EFFECTS OF PULSATING FLOW ON THE EVAPORATION RATE FROM WATER SURFACE

Hesham M. Mostafa[1], M. Abdelkreem[2]

[1]Mechanical, [2]Chemical Engineering Dept.,

Higher Technological Institute, 10th of Ramadan City, Egypt

ABSTRACT

Experiments were performed to investigate the effects of pulsating air flow on the evaporation rate from a water pan, which fixed in the floor of a rectangular cross-section wind tunnel. A test rig was designed and constructed to carry out the experiments. Steady and pulsating air flows were considered. The effect of pulsation frequency on heat and mass transfer coefficients was examined at different mass air flow rates, and subsequently its impact on Nusselt and Sherwood numbers. The pulsation frequency was varied from 0.067 Hz to 1.2 Hz and Reynolds number ranged from 1.2×10^4 up to 3.8×10^4. The results indicated that, Nusselt number and Sherwood number in the presence of pulsation were considerably higher than those for steady air flow. It was also observed that Nusselt and Sherwood numbers get higher values with increasing pulsation frequency. The obtained results are correlated and those of steady flow are compared with the previous work.

Key Words: Evaporation Rate, Heat And Mass Transfer, Pulsating Flow, Water Surface.

I. INTRODUCTION

Gas-liquid flow systems with coupled heat and mass transfer are widely encountered in practice. Drying, evaporative cooling, liquid film evaporator, turbine blade cooling, cooling of microelectronic equipments, and the simultaneous diffusion of metabolic heat and perspiration in the control of our body temperature are just some examples.

The evaporation of water and its diffusion into the flowing air requires the transfer of the latent heat of vaporization to the water in order to vaporize it. The required heat for vaporization at the surface to the flowing air is transferred by convection from the flowing air stream and by conduction from underneath of the water surface due to the sensible energy of the water itself. Therefore, combined heat and mass transfer processes occur. In such situation, the boundary layer incorporates a resistance for both heat and mass transport processes. At low temperatures the process is mainly heat transfer controlled. However, with increasing the temperature difference between water surface and the flowing air, both the heat transfer rate and concentration difference increase. This concentration difference reaches a maximum when the interface saturation pressure reaches the vapor pressure in the flowing air. Consequently, the whole process of evaporation becomes mass transfer controlled.

Evaporation to gas stream flow and related problems have received considerable attention due to their numerous applications. A comprehensive transient-state, three-dimensional model for heat transfer and fluid dynamics in a channel was presented by Wang et al. [1]. After model validation, the pulsating flow with four different cases is numerically investigated. The results indicate that both amplitude and period are very important parameter, which affect the flow and heat transfer performance. Mass transfer measurements have been reported for internal flows with moving liquid vapor interface [2]. They showed that the effect of the reverse velocity at the gas-liquid interface reduces the mass transfer rate. In these studies, heat transfer was not considered. Chow and Chung [3] investigated, numerically, the evaporation of water over a flat plate into a laminar gas stream using the governing equations for heat and mass transfer. They derived an iterative similarity solution to the problem. Their numerical results showed that, for the same mass flux of the free stream of low temperatures, water evaporates faster in dry air than in humid air and in superheated steam. However, the trend is reversed at high free stream temperatures.

Evaporation of water film into a gas stream along a flat plate was investigated by Schroppel and Thiele [4], Chow and Chung [5]. Their analyses were restricted to the processes with negligible effects of liquid film; therefore, only the heat and mass transfer in the air stream were considered. The experiments reported by Sparrow and Tao [6] were performed in a flat rectangular duct. Their investigation aimed to determine the mass/heat transfer and pressure drop response to periodic, rod-type disturbance elements situated adjacent to one of the principal wall and oriented transverse to the flow direction. Cycle-average, fully developed Sherwood numbers displayed substantial enhancement compared with the smooth-wall duct. At the wall, that contained rod-type disturbance elements, enhancement in Sherwood numbers of up to 140 % have been occurred as reported.

The evaporation rates of water were measured by Haji and Chow [7] and the results agreed well with the predictions of [3, 5] if the heat loss from the water pan was accounted for. Sheikholeslami and Watkinson [8] examined the effect of water vapor content in humid air and superheated steam at elevated temperatures on the rate of evaporation of water. Their experimental results confirmed the existence of inversion point temperature, above which the rate of evaporation of water increases with increasing in the water vapor content of the medium. Yan and Soong [9] investigated liquid film cooling in a turbulent gas stream. In their study, it is disclosed that the introduction of a thin continuous liquid film onto a given surface can effectively protect the wetted surface from thermal damage by the proximate hot gas stream. Combined heat and mass transfer processes that occur when water evaporates from a flowing film in an inclined channel to air stream, have been investigated by Zheng and Worek [10]. It was found that the combined heat and mass transfer in film evaporation can be enhanced by adding rods to the plate surface which result in agitation of the flowing water film and air stream.

Eames et al. [11] has collected a review on the evaporation coefficient of water. It is concluded that, molecular collision in the vapor phase and heat transfer limitation in the liquid phase can have a considerable influence on experimental evaporation rates. The evaporation of liquid into its own vapor was studied by Schwartze and Brocker [12]. They developed a model to predict the evaporation rate of water into humid air and the inversion temperature above which the evaporation rate into pure superheated steam is higher into dry air for different drying processes of constant mass and volumetric flows. It is reported that the calculated evaporation rates were in good agreement with literature data.

Sultan [13] investigated experimentally the rate of evaporation from a water pan to airflow in a rectangular cross section wind tunnel. Wedges are fixed on the inner vertical walls of the test section above the water panel. The effect of area ratio on the evaporation rate and hence the mass transfer coefficient was considered at different values of Reynolds number. It is reported that the convergent pressure gradient leads to an increase in the mass transfer coefficient.

The rate of evaporation from water surface is dependent on several parameters. These include the air stream parameters such as its mass flow rate, its temperature and its moisture content as well as the water temperature. Recent analysis carried out by Marek and Straub [14] indicated that water evaporation rate increased with the increase of surface water temperature, but the rate of this increase gradually slowed down with the surface water temperature increase. This implies that the water evaporation rate is not directly proportional to vapor pressure difference, and may relate to its exponent, i.e. $(P_{v,ws} - P_{v,\infty})^n$, with the power of n <1. Tang and Etzion [15] executed measurements to investigate the evaporation process from a wetted surface and that from a free water surface into the ambient under a wide range of climatic conditions. It is reported that the water evaporation rate is proportional to the exponent of water vapor pressure in the power of 0.82 for evaporation from a free water surface and of 0.7 when water evaporates from the wetted towels. A comparison of water evaporation rates from both wetted surface and free water surface has shown that when wind velocity across the water surfaces is very low, the rate of water evaporation from the wetted surface is greater than that from the free water surface. However, with higher wind velocity, this is reversed, and the evaporation rate from the free water surface was higher.

Pulsating flow is one of the unsteady flows that are characterized by periodic fluctuations of the mass flow rate and pressure. It can be produced by reciprocating pump or by steady flow pump/blower together with some mechanical pulsating devices. It is expected that the heat transfer to or from the flow would be changed since the pulsation would alter the thickness of the boundary layer and hence the thermal resistance. Pulsating flow is assumed to be consisted of a steady Poiseuille flow and purely oscillatory. Experimental investigation is still the most reliable way to deal with the pulsating flow. Most of investigators [16-22] considered in their studies a small number of operating variables and confined it to relatively narrow range. Many parameters have an influence on heat transfer characteristics of pulsating turbulent flow. Among those, pulsation frequency, its amplitude, axial location, Reynolds number, Prandtl number and pulsator type and its location. As a result, some investigators reported little increase, no increase, and even decrease in the rate of heat transfer. These conflicting in results showed that the heat transfer characteristics in pulsating flow are still not clearly understood. Heat transfer characteristics of pulsating turbulent air flow in a pipe heated at uniform heat flux were experimentally investigated by El-shafei et al., [23]. The experiments were performed over a range of $10^4 < Re < 4 \times 10^4$ and $6.6 \le f \le 68$ Hz. With installing the oscillator downstream of the tested tube exit, results showed that the local value of Nusselt number either increases or decreases over the steady flow value, depending on the pulsation frequency and Reynolds number.

Due to the complicated nature of unsteady turbulent flow, too much theoretical investigations are needed to find a solution for the problems of hydrodynamics and heat transfer of such a flow. An analytical study on laminar pulsating flow in a pipe by Faghri et al. [24], reported that higher heat transfer rates are produced. They related that to the interaction between the velocity and temperature oscillation which introduces an extra term in the energy equation that reflects the effect of pulsations. On the other hand, Chang et al. [25] concluded that the pulsation has no effect on the time averaged Nusselt number. In order to check whether the steady flow analysis is applicable to the prediction of heat transfer analysis in a pulsating turbulent flow, Park et al. [26], carried out a series of measurements on heat transfer to a pulsating turbulent flow in a vertical pipe subjected to a uniform heat flux over a range of $1.9\times10^4 < Re < 9.5\times10^4$. It is reported that for the cases of high mean Reynolds number ($Re>5.5\times10^4$), the experimental data approaches to the quasi-steady predictions. However, at low mean Reynolds number ($Re<4\times10^4$), the data showed an increasing discrepancy with the increase of frequency.

In this work, simultaneous heat and mass transfer accompanied with evaporation process from water in a pan, located at the lower wall of a rectangular duct, to steady and low frequency pulsating air flows are investigated. The effect of pulsation frequency on both heat and mass transfer coefficients, and subsequently on Nusselt and Sherwood numbers are examined at different values of airflow rates.

II. EXPERIMENTAL TEST RIG

The experimental test rig is illustrated in Fig.(1). It mainly consists of a circular inlet section (which contains a control valve, pulsator, blower, and orifice plate), a rectangular cross section duct in which the test section is installed, and the exit section. Air is drawn from the ambient by the blower (2) into the inlet section via a throttle valve (3) that controlling the amount of air flow rate through the tunnel and a pulsator (6). The pulsator consists of a circular plate (5) which can rotate about its axis freely inside the inlet section by a small DC motor (7) provided with speed controller in order to obtain different pulsation frequency. A flexible connection (8) separates the blower section and the rest of the wind tunnel to eliminate any vibrations promoted. The leaving air from the blower flows through a calibrated orifice meter (9) to measure the average airflow rate. To insure that a fully developed flow is achieved at the entrance of the test section (11), air is traveled through the entrance length (19) of 2000 mm long, which having the same rectangular cross section as that of the test section.

The test section (11) walls are made of clear acrylic plate sheets of 6 mm thick. The basic dimensions for the test section are 1200 mm long, 188 mm wide and 200 mm high. Water pan of 400 mm long, 188 mm wide and 15 mm deep is fitted, horizontally, at the middle of the test section, as shown in Fig. (2). The outer surfaces of the water pan and tunnel walls are completely insulated with 50 mm glass wool to minimize the heat loss.

III. MEASUREMENTS AND METHODOLOGY

Prior to start of the experiments, water pan was first cleaned, and then filled with a known mass of water. Air was drawn from the ambient and controlled, using the throttle valve. The frequency of pulsating flow was varied and controlled by the variable speed DC motor. All the required measurements for air velocity, relative humidity and dry bulb temperature across the height of the cross section at upstream and downstream locations of the test section are executed, using a traversing mechanism provided with the measuring probes and indicators as shown in Fig. (1). The water temperature was measured during the experiment, and the duration time for each experiment was recorded. Mass of water in the pan was measured before and after each experiment by a digital mass balance (16) with a sensitivity of 0.1 g. Static pressure difference across the orifice plate was measured by a calibrated U-tube water manometer (18) with a scale division of 1 mm to measure the average air flow rate and in turn the average air velocity, which was checked against that measured by a hot wire anemometer sensor (type Testo 605-V1, of 8mm probe diameter), with a resolution of 0.01 m/s. Relative humidity was measured by a hygrometer sensor (type Testo 605-H1of 8 mm probe diameter), with a resolution of 0.1%. All temperatures were measured by 0.5 mm copper-constantan thermocouple (type k), which were connected to a temperature recorder via multi point switch having an accuracy of ± 0.1 °C. The largest calculated uncertainties in measurements were less than 7% for Reynolds number, 9.7% for Nusselt number and 8.5% for Sherwood number.

IV. DATA REDUCTION

The evaporation of water into air flow is a problem of coupled heat and mass transfer. This problem is easy to describe since it only deals with water vapor and air that can be considered as a binary mixture. Figure (2) illustrates a water surface exposed to steady and pulsating air flows. Due to the difference between the vapor pressure at surface water temperature and that at the dew point temperature of air stream, some water evaporates from the water pan, $\dot{m}_{ev}$, where it flows to the air flow and heat flows in the opposite direction. If the rate of evaporation is so small that its cooling effects are negligible, the air temperature and its composition can be assumed constant within the evaporation process in the direction of the air flow. Consequently, the free surface of water inside the pan is assumed to be flush with the lower base of the rectangular duct so that it does not occupy an appreciable part of the inner airflow area. In such situation, the mass transfer for the air stream can be described as:

$$h_m = \frac{\dot{m}_{ev}}{A(\rho_{v,ws} - \rho_{v,\infty})} \qquad (1)$$

Where h_m, $\dot{m}_{ev}$, A, and ρ are the mass transfer coefficient, the rate of water evaporation, the water surface area, and the water vapor density, respectively. The mass balance for the gaseous phase gives

$$\dot{m}_{a,in} . x_{in} + \dot{m}_{ev} = \dot{m}_{a,out} . x_{out} \qquad (2)$$

Where, x is the moisture content of air flow, and $\dot{m}_{a,in} = \dot{m}_{a,out} = \dot{m}_a$ = constant, with the air as the non-transferring component.

It is assumed that the heat is transferred from the air to the water only by convection and the evaporation occurred at the water surface temperature. With these assumptions, the heat balance for the flowing air is given by:

$$q'' \times A = (\dot{m}_a . c_{pa})(T_{\infty,out} - T_{\infty,in}) + \dot{m}_{ev} \times i_{fg} - \dot{m}_{ev} c_{pv} (T_{\infty,out} - T_{ws}) \qquad (3)$$

Where i_{fg} is the latent heat of vaporization predicted at the water temperature, and the water surface temperature, T_{ws} is assumed to be equal to that of water. The convective heat transfer coefficient is then follows as:

$$h = q''/(T_{\infty,m} - T_{ws}) \qquad (4)$$

Where, $T_{\infty,m} = (T_{\infty,in} + T_{\infty,out})/2$, is the mean bulk air temperature.

All thermodynamic properties of the water vapor and the flowing air are functions of temperature and pressure. Considering the interface at equilibrium and treating the air and water vapor as perfect gases, their properties can be predicted from the pure components and the partial pressures using the of standard thermodynamic relations. The unknown variables that must be accurately measured are the temperature of the flowing air, $T_{\infty,m}$; the water temperature, T_{ws}; the rate of evaporation, $\dot{m}_{ev}$; the relative humidity of the flowing air, Φ, and the ambient pressure.

The partial vapor pressure of the bulk humid air can be calculated from the measured relative humidity, Φ as:

$$P_{v,\infty} = \Phi \times P_{v,sat} \tag{5}$$

Where $P_{v,sat}$ can be determined at the measured dry bulb temperature of the flowing air. The partial density of water vapor away from the interface in the bulk air is given by:

$$\rho_{v,\infty} = \frac{P_{v,\infty}}{R_v T_{db}} \tag{6}$$

The water vapor density at the interface can be also calculated from

$$\rho_{v,ws} = \frac{P_{v,ws}}{R_v T_{v,ws}} \tag{7}$$

Where $P_{v,ws}$, $T_{v,ws}$ and R_v are the saturation pressure of water vapor at water temperature, water vapor temperature at water surface and gas constant for water vapor, respectively.

Then, with $P_{v,\infty}$ and ambient pressure, the moisture content in the bulk air follows as

$$x = 0.622\, P_{v,\infty} / (P_{atm} - P_{v,\infty}) \tag{8}$$

The heat and mass transfer coefficients for the steady airflow in the rectangular duct, over the water pan are expressed in dimensionless numbers, Nusselt number and Sherwood number, respectively. The variations of these dimensionless numbers with Reynolds number will be presented and discussed. The Nusselt number is defined as

$$Nu = \frac{h\ L_c}{k} \tag{9}$$

Sherwood number, which is analogous to the Nusselt number, given by

$$Sh = \frac{h_m\ L_c}{D} \tag{10}$$

The Reynolds number is calculated based on the characteristic length, L_c is given by:

$$Re = U_m L_c / \nu \tag{11}$$

The characteristic length appeared in the above dimensionless numbers is described as [13]:

$$L_c = \frac{4(\text{Volume occupied by the flow})}{\text{Wetted surface area}} = \frac{4WHL}{2(WL + HL)} \tag{12}$$

V. RESULTS AND DISCUSSION

Appropriate analysis for the experimental measurements; lead to obtain the convective heat and mass transfer coefficients and consequently Nusselt and Sherwood numbers for steady and pulsating airflow over a water surface. The effect of pulsation frequency variations on heat and mass transfer coefficients is examined over a range of $1.9\times10^4 < Re < 9.5\times10^4$.

As a preliminary test, the measured values of air velocity; relative humidity and dry bulb temperature along the vertical distance downstream of the water pan for steady flow are represented as shown in Fig. (3). Local values for air velocity are measured to obtain the mean value of air velocity. As shown in Fig. (3.a), the velocity of air increases sharply in the boundary layer zone, but out of this layer it takes, approximately, constant value. Also, it is noticed from Fig. (3.b) and Fig. (3.c) that, in the boundary layer zone, near the surface of the water pan, there is a variation of relative humidity and dry bulb temperature, but out of this layer, they take nearly constant value.

The average values of convective heat transfer coefficient, *h* and mass transfer coefficient, h_m are computed from the measured data for steady air flow of different mass rates and presented in Fig. (4). As expected, both h and h_m increases with increasing Reynolds number, *Re*. Consequently, both Nusselt number, *Nu* and Sherwood number, *Sh* increases with increasing *Re* as shown in Fig. (5).

The heat transfer results are compared with the available results reported by other workers. Two cases from the previous work are considered, for airflow over naphthalene surface fixed on the lower wall [6], and over a horizontal water pan inside a rectangular duct [13]. Figure (6) shows a comparison between the present results and theirs for steady air flow. It can be observed that, *Nu* has the same trend as those of the previous ones. The discrepancies may be attributed to the difference in geometry and type of evaporated fluid when compared with [6], and difference in geometry of the tested wetted surface when compared with [13].

The measured data for pulsating airflow with different frequencies, at steady state conditions are used to calculate the convective heat and mass transfer coefficients. At a certain frequency, the mean value of air mass flow rate for each run was determined. The variations of the convective heat transfer coefficient with Reynolds number at different frequency are shown in Fig. (7). At a certain pulsating frequency, it can be observed that *h* increases with increasing *Re* and higher than those of the steady flow. At the same time, as the pulsation frequency increases, *h* gets higher value and its enhancement over that of steady flow becomes bigger. Consequently, the same behavior for Nusselt number variation with Reynolds number as can be noticed from Fig. (8). The percentage increase in Nusselt number for pulsating flow of $f = 1.2$ *Hz* is nearly 30 % higher than that of the steady flow over the tested range of Reynolds number. This may be attributed the promotion of waves at the water surface which in turn result in the agitation of the boundary layer, leading to heat transfer enhancement.

Figure (9) shows the variation of mass transfer coefficient with Reynolds number for different pulsation frequency. It can be noticed that, at a certain frequency h_m increases with increasing *Re*. In the mean time, as the pulsating frequency increases, h_m gets higher value and the percentage enhancement over that for steady flow increases as well. Figure (10) shows the effect of varying pulsation frequencies on Sherwood number for different mass airflow rates. It can be noticed that, *Sh* takes the same behavior as *Nu*. At a certain pulsating frequency, its value increases with increasing Reynolds number. Over the tested range of Reynolds number, for pulsating airflow of frequency, $f = 1.2$ *Hz,* the percentage increase in Sherwood number is about 50 % higher than that for steady flow. As discussed above, this increase related to the agitation of the boundary layer resulted from pulsation. The increase of pulsation frequency leads to more agitation, resulting in more evaporation.

Pulsation frequency is normally expressed in a dimensionless form as Strouhal number, given by:

$$St = f\, L_c / U_m \qquad (13)$$

The effect of pulsation frequency variation on heat and mass transfer during the evaporation process can be discussed by plotting *Nu* and *Sh* versus Strouhal number. Figure 11 shows the variation of *Sh* and *Nu* with *St* at a certain value of Reynolds number ($Re = 2.4\times10^4$). It can be observed that *Sh* and *Nu* increase with increasing Strouhal number.

The present experimental results for heat and mass transfer of pulsating airflow are correlated as:

$$Nu = 0.023\, Re^{0.94}\, Pr^{0.4}\, (1 + 2.05\, St - 4.2\, St^2) \qquad (14)$$

$$Sh = 0.023\, Re^{0.83}\, Sc^{0.4}\, (1 + 4.02\, St - 4.82\, St^2) \quad (15)$$

Where *Pr* and *Sc* are the Prandtl number and Schmidt number, respectively, given by

$$Pr = \nu / \alpha, \text{ and}$$

$$Sc = \nu / D \quad (16)$$

The above correlations are related to the considered operating conditions in this investigation ($1.2\times104 < Re < 3.8\times104$, and $0 \le St \le 0.1$). The maximum error in predicted values of both Nusselt number and Sherwood number by the above suggested correlations was found to be nearly ± 20% as can be seen in Fig 12 and Fig.13, respectively.

VI. CONCLUSIONS

The evaporation of liquids into their own vapor air mixtures is a problem of coupled heat and mass transfer. The evaporation of water from a water pan into airflow inside a rectangular duct is investigated, experimentally for steady and pulsating airflow. The effect of pulsation frequency variation on the heat and mass transfer coefficients is presented and discussed at different values of Reynolds number and its impact on Nusselt and Sherwood numbers. The obtained results indicated that Nusselt and Sherwood numbers increase with increasing Reynolds number for steady and pulsating airflows. For pulsating airflow, Nusselt and Sherwood numbers increase by a considerable value than those for steady airflow. For $St = 0.1$ and at $Re = 2.4\times10^4$, the percentage increase in *Nu* and *Sh* was about 30% and 50% with respect to those for steady flow, respectively. The obtained results are correlated, and those for steady airflow are compared with previous work. Fair agreement is found between the present results and those reported in the available literature.

REFERENCES

[1] Y. Wang, Y. He, W. Yang and Z. Cheng, Numerical analysis of flow resistance and heat transfer in a channel with delta winglets under laminar pulsating flow, Int. J. Heat and Mass Transfer Vol. 82, (March 2015), pp. 51-65.

[2] M. W. Clarkand, C.J. King, Evaporation rates of volatile liquids in a laminar flow system: Part 1 and 2, A.I.Ch.E. JI 16, (1970), 64-75.

[3] L.C Chow and J.N. Chung, Evaporation of water into a laminar stream of air and superheated steam, Int. J. Heat Mass Transfer 26, (1983), 373-380.

[4] J. Schroppel and F. Thiele, On the calculation of momentum, heat and mass transfer in laminar and turbulent boundary layer flow along a vaporizing liquid film, Numer. Heat Transfer 6, (1983), 475-496.

[5] L.C. Chow and J.N. Chung, Water evaporation into a turbulent stream of air, humid air or superheated steam, 21st ASME/AIChE National Heat Transfer Conference, Seattle, WA, ASME Paper No. 83-HT-2 (1983).

[6] E.M. Sparrow and W. Q. Tao, "Enhanced heat transfer in a flat rectangular duct with stream wise-periodic disturbances at one principal wall" ASME J. Heat Transfer Vol. 105, (1983), pp. 851-861.

[7] M. Haji and L.C. Chow, Experimental measurement of water evaporation rates into air and superheated steam, J. Heat Transfer 110, (1988), 237-242.

[8] R. Sheikholeslami and A. P. Watkinson, "Rate of evaporation of water into superheated steam and humidified air" Int. J. Heat and Mass Transfer Vol. 35, No.7, (1992), pp. 1743-1751.

[9] W. M. Yan, and C. Y. Soong, Numerical study of liquid film cooling in a turbulent gas stream, Int. J. Heat Mass Transfer 36, (1993), 3877-3885.

[10] G. S. Zheng and W. M. Worek, “Method of heat and mass transfer enhancement in film evaporation” Int. J. Heat and Mass Transfer Vol. 39, No. 1, (1996), pp. 97-108.

[11] I. W. Eames, N.J. Marr and H. Sabir, “The evaporation coefficient of water: a review,” Int. J. Heat and Mass Transfer Vol. 40, No. 12, (1997), pp. 2963-2973.

[12] J. P. Schwartze and S. Brocker, “The evaporation of water into air of different humidity’s and the inversion temperature phenomenon” Int. J. Heat and Mass Transfer Vol. 43, No. 6, (2000), pp. 1791-1800.

[13] G. I. Sultan, “Effect of pressure gradient on the evaporation rate from a horizontal water panel” Mansoura Engineering Journal Vol. 27, No. 1, (2002), pp. M1-M12

[14] R. Marek and J. Straub, Analysis of the evaporation coefficient and the condensation coefficient of water., International Journal of Heat and Mass Transfer, 44, (2001), 39–53.

[15] R. R. Tangi and Y. Etzion, "Comparative studies on the water evaporation rate from a wetted surface and that from a free water surface", Building and Environment 39, (2004), 77 – 86.

[16] T. S. Zhao and P. Cheng, “Heat transfer in oscillatory flows” Annual Review of Heat Transfer Vol. IX, Chapter 7, (1998), pp. 1-64.

[17] A. Al-Haddad and N. Al-Binally, Prediction of heat transfer coefficient in pulsating flow. Int. J. Heat Fluid Flow, 10, 2, (1989), 131–133.

[18] M. A. Habib, S. A. M. Said, S. A. Al-Dini, A. Asghar, and S. A. Gbadebo, Heat transfer characteristics of pulsated turbulent pipe flow. J. Heat Mass Transfer, 34, 5, (1999), 413– 421.

[19] H. M. Mostafa, A. M Torki and K. M. Abd-Elsalam, Experimental study for forced convection heat transfer of pulsating flow inside horizontal tube. In: Fourth International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics. Cario, Ak3, (2005a).

[20] M. A. Habib, A. M. Attya, S. A .M. Said, A.I. Eid, and A. Z. Aly, Convective heat transfer characteristics of laminar pulsating air flow. J. Heat Mass Transf. 38, (2002). 221 – 232.

[21] J. Zheng, D. Zeng, P. Wang and H. Gao, An experimental study of heat transfer enhancement with pulsating flow. Heat Transfer, Asian Res. 23 (5), (2004), 279–286.

[22] A. E. Zohir, M. A. Habib, A. M. Attya, and A. I. Eid, An experimental investigation of heat transfer to pulsating pipe air flow with different amplitudes. J. Heat Mass Transfer, 42 (7), (2005), 625–635.

[23] E. A. M. Elshafei, M. Safwat , H. Mansour, and M. Sakr, "Experimental study of heat transfer in pulsating turbulent flow in a pipe", International Journal of Heat and Fluid Flow 29 (2008) 1029–1038

[24] M. Faghri, K. Javdani, and A. Faghri, Heat transfer with laminar pulsating flow in pipe. Heat Mass Transfer, 6 (4), (1979), 259–270.

[25] J. Chang, X.Y.L. Zhi, and T.S. Zhaoan, Analytical study of pulsating laminar heat convection in a circular tube with constant heat flux. J. Heat Mass Transfer,. 47, (2004), 5297–5301.

[26] J. S. Park, M. F. Taylor, and D. M. McEligot, Heat transfer to pulsating turbulent gas flow. In: Seventh International Heat Transfer Conference, vol. 3. Munich, W.G. (1982), pp. 105–110.

NOMENCLATURE

U Air velocity, m/s.

W Width of water pan, m

x Moisture content of air, kg_{wv} /kg_a

Greek symbols

α Thermal diffusivity, m^2/s

β Ackermann correction factor, -

Φ Relative humidity, %

μ Dynamic viscosity, kg/m.s

ν Kinematic viscosity, m^2/s

ρ Density, kg/m^3

Subscripts

a air

atm atmospheric

db dry bulb

ev evaporation

in upstream

out downstream

m mean

sat saturation

v water vapor

w water

ws water surface

∞ far from the water surface

A Surface area of water pan, m^2

c_p Specific heat at constant pressure, J/(kg.K)

D Mass diffusion coefficient, m^2/s

f Frequency, 1/s

H Height of wind tunnel, m

h Heat transfer coefficient, W/(m^2 K)

h_m Mass transfer coefficient, m/s

i_{fg} Latent heat of vaporization, J/kg

k Thermal conductivity, W/(m K)

L Length of water pan, m

L_c Characteristic length, m

$\dot{m}_a$ air flow rate, kg/s

$\dot{m}_{ev}$ Evaporation rate, kg/s

Nu Nusselt number (h L_c/k), -

P Pressure, Pa

Pr Prandtl number (Pr = ν /α), -

Q Rate of heat transfer, W

q" Heat flux, W/m^2

R_v vapor gas constant, J/kg K

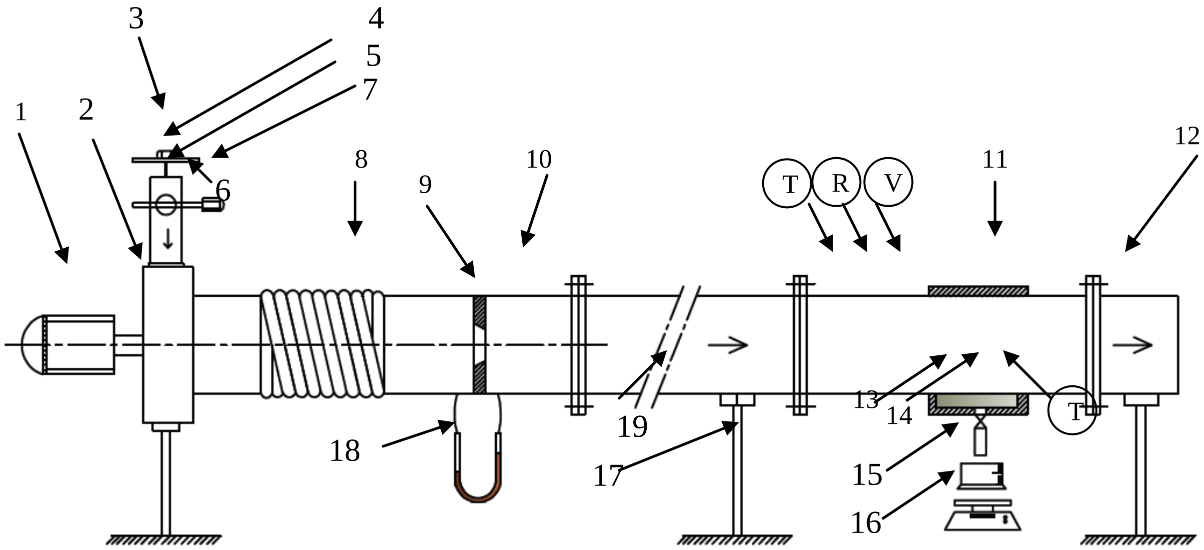
Re Reynolds number (Re = U_m L_c /ν), -

Sc Schmidt number (Sc= ν / D), -

Sh Sherwood number (Sh= h_m L_c/D), -

St Strouhal number, (St = f L_c /U_m), -

T Temperature, K



والشكر لله رب
العالمين

1- Electric motor
2- Blower
3- Throttle valve
4- Inlet section
5-Circular plate
6- Pulsator
7- DC motor with speed controller
8- Flexible connection

9- Orifice plate
10- Circular duct
11- Test section
12- Exit section
13- Glass wool
14- Water pan
15- Calibrated glass
16- Digital balance

17- Stand
18- U tube manometer
19- Entrance section
R Relative humidity
V Velocity
T Temperature

Fig. (1) Schematic of the Experimental Test Rig

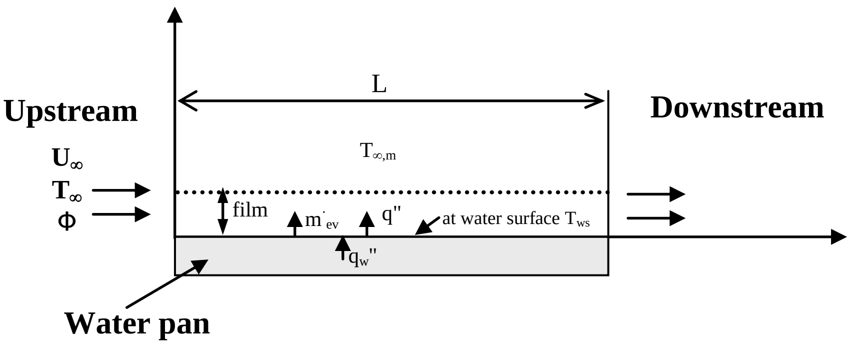


Fig. (2) Schematic of the Physical System

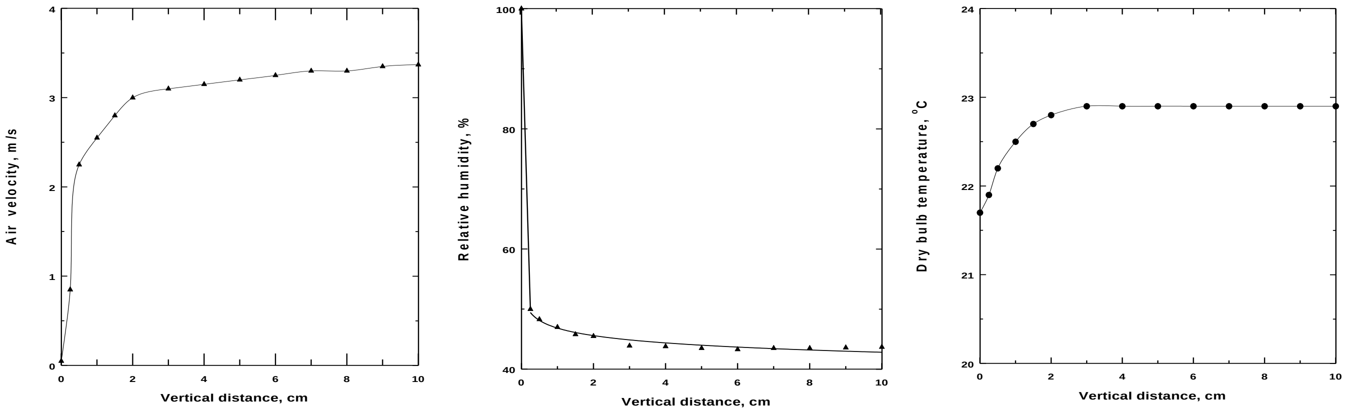


Fig. (3) Variations of the measuring parameters along the vertical distance downstream of the water pan.

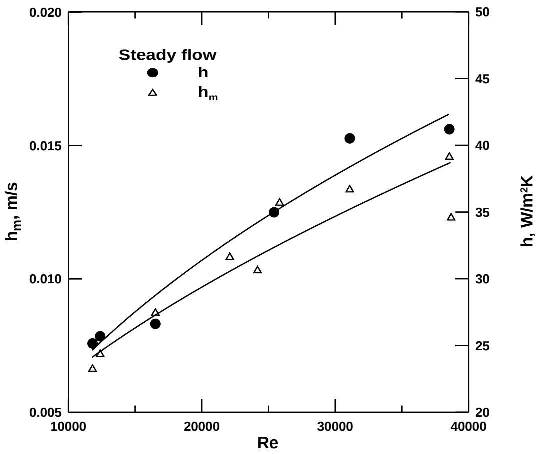


Fig. (4) Heat and mass transfer coefficient versus Reynolds number for steady flow.

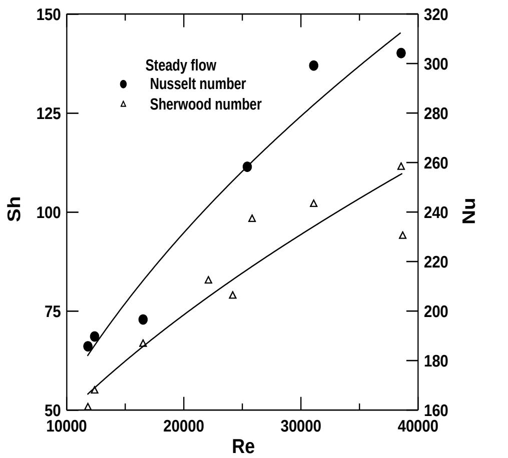


Fig.(5) Sherwood and Nusselt numbers versus Reynolds number for steady flow.

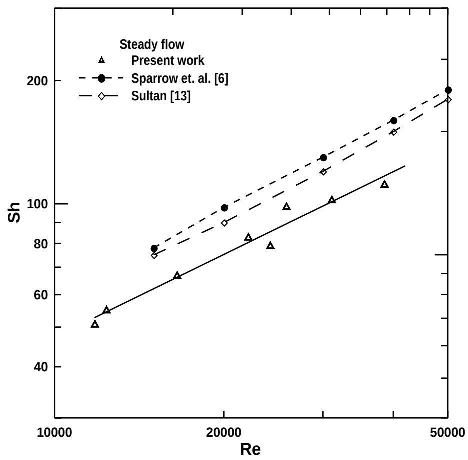


Fig. (6) Comparison between the present results of steady flow and previous work.

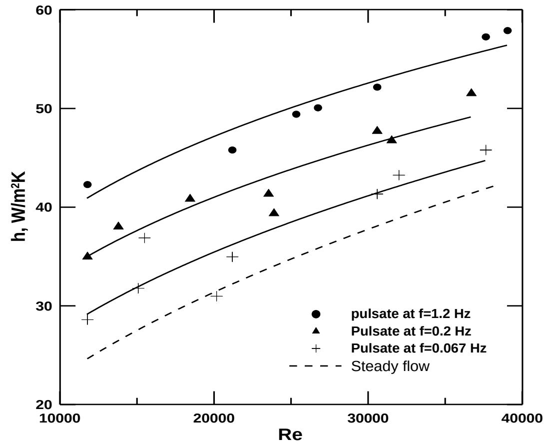


Fig. (7) Heat transfer coefficient versus Reynolds number for pulsating flow at different frequencies.

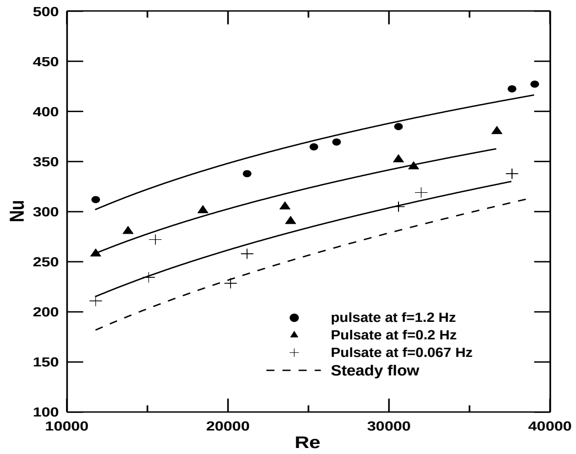


Fig.(8) Nusselt number versus Reynolds number for pulsating flow at different frequencies

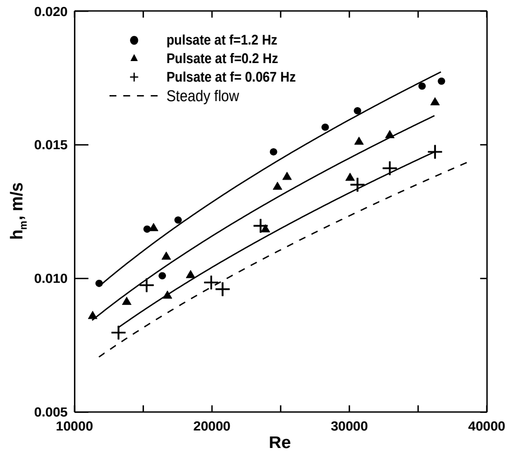


Fig. (9) Mass transfer coefficient versus Reynolds number for pulsating flow at different frequencies

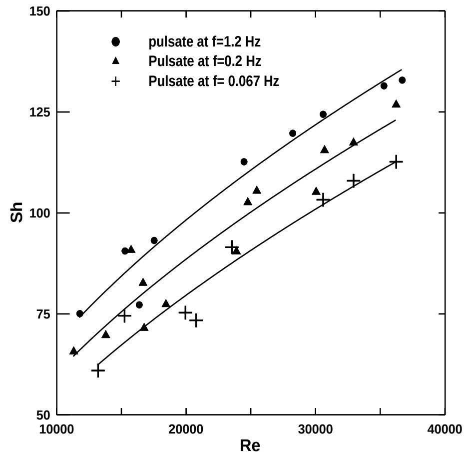


Fig.(10) Sherwood number versus Reynolds number for pulsating flow at different frequencies

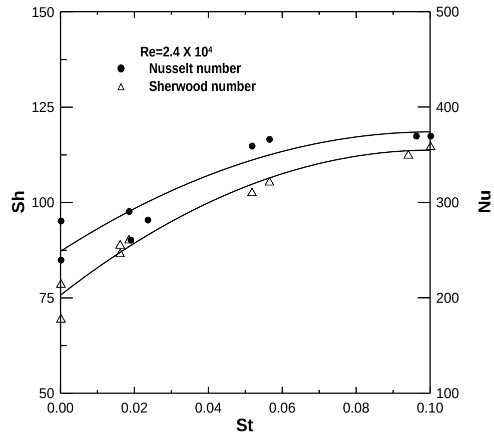


Fig. |(11) Nusselt and Sherwood numbers versus Strouhal number at Re=2.4×10^4

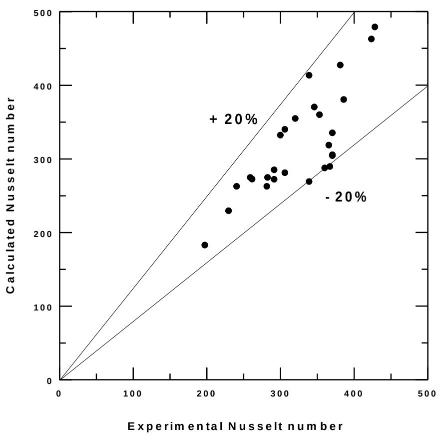


Fig. (12) Comparison between the proposed correlation for Nusselt number and experimental data

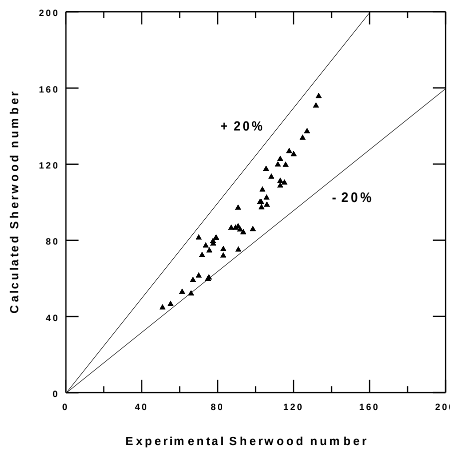


Fig.(13) Comparison between the proposed correlation for Sherwood number and experimental data