

# DEVELOPMENT OF AN EXPERIMENTAL METHOD TO GENERATE A NON-UNIFORM AIR FLOW DISTRIBUTION AT THE INLET OF A HEAT EXCHANGER

PISANO, Alessandro; MARTÍNEZ BALLESTER, Santiago; CORBERÁN, José Miguel

apisano@iie.upv.es (A. Pisano).

Universitat Politècnica de València, Institute for Energy Engineering, Camino de Vera, s/n, Valencia, 46022, Spain

## SUMMARY

As already demonstrated by many authors, the non-uniformity of the air distribution at the inlet of both condensers and evaporators entails not negligible performance degradation. Its experimental quantification is mainly complicated by the difficulties connected to the air velocity measurement in a traditional air-conditioning installation. For this reason, many works are supported by the application of CFD models or others simulation tools. In order to analyze experimentally the effect of the non-uniform air distribution on the performance of a condenser, in this work a simple method for generating a specific air velocity profile at the inlet of a heat exchanger will be presented. It consist in allocating several filters along the heat exchanger height. Once characterized the filter, identifying the relation between the length of the filters, the pressure drop and the air velocity, any type of air velocity profile can be generated. The method presents high flexibility and allows being used with any typology of heat exchanger. In this work, the method will be used for generating a non-uniform air velocity profile at the inlet of two condenser: a round tube plate fins heat exchanger and a microchannel heat exchanger. The results show that, with a specific distribution of the filters, the non-uniform air velocity profile characterizing the A-shape condensation units has been reproduced with a very good agreement.

**Keywords:** airflow malditribution, heat exchanger, experimental method

## 1. Introduction

The performance of a refrigeration system, usually expressed in term of thermal capacity and coefficient of performance (COP), are strongly connected to the working conditions of all its components. As already demonstrated by other authors, heat pumps and air conditioning systems can undergo a substantial performance reduction if the evaporator or the condenser are characterized by non-uniform air distribution.

Due to the difficulties connected to the air-velocity measurement, most of the studies concerning the effect of the air maldistribution on the thermal capacity degradation of evaporators and condensers have been carried out by using specific models and software. Lee and Domanski [1] studied the effects of air and refrigerant maldistribution on the performance of a round-tube condenser by means of the software EVAP5M. Yaïci *et al.* [2] proposed an optimization study by using a three-dimensional CFD simulation software. Similarly, Bach *et al.* [3] modified the configuration of a two-circuits evaporator in order to mitigate the effects of the airflow maldistribution on its performance. They implemented the evaporator model in ACHP [4] taking into account the non-uniformity of the air distribution.

Mao *et al.* [5] studied the performance degradation of a plate-tube condenser. Comparing the effects of different air velocity profiles, they evaluated a maximum performance degradation of about 6% using a semi-empirical model based on a Finite Volume Method (FMV).

Some of technics resumed by Mc Williams [6] have been used for evaluating experimentally the air velocity at inlet of a heat exchanger. Yashar *et al.* [7] measured the air velocity profile for A-shape evaporation unit using Particle Image Velocimetry. Domanski *et al.* [8] used their results for optimizing the evaporator circuitry. They combined the commercial software EVAP-COND and ISHED (Intelligent System for Heat Exchanger Design).

Datta *et al.* [9] studied experimentally the performance degradation due to the partial blockage of the airflow through the compact parallel tube condenser. Considering different type of blockage of the air, they measured the air velocity trough a hot-wire anemometer. The results show how the performance degradation is due to the combined effects of the airflow reduction and air maldistribution.

This work is not directly focused on the analysis of the performance degradation connected to the air flow maldistribution, but it is aimed to present a methodology to generate a specific air velocity profile at the inlet of any type of heat exchanger. The method consists in allocating several filters with different length along the heat exchanger height by using a suitable support structure. A process of characterization, carried out in an experimental facility, allows determining the length of each filter as function of the velocity profile desired. With the method proposed, the air velocity profile characterizing a A-shape condensation units has been generated at the inlet of two different heat exchangers: a round tube plate fins heat exchanger and a plate tube heat exchanger. The results show very good agreement between the calculated air velocity profile and the measured one.

## 2. Experimental work

The experimental work has been carried out in several steps. The objective of the first part of the work was to reproduce in an experimental facility the phenomena occurring in the air side of a traditional air-to-water heat pump. With this end, in a long duct equipped with an axial fan, the air velocity at the inlet of a microchannel exchange was measured by means of a hot-wire anemometer (Fig. 1). Initially, in order to generate a non-uniform velocity profile, filters with different density and thickness were positioned along the inlet surface of the heat exchanger without using any support structure. Due to the poor quality of the results of the measurements, negatively influenced by the presence of high turbulent flow inside the duct, the original installation was modified: the duct was enlarged and a honeycomb air straightener was added (Fig. 2). The second step of the experimental work was mainly focused on the process of characterization of the filter. As will be dealt lather, the process consists in evaluating quantitatively the relation between the length of the filters, the pressure drop and the air velocity profile upstream the heat exchanger. Hence, once defined a theoretical velocity profile, the

length and the position of the filters along the inlet surface of the heat exchanger are univocally determined.

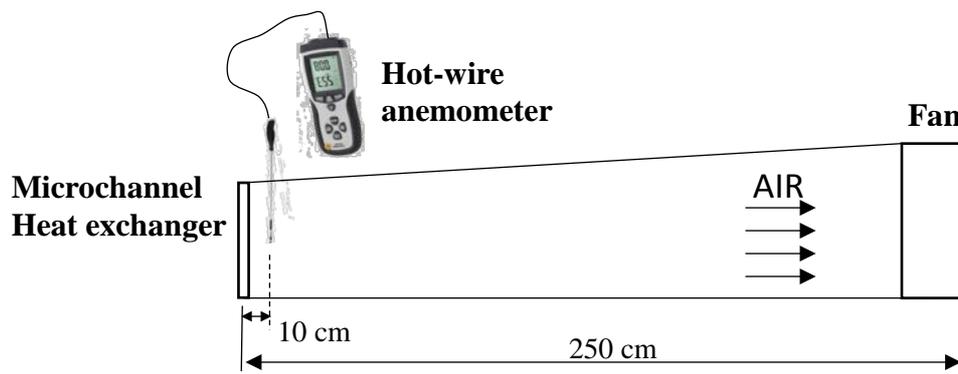


Figure 1: First installation for measuring the air flow through a heat exchanger.

Nevertheless, the effects of the turbulence were only partially nullified. Therefore, the method for measuring the air velocity changed substantially. The hot-wire anemometer, which previously was inside the duct (Fig. 2) downstream the heat exchanger, was inserted inside an additional support structure. In Fig. 5 is shown the position of the structure, slabs and filters respect to the heat exchanger. During the tests, the probe was inserted between two consecutive slabs through a perforated transparent window. The window allowed checking the position of both filters and probe inside the structure.

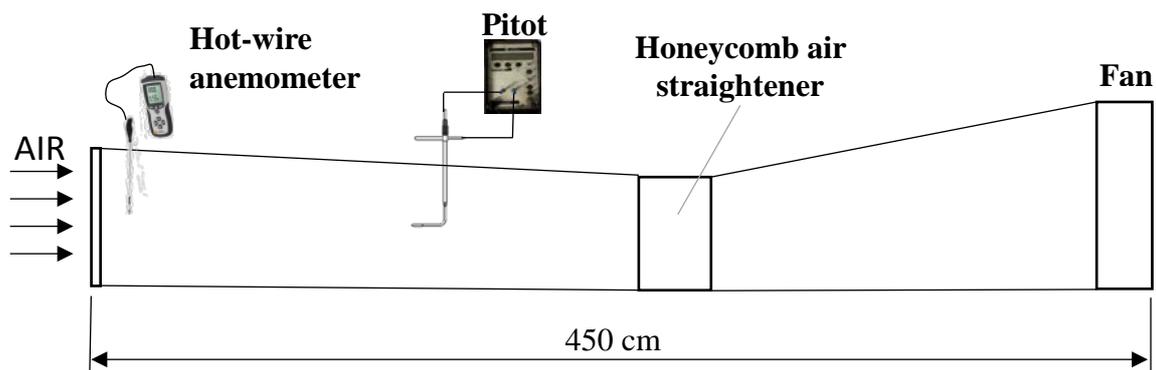


Figure 2: Modified installation for measuring the air flow through a heat exchanger.

Nullify totally the effect of the turbulence, this solution allowed obtaining very good agreement between the air velocity profile required and the measured one. In the last part of the work, two different new support structures were designed according to the geometric characteristics of a round tube plate fins heat exchanger and a microchannel heat exchanger. Both the heat exchangers have been tested in an experimental air conditioning system. The results of the air velocity measurement will be presented in this paper, while the effects of the air maldistribution on the performance of the heat exchanger will be presented and discussed in another work.

### 3. Air duct: design and first results

In Fig. 1 is depicted the scheme of the experimental facility for measuring the air velocity at the inlet of the heat exchanger. The air is moved inside a 2.5 m long duct by means of an axial fan. At the other extremity of the duct, a microchannel heat exchanger was positioned. The unit is characterized by a core length equal to 471 mm, core width of 334 mm and a core depth equal to 19 mm. The fins are louvered and the tube thickness is 2 mm. In the upper panel of the duct, connected to the heat exchanger, a transparent window allowed positioning the probe for measuring the air velocity. The probe was 10 cm far from the heat exchanger during the measurements. The first tests were carried out

according to the log-T method [10]. The original log-T method was slightly modified increasing the number of point where the air velocity is measured.

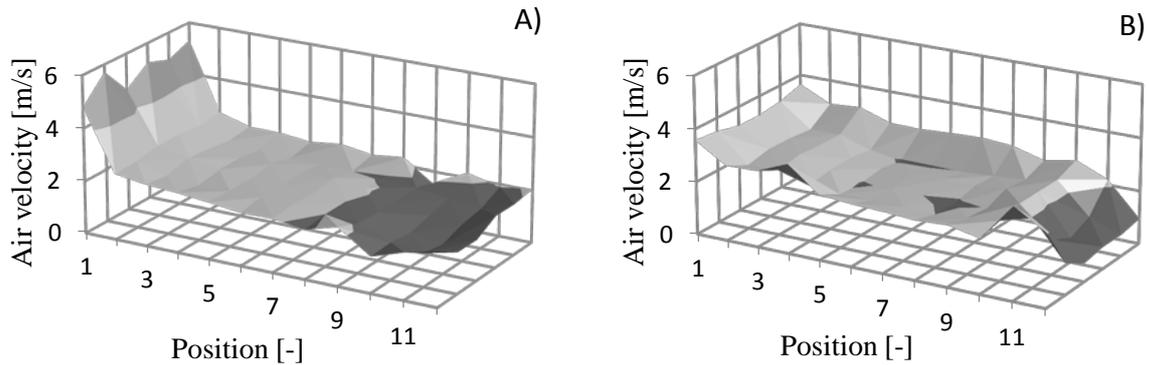


Figure 3: Air velocity profile at the inlet of the microchannel heat exchanger.

When the filters occupied the heat exchanger inlet surface, a high turbulent flow affected negatively the measurement with the hot-wire anemometer. In Fig. 3 is represented the experimental data obtained considering two different configurations of the filters. In the configuration A the upper part of the heat exchanger was uncovered, while the lower part was plugged with two filters with different density. In the case B, only one type of filter was positioned along the heat exchanger height. Differently from the previous case, the length of the filters decreased from the bottom to the top of the heat exchanger. In both cases, the air velocity surface was quite corrugated due to the turbulent flows and a not clear relation between filters length and air velocity could be defined. In order to found this relation, the original installation was modified as depicted in Fig.2.

#### 4. Characterization of the filter

The process of characterization of the filter was performed as follows. By changing the rotational speed of the fan (six operating points), the pressure drop due to the heat exchanger ( $\Delta P_{HEX}$ ) was measured by means of the Pitot tube. Once established the thickness ( $L_{filter}$ ), a layer of filter was positioned forward the whole heat exchanger inlet surface (Fig.4) and the previous tests were repeated. In this case, the total pressure drop ( $\Delta P_{total}$ ) and the average value of the air velocity ( $V$ ) were measured. In Fig.4 is depicted the experimental trend of pressure drop generated exclusively by the filter ( $\Delta P_{filter}$ ). Its value has been evaluated as follows:

$$\Delta P_{filter} = \Delta P_{total} - \Delta P_{HEX} = f \cdot \frac{L_{filter}}{D} \cdot \rho \frac{V^2}{2} = K_{filter} \cdot L_{filter} \cdot V^2 \quad (1)$$

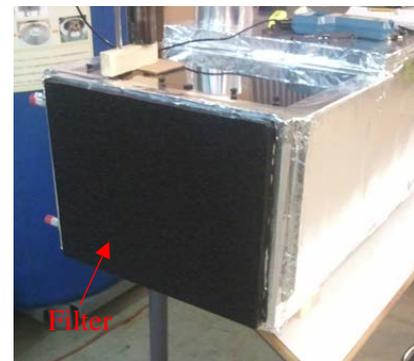
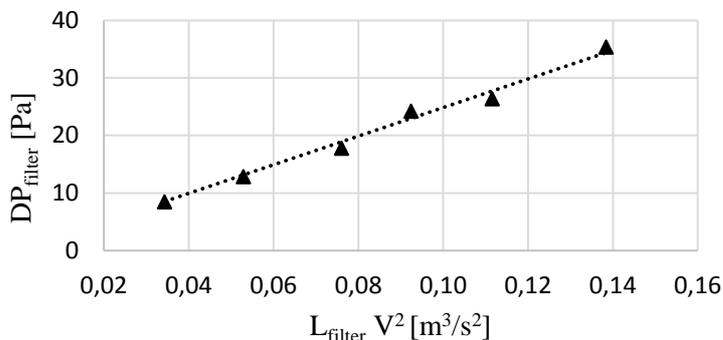


Figure 4: Filter characterization.

Starting from this equation, the  $K_{filter}$  that was calculated as:

$$K_{filter} = \frac{\Delta P_{filter}}{(L_{filter} \cdot V^2)} \quad (2)$$

Evaluated experimentally the parameter  $K_{filter}$ , once defined the desired air velocity profile as function of the heat exchanger height, if the  $\Delta P_{design}$  is defined as:

$$\Delta P_{design} = K_{filter} L_{filter, min} V_{max}^2 \quad (3)$$

The length and the position of the filter (i) can be evaluated by using the following equation:

$$L_{filter, i} = \frac{\Delta P_{design}}{(K_{filter} \cdot V_i^2)} \quad (4)$$

In the Eq.4,  $V_i$  represents the local value of the desired air velocity profile.

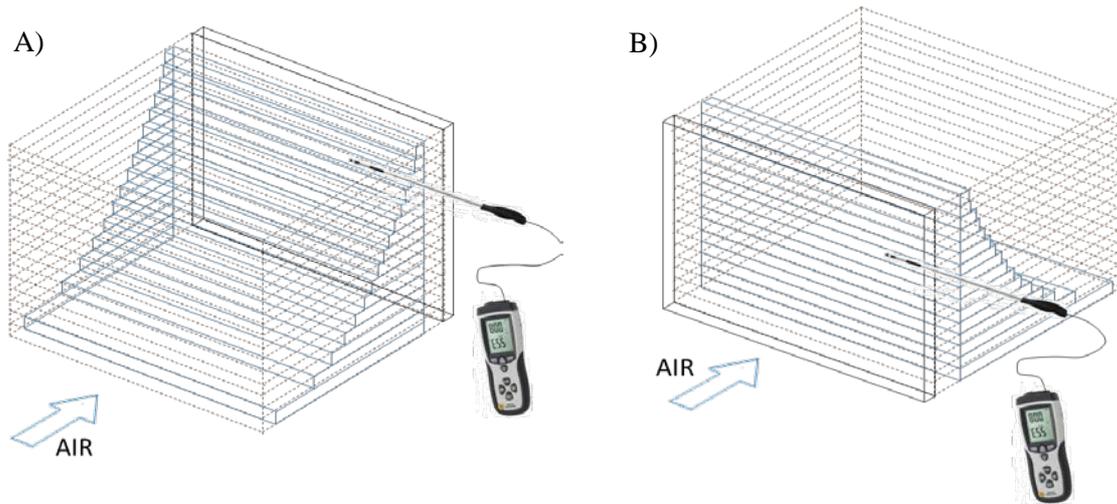


Figure 5: position of the filters, support structure and probe during the air velocity measurement

A linear air velocity profile has been reproduced at the inlet of the heat exchanger tested. Sixteen filters have been positioned inside the support structure as depicted in Fig.5. In order to nullify the effects of the turbulence, the probe is inserted in the structure measuring the air velocity between two consequent slabs. In Fig.6 the configuration of the filters (Eq. 4) and the comparison between the linear and the measured velocity profile are shown.

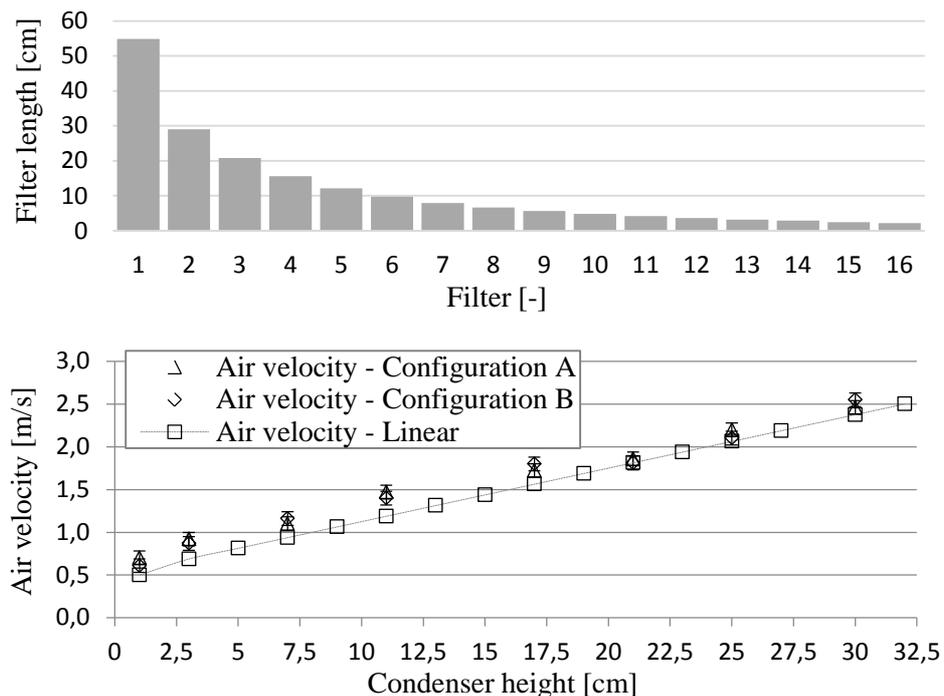


Figure 6: Linear air velocity profile

Two different positions of the structure and filters respect to the heat exchanger have been tested. It does not influence negatively the air velocity measurement. Indeed, with both the configurations A and B Fig.5 a very good agreement between the linear and the measured velocity profile has been obtained.

## 5. Experimental air-conditioning system and final results

Since the good results obtained in the air duct, the experimental method was used for analyzing the effects of the air maldistribution on the performance of an experimental air-conditioning system. The irregular profile showed in Fig.7 and Fig.8 was generated positioning several filters along inlet surface of the condenser. As proposed by Lee *et al.* [11], this air velocity profile is typical of one heat exchanger of an A-shape condensation unit.

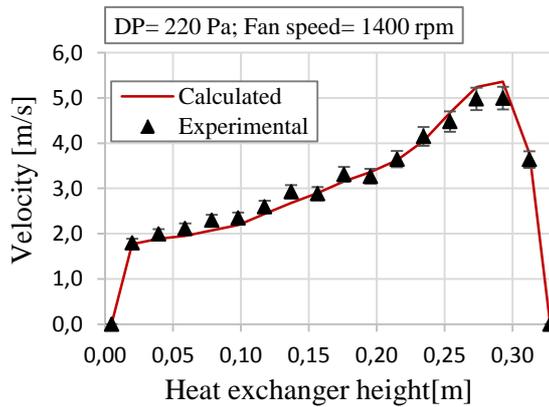
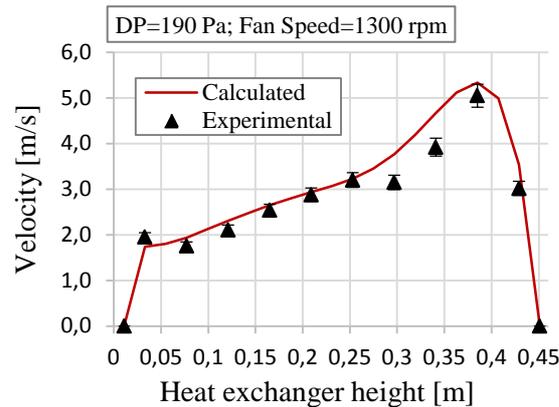
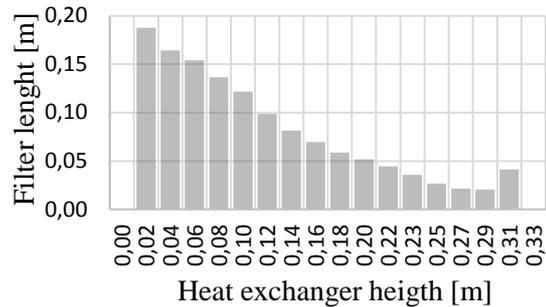
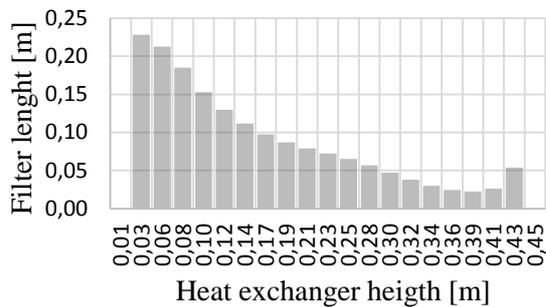


Figure 7: Air velocity measurement at the round tube heat exchanger inlet

Figure 8: Air velocity measurement at the microchannel heat exchanger inlet

Due to the position of the heat exchanger respect the stream of air, the airflow rate at the upper part of the condenser is clearly higher. In the cases tested, the highest air velocity (5 m/s) favors the heat transfer when the refrigerant is superheated. Otherwise, when the condensation process is finishing, the heat transfer is penalized due to the lowest value of the air velocity (1.5 m/s). At the extremities, the velocity has been maintained equal to zero. Depending on the heat exchanger geometry, this type of air maldistribution could modify in a different way the performance of the condenser. Therefore, the same velocity profile has been generated at the inlet of a round tube heat exchanger (RTPFs) and a microchannel heat exchanger. The condensers have been equipped with two specific support structures especially designed for them. In Fig.7 and Fig.8 is shown the position and the length of the filters. According to the characteristics of the RTPFs condenser, already widely described by Pisano *et al.* [12], and those of the microchannel (330 mm tall, 510 mm wide and 27 mm thick) a different number of filters were positioned.

The calculated velocity profile has been obtained by using the Eq.2 in which the experimental value of the pressure drop (Fig.7 and Fig.8) was used instead of the  $\Delta P_{design}$ . The rotational speed of the fan was

changed until reaching the calculated average value of the air velocity equal to 2.5 m/s (Fig.7 and Fig.8). Finally, the air velocity was measured with the hot-wire anemometer. For both the heat exchangers, the trend of the measured data fit with good accuracy the calculated air velocity profile.

## 6. Conclusions

In the present work, an experimental method for generating a non-uniform profile at the inlet of a heat exchanger has been presented. After the analysis of the first results, the description of the effective method can be summarized as follows:

- In a specific experimental facility, where the air is moved inside a long duct by means of an axial fan, the characterization of the filter has to be carried out. It consists in evaluating experimentally the relation between the filter length, the pressure drop and the air velocity at the inlet surface of a heat exchanger.
- Once defined the desired air velocity profile, according to the heat exchanger geometry, a certain number of filters have to be positioned. Their length is strongly connected to the results of the characterization process.
- The filters have to be positioned inside a support structure, where the filters are separated with thin slabs. In this way, the air velocity can be measured with a hot-wire anemometer without undergoing the negative effects of the turbulent flows.
- The method is highly flexible. It can be used for different types of heat exchanger and in different experimental facilities.

In this paper, the air velocity profile characterizing the inlet surface of an A-Shape condensation unit has been generated at the inlet of a Round Tube Plate Fins heat exchanger and a microchannel heat exchanger. Both the heat exchangers have been tested in an experimental air-conditioning system. The results of the air measurement show a very good agreement between the calculated profile and the measured one. In a further paper the analysis of the performance degradation due to this type of air maldistribution will be dealt.

## 7. Acknowledgement

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## 8. References

- [1] LEE, J., DOMANSKI, P. *Impact of air and refrigerant maldistributions on the performance of finned-tube evaporators with R-22 and R-407C*, s.l.: The Air-Conditioning and Refrigeration Technology Institute, 1997.
- [2] YAĀCI, W., GHORAB, M. and ENTCHEV, E. *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, 2014. Volumen 74, p. 490–500.
- [3] BACH, C., GROLL, E., BRAUN, J. and HORTON, W. *Mitigation of air flow maldistribution in evaporators*, Applied Thermal Engineering, 2014, Volumen 73, pp. 879-887.
- [4] BELL, I. *Air conditioning and Heat Pump Model (ACHP)*, s.l.: Sourceforge SCM Repository, 2012.
- [5] MAO, J., CHEN, H.X., JIA, H., WANG Y.Z., HU, H.M. *Effect of air-side flow maldistribution on thermal-hydraulic performance of the multi-louvered fin and tube heat exchanger*, 2013, International Journal of Thermal Sciences, Volumen 73, pp. 46-57.

- [6] MC WILLIAMS, J. *Review of Airflow Measurement Techniques*, 2002, Energy Performance of Buildings Group, Berkeley, USA.
- [7] YASHAR, D., CHO, H. and DOMANSKI, P. *Measurement of Air-Velocity Profiles for Finned-tube heat Exchangers Using Particle Image Velocimetry*, 2008, West Lafayette, Proceedings of International Refrigeration and Air Conditioning Conference.
- [8] DOMANSKI, P., YASHAR, D. and LEE, S. *Evaporator optimization for non-uniform air distribution*, Stockholm, 2010, Proceedings of Sustainable Refrigeration and Heat Pump Technology.
- [9] DATTA, S., DAS, P. and MUKHOPADHYAY, S. *Obstructed airflow through the condenser of an automotive air conditioner – Effects on the condenser and the overall performance of the system*, Applied Thermal Engineering, 2014, Volumen 70, pp. 925-934.
- [10] ASHRAE. *Fundamentals*, 2009, s.l.: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
- [11] LEE, T., WU, W., CHUANG, Y. and WANG, S. *An improvement of airflow and heat transfer performance of multi-coil condensers by different coil configurations*, 2010, International Journal of Refrigeration, Volume 33, pp. 1370-1376.
- [12] PISANO, A., MARTINEZ-BALLESTER, S., CORBERÁN, J.M., HIDALGO, F., ILLÁN, F., CASCALES, J.R. *A Discussion about the Methodology to Validate the Correlations of Heat Transfer Coefficients and Pressure Drop during the Condensation in a Finned-tube Heat Exchanger*, 2014, West Lafayette, Proceedings of International refrigeration and Air conditioning Conference.