

Experimental study on enhanced heat transfer by water spraying in the cooling air flow

D T Balanescu, V M Homutescu* and A G Lupu

Department of Mechanical and Automotive Engineering, “Gheorghe Asachi” Technical University of Iasi, Iasi, Romania

*E-mail: mariohomutescu@gmail.com

Abstract. Water spraying into the cooling air flow is one of the most efficient methods for heat transfer enhancement. Consequently, it is currently used in several applications and is analysed as a solution for others, where compactness is very important. Considering two practical cases, an air cooled steam condenser and a compact automotive radiator, authors performed an experimental study applying this method for a fin-and-tube heat exchanger with inline tube arrangement. Experiments were carried out using a wind tunnel. Reynolds number varied in the range 2082...4432, which includes entire transient flow range. Maximum specific sprayed water flow was $1.5 \cdot 10^{-3}$ kg of water per kg of dry air. For this specific flow, an increase of 52 to 73% was achieved for the overall specific heat transfer coefficient. The cooling air pressure loss in heat exchanger almost doubled at maximum Reynolds number, compared to the case described by minimum Reynolds number.

1. Introduction

Moisturizing of gaseous thermal agents (air or other gas types) is one of the most efficient way to enhance the heat transfer in heat exchangers. This method is not quite new, one application being the automotive radiator. Thus, the feasibility of a compact radiator with water or glycol spraying in cooling air flow was experimentally studied and an improvement of 40-45% of the heat transfer coefficient was achieved, [1].

There are several experimental studies on heat transfer enhancement by water spraying into the cooling air. The optimization of process parameters in the case of steel plate cooled by water sprayed air was analyzed in [2]. The influence of both spraying axis angle and geometry of heat exchanger fins was studied in [3] while the effect of the relative humidity of air on the heat transfer was investigated in [4]. Also, improvement of the heat transferred into a heat exchanger was studied [5], using water with nanofluid (Al_2O_3 particles) added to air moisturized by water spraying; an increase of heat transfer coefficient up to 57.7% was achieved by water spraying in cooling air and up to 76.3% when nanofluid was added into the water. An increase of heat exchanged up to 3 times is indicated in [6] when droplets of water smaller than 25 μm are sprayed in the cooling air flow. The experimental results were used to validate a prior numerical simulation.

Based on the experimental results, some mathematical models were developed. Thus, in [7] is proposed a model for analysis of performance and temperature distribution in parallel and counter-flow wet heat exchangers. Another model referring to moisturized air cooling is presented in [8]. This model optimizes the separation distance between the parallel plates of a heat exchanger.



Currently, spray cooling is applied in air conditioning, commercial and industrial cooling [7], [9], steel industry, cooling of electronic chips and microprocessors, laser techniques and heat exchangers of power plants [2]. Results of an experimental study on the air cooling condenser of an air cooling power generating unit are presented in [10]. The experiments on the condenser, with wavy finned flat tubes, were performed by wind tunnel for Reynolds number of the cooling air flow in the range 210-680; it was indicated that water spraying in the cooling air could increase Nusselt number up to 68%.

An application of the moisturized air cooling in power plants is the forced draught wet-cooling tower, where the process fluid flow (inside finned tubes) is cooled by the water-sprayed air flow [11]. Another application is the air-cooled steam condenser, recommended in arid places. A concentrated solar thermal power plant based on this type of condenser is presented in [12]. Aiming to increase the exergetic efficiency of the condenser, water spraying in cooling air flow is proposed as a method for heat transfer enhancement.

Impact of relative humidity on the heat transfer is also a matter of great interest in the case of condensing boilers, when the gaseous thermal agent is the flue gas. Experimental studies on this subject are presented in [13] and [14]. Based on experimental results, a formula for convective heat transfer coefficient was also obtained in [14]. A simplified mathematical model for the assessment of boiler efficiency function by relative humidity is proposed in [15].

The present study was performed by considering two applications of the wet heat exchangers, namely air cooled steam condenser – as component of a power plant or even of a mobile power generating unit – and compact automotive radiator. Therefore, an experimental investigation on the influence of the sprayed water flow over the heat transfer coefficient was performed in the case of a wet air cooled fin-and-tube heat exchanger with inline arrangement of tubes. Variation of the air pressure loss in heat exchanger was also analyzed.

2. Experimental test rig

Schematic and images of the experimental test rig are presented in figures 1 and 2. The experiments were carried out using a wind tunnel.

In order to perform the study, both air flow and sprayed water flow had to be determined. Air flow was determined as a function of air velocity, measured with an Anubar tube, AT, connected to a differential manometer, DM₂. For higher precision, DM₂ is inclined at 8° relative to horizontal.

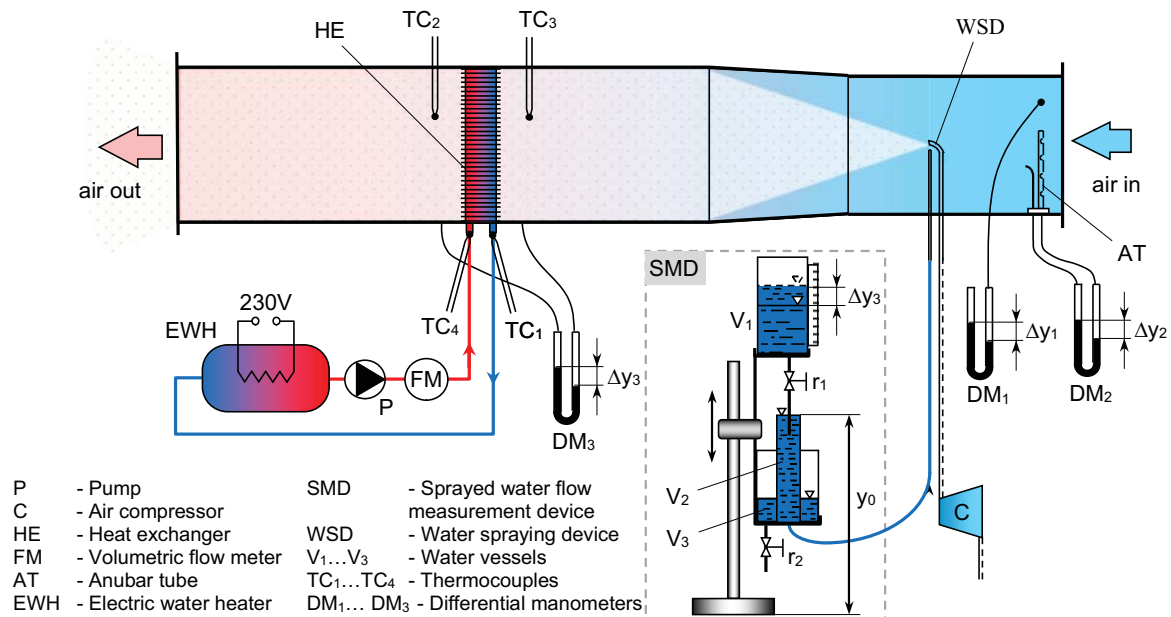


Figure 1. Schematic of the test rig.

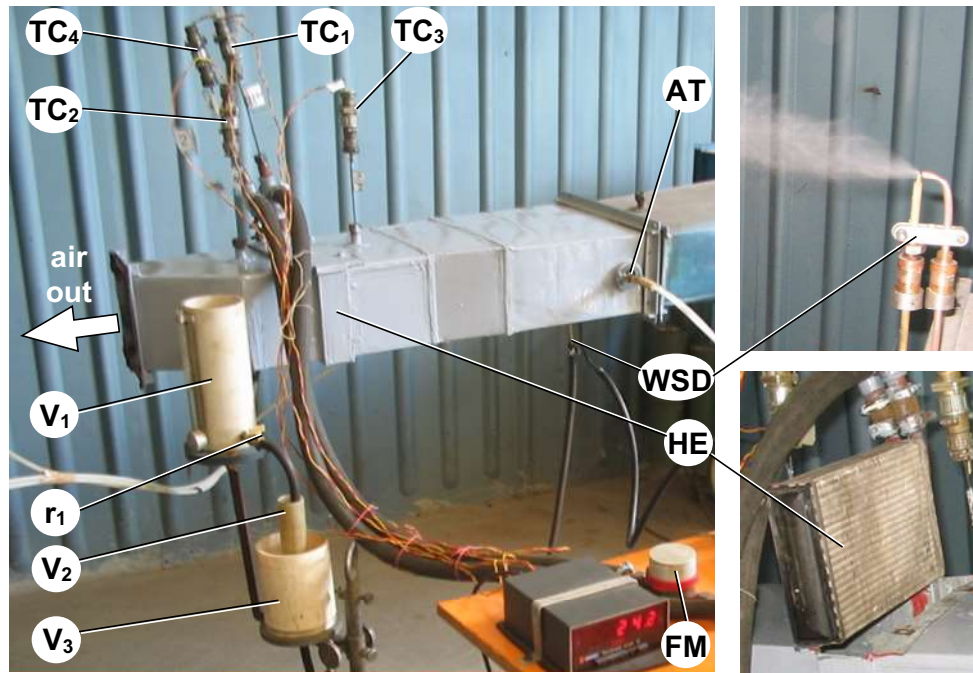


Figure 2. Test rig used in experiments.

The sprayed water flow is measured with SMD, by counting for each test the time and water level (initial and final) in V1; the internal diameter of V1 is 79 mm. SMD consists in an assembly of three water vessels (V1, V2 and V3). Each vessel is fixed relative to the others but the entire assembly can be moved up and down on the sliding rod SR. In order to keep constant the suction pressure of the water spraying device WSD, the level of water in V2 (y_0) was maintained constant on each tested regime by acting the tap t_1 . At the end of each test, the excess water, drained from V2 in V3, was moved back in V1; thus, resulted the final level of water in V1. The sprayed water flow was increased or reduced by changing the suction pressure (by moving up or down the assembly of vessels, thus modifying y_0).

3. Mathematical model and experiments

Experiments were carried out at several cooling air flows in HE and several sprayed water flows. The air flow was adjusted by the flow obstruction of the wind tunnel, while sprayed water flow was adjusted by modifying the level y_0 , as described above. At each regime, were recorded $\Delta y_1, \Delta y_2, \Delta y_3$, in mm H₂O, as well as indications of TC1...TC4 and FM. Based on the measured values, the specific sprayed water flow (ς), Reynolds number in heat exchanger (Re_{HE}) and overall heat transfer coefficient of HE (k) were determined by the calculation procedure described below.

Specific sprayed water flow was defined as

$$\varsigma = \dot{m}_{sw} \cdot \dot{m}_{da}^{-1} \quad [\text{kg of water} / \text{kg of dry air}], \quad (1)$$

where $\dot{m}_{sw}$, in kg/s, is the sprayed water flow, determined as described above, while $\dot{m}_{da}$, in kg/s, is the dry air flow. We note that ambient air was assumed as “dry air” in the study and the standard pressure $p_0 = 101325$ Pa was considered for the ambient (dry) air.

Dry air flow is expressed as

$$\dot{m}_{da} = \rho_A \cdot w_A \cdot A_A \quad [\text{kg} / \text{s}], \quad (2)$$

where: ρ_A - dry air density in the cross-section of the air duct where AT is placed, in kg/m³;
 w_T - velocity of dry air in the cross-section of the air duct where AT is placed, in m/s;
 A_A - cross-sectional area of air duct in the section where AT is placed; $A_A = 4.67 \cdot 10^{-3}$ m².

Dry air density in the section where AT is placed is calculated as

$$\rho_A = p_A \cdot (R_a \cdot T_a)^{-1} \quad [kg / m^3], \quad (3)$$

where: R_a - air constant; $R_a = 287.04 \text{ J/(kg}\cdot\text{K)}$;
 T_a - absolute temperature of ambient air;
 p_A - absolute pressure of air in the cross-section of the air duct where AT is placed, in Pa; it is given by

$$p_A = p_0 + \Delta p_1 = p_0 + 9.81 \cdot \Delta y_1 \quad [Pa]. \quad (4)$$

It should be noted that in formula (3) absolute ambient temperature, T_a , was used instead of the absolute temperature in the air duct section of AT, for simplification. All verification tests indicated that difference between the two temperatures is less than 1 K. This induces differences of maximum 0.3% in calculation of dry air density, which is not significant for the current study.

Velocity of dry air in the section where AT is placed, as well as mean velocity of dry air in HE are given by the calibrating equations of these two components of the installation. They are expressed as functions of Δy_2 , which is indicated by DM₂:

$$w_{A,HE} = C_{1A,HE} + C_{2A,HE} \cdot \Delta y_2 - C_{3A,HE} \cdot (\Delta y_2)^{-1} \quad [m / s]. \quad (5)$$

Coefficients used in formula (5) are

$$\begin{array}{lll} C_{1A} = 5.058263 & C_{2A} = 0.217592 & C_{3A} = 6.519252 \\ C_{1HE} = 18.410727 & C_{2HE} = 0.7999344 & C_{3HE} = 23.686573 \end{array}.$$

Once w_{HE} is determined, Reynolds number in HE was calculated with the known formula

$$Re_{HE} = \rho_{HE} \cdot w_{HE} \cdot D_H \cdot \mu_{HE}^{-1}, \quad (6)$$

where: D_H - hydraulic diameter of HE; $D_H = 1.367 \cdot 10^{-3} \text{ m}$;
 μ_{HE} - mean dynamic viscosity of air in HE, in kJ/(kg K) ;
 ρ_{HE} - mean density of air in HE, in kg/m^3 ; it is calculated similar to ρ_A , with formula (3), but replacing T_a and p_a with the absolute temperature and pressure of air in HE

$$T_{HE} = 0.5 \cdot (t_3 + t_2) + 273.15 \quad [K], \quad (7)$$

$$p_{HE} = p_0 + 0.5 \cdot \Delta p_3 = p_0 + 9.81 \cdot 0.5 \cdot \Delta y_3 \quad [Pa]. \quad (8)$$

Air temperatures t_2 and t_3 from formula (7) are indicated by thermocouples TC₂ and TC₃ (figures 1 and 2). In formula (8), Δp_3 is the air pressure loss in HE, indicated by the differential manometer DM₃.

Mean dynamic viscosity of air is calculated with formula [16]

$$\mu_{HE} = 1.458 \cdot 10^{-6} \cdot T_{HE}^{1.5} \cdot (T_{HE} + 110.4)^{-1} \quad [N \cdot s / m^2]. \quad (9)$$

Heat transferred from water to the moist air in HE is a sum of two components – sensible heat and latent heat, involved in vaporization of water droplets sprayed in installation. The heat flow rate is

$$\dot{Q}_{HE} = \dot{Q}_s + \dot{Q}_l = \dot{V}_w \cdot \rho_w \cdot c_{pw} \cdot (t_4 - t_1) \quad [W], \quad (10)$$

where: $\dot{V}_w$ - volumic flow rate of water in radiator, in m^3/s ; it is indicated by the volumetric flow meter FM (see figure 1 and figure 2);
 ρ_w - mean density of water in HE, in kg/m^3 ;
 c_{pw} - mean specific isobaric heat capacity of water in HE, in J/(kg K) ;
 t_4, t_1 - HE water inlet and outlet temperature, in K; are indicated by TC₄ and TC₁.

Mean density and mean specific isobaric heat capacity of water in HE were calculated according to [17] by considering the mean temperature of water in HE.

The overall heat transfer coefficient of HE is

$$k = \dot{Q}_{HE} \cdot (S_{HE} \cdot LMTD)^{-1} \quad \left[W / (m^2 \cdot K) \right], \quad (11)$$

where S_{HE} is the heat transfer surface of HE (2.496 m²) while $LMTD$ is the logarithmic mean temperature difference; it is expressed as a function of the temperatures of water (t_1 , t_4) and (t_2 , t_3), indicated by thermocouples TC₁...TC₄.

4. Results and interpretation

Results of the experimental study are presented in figures 3 and 4. The range of Reynolds number (Re_{HE}) covered by experiments was 2082...4432, which is larger than transient flow range ($2300 < Re < 4000$). As figure 3 indicates, the air pressure loss in HE (Δp_3) increased 2.7 times (from 961 Pa to 2560 Pa) over the entire range of Re_{HE} , which corresponds to a variation of air velocity from 24.6 m/s to 52.3 m/s.

In figure 4 there are presented variation curves of the overall specific heat transfer coefficient (k) with Re_{HE} for several values of the specific sprayed water flow (ζ). All the curves represent logarithmic interpolation of the experimental results.

The dashed curve in figure 4 is the reference curve and corresponds to the case when dry cooling air was used in experiments (no water spraying, $\zeta = 0$). For this case, over the entire range of Re_{HE} mentioned above, k increased from 70 to 198 W/(m² K).

It should be noted that in this study, the maximum specific sprayed water flow was 1.5 g of water/kg of dry air. For example, at 30°C, when the specific enthalpy of vaporization is 2430 kJ/kg, the complete vaporization of this flow requires a thermal heat rate of 3.6 kW. This thermal potential of the latent heat associated to sprayed water is impressive, since it is close to the maximum thermal heat rate transferred in HE. But, with respect to the density or velocity of air, a sprayed water flow of 1.5 g per kg of dry air has no significant influence. As a reference, standard moisture content usually considered in engineering is 8 g/kg of dry air or 10 g/Nm³ of dry air, while maximum moisture content in air is 30.4 g/m³ – at saturation, at 30°C. That is why it was assumed that Reynolds number was not significantly influenced by sprayed water.



Figure 3. Variation of Δp_3 with Re_{HE} and w_{HE} .

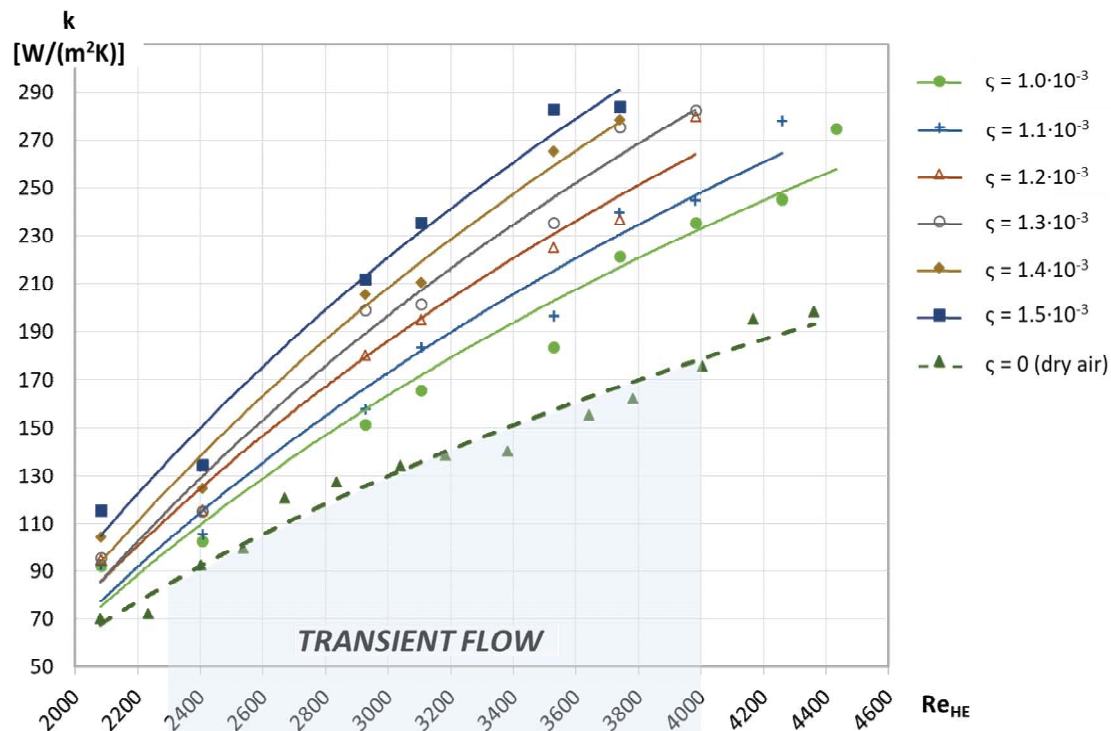


Figure 4. Variation of k with Re_{HE} for several values of ζ .

Analysing the data presented in figure 4, it can be inferred that water spraying in the cooling air has a significant influence on the heat transfer. The value of the overall heat transfer coefficient increased continuously when ζ increased from 10^{-3} to $1.5 \cdot 10^{-3}$ kg of water / kg of dry air, as well as with Re number.

The minimum gain in heat transfer enhancement was obtained for $\zeta = 1.0 \cdot 10^{-3}$ kg of water / kg of dry air and $Re_{HE} = 2082$. In this case, $k = 73$ W/(m²·K). For the same Reynolds number, the equivalent value of k from the reference dry air curve is 69 W/(m²·K). Thus, there is a gain, but of only 6%.

The most intense heat transfer was measured for $\zeta = 1.5 \cdot 10^{-3}$ kg of water / kg of dry air. The values for k on this curve increase from 105 W/(m²·K) when $Re_{HE} = 2082$, to 290 W/(m²·K) when $Re_{HE} = 3740$ and representing the largest value of k in the entire study. Since the equivalent value of k , indicated by the dashed reference curve, is 168 W/(m²·K) for $Re_{HE} = 3740$, it means that k increases with 52 to 73% – relative to the reference case – when $\zeta = 1.5 \cdot 10^{-3}$ kg of water / kg of dry air.

5. Conclusions

In the experiments carried out in this study, the range of Reynolds number, Re_{HE} , was 2082...4432, larger than transient flowing regime range. The corresponding range of mean air velocity in the fin-and-tube heat exchanger, w_{HE} , was 24.6...52.3 m/s. The mentioned flow conditions are correlated with two applications considered, an air cooled steam condenser (as component part of a power generating unit – steady or even mobile) and a compact automotive radiator.

Air pressure loss increased 2.7 times, from 961 Pa to 2560 Pa, when Re_{HE} and w_{HE} varied from 2082 and 52.3 m/s to 4432 and 52.3 m/s.

Overall heat transfer coefficient, k , was very sensitive to the increase of the specific sprayed water flow; at minimum Re_{HE} and w_{HE} (2082 and 24.6 m/s, respectively), k has increased with 6 to 52% – relative to the reference case of dry air cooling, when ζ varied from 10^{-3} to $1.5 \cdot 10^{-3}$ kg of water / kg of dry air.

Maximum gain in what concerns value of k was 73% – achieved at maximum value of ζ ($1.5 \cdot 10^{-3}$ kg of water / kg of dry air), when $Re_{HE} = 3740$ and $w_{HE} = 44.1$ m/s.

6. References

- [1] Yang W J and Clark D W 1975 Spray cooling of air-cooled compact heat exchangers *International Journal of Heat and Mass Transfer* **18** 2 pp 311-317
- [2] Nayak S K, Mishra P C and Parashar S K S 2016 Experimental investigation and optimization of air-water spray impingement cooling to enhance heat transfer *International Journal of Engineering and Technology* **8** 1 pp 379-390
- [3] Silk E A, Kim J and Kiger K 2006 Spray cooling enhanced surfaces: impact of structured surface geometry and spray axis inclination *International Journal of Heat and Mass Transfer* **49** 25-26 pp 4910-4920
- [4] Fujita A, Kurose R and Komori S 2010 Experimental study on effect of relative humidity on heat transfer of an evaporating water droplet in air flow *International Journal of Multiphase Flow* **36** 3 pp 244-247
- [5] Abed Y H, Karamallah A A and Saleh A M 2015 Experimental study to enhanced heat transfer on a heat exchanger AL2O3 water-nanofluid using an air flow with water droplets *International Journal of Application or Innovation in Engineering & Management* **4** 1 pp 33-45
- [6] Tissot J, Boulet P, Labergue A, Castanet G, Trinquet F and Fournaison L 2012 Experimental study on air cooling by spray in the upstream flow of a heat exchanger *International Journal of Thermal Sciences* **60** pp 23-31
- [7] Mehrabian M A and Samadi B 2010 Heat-transfer characteristics of wet heat exchangers in parallel-flow and counter-flow arrangements *International Journal of Low-Carbon Technologies* **5** pp 256-263
- [8] Issa R J 2010 Heat transfer optimization for air-mist cooling between a stack of parallel plates *Journal of Thermal Science* **19** 3 pp 253-260
- [9] Papakostas K T and Kiosis G C 2015 Heat recovery in an air-conditioning system with air-to-air heat exchanger *International Journal of Sustainable Energy* **34** 3-4 pp 221-231
- [10] Xiao L, Wu T, Feng S, Du X and Yang L 2017 Experimental study on heat transfer enhancement of wavy finned flat tubes by water spray cooling *International Journal of Heat and Mass Transfer* **110** pp 383-392
- [11] Kroger D G 2004 *Air-cooled heat exchangers and cooling towers. Thermal-flow performance evaluation and design* (Tulsa, Oklahoma: PennWell Corporation)
- [12] Dirbude S, Khalifa N and Chandra L 2017 Selective design of an experiment for evaluating air-water hybrid steam condenser for concentrated solar power, in book *Concentrated solar thermal energy technologies: recent trends and applications* (Singapore: Springer Verlag) pp 89-102
- [13] Zhu X, Sui X, Zhao Y, Meng J and Li Z 2016 Experimental study on the flow and heat transfer of a gas-water mixture through a packed channel *Science Bulletin* **61** 5 pp 406-415
- [14] Che D, Da Y, Zhuang Z 2005 Heat and mass transfer characteristics of simulated high moisture flue gases *Heat and Mass Transfer* **41** 3 pp 250-256
- [15] Barros J P, Azevedo J L T and Monteiro L 2014 Effect of relative humidity in the efficiency of condensing gas water heater appliance *Applied Thermal Engineering* **65** 1-2 pp 66-73
- [16] Dixon J C 2007 *The Shock Absorber Handbook – Second Edition* (John Wiley & Sons)
- [17] Wagner W, Kretzschmar H J 2008 *International Steam Tables. Properties of Water and Steam Based on the Industrial Formulation IAPWS-IF97 – Second Edition* (Berlin: Springer)

Acknowledgments

This work was partly supported by the project POSCCE-A2-O2.2.1-2009-4-ENERED, ID nr. 911, co-financed by the European Social Fund within the Sectoral Operational Program “Increase of Economic Competitiveness”.