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# Heat and mass transfer in a direct evaporative cooler using new material

# إنتقال الحرارة والكتلة خلال مبرد تبخيرى مباشر بإستخدام مادة جديدة

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**ملخص**

يعرض هذا البحث تحليل لاداء مبرد تبخيرى مباشر باستخدام مادة جديدة "اللوف", واللوف يتم استخدامه فى تقشير الجلد الميت ويوجد فى المتاجر على شكل فرش للظهر او قفازات للتقشير ويستخدم ايضا لغسيل الاوانى والاسطح واللوف من النباتات التى تنبت فى المناخ الدافىء.

تم تصميم نفق هواء خصيصا لتحليل اداء هذه المادة الجديدة تحت تأثير عدة متغيرات منها سرعة الهواء ومعدل سريان الماء وسمك المادة المستخدمة. يتم تسجيل خصائص الهواء عند الدخول والخروج من النفق ومن ثم حساب معاملات انتقال الحرارة والكتلة. ويتم مقارنة كفاءة المبخر ومعامل انتقال الحرارة ومعامل انتقال الكتلة ورقمى نسلت و شرود فى كل مرة.

هذه المادة الجديدة ذات مسامية جيدة جدا وبالتالى الفقد فى ضغط الهواء اقل ما يمكن بالاضافة الى ارتفاع كفاءة المبخر الذى يصل الى 94%.

ونظرا لهذه الكفاءة العالية والتكلفة الضئيلة لهذ المادة مع توفرها وانتشارها, لهذا يمكن استخدامها فى المبخرات فى الاغراض المنزلية والتجارية ايضا.

## ABSTRACT

This paper presents the performance analysis of a new evaporative cooling pad material "Luffa", or luffa, sponges which are produced from long, thin gourds and are much harder and more abrasive than sea sponges. Most often, luffa sponges are used for scrubbing and exfoliating dead skin. Often seen for sale in stores in the form of back brushes or exfoliating mitts, luffa sponges also make ideal scrubbers for pots, pans and surfaces, such as counter tops. The luffa gourd is an amazingly versatile, no-fuss plant that is relatively easy to grow in warm climates [1]. A test tunnel is constructed for this particular work and the behavior of this new evaporative cooling pad material is analyzed while changing of three parameters, Air velocity, water flow rate and the bad thickness. Inlet and outlet air characteristics are recorded with time and transient variation of thermo-physical properties of air is then evaluated from the measured data and heat and mass transfer equations. Cooling effectiveness, heat transfer coefficient, mass transfer coefficient and accordingly Nusslet and sherowd numbers are obtained and compared each time. This new bad material offers the lowest pressure drop among other materials ($<10$ N/m$^2$) with high effectiveness (94%). Because of better performance, lower costs and easy availability of this new bad material, Using it as wetted media may enhance the scope of using this material in domestic and commercial evaporative cooling systems for sustainable development.

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## Nomenclature

| | |
|---|---|
| Cp | specific heat of air, kJ/kg K |
| w | air humidity ratio, $g_v/kg_a$ |
| D | mass diffusion coefficient of vapor in the air, $m^2/s$ |
| $w_{st.}$ | saturated air humidity ratio, kg/kg |
| h | enthalpy of air, kJ/kg |
| $\eta$ | effectiveness of the cooler % |
| $Q_{lat.}$ | latent heat KW |
| $Q_{sens.}$ | sensible heat KW |
| $m_w$ | rate of water evaporation |
| $m_a$ | air mass flow rate, kg/s |
| $P_a$ | atmospheric pressure, Pa |
| $P_{st.}$ | saturation pressure, Pa |
| $P_v$ | vapor pressure, Pa |
| K | conductive heat transfer coefficient W/m K |
| $h_c$ | convective heat transfer coefficient $W/m^2K$ |
| $h_m$ | mass transfer coefficient m/s |
| t | air temperature, °C |
| T | air temperature, K |
| $v_a$ | frontal air velocity, m/s |
| R | air gas constant J/kg K |
| f | the wetted bad surface area $m^2$ |
| A | cross section area of bad material $m^2$ |
| a,b | length m , width m |
| x | bad material thickness m |
| $h_{fg}$ | enthalpy of eaporation KJ/Kg |
| Pr | Prandtl nomber |
| Nu | nusslet nomber |
| Sh | sherwood nomber |
| Sc | schmidt nomber |
| Re | reynold nomber |
| Dh | hydraulic diameter (4*A)/P |
| ***Subscripts*** | |
| in | inlet |
| out | outlet |
| db | dry-bulb |
| wb | wet-bulb |
| a | air |
| w | water |
| st | saturated |
| v | vapour |
| ***Greek symbols*** | |
| $\delta$ | thickness of pad material, m |
| $\eta$ | cooling efficiency % |
| $\rho$ | air density, $kg/m^3$ |
| $\Phi$ | relative humidity, % |
| $\mu$ | dynamic viscosity, kg/m s |
| $\xi$ | bore surface area per pad unit volume $m^2/m^3$ |

# 1. Introduction and literature review

The higher living and working standards associated to the reduced prices of air conditioning systems led to a considerable increase in demand for air conditioning in buildings. In European Union (EU) energy demand for space cooling applications has grown 14.6% per annum between 1990 and 2000 and it is expected an annual growth of 3.4% in the 2000–2030 period [2]. The number of room air-conditioners in use in EU has grown from a value of 1.2 millions in 1990 to 7.4 millions in 1996, and should be close to 33.0 millions in 2020, with an estimated electricity consumption of 43928 GW h/year, four times greater than the 1996 value. In a conservative perspective, the associated greenhouse gases emissions (particularly $CO_2$ emissions) are projected to increase by a factor of 35 from 1990 to 2020 [3]. Concerning central air conditioning systems – with more than 12 kW of cooling capacity – the scenario also deserves special attention. In EU, between 1985 and 2000 the annual addition of building cooled-floor area by this kind of systems, really added or simply replaced, grew from 40 million to 150 million square meters [4]. Important consequences are, however, associated to such a massive use of air conditioning systems. Major ones are, accentuation of fossil fuels dependence and its related need to consider the immediate and future availability of energy products at affordable prices, greenhouse effect due to CO2 emissions, ozone layer depletion and occurrence of electrical peak loads in hot summer days which often conducts to brown-out situations.

The evaporative cooler works on one of the oldest principles of air conditioning known to Man: Cooling of air by the evaporation of water. It is the most common form of household cooling found in arid areas. The popularity of evaporative cooling in such areas is due to its relatively low initial and operational cost compared to refrigerated cooling. Conventional direct evaporative coolers consist of a large water reservoir, a pump that draws water from the reservoir and discharges it through spray nozzles directly into air stream or through cooling pads. The direct evaporative cooler cools the air when the air comes in contact with water in the wetted media (cooling pads). During evaporation of water in air stream, the required heat is taken from air itself.

Now-a-days some buildings use air conditioning systems which are based on vapour compression refrigeration system. These systems consume substantial power and may be harmful to environment. Therefore it is very much needed to have low energy consuming devices such as evaporative cooling systems for providing thermal comfort in buildings. Many researchers have carried out research in order to improve performance of such systems. Dowdy, Reid, and Handy (1986) have tested aspen pads for the evaporative cooling process and heat & mass transfer coefficients were experimentally obtained for various thickness of aspen pads. Dutta et al. (1987) described the suitability of evaporative cooling system (both direct and indirect type) for most zones of India and large areas of Australia. Navon and Arkin (1994) had conducted feasibility study on utilization of direct–indirect evaporative cooling for residences in Israel and found that this system can provide 40 a significantly higher level of thermal comfort. Bajwa, Aksugur, and Al-Otaibi (1993) and Heidarinejad et al. (2008) had conducted similar studied on various kinds of evaporative cooling systems to find their suitability in multi-climate country. Eldessouky Hisham and Al-haddad Amir (1996) evaluated the thermal and hydraulic performance of a modified two stage evaporative cooler, Various considerations were the mode of operation, packing thickness, mass flow rate of the water. Liao and Chiar (2002) developed a compact wind tunnel to simulate evaporative cooling pad-fan systems for direct measurement of system performance. Two alternative materials of coarse and fine fabric PVC spongy mesh were tested as pads in wind tunnel experiment. The effect of air velocity, water flow rate, static pressure drop across pad and pad thickness were experimentally examined. El-Dessouky, Ettouney, and Al-Zeefari (2004) developed an experimental rig of two stage evaporative cooling and tested in the Kuwait environment. The system was formed of an indirect evaporative cooler unit followed by direct evaporative cooler. Gunhan, Demir, and Yagcioglu (2007) evaluated the suitability of some local materials as evaporative cooling pads. Rawangkul, Khedari, Hirunlabh, and Zeghmati (2008) had tested evaporative cooling pad made of coconut coir in Thailand region. They showed that evaporative cooling efficiency of coconut coir's pad was around 50%. Camargo, Ebinuma, and Silveira (2005) conducted experimental studies on direct evaporative cooler and determined convective heat transfer coefficients. Maheshwari, Al-Ragom, and Suri (2001) had demonstrated energy saving potential of an indirect evaporative cooler in Kuwait. Sodha, Singh, and Sawhney (1995) evolved a rule of thumb in terms of the size of the floor area to be cooled for two different climates, namely hot dry and composite for direct evaporative coolers. Zhao, Liu, and Riffat (2008) had tested several types of materials, namely metals, fibers, ceramics, zeolite and carbon as heat and mass transfer medium in the indirect evaporative cooling systems, and from the investigation, the most adequate material and structure were identified. Jain, J. K., & Hindoliya, D.A. (2011), had tested twonew types of materials, (Palash and Coconut), heat and mass transfer were obtained throw the indirect evaporative cooler system, and pressure drop also measured from the investigation, these two new materials had offered low pressure drop and high effectiveness.

From the above literature review, one can conclude that researchers from all over the world are trying to search or develop new efficient and sustainable pad materials which is very much required for further enhancing cooling potential of evaporative cooling devices.

In the present work a new material, i.e. luffa as shown in Fig.1(are produced from long, thin gourds and easy to grow in warm climates) has been experimentally tested while changing of three parameters, Air velocity, water flow rate and the bad thickness in a test tunnel which is constructed for this particular work.

## 2. Experimental set-up

This paper presents the performance analysis of a new evaporative cooling pad material "Luffa", sponges which are produced from long, thin gourds these gourds are cut down in suitable size and distributed them inside a wire mish frame of size 30 cm × 30 cm cross section with the required thickness as shown in Fig.1

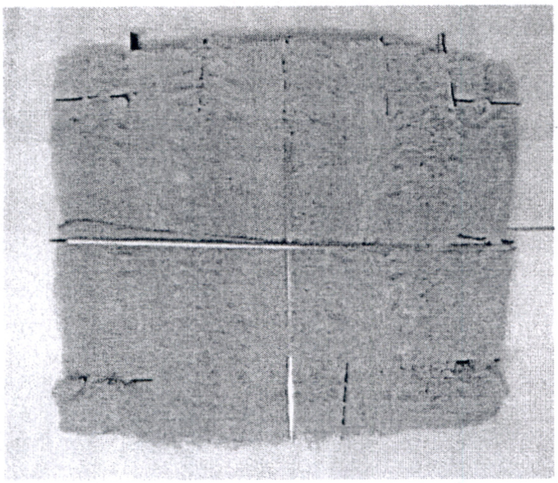


*Fig. 1 Photogragh of luffa sponge.*

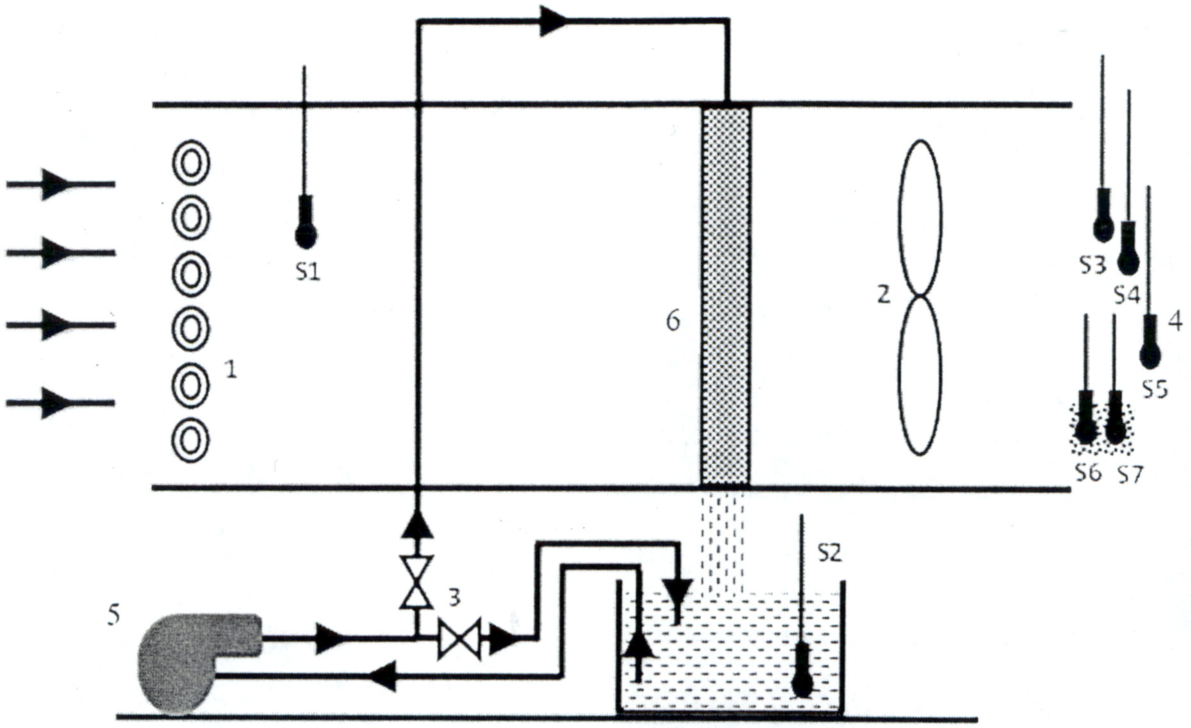
Fig. (2) shows the entire construction of the humidifier in test section. An air wind tunnel has been designed and prepared to operate adirect evaporative cooler. The set up includes a wind tunnel from fiber glass, electerical heater(1), blower(2), rigid pad media(6), re-circulating pump(5), a sprinkler and a water collecting tank. A test section having 30 cm × 30 cm cross section and 120 cm long is used to accommodate pads of desired thickness.

Ambient air is forced to circulate through the tunnel by blower. Cooling water is sprayed from above the test section by sprinkler onto the top surface of cooling pad. The falling water is collected in the water collecting tank and then re-circulated through the pump.

A provision is made for easy changing of cooling pad of different thicknesses, different water flow rates and different air velocities.

A number of calibrated sensors are fitted and connected to data logger for recording various temperatures to measure dry bulb and wet bulb temperatures at the inlet and outlet of the test section and also TM-414 (4) sensor is fitted to measure air velocity and outlet air temperature and outlet humidity. The sensors are connected to data logger for recording various temperatures.

The system piping diagram and control valves were designed to facilate changing water flow rate.




1 - Air heater
2 - Air blower
3 - Water valves
4- S5 TM-414 sensor
5 - Water pump
6 - Bad material "Lufa"
7 - S1 -- S7 LM35 temperature sensors

*Fig. 2 The air test tunnel*

In order to measure the pressure drop across the cooling pads of different materials, an arrangement is made to measure static pressure difference across the pad using inclined tube manometer.

## 3. Measurment

A number of calibrated sensors are fitted and connected to data logger for recording various temperatures to measure dry bulb and wet bulb temperatures at the inlet and outlet of the test section.

There are eight sensors of LM35 to measure dry bulb and wet bulb temperatures as shown in Fig.3,

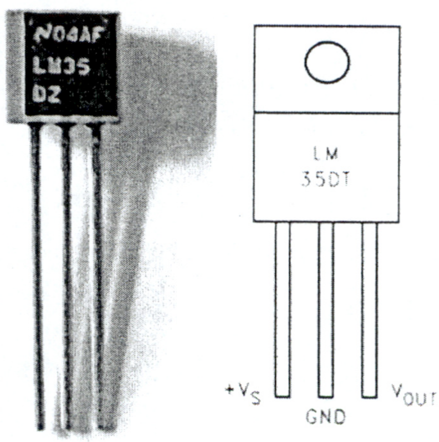



***Fig. 3** the LM35.*

The LM35 series are precision integrated-circuit temperature sensors, whose output voltage is linearly proportional to the Celsius (Centigrade) temperature.

The LM35 does not require any external calibration or trimming to provide typical accuracies of ±0.25°C at room temperature and ±0.75°C over a full −55 to +150°C temperature range.

These number of LM35 sensors are connected to data logger as shown in Fig.4 for recording various temperatures each specified time and save all data in database.

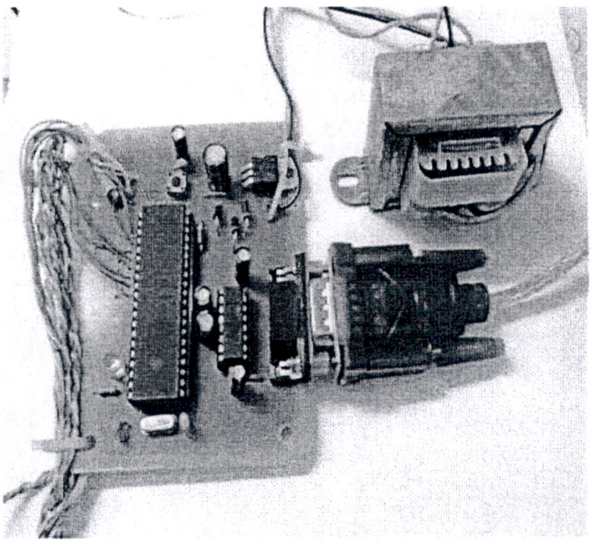


*Fig. 4 the data logger.*

Another sensor TM-414 also is fitted to measure air velocity and outlet air temperature and outlet humidity as shown in Fig.5.

*Fig. 5 the TM-414 sensor.*

TM-414 is an advanced sensor which measure air velocity within range 0.4 ~ 45 m/s with an accuracy of ±3% +0.2 and measure temperature within range -20 ~ 60 °C with an accuracy of ±1 °C and measure air humidity within range 20 ~ 80 % with an accuracy of ±3% and if the humidity is < 20 or > 80% the accuracy will be ±5%.

## 4. Methodology

The experiments are conducted in the month of July in Egypt with variable air flow rates. three different materials thickness, three different water flow are tested one by one. Initially set up was run for about 15 min to ensure near steady state condition. During each experiment dry bulb, wet bulb temperatures and relative humidity of air at inlet and outlet are measured each 5 sec using advanced sensors and recorded using data logger.

Temperature of water in the collecting tank is also recorded.

Neglecting heat flux transferred from surroundings, air is cooled and humidified with constant enthalpy, i.e. air losses a certain amount of sensible heat as well as gains an equal amount of latent heat of water evaporation. Water temperature in the basin will almost be a certain value that is slightly higher than the wet-bulb temperature of inlet air during a stable operation period of the cooler. Thus, the air process in the direct evaporative cooler is different from that of cooling tower in some extent.

Fig.6 shows the configuration of a pad module, x-axis donates air flow direction; z-axis donates the water sprinkling direction.

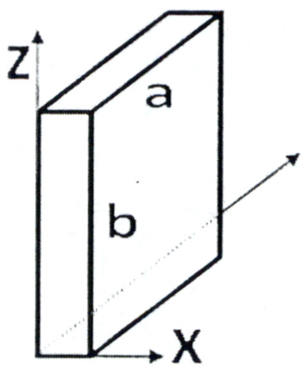



*Fig. 6 the pad module.*

To simplify the heat and mass transfer analysis, the following assumptions are made:

(1) the pad material is wetted uniformly and fully;

(2) the convective heat transfer coefficient $h_c$ and mass transfer coefficient $h_m$ of moist air on the surface of water film are constant;

(3) the thermal properties of air and water are constant;

(4) water–air interface temperature is assumed to be uniform and constant;

(5) Lewis number Le = 1;

(6) the heat flux transferred from the surroundings is neglected;

(7) air near the water–air interface is saturated, its temperature is assumed to be that of sprinkled water;

(8) air temperature changes only in flow direction denoted by x.

- The cooling efficiency of the evaporative cooler is given by:

$$\eta = \frac{t_{in} - t_{out}}{t_{in} - t_{wb}} \quad \dots (1)$$

where $t_{in}$ is the dry bulb temperature of air at inlet, $t_{out}$ is the dry bulb temperature of air at outlet, $t_{wb}$ is the wet bulb temperature of air at inlet.

- Then we get enthalpy:

$$h = cpt_{db} + w(2500 + 1.84t_{db}) \quad \dots (2)$$

where $cp$ is the heat capacity of air " $cp = 1.005\frac{Kj}{Kg.k}$ ", $t_{db}$ is the dry bulb temperature of air, $w$ is the specific humidity of air.

- Air mass flow rate:

$$m_a = \rho_a\, A\, V_a \qquad ...(3)$$

where $\rho_a$ is the density of air " $\rho_a = 1.124\ kg/m^3$"

- Rate of water evaporation:

$$m_w = m_a\, \Delta w \qquad ...(4)$$

where $\Delta w$ is the difference between specific humidity of air at outlet at steady and specific humidity of air at inlet "$\Delta w = w_{out,st.} - w_{in}$".

- Amount of latent heat of water evaporation:

$$Q_{lat} = m_w\, h_{fg} \qquad ...(5)$$

Where $h_{fg}$ is the enthalpy of evaporation "$h_{fg} = 2200\ kJ/kg$".

- Amount of sensible heat equal amount of latent heat of water evaporation:

$$Q = Q_{sens} = Q_{lat} \qquad ...(6)$$

- Amount of gained sensible heat by air:

$$Q = h_c\, f\, \Delta t \qquad ...(7)$$

Where $\Delta t$ is logarithmic mean temperature difference $\Delta t = \frac{(t_{in,db}-t_w)-(t_{out,db}-t_w)}{ln\left(\frac{t_{in,db}-t_w}{t_{out,db}-t_w}\right)}$,

$f = \xi.a.b.x$ [18].

- When thermal equilibrium is reached, the water temperature theoretically approaches the wet bulb temperature of entering air. An energy balance for air/water vapour mixture across the pads can be written as follows:

$$m_a h_{a,in} + m_{v,in} h_{v,in} + m_w h_w - m_a h_{a,out} - m_{v,out} h_{v,out} = Q \quad ...(8)$$

From Eq.(4) and (8), one get,

$$Cp_a(T_{db,in} - T_{db,out}) + w_{in}(h_{v,in} - h_w) - w_{out}(h_{v,out} - h_w) = Q/m_a \qquad ...(9)$$

- The convective heat transfer coefficient $h_c$ :

$$h_c = \frac{Q}{f\, \Delta t} \qquad ...(10)$$

- The mass transfer coefficient $h_m$ :

$$m''_w = \frac{m_w}{f} \quad , \quad h_m = \frac{m''_w}{\Delta\rho} \qquad ...(11)$$

Where $\Delta\rho$ is the density difference of water vapor "$\Delta\rho = (\rho_{v@t_{wb,st.}} - \rho_{v@t_{db,st.}})$",

$\rho_{v@t_{db,st.}}$ is the vapor density at dry bulb temperature of air outlet at steady "we get $p_{st@t_{db}}$ then $p_v = p_{st}\,.\,\emptyset$ where $\emptyset$ is the relative humidity then $\rho_{v@t_{db,st.}} = \frac{p_v}{R\,T_{db}}$ "

$\rho_{v@t_{wb,st.}}$ is the vapor density at wet bulb temperature of air outlet at steady "we get $p_{st@t_{wb}}$ and $\emptyset$=1 then $\rho_{v@t_{wb,st.}} = \frac{p_v}{R\,T_{wb}}$ "

- Nusslet number Nu :

$$Nu = \frac{h_c\, Dh}{K} \qquad ...(12)$$

- The mass diffusion coefficient of vapor in the air D :

$$D = 2.256\left(\frac{T_{db}}{256}\right)^{1.81} * 10^{-5} \qquad ...(13)$$

- Sherowd number Sh :

$$Sh = \frac{h_m\, Dh}{D} \qquad ...(14)$$

## 5. Results and discussion

In the present work a new material, i.e. Luffa has been experimentally tested and performance of the evaporative cooler has been compared with those of other existing materials as shown in table 1.

An air tunnel is constructed for this particular work and the performance of this new evaporative cooling pad material is analyzed while changing the value of three parameters, Air velocity, water flow rate and the bad material thickness.

The difference in mass between the bad material in case of dry and throughly wetted represents the amount of water carried by the bad and subsequently the volume of water which represents the space volume in the bad, we devide this volume by the bad thickness and result then devided by the bad volume to get $\xi$ which represents the bore surface area per unit badding volume which is constant for each material and in case of our new material "luffa" $\xi$ =27 $m^2/m^3$, by this method we can get the actual surface area of the bad material.

*Table (1) Comparison between present and previous published works*

| no. | Pad media | Mass flow rate $m_a$ in Kg/s | Inlet DBT | Outlet DBT | Wetted surface area in $cm^2$*100 | Saturation effectiveness in % | Pressure difference across the pad in N/m2 | Heat transfer coeff. in $Kw/m^2.k$ | Mass transfer coeff. In m/s |
|---|---|---|---|---|---|---|---|---|---|
| 1 | Aspen fibers [5] | 0.029 | 42.25 | 28.81 | 2.991 | 75.37 | 88.18 | 2.009 | 1.6 |
| 2 | Aspen fibers [5] | 0.045 | 42.46 | 29.495 | 1.495 | 71.59 | 48.99 | 6.789 | 3.4 |
| 3 | Khus fibers [11] | 0.045 | 42.14 | 30.84 | 0.624 | 64.16 | 14.69 | 5.544 | 2.8 |
| 4 | Khus fibers [11] | 0.029 | 42.82 | 29.12 | 1.248 | 73.66 | 29.39 | 1.909 | 1.5 |
| 5 | Coconut fibers [11] | 0.045 | 43.78 | 29.894 | 1.045 | 69.39 | 39.19 | 6.375 | 3.2 |
| 6 | Coconut fibers [11] | 0.029 | 43.12 | 28.225 | 2.091 | 78.6 | 68.59 | 2.208 | 1.8 |
| 7 | Palash fibers [11] | 0.029 | 43.5 | 27.267 | 1.489 | 83.89 | 39.19 | 2.615 | 2.1 |
| 8 | Palash fibers [11] | 0.045 | 43.35 | 27.783 | 0.744 | 81.04 | 29.39 | 8.98 | 4.6 |
| 9 | Loufa (new) | 0.2 | 39 | 30.8 | 9 | 88.13 | < 10 | 14.6936 | 18.1693 |
| 10 | Loufa (new) | 0.28 | 39 | 28 | 9 | 87.9 | < 10 | 30.4 | 86.95 |
| 11 | Loufa (new) | 0.22 | 39 | 29.8 | 9 | 73.6 | < 10 | 4.598 | 7.016 |
| 12 | Loufa (new) | 0.182 | 39 | 30.4 | 9 | 94.6 | < 10 | 16.8165 | 22.2097 |

**5.1 The effect of Reynold number on the evaporative cooler**

➢ The outlet dry bulb temperature decrease with the increase of air velocity at constant water flow rate and bad thickness 10mm and also decrease with the increase of water flow rate at constant air velocities and the same bad thickness 10 mm as shown in Fig.(6-a)

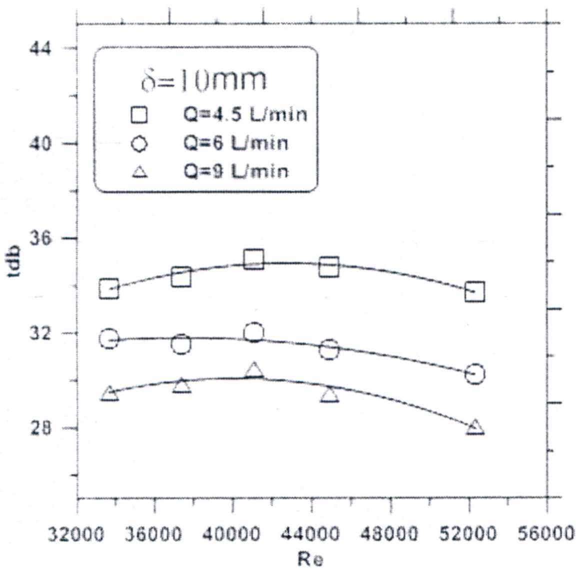



Fig.(6-a) Variation of outlet air dry bulb temperature with Reynold number for different water flow rates at 10 mm bad thickness

➢ The outlet dry bulb temperature increase with the increase of air velocity from 1.8 to 2.4 m/s at constant water flow rate and bad thickness 18mm and when we increase air velocity more than 2.4 m/s the output dry bulb temperature tend to decrease, also the output dry bulb temperature decrease with the increase of water flow rate at constant air velocities and the same bad thickness 18 mm as shown in Fig.(6-b)

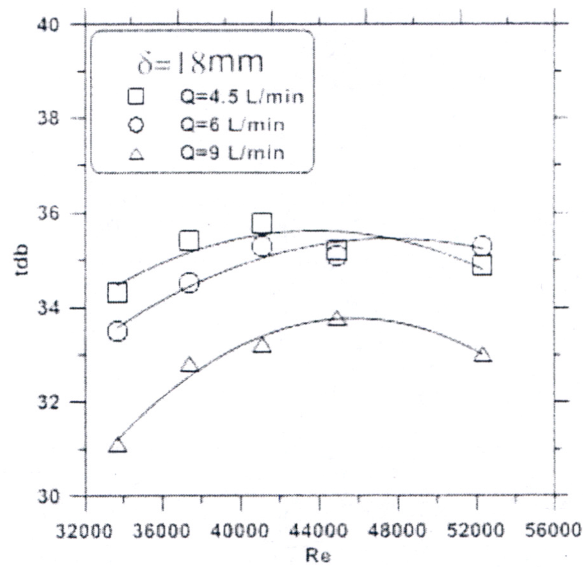



Fig.(6-b) Variation of outlet air dry bulb temperature with Reynold number for different water flow rates at 18 mm bad thickness

➢ The outlet wet bulb temperature decrease with the increase of air velocity for 4.5 and 6 L/min water flow rates at bad thickness 10 mm but increase with the increase of air velocity for 9 L/min water flow rate at the same bad thickness 10 mm as shown in Fig.(6-c)

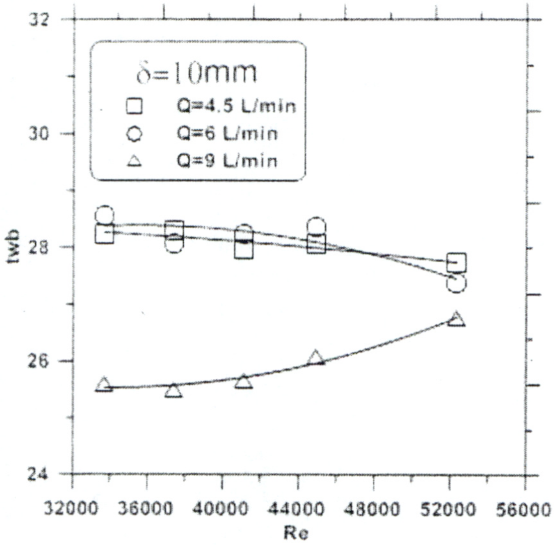



Fig.(6-c) Variation of outlet air wet bulb temperature with Reynold number for different water flow rates at 10 mm bad thickness

➢ The outlet wet bulb temperature increase with the increase of air velocity for 6 and 9L/min water flow rates at bad thickness 18 mm but decrease with the increase of air velocity for 4.5 L/min water flow rate at the same bad thickness 18 mm as shown in Fig.(6-d)

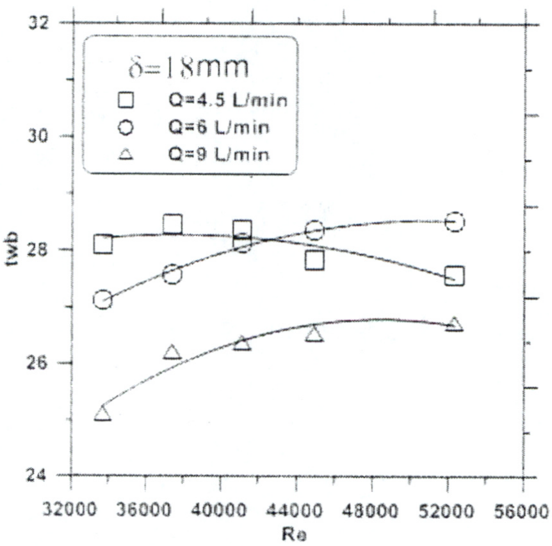



Fig.(6-d) Variation of outlet air wet bulb temperature with Reynold number for different water flow rates at 18 mm bad thickness

➢ It is noted that at 2.2m/s air velocity (Re=41147.13) represents a turning point in almost curves that shows the relation between Nusslet and Sherowd numbers with Reynold nomber as the values of Nusslet and Sherowd numbers tend to increase at air velocity greater than 2.2m/s (Re=41147.13) at constant water flow rates and 10mm bad thickness, and that due to the increase in the rate of water evaporation which causes the increase in heat mass transfer coefficients and subsequently the increase in Nusslet and Sherwood numbers as show in Fig.(7-a,b)

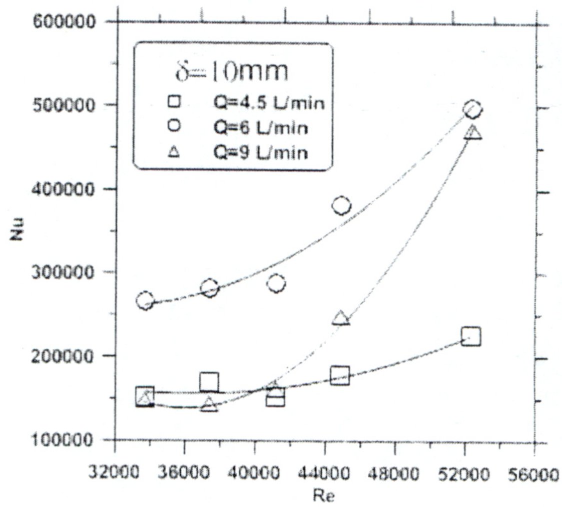



Fig.(7-a) Variation of Nusslet number with Reynold number for different water flow rates at 10 mm bad thickness

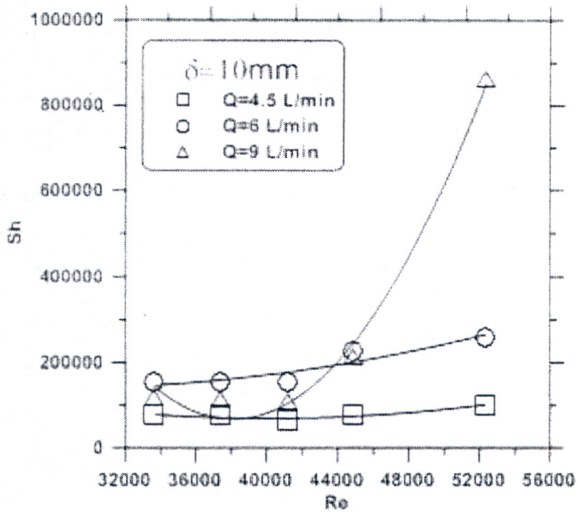



Fig.(7-b) Variation of Nusslet number with Reynold number for different water flow rates at 10 mm bad thickness

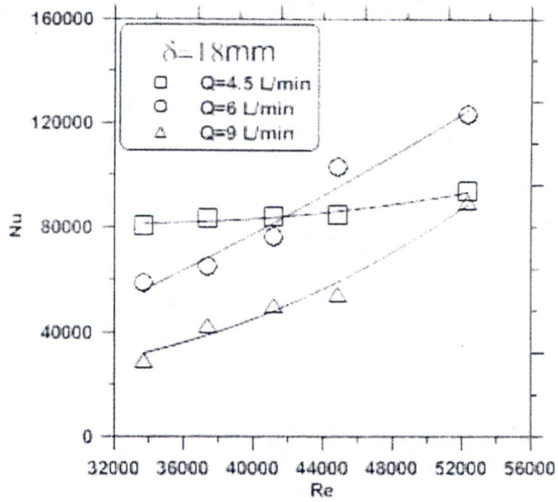



Fig.(7-c) Variation of Nusslet number with Reynold number for different water flow rates at 18 mm bad thickness

➢ The values of Nusslet and Sherowd numbers tend to increase with the increase of air velocity for 6 and 9 L/min constant water flow rates and 18mm bad thickness, and that due to the increase in the rate of water evaporation as the wet bulb temperature of air outlet increase which causes the increase in heat mass transfer coefficients and subsequently the increase in Nusslet and Sherwood numbers, but the values of Nusslet and Sherwood numbers tend to keep constant at 4.5 L/min water flow rate and at the same thickness as the wet bulb temperature of air outlet in this case tends to keep constant causing constant water evaporation as show in Fig.(7-c,d)

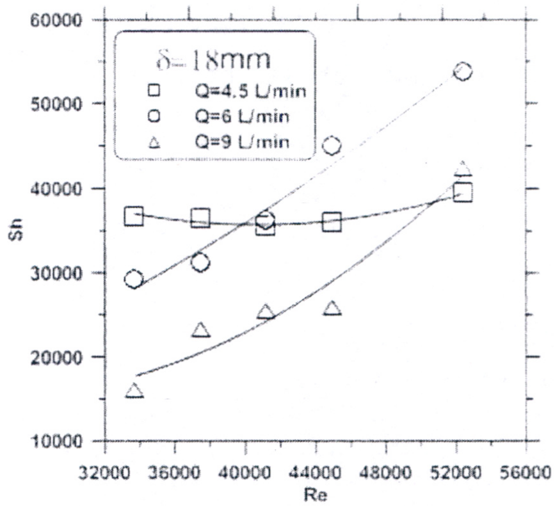



Fig.(7-d) Variation of Sherwood number with Reynold number for different water flow rates at 18 mm bad thickness

➢ The results also indicated that at constant thickness 10mm at different water flow rates the cooler effictiveness decreases with the increase of air velocity from 1.8 to 2.2 m/s as the output dry bulb increases ,but when we increase air velocity more than 2.2 m/s the effectiveness increases as the output dry bulb decreases as show in Fig.(8-a).

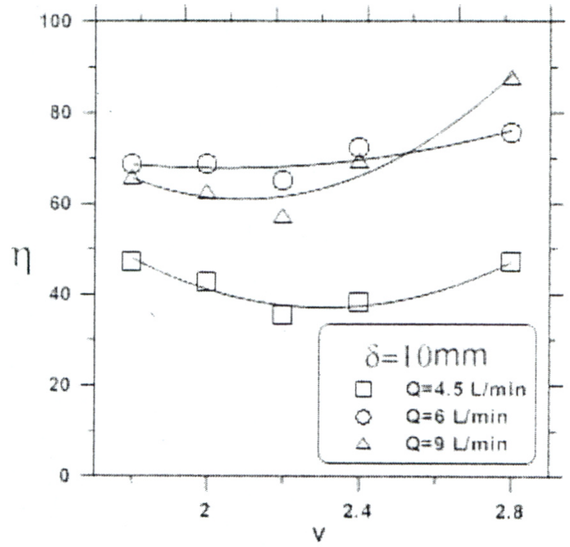



Fig.(8-a) Variation of performance of evaporative cooler with air velocity for different water flow rates at 10 mm bad thickness

➢ At 18mm bad thickness at different water flow rates the evaporative cooler performance decreases with the increase of air velocity as the output dry bulb increases as shown in Fig.(8-b).

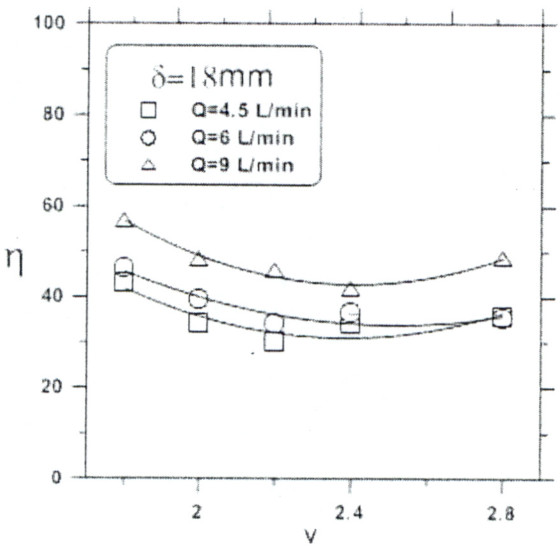



Fig.(8-b) Variation of performance of evaporative cooler with air velocity for different water flow rates at 18 mm bad thickness

➢ At 22 mm bad thickness at 6 L/min water flow rate the evaporative cooler performance has a very big values at lower velocities and decreases with the increasing of air velocities as the output dry bulb increases as shown in Fig.(8-c).

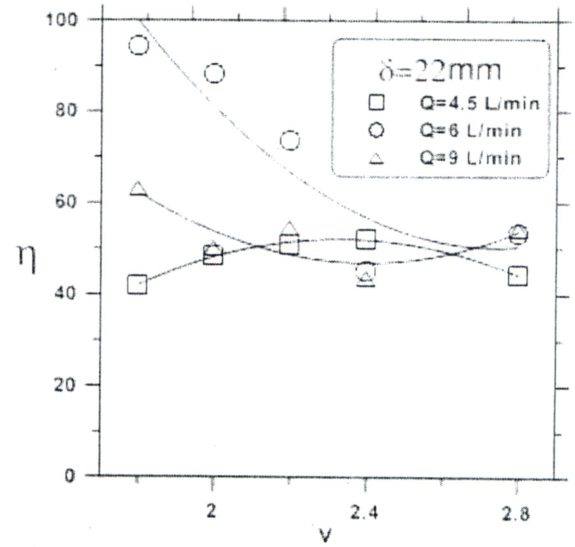



Fig.(8-c) Variation of performance of evaporative cooler with air velocity for different water flow rates at 22 mm bad thickness

➢ Depending on the relationship between evaporative saturation efficiency and air velocity. Mentioned above, one can say that the temperature drop of air passing through the pad decreases with the increase of air velocity.

The relative humidity of airflow increases as it passes through the pad media as a result of evaporation, the results of the measurements of the relative humidity indicated that ralative humidity difference decreases with the increase of air velocity.

Generally higher efficiencies are obtained with thicker pads, and slower air velocities. This result reflects greater evaporative rates as air takes more time to travel through the pad.

### 5.2 The effect of water flow rate on the evaporative cooler

➢ The output dry bulb and wet bulb temperatures obviously decrease with the increase of water flow rate as the increasing of water flow rate increases the opportunity for air to saturate with vapor and subsequently increases the saturation effectiveness of the cooler due to the decrease in the outlet dry bulb temperature as shown in Fig.9,

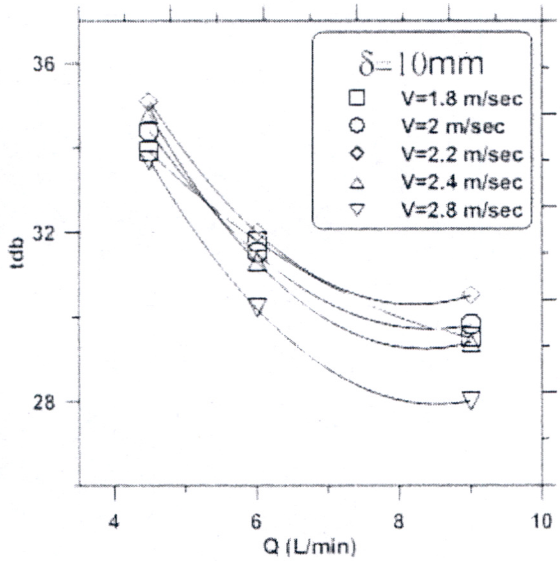



Fig.(9-a) Variation of outlet air dry bulb temperature with water flow rate for different air velocities at 10 mm thickness

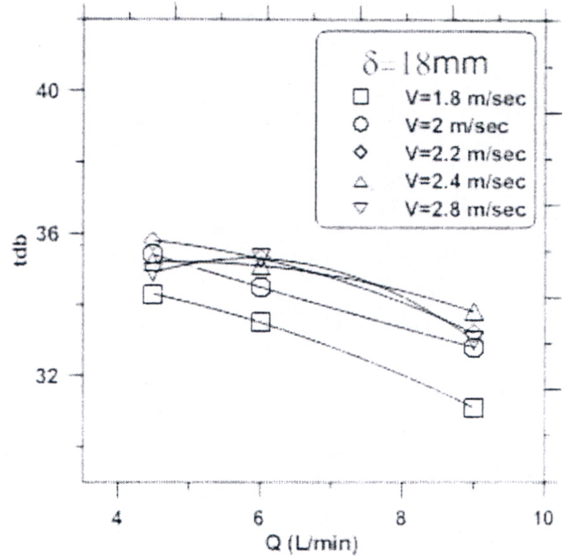



Fig.(9-b) Variation of outlet air dry bulb temperature with water flow rate for different air velocities at 18 mm thickness

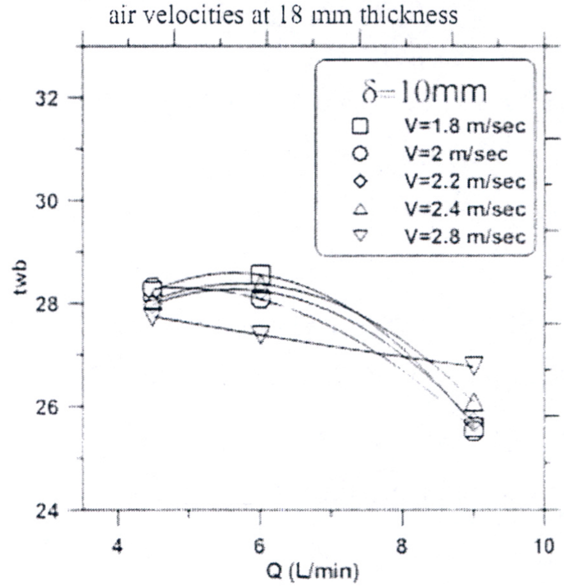



Fig.(9-c) Variation of outlet air wet bulb temperature with water flow rate for different air velocities at 10 mm thickness

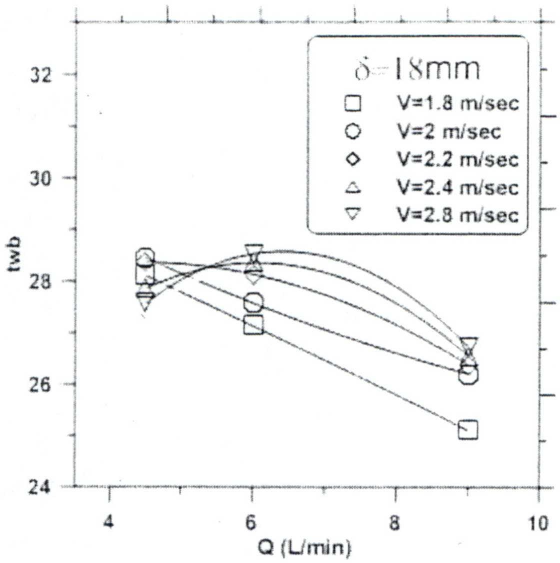



Fig.(9-d) Variation of outlet air wet bulb temperature with water flow rate for different air velocities at 18 mm thickness

## 5.3 The effect of bad thickness on the evaporative cooler

➢ Generally the saturation effictiveness of the evaporative cooler increases with the increase of bad thickness, as with thin pads, the porosity of the pad is enough to allow fast passage of air thus reducing the heat exchange period of evaporation. On the other hand, increasing the pad thickness reduces the porosity of pad and increase the passing time of air which increases the heat exchange period, subsequently improving the evaporation and cooling efficiency.

But in this study the effect of the pad thickness on the saturation efficiency is not agree with what was mentioned above as results show that at 4.5 L/min water flow rate the outlet air dry bulb temperature increase with the increase of bad thickness from 10mm to 18mm and decrease when we increase thickness more than 18 mm at different air velocities as shown in

Fig.(10-a).

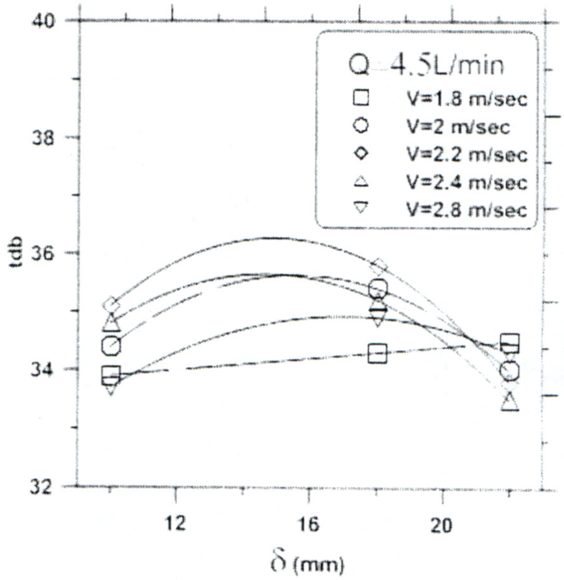



Fig.(10-a) Variation of outlet air dry bulb temperature with bad thickness for different air velocities at 4.5 L/min water flow rate

and the wet bulb temperature tend to increase with the increase of bad thickness at the same conditions as shown in Fig.(10-b).

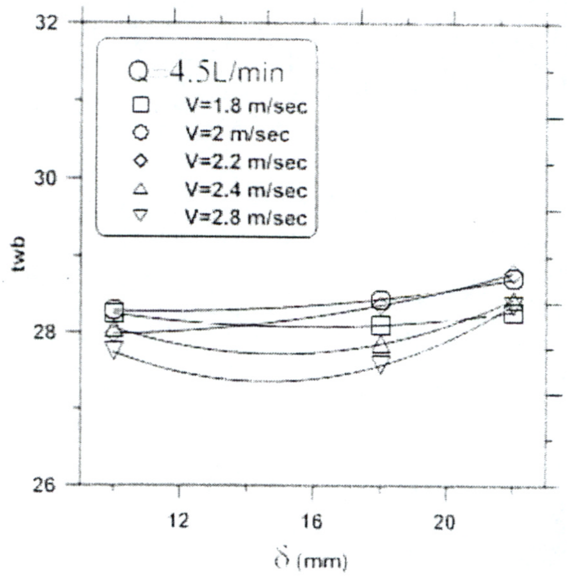



Fig.(10-b) Variation of outlet air wet bulb temperature with bad thickness for different air velocities at 4.5 L/min water flow rate

➢ At 6 L/min water flow rate the outlet air dry bulb temperature increase with the increase of bad thickness from 10 mm to 18 mm and decrease when we increase thickness more than 18 mm at different air velocities as shown in Fig.(10-c).

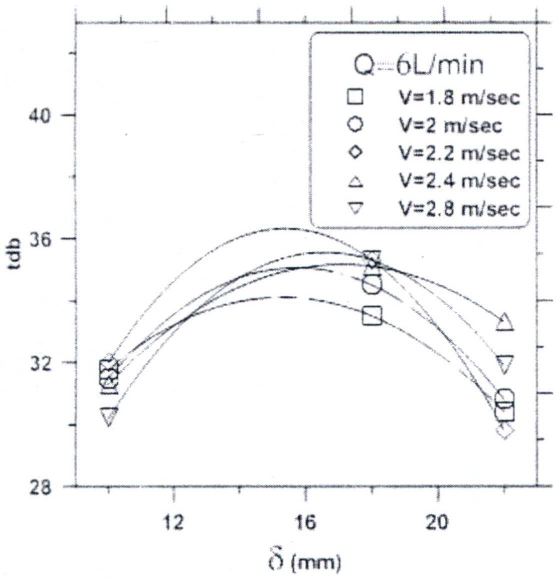



Fig.(10-c) Variation of outlet air dry bulb temperature with bad thickness for different air velocities at 6 L/min water flow rate

and at the same conditions the outlet air wet bulb temperature decrease with the increase of bad thickness from 10mm to 18mm and increase when we increase thickness more than 18mm at lower air velocities "less than 2.2 m/s", but at air velocity more than 2.2 m/s the outlet air wet bulb temperature decrease while increasing the bad thickness as shown in Fig.(10-d).

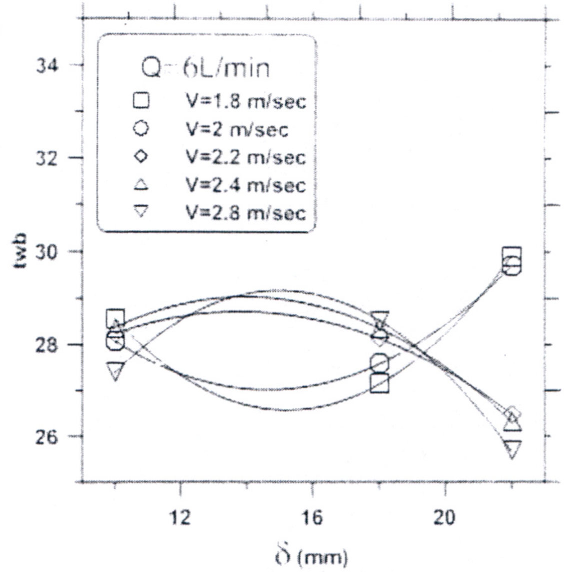



Fig.(10-d) Variation of outlet air wet bulb temperature with bad thickness for different air velocities at 6 L/min water flow rate

➢ At 9 L/min water flow rate from results one can say that the outlet air dry bulb and wet bulb temperature increases with the increase of bad thickness as shown in Fig.(10-e,f).

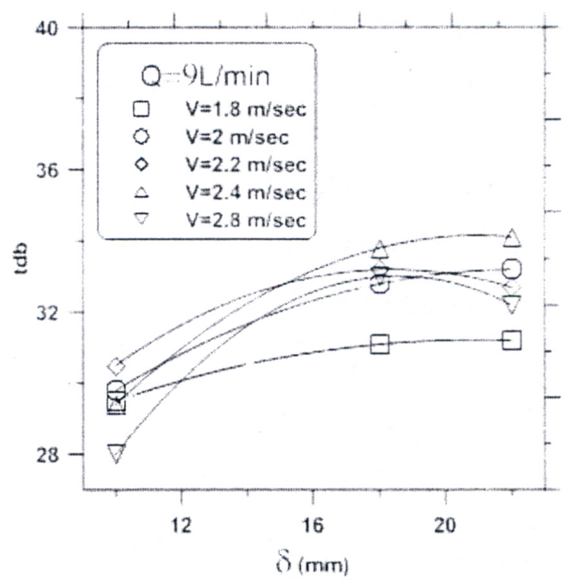



Fig.(10-e) Variation of outlet air dry bulb temperature with bad thickness for different air velocities at 9 L/min water flow rate

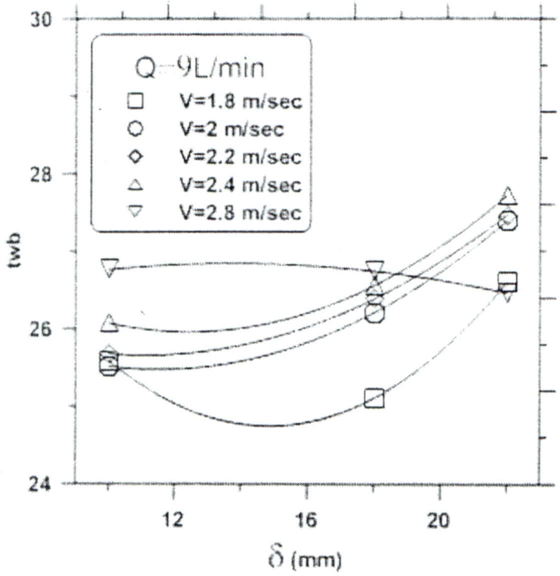



Fig.(10-f) Variation of outlet air wet bulb temperature with bad thickness for different air velocities at 9 L/min water flow rate

➤ It is noted from curves that shows the relation between Nusslet and Sherowd with Reynold nomber the pad of 22 mm both Nusslet and Sherwood nombers increases with the increase of air velocity at 4.5 and 9 L/min water flow rate due to the increase in the outlet wet bulb temperatures which causes increase in rate of water evaporation, however at 6 L/min water flow rate both nusslet and sherowd nombers descend while decresing air velocity and that due to the increase in the rate of water evaporation at 22 mm bad thickness with 6 L/min water flow rate at lower air velocities which causes the increase in heat and mass transfer coefficients and subsequently the increase in Nusslet and Sherwood nombers as shown in Fig.11.

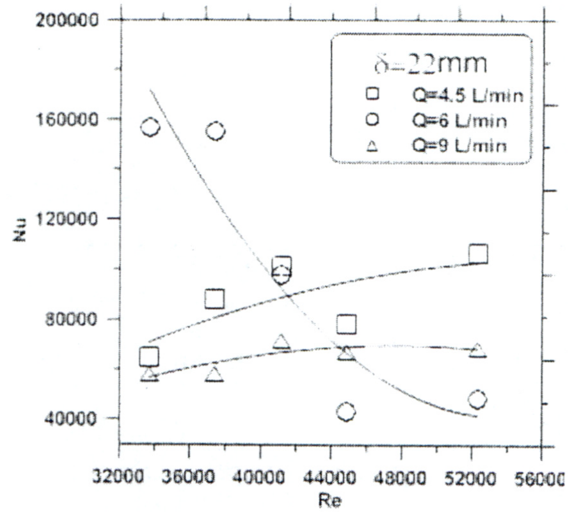



Fig.(11-a) Variation of outlet air Nusslet nomber for different air velocities and water flow rates at 22 mm bad thickness

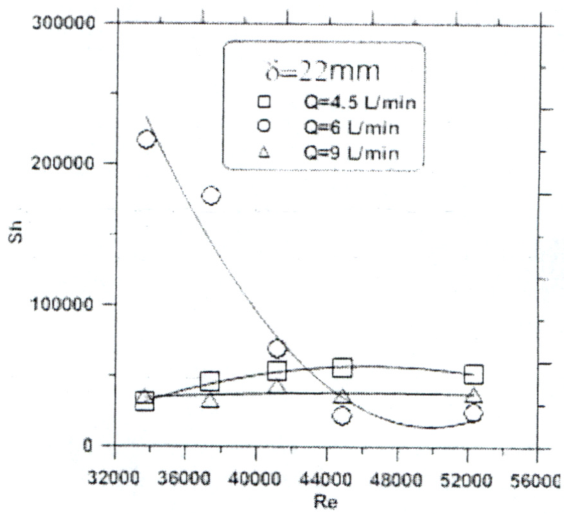



Fig.(11-b) Variation of outlet air Sherwood nomber for different air velocities and water flow rates at 22 mm bad thickness

## 5.4 Choosing the optimum condition for better performance of the cooling systems

An evaporative cooling system must decrease the air temperature to the desired degree by minimum power consumption and expenses. Thus, an ideal pad media must have the highest evaporative saturation efficiency and the lowest airflow resistance. For this reason, when selecting the optimal pad media for cooling systems evaporative efficiency and resistance against airflow of the pad materials must be considered together. According to the data obtained, the highest evaporative saturation efficiency and the working condition of the selected pad materials.

Over the tested range of operating condition, it is obsered that the most reliable pad media is at 22 mm pad thickness, 6 L/min water flow rate and 1.8 or 2 m/s air velocity resulted in high evaporative saturation efficiency the point of view.

And is at 10mm pad thickness, 9 L/min water flow rate and 2.8 m/s air velocity from the point of view of the high evaporative saturation efficiency and the lowest pressure drop level.

### 5.5 Emperical correlations for Nusslet and Sherwood nombers

Based on results the eqautions of Nusslet and Sherwood nombers can be written as following:

$$Nu = 16.69\, Re^{0.835}\, Pr^{0.3} \quad \text{...(15)}$$

At $3 * 10^4 > Re > 5 * 10^4, Pr \approx 0.7$

$$Sh = 14\, Re^{0.8}\, Sc^{0.2} \quad \text{...(16)}$$

At $3 * 10^4 > Re > 5 * 10^4, Sc \approx 0.5$

## 6. Conclusions

Researchers from all over world are trying to develop a new efficient and sustainable pad materials which is very much required for further enhancing cooling potential of evaporative cooling devices.

The present investigation deal with a new pad material (luffa) and the performance of the evaporative cooler is experimentally studied at several operating conditions, the following can be concluded:

(1) During each experiment there is an amount of water evaporates in air increasing the humidity of air and decreasing its temperature as well, this amount of water that evaporates in air taken from the water which is sprayed on the bad material causing lack in the amount of water in the tank, so we have to make up this lack.

From results one can say that the amount of make up water ranging between 0.659 to 1.657 Lit.

(2) One can conclude that luffa show great potential for use as a wetted media in domestic and commercial direct evaporative coolers. It is found that the performance of the cooler is much better than that of other materials tested before with very low pressure drop.

(3) The comparison of saturation effectiveness and heat and mass transfer coefficients for five different pad materials

shows that using the new evaporative bad material gives the highest saturation effectiveness and lowest pressure drop.

(4) It may also be noted that using the new pad material "Luffa", increasing pad thickness does not increase the saturation effectiveness significantly and that could be due to the small scale of bads thickness used in this study and emphasizes the great impact of water flow rate and air velocity on the effictiveness.

(5) Generally higher efficiencies are obtained with thicker pads, and slower air velocities. The highest effectiveness is found 94% at 22 mm pad thickness, 6 L/min water flow rate and 1.8 or 2 m/s air velocity.

(6) the temperature drop and the relative humidity of air passing through the pad decreases with the increase of air velocity.

(7) Results showed that there is an optimum condition in each parameter:
Thickness to be greater than 18 mm,
Water flow rate is about 6 L/min,
Air velocity should be less than 2 m/s.

## 7. References


[1] http://www.ehow.com/about_5137588_loofah-sponge.html

[2] European Commission, European Energy and Transport – Trends to 2030, Directorate General for Energy and Transport, 2003.

[3] European Commission, EERAC – Energy Efficiency of Room Air Conditioners, Final Report, Directorate General for Energy, 1999.

[4] European Commission, EECCAC – Energy Efficiency and Certification of Central Air Conditioners, Final Report, Directorate General Transportation – Energy, 2003.

[5] Dowdy, J. A., Reid, R. L., & Handy, E. T. (1986). Experimental determination of heat and mass transfer coefficients in Aspen pads. ASHRAE Transaction, 92(2A), 60–68.

[6] Dutta, S., et al. (1987). Design and operation of evaporative cooling system. Journal of Australian Refrigeration, air Conditioning and Heating, 41(2), 28–36.

[7] Eldessouky Hisham, T., & Al-haddad Amir, A. (1996). Thermal and hydraulic performance of a modified two stage evaporative cooler. Renewable Energy, 7(2), 165–176.

[8] El-Dessouky, H., Ettouney, H., & Al-Zeefari, A. (2004). Performance analysis of two-stage evaporative coolers. Chemical Engineering, 102(3), 255–266.

[9] Gunhan, T., Demir, V., & Yagcioglu, A. K. (2007). Evaluation of the suitability of some local materials as cooling pads. Biosystems Engineering, 96(3), 369–377.

[10] Heidarinejad, G., et al. (2008). Feasibility of using various kinds of cooling systems in a multi-climates country. Energy and Buildings, 40(10), 1946–1953.

[11] Liao, C.-M., & Chiar, K.-H. (2002). Wind tunnel modeling the system performance of alternative cooling pads in Taiwan region. Building and Environment, 37(2), 177–187.

[12] Maheshwari, G. P., Al-Ragom, F., & Suri, R. K. (2001). Energy-saving potential of an indirect evaporative cooler. Applied Energy, 69(1), 69–76.

[13] Navon, R., & Arkin, H. (1994). Feasibility of direct–indirect evaporative cooling for residences, based on studies with a desert cooler. Building and Environment, 29(3), 393–399.

[14] Rawangkul, R., Khedari, J., Hirunlabh, b., & Zeghmati, B. (2008). Performance analysis of a new sustainable evaporative cooling pad made from coconut coir. International Journal of Sustainable Engineering, 1(2), 117–131.

[15] Sodha, M. S., Singh, S. P., & Sawhney, R. L. (1995). Evolution of design patterns for direct evaporative coolers. Building and Environment, 30(2), 287–291.

[16] Zhao, X., Liu, S., & Riffat, S. B. (2008). Comparative study of heat and mass exchanging materials for indirect evaporative cooling systems. Building and Environment, 43(11), 1902–1911.

[17] Jain, J. K., & Hindoliya, D.A. (2011), Experimental performance of new evaporative cooling pad materials. Sustainable Cities and Society (2011).

[18] J.M. Wu, X. Huang , H. Zhang (2005), Theoretical analysis on heat and mass transfer in a direct evaporative cooler.