Experimental Study of the Effect of Non-Uniform Airflow Velocity Distribution on the Thermal Performance of a Water-Air Heat Exchanger

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Abstract. *The present work concerns an experimental study on the thermal performance of a fin-and-tube heat exchanger (HX) under the effect of a non-uniform airflow velocity distribution upstream of the HX. To this end, an experimental setup is designed and implemented. Its main objective is obtaining experimentally non-uniform velocity distributions upstream of a HX with prescribed mean and standard deviation. It is shown that HX thermal performance deteriorates due to the maldistributed upstream velocity. The obtained results agree with numerical results reported in literature.*

1. Introduction

HXs are mostly used in applications demanding heat transfer between two fluids [1-2] such as space heating, refrigeration, air conditioning, power plants [3], nuclear reactors, chemical plants, petrochemical plants, automotive industries, natural gas processing, and sewage treatment [4-8]. A common example of a HX can be found in a vehicle underhood where a circulating fluid, known as engine coolant, flows through the radiator coils with air flowing past the coils to cool the fluid.

Different brands of HXs are present and depend mainly on how the fluid flows and on the geometry. Fins-and-tubes, counter-flow channels and micro-channels, reactors, shell and tube coil, and multifunctional HXs [9-15] are some of the most common types of HXs. Fins-and-tubes HXs [16] consist of continuous parallel fins located among elliptical-cross-sectional tubes. HXs of the fins-and-tubes type are used in several engineering applications, especially automotive applications due to their high thermal efficiency, light weight and compactness.

Preceding studies [17-21] on fins-and-tubes HXs revealed that their thermal performance is mainly dependent on fin space, tube pitch, louver angle and the operating parameters in such as fluid flow rates and temperatures. Nonetheless, for a specified fins-and-tubes HX geometry and given working conditions, it was proved that the topology of the airflow has superior effect on HX performance compared to the tubular flow (flow to-be-cooled) [22-23]. In other terms, fins-and-tubes HXs subjected to two dissimilar upstream velocity layouts of equal flow rate exhibit different thermal performances. Actually, when incorporated in complex geometries (for example in the car underhood), a fins-and-tubes HX is constantly exposed to a maldistributed upstream flow velocity and temperature distributions. The velocity mal-distribution degrades HX thermal performance [24-26]. Nevertheless, little investigations concentrated explicitly on the relation between a HX thermal performance and the mal-distributions in the velocity and temperature fields. Khaled et al. proved numerically [27-29] that non-uniform flows through a fins-and-tubes HX might reduce its performance by up to 40%. Instead, the mal-distribution in the upstream temperature distribution can either enhance or reduce the performance of a fins-and-tubes HX by up to 5% depending on configuration.

This work presents an experimental study of the effect of airflow velocity maldistribution upstream of a HX on the thermal performance, with an ultimate aim of validating the numerical results of the literature. This study is made possible by an experimental setup that provides a real exchanger with hot water and allows setting and controlling a prescribed velocity distribution. The obtained experimental results confirm thermal performance degradation due to non-uniformity as reported in the numerical results of the literature.

The rest of this paper is organized as follows. Section 2 is devoted to an overview on numerical study of performance of HXs with velocity and temperature distribution non-uniformities. In section 3, the experimental setup will be exposed. Section 4 is then devoted to the tested configurations as well as the corresponding results. Finally, section 5 draws the main conclusions of the work.

2. Overview on Numerical Studies

There are few experimental studies that tackle the thermal performance of different types of HX with respect to the non-uniformities in airflow, water flow and temperature distributions of both flows. The existing studies are mainly numerical or analytical and lack experimental validations and comparison calculations/experiments. Since the objective of this study is to obtain experimentally a database on HX thermal performance exposed to maldistributed velocity and temperature profiles and to validate numerical results done before, the main numerical results obtained and how non-uniformities are modeled will be overviewed below.

2.1. Effect of Velocity Nonuniformity

In 2007, Beiler and Kroger [30] showed that airflow maldistribution degrades the performance of the HX. However, with optimized designs of these exchangers (air-cooled), the degradation can be minimized. T' Joen et al. [31] aimed to predict through simulation the effect of airflow non-uniformity on an exchanger with uniform liquid flow. It was shown that airflow non-uniformity reduces the exchanger's performance, in addition, the developed numerical tool allows for achieving HX designs with enhanced efficiencies. For turbulent flow, Mueller [32] found that certain types of HXs exhibit slight reductions in performance under the effect of airflow non-uniformity, the reduction is more severe with laminar flows.

In 2011, Mao et al. [33] investigated how the thermal performance and pressure drop are affected due to airflow non-uniformity in a louvered fin-and-tube HX. A maximum capacity reduction of 6% was obtained along with a 34% maximum pressure drop increment. Chin and Raghavan [34] proved that for a non-uniform airflow profile, the fin-tube HX thermal performance is influenced by statistical parameters of the profile such as the mean, standard deviation and the skew. The kurtosis however, has no effect. Kaern et al. [35] numerically investigated the performance of an evaporator subjected to airflow, feeder tube bending and liquid-vapor phase non-uniformities. It was observed that these maldistributions harm the cooling capacity and the coefficient of performance of the evaporator.

In 2013, Huang and Wang [36] provided optimal configurations of 3D U-type compact parallel flow heat exchanger to obtain uniformly distributed tube flow rates.

In 2014, Chu et al. [37] investigated numerically four designs of a high temperature HX for flow uniformity. It was concluded that a reduction of 52% in the maldistribution for the inlet manifold was achieved with helical baffles, which also provided a 24% increase in the Nusselt number. Yaici et al. [38] examined through 3D-CFD simulations the airflow non-uniformity impacts on the thermo-hydraulic performance of HXs. 50% enhancement or degradation in the Colburn j-factor was obtained in comparison to a HX with a uniformly distributed inlet air velocity.

2.2. Effect of Temperature Nonuniformity

Kou et al [39], in 1997, examined the influence of temperature non-uniformity on a two-hot-one-cold direct transfer HX thermal performance. It was noticed that the maldistribution in the temperature profile enhanced the performance.

In 1998, the thermal performance of HXs was shown by Kou et al. to be deteriorated by inlet temperature non-uniformity and longitudinal wall conduction [40].

In 1999, Ranganayakulu and Seetharamu [41] performed numerical calculations of the performance of a crossflow compact plate-fin HX and its degradation in response to the simultaneous influences of longitudinal wall conduction and velocity and temperature non-uniformities for numerous designs and working circumstances. Authors established that the performance deviations are relatively important in certain characteristic applications.

In 2002, Guo et al. [42] performed theoretical and experimental investigations of the effect of uniformity of the temperature difference field on the effectiveness of several types of HXs. Authors indicated that the maldistribution in the temperature difference reduces the HX effectiveness.

In 2008, Mishra et al. [43] conducted numerical calculations of transient temperature response of a crossflow HX to the temperature and flow mal-distributions for several conditions. It was observed that the responses depend on the individual temperature profile position for the temperature maldistribution, and on the fluid moving device's position for flow non-uniformity.

Passing through the above studies and others done by the authors of this paper [27-29], it can be noticed that numerical tools and methods continue to change and there has been a better understanding of the impact of airflow velocity and temperature maldistributions on HXs thermal performance. At the same time and despite the differences among these methods, they are all based on a common fundamental as to dividing the HX matrix into $n \times m$ cells and of applying the energy balance at each cell level.

For each cell, given values of air temperature and velocity are considered upstream of the cell in order to describe the air temperature and velocity non-uniformities over the entire surface of the exchanger. Output parameters calculated for each cell are then considered input parameters for the following cell.

The main concern of the various studies presented above is to provide an analysis on how airflow velocity and temperature non-uniformities affect the thermal performance of a HX. This study provides a procedure towards optimizing the cooling air arrangement of a certain tube-fin HX. To seek this objective, experimental studies are required to validate numerical tools and to provide answers when the numerical models are incapable. To proceed, a modular and flexible experimental setup should be conceived and implemented. The main features of this setup should be its ability to control air temperature and velocity distributions upstream of the exchanger in order to perform parametric studies. This setup will be exposed in the next section.

3. Experimental Setup

The experimental setup consists of a thermal part and an aerodynamic part. The thermal part of the experimental setup forms a closed loop, starting by the water heater and finishing by it. It is made up of the following components:

1- A 100-L water storage tank equipped with a gate valve and used to fill the electrical water heater for the first time. If there are leakages or losses in the system, it can be used to refill the system.

2- A 100-L electric water heater used for heating water up to 80 ^{0}C, with two gate valves to control the flow needed for the system.

3- Hot water return pump with a maximum head of 10 m, a flow rate of 18 L min-1 and a rotational speed of 2900 rpm. The pump is used to circulate hot water of the electric water heater in the HX in a closed thermal loop.

4- A 70 cm × 55 cm radiator with two thermostats (temperature sensors) positioned at the inlet and outlet of the exchanger.

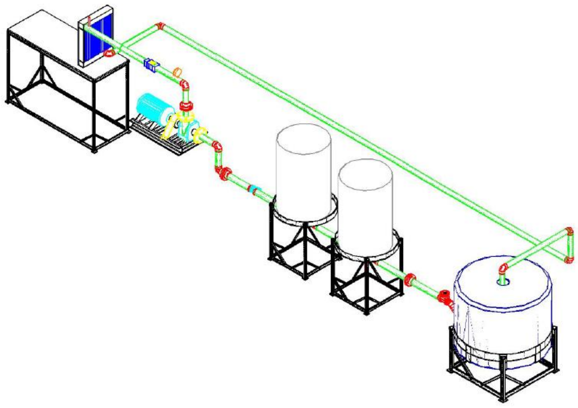

Fig. 1. Schematic of the thermal part.

The connections among the different components of the thermal part are shown in Fig. 1.

The aerodynamic part (Fig. 2) consists of:

1- A 70-cm duct, divided into 15 cells with area of 18.3 cm × 14 cm each, used for guiding airflow in each cell to have a constant velocity over the cell surface. It permits at the same time to prescribe different velocities in the cells to have a maldistributed velocity profile covering the whole surface of the HX. The reason for dividing the duct into several cells is that numerical computational codes with which the experimental results are to be compared are based on the same principle.

2- Fifteen fans with 100-W power, 12-V voltage and 13 m s-1 velocity for the given cell area and distance.

3- Five-speed regulators, each used to control the speed of three fans. The fans and their regulators allow creating and achieving prescribed velocities in the different cells.

4- Five 40-A/ 12-V batteries used to supply the fans through the regulators.

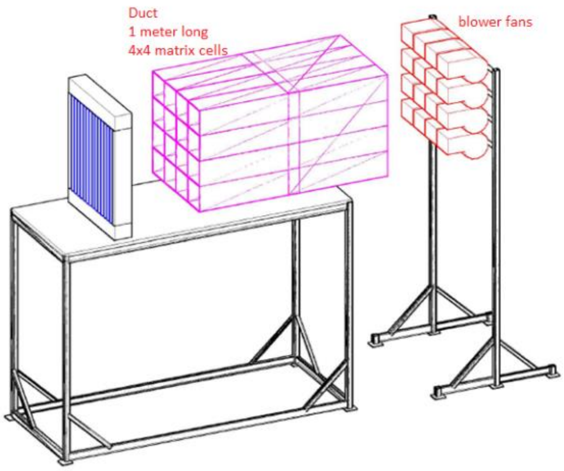

Fig. 21. Schematic of the aerodynamic part.

The two parts were finally assembled together to form the overall experimental setup shown in Fig. 3. The flexible and practical design of this setup allows it to be readily adapted to studying the non-uniformity of the upstream velocity and temperature distributions. The following section discusses different configurations of velocity distribution that can be achieved with the experimental setup as well as the corresponding impacts on the exchanger's thermal performance.

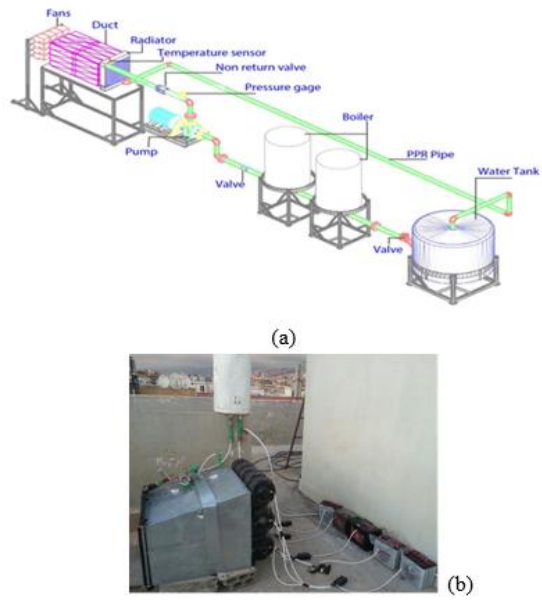

Fig. 3. a) Schematic of the experimental setup, b) the complete experimental setup.

4. Configurations and Results

Testing will be based on several values of water flow rate and mean air velocity. The water flow rate values are 200, 400, and 600 L/h and the mean air velocity values are 2 and 3 m/s. The testing protocol is as follows:

1- Take a fixed value for Q water flow rate;
2- Take a fixed mean air velocity value;
3- Take results for different configurations of standard deviations for the same mean velocity values;
4- For each configuration read the inlet and outlet temperatures.

A total of 36 configurations are obtained with the modular experimental setup and tested. These 36 configurations correspond to six configurations of velocity profiles entering (upstream) the exchanger for each mean air velocity (2 and 3 m/s) and for each of the three water flow rates of 200, 400, and 600 L/h. The configurations of velocity distributions are shown in Fig. 4 and Fig. 5.

Fig. 6 shows the water outlet temperature and the thermal performance of the HX with respect to the standard deviation of the upstream velocity distribution for several water flow rates and a mean velocity of 2 m/s.

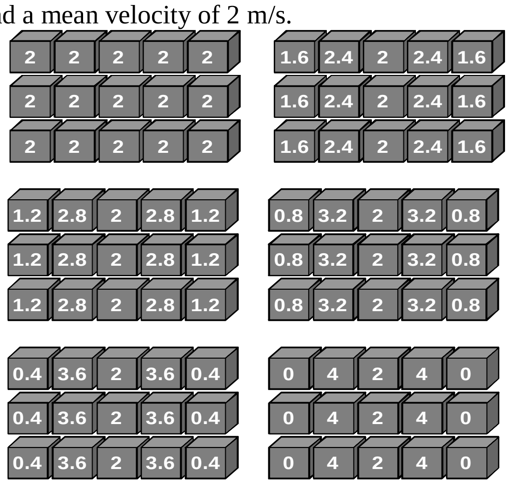

Fig. 42. *Configurations of velocity distribution for a mean value of 2 m/s.*

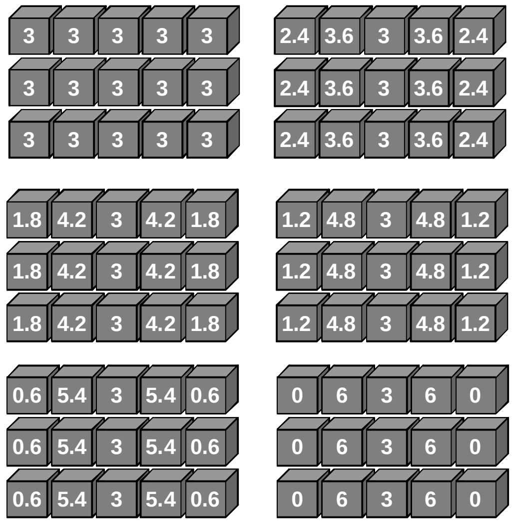

Fig.5. Configurations of velocity distribution for a mean value of 3 m/s.

It is noticed that the upstream velocity maldistribution raises the water temperature outlet of the HX. For example, for a water flow rate of 200 L/h, increasing the standard deviation from 0 to 2 m/s causes the outlet temperature to increase from 53.8 °C to 58.2 °C. The HX thermal performance (Fig 6-b) drops from 3.78 kW to 2.75 kW when the standard deviation is increased from 0 to 2 m/s, a 27% reduction in comparison to a uniformly distributed upstream velocity. These magnitude orders are close to that obtained numerically by Khaled et al. [27]. For the water flow rates of 400 and 600 L/h, the thermal performance of the exchanger decreases from 7.56 to 5.65 kW and from 10.99 to 8.26 kW respectively. The same trends are noticed for a mean velocity of 3 m/s.

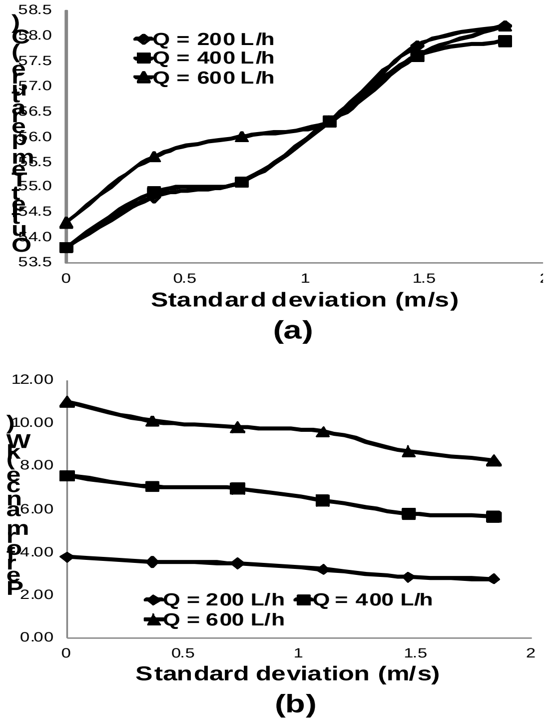

Fig. 63. Variation of a) outlet temperature, b) thermal performance in function of the standard deviation for different water flow rates.

5. Conclusions

This work concerned an experimental study of HX thermal performance in relation with the non-uniformity of the upstream velocity distribution. A prototype with its corresponding setup was designed and implemented. The prototype consists of two parts: thermal and aerodynamic. The aerodynamic part permits to fix and control a prescribed velocity distribution upstream a real HX. The thermal part is a closed loop permitting to provide a real exchanger with hot water. This part permits to maneuver the thermal performance of the HX for different velocity distributions.

It was concluded that the upstream velocity maldistribution degrades the thermal performance of the HX. For example, for a water flow rate of 200 L h-1 and a mean velocity of 2 m s-1, increasing the standard deviation from 0 to 2 m s-1 decreases the thermal performance of the exchanger from 3.78 kW to 2.75 kW, a 27% decrease compared to a uniform upstream velocity distribution. These tendencies are coherent with the numerical results reported in the literature.

References

[1] M. S. Krishnakanth, C. Aravind and A. Ashok, Numerical Investigation of Concentric Tube Heat Exchanger using a Nanofluid, International Journal of Engineering Research & Technology (IJERT), 8 (2019) 1339-1344

[2] N. Azari, I. Chhaya, Y. Ghamat, D. Kanthariya, M. Patel and F. Bodiwala, Experimental Investigation of Heat Transfer in Compact Heat Exchanger using Water-Ethylene Glycol, International Journal of Engineering Research & Technology (IJERT), 8 (2019) 666-669

[3] Aneeq, Z.G Fu and H. I. Khan, A Comprehensive Study on Integration Technique on Air Separation Unit and Gas Turbine for IGCC Power Plant, International Journal of Engineering Research & Technology (IJERT), 8 (2019) 1493-1499

[4] J.S. Mate, Experimental and CFD analysis of heat exchangers using different orifice baffles design, International Journal of Engineering Research and Technology, 7 (2018) 77-84

[5] B. Raja and R. L. Jhala, Optimization of shell and tube heat exchangers using teaching-learning based optimization algorithm, International Journal of Engineering Research and Technology, (2016) 1-5

[6] Q. Zhang, S. Qin, and R. Ma, Simulation and experimental investigation of the wavy fin-and-tube intercooler, Case Studies in Thermal Engineering, 8 (2016) 32-40.

[7] S. Tamna, Y. Kaewkohkiat, S. Skullong, and P. Promvonge, Heat transfer enhancement in tubular heat exchanger with double V-ribbed twisted-tapes, Case Studies in Thermal Engineering, 7 (2016) 14-24.

[8] Mahmoud Khaled, Mohamad Ramadan, Heating fresh air by hot exhaust air of HVAC systems, Case Studies in Thermal Engineering, 8 (2016) 398-402

[9] K. Amarnath, A. Kishore, A. Kathaperumal, K. Karthickraja and K. Karthikeyan, Design and Fabrication of Bank of Tubes Counter Flow Heat Exchanger, International Journal of Engineering Research & Technology (IJERT), 7 (2019) 1-8

[10] D. Bhanuchandrarao, M. A. chakravarthy, Y. Krishna, V .V. Subba Rao and T. Hari Krishna, CFD Analysis And Performance Of Parallel And Counter Flow In Concentric Tube Heat Exchangers, International Journal of Engineering Research & Technology (IJERT), 11 (2013) 2782-2792

[11] A. Sivakumar, K. Thamizhmaran, G. Kadiravan, B. Porselvan and G.Gunaseelan, Analysis of Microchannel Heat Exchanger for Electronic

Equipments Cooling using SiO2/Water Nanofluid, International Journal of Engineering Research & Technology (IJERT), 3 (2015) 1-3

[12] A. Raj Singh Suri, A. Kumar, and R. Maithani, Effect of square wings in multiple square perforated twisted tapes on fluid flow and heat transfer of heat exchanger tube, Case Studies in Thermal Engineering, 10 (2017) 28-43.

[13] M. Izzedine, S. Adjibade, A. Thiam, C. Awanto, and D. Azilinon, Experimental analysis of diffusion absorption refrigerator driven by electrical heater and engine exhaust gas, Case Studies in Thermal Engineering, 10 (2017) 255-261.

[14] A.M. Hussein, H.K. Dawood, R.A. Bakara, and K. Kadirgamaa, Numerical study on turbulent forced convective heat transfer using nanofluids TiO2 in an automotive cooling system, Case Studies in Thermal Engineering, 9 (2017) 72-78

[15] A. K. Prasad and K. Anand, Design & Analysis of Shell & Tube Type Heat Exchanger, International Journal of Engineering Research & Technology (IJERT), 9 (2020) 524-539

[16] I. J. Dhangar and M. Chopra, Experimental Investigation and CFD Analysis Performance of Fin and Tube Heat Exchanger with Different Types of Fins, International Journal of Engineering Research & Technology (IJERT), 5 (2017) 1-6

[17] M. Ramadan, M. Gad El Rab, and M. Khaled, Parametric analysis of air–water heat recovery concept applied to HVAC systems: Effect of mass flow rates, Case Studies in Thermal Engineering, 6 (2015) 61-68.

[18] M. Ramadan, S. Ali, H. Bazzi, and M. Khaled, New hybrid system combining TEG, condenser hot air and exhaust airflow of all-air HVAC systems, Case Studies in Thermal Engineering, 10 (2017) 154-160.

[19] P. Blecich, Experimental investigation of the effects of airflow nonuniformity on performance of a fin-and-tube heat exchanger, International Journal of Refrigeration, 59 (2015) 65-74.

[20] J. Fernandez-Seara, A. A. Pardinas, and R. Diz, Heat transfer enhancement of ammonia pool boiling with an integral-fin tube, International Journal of Refrigeration, 69 (2016) 175-185.

[21] M.A. Abdous, H. Saffari, H. B. Avval, and M. Khoshzat, The study of entropy generation during flow boiling in a micro-fin tube, International Journal of Refrigeration, 68 (2016) 76-93.

[22] A. Waschle, Numerical methods, Three-Dimensional, in: J. Wiedemann, W.H. Hucho, Progress in Vehicle Aerodynamics II Thermo-Management, Expertverlag, Stuttgart, 2002, pp. 115-128.

[23] L.Z. Zhang, Flow maldistribution and thermal performance deterioration in a cross-flow air to air heat exchanger with plate-fin cores, International Journal of Heat and Mass Transfer 52 (2009) 4500-4509.

[24] K. Shaji, S.K. Das, The effect of flow maldistribution on the evaluation of axial dispersion and thermal performance during the single-blow testing of plate heat exchangers, International Journal of Heat and Mass Transfer 53 (2010) 1591-1602.

[25] N. Srihari, B.P. Rao, B. Sunden, S.K. Das, Transient response of plate heat exchangers considering effect of flow maldistribution, International Journal of Heat and Mass Transfer 48 (2005) 3231-3243.

[26] L. Cabezas-Gòmez, H.A. Navarro, S.M. De Godoy, A. Campo, J.M. Saiz-Jabardo, Thermal characterization of a cross-flow heat exchanger with a new flow arrangement, International Journal of Thermal Sciences 48 (2009) 2165-2170.

[27] M. Khaled, H. El Hage, A. El Marakbi, F. Harambat, and H. Peerhossaini, Aerothermal management of vehicle heat exchangers – Parametric analysis, Proc. ASME Fluids Engineering Division Summer Conference, Nevada, USA, 2013.

[28] M. Khaled, F. Harambat, and H. Peerhossaini, Analytical and empirical determination of thermal performance of louvered heat exchanger – effects of air flow statistics, International Journal of Heat and Mass Transfer, 54 (2011) 356-365.

[29] M. Khaled, M. Ramadan, A. Shaito, H. El Hage, F. Harambat, and H. Peerhossaini, Parametric analysis of heat exchanger thermal performance in complex geometries – Effect of velocity and temperature distributions, Heat Transfer Engineering Journal, 37 (2016) 1027-1037.

[30] M.G. Beiler and D.G.Kroger, Thermal performance reduction in air-cooled heat exchangers due to non-uniform flow and temperature distributions, Heat Transfer Engineering, 17 (2007) 82-92.

[31] T'Joen, C., Willockx, A., Steeman, H.J., and De Paepe, M., Performance prediction of compact fin-and-tube heat exchangers in maldistributed airflow, Heat Transfer Engineering, 28 (2007) 986-996.

[32] Mueller, A.C., Effects of some types of maldistribution on the performance of heat exchangers, Heat Transfer Engineering, 8 (2007) 75-86.

[33] J.N. Mao, H.X. Chen, H. Jia, Y.Z. Wang, and H.M. Hu, Effect of air-side flow maldistribution on thermal–hydraulic performance of the multi-louvered fin and tube heat exchanger, International Journal of Thermal Sciences, 73 (2013) 46-57.

[34] W. M. Chin and V. R. Raghavan, The influence of the moments of probability density function for flow maldistribution on the thermal performance of a fin-tube heat exchanger, International Journal of Thermal Sciences, 50 (2011) 1942-1953.

[35] M.R. Kærn, W. Brix, B. Elmegaard, and L.F.S. Larsen, Performance of residential air-conditioning systems with flow maldistribution in fin-and-tube evaporators, International Journal of Refrigeration, 34 (2011) 696-706.

[36] C.H. Huang and C.H. Wang, The study on the improvement of system uniformity flow rate for U-type compact heat exchangers, International Journal of Heat and Mass Transfer, 63 (2013) 1-8.

[37] W.X. Chu, T. Ma, M. Zeng, T. Qu, L.B. Wang, and Q.W. Wang, Improvements on maldistribution of a high temperature multi-channel compact heat exchanger by different inlet baffles, Energy, 75 (2014) 104-115.

[38] W. Yaïci, M. Ghorab, and E. Entchev, 3D CFD analysis of the effect of inlet air flow maldistribution on the fluid flow and heat transfer performances of plate-fin-and-tube laminar heat exchangers, International Journal of Heat and Mass Transfer, 74 (2014) 490-500.

[39] H.S. Kou and P. Yuan, Thermal performance of cross flow heat exchanger with nonuniform inlet temperatures, International Communications in Heat and Mass Transfer, 24 (1997) 357–370.

[40] H.S. Kou and P. Yuan, The effect of longitudinal wall conduction on the cross flow heat exchanger with nonuniform inlet temperatures, Heat Transfer Engineering, 19 (1998) 54-63.

[41] Ch. Ranganayakulu and K.N. Seetharamu, The combined effects of longitudinal heat conduction, flow nonuniformity and temperature nonuniformity in crossflow plate-fin heat exchangers, International Communications in Heat and Mass Transfer, 26 (1999) 669-678.

[42] Z.Y. Guo, S.Q. Zhou, Z.X. Li, and L.G. Chen, Theoretical analysis and experimental confirmation of the uniformity principle of temperature difference field in heat exchanger, International Journal of Heat and Mass Transfer, 45 (2002) 2119-2127.

[43] M. Mishra, P.K. Das, S. Sarangi, Effect of temperature and flow nonuniformity on transient behaviour of crossflow heat exchanger, International Journal of Heat and Mass Transfer, 51 (2008) 2583-2592.