figs. 5(A), 5(B) and 5(C) shows the frequency spectrum of stream-wise velocity oscillation when Reynolds numbers equal to 165, 550 and 1100, respectively. As shown in the figure, with the increase of Reynolds number and turbulence, the main frequency of main flow is decreasing. When Reynolds number equal to 550 or 1100, there is no obvious main frequency at all, which indicates that the main flow in the channel has tended to turbulence or complete turbulence. That means that the internal flow in the cross-corrugated triangular channels is easier to achieve turbulent state under the lower Reynolds number, because the kind of flow channels greatly strengthen the heat transfer and momentum transfer. The three-dimensional velocity components U_x , U_y and U_z with time were plotted in Fig. 5(D) when Reynolds number was 1100. It can also be found that three directions velocity were significant different fluctuation amplitudes. And the fluctuation amplitude of U_x was about 5 times than that of U_y and U_z .

3.2 Heat Transfer Experiment Results and Discussion

In the heat transfer experiment, the cross-corrugated triangular ducts with a fold angle of 60° was selected. When the velocity of the inlet air was changed, we could obtain the macroscopic heat transfer parameters, such as fully developed local or cyclic mean friction factor (f_D), fully developed local or cyclic mean Nusselt numbers (Nu_D), heat transfer factor (j), performance evaluation criteria value of heat transfer (PEC). With the increase of Reynolds number from 250 to 5000, the mean friction factor decreases from 0.52 to 0.17, which satisfies the linear relationship in the double logarithmic coordinates. And there are similar patterns for Nu_D and j . The linear fitting equation is as following:

$$f_D = 4.17254 \text{Re}^{-0.39101}, \text{ and the value of linearly dependent is } 0.985.$$

$$Nu_D = 0.07724 \text{Re}^{0.74353}, \text{ and the value of linearly dependent is } 0.992.$$

$$j = 0.081255 \text{Re}^{-0.26488}, \text{ and the value of linearly dependent is } 0.975.$$

Fig. 5. (A–C) Frequency spectrum of velocity oscillation for different Reynolds number and (D) time dependency of three velocity components.

With the increase of Re number from 250 to 5000, the performance evaluation criteria value of heat transfer gradually increases from 0.75 to 3.9, which satisfies the linear relationship in the double logarithmic coordinates as well.

3.3 Simulations Results and Discussion

Dealing with the experimental data about the velocity data and pressure drop of the first kind of structure for the cross-corrugated triangular ducts with 90° included angle, the values of the fully developed average friction factor (f_D) between numerical simulation and experiment is smaller than 10%. Similarly, the fully developed average Nusselt Numbers (Nu_D) has 12% between numerical simulation and experiment. The deviation may be attributed to entrance effect and thermal losses. Anyway, such deviation is acceptable in engineering. And the consistency of change of f_D or Nu_D proves that those parameters, solutions and models are appropriate, including low Re turbulence model number $k-\varepsilon$ turbulence model and grid meshing.

Select the same parameters as the above, and complete other numerical simulations for several included angles, and obtain some simulation data and processing results. When the included angle of cross-corrugated triangular ducts is 90° , the fully developed average (f_D) and the fully developed average Nusselt Numbers (Nu_D) is maximum, and then in that order, 120° , 45° and 60° (see in Fig. 6). When Re number is 1000, f_D of 45° is 20% smaller than that of 90° . In the meantime, Nu_D value of 120° , 60° and 45° is 30%, 45% and 75% smaller than that of 90° , relatively. What is more, as shown in Fig. 6, the j value is 14%, 47% and 76% smaller than that of 90° . And the PEC value is 24%, 52% and 73% smaller than that of 90° . These results make clear that the included angle has a great impact on the effect of heat transfer. When the included angle of cross-corrugated triangular ducts take the value of 90° , momentum transfer and heat transfer enhancement work best.

4 FIELD SYNERGY PRINCIPLE AND SYNERGY ANGLES

4.1 Field Synergy Principle and the Plot of the Local Synergy Angle

Guo (2004) studied the two-dimensional flat laminar boundary layer (parabolic flow) heat transfer and analyzed the energy equation of boundary layer flow. And the Field Synergy Principle

Fig. 6. Variation of f_D , Nu_D , j and PEC with Reynolds numbers for different included angles.

was founded (Guo, 2004). According to the principle, the performance of convection heat transfer not only depends on the fluid velocity and temperature fields, but also depends on the angle between velocity and temperature gradient, which is expressed as follows:

$$\beta = \arccos \frac{\mathbf{U} \cdot \nabla T}{|\mathbf{U}| |\nabla T|} = \arccos \frac{u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y}}{\sqrt{u^2 + v^2} \sqrt{\left(\frac{\partial T}{\partial x}\right)^2 + \left(\frac{\partial T}{\partial y}\right)^2}} \quad (12)$$

Based on the energy conservation equation, using the FLUENT simulation with the custom functions UDF, which is attached to the appendix, the cloud distribution of the local synergy angle (β_{Li}) is achieved.

Fig. 7 shows the cloud distribution of β_{Li} at the cross section where $x^* = 0, 0.5, 1, 1.5$, etc. Both the two figures have cyclical repeatability and fully be developed from the second cyclical unit.

Contours of β_{Li} in the ducts for different included angles, as shown in Fig. 7, provide the concrete images and perceptual intuitions how the included angles have a huge impact on the local synergy angle. In the upper part of the corrugated section, the blue-green areas with lower value of β_{Li} , which means synergy between velocity gradient, temperature gradient and pressure gradient in these areas is better than other areas, take a "Y" shape in the cross-corrugated triangular flow ducts with 45° and 60° included angles. In the meantime, the blue-green areas take "∧" shape for 90° included angles. And the areas take "∟" shape only on the windward side for 120° included angles. In the lower part of the triangle straight channel, the areas with good synergy are mainly around the region $z^* = 0.5$, and the sizes of these areas are biggest for 120° included angles, and then 90°, 60°, 45° in turn.

Fig. 7. Contours of local field synergy angle (β_{li}) in the ducts for different included angles.

4.2 Field-averaged Synergy Angle

Although the cloud distribution of the field synergy angle can accurately reflect the synergy

situation, the overall performance of the heat transfer coefficient is needed for the whole area in most cases. The field-averaged synergy angle (β_{VOL}) is proposed, which can be calculated by

$$\beta_{VOL} = \sum \frac{dV_i}{\sum dV_i} \cdot \beta_{Li} \quad (13)$$

where Li is a local synergy angle at any point.

Analyzing the numerical simulation data, the variation of the field-averaged synergy angle (β_{VOL}) of cross-corrugated triangular channel can be calculated with several Reynolds numbers for different included angles (Fig. 8).

When Reynolds number is 2000 for the cross-corrugated triangular channel with 90° included angle, the field-averaged synergy angle (β_{VOL}) is 74.1° , which is $1\text{--}2.3^\circ$ less than the value of β_{VOL} with other included angles. The smaller of β_{VOL} , the better of the collaboration of the speed and the temperature gradient. What's more, the result of β_{VOL} has the same law as the macroscopic heat transfer parameters, such as f_D , Nu_D , j , PEC . For different included angles, the value of β_{VOL} takes the minimums when the included angle takes 90° . Similarly, those macroscopic heat transfer parameters achieve optimum when the included angle is 90° .

4.3 Comparison of Synergy Angle for Different Flow Channels

In order to have a more comprehensive understanding about cross-corrugated triangular ducts, a comparison of the field-averaged synergy angle is done among the cross-corrugated triangular ducts (CCD) with 90° included angle, Circular tube, Parallel Flat Plates (PFP), Spirally Twisted Flats (STF), Spiral Straight Pipe (SSP). This comparison has been described before (Chen, 2012) and the results are shown in Fig. 9. It's evident that the β_{VOL} of CCD is minimal, and the followings are STF, SSP, Circular, PFP in sequence. And the β_{VOL} of CCD is 14.1° less than Circular tube and 14.5° less than parallel flat plates. The advantage of the field synergy for cross-corrugated triangular ducts is obvious.

5 EVALUATION OF ENERGY CONSERVATION

Due to the hot and humid climates in south China, energy for moisture control with an independent humidity control system still accounts for 27% of the total energy for air conditioning. Of which, 20–50% is used for latent load treatment. Zhang (2006) suggested that it is necessary to pay equal attention to eliminating the reheat energy consumption of fresh air and recovering the latent heat of exhaust air. Fig. 10 plots the primary energy requirements by dehumidification system for each person from January to December. The energy values vary from 1.5×10^5 kJ to 5.3×10^5 kJ per month per person. In the dry season (January and February), energy requirements are the least energy. Corresponding, in the most humid season (August), the required energy is the largest.

Fig. 8. Variation of β_{VOL} with Reynolds numbers for different included angles.

Fig. 9. Variation of β_{VOL} with Reynolds numbers for different channels.

Fig. 10. Percentages of primary energy consumed in each month in a year.

Modification of polyvinyl alcohol (PVA) by blending of montmorillonite (MMT) and anhydrous calcium chloride (CaCl_2), the moisture-permeable membrane is obtained, which was covered on a cross-corrugated triangular channel to prepare several total heat exchangers. This research has been studied before (Liang, 2014).

5.1 Calculation Examples of Air Conditioning System

The air conditioning system has been described before (Zhang, 2006). In the investigation, an office building with five occupants in a 20 m^2 room is considered. The set points for indoor air are as follows: temperature, 25°C ; relative humidity, 50%. Fresh air is supplied at $37.5 \text{ m}^3 \text{ h}^{-1}$, 20°C in temperature, and 7 g kg^{-1} in humidity ratio. The air dehumidification systems are operated when the offices are opened, i.e., from 9:00 to 18:00. At night, the systems are shut down, to save energy. The sensible load of the room is treated by chilled-ceiling panels. The total ventilation load includes two fractions: moisture load from fresh air and moisture load from human activities. The fresh air sensible and latent load is a variable relating to weather conditions, while the load from human activities is assumed to be kept at 50 g h^{-1} .

The system with no heat recovery is shown in Fig. 11 (left). It is the base system studied. Four independent air dehumidification systems with heat recovery measures are proposed. The operations of the four systems are controlled to satisfy the load and weather conditions. System 1 comprises a heat pump: the cooling coil acts as the evaporator, and the heating coil acts as a

Fig. 11. The base system (air dehumidification with no recovery) (left) and membrane core of total heat exchangers with the structure of cross-corrugated triangular ducts (right).

condenser. Fresh air (point A) first flows through a cooling coil where it is dehumidified below the dew point (point B). Then the air flows through a heating coil where it is heated to the set points of supply air. System 2 uses a sensible heat exchanger to recover the heat of the supply air itself after it flows through the dehumidification cooling coil. System 3 comprises a membrane-based total heat exchange. In this system, a membrane based total heat exchanger is used before the fresh air is pumped to a heat pump for air dehumidification. The total heat exchanger has a membrane core where the incoming fresh air exchanges moisture and temperature simultaneously with the exhaust air. In System 3, the total heat or enthalpy from the exhaust is recovered. As the improved version of System 3, System 4 change the membrane core into cross-corrugated triangular ducts (see in Fig. 11 right). To make comparisons, the annual energy requirements of the four proposed systems and a dehumidification system with no heat recovery measures as shown in Fig. 11 (left) are computed and listed in Table 1.

All the energy values are calculated on a per-person basis. In the analysis, the systems are used to treat the latent load solely, the sensible load of the room is around 50 W m^{-2} , which will be extracted by chilled-ceiling panels. Concerns on indoor air quality have prompted the research of novel air dehumidification techniques. This study proposed four systems in which mechanical dehumidification is combined with energy recovery measures like a heat pump, membrane enthalpy recovery, sensible heat exchanger. An hour-by-hour simulation reveals that 29–52% of primary energy can be saved by the independent air dehumidification with heat recovery, depending on the system involved. It is found that the mechanical dehumidification with a sensible heat exchanger consumes the largest energy in all systems proposed. By contrast, the one that consumes least energy is the mechanical dehumidification with a membrane enthalpy exchanger (System 4). The energy consumptions of all four systems are in the same order, because they use recovery measures.

Similar studies have been conducted (Zhong *et al.*, 2010). Zhong *et al.* (2010) chose four cities and test the difference between sensible heat exchangers and total heat exchangers or mixed use of two kinds of the heat exchanger. In hot and humid areas such as Shanghai and Guangzhou in summer, it is necessary to pay equal attention to eliminating the reheat energy consumption of fresh air and recovering the latent heat of exhaust air. It's a safe conclusion that the total heat

Table 1. Annual primary energy requirements ($10^3 \text{ kJ person}^{-1}$) for each person under various dehumidification strategies.

	Cooling	Heating	Electricity	Condensing	Auxiliary	Fan	Total
No recovery#	5500	2100	3700	6600	91	260	6200
System 1#	5500	1600	3700	6700	38	390	4100
System 2#	4200	1600	3800	5400	33	590	4400
System 3#	4400	580	3000	5400	45	590	3600
System 4	3800	430	3000	5400	45	590	3000

Note: The data of the row with # are referenced from the Zhang (2006).

Table 2. COP, electricity expense and payback period for different air conditioning system.

Combination mode	Beijing			Xi an			Shanghai			Guangzhou		
	COP	ΔMe	Y	COP	ΔMe	Y	COP	ΔMe	Y	COP	ΔMe	Y
Mode 1	9.6	0.01	185	10.7	0.01	102	9.0	0.038	37.6	10.3	0.086	16.9
Mode 2	31.1	0.13	17.6	30.2	0.13	18.0	49.7	0.20	11.9	49.7	0.45	5.2
Mode 3	30.8	0.23	12.5	32.0	0.25	11.6	29.0	0.25	11.4	30.0	0.57	5.1
Mode 4	47.5	0.32	11.9	47.5	0.33	11.5	66.6	0.41	9.2	68.5	0.95	4.0

Notes:

1. Combination mode of heat exchanger: Mode 1 contains a single sensible type; Mode 2 contains a single total type; Mode 3 contains two sensible types; Mode 4 contains one sensible and one total.
2. The unit of ΔMe is $\$ (m^3 h a)^{-1}$, which means annual electricity saving per unit flow of fresh air. And The unit of Y is the years of the payback period for the equipment input.

exchangers use the structure of Fig. 11 can improve COP value higher than 68.5 (see in Table 2). The equipment price is higher, even though the energy saving effect of total heat exchanger and its combination is remarkably better than sensible heat exchanger. The initial investment recovery time of different air heat exchanger combinations is analyzed and calculated to facilitate the selection of air heat exchanger.

$$Y = \frac{G \cdot N}{Me} \quad (14)$$

$$Me = \frac{0.15G\eta \cdot Q_t}{1000 \times 3600 \times COP} \quad (15)$$

In which Y represents the payback period and Me is the electricity fees for the whole cooling season, G means the fresh air volume required by the system. On the other hand, Q_t means the total energy consumption of fresh air, and η means the total energy-saving efficiency corresponding to various air heat exchangers and their combinations. The average COP of the cooling system in summer is 3.0, and $0.15 \$ (kW h)^{-1}$ for the electricity charge. N is the average initial investment price. The value of N is $1.45 \$ (m^3 h)^{-1}$ for sensible heat exchange, and meanwhile $2.35 \$ (m^3 h)^{-1}$ for total heat exchanger. And the service life of the air heat exchanger in European and American markets is considered to be about 20–30 years.

5.2 Calculation of Energy Saving and Emission Reduction in Canton

According to the statistical data of Guangdong Province in 2020, the annual power consumption of air-conditioning system (including household and office buildings) was about 187 billion kW·h. If the total heat exchanger of System 4 was used to replace the sensible heat exchanger system, 35%–45% energy could be saved. Of these, 45% energy efficiency was achieved in June to October and 35 per cent in the remaining seven months. According to the emission reduction data of National Bureau of Statistics of P.R.C, fossil-fuel power stations produced 5280 billion kW·h and reduced 798 million carbon emission in 2020. By that standard, the total heat exchanger with System 4 and cross-triangular corrugated flow duct in Guangdong Province can save 76.2 billion kW·h of electricity in the whole year. In another words, using this technology can bring 1.4×10^5 tons of carbon emission reduction, which equal to 12% of 113 million tons of carbon emission (the target of reducing carbon emissions in Guangdong). Therefore, the application of this single technology is of great significance to the National Energy Conservation and emission reduction, especially for the high temperature and humidity areas like Canton. The data cited above refer to Fig. 12.

6 PERSPECTIVES

It should be noted that till date theoretical research on analyzing the heat transfer processes

Fig. 12. Energy Savings and Carbon reduction (Using this technology) in Canton in 2020.

over a heat exchanger is not perfect. Based on a combined simulation and experimental approach, the results we presented here could provide a solution for the design and optimization of heat exchangers, which helped to increase energy efficiency and decrease CO₂ emission, as a result reducing the annual cost especially in winter. Continuous effort should be put for further optimization and innovation of the key equipment and facilities, which are expected to contribute to a low-carbon society.

7 CONCLUSIONS

The included angle of cross-corrugated triangular ducts has a great effect on the macroscopic heat transfer parameters, such as f_D , Nu_D , j , PEC . It's found that when the included angel takes 90°, the performance of enhanced heat transfer is the highest among four included angels. Local field synergy angle (β_{Li}) distribution inside ducts is shown by contours for different included angle and the field-averaged synergy angles (β_{VOL}) are calculated. A comparison of field-averaged synergy angles is done among the cross-corrugated triangular flow channel, Circular tube, Parallel Flat Plates, Spirally Twisted Flats, Spiral Straight Pipe. The result shows that the field synergy of cross-corrugated triangular ducts is the best.

In humid and hot areas in summer, the total heat exchanger or the combination of total heat and sensible heat exchanger should be used to pretreat the fresh air. And the total heat exchangers with membrane core, which consist of cross-corrugated triangular ducts can bring 1.4×10^5 tons of carbon emission reduction or 12% of the target in Canton, and the capital investment on equipment can get back as soon as four years.

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DISCLAIMER

The authors confirm that there are no conflicts of interest.

NOMENCLATURE

A	heat transfer area (m^2)
A_{ci}	cross-sectional area at inlet or outlet of a cycle (m^2)
A_{cyc}	surface area of the channel (m^2)
c_p	specific heat [J (kg K)^{-1}]
D_h	hydraulic diameter of the channel (m)
f	friction factor
f_D	fully developed local or cyclic mean friction factor
G	volume flow ($\text{m}^3 \text{m}^{-1}$ or $\text{m}^3 \text{h}^{-1}$)
h	static enthalpy (kJ kg^{-1}); or heat transfer coefficient [$\text{W (m}^2 \text{K)}^{-1}$]
j	heat transfer factor
L_{cyc}	length of a cycle in flow direction (m)
Nu	Nusselt Numbers
Nu_D	fully developed local or cyclic mean Nusselt numbers
p	pressure (Pa)
PEC	performance evaluation criteria value of heat transfer
Pr	Prandtl Numbers
Re	Reynolds numbers
t	time (s)
T	temperature (K)
u	flow velocity (m s^{-1})
V_{cyc}	volume of channel (m^3)
ΔT	logarithmic mean temperature difference (K)
x, y or z	coordinates (m)

Greek Symbols

α	degree of included angle ($^\circ$)
ρ	fluid density (kg m^{-3})
μ	molecular viscosity (Ns m^{-2})
λ	thermal conductivity [W (m K)^{-1}]
β	angle between velocity and temperature gradient ($^\circ$)

Superscripts

*	dimensionless
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Subscripts

b	bulk
i	inlet
Li	local value at i point
o	outlet
w	wall
VOL	volume or field averaged

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