EXPERMENTAL INVESTIGATION OF AIR HUMIDIFICATION THROUGH CONVERGENT AND DIVERGENT RECTANGULAR DUCT

"دراسة معملية لترطيب الهواء خلال نفق مستطيل متقارب و متباعد "

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خلاصة البحث:

فى هذا البحث تم عمل دراسة معملية لإنتقال الحرارة والكتلة الأنيين لسريان منتظم لهواء يمر على سطح ماء موجود فى حوض مستطيل الشكل ومثبت فى قاعدة نفق هوائى متقارب ومتباعد التكوين ولاحداث هذا التكوين يتم تغيير مساحة مقطع النفق الهوائي بوضع اجنحة مستوية بزوايا ميل مختلفة في سقف النفق الهوائي .وقد تم حساب رقم نوسلت ورقم شرويد عند قيم مختلفة لرقم رينولدز (40000 > Re>11000) وزوايا ميل مختلفة ($4.8^o > \theta > 0.0$). وقد أظهرت النتائج أن رقم نوسلت ورقم شرويد يزيد مع زيادة رقم رينولدز في كلا من النفق الهوائي المتباعد والمتقارب وتصل هذه الزيادة الي 150% عند رقم رينولدز Re=11446. وقد تم استنتاج صيغة رياضية لا بعدية لحساب كلا من رقم نوسلت ورقم شرويد .

ABSTRACT

Evaporation rate of water into air flow over water panel which fixed in the floor of a rectangular wind tunnel is investigated, experimentally. Flow passage area variation is constructed by fixing wedges in the ceiling of the test section; in front of water panel. Two configurations are achieved for flow passage in the test section; convergent configuration and divergent configuration. Convergent configuration increases air velocity accordingly there is a negative pressure gradient, and divergent configuration decreases air velocity accordingly there is a positive pressure gradient. The required measurements for velocity, temperature and humidity, are taken at upstream and downstream over the water panel. Also, the amount of water evaporated through each experiment was measured. The effect of flow passage area variation on the evaporation rate is investigated at different values of Reynolds number and its impacts on Sherwood number are considered. The ranges tested for Reynolds number is (40000>Re>11000). The obtained experimental results show that, Nusselt and Sherwood numbers increases with increasing Reynolds number for both convergent and divergent configurations. Sherwood number takes higher values in case of convergent configuration than the divergent configuration. The maximum enhancement in Nusselt about 150% and Sherwood numbers is about 140%at Re=11446 and area ratio parameter(β=0.05) in the considered operating ranges. Empirical correlations for Nusselt and Sherwood numbers as a function of Reynolds number ,prandtl and Schmidt numbers.

Key words: Heat and mass transfer, Humidification, moist air, duct flow

1. INTRODUCTION

The evaporation of a liquid into a gas consisting of its own vapour, air or a mixture is important in many chemical and industrial engineering processes. The gas can be

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considered as a binary mixture. Simultaneous heat and mass transfer govern the process of evaporation from a water surface into airflow. The boundary layer incorporates a resistance for both heat and mass transport processes. At low temperatures the process is mainly heat transfer controlled, since the concentration difference across the boundary layer is large as compared to a rather small temperature difference. At low temperature difference, the required heat for vaporization of water at the surface to the flowing air is transferred by convection and radiation from the flowing air stream and by conduction from the water under the surface due to the sensible energy of the water itself.

Sultan (2002) Experimentally investigated the rate of evaporation from a water panel to airflow in a rectangular 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 is considered at different values of Reynolds number. Convergent pressure gradient leads to an increase in the mass transfer coefficient.

Hesham M.Mostafa (2002) Investigated, theoretically and experimentally simultaneous heat and mass transfer from a horizontal water panel to airflow inside a rectangular wind tunnel. In the theoretical model, the flow of air over a water panel is assumed to be laminar and steady in Cartesian coordinates. Air flowed inside a rectangular tunnel over a water panel, which is fixed in the floor of the tunnel. Steady and pulsating airflow was considered. The effect of pulsation frequency on the heat and mass transfer coefficients is investigated, experimentally. The results show that, Nusselt and Sherwood numbers increase with increasing Reynolds number for pulsating flow and steady flow. For pulsating flow,the increase in Nusselt and Sherwood numbers is higher than these for steady flow by a considerable value.

Ahmed F.khudheyer (2011) investigated, experimentally heat and mass transfer from or to a horizontal moving water film to airflow flowing over the film inside a rectangular duct wind tunnel. To perform the experimental study for this phenomenon air flows inside a rectangular duct over a moving water film flow inside the duct, which is fixed in the floor of the tunnel. Therefore, some of the water film is evaporated from the water surface to the air flowing over it, the amount of water evaporated through each experiment is measured. The results show that, Nusselt and Sherwood numbers increase with increasing Reynolds number.

Eames et al., (1997) 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 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 (1996). It was found that the combined heat and mass transfer in film evaporation can be enhanced by adding rods to the plate surface to agitate mechanically the flowing water film and air stream.

Kuan-Tzong Lee, (1998) studied mixed convection heat transfer in horizontal rectangular ducts with wall transpiration effects. A detailed numerical study was carried out to examine the effects of wall transpiration on laminar mixed convection flow and heat transfer in the entrance region of horizontal rectangular ducts. The vorticity velocity method was employed in the formulation of both thermal boundary condition of uniform heat flux and uniform wall temperature. Predicted result was presented for airflow over a wide range of governing parameters. In this work the wall Reynolds number are varied from -2 (suction) to 4 (injection) and Rayleigh number was ranging from 0.0 to $2x10^5$ for aspect ratio (0.2, 0.5, 1.2 and 5).

M. Haji and L. C. Chow (1988) studied heat and mass transfer with dehumidification in laminar boundary layer flow along a cooled flat plate. The effects of dehumidification, from laminar flow humid air, over isothermal cooled flat plate have been investigated using the similarity principle. The results obtained for both saturated and unsaturated humid air is compared with those of dry air. It can be concluded that dehumidification hasa significant effect of both sensible and latent heat transfer as well as the friction factor. However, the difference in heat transfer coefficient of humid and dry air is reduced as the relative humidity is decreased.

Wel-monYan,(1995) studied transport phenomena of developing laminar mixed convection heat and mass transfer in inclined rectangular ducts. The developing laminar mixed convection heat and masstransfer in inclined rectangular ducts has been studied numerically, with six independent parameters; Prandtl number Pr, mixed convection parameter (Pr Gr/Re), modified Rayleigh number Ra*, buoyancy ratio N, Schmidt number Sc anaspect ratio.Typical developments of velocity, temperature and concentration profiles are shown to include the limiting cases of horizontal and vertical rectangular ducts.

Sheikholeslami and Watkinson (1992) examined the effect of steam content on the rate of evaporation of water into moist air and superheated steam at elevated temperatures. Their experimental results confirmed the existence of inversion point temperature, above which the rate of evaporation of water increases with

Increasing in the steam content of the medium.

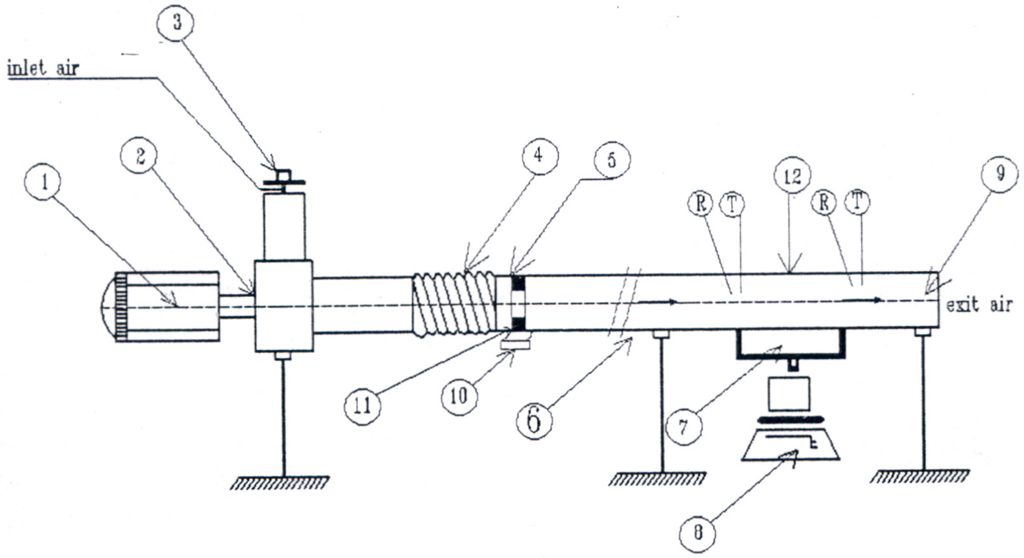
Schwartze and Brocker (2000) carried out a theoretical study of the evaporation of water into its own vapour, air and a mixture thereof. They introduced refined definitions of the inversion point temperature they defined firstly a local inversion temperature as a temperature at which the evaporation rates from an infinitesimal area for two gas flows with different vapour mole fractions is equal. Then, they defined the apparent inversion temperature as the temperature at the beginning of evaporation area for which. average evaporation rate into two gas flows with different vapour mole fractions is equal. These new and precise definitions enable a more precise description of the inversion temperature phenomenon and permit to explain the variation of the inversion temperature values obtained in earlier studies.

In the present work, the influence of area ratio parameter of the wedges on heat and mass transfer from water panel to air flow inside the tunnel(Nusselt and Sherwood numbers) is investigated experimentally at different values of Reynolds number .

2. EXPERIMENTAL TEST RIG

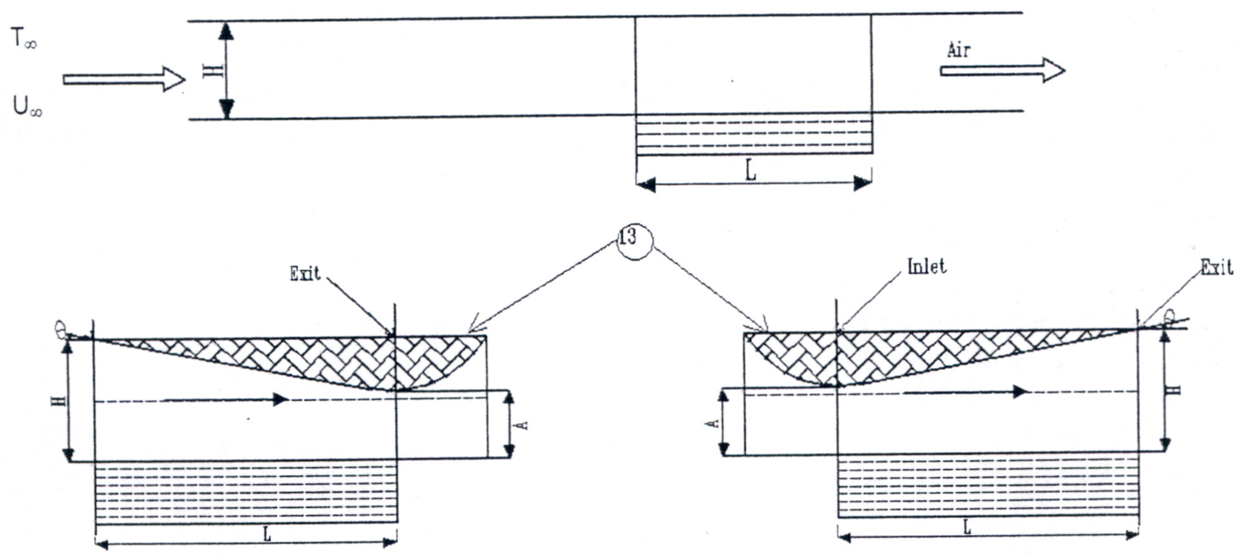
The schematic diagrams of the experimental test rig and the test section are shown in figure (1) and figure (2). Test rig is consisted mainly of a rectangular wind tunnel with different cross sectional area is constructed by fixing wedges in the ceiling of the test section ;in front of water panel. Air withdrawn from the ambient by an air blower (2), which is connected the rectangular wind tunnel using flexible connection (4) to absorb the vibration from the fan The inlet section is circular tube (5).At the beginning of the inlet section a throttle plate valve (3) is fixed and used to control the amount of airflow rate through the tunnel. Air is passed through the test section of 2000 mm long with 120 mm×100 mm cross sectional area. A water panel (7) of 600 mm long, 120 mm wide, 20 mm depth is located at 1200 mm from the test rig inlet.

A wooden wedge (13) is used to create convergent and divergent configurations are achieved for flow passage in the test section. The wooden wedge is fixed in the ceiling of the rectangular cross section of the test rig above the water panel as shown in figure (2).Five wooden wedge "for both convergent and divergent configurations "are used, whose dimensions are given in Table (1).The mass of water in the panel is measured before and after each experimental by using a digital mass balance (8) with a sensitivity of 0.1 g. The static pressure at inlet and exit of the test section is measured by a micro – manometer through the pressure taps (10) with scale division 1Pa.



1)Electric motor for blower.
2) Air blower.
3) Throttle valve
4)Flexible connection
5)Circular tube
6)Entrance region
7)Water panel.
8)Mass balance.
9)Air exit.
R - Relative Humidity
10)Micro - manometer
11)Orifice meter
T - Dry Bulb Temperature
12)Test Section

Figure (1) Schematic diagram of the experimental test rig.



a) Convergent configuration

b) Divergent configuration

Figure (2) Schematic diagram of test section.

Table (1) dimensions of wooden wedges

A m	H m	θ Deg.	β=(A-H)/L
0.1	0.1	0	0.0
0.9	0.1	0.95	-0.0166
0.8	0.1	1.9	-0.033
0.7	0.1	2.86	-0.05
0.6	0.1	3.81	-0.0667
0.5	0.1	4.8	-0.08

Where :

A exit hieght of the test section

H hieght of the test section.

θ wedge angle.

L wedge length.

β Area ratio parameter

3. EXPERMENTAL MEASURMENTS AND METHODOLOGY

Before starting the run, water panel is first cleaned, then the water panel is filled with a known mass of water. Air was drawn from the ambient and controlled by using a throttle valve to a certain value. All required measurements for air velocity, relative humidity and dry bulb temperature at upstream and downstream locations are achieved The average water panel temperature was measured during the experiment . Also, the duration time for each experiment was measured. Mass of water in the panel was measured before and after each experiment by using a digital mass balance with a sensitivity of 0.1 g. . Relative humidity was measured by using a hygrometer sensor (type Testo 605-H1), with a resolution of 0.1%. All temperatures were measured by using thermometer.

To analyze the obtained measurements data, the air and water vapor in the wind tunnel was treated as ideal gases. The air at the water-air interface is saturated. Then, the water vapor density at this position is

$$\rho_{v,ws} = \frac{P_{v,ws}}{R_v T_{v,ws}} \tag{1}$$

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 .

The water vapor pressure far from the water surface was determined from;

$$P_{v,\infty}=(RH)P_{sat} \tag{2}$$

Where RH and P_{sat} are the relative humidity and saturation pressure of water vapor in the flowing air at a temperature equal to dry bulb density far away from the water surface is;

$$\rho_{v,\infty} = \frac{P_{v,\infty}}{R_v T_{db}} \tag{3}$$

Then, the net amount of heat transfer can be defined as the difference between the outlet and inlet conditions as;

$$Q = \dot{m}_{air} (H_2-H_1) \tag{4}$$

Where;

Q = rate of heat transfer, W

m_{air} = airflow rate, kg/s.
H_2= enthalpy of exit air, j/kg
H_1=enthalpy of inlet air, j/kg.
Heat and mass transfer coefficients (h and h_m) can be defined as;

$$h = \frac{Q}{A(T_\infty - T_w)} \qquad (5)$$

$$h_m = \frac{m_{ev}}{A(\rho_{v,ws} - \rho_{v,\infty})} \qquad (6)$$

where: A = surface area of the water panel (A=W.L), m^2
W = width of the water panel, m
L = length of the water panel, m
T_w = water temperature,°C
T_∞ = ($T_{\infty,1}$+ $T_{\infty,2}$)/2 = average air temperature, °C.
$T_{\infty,1}$ = air temperature at upstream, °C.
$T_{\infty,2}$ = air temperature at downstream, °C.
Nusselt number, Sherwood number, Reynolds number, Prandtl number and Schmidt number can be calculated from the following relations;

$$Nu = \frac{h\ D_H}{k}, \qquad Sh = \frac{h_m\ D_H}{D},$$

$$Re = \frac{U_{av}\ D_H}{\nu}, \qquad D_H = \frac{4\,(W.H)}{2(W+H)},$$

$$Pr = \frac{C_p \mu}{k} \quad \text{and} \quad Sc = \frac{\nu}{D}$$

Where D, k, D_H and ν are diffusion coefficient of water vapor in air, thermal conductivity of air at T_∞,hydraulic diameter and kinematic viscosity for air at T_∞ respectively.

4. RESULTS AND DISCUSSIONS

In this section the obtained experimental results are presented and discussed. Accordingly, the experimental results, dry bulb- and wet bulb temperatures, local and average heat transfer, as well as mass transfer coefficients area ratio parameter and local, average Nusselt and Sherwood numbers are depicted. Also a comparison between the present results and that of previous available works are also presented.

4.1. Results of rectangular duct.

Figures (3) and (4) show the variation of the direct measured values of dry bulb temperature and relative humidity along the vertical distance at the end of water panel (downstream of the water panel) for steady flow. the dry bulb temperature has minimum value (temperature of water panel) at zero vertical distance. and takes an asymptotic value far from the water panel and relative humidity has a maximum value at zero vertical distance and then decrease far from the water panel.

Figures(5) and (6)show the relation between Nusselt number and Sherwood number versus Reynolds number in case of zero pressure gradient for air flow over water panel ,it is noticed that Nusselt and Sherwood number increase with increasing Reynolds number.
Figure (7) shows the variation of Sherwood number versus Reynolds number for the present work compared with previous works, for steady airflow. One case from the

Previous works are considered, for airflow inside a rectangular duct and airflow in a rectangular wind tunnel over a horizontal water panel . It is observed that, the present results have a good agreement with the previous results.

4.2.Results of convergent configuration

Figure (8) shows the relation between Nusselt number versus Reynolds number ,Nusselt number increase with increasing Reynolds at different values of area ratio parameter for convergent configuration.

Figure (9) shows the relation between Sherwood number versus Reynolds number ,Sherwood number increase with increasing Reynolds at different values of area ratio parameter for convergent configuration.

Figures (10) and (11) show the relation between Sherwood and Nusselt numbers versus area ratio parameter,β for different value of Reynolds number , Sherwood and Nusselt numbers are increase with area ratio parameter until it reaches a certain value at which it begin to decrease . This may be due to the migration of water bubble with the moist airflow. Because of this migration, the rate of evaporation, and in turn, rate of heat transfer decrease. Also the negative pressure gradient associated with the flow through the convergent configuration extract the water vapour out of the pan and leads to the increase of evaporation rate and the decrease in water panel temperature due to evaporation tends to decrease the diffusion coefficient.

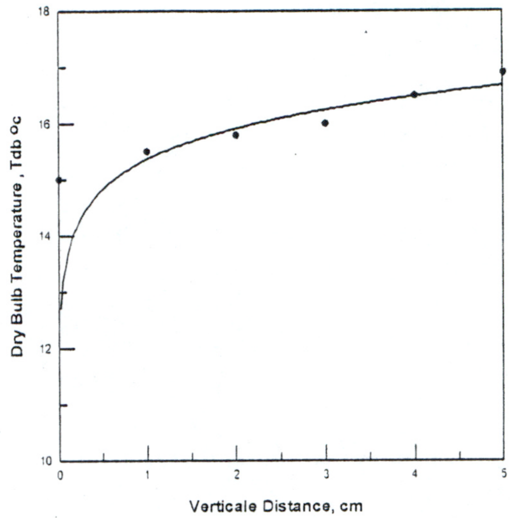


Figure (3) variation of dry bulb temperature with vertical distance.

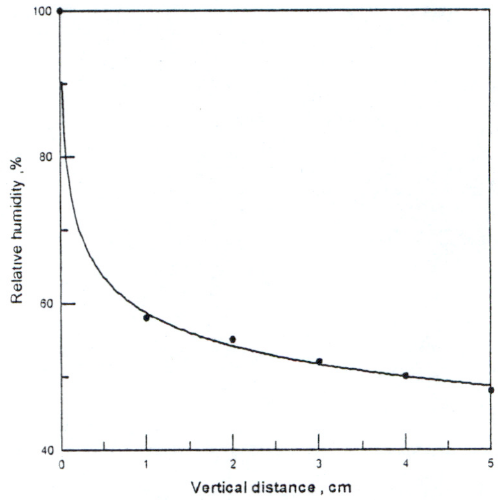


Figure (4) variation of relative humidity with vertical distance.

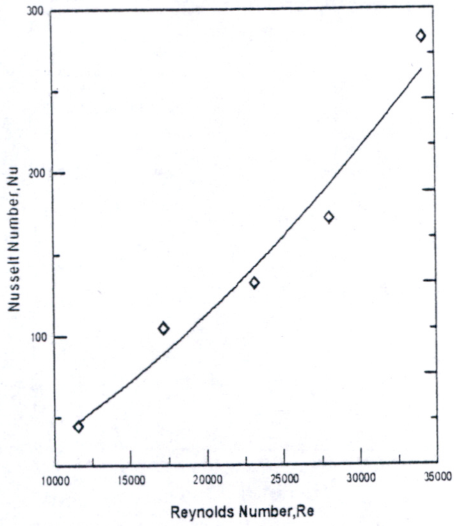


Figure (5) Relation between Nusselt number and Reynolds number for rectangular duct (β=0).

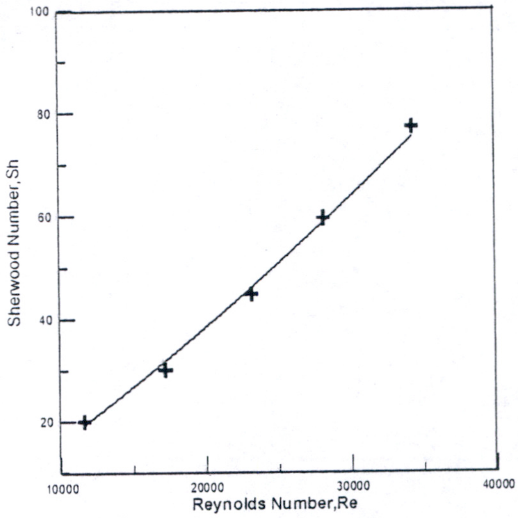


Figure (6) Relation between sherwood number and Reynolds number for rectangular duct (β=0).

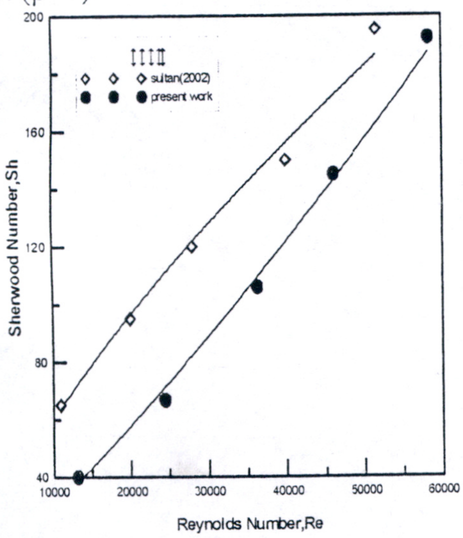


Figure (7) comparison of the present experimental data with the data of sultan(2002).

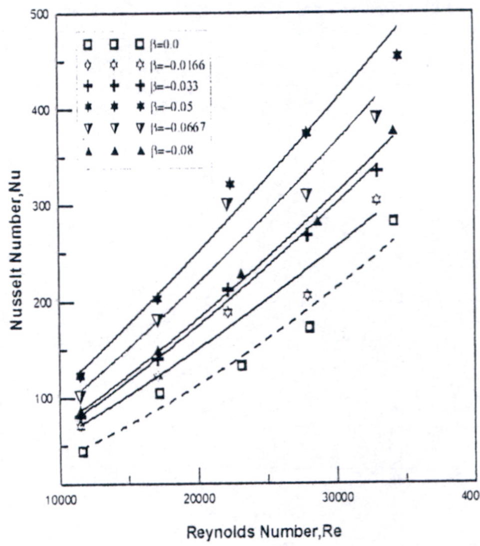


Figure (8) Relation between Nu and Re at different value of β for convergent configuration.

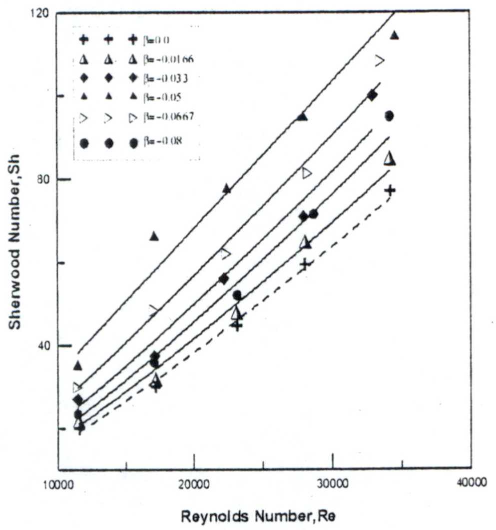


Figure (9) Relation between Sh and Re at different value of β for convergent configuration.

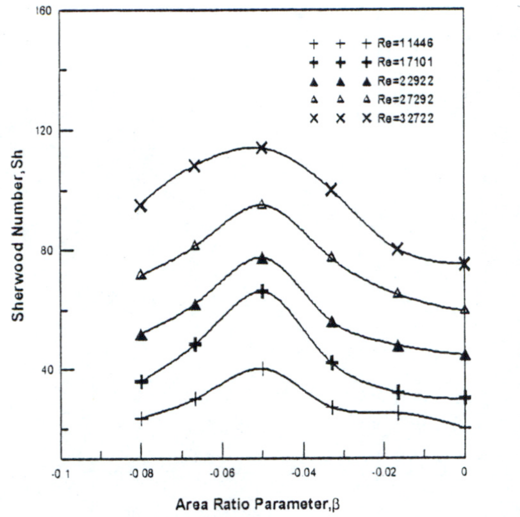


Figure (10) Relation between Sh and β at different value of Re for convergent configuration.

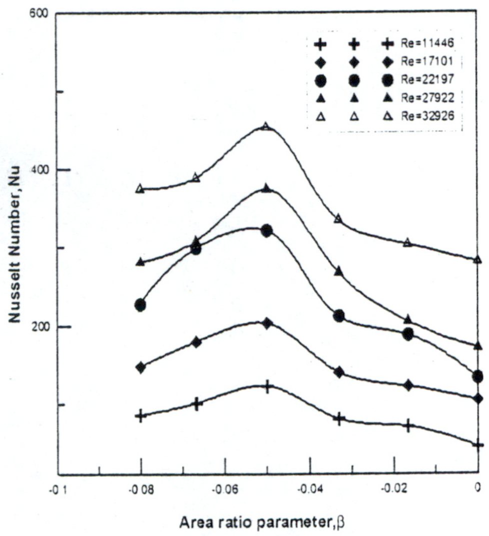


Figure (11) Relation between Nu and β at different value of Re for convergent configuration.

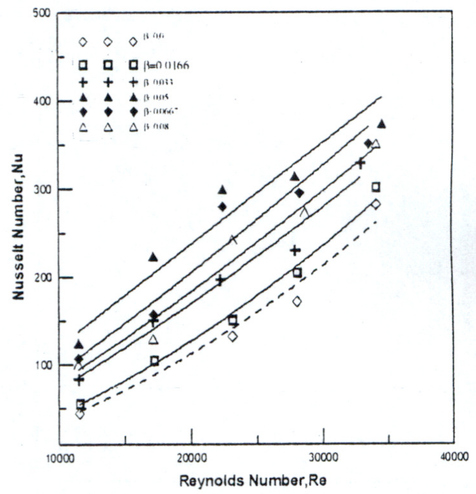


Figure (12) Relation between Nu and Re at different value β for divergent configuration.

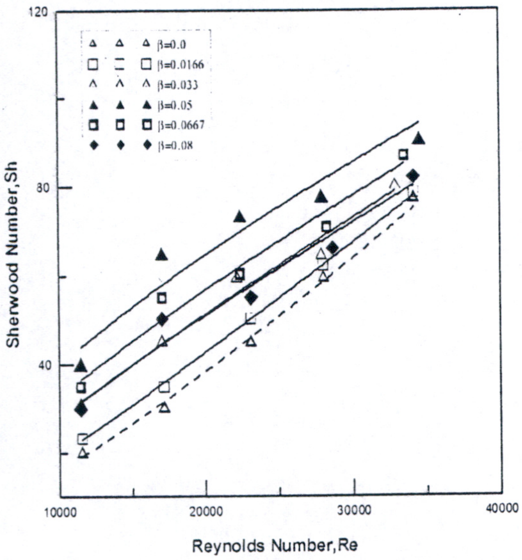


Figure (13) Relation between Sh and Re at different value of β for divergent configuration.

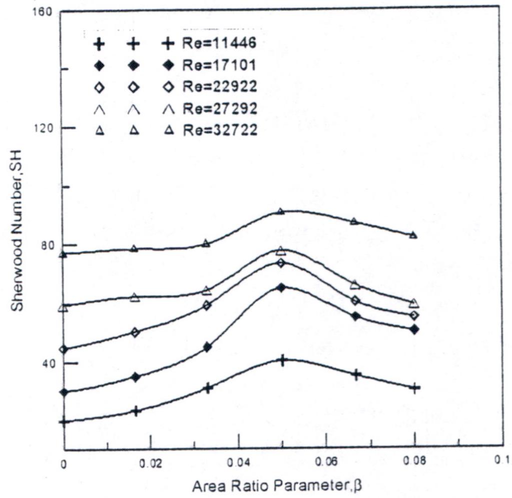


Figure (15) Relation between Sh and β at different value of Re for divergent configuration.

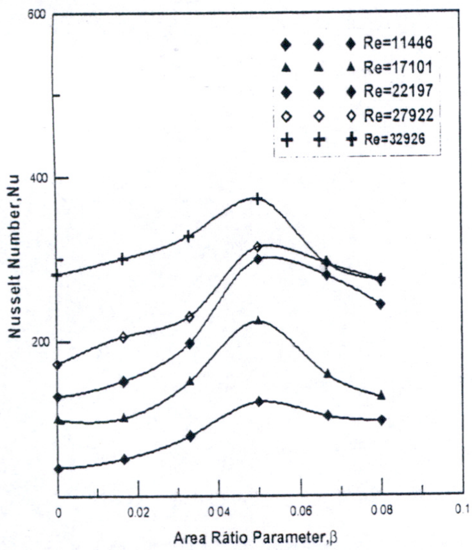


Figure (14) Relation between Nu and β at different value of Re for divergent configuration.

4.3. Results of divergent configuration

Figures (12) and (13) show the relation between Nusselt and Sherwood numbers versus Reynolds number ,Nusselt and Sherwood numbers increase with increasing Reynolds at different values of area ratio parameter for divergent configuration..

Figures (14) and (15) show the relation between Nusselt and Sherwood number versus area ratio parameter,β for different value of Reynolds number .Nusselt and Sherwood number are increase with increasing area ratio parameter until it reaches a certain value at which it begins to decrease .This may be due to the migration of water bubble with the moist airflow with the higher value of area ratio parameter.

Because of this migration, the rate of evaporation, and in turn, rate of heat transfer are decreased.

The following relations correlate the experimental results, where Nusselt and Sherwood numbers are expressed as functions of Reynolds number and area ratio parameter:-

For convergent configuration:

$$Nu=(0.0004-0.017479\beta-0.0159\beta^2)Re^{1.25}Pr^{0.33}$$
$$Sh=(0.001253-0.035175\beta-.355\beta^2)R^{1.0325}Sc^{0.33}$$

For divergent configuration :

$$Nu=(0.00128+0.0442\beta-0.3757\beta^2)Re^{1.14}Pr^{0.33}$$

$$Sh=(0.00063+0.01504\beta-0.1402\beta^2)Re^{1.1}Sh^{0.33}$$

These correlations are valid for
$0.0 \leq \beta \leq 0.1$ and $40000 > Re > 11000$

CONCLUSIONS:

An experimental investigation was conducted to study the effect of pressure gradient on the heat and mass transfer for turbulent flow of air at different values of Reynolds number in a rectangular duct over a water panel.
It is concluded that :
1. Convergent configuration leads to increase in the mass and heat transfer coefficient in all cases, and the maximum increase is found to be 150% and 140% respectively at Re=11446 and β=0.05.
2. The maximum enhancement in mass and heat transfer coefficient in case of divergent Configuration is 122.2% and 133.3% respectively at Re=11446 and β=0.05.
3. Nusselt and Sherwood numbers increase with increasing Reynolds number for convergent and divergent configuration.

NOMENCLATURE

A : Surface area of water panel, m^2
β : Dimensionless parameter, -
D : Diffusion coefficient, m^2/s
H : Height of wind tunnel, m
h : Heat transfer coefficient, W/(m^2. °C)
h_m : Mass transfer coefficient, m/s
k : Thermal conductivity, W/(m. °C)
L : Length of water panel, m
$m^{\cdot}_{air}$: airflow rate, kg/s
$m^{\cdot}_{ev}$: Evaporation rate, kg/s
Nu : Nusselt number ($Nu=hD_H/k$), -
Q : Rate of heat transfer, W
T_∞ : temperature
Cp : Specific heat at constant pressure, J/(kg.°C)
U : Air velocity, m/s

Greek symbols

μ : Dynamic viscosity, kg/m.s
ν : Kinematic viscosity, m^2/s
ρ : Density, kg/m^3

Subscripts

air air
sat saturation
av average
v water vapor
db dry bulb
w water
∞ far from the water surface
ws water surface

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