INVESTIGATION OF THE THERMAL PERFORMANCE OF A CROSS-FLOW WATER COOLING TOWER WITH DIFFERENT PACKINGS

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ABSTRACT

The present research involves experimental and theoretical work to study the performance of three kinds of packing (PVC, corrugated and flat plate asbestos) in a cross-flow water cooling tower under different atmospheric conditions and then comparing the performance of them with each other. The experimental work included the design, constructions and installation of a cross-flow cooling tower test rig suitable for measuring the individual coefficients (k_Ga) and (k'a). The experimental results were found by varying the inlet air flow rate and inlet water flow rate which are used as an input data to the computer program for finding the available performance coefficient (NTU) using the method of Webb with some modifications. Least square method was then used to correlate the experimental results of (NTU) in terms of water to air ratio (L/G).

الخلاصة

يتضمن البحث الحالي دراسة نظرية وعملية لاداء ثلاث انواع من الحشوات (بلاستيكية PVC ، الواح اسبست متموجة و مستوية) في برج تبريد متعامد الجريان تحت ظروف جوية متغيرة و ظروف تشغيلية مختلفة، ثم مقارنة اداء هذه الحشوات مع بعضها. و قد تطلب العمل تهيئة منظومة برج التبريد و حساب معامل انتقال المادة و الحرارة للبرج بتحليل القياسات العملية. و قد تم في التجارب تغيير كمية الهواء و الماء الداخل و التي اصبحت بيانات دخول الى برنامج حسابي للحصول على معامل الاداء باستعمال طريقة الباحث (Webb) مع اجراء بعض التحويرات. و قد استعملت طريقة المربعات الصغرى للحصول على علاقات بين معامل الاداء و نسبة معدل جريان الماء الى الهواء.

Key words: Cooling Tower, Cross-Flow, Thermal Performance, Different Packings

INTRODUCTION

Probably the most important device utilizing direct contact between water and atmospheric air is the cooling tower. The objective of cooling tower is not the processing of the air, but cooling of the spray water. Cooling towers may be used thermally to reclaim circulating water for re-use in refrigerant condensers, power plant condensers and other heat exchangers.

There are two basic cooling tower heat transfer packing or fill arrangements, namely, counter-flow and cross-flow. In the counter-flow cooling towers, the air is directed to flow vertically upwards against the descending flow of water. In the cross-flow arrangement, the air flow is directed in a perpendicular direction to the flow of water, as shown in **Fig (1)**. For both packing configuration, drift eliminators are placed beyond the packing to prevent loss of water droplets into the leaving air stream.

The advantages and disadvantages of the counter-flow cooling towers:

<u>Advantages</u>:

1. It has the most efficient means for heat transfer, since the coolest water contacts the coolest air.

2. Close control of cold water temperature.

3. Small ground area in which it can be operated.

4. Close approach and long cooling range possible.

<u>Disadvantages</u>:

1. Restricted louver area at the base with high velocity of inlet air increases the fan horsepower.

2. Resistance of upward air travel against the falling water, results in higher static pressure loss and a greater fan horsepower than with air/ water flow in cross-flow towers.

3. Uneven distribution of air velocities through the packing with very little movement near the walls and center of the tower.

4. High pumping head necessary because of the tower height and nozzle pressure require the packing to be placed high up in the tower because the intakes at the base must be unobstructed.

5. The hot water system is inaccessible for ready maintenance, because the water spray system is sandwiched between the top of the fill and the drift eliminators.

The advantages and disadvantages of the cross-flow cooling towers:

Advantages:

1. Low pumping head.

2. Low static pressure drop at the air side.

3. Low fan horsepower.

4. Convenient arrangement of the distribution system requiring only (6-8in) depth of water over the top of the tower.

5. It is possible to clean the distribution system while the tower is in operation.

6. Higher water loadings are possible for a given height.

Disadvantages:

1. A substantial cross-flow correction factor needs to be applied to the driving force, particularly where long range and close approach performance are required.

2. A cross-flow tower may need more ground area and more packing or fill material to transfer the same amount of heat, because the performance calculated has a greater NTU and this has been interpreted to indicate the cross-flow tower will have a larger required coefficient so must be physically larger to meat this condition.

3. In a cross-flow cooling tower the water temperature decreases vertically while the air enthalpy increases horizontally, requiring a double integration.

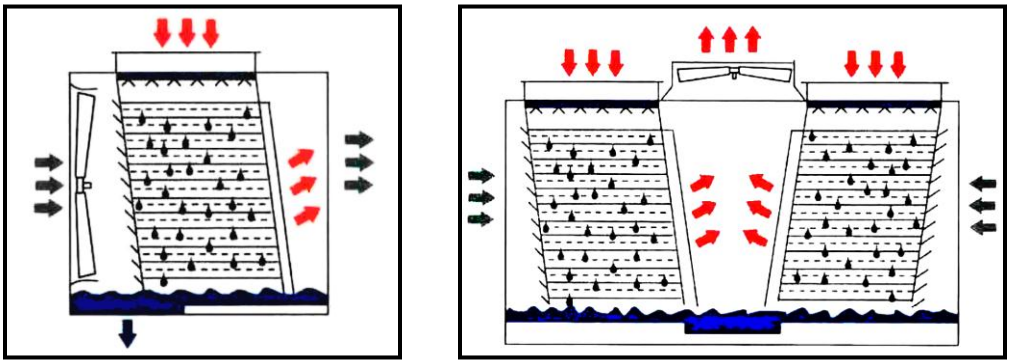


Fig (1) : Cross-Flow Cooling Tower

Merkel developed the first practical use of the differential equations in 1925. He combined the equations for heat and water vapor transfer and used enthalpy as the driving force to allow for both sensible and latent heat transfers. Heat is removed from the water by a transfer of sensible heat due to a difference in temperature levels and by the latent heat equivalent of the mass transfer resulting from the evaporation of a portion of the circulating water. Merkel's method has been the basis of most cooling tower analysis. His analysis is based on the assumptions that the water evaporation loss in the energy balance equation is negligible and that the Lewis number for air/ water vapor system is unity.

(Molyneux 1967)theoretical and experimental work to determine the difference between counter-flow and cross-flow cooling towers. He concluded that for the large industrial units with relatively small ranges, an approach of less than ($15^{\circ}F$) and operating with large flows of cooled water at (L/G) ratios greater than (1), the double induced draught cross-flow cooling tower can give economic advantages and the capital cost need to be no higher than that of the counter-flow tower having the same duty, but there can be significant saving in air and water pumping cost.

(Sutherland 1983) covered a wide range of air conditioning applications of cooling tower and should be of real value for design purposes. The accurate thermodynamic predictions apply generally and not to one particular tower and packings. His analysis did not utilize the assumptions of Merkel.

(Webb and Villacres 1984) presented algorithms for the performance simulation of evaporative cooling equipments. The algorithm performs a rating calculation for a given cooling tower fluid cooler or evaporative condenser. The most common rating calculation seeks the heat duty of the evaporative heat-exchanger for a specified operating condition (fluid flow rates, inlet process conditions and ambient air wet-bulb temperature).

Two types of packings were used in the experimental research work. The first is made of PVC sheets which are compressed thermally and having corrugations and extensions to increase the wetted surface area and to increase the spraying of the water. The second type is manufactured from asbestos cement, either as a flat plate or corrugated sheets.

CROSS-FLOW COOLING TOWER THEORY

The present analysis is based on the theory proposed by **(Baker and Shryock 1961)**, which is an extension and development of the original theory of cooling tower by **(Merkel 1925)**.

Both mass transfer (water evaporating from the droplets falling through the tower) and heat transfer (cooling of the water due to evaporation) are incorporated in the single coefficient ($K'aV/L$). The driving force for both is represented by the enthalpy potential difference between the air at the local wet-bulb temperature (h_a) and air at the local water temperature (h').

As shown in **Fig (2)**, water falls through the tower and is cooled by sensible heat loss and latent heat loss due to evaporation. The air is drawn across the tower by the fan, picking up sensible heat and water vapor in the process. Starting at the top left corner where the water temperature and air enthalpy are known, calculating the air enthalpy horizontally and water temperature vertically, as represented by the equations:

$$K'aV / L = \int_{t_1}^{t_2} C_{pw} \frac{dt}{h' - h_a} \qquad (1)$$

and

$$K'aV / G = \int_{h_{a1}}^{h_{a2}} \frac{dh_a}{h' - h_a} \qquad (2)$$

Numerically, the tower volume is divided into incremental volumes each of which accomplishes an identical fraction of the total transfer duty. The typical increment framed in **Fig (2)** and **Fig (3)**. The air enters at the left of the increment, its enthalpy increases as water vapor is added, and it exits to the right. The water is cooled from (t_1) to (t_2) due to the evaporation. After all increments have been traversed (M=10 is the number of increments), the calculation is terminated by averaging the temperature of the water leaving the bottom group of increments to yield the water outlet temperature.

A computer program was prepared based on Webb's method with some modifications to calculate the required conditions such as, temperatures and enthalpies for the bulk water, bulk air and interface for all the increments of the tower.

The algorithm for performance simulations of cross-flow cooling tower in the computer program consists of two iterations (inner and outer), starting from integrating **eq. (2)**, which gives:

$$K'aV / G = \frac{\Delta h_a}{h' - h_a} = (NTU)_G \qquad (3)$$

The above equation is applied to each incremental unit volume. The inlet and outlet water and air properties in the incremental volume are symbolized as shown in Fig (3). The enthalpy of air entering to each increment of the first column is ($h_W= h_a$) and the temperature of water entering to each increment of the first row is ($t_N=t_1$).At the bottom of grid we found the water outlet temperature of the last vertical increment (M) and for each of the horizontal increments (i= 1, 2, 3, …, M) or t_2 (M,i) from the equation :

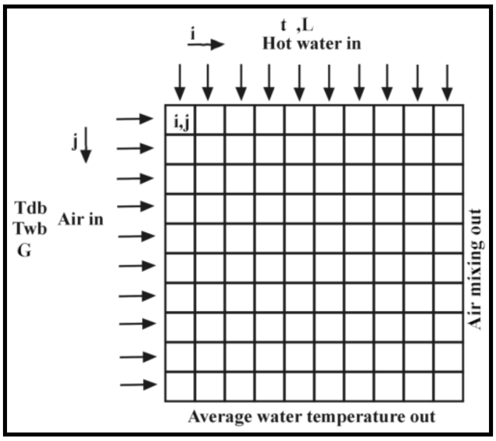
$$t_2 = \frac{1}{M}\sum_{i=1}^{M} t(M,i) \tag{4}$$

The outer iteration was developed for comparing the water outlet temperature (t_2) with a specified water outlet temperature, and if thy are not in agreement a new value of $(NTU)_G$ is assumed until the difference between them meet a specified value. The enthalpy for the outlet air is then computed from the following equation:

$$h_{a_2} = \frac{1}{M}\sum_{j=1}^{M} h(j,M) \tag{5}$$

Then we can find $(NTU)_L$ from the following equation:

$$(NTU)_L = (NTU)_G \times \frac{G}{L} \tag{6}$$



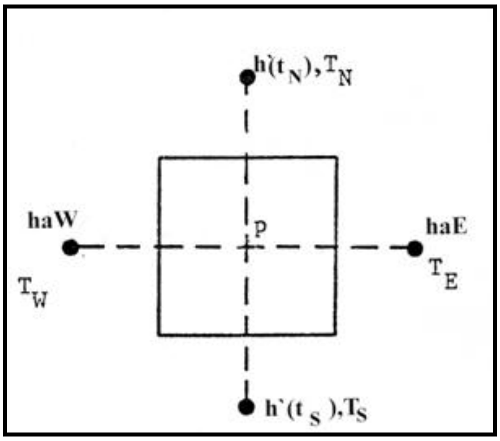


Fig (2): Cross-flow cooling tower Incremental volume

Fig (3): Incremental unit volume

RESULTS AND DISCUSSIONS

Results:

Results of this study are shown in **Figs. (4 to 7)**.

Discussions:

On comparing the performance of the three packings with each other it appears from **Fig.4** that the performance of the PVC packing is higher than the other two by about (98%) because its wetted surface area per unit volume is higher than the other two packing. In addition to that the ascending water spray down as small droplets, because of the packing corrugation and slots which increase the period of contact between air and water. This figure also shows that the performance of the corrugated asbestos packing is slightly higher than the flat plate asbestos packing by about (10%) because the wetted surface area per unit volume for the corrugated asbestos packing is higher than corresponding value for the flat plate asbestos packing.

As a practical application, the correlation equation for each of the three packing used in the tower (PVC, flat plate asbestos and corrugated asbestos packing) had been obtained for the range of air flow rates (1.3964 kg/s.m^2) to (0.6991 kg/s.m^2). The correlation equations are:

$NTU = 0.2660\,(L/G)^{-0.9144}$ (For flat plate asbestos packing)

$NTU = 0.4581\,(L/G)^{-0.7171}$ (For PVC packing)

$NTU = 0.2567\,(L/G)^{-0.5815}$ (For corrugated asbestos sheets spaced 15mm)

$NTU = 0.2438\,(L/G)^{-0.5141}$ (For corrugated asbestos sheets spaced 25mm)

The volumetric mass transfer coefficient ($K'a$) for the PVC packing is higher than ($K'a$) for the other two types as shown from the same **Fig.5**. This figure also show that the volumetric mass transfer coefficient ($K'a$) for the corrugated asbestos sheets with (S=15mm) is higher than ($K'a$) of the flat plate asbestos packing with (S=25mm).

The correlation of each line is represented in the following relation:

$\mathbf{K'a = cG^{b}L^{d}}$, where: c, b and d are constant

From the above relation, the volumetric mass transfer coefficients equations which had been obtained for the range of air flow rates (1.3964 to 0.6991kg/s.m^2) are:

$$K'a = 1.5976\, G^{0.9144}\; L^{0.0856}$$ (Flat plate asbestos packing)

$$K'a = 2.7513\, G^{0.7171}\; L^{0.2829}$$ (PVC Packing)

$$K'a = 1.5413\, G^{0.5815}\; L^{0.4185}$$ (Corrugated asbestos sheets spaced 15mm)

$$K'a = 1.4643\, G^{0.5141}\; L^{0.4859}$$ (Corrugated asbestos sheets spaced 25mm)

The volumetric heat transfer coefficient (K_Ga) is highly increased with (G), but slightly increased with (L) as shown in **Fig.6**. Also it can be seen from this figure that the volumetric heat transfer coefficient (K_Ga) is higher for PVC packing than for the flat plate and corrugated asbestos packings. The volumetric heat transfer coefficient (K_Ga) for the corrugated asbestos packing is also higher than the corresponding value for the flat plate asbestos packing.

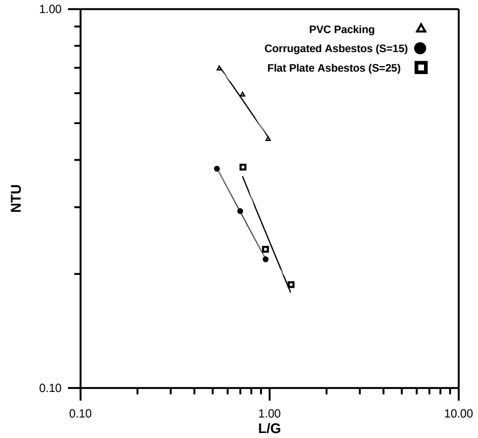
The above influences, the volumetric heat transfer coefficient (K_Ga) as with the volumetric mass transfer coefficient ($K'a$), because of the Lewis relation which relates them:

$$\frac{K_G a}{k'a \times c_{pm}} = Le \tag{7}$$

This relation is considered unity.

Fig.7 is a plot of volumetric rate of mass transfer coefficient ($K'a$) in ($kg/s.m^3$) versus air flow rate (G) in ($kg/s.m^2$) for the results of Molyneux[3] using fiber glass packing and for the three packing used in this experimental work and for a constant water flow rate of ($1.358 kg/s.m^2$). This Figure shows a good matching with the results of Molyneux, especially for the corrugated asbestos packing (S=25mm).

The reason for the differences in the slope of these lines is because of the differences in types of packing.



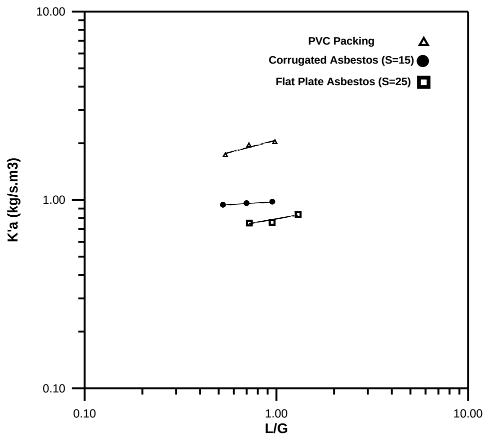
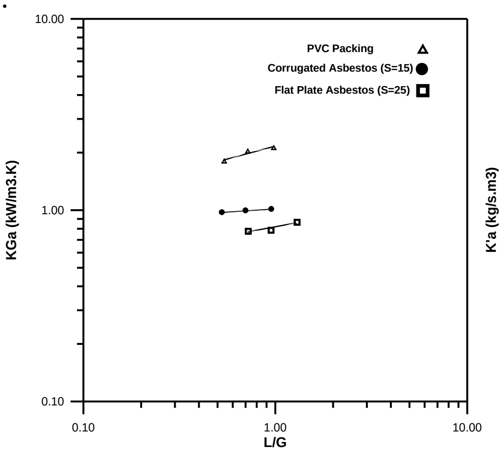


Fig (4) The (NTU) Vs (L/G) For Flat Asbestos Corrugated Asbestos And PVC Packings For Air Flow Rate About (0.775kg/S)

Fig (5) Volumetric Mass Transfer Coefficient Vs. L/G For Different Packings At Air Flow Rate About (0.775kg/S)



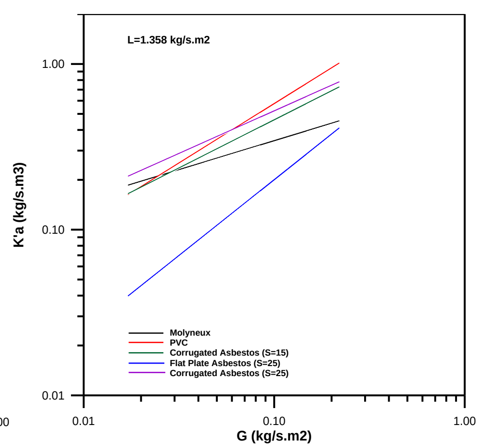


Fig (6) Volumetric Heat Transfer Coefficient Vs. L/G For Different Packing At Air Flow Rate About (0.775kg/S)

Fig (7) K′A Vs. G For Molyneux (1967) Data Compared With The Present Work

CONCLUSIONS

1. The performance of a given volume of tower packing is affected mainly by the ratio of water flow rate to the air flow rate. This means that the maximum performance can be obtained with minimum water and air flow ratio (L/G).
2. The mass and heat transfer coefficients ($K'a$) and (K_Ga) are mainly affected by the air flow rates and slightly by the water flow rates.
3. For a given volume of packing, air flow rates and water flow rates, the performance of PVC packing exceeds that of the two types of the asbestos packings by about (45%-50%). Also the performance of the corrugated asbestos sheets is more than the corresponding performance of the flat plate asbestos by about (6.5%-7%).
4. We found that in the case of using PVC packing, substantial decrease in volume of the cooling tower is obtained compared to using the other two kinds of asbestos packings to achieve the same operating conditions due to the performance of PVC packing being higher than the corresponding value of the other types.

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NOMENCLATURE

A	*Area of water surface per unit volume*
C_{pw}	*Specific heat of water assumed, 4.19*
G	*Inlet air mass flow rate (dry air)*
h_a	*Enthalpy of moist air, h_{a1} entering, h_{a2} leaving (dry air)*
H'	*Saturated enthalpy of moist air at bulk water temperature*
K'	*Unit conductance, mass transfer, interface to main air stream*
L	*Inlet water mass flow rate*
M	*Number of horizontal and vertical increments*
NTU	*Number of transfer units*
V	*Cooling volume*