Journal of Physics: Conference Series

PAPER • OPEN ACCESS

Numerical Model of an Evaporative Condenser for the Food Refrigeration Industry

To cite this article: Michael Giovannini and Marco Lorenzini 2022 *J. Phys.: Conf. Ser.* **2177** 012009

Numerical Model of an Evaporative Condenser for the Food Refrigeration Industry

Michael Giovannini, Marco Lorenzini

Università di Bologna, DIN – Dipartimento di Ingegneria Industriale, Via Fontanelle 40, I-47121, Forlì (FC) Italy,

marco.lorenzini@unibo.it

Abstract. Evaporative condensers play a key role in refrigeration plants for industrial applications, as they are needed to dispose of the thermal and electric power required by the process. Modelling them as lumped capacitance systems is very expedient when trying to simulate e. g. the steady-state and dynamic behaviour of the whole refrigeration plant, but at the expenses of loss of information and often oversimplifying assumptions. This paper analyses two models for heat and mass transfer in cooling equipment, Poppe's and its earlier, simpler version, Merkel's. The two approaches are applied to describe the transport phenomena in an evaporative condenser and the predicted cooling power compared to the data declared by the manufacturer under certain operating conditions. It is demonstrated that predictions are accurate within less than 10%.

1. Introduction

Evaporative condensers have come into use at food processing and storage plants comparatively recently. This piece of heat transfer equipment basically combines an air-cooled and a water-cooled condenser together, with benefits in terms of overall costs.

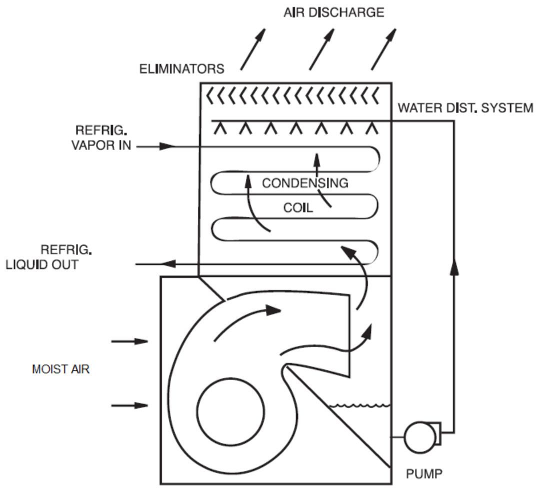


Figure 1: Schematic drawing of an evaporative condenser.

In this type of condenser the heat is transferred from the refrigerant through tube walls to a thin film of water, which flows over the outer surface of the tubes and then to air, mainly due to partial evaporation of water. These condensers can be profitably employed especially in dry and hot climates. Water consumption in the condenser is moderate and can be estimated to amount to 0.4 kg per 1000 kJ of energy transferred as heat, in the case evaporation is the only means of heat removal. During summertime up to 80% of the thermal energy removed may be due to evaporation. As sketched in Fig.1, water is sprayed onto the tubes by means of nozzles and partly evaporates, but mostly flows by gravity into a sump, where it is collected and pumped back to the nozzles above the tube bank. The evaporative process is enhanced by radial or axial fans, which either blow or draft air over the tubes.

On one hand, there is no longer the need to treat and circulate large amounts of water, on the other, fans needed to maintain the airflow are smaller and require significant less power than in air-cooled systems. The most prominent feature, however, is that evaporative condensers can be designed for a lower condensation temperature than single-fluid condensers, which results in lower power consumption and smaller operation pressure designs for the compressors. In the winter months, evaporative condensers may be operated as air-cooled machines, thus eliminating water pumping and reintegration costs. Also, the use of a fluid collector and separator downstream of the condenser ensures a vapor quality of the refrigerant at the outlet close to zero (saturated liquid). The disadvantages of evaporative condensers are related to fouling issues, as scales tends to form rather quickly on the outside of the tubes, and some cleaning difficulties of the inside of the ducts may also be encountered [1,2].

Given the wide range of transport phenomena involved and the practical significance of the problem, the mechanisms of heat and mass transfer occurring have been the subject of several investigations in the past: the earliest studies, which all successive research refers to, are those of Merkel [3] and Poppe [4], and they will be further detailed in the following. Merkel’s model was applied with additional simplifying assumptions by Baker and Shyrock [5] to study a condenser. Extensive work has been carried out by Dreyer, starting from his Doctoral Thesis [6], in which some calculation methods for heat transfer are outlined; Poppe’s model is the starting point, from which a simplified formulation is obtained, close to Merkel’s, and a lumped-parameter model is also hinted at. In this work [6] a list of correlation to compute the heat and mass transfer coefficients is given, according to the type of evaporative condenser considered: the models differ for the fluid flow direction, layout and presence or lack of additional heat transfer blocks to cool the water dripping from the tubes, known as fill packs. Further explanations on the calculation processes in [6] are given in a later work [7], which examines the modes of computation best suited for a given layout and geometry of the tubes.

A rather involved, if general, methodology to determine evaporative heat transfer for different arrangements and two or three participating fluids is given by Halàsz [8].

Merkel’s model was used by Hasan and Siren [9] to describe evaporative heat transfer in closed wet cooling towers used for climatization of office buildings; in [10] the same subject was further investigated employing simplified analytical formulations in CFD simulations.

Air-cooled heat exchangers and cooling towers are treated extensively in the book by Kröger [11], which describes both Merkel’s and Poppe’s approaches and considers them for several practical applications. Heat transfer and pressure drop in heat exchangers are discussed, and a wide selection of applicable correlations offered, also regarding the heat and mass transfer coefficients in the liquid film around tubes.

In a further work, [12], Kloppers and Kröger discuss the effect of the Lewis factor (which had already been related to the Lewis number in [4]) on the heat exchanger performance for different layouts of cooling towers.

Facao and Oliveira, [13], discuss the possibility of employing computer simulations to obtain mass transfer correlations based on their comparison of experimental data for an indirect contact cooling tower with the results of CFD simulations.

Expanding on the work of Dreyer, [4], Yilmaz [14] has suggested more recently an integral analytical approach to obtain the performance of wet cooling towers.

The literature survey demonstrates that the modelling of evaporative coolers, wet cooling towers and air-cooled condenser continues to be a lively area of interest, given the wide range of modelling issues which are also dependent on the details required by the application for which they are devised.

This work intends to investigate the behaviour of an evaporative condenser without fill pack to use it as a simulation block in the framework of a larger, mostly lumped-parameter, model of a refrigeration plant for food (fruit and vegetables) storage located in northern Italy. The full model is aimed towards analysis of plant performance when condensation temperature is lowered following more favourable ambient conditions, that is cooler weather. Evaporative condensers allow this, which has benefits in terms of energy consumption and service life, but there is a strong resistance from the conductors, lest storage conditions in the cells should fail to ensure perfect conservation of the goods. Simulation thus becomes a way to demonstrate the feasibility and profitability of this choice.

2. Modelling approaches: Poppe’s and Merkel’s

The choice of the model for the evaporative condenser was dictated, as mentioned above, by the final use of the model itself, i.e. the simulation of the behaviour of a refrigeration plant for fruit and vegetable storage. A full CFD model would be computationally too intensive and difficult to integrate into a larger framework, potential candidates remain therefore lumped-parameter and distributed-parameter models. The former would be a computationally better choice, as quantities are at most time-dependent only, and the model becomes algebraic for steady state conditions. Unfortunately, the underlying assumptions required to use a lumped parameter formulation (as discussed by Kröger in [11]), especially those concerning water temperature distribution throughout the tube banks are not verified in practice. For this reason, the choice fell on a distributed-parameter model, in which the quantities not only depend on time, but also on some spatial coordinate, which is determined by the physical process to be described.

Several models found in literature, [3-14] have been considered, all dealing with three fluids, of which two (air and water) interact directly, whereas the third (a refrigerant of some kind) undergoes condensation behind a physical barrier (the pipe wall). The governing equations can be applied to co-flow counterflow and crossflow arrangements, which determine the way in which numerical integration of the equations is carried out.

2.1. Merkel’s simplified model

The more extensive version of the model employed, the one given by Poppe [4] will be detailed in the following subsection. Here the simplifying assumptions underlying it are recalled, as they are shared by Merkel’s model as well. These are:

- negligible radiation heat transfer;
- uniform distribution of recirculated water, both over the coil and the fill-pack, if present;
- uniform temperature in the water film over the coil, owing to its small thickness;
- area of the heat transfer surface between liquid film and water equals that of the outer pipe wall;
- contributions to heat transfer by end-elements of coil is negligible;
- the enthalpy increase in the water due to circulation pump is neglected.

In order to reduce the number of equations to be solved, further simplifying assumptions may be made, as suggested by Merkel, namely:

- variations in the mass flowrate of water through the coil due to evaporation of the liquid film are neglected;
- the heat transfer rate during the evaporative process is comparable to the mass transfer rate, which is the same as taking the Lewis factor Le_f, i.e. the ratio of the Stanton number for mass transfer to the Stanton number for heat transfer close to unity.

Within this framework, the governing equations that describe the problem are all derived from heat and mass transfer balances over a control volume and are listed below for dry air specific enthalpy, water temperature and refrigerant specific enthalpy.

$$dh_a=\frac{\beta}{\dot{m}_a}\left(h_{a,sw}-h_a\right)dA_o \tag{1}$$

$$dT_w=\frac{1}{\dot{m}_w\cdot c_{p,w}}\left(\dot{m}_a dh_a+\dot{m}_r dh_r\right) \tag{2}$$

$$dh_r=\frac{U}{\dot{m}_r}\left(T_r-T_w\right)dA_o \tag{3}$$

To which a further equation is added, in case the refrigerant is superheated:

$$dT_r=\frac{1}{c_{p,r}}dh_r \tag{4}$$

The subscripts ‘w’, ‘r’ and ‘a’ refer to the three fluids, namely water, refrigerant and air, whilst ‘s’ stands for ‘saturation and ‘o’ for ‘outer’. The mass transfer coefficient is β, c_p is the specific thermal capacity at constant pressure, T is the temperature and A the outer area of the pipe. $\dot{m}$ is the mass flowrate and U is the overall heat transfer coefficient, as defined by Eq. (5):

$$U=\left(\frac{1}{\alpha_r}\left(\frac{d_o}{d_i}\right)+\frac{d_o}{2\lambda}\ln\left(\frac{d_o}{d_i}\right)+\frac{1}{\alpha_w}\right)^{-1} \tag{5}$$

In Eq. (5) d is the pipe diameter, the subscript ‘i’ is for ‘inner’, λ is the thermal conductivity of the pipe and α the heat transfer coefficient. The pressure drop of the refrigerant flowing within the cooling coil is neglected and all thermodynamic and thermophysical properties needed in the computations are obtained from data fits.

Several correlations for the heat and mass transfer coefficients are available in the literature [11], and an extensive discussion on them for a similar case, albeit with a fill pack has been given in [15]. In the calculations, for the refrigerant, Shah’s correlation is used during condensation, and Dittus-Boelter for the superheated fluid [11,15]. The mass transfer coefficient b is computed according to the correlations by either Parker and Treybald [11,15] or Nitsu [11,15], Eqs. (6) and (7) respectively:

$$\beta=0.04935\cdot\left(\frac{\dot{m}_a}{A_c}\right)^{0.905} \tag{6}$$

$$\beta=0.076\cdot\left(\frac{\dot{m}_a}{A_c}\right)^{0.8} \tag{7}$$

where the subscript ‘c’ is for ‘cross-section’. For the heat transfer in the water film, the operating conditions, which are detailed below, fell out of the bonds of the available correlations. It was therefore chosen to use the one closest to the operating conditions, i.e. Erens and Dreyer’s [11,15]:

$$\alpha_w=2843\cdot\left(\frac{\Gamma_m}{d_o}\right)^{0.384} \tag{8}$$

Where Γ_m is the average mass flowrate of water over the pipes per unit length.

2.2. Poppe's extended model

The more complex formulation of the governing equations proposed by Poppe [4] was also implemented. This is obtained by dropping the last two assumptions mentioned in the previous subsection. Equations (3) and (4) remain unchanged, whilst Eqs. (1) and (2) become:

$$dh_a=\frac{\beta}{\dot{m}_a}\left(Le_f\left(h_{a,sw}-h_a\right)-(Le_f-1)\left(w_{a,sw}-w_a\right)h_v\right)dA_o \tag{9}$$

$$dT_w=\frac{1}{\dot{m}_w\cdot c_{p,w}}\left(\dot{m}_a dh_a+c_{p,w}T_w d\dot{m}_w+\dot{m}_r dh_r\right) \tag{10}$$

To which two more equations are added:

$$dw_a=\frac{\beta}{\dot{m}_a}\left(w_{a,sw}-w_a\right)dA_o \tag{11}$$

$$d\dot{m}_w=-\dot{m}_a dw_a \tag{12}$$

In Eqs. (11) and (12) w is the humidity ratio and the superscript 'v' is for 'vapour'. The set of equations constituted by Eq. (3) with Eqs. (9)-(12) is valid for unsaturated air; if this is not the case, Eqs. (9) and (11) are replaced by Eqs. (13) and (14) respectively:

$$dh_a=\frac{\beta}{\dot{m}_a}\left(Le_f\left(h_{a,sw}-h_a\right)-(Le_f-1)\left(w_{a,sw}-w_{as}\right)h_v + Le_f(w_a-w_{as})c_{p,w}T_w\right)dA_o \tag{13}$$

$$dw_a=\frac{\beta}{\dot{m}_a}\left(w_{a,sw}-w_{as}\right)dA_o \tag{11}$$

The Lewis factor is computed for both unsaturated and supersaturated conditions using Bosnjacovic's equations [11].

3. Model implementation

The model was implemented for the case in which no fill pack is installed, and the results of the simulation were checked against the data declared by one manufacturer (BAC-Baltimore, USA) for their VXC model; in [15] the results for an evaporative condenser with fill pack (BAC – CXVE) are reported and the influence of the choice of the correlations for heat and mass transfer coefficients are thoroughly discussed.

Figure 2 shows how each rank, that constitutes a separate circuit, through which the condensing refrigerant flows, is subdivided in control volumes, together with the flow paths for the refrigerant, air and water. Balance equations are applied for each control volume and are discretised with fixed steps through a forward scheme. Integration is carried out using an in-house code developed in Python, starting from the topmost element where the superheated ammonia enters the circuit and following the path of the refrigerant. The outputs for Poppe's model are air enthalpy, humidity ratio and temperature, water temperature over the coil and in the sump, and refrigerant temperature, which are mapped unto the elements of the domain (hence the distributed-parameter nature of the method). If Merkel's approach is considered, no information on air temperature and humidity ratio is obtained.

The heat transfer from the refrigerant to the cooling fluid is also computed. A predictor-corrector approach is used, whilst an iterative (control) is employed to obtain the inlet values of enthalpy for the air in the element corresponding to the refrigerant inlet. When Poppe's model is used, the temperature of the air must at the inlet element must be determined likewise.

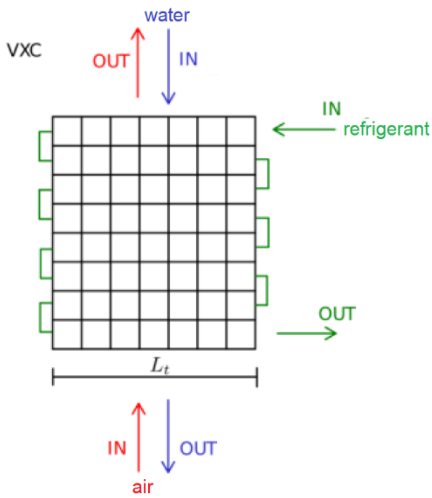
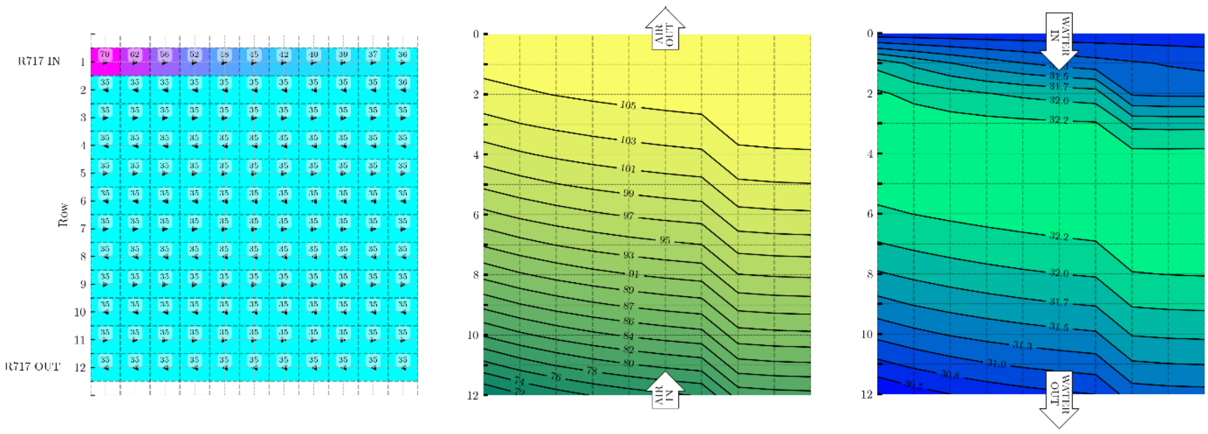


Figure 2. Flow directions for the fluids and element discretisation.

4. Results and discussion

The operating conditions against which the model is tested are air dry bulb and wet bulb temperatures, T_{db}=37 °C, T_{wb}=25.3 °C, corresponding to a relative humidity φ=38 %, and condensation pressure for ammonia $p_{r,s}$=13.50 bar (which corresponds to a saturation temperature $T_{r,s}$=35.0 °C). The cooling power stated by the manufacturer under these operating conditions (steady state) is P_c=1410 kW.

Figure 3 shows the plots for Merkel's model, whilst those for Poppe's are shown in Fig. 4.



(a) (b) (c)

Figure 3 Refrigerant temperature (°C) (a), enthalpy of air (kJ·kg^{-1}) (b) and water temperature (°C) (c), Merkel's model.

The refrigerant temperature is plotted in Fig. 3a: as is consistent with the operation mode of evaporative condensers, the fluid enters at superheated state (70 °C, topmost left element) and quickly reaches saturation temperature, exiting at a quality close to zero (0.01 in this case). The thermal power removed from R717 is P_c=1374.1 kW, less than 2.5% below the value declared by the manufacturer. Figure 3b is the plot of air enthalpy: as expected, isolines run almost horizontally, as is the case for the temperature of the water dripping from the tubes, Fig. 3c. It can be noticed that the higher refrigerant temperature at the inlet creates higher temperature gradients in the water, and the interaction of the liquid dripping downwards with the air being blown from below is responsible for the ensuing pattern. A distributed parameter formulation is thus demonstrated as being necessary, since the water temperature over the tubes is not uniform, and in [15] it was proven that this approach allows to discard correlations for heat and mass transfer which apparently yield even better results, but which would cause the temperature of the water in some volumes to fall several degrees below the wet bulb temperature, which is unphysical. The model also yields the amount of heat transfer to the air due to evaporation, which amounts to around 98% of the total, again as expected for an evaporative condenser. It is also to be remarked that the convective heat transfer coefficient in the water film, given by Eq. (8) is obtained outside the bounds of the correlation, and that the value obtained α_w=2769 W m^{-2} K^{-1} is significantly lower than the values obtained experimentally in [6].

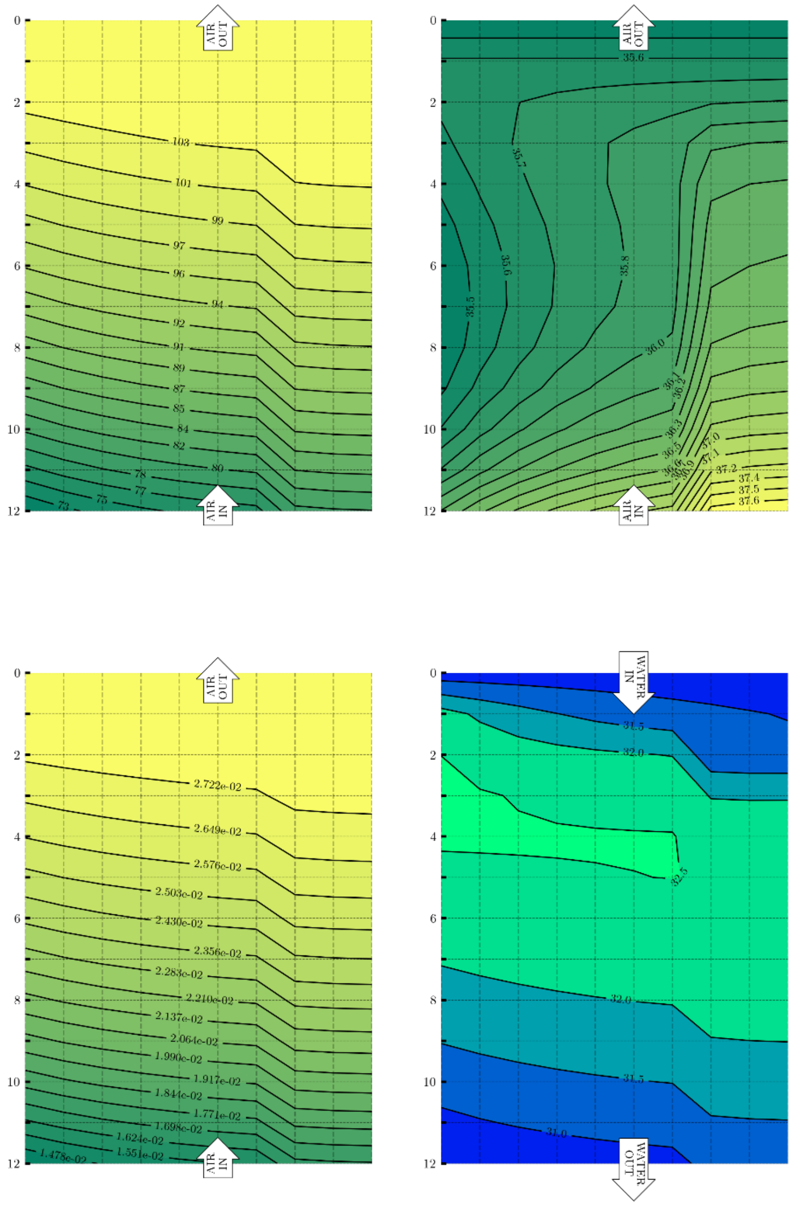


Figure 4. Air enthalpy (kJ·kg^{-1}) (top left), temperature (°C) (top right), humidity ratio (kg$_w$·kg$_a$$^{-1}$) (bottom left) and water temperature (°C) (bottom right) over the cooling coil, Poppe's model.

The results of the simulation with Poppe's model are shown in Fig. 4 (the temperature distribution for the refrigerant is omitted, since it is the same as Fig.3): in this case the temperature of moist air and the humidity ratio over the cooling coil are also obtained. The air enthalpy and water temperature follow a similar pattern to those found with Merkel's model. The water temperature in the sump is T_s=30.8 °C whilst Merkel's model would yield T_s=30.6 °C: even if the difference is small, it has nonetheless an influence on the cooling capability, which, in this case is P_c=1282.6 kW, that is, about 9% below the reference value. Poppe's model, although more complex and richer in output information, appears to perform slightly worse than Merkel's in predicting the cooling performance of the evaporative condenser. One element which deserves further analysis, though, is the actual value of the heat transfer coefficient in the film, which was obtained outside of the bounds for the correlation, and which might be somewhat higher: in this case, Merkel's model might overestimate the results and Poppe's prove to be more accurate.

5. Conclusions

Two different models to capture the transport phenomena occurring in evaporative condensers were described and implemented to check their capability to correctly predict the cooling performance of a piece of equipment as declared by the manufacturer. Both models were accurate within 10% and but the simpler (Merkel's) came closer, underestimating by little more than 2% the cooling power, whereas the more complex formulation (Poppe's) underestimated the quantity by about 9%. An element of uncertainty turned out to be the convective heat transfer coefficient in the film, for which no suitable correlation for the operating conditions of the problem was found. As the value extrapolated from Eq. (8) seems to be lower than those reported from experiments in [6], the matter deserves further investigation. To this aim, some field measurements on units installed in a food storage facility are planned.

The model of the evaporative condenser is thought as a block in a larger system which simulates the behaviour of a refrigeration loop for industrial applications, especially the savings in terms of energy which can be obtained by letting the condenser operate at temperatures closer to the wet bulb, especially on colder days. Although developed for steady state situations, the model can easily be adapted to transient conditions by introducing the storage term in the energy and mass balance equations.

References

[1] Lebedev V, *Refrigerating engineering*, 1986, (MIR Moscow, Russia)
[2] Bonauguri E, Miari D, *Tecnica del freddo* 1988, (Hoepli, Milan, IT)
[3] Merkel F, *1925 VDI-Zeitchrift* **70** pp 123–128
[4] Poppe M., and Rögener H., 1991 *VDI-Wärmeatlas*, pp Mi 1–Mi 15.
[5] Baker D R and Shryock H A, *1961 Journal of Heat Transfer*, **83**(3) pp 339–349
[6] Dreyer A A, *PhD thesis,* 1988 (Stellenbosch University, Stellenbosch, RSA).
[7] Erens P J, and Dreyer A A, *1989 N and O Journal*, **5**(1) pp 1–10
[8] Halasz B, *1998 Revue generale de thermique*, **37**(4) pp 245–255
[9] Hasan A A, and Siren K, *2002 Energy and buildings*, **34**(5) pp 477–486
[10] Hasan A A, *PhD thesis*, 2005 (Helsinki University of Technology, Helsinki, Finland)
[11] Kröger D G, *Air-cooled heat exchangers and cooling towers*, *v.1.* 2004 (Pennwell Books, Tulsa OK, USA).
[12] Kloppers J C, and Kröger D G, *2005 Int. journal of thermal sciences*, **44**(9) pp 879–884
[13] Facao J, and Oliveira A C, *2008 Num. Heat Transfer, Part A*: *Applications*, **54**(10) pp 933–944
[14] Yilmaz A., *2010 Journal of Thermal Science and Technology*, **30**(2) pp 45–56
[15] Giovannini M, Lorenzini M *2021 Proc. of ATE-Hefat 2021* (Amsterdam, The Netheralands).