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

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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 \times 10^{-3} \text{ 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 nano-fluid (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 nano-fluid 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 \mu m 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 moistened 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.](image-url)
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 ($\zeta$), 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

$$\zeta = \frac{\dot{m}_{sw}}{\dot{m}_{da}}$$

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

$$\dot{m}_{da} = \rho_a \cdot w_T \cdot A_d \ [kg/s],$$

where: $\rho_a$ - 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;
$A_d$ - cross-sectional area of air duct in the section where AT is placed; $A_d = 4.67 \cdot 10^{-3}$ m².
Dry air density in the section where AT is placed is calculated as

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

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

\[ p_A = p_0 + \Delta p_1 = p_0 + 9.81 \cdot \Delta y_1 \quad \text{[Pa]}, \quad (4) \]

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 \( \Delta y_2 \), which is indicated by DM2:

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

Coefficients used in formula (5) are

\[ C_{1,HE} = 5.058263 \quad C_{2,HE} = 0.217592 \quad C_{3,HE} = 6.519252 \]

\[ C_{1,HE} = 18.410727 \quad C_{2,HE} = 0.7999344 \quad C_{3,HE} = 23.686573 \]

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; \( D_H = 1.367 \cdot 10^{-3} \text{ m} \);  
- \( \mu_{HE} \) - mean dynamic viscosity of air in HE, in kJ/(kg K);  
- \( \rho_{HE} \) - mean density of air in HE, in kg/m\(^3\); it is calculated similar to \( \rho_A \) with formula (3), but replacing \( T_a \) and \( p_a \) with the absolute temperature and presure of air in HE

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

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

Air temperatures \( t_2 \) and \( t_3 \) from formula (7) are indicated by thermocouples TC2 and TC3 (figures 1 and 2). In formula (8), \( \Delta p_3 \) is the air pressure loss in HE, indicated by the differential manometer DM3.

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

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

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

\[ Q_{HE} = \dot{Q}_v + \dot{Q}_l = \dot{V}_w \cdot \rho_w \cdot c_{pw} \cdot \left( t_4 - t_1 \right) \quad \text{[W]}, \quad (10) \]

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);  
- \( \rho_w \) - 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

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

where \( S_{\text{HE}} \) is the heat transfer surface of HE (2.496 m\(^2\)) while \( \text{LMTD} \) 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 TC\(_1\)…TC\(_4\).

4. Results and interpretation

Results of the experimental study are presented in figures 3 and 4. The range of Reynolds number \((Re_{\text{HE}})\) 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 \((\Delta p_3)\) increased 2.7 times (from 961 Pa to 2560 Pa) over the entire range of \(Re_{\text{HE}}\), 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 \(Re_{\text{HE}}\) for several values of the specific sprayed water flow \((\varsigma)\). 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, \(\varsigma = 0\)). For this case, over the entire range of \(Re_{\text{HE}}\) mentioned above, \(k\) increased from 70 to 198 W/(m\(^2\) 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/Nm\(^3\) of dry air, while maximum moisture content in air is 30.4 g/m\(^3\) – 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 \(\Delta p_3\) with \(Re_{\text{HE}}\) and \(w_{\text{HE}}\).](image-url)
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 $\zeta$ 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, $k = 73 \text{ W/(m}^2\cdot\text{K)}$. For the same Reynolds number, the equivalent value of $k$ from the reference dry air curve is 69 W/(m$^2$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/(m$^2$K) when $Re_{HE} = 2082$, to 290 W/(m$^2$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/(m$^2$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 24.6 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 $\zeta$ 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 $\zeta$ ($1.5 \cdot 10^{-3}$ kg of water / kg of dry air), when $Re_{HE} = 3740$ and $w_{HE} = 44.1$ m/s.
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