PERFORMANCE PREDICTION ON THE FINNED-TUBE DRY COOLER IN WINTER PERIOD

Bing Yu[1,2,3], Xichun Wang[1], Peter Luscuere[2,3]
[1]Royal Haskoning Consulting (Shanghai) Co. Ltd, Shanghai, China
[2]Division Building Services, Haskoning Nederland BV, Nijmegen, the Netherlands
[3]Faculty of Environmental Science and Engineering, Tianjin University, Tianjin, China

ABSTRACT

A mathematical model is presented to predict the behaviour of condensation and frost formation on heat transfer surfaces by simultaneously considering the condensing and frost layer. The model employs one-dimensional transient formulation based upon a local averaging technique, taking into account the variation of the frost density and thickness. The presented model is validated by comparing to experimental data provided by the dry cooler manufacturer. It is found that the model can predict the heat transfer performance of the dry cooler with an accuracy of within 2.19%. The dry cooler's heat transfer performance are predicted at different air and water flow rates operating with water as heat transfer fluid during the winter period. The model and discussion are helpful for the operation of such a kind of dry cooler with water during the winter period.

INTRODUCTION

Frost phenomena are encountered in numerous fields of industry dealing with low temperatures, including refrigerators, air conditions, gas coolers and cold stores. In air-cooled heat pumps, frost phenomenon on heat transfer surfaces of fined-tube heat exchangers is a common problem. The heat resistance between the heat transfer surface and air flow is increased and the air flow rate through the heat exchanger is decreased due to the frost on the heat transfer surface, all of which unfortunately degrade the performance of the heat exchanger (Yu et al., 1998, Yu et al., 1998 and Yu and Feng, 2000). Furthermore: due to the frost on the surface of the heat exchanger, the size of the refrigeration unit used for industry is 50% larger in size and 25% higher in energy consumption, compared with the unit without frost (O'Neal and Tree, 1985).

For dry cooling operation with water as heat transfer fluid during the winter period, freezing is a problem. The viscosity of water also increases significantly with decreasing temperature, which results in higher energy consumption of the pump system. In addition, the phenomenon of pipe breaking occurs with ice formation in the pipe.

For these reasons, it is obviously necessary to use a numerical model to predict accurately heat transfer characteristics of the finned-tube heat exchanger under frost and frozen conditions.

According to the previous studies about the frost formation on the surface of the heat exchanger, most researchers have developed a numerical model based on the lump parameter method assuming ice formation on the heat transfer surface to be uniform and measured heat transfer characteristics of the typical heat exchanger during the ice formation process (O'Neal and Tree, 1985, Padki et al., 1989). However, an accurate model of dynamic frost formation on the heat transfer surface has not been integrated into the heat exchanger model in this paper.

In this paper, a mathematical model has been developed to predict the behaviour of condensation and frost formation on the heat transfer surface. The model concerns the dynamic formation of frost and condensing. Therefore, it has a high accuracy in the prediction of both the frost formation on one side and freezing phenomena on the waterside. In addition, the optimal control strategy has been developed to run the dry cooler with water as heat transfer fluid without freezing phenomena in the winter period.

MATHEMATICAL MODEL

The freezing process on the heat exchanger surface involves simultaneous heat and mass transfer. The following assumptions have been made in the development of the model:

1) The freezing process can be considered as a quasi-steady-state process, thus in a time step, the air flow is stable and the properties of air and the frost are constants;
2) Since heat resistances of the fins and tube is much lower than the heat resistances of the frost layer and the air flow, thermal conductivity of fins and tube are neglected;
3) The radiation between the heat transfer surface and air is negligible.

Numerical analysis is performed to predict the characteristics and frost layer growth of the heat and mass transfer in the configuration of Figure 1.

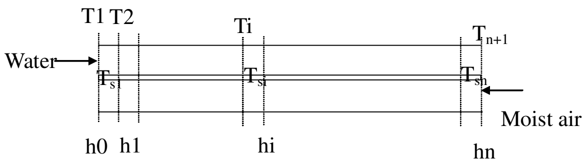

Figure 1 Calculation element of the heat exchanger surface

For the dry surface on the air side when humidity based on the fin surface temperature is higher than the air humidity ($x(T_{si}) \geq x_i$), the heat and mass equations can be expressed as :

$$m_a C_{p,a}(T_{a,i} - T_{a,i-1}) = \alpha_a A_a (T_{a,i} - T_{s,i}) \quad (1)$$

$$x_{a,i} - x_{a,i-1} = 0 \quad (2)$$

$$\alpha_a A_a (T_{a,i} - T_{s,i}) = \alpha_w A_w (T_{s,i} - T_{w,i}) \quad (3)$$

When humidity, based on the fin surface temperature is lower than the air humidity ($x(T_{si}) < x_i$), the heat and mass equations, for the wet or frost surface on the side can be expressed as:

$$m_a (h_{a,i} - h_{a,i+1}) = \alpha_a A_a (T_{a,i} - T_{s,i}) + m_a (x_{a,i} - x_{a,i+1}) c_1 \quad (4)$$

$$m_a (x_{a,i} - x_{a,i+1}) = \frac{\alpha_a}{C_{p,a}} A_a (x_{a,i} - x_{si}) \quad (5)$$

$$\alpha_w A_w (T_{si} - T_{w,i}) = \frac{\alpha_a}{C_{p,a}} A_a (h_{a,i} - h_{si}) \quad (6)$$

The heat transfer on the water side can be expressed as:

$$m_w C_{p,w}(T_{w,i} - T_{w,i-1}) = \alpha_w A_w (T_{w,i} - T_{si}) \quad (7)$$

The enthalpy of humid air can be express as:

$$h_{a,i} = C_p (T_{a,i} - 273) + (2500 + 1.84(T_{a,i} - 273)) x_{a,i} \quad (8)$$

Considering the item of $1.84T_{a,i}$ is much less than 2500, it is neglected in the model, thus the enthalpy difference between neighboring elements can be expressed as:

$$h_{a,i} - h_{a,i+1} = C_{p,a}(T_{a,i} - T_{a,i+1}) + C_1 (x_{a,i} - x_{a,i+1}) \quad (9)$$

C_1 is the latent heat and is 0, 2500 and 2835 (dry, wet and frost).

By arranging the above equations for three different conditions (dry, wet and frost) and four local variables, the problem can be solved.

For dry surface conditions ($x_{a,i} = const$, $(i = 1 \ldots n)$), three sets of equations (Equations 1, 3 and 7) are sufficient to solve the problem:

$$(\alpha_w A_w - m_w C_{p,w}) T_{w,i} + m_w C_{p,w} T_{w,i+1} - \alpha_w A_w T_{s,i} = 0 \quad (10)$$

$$(\alpha_a A_a - m_a C_{p,a}) T_{a,i} + m_a C_{p,a} T_{a,i-1} - \alpha_a A_a T_{s,i} = 0 \quad (11)$$

$$-\alpha_w A_w T_{w,i} - \alpha_a A_a T_{a,i} + \left(\alpha_w A_w + \alpha_a A_a\right) T_{s,i} = 0 \quad (12)$$

For wet and frost surface conditions ($x(T_{si}) \leq x_{a,i}$), four sets of equations with four sets of variables and more general equations with c1 (c1=0, 2500, 2835 ~ dry, wet, frost) and c0=2500 for enthalpy calculations are used. The equations used for the calculation are arranged as:

$$\left(\alpha_a A_a - m_a C_{p,a}\right) T_{a,i} + m_a C_{p,a} T_{a,i+1} - \alpha_a A_a T_{s,i} + \left(\frac{\alpha_a A_a C_1}{C_{p,a}} - m_a C_1\right) x_{a,i} + m_a C_1 x_{a,i+1} = \frac{\alpha_a A_a C_1}{C_{p,a}} x\left(T_{s,i}\right) \quad (13)$$

$$\left(\alpha_w C_{p,w} - m_w C_{p,w}\right) T_{w,i} + m_w C_{p,w} T_{w,i-1} - \alpha_w A_w T_{s,i} = 0 \quad (14)$$

$$-\alpha_a A_a T_{a,i} - \alpha_w A_w T_{w,i} + \left(\alpha_w A_w + \alpha_a A_a\right) T_{s,i} - \frac{\alpha_a A_a}{C_{p,a}} C_1 x_{a,i} = -\frac{\alpha_a A_a}{C_{p,a}} C_1 x\left(T_{s,i}\right) \quad (15)$$

$$\left(\alpha_a A_a - m_a C_{p,a}\right) x_{a,i} + m_a C_{p,a} x_{a,i+1} = \alpha_a A_a x\left(T_{s,i}\right) \quad (16)$$

The heat transfer coefficient for the air side of the heat exchanger can be calculated by the correlation equation proposed by Vampola (1966):

$$Nu = \frac{h d_b}{k} = 0.251 \mathrm{Re}^{0.67} \left(\frac{P_t - d_b}{d_b}\right)^{-0.2} \times \left(\frac{P_t - d_b}{P_d - \delta_f} + 1\right)^{-0.2} \left(\frac{P_t - d_b}{p_d - d_b}\right)^{0.4} \quad (17)$$

Where d_b=external diameter of the tube, P_d=diagonal pitch, P_t=Transverse pitch, δ_f =fin thickness

The heat transfer coefficient for the water side of the dry cooler can be calculated by the following equations.

For turbulent and transition water flows (Re=2300~5×10^6), heat transfer coefficient can be calculated by the correlation equation proposed by Gnielinski (1976)

$$Nu = \frac{hd_b}{k} = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3}-1\right)} \tag{18}$$

where
$$f = \frac{1}{\left(1.82\log_{10}\mathrm{Re}-1.64\right)^2}$$

For laminar water flow (Re ≤ 2300) a uniform heat flux and heat transfer coefficient can be calculated by (Frank and David, 1996)

$$Nu = \frac{hd_b}{k} = 4.36 \tag{19}$$

There are many factors, such as air velocity, surface temperature, air temperature and humidity, which influence the density of the frost on the heat transfer surface of the heat exchanger. In this paper, the correlation equation proposed by Bing (1996) is used to calculate the density variations of the frost on the heat transfer surface; it is expressed as the function of the heat exchanger surface temperature:

$$\rho_{fr} = 650\exp\left(0.227t_s\right) \tag{20}$$

Thermal conductivity of the frost can be expressed as a function of the density as follows:

$$k_{fr} = 0.001202\rho_{fr}^{\ 0.963} \tag{21}$$

In the dynamic process of freezing, the heat exchanger surface temperature is a function of the time, therefore, the density of the frost changes at different periods. The thickness of the frost layer for each time interval on the heat exchanger surface can be calculated as follows:

$$\Delta\delta_{fr} = \frac{w\Delta\tau}{\rho_{fr}A_e} \tag{22}$$

MODEL VALIDATION

In order to exam the validity of the calculation model described in the section of the mathematic model, numerical simulations were implemented with the properties and conditions of the dry cooler provided by the AIA and the simulation results are then compared to the experimental results.

The front and side views of the dry cooler used for the simulation are shown as follows.

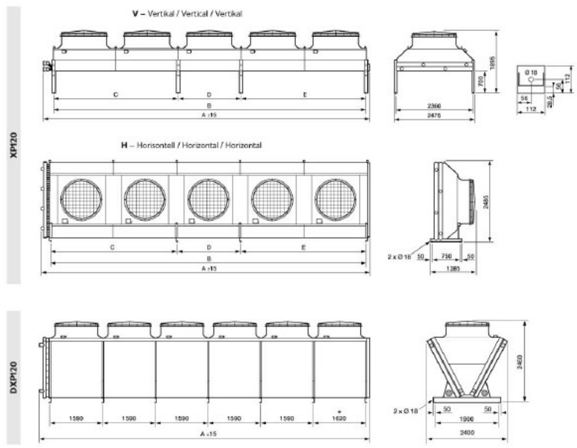

Figure 2 Front and side views of the dry cooler

The dry cooler is actually a kind of finned-tube heat exchanger with water flowing inside the tubes and air flowing around the outside of the tubes and fins. The tubes and fins are made from copper and aluminum, respectively. The detailed configuration of the fins and tubes are shown as follows:

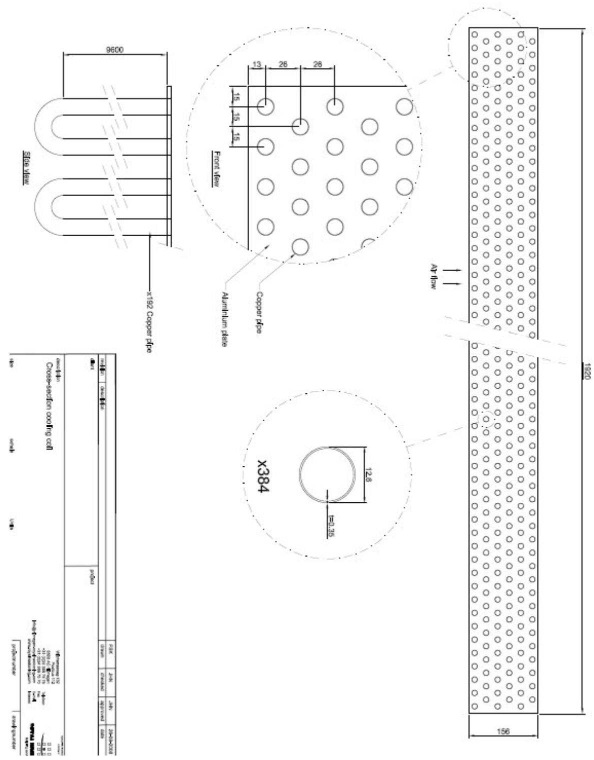

Figure 3 Detailed configuration of fins and tubes of the dry cooler

The dimension of the air cooler used for simulation is showed in Table 1 as follows.

Table 1 Dimension of air cooler

DRY COOLER DIMENSION	DIMENSION
Fin type	30×26
Fin material	Epoxy@2.5 mm
Fin Thickness	0.13 mm
Fin width	9600 mm
Fin length	1920 mm
Total loop	192
Heat transfer area at water side	1075.6 m^3
External diameter of the tube	12.6 mm
Thickness of the tube	0.35 mm
Total heat transfer area at water side	2075.6 m^3

For the measurements implemented by the manufactures, a water-glycol solution is used as heat transfer fluid in the tubes. The heat transfer fluid properties and the experimental results of the air cooler are shown in Table 2.

Table 2 Properties and boundary conditions of water-glycol and air

AIR	PROPERTIES & BOUNDARY
Air flow rate	47.4 m^3/s
Air velocity	2.57 m/s
Design air inlet temperature	35 °C
Design air outlet temperature	46 °C
Air density	1,146 kg/ m^3
Relative humidity	60%
GLYCOL-WATER MIXTURE	
Flow rate	25.57 l/s
Flow velocity	1.27 m/s
Density	1025.6 kg/ m^3
Specific heat	3817.8 J/kg.K
Thermal conductivity	0.488 W/m.K
Design inlet temperature	50 °C
Design outlet temperature	44 °C
Total capacity	600.6 kW

Based on the simulation model, the outlet temperatures of both air and glycol-water solution can be calculated. The simulated results and the measured data are compared in Table 3.

Table 3 Outlet temperature of air and glycol-water at design condition

OUTLET TEMP (°C)	MEASURED	SIMULATED	DEV
T_{glycol_water} (°C)	44	43.87	-0.13°C
T_{air} (°C)	46	45.87	-0.13°C
Total Capacity (kW)	600.6	613.7	2.19%

As can be seen from Table 3, measured and simulated outlet temperatures of the dry cooler are matched very well. On the other hand, the deviation between the measured and simulated heat transfer capacity of the heat exchanger is 2.19%, this is mainly due to the neglected heat transfer resistance of the tubes and fins of the dry cooler.

CASE STUDIES

A new KPN datacenter is going to be constructed in Arhnem, the Netherlands. Dry coolers, as shown in Figure 2, are used to produce cooling water to remove the heat generated by the computers of the datacenter during the winter period. It is planned to use water as heat transfer fluid in the dry cooler due to the lower viscosity compared with a water-glycol solution or glycol solution. However, there will be a risk of freezing of water during the winter period, so the heat transfer characteristics should be predicted before implementation of the project.

According to the design schedule, the cooling demand will be implemented in three construction phases:

- Phase 1: 0 ~ 1100kW of cooling demand, with 6 heat exchangers;
- Phase 2: 1101 ~ 2200kW of cooling demand, with 8 heat exchangers;
- Phase 3:: 2201 ~ 3300kW of cooling demand, with 10 heat exchangers.

In the first stage, almost 1100kW of cooling demand is required. Divided by 6 heat exchangers, this accounts for 183.33kW per unit.

Based on the different extreme operation conditions, the following cases will be checked:

1. Water supply & return temperature of 18/10°C and ambient air temperature -15°C and 50% RH
2. Water supply & return temperature of 16/10°C and ambient air temperature -15°C and 50% RH
3. Water supply & return temperature of 24/10°C and ambient air temperature -15°C and 50% RH
4. Water supply & return temperature of 16/10°C and ambient air temperature -20°C and 50% RH
5. Water supply & return temperature of 16/10°C and ambient air temperature -15°C and 90% RH

The critical situation is the period when Phase 2 starts but cooling demand is still 1101kW, i.e. 137.5kW per unit, since there is a high possibility of freezing of water during the winter period. The heat transfer behaviour of water in the dry cooler should be examined at such critical situations.

Required water flow condition for five different cases

When Phase 2 starts but cooling demand is 1101kW, the cooling capacity for each exchanger is 137.5kW. Based on a cooling capacity of 137.5kW for each exchanger and the different extreme operations of the dry coolers, the water flow rates and flow conditions for five different cases can be obtained as follows:

Table 4 Flow conditions of the five cases

CASES	$T_{w,in}$ (^{O}C)	$T_{w,out}$ (^{O}C)	$T_{a,\ in}$ (^{O}C)	$RH_{s,out}$ (%)	M_{water} (l/s)	Re_{water}
Case 1	18	10	-15	50	4.11	2046
Case 2	16	10	-15	50	5.47	2620
Case 3	24	16	-15	50	4.11	2419
Case 4	16	10	-20	50	5.47	2620
Case 5	16	10	-15	90	5.47	2620

As can be seen from Table 4, when cooling demand for each heat exchanger is set to 137.5kW, the water flow in Case 1 shall stay laminar whereas the water flow in Cases 2, 3, 4 and 5 will be in transition from laminar to fully turbulent.

Keeping the water flow at required values as shown in Table 4 for five cases and tuning the air flow rates to meet the required cooling capacity of 137.5kW of each heat exchanger, the required air flow rates and outlet temperatures of air for five cases can then be obtained as follows:

Table 5 Air flow rates and outlet temperatures for five cases

CASES	Q_{air} (m^3/s)	$T_{air,out}$ (^{o}C)
Case 1	4.22	14.25
Case 2	4.37	15.09
Case 3	3.28	22.87
Case 4	3.66	15.18
Case 5	4.34	15.09

As can be seen from Table 5, the cooling capacity of the heat exchanger can also be controlled for five cases by keeping the water flow rate at constant values as shown in Table 4 and tuning the air flow rates. The air outlet temperatures for all five cases are higher than the dew point temperature. Therefore, there are no freezing phenomena on the surface of the heat exchangers. However, the minimum controllable fan air flow rate is 9.48m^3/s (V_{air}=0.514m/s), which is higher than the required air flow rate as shown in Table 5. From a system control point of view, it is not feasible to realize such a control strategy.

Risk analysis

As discussed in the above section, it is not possible to keep the variable speed fan at the controllable range whith the control strategy of keeping a constant water flow rate whilst tuning the air temperature. In order to avoid the risk of freezing of the water in the pipes and keep the air at the controllable flow range, it is necessary to find the lowest outside environmental temperature allowed for the proper operation of the dry coolers. The calculation routine to find the lowest outside temperature allowed is described as follows:

1. Keep the air flow velocity at the minimal valid value (*Vair*=0.514m/s).
2. Calculate the required water flow rate, based on: an outlet water temperature of 3^{o}C to avoid freezing, required inlet water temperatures and a cooling capacity of 137.5kW.
3. Find the lowest outside environmental temperature allowed for, using the simulation program.

The lowest outside temperatures allowed for the operation of the fan at the controllable range for five cases are obtained as follows:

Table 6 The lowest outside temperatures allowed for running the fan

CASES	$T_{air,\ lowest\ allowed}$ (^{O}C)	Q_{water} (l/s)	$T_{water,out}$(^{O}C)
Case 1	-4	2.19	3
Case 2	-5	2.48	3
Case 3	-1.5	1.57	3
Case 4	-5	2.48	3
Case 5	-5	2.48	3

Based on the lowest outside temperature for each case, it is possible to define the control scheme for a proper operation of the dry cooler. Taking case 1 for example, when the outside temperature is lower than -4^{o}C, the control system shall give a warning or shut down the fan in order to avoid freezing of water in the pipes.

However, we still have the risk of freezing of water in the pipes when the fan of the dry cooler is switched off. In this case, the air flow in the dry cooler takes the form of natural ventilation. The calculation results for outlet temperatures for five cases when the fan is switched off are shown as follows:

Table 7 Predicted results for an inoperative fan

CASES	$T_{water,out}$ (PREDICTED WATER TEMPERATURE)	Q_{water} (l/s) (MINIMAL WATER FLOW RATE)
Case 1	7.4176	4.11
Case 2	6.3795	5.47
Case 3	10.745	4.11
Case 4	4.6938	5.47
Case 5	6.3795	5.47

It is found that the predicted outlet temperatures of water from the dry cooler for all five cases whith water at minimal flow rates (see Table 4) are higher than 3°C. Therefore, there is no risk of freezing of water in the pipes in case of an inoperative fan. At the same time, it is also found that there is no frosting or condensation of the air on the surface of the dry cooler for all five cases.

Suitable control strategy

From the above discussion, it is possible to develop a feasible control strategy for the proper operation of the dry cooler during the winter period. The water outlet temperature of the dry cooler is taken as the setting value to control the air flow rate and water flow rate. On the other hand, the transition of the water flow from turbulent to laminar significantly degrades the heat transfer coefficient, which can be used to reduce the cooling capacity as to avoid freezing of the water in the dry cooler when the required cooling capacity is reduced. An example of a control strategy of the dry cooler is showed as follows (Figure 4):

Initially, assuming the water flow rate at a minimum of 4.11 l/s and fans switched off.

When the water outlet temperature of the dry cooler is higher than 10°C, the fan is switched on and is set to the minimum flow rate of $9.48 m^3/s$ (0.514m/s), if the water outlet temperature is still higher than 10°C, the fan speed is continuously increased until fulfillment of the set point temperature of 10°C.

When the fan is at full speed ($Q_{air} = 47.4 m^3/s$) and the set point temperature of 10°C cannot be fulfilled, the pump speed is increased to fulfill the set point of 10°C.

When cooling demand of the dry cooler is decreased, the outlet temperature will also decrease. Taking the water outlet temperature of 5°C as the set point temperature in this case, the fan is switched off and water flow rate is decreased sharply to minimum value of 4.11 l/s when the water outlet temperature is below 5°C.

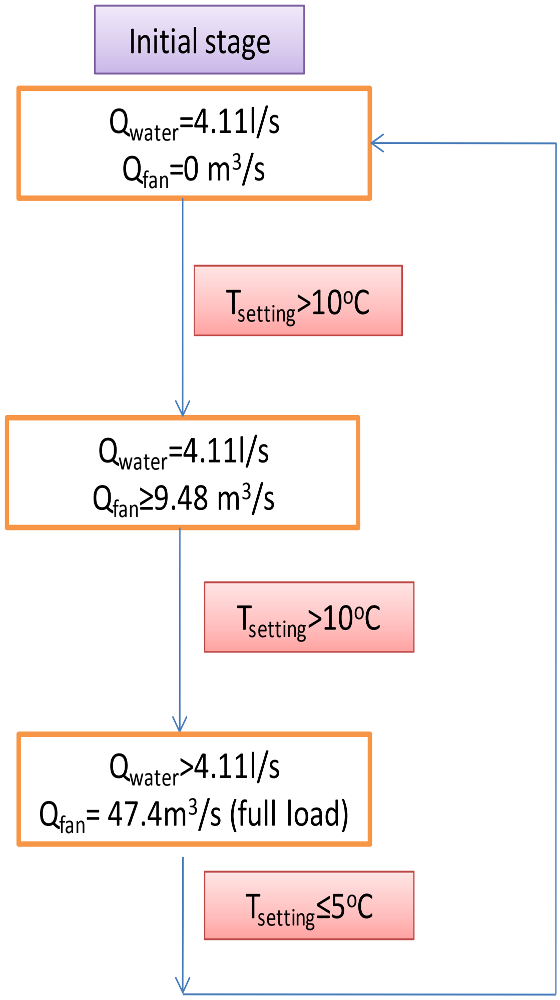

Figure 4. A suitable control strategy

CONCLUSIONS

A mathematical model, which can predict the behaviour of condensation and frost formation on the heat transfer surfaces has been developed by simultaneously considering condensation and frost layer,. The model for the prediction of the performance of the heat exchanger can be used to accurately predict the heat transfer behaviour of the dry cooler operated in a cold winter period. The control strategy of keeping water at required values based on the cooling capacity and inlet and outlet water temperature difference and tuning the air flow rate has been proved to be not feasible due to the limitation of the minimum controllable flow rate of the variable speed fan. The control strategy of keeping the air flow rate at a minimum controllable value of the fan whilst tuning the water flow rate to obtain a required cooling capacity has been proved to be limited by the outside air temperature. In addition, there is no condensation or frosting phenomena found on the heat transfer surface of the dry coolers for all five cases, since the air temperature flowing through the dry cooler is always keeping a temperature above the dew point temperature of the air. It is feasible to use water as a heat transfer medium in the dry cooler during extremely cold winter periods if the proper control strategy is applied.

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