https://doi.org/10.1016/j.apenergy.2014.05.024

An optimal design analysis method for heat recovery devices in building applications

X.P. Liu[1], J.L. Niu[1,*]

[1] Department of Building Services Engineering, The Hong Kong Polytechnic University, Hong Kong

[*]Corresponding author. Tel: (852) 2766 7781; fax: (852) 2774 6146.

E-mail address: bejlniu@polyu.edu.hk

Abstract

Air-to-air heat recovery system is widely used in building applications to reduce the energy used for conditioning the fresh air. The heat exchanger core geometry is one of the key factors that affect the overall performance of a heat recovery system. To better guide the development of high performance heat exchangers in building applications, a new analysis method is proposed in this work from the practical application point of view. The objective of the new optimization method is: at any given mass flow rate, temperature difference and desired heat recovery effectiveness, to minimize the material cost at a specified fan energy use, or alternatively, to minimize the fan energy use at a given material cost. Different duct geometries are analyzed together with the classical *j/f* factor method: equilateral triangle (Tri), circular (Cyl), square (Squ), rectangle with aspect ratio 1/2 (Rec(1/2)), 1/4 (Rec(1/4)), and 1/8 (Rec(1/8)). A novel channel structure named cross-corrugated triangular (CCT) duct is also considered for comparison. From the energy saving point of view, under the same hydraulic diameter, the pumping power requirements for Rec(1/8) are the lowest when compared with the other shapes in the laminar flow region,while the pumping power requirements for CCT duct are the highest, indicating larger energy consumptions when using such structure. Conversely, with a specified fan power consumption, the required total surface areas of Rec(1/8) are the smallest, which means that a parallel plate channel is the best geometry from the material saving point of view. By employing this method, the manufacturing and operating cost can be considered synthetically for achieving an optimal design. The proposed method can be used to select target-oriented high performance heat recovery core geometry for desired heat recovery performance, resulting in reduced space, weight, support structure, energy requirement and lifetime cost.

Keywords

Heat recovery, Building application, Optimal design, Performance analysis

1. Introduction

With rapid economic growth, there is a growing desire for better indoor built environment. Both thermal comfort and indoor air quality issues have gained increasing attention. Adequate ventilation is necessary to maintain a desired indoor air quality [1-2]. As a result, the energy consumption in buildings is increased due to the expansion of building sectors and the growth of ventilation system [3-5]. Energy saving technologies have attracted increasing attention due to global warming and environmental impact issue [6-9]. Air-to-air heat recovery systems employed in buildings could save a large fraction of the energy that is used for conditioning the fresh air, and theoretically reduce the energy consumption by a few significant percentage points [10]. Thus it comes into widespread use and has currently become a requirement in building designs [11].

Heat recovery systems include sensible heat recovery and enthalpy (sensible and latent) heat recovery. Different types of heat recovery systems, such as heat pipe [12-13], rotary wheel [14], run-around coil [15], and fixed plate [16], have been utilized to recover energy between the supply and exhaust airflow. A comprehensive review about the heat recovery technologies used in building applications has been made by Mardiana-Idayu and Riffat [17]. The efficiency and advantages of these technologies were summarized in that study. It is found that the works in the past had emphasized more on heat or mass transfer augmentation in heat exchanger. When the building standards are introduced, the performance of heat recovery becomes a concern [18-20].

The overall performance of a heat recovery system is affected by many factors that include the material property, operating condition, and heat exchanger core geometry. Various technologies have been developed to increase sensible and latent effectiveness of a stationary heat recovery heat exchanger, including material side intensification and air side intensification. From material side, membrane based heat recovery ventilators have attracted much attention due to its superior moisture-recovery effectiveness. Encouraging results are emerging with the introduction of new materials that can offer moisture recovery at the same time [21], and Nano materials breakthroughs are offering new opportunities [22]. From air side, the use of compact heat exchangers is an effective method to intensify heat mass transfer [23,24]. Related investigations have been conducted to reveal the effect of different surface geometries on the performance of heat recovery heat exchangers. In practice, many other factors affect the end performance of a heat exchanger, such as limited heat exchanger size, reduced temperature difference, and reduced fan power [25].

The basic performance data for a heat exchanger are often shown as *j* (Colburn factor) or *Nu* (Nusselt Number) and *f* (Fanning friction factor) vs. Re (Reynolds number) curves. Since Kays and London presented *j* and *f* vs. *Re* for a large number of compact surfaces in 1984 [26], this kind of plot curves have become a customary tool for presenting performance data for heat transfer surface geometries [27-34]. Yet, such evaluation method will only give a partial indication of performance. It is a common knowledge that heat transfer enhancement can be obtained but with a larger pressure drop penalty. Sometimes, the benefits gained from heat transfer enhancement are not great enough to offset the increased friction losses. Sparrow and Comb [35] evaluated the effect of channel height on the flow and heat transfer characteristics on a corrugated-wall heat exchanger. It was found that the increase of the channel height resulted in a substantial increase in the Nusselt number but the friction factor increased to a greater extent. This kind of enhancement may be acceptable when the designer's objective is focused on increase of heat transfer, but is not desired in heat recovery applications. Heat recovery systems used in building applications recover thermal energy, but use electrical energy for the fans, which may be more precious than the saved heat [36-37]. Other performance parameters were developed due to the failure of *j* and *f* curves to portray the relative performance. The ratio of the Colburn factor to friction factor (*j/f*) establishes a relation between friction and heat transfer coefficient, and is widely used for many comparative studies [28, 30, 38-41]. This factor is a useful parameter when comparing surfaces with different cross-sectional shapes. However, the influence of the scale of the geometry is not taken into consideration when using this parameter. Also, the performance evaluation is made through the comparisons of the corresponding *j/f* values under the same Reynolds numbers, which means the air flow rate may not remain constant for different surface geometries. Thus the significant effect introduced by the airflow rate is overlooked by using this parameter. The performance analysis on heat recovery heat exchangers should be conducted based on the same environmental and operating conditions. From an energy saving point of view, which is the primary purpose for using heat recovery system in building applications, the pressure drop, i.e. electricity consumption for fans, should be seriously considered. On the other hand, the designer may not always choose the best heat exchanger with minimal energy use, since he must consider the space available, and the required total surface area which is related to the material costs.

Generally there are two categories of methods for performance analysis on heat exchangers. One is the performance evaluation criteria (PEC) based on the first law of thermodynamics, which define the performance benefits of an exchanger having enhanced surfaces, relative to a standard exchanger with smooth surfaces subject to various design constraints [42-44]. Twelve criteria include three different types of constraints on the geometry of the heat exchanger are developed: Fixed geometry (FG), fixed flow area (FN), and variable geometry (VG) criteria. However, these criteria are segregated by corresponding constraints, which result in that the criteria cannot be widely used in comparative studies. For example, one of the FG criteria seek reduced fan power for fixed heat duty, but the cross-sectional flow area and tube length are required to be held constant. The other category for performance evaluation is based on the second law of thermodynamics, which is focused on entropy production and the destruction of useful energy (exergy) that results from heat transfer enhancement [45-46]. The aim of all second law criteria is to minimize the net exergy destruction or the net entropy generation resulting from the enhancement. Zimparov [47-48] further extended PEC equations in connection with the entropy generation and exergy destruction, and added new PEC for enhanced heat transfer surfaces when assessing the merits of augmentation techniques. It should be noted that by employing different performance analysis method, the results can be different. Chakraborty and Ray [49] presented an analysis study on combined first and second law based optimization of thermal-hydraulic performance of fully developed laminar flow through square ducts with rounded corners. It was found that when using the entropy generation minimization as a tool for thermal-hydraulic optimization, it may lead to contradictory results for some of the performance evaluation criteria (PEC). No general comment can be made with respect to the superiority of a particular geometry of the ducts, unless the geometric and thermal-hydraulic constraints plus the objective functions are carefully considered in view of the real application scenario.

To better guide the development of high performance heat exchanger in building applications, a straightforward analysis method for comparing the performance of different heat recovery heat exchangers is proposed in this work. As an initial demonstration of the method, comparisons are made between different heat exchanger core geometries, focused on the objective of minimizing the pressure drop and total surface area in terms of both energy saving and material cost reduction, which are particularly important for heat recovery applications.

2 Methodology

Based on the essential characteristics when employing heat recovery in building applications, the objective of the new optimization method is defined: at any given mass flow rate, temperature difference and desired heat recovery effectiveness, minimizing the material cost at a specified fan energy use, or alternatively, minimizing the fan energy use at a given material cost. The analysis process is presented in Figure 1.

Fig.1 The presented analysis process for the optimal design

The basic equations are introduced as follows. With regard to the material cost and the total surface area of the heat exchanger core considered, heat transferred by the air can be calculated by

$$\dot{Q} = \dot{m} c_p (T_{fo} - T_{fi}) \tag{1}$$

where $\dot{Q}$ is the heating power (kW), $\dot{m}$ is mass flow rate of air (kg s^{-1}), c_p is specific heat of air (kJ kg^{-1} K^{-1}). The heat transfer rate can also be calculated by

$$\dot{Q} = h A_t \Delta T_{\lg} \tag{2}$$

A_t is the total surface area of the exchanger (m^2), $\Delta T_{\lg}$ is the log mean temperature difference between the wall and fluid, which is

$$\Delta T_{\lg} = \frac{(T_{fi} - T_w) - (T_{fo} - T_w)}{\ln(T_{fi} - T_w)/(T_{fo} - T_w)} \tag{3}$$

where subscripts "*f, w, i, o*" refer to "air, wall, inlet and outlet" respectively. h is the convective heat transfer coefficient,

$$h = \frac{Nu\lambda}{D_h} \tag{4}$$

Nu is Nusselt number, λ is air conductivity (kW m^{-1} K^{-1}). The hydraulic diameter (m) of the channel is defined as

$$D_h = \frac{4V_t}{A_t} \tag{5}$$

where V_t is exchanger core volume (m^3).

From equation (1) , (2) and (4), the total surface area of the heat exchanger can be expressed as

$$A_t = \frac{\dot{m}c_p(T_{fo} - T_{fi})}{\Delta T_{\lg}} \frac{Nu\lambda}{D_h} \tag{6}$$

For a heat exchanger core with different surface geometries, Nusselt number can be identified under specified flow conditions. From this equation, with specified mass flow rate and temperature difference, the required total surface area of a heat exchanger can be calculated, and be varied with the hydraulic diameter.

With regard to the fan energy use, the required fan power is considered. The fan power is proportional to the exchanger pressure drop and is given by

$$P_{fan} = \frac{Q\ \Delta p}{\eta_{fan}} = \frac{\dot{m}\Delta p}{\rho\eta_{fan}} \tag{7}$$

where Q is the volume air flow rate (m^3 s^{-1}), Δp is the pressure drop (Pa), η_{fan} is the fan efficiency, and ρ is the density of air (kg m^{-3}).

The expression for exchanger pressure drop is

$$\Delta p = f \frac{4L_i}{D_h} \frac{G^2}{2\rho} \quad (8)$$

where f is the friction factor, L_i is the length of the duct (m), G is referred to as the mass velocity ($G = \rho u_m = \dot{m} / A_0$) (kg m^{-2} s^{-1}), A_0 is the minimum free flow area (m^2).

The Reynolds number, Re, is defined as

$$\mathrm{Re} = \frac{\rho u_m D_h}{\mu} \quad (9)$$

where u_m is the inlet velocity (m s^{-1}), μ is air dynamic viscosity (kg m^{-1} s^{-1}). Substituting the expressions of Equation 8 into Equation 7, the required fan power can be expressed as:

$$P_{fan} = \frac{2\mu}{\rho^2 \eta_{fan}} \frac{L_i}{D_h^2} \frac{\dot{m}^2}{A_0} \left(f \cdot \mathrm{Re} \right) \quad (10)$$

When considering various channel structures, for particular application scenarios, the mass flow rate and temperature difference can be identified. Under the same heat recovery effectiveness, the recovered heat is fixed. Using Equation 6 and the definition of hydraulic diameter, the length of heat exchanger core with various surface geometries can be obtained. Both the required fan power and total surface area can therefore be calculated and be varied with the hydraulic diameter. The following comparisons can be made directly to guide the optimal design: 1) under the same hydraulic diameter, the fan power requirements for different surface geometries; 2) with the same surface geometry, the size effect on fan power requirements; 3) under the same power consumption, the required total surface area or total volume for different duct shapes.

3. Main applications of the evaluation method

When the relationships of fan power (total surface area) and hydraulic diameter are established under the same recovered heat, how A_t (or D_h) varies with fan power can be investigated through the comparisons between the required A_t (or D_h) values at a specified fan power.

Considering various duct geometries, this analysis method can be employed in two different ways, depending on the proportion of fan power to recovered heat. If the fan power consumption takes an important part in the overall operating cost, which is also called lifetime costs, attention must be given to the fan power requirements for the system as a whole to be effective. However, the ratio between fan power and heat transfer rate can be significantly different under different scenarios. When $P_{fan}/\dot{Q}$ is insignificant, the total area or volume will be the only design factor. A lower total surface area means less material costs. The purpose of heat transfer enhancement is aimed at reducing the material use, or weight of the heat exchanger. On the other hand, when $P_{fan}/\dot{Q}$ cannot be neglected, considering the energy conversion efficiency (primary energy to electricity) and the fan efficiency, minimizing fan power consumption will be the prime objective. The desired heat transfer rate should be obtained by using a proper capacity fluid pumping/fan device, which in turn has a major impact on the operating cost. A practical air to air heat recovery system used in building applications and combined with different surface geometries are analyzed here to illustrate this optimization framework.

4 Results for air-to-air heat recovery units

Several typical-shaped duct geometries are analyzed here. From the point of view of either total surface area or fan power requirement, the comparisons are made in order to determine the possible optimum duct geometry which minimizes the material cost or energy use. With regard to the characteristics of using heat recovery in building application, a typical scenario is assumed. The assumed environmental and operating conditions are listed in Table 1. The duct geometries used are: equilateral triangle (Tri), circular (Cyl), square (Squ), rectangle with aspect ratio 1/2

(Rec(1/2)), 1/4 (Rec(1/4)), and 1/8 (Rec(1/8)). A novel channel structure named cross-corrugated triangular (CCT) duct developed by Zhang et al. [34] is also considered for comparisons. During the analysis, the hydraulic diameters for all the compared shapes are varied from 5mm to 20mm.

Tab.1 The assumed operating and environmental conditions.

For direct comparison purpose, all the required fan power and total surface area are normalized by the corresponding values from a circular tube heat exchanger with D_H = 5mm, since the circular tube and parallel plates are the commonly used geometries in heat transfer devices. The Nusselt numbers and friction factors for different ducts are shown in Table. 2.

Tab.2 Nusselt numbers and friction factors for flow in ducts with different cross section.

The variations of fan power and total surface areas along with different hydraulic diameters are presented in Figure 2, with constant heat flux boundary condition. The X-axis is the hydraulic diameter. The primary Y-axis on the left side is the normalized fan power, while the secondary Y-axis on the right side is the normalized total surface area requirement. The Reynolds numbers for all the geometries are smaller than 2300, which are in the laminar flow region. Firstly, the analysis results illustrate the fan power requirements for different channel structures, and the geometric effect from both size and shape points of view are revealed from the figure.

Fig.2 Fan power and total surface area variation under specified recovered heat

(Laminar flow region; Constant heat flux condition)

With regard to the shape effect, under the same hydraulic diameter, the fan power requirements for CCT duct are the highest in the comparison range, indicating larger energy consumption when using such structure. The fan power requirements for the rectangular channel with aspect ratio 1/8 are the lowest when compared with the other shapes. With regard to the size effect, it is shown that the fan power requirement is decreased with increasing hydraulic diameter under the same surface geometry. Secondly, the required total surface areas for different structures are revealed in this figure under the same recovered heat. With the same hydraulic diameter, the required total surface areas for CCT channel are always the smallest among the compared shapes, and the required total surface areas for triangular-shaped duct are the largest.

The energy consumptions should be taken into consideration when comparing the material cost. At any specified fan power (for example P_{nor}=0.05, as shown in the figure), the corresponding total surface areas for different duct geometries can be identified. Or conversely, at a specified total surface area, the corresponding fan power requirements can also be obtained. The line P_{nor}=0.05 intersects the fan power curves at different points. From these points, the corresponding hydraulic diameters can be identified. Then from these hydraulic diameters, the corresponding total surface area can be obtained, as shown in the secondary Y-axis. Such comparison reveals the related material cost under the same energy use when the operating and environmental conditions are the same. When $P_{fan}/\dot{Q}$ is insignificant, this is one of the key considerations for achieving an optimal design. It is shown under the same fan power, the rectangular shape with aspect ratio 1/8 need the smallest total surface area, which means the best geometry from material cost saving point of view. The required total surface area for the channel with triangular cross sectional geometry is the largest among the selected shapes.

Fig. 3 *j/f* values variation with Reynolds numbers

($500 \le \mathrm{Re} \le 2300$; Constant heat flux condition)

The results of performance analysis using *j/f* values for different channel structures, including the CCT channel, are presented in Figure 3 for comparison where the working fluid is air. The *j/f* values for the CCT channel are the smallest when $500 \le \mathrm{Re} \le 2300$. Since the Nusselt numbers of the CCT channel are greater than the other ducts within almost all the range (when $\mathrm{Re} \ge 800$), these results indicate that using the CCT channel can increase heat transfer, but the friction loss could be increased to a greater extent under the same Reynolds number. The *j/f* value for Tri, Cyl, Squ, Rec(1/2), Rec(1/4), and Rec(1/8) is constant in the laminar flow region. It can be seen that the influence of the scale of the geometry and the corresponding volume requirement can't be determined by using this j/ffactor. By employing the analysis method proposed in this study, the scale of the geometry can be taken into considerations, and the required total volume can be calculated and compared under the same operating and environmental conditions. Furthermore, the hydraulic diameter, which is not a practical design constraint, does not need to be fixed during the comparisons.

Fig. 4 Fan power and total volume variation under specified recovered heat

(Laminar flow region; Constant heat flux condition)

The required total volume under the same condition and the heat recovery effectiveness can also be calculated and is presented in Figure 4. The X-axis is the hydraulic diameter. The primary Y-axis on the left side is the normalized fan power, while the secondary Y-axis on the right side is the normalized total volume requirement. The space requirements for different channel structures are illustrated from this figure. The results reveal the space occupancy for different channel structures under the same conditions. With the same hydraulic diameter, the required total surface areas for the CCT channel are generally the smallest among the compared shapes. While the required total surface areas for triangular-shaped duct are the largest when compared with others. When considering the energy consumptions, at a specified fan power (P_{nor}=0.05), the corresponding total volumes requirements for different duct geometries can be identified. It is shown that the required volume for rectangular channel with aspect ratio 1/8 is the lowest of all the studied shapes.

Fig 5 Fan power and total surface area variation under specified recovered heat

(Laminar flow region; Constant wall temperature condition)

Furthermore, from Figure 2 and 4, when the material cost or the available space is fixed, the related fan power requirements for different channel structures can also be identified by this analysis method, regardless of any additional constraints.

Under constant wall temperature boundary condition, the corresponding results are presented in Figure 5 and Figure 6. The basic variation trends for fan power and surface area requirements are similar with that under constant heat flux condition. With the same hydraulic diameter, the required total surface areas for triangular channel are always the largest among the compared shapes. Considering energy consumptions, when P_{nor}=0.05, the required volume for Tri, Cyl, Squ are almost the same.

Fig 6 Fan power and total volume variation under specified recovered heat

(Laminar flow region; Constant wall temperature condition)

The comparisons can be extended to transitional and turbulent flow region. For transition or turbulent flow condition, the Nusselt numbers are varied with Reynolds numbers. Since the error of using Dittus-Boelter equation may be as large as 25%, a more accurate correlation provided by Gnielinski is used here ($3000 < Re < 5 \times 10^6$, including the transition region).

$$Nu = \frac{(f/8)(\mathrm{Re}_D - 1000)\,\mathrm{Pr}}{1 + 12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \qquad (11)$$

where the friction factor is obtained from the correlation developed by Petukhov:

$$f = (0.790 \ln \mathrm{Re}_D - 1.64)^{-2} \qquad (12)$$

The analysis results are shown in Figure 7. The required duct length for different channel structures is not the same within the comparisons range. Thus the comparisons among required total surface area under the same fan power are different from that under laminar flow condition.

Fig 7 Fan power and total surface area variation under specified recovered heat

(Transitional flow and turbulent flow region)

5. Conclusions

For heat recovery devices in building applications, when the operating and environmental conditions are known, the present study proposed a straightforward analysis method for optimization design. From both energy saving and reducing material costs (or space occupancy) points of view, the analysis method combines the required total surface area (or total volume) with fan power requirements under specified operating conditions. By employing this method, the material cost (or required space area) at a specified fan energy use can be minimized. Conversely, at a given material cost (or space area), the fan energy use can be minimized. The construction cost and the lifetime cost can be taken into consideration at the same time. As shown in the presented demonstration, from energy saving point of view, the fan power requirements for Rec(1/8) are the lowest when compared with the other shapes in the laminar flow region under the same hydraulic diameter. While the values for the CCT duct are the highest, indicating larger energy consumption when using such novel structure. Conversely, with a specified fan power consumption, the required total surface areas of Rec(1/8) are the smallest, which means the best geometry from material cost saving point of view. It should be noticed that the operating strategy with various ambient environment conditions of a heat recovery system have obvious influence on the optimization results. For example, different air flow rates and temperature differences lead to different Re numbers and heat transfer rates, which result in different total surface areas and fan power requirements. Thus the optimization results could be different. The proposed optimal analysis method need be employed under real scenarios with specified conditions. And it can be used to select target-oriented high performance heat recovery core geometry for desired heat recovery performance, resulting in reduced space, weight, support structure, energy requirement and lifetime cost. Further studies could be focused on developing a nonlinear optimization method that can determine the best design directly.

Acknowledgement

The Project is supported by NSFC-RGC joint scheme of Natural Science Foundation of China and Research Grant Council of Hong Kong under the project No. N_PolyU513/11.

References

[1] Calautit JK, Hughes BR, Chaudhry HN, Ghani SA. CFD analysis of a heat transfer device integrated wind tower system for hot and dry climate. Appl Energy 2013; 112: 576-591.

[2] Hughes BR, Chaudhry HN, Calautit JK. Passive energy recovery from natural ventilation air streams. Appl Energy 2014; 113: 127-140.

[3] Pérez-Lombard L, Ortiz J, Pout C. A review on buildings energy consumption information. Energ Buildings 2008; 40: 394–398.

[4] Chua KJ, Chou SK, Yang WM, Yan J. Achieving better energy-efficient air conditioning - A review of technologies and strategies. Appl Energy 2013; 104: 87-104.

[5] Zeng R, Wang X, Di H, Jiang F, Zhang Y. New concepts and approach for developing energy efficient buildings: Ideal specific heat for building internal thermal mass. Energ Buildings 2011; 43: 1081-1090.

[6] Pruitt KA, Braun RJ, Newman AM. Evaluating shortfalls in mixed-integer programming approaches for the optimal design and dispatch of distributed generation systems. Appl Energy 2013; 102: 386-398.

[7] Bianchi M, Ferrari C, Melino F, Peretto A. Feasibility study of a Thermo-Photo-Voltaic system for CHP application in residential buildings. Appl Energy 2012; 97: 704-713.

[8] Zhou D, Zhao CY, Tian Y. Review on thermal energy storage with phase change materials (PCMs) in building applications. Appl Energy 2012; 92: 593-605.

[9] Chesi A, Ferrara G, Ferrari L, Magnani S, Tarani F. Influence of the heat storage size on the plant performance in a Smart User case study. Appl Energy 2013; 112: 1454-1465.

[10] Zhang LZ, Niu JL. Energy requirements for conditioning fresh air and the long term savings with a membrane-based energy recovery ventilator in Hong Kong. Energy 2001;26(2): 119-135.

[11] Zhong K, Kang Y. Applicability of air-to-air heat recovery ventilators in China. Appl Therm Eng 2009; 29: 830–840.

[12] Riffat SB, Gan G. Determination of effectiveness of heat-pipe heat recovery for naturally-ventilated buildings. Appl Therm Eng 1998; 18: 121–130.

[13] Calautit JK, Chaudhry HN, Hughes BR, Ghani SA. Comparison between evaporative cooling and a heat pipe assisted thermal loop for a commercial wind tower in hot and dry climatic conditions. Appl Energy 2013; 101: 740-755.

[14] Zhang LZ, Niu JL. Performance comparisons of desiccant wheels for air dehumidification and enthalpy recovery. Appl Therm Eng 2002; 22: 1347–1367.

[15] Vali A, Simonson CJ, Besant RW, Mahmood G. Numerical model and effectiveness correlations for a run-around heat recovery system with combined counter and cross flow exchangers. Int J Heat Mass Transfer 2009; 52(25–26): 5827–5840.

[16] Zhang LZ, Jiang Y. Heat and mass transfer in a membrane-based energy recovery ventilator. J Membr Sci 1999; 163: 29–38.

[17] Mardiana-Idayu A, Riffat SB. Review on heat recovery technologies for building applications. Renew Sustain Energy Rev 2012; 16 :1241–1255.

[18] Fehrm M, Reiners W, Ungemach M. Exhaust air heat recovery in buildings. Int J Refrig 2002; 25: 439–449.

[19] Yun KT, Cho H, Luck R, Mago PJ. Modeling of reciprocating internal combustion engines for power generation and heat recovery. Appl Energy 2013; 102: 327-335.

[20] Chua KJ, Chou SK, Yang WM. Advances in heat pump systems: A review. Appl Energy 2010; 87 : 3611-3624.

[21] Zhang LZ. Coupled heat and mass transfer through asymmetric porous membranes with finger-like macrovoids structure. Int J Heat Mass Transfer 2009; 52: 751–759.

[22] Nair RR, Wu HA, Jayaram, PN, Grigorieva IV, Geim AK. Unimpeded permeation of water through helium-leak-tight graphene-based membranes. Science 2012; 335: 442–444.

[23] Stasiek JA. Experimental studies of heat transfer and fluid flow across corrugated-undulated heat exchanger surfaces. Int J Heat Mass Transfer 1998; 41: 899–914.

[24] Zhang LZ. Turbulent three-dimensional air flow and heat transfer in a crosscorrugated triangular duct. J Heat Transfer 2005; 127:1151–1158.

[25] Webb RL. Principles of Enhanced Heat Transfer. John Wiley & Sons, New York 1994.

[26] Kays WM, London AL. Compact Heat Exchangers, 3rd Edition. McGraw-Hill, New York 1984.

[27] Stasiek JA, Collins MW, Ciofalo M, Chew PE. Investigation of flow and heat transfer in corrugated passages—I. Experimental results. Int J Heat Mass Transfer 1996; 39 (1): 149–164.

[28] Wang CC, Hsieh YT, Lin YT. Performance of plate finned tube heat exchanger under dehumidifying conditions. J Heat Transfer 1997; 119: 109-117.

[29] Ozceyhan V, Gunes S, Buyukalaca O, Altuntop N. Heat transfer enhancement in a tube using circular cross sectional rings separated from wall. Appl Energy 2008; 85: 988–1001.

[30] Metwally HM, Manglik RM. Enhanced heat transfer due to curvature-induced lateral vortices in laminar flows in sinusoidal corrugated-plate channels. Int J Heat Mass Transfer 2004; 47 (10–11): 2283–2292.

[31] Zhang LZ. Numerical study of periodically fully developed flow and heat transfer in cross-corrugated triangular channels in transitional flow regime. Numer Heat Tr A-Appl 2005; 48: 387–405.

[32] Dong J, Chen J, Chen Z, Zhou Y, Zhang W. Heat transfer and pressure drop correlations for the wavy fin and flat tube heat exchangers. Appl Therm Eng 2007; 27 (11–12): 2066–2073.

[33] Ismail LS, Ranganayakulu C, Shah RK. Numerical study of flow patterns of compact plate-fin heat exchangers and generation of design data for offset and wavy fins. Int J Heat Mass Transfer 2009; 52 (17–18): 3972–3983.

[34] Zhang LZ, Chen ZY. Convective heat transfer in cross-corrugated triangular ducts under uniform heat flux boundary conditions. Int J Heat Mass Transfer 2011; 54: 597–605.

[35] Sparrow EM, Comb JW. Effect of interwall spacing and fluid flow inlet conditions on a corrugated-wall heat exchanger. Int J Heat Mass Transfer 1983; 26 (7): 993–1005.

[36] Roulet CA, Heidt FD, Foradini F, Pibiri MC. Real heat recovery with air handling units. Energ Buildings 2001; 33 (5): 495–502.

[37] Dodoo A, Gustavsson L, Sathre R. Primary energy implications of ventilation heat recovery in residential buildings. Energ Buildings 2011; 43(7): 1566–1572.

[38] Islamoglu Y, Parmaksizoglu C. The effect of channel height on the enhanced heat transfer characteristics in a corrugated heat exchanger channel. Appl Therm Eng 2003; 23 (8): 979–987.

[39] Zhang J, Kundu J, Manglik RM. Effect of fin waviness and spacing on the lateral vortex structure and laminar heat transfer in wavy-plate-fin cores. Int J Heat Mass Transfer 2004; 47(8–9): 1719–1730.

[40] Elshafei EAM, Awad MM, El-Negiry E, Ali AG. Heat transfer and pressure drop in corrugated channels. Energy 2010; 35 (1): 101–110.

[41] Wang Y, He YL, Mei DH, Tao WQ. Optimization design of slotted fin by numerical simulation coupled with genetic algorithm. Appl Energy 2011; 88 (12): 4441–4450.

[42] Webb RL, Eckert ERG. Application of rough surfaces to heat exchanger design. Int J Heat Mass Transfer 1972; 21: 1647–1658.

[43] Bergles AE, Bunn RL, Junkhan GH. Extended performance evaluation criterion for enhanced heat transfer surfaces. Lett Heat Mass Trans 1974; 1: 113–120.

[44] Webb RL. Performance evaluation criteria for use of enhanced heat transfer surfaces in heat exchanger design. Int J Heat Mass Transfer 1981; 24: 715–726.

[45] Bejan A. General criterion for rating heat exchanger performance. Int J Heat Mass Transfer 1978; 21: 655–658.

[46] Bejan A. Entropy generation minimization: the method of thermodynamic optimization of finite-size systems and finite-time processes. CRC Press; 1996.

[47] Zimparov V. Extended performance evaluation criteria for enhanced heat transfer surfaces: heat transfer through ducts with constant wall temperature. Int J Heat Mass Transfer 2000; 43(17): 3137–3155.

[48] Zimparov V. Extended performance evaluation criteria for enhanced heat transfer surfaces: heat transfer through ducts with constant heat flux. Int J Heat Mass Transfer 2001; 44 (1): 169–180.

[49] Chakraborty S, Ray S. Performance optimisation of laminar fully developed flow through square ducts with rounded corners. Int J Therm Sci 2011; 50(12): 2522–2535.