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Review Article

Review on Mathematical Modelling of Evaporative Condenser for Refrigerant inside the tube.

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Abstract

Air-Conditioning plays an essential role in ensuring occupants thermal comfort. However, buildings electricity bills have become unaffordable. Yet the commercially used cooling systems are highly power-consuming, i.e. vapor compression systems. This paper aims to review the mathematical modelling of evaporative condenser for VCC (vapor compression cycle). Evaporative condenser helps to achieve removal high heat load from refrigerant that is coming from compressor. This paper will include new condensation correlation for local heat transfer coefficient. This paper aims to give all the mathematical correlation which will require to design the evaporative condenser for refrigerant inside the tube and water spray and air in counter flow direction.

Keywords: *Evaporative condenser, Refrigerant inside the tube banks, condensation inside the tubes.*

1. Introduction

The analysis of equipment's for fluid refrigerant condensation in large size refrigeration system are of great importance, since condenser with better performance can help with lower initial and operational cost.

The goal of this paper is to study heat and mass transfer in an evaporative condenser as well as to evaluate the relationship among some of the measured quantities.

The correct mathematical modelling offers great advantages when used for operation and performance analysis of equipment. Several work have been developed by many scientist for better understanding the heat and mass transfer phenomena that occurs in evaporative condensers.

Mathematical modelling of heat and mass transfer processes in evaporative condenser was a subject of numerous works. Among them the works of Parker and Treybal, in condensation inside the tubes the work of Thome-El Hajal-Cavallini should be mentioned.

Parker and Treybal first took into account the changes of the temperature of water which is spraying the tube surface. It was assumed that the amount of water into air could be neglected and that in the considered range of temperatures, the enthalpy of saturated air was a linear function of temperature.

Thome-El Hajal-Cavallini gives flow pattern based on intube condensation heat transfer model. Which can be used in the determination of heat transfer coefficient of tube side refrigerant which is phase changing.

Fig. 1 presents the scheme of evaporative condenser. Its essential component is a bare-tube pipe coil inside which a refrigerant flows downwards. The outer surface of the tube is sprayed with water in a closed cycle. Air flow in the direction opposite to the water flow is forced by a ventilator.

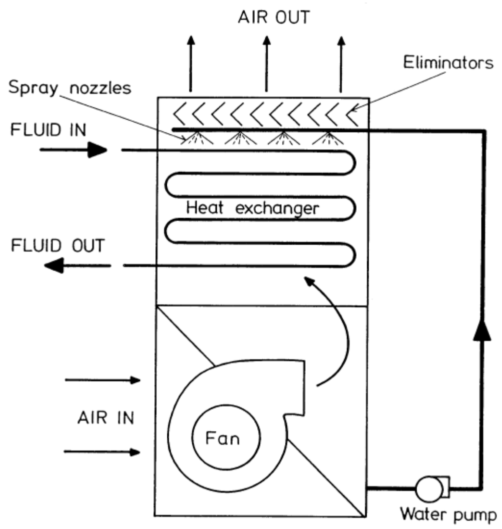

Fig.1 Schematic diagram of evaporative condenser

The heat exchange process that takes place in evaporative condenser is complex. Heat transfer from

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the refrigerant into the water sprayed in the outer coil surface and from the water into the air counter flowing through the condenser. Direct contact of air with water at simultaneous heat inflow results in the process of non-adiabatic water evaporation into air. As a result, the air carries away latent heat of water vapor that humidifies this air and, to a much smaller degree, non-latent heat penetrating from the water surface into air. The air temperature is of no special importance. It can be lower, equal or even higher than the temperature of water, depending on the operation conditions (compare the points 1, W and 1', W' in fig. 2).

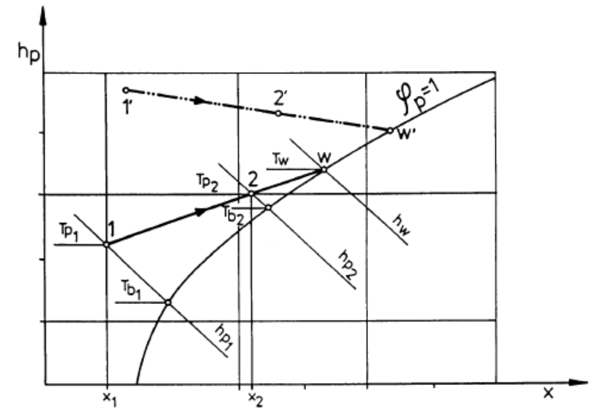

Fig. 2 Changes of air parameters; 1, 1'-intial state of air, 2, 2'-final state of air

The important factor which influences the heat exchange between water and air is the air enthalpy. To make the process possible it must be smaller than the enthalpy corresponding to the state of saturation at water temperature (points W and W'). The changes of air parameters in the exchanger are presented in the $(h_p - x)$ diagram (Fig. 2).

The mechanism of heat exchange in evaporative fluid cooler is connected with the heat exchange among three types of fluids: air, spraying water and cooled liquid. At constant air parameters the spraying water and cooled liquid. At constant air parameters the spraying water temperature stabilizes automatically at a level which depends on heat load. At constant heat load the temperature of water depends on the value of air parameters. The changes of the water temperature result from the tendency of the cooled-water-air system to reach the thermal equilibrium state.

2. Mathematical model of evaporative condenser

2.1 Assumptions

1. The heat and mass transfer processes are performed in steady state conditions, in the direction perpendicular to the wall.
2. Owing to the small temperature differences at which the process occurs heat transferred by radiation is not taken into account.
3. At the interfacial surface air reaches the temperature of water and its humidity corresponds to the state of equilibrium.
4. The resistance of the heat transfer from the water film core to its surface is neglected.
5. Water and air in counter flow.
6. Spraying water flows in a closed cycle.

2.2 Basic Equation of heat and mass transfer.

To evaluate the thermal capacity of an evaporative condenser, ambient heat transfer rate or rejected heat $\dot{q}$ (KW). Must be determined. This heat transfer rate can be calculated by two ways.

1st thermal balance involving the water spray stream and the air flow, as represented in Eq (1) (ASHRAE, 2005)

$$q = \dot{m}_{air}\left(h_{air,out} - h_{air,in}\right) - \dot{m}_{wmu}h_w \quad (1)$$

2nd one is establishing a thermal balance with the refrigerant and make up water flows (ANSI/ASHRAE, 1995)

$$\dot{q} = \dot{m}_r\left(h_{r,int} - h_{r,out}\right) - \dot{m}_{mwu}\left(h_w - h_{wmu}\right) \quad (2)$$

In this work, the overall heat transfer coefficient is determined for three heat transfer zones: dessuperheating, condensation and subcooling. In all zones, the heat transfer rate is determined by correlations. The dessuperheating is estimated at 1/6th of the total surface area, as well as for the subcooling heat transfer surface. Fig. 3 shows a scheme of refrigerant, spray water and air flows over an elementary cross section of the tube. For simplicity, the refrigerant and spray streams flow in the same sense.

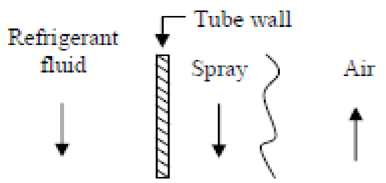

Fig. 3: Schematic water, refrigerant and air flow.

The overall heat transfer coefficient is calculated for the region depicted in this fig by equation (3) in respect to the outer diameter d_{ext} (ASHRAE, 2000).

$$U = \frac{1}{\frac{d_{ext}}{d_{int}}\left(\frac{1}{\alpha_{int}}\right) + \frac{d_{ext}}{d_{in}}\left(\frac{L}{k_r}\right) + \frac{1}{\alpha_{ext}}} \quad (3)$$

3. Refrigerant side heat transfer coefficient

The internal heat transfer coefficient can be evaluated as an average or local value. The local value must be determinate with the properties of the refrigerant fluid that are dependent mainly on the temperature of the single phase case, as well as in dessuperheating and sub cooling regions. On these two last regions, the

$$\alpha_f = 0.728\left[\frac{\rho_L(\rho_L-\rho_g)gh_{LG}K_L^3}{\mu_L d_{int}(T_{sat}-T_w)}\right] \quad (12)$$

internal heat transfer coefficient can be determine using Dittus-Boelt (1985) correlation that results in:

The exponent of the Prandtl number (n) is 0.3 for dessuperheating and subcooling zones.α_{int}

For the average heat transfer coefficient, the thermophysical properties in Eq. (4) are evaluated at the refrigerant mean temperature, taken at the inlet and outlet considered regions.

3.1 Condensation Heat Transfer model

The Thome-El Hajal-Cavallini flow pattern based in tube condensation heat transfer model is implemented

$$\varepsilon = \frac{\varepsilon_H - \varepsilon_r}{\ln(\varepsilon_H/\varepsilon_r)} \quad (5)$$

as follows:

1. Determine the local vapor void fraction using:
2. Determine the local flow pattern using the flow pattern map and an necessary transition velocities at the sum of X;
3. Identify the type of flow pattern (annular, intermittent, mist, stratified-wavy or

$$\alpha_c = cRe_L^n Pr_L^m \frac{k_L}{\delta} f_i \quad (6)$$

stratified);

$$\alpha(x) = \frac{\alpha_f\theta + (2\pi-\theta)\alpha_c}{2\pi} \quad (7)$$

4. If the flow is annular or intermittent or mist,

$$A_L = \frac{(2\pi-\theta)}{8}[d_{int}^2 - (d_{int}^2 - 2\delta)^2] \quad (8)$$

then $\theta = 0$ and α_c is determined with and

$$f_i = 1 + \left(\frac{u_g}{u_L}\right)^{1/2}\left(\frac{(\rho_{L}-\rho_G)g\delta^2}{\sigma}\right)^{1/4} \quad (9)$$

$$\theta_{strat} = 2\pi - 2\Big\{\pi(1-\varepsilon) + \left(\frac{3\pi}{2}\right)^{1/3}\Big[1-2(1-\varepsilon)+(1-\varepsilon)^{\frac{1}{3}} - \varepsilon^{1/3}\Big] - \frac{1}{200}(1-\varepsilon)\varepsilon[1-2(1-\varepsilon)][1+4((1-\varepsilon)^2+\varepsilon^2)]\Big\} \quad (10)$$

$$\theta = \theta_{strat}\left[\frac{\dot m_{wavy}-\dot m}{\dot m_{wavy}-\dot m_{strat}}\right]^{0.5} \quad (11)$$

$\alpha(x) = \alpha_c$ in
Where δ is obtained by solving
And f_i with

5. If the flow is stratified-wavy, θ_{strat} and θ calculated using

Then α_c can be calculated using (6)

$$Nu = \frac{\alpha_{int}d_{int}}{k} = 0.023Re^{4/5}Pr^n \quad (4)$$

and α_f Using,

And finally $\alpha(x)$ is determined using (7) where δ is obtained with (8) and f_i with (9).

6. If the flow is fully stratified, use equation (10) and θ_{strat} is set equal to θ, then α_c and α_f are calculated using (6) and (12) and $\alpha(x)$ is determined using (7) where δ is obtained with (8) and f_i is determined with (9).

4. Spray water heat transfer coefficient

The heat transfer coefficient between external tube surface and spray water was calculated from some well-known correlations.

Tovaras et.al.(1984) (appud Zalewski and Gryglaszwski, 1997) proposed a correlation for this heat transfer coefficient as a function of water Prandtl number (Pr_w) water and air Reynolds number (Re_w)and (Re_{air}). This correlation for water flowing

$$Nu_{ext} = 3.3\times10^{-3}Re_w^{0.3}Re_{air}^{0.15}Pr_w^{0.61} \quad (13)$$

downstram across the horizontal tubes has the

$$\mathrm{Nu_{ext}} = 1.1\times10^{-2}\mathrm{Re_w^{0.3}Pr_w^{0.62}} \quad (14)$$

following form.[4]

In the range Re_{air}= 690 to 3000:

$$Nu_{ext} = 0.24Re_w^{0.3}Re_{air}^{-0.36}Pr_w^{0.66} \quad (15)$$

For Re_{air} = 3000 to 6900:

For Re_{air}> 6900:

$$Re_{air} = \frac{w_o d_z \rho_p}{\mu_p} \quad (16)$$

The Reynolds and Nusselt numbers in the above

$$Re_w = \frac{4G}{\mu_w} \quad (17)$$

relations are defined in the following way:

$$Nu_w = \left(\frac{v_w^2}{g}\right)^{1/3}\frac{\alpha_{ext}}{\lambda_w} \quad (18)$$

The formulae are valid in the range: Pr_w = 4.3 to 11.3; Re_w = 160 to 1360.

5. Result and discussion

The heat transfer coefficients were calculated and compared with experimental result. The result obtained from Tovaras *et al.* is close to experimental result. Therefore we must use Tovaras *et al.* correlation for spray side heat transfer coefficient fig 4(a) and (b)

EL Hajal, Thome and Cavallini (2003) proposed a phenomenological condensation model based on local flow patterns and interfacial wave effects for condensation inside plain tubes for a very wide range of parameters: mass velocities from 16 to 1532 Kg/m^2s, tube internal diameters from 3.14 to 21.4 mm reduced pressure from 0.02 to 0.8 and vapor qualities from 0.03 to 0.97. El Hajal, Thome Cavallini model is used to predict local flow patterns in their heat transfer model. So far for heat transfer and flow patterns, the method has been compared with major 20 refrigerant. They showed not only that the heat transfer model was accurate statistically 85% of the eleven original refrigerant in their database from nine different labs, representing a total of 1850 data points, were predicted within ±20% (fig. (5)) but also it followed the trends within the database well with respect to vapor quality, tube diameter, mass velocity, reduced pressure, void fraction etc.

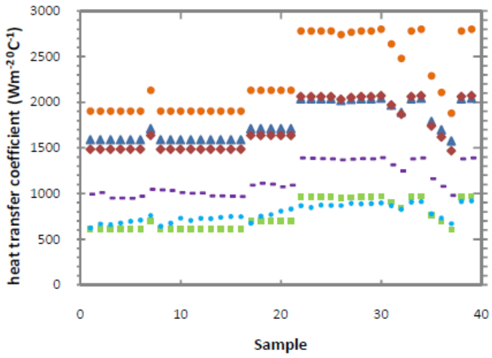

Fig.4 a) external heat transfer coefficient (h_{ext})

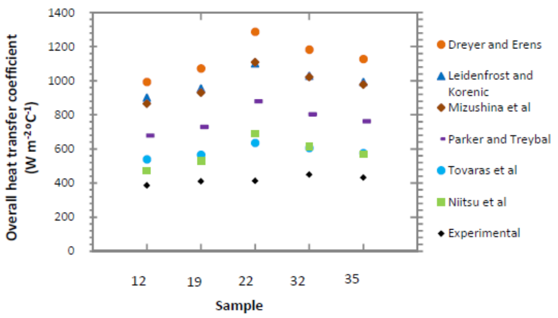

Fig. 4b) overall heat transfer coefficient (U)

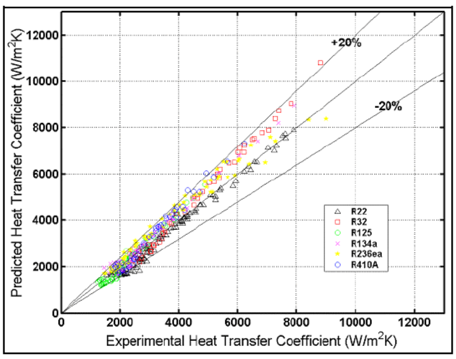

Fig.5 Comparison of the Thome-El Hajal-Cavallini model to the data of Cavallini *et al*. (1999, 2001) for six refrigerants in an 8 mm tube

Conclusion

Mathematical correlation represented in this paper can be used for design and develop an Evaporative Condenser with refrigerant in the tube. From the result it can be seen that this correlation closely simulate the heat transfer coefficient of Condensation process and heat transfer between water spray and air. Above correlation can be used to make a software for design and development of Evaporative Condenser with tube size of 3.14 to 21.4 mm.

Nomenclature

A_L	Cross section area occupied by liquid phase(m^2)
c	Empirical constant(0.003 from condensation database)
d_{ext}	External diameter of tube(m)
d_{int}	Internal diameter of tube(m)
f_i	Roughness correction factor
g	Acceleration due to gravity(m/s^2)
$h_{r,int}$	Refrigerant specific enthalpy at inlet(kJ/kg)
$h_{r,out}$	Refrigerant specific enthalpy at outlet(kJ/kg)
h_w	Sump water enthalpy(kJ/kg)
h_{LG}	Enthalpy of condensation(kJ/kg)
h_{wmu}	Specific enthalpy of make-up water(kJ/kg)
$k = k_L$	Refrigerant liquid thermal conductivity(w/mk)
k_r	Thermal conductivity of tube walls(w/m-k)
L	Length of the tubes(m)
$\dot{m}_{air}$	Mass flow rate of air(kg/s)
$\dot{m}_{mwu}$	Mass flow rate of make-up water (kg/s)
$\dot{m}_r$	Mass flow rate of refrigerant(kg/s)
$\dot{m}_{strat}$	Mass flow rate of stratified refrigerant(kg/s)
Nu_{ext}	Nusselt no at external side of tubes
Pr_L	Prandtl no of liquid refrigerant
$\dot{q}$	Heat rejected or ambient heat transfer rate (kW)
Re_w	Reynold no of spray water
Re_L	Reynold no of liquid refrigerant
Re_{air}	Reynold no of air
T_{sat}	Temperature of refrigerant film(k)
T_w	Temperature of tube wall(k)
U	Overall heat transfer coefficient(W/m^2k)
u_g	Velocity of gaseous refrigerant(m/s)
u_L	Velocity of liquid refrigerant(m/s)

Greek letters

α_c	Convective condensation heat transfer coefficient (W/m^2k)
α_{int}	Average heat transfer coefficient at refrigerant side(W/m^2k)
α_{ext}	Average heat transfer coefficient at water air side(W/m^2k)
α_f	Mean heat transfer coefficient of film(W/m^2k)
$\alpha(x)$	Local heat transfer coefficient at this vapor quality(W/m^2k)
δ	Liquid refrigerant film thickness(m)

ε	Logarithmic mean void fraction
ε_H	Horizontal void fraction
ε_r	Rouhani void fraction
θ	Stratified wavy angle
θ_{strat}	Stratified liquid angle
λ_w	Thermal conductivity of spraying water(w/m-k)
μ_L	Dynamic viscosity of liquid refrigerant(kg/m-s)
ρ_G	Density of gaseous refrigerant(kg/m^3)
ρ_L	Density of liquid refrigerant
σ	Viscous shear stress
υ_w	Velocity of spray water

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