

Investigations of Natural Convective Heat Transfer in Rectangular Thermal Passages

Hussain H. Al-Kayiem, Ahmed K. Hussein, Toh Seng Peow

Abstract—The evaluation of the convective heat transfer of flow in passages with rectangular cross section is still of interest for the heat transfer investigators, as in the air heater solar collectors. The aim of this paper is to present investigation results on the natural convection heat transfer in a solar air heater. The effect of the channel length as heat transfer surface and the inclination of the passage were investigated. The results were obtained experimentally and theoretically. For that, an experimental test rig was fabricated with channel lengths of 1m, 1.5m, and 2m. For each length, the air outlet and inlet temperatures, absorber and cover temperatures, solar radiation intensity and air flow rate were measured at 10°, 30°, 50°, 70°, and 90° tilt angles. Measurements were recorded every 2 hours interval to investigate the transient behavior of the system. The experimental and theoretical results are presented in terms of Nu number versus Ra number and discussed. The percentages of differences between experimental and theoretical results are within the margin of 6% to 13%, effectively. It is recommended to extend the investigation to study the same configurations with different artificial surface roughing by ribs or pins.

Keywords—Convective heat transfer, Flat plate, Natural convection, Passage flow, Solar energy.

I. INTRODUCTION

SOLAR energy collectors are a type of heat exchangers that transform solar radiation energy to internal energy of the transport medium. The major components of any solar system are the solar collector, which absorbs the incoming solar radiation, converts it into thermal energy, and transfers this energy as heat to a working fluid (usually air, water or oil) flowing through the collector.

Air heating flat-plate solar collectors are usually rectangular passages with two openings at both of their ends, inlet and outlet. The main engineering concept behind the air heating solar collector is natural convection. The most common applications of air heating solar collectors are the domestic heating, solar dryer, solar chimney and Trombe Wall.

The evaluation of the outlet temperature of flow in passages with rectangular cross section is still of interest for the convective heat transfer investigators. Many parameters are involved in such case, for example the passage length, gap size, inclination angle, walls temperature, ambient conditions like temperature and humidity, insulation, etc. Accordingly,

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correlating the outlet temperature to the inlet at different conditions will simplify the mathematical modeling of the phenomena.

Many research studies, in terms of experiments, analytical and numerical simulations have been carried out related to the evaluation of the transfer of radiant energy to kinetic energy in the working fluids of thermal passages. Most of them involved with the solar chimneys and the Trombe walls. Evaluation of the performance of special chimney configuration, such as roof solar collectors and Trombe walls had been made to achieve optimum and energy conservation in buildings (e.g. [1]-[4]). Reference [5] showed that there is an optimum chimney length/gap width ratio for maximum air flow rate for the 1.95m high and variable width chimney which was electrically heated. Reference [6] had proven experimentally that the velocity and temperature magnitudes in width size varying from 2.5 to 20cm of a vented 2.2m high Trombe wall are functions of the gap width, ambient air temperature, insulation rate, wall temperature and the elevation above the Trombe wall inlet duct. Reference [7] developed a steady state analytical model for uniform wall temperature applied to a solar system consisting of a solar air heater connected to a conventional chimney.

Reference [1] used 3D computational fluid mechanics (CFD) techniques to study the parameters that influence the performance of a Trombe wall. Using the concept of a thermal resistance network, [2] developed an analytical model to examine the effects of air gap and solar irradiation intensity on the performance of different chimneys assuming uniform heat flux on the heated wall, solved using matrix inversion. Reference [8] had also performed an assessment of the impacts of the inclination angle on the induced ventilation rate by CFD modeling technique.

It is clear from the literature that most of the works are focusing on the gap between the cover and the heated surface. The only work considered the effect of inclination is carried by [8], by CFD. The length effect of the passage has not been well covered in the literature. Also, the transient behavior of such configuration over the solar day is not studied at large range of inclination. It is essential to examine the system at low inclination as the case of ground solar chimney and absorbers up to 90°, as in the case of Trombe wall of vertical roof top solar chimney.

The objective of this work is to further investigate the heat transfer and the outlet temperature due to absorbed solar radiation at various lengths and inclination angles in rectangular passage. Three different lengths of absorber plates; 1.0, 1.5, and 2.0m long in the flow direction, and five different

tilt angles; 10, 30, 50, 70, and 90 degree inclination have been tested to investigate the natural convection heat transfer during the day hours. The investigations have been carried out experimentally and mathematically.

II. EXPERIMENTAL IMPLEMENTATIONS

The scope of this study was to investigate the natural convection heat transfer experimentally and analytically. For that, an experimental test rig was designed and fabricated in the solar research site (SRS) in Universiti Teknologi PETRONAS (UTP), as shown in Fig. 1. The selection of materials, the measuring instruments and the measurement procedure are presented in this section.

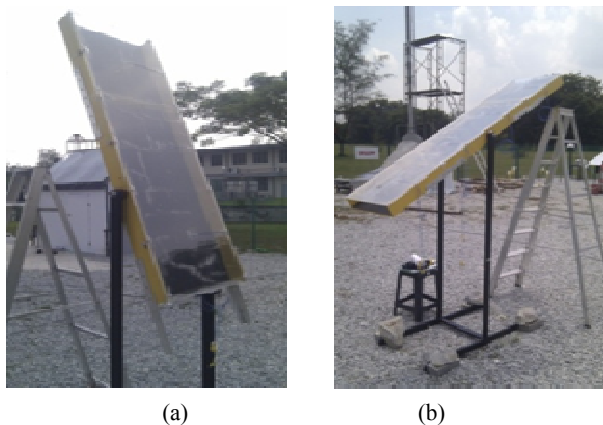


Fig. 1 (a) 1.5m wall channel at 70°, (b) 2.0m wall channel at 30°

A. Material Selection

Selection of proper material to construct a rig that is able to operate and stand the harsh rainy, sunny, and humid weather, like Malaysian weather, is a critical issue. Table I shows the material chosen for each component of the test rig and their justifications.

TABLE I
 MATERIAL SELECTION MATRIX

Component	Material	Justification
Double glazing	√ Perspex	• Ease of fabrication
	× Glass	• Sheer clarity • Lighter weight • Less fragile than glass
Absorber	√ Aluminum	• Ease of fabrication
	× Copper	• High thermal conductivity • Lighter weight and more economical than copper
Insulator and body	√ Plywood	• Low thermal conductivity
	× Metal	• Cost effectiveness

B. Instrumentations

The acquired data from each experiment are solar radiation measured on horizontal surface, ambient, inlet, and outlet mean air temperatures, inner and outer glazing temperature, absorber plate temperature at different locations. Meanwhile, the tools and equipment used for the data collection purpose was data logger; thermocouple wires type K for the surfaces temperature measurements, thermocouple probes for the fluid

temperature measurements, and a solar meter. All are connected to a GRAFTECH GL800 midi data logger.

C. Procedure

Thermocouples are placed along the length of the surface of the inner and outer glazing and absorber in order to take their mean temperature. Four thermocouples are placed along the wall surfaces of 2.0m length; three thermocouples are placed for 1.5m length; and 2 thermocouples for the case of 1.0m length. Measurements are acquired at every 2 hours interval, at 8am, 10am, 12pm, 2pm, 4pm, and 6pm to observe the transient behavior of the system.

Each experiment is commenced with 2m absorber length at 90° inclination angle, then at 70°, 50°, 30, and 10° from the horizontal axis. Due to the unpredictability of weather, the measurements are repeated for 3 consecutive days for each set of tilt angle at 2m, and the average is considered. After 2m configuration is completed, the same procedure is repeated for the 1.5m and 1.0m absorber length.

III. MATHEMATICAL SIMULATION

Fig. 2 shows the physical model of the double glazing solar collector that is developed for this study. To establish the mathematical model, the following assumptions are made:

- The entire system is in steady state condition.
- Friction losses are neglected due to the very low order of air flow velocity.
- The flow of air in the channel is laminar.
- The inlet air temperature is the same as the ambient temperature.
- All energy transfer processes through the glass cover, absorber plate and air channel are one dimensional in nature.

The heat transfer from the cover and absorber to the air stream is by natural convection.

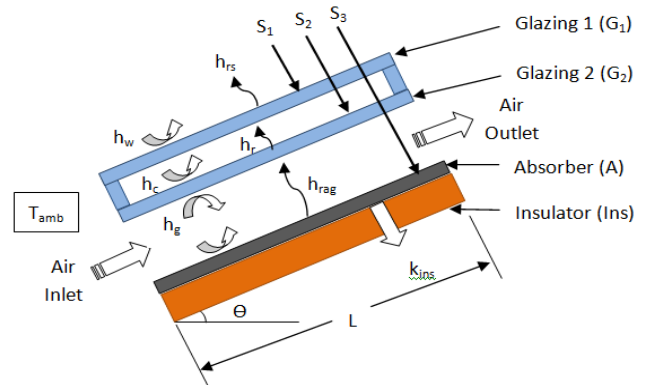


Fig. 2 Physical Model of Double Glazing Solar Collector

Subsequently, mathematical model was then developed based on the physical model above. The mathematical model is mainly used to predict the outlet air temperature of the system. The major parameters included in this study are: temperature of absorber, T_A , glazing surface, T_{G1} and T_{G2} , ambient air temperature T_{amb} , temperatures of air at inlet, T_{fi}

and outlet, T_{fo} , mean air temperature T_f , inclination angle from horizontal plane, θ and length of channel, L .

The following energy equations for the glazing surfaces, air passage, and absorber surfaces were established to find the desired correlation.

At Glazing-1,

$$S_1 + h_c(T_{G2} - T_{G1}) + h_r(T_{G2} - T_{G1}) = h_{rs}(T_{G1} - T_{amb}) + h_w(T_{G1} - T_{amb}) \quad (1)$$

At Glazing-2,

$$S_2 + h_g(T_f - T_{G2}) + h_{rag}(T_A - T_{G2}) = h_c(T_{G2} - T_{G1}) + h_r(T_{G2} - T_{G1}) \quad (2)$$

Substituting (2) into (1),

$$S_1 + S_2 + h_g(T_f - T_{G2}) + h_{rag}(T_A - T_{G2}) = h_{rs}(T_{G1} - T_{amb}) + h_w(T_{G1} - T_{amb}) \quad (3)$$

In Air Passage,

$$h_a(T_A - T_f) = h_g(T_f - T_{G2}) + q'' \quad (4)$$

At Absorber,

$$S_3 = h_a(T_A - T_f) + h_{rag}(T_A - T_{G2}) + \left(\frac{k_{ins}}{\Delta w_{ins}}\right)(T_A - T_{amb}) \quad (5)$$

The useful heat gain by air in equation (4) can be found from:

$$q'' = \dot{m} C_f (T_{fo} - T_{fi}) \quad (6)$$

where,

$$\dot{m} = C_d \rho_f A_0 \sqrt{\frac{2gL \sin \theta (T_f - T_{amb})}{(1 + A_r)^2 T_{amb}}} \quad (7)$$

where, $A_r = A_0/A_i$; $C_d = 0.57$, [9]

The mean air temperature is given as

$$T_f = \gamma T_{fo} + (1 - \gamma) T_{fi} \quad (8)$$

where, γ is constant for mean temperature approximation equal to 0.74 according to [10].

The solar radiation heat flux absorbed by glazing 1, S_1 , glazing 2, S_2 and absorber, S_3 are given as

$$S_1 = \alpha_1 I \quad S_2 = \tau \alpha_1 I \quad S_3 = \tau^2 \alpha_1 I$$

The radiation heat transfer coefficient between glazing 1 and sky is given as

$$h_{rs} = \frac{\sigma \epsilon_{G1} (T_{G1} + T_s) (T_{G1}^2 + T_s^2) (T_{G1} - T_s)}{T_{G1} + T_{amb}} \quad (9)$$

where sky temperature, [11]

$$T_s = 0.0552 (T_{amb})^{1.5}$$

Convective wind heat loss coefficient is given by [12], as

$$h_w = 5.7 + 3.8 V_w$$

where, V_w = wind velocity.

Radiative heat transfer coefficient between absorber and glazing 2, h_{rag} is given as

$$h_{rag} = \frac{\sigma (T_{G2}^2 + T_A^2) (T_{G2} + T_A)}{\left(\frac{1}{\epsilon_{G2}} + \frac{1}{\epsilon_A} - 1\right)} \quad (10)$$

Convective heat transfer coefficients between glazing 2 and air channel and absorber and air channel are given by (11 a) and (11 b), respectively.

$$h_g = \frac{Nu k_f}{L} \quad (11 a)$$

$$h_a = \frac{Nu k_f}{L} \quad (11. b)$$

where, for glazing 2 to air,

$$Nu = 0.6 (Gr \cdot \cos \theta \cdot Pr)^{0.2} \quad (12)$$

and from absorber plate to air,

$$Nu = 0.6 (Gr \cdot \cos \theta \cdot Pr)^{0.2} \quad (13)$$

where,

$$Pr = \frac{\mu_f C_f}{k_f}, \text{ and } Gr = \frac{g \beta S_2 L^4}{k_f \nu_f^2}$$

Properties of air, as working fluid, $\mu_f, C_f, k_f, \beta, \nu_f$ can be found from the empirical relationships proposed based on tabulated data from handbooks for air properties by using the mean temperature value of $T_m = 0.5(T_{G2} + T_f)$ for the glazing 2 to air case, and $T_m = 0.5(T_A + T_f)$ for the absorber to air case.

The air properties are obtained from the empirical relationships:

$$\mu_f = [1.846 + 0.00472 (T_m - 300) \times 10^{-5}]$$

$$C_f = [1.007 + 0.00004 (T_m - 300) \times 10^3]$$

$$k_f = [0.0263 + 0.000074 (T_m - 300)]$$

$$\rho_f = [1.1614 + 0.00353 (T_m - 300)]$$

$$\nu_f = \frac{\mu_f}{\rho_f} \beta = \frac{1}{T_m}$$

In the above mathematical model, $T_A, T_{G1}, T_{G2}, T_{fi}$ are the four unknowns temperatures that are to be estimated, and subsequently, T_m and T_{fo} can be obtained. However, in order to accomplish the solution, T_A, T_{G2} and T_{fi} are first required to be

known in order to calculate the outlet air temperature, T_{fo} and then the mean air temperatures, T_m . In other words, T_A , T_{G2} and T_{fi} are first needed to be guessed in order to find their real values through the method of iterations, which is very complex due to the many unknowns involved. Thus, in order to simplify this problem, T_A , T_{G2} , and T_{fi} are assumed to be known as measured values from the experiments. Hence, T_{fo} and T_m are left as the unknown parameters. The problem can be solved by substituting (8) into (5), getting

$$S_3 = h_a \{ T_A - [\gamma T_{fo} + (1 - \gamma) T_{fi}] \} + h_{rag} (T_A - T_{G2}) \left(\frac{k_{ins}}{\Delta w_{ins}} \right) (T_A - T_{amb}) \quad (14)$$

Rearranging,

$$T_{fo} = T_{fi} + \frac{T_A - T_{fi}}{\gamma} + \frac{h_{rag} (T_A - T_{G2}) - S_3}{h_a \gamma} - \frac{k_{ins} (T_A - T_{amb})}{h_a \gamma (\Delta w_{ins})} \quad (15)$$

The set of equations is converted into computer program **MATLABv7.1** environment. Primary guess value for T_{fo} was used and the final value was obtained via iterative solution.

IV. RESULTS AND DISCUSSION

Despite that average readings have been considered, it is necessary to clarify that the data collected are sometimes inconsistent, due to the inevitable sudden change of weather (cloudy and windy). Therefore, the following discussions are made based on the overall observation, rather than by individual configuration.

A. Insolation Measurement

The solar radiation during the repeated days of experiments is measurement, averaged and presented in Figs. 3 (a)-(c) for the 1.0m, 1.5m, and 2.0m length configurations, respectively. The covered period in each case is the sunny period during the day, from 8.00am to 6.00pm. The measurements were carried out by positioning the solarimeter perpendicular to the glazing.

It is clear that the insolation is going higher as the channel is oriented toward the horizontal configuration. Also, for all cases of inclinations, the maximum solar insolation is around 12:00 noon to 1:00pm. These measurement results will be used for further analysis in the coming paragraphs. The trend of the measured intensity is similar to the trend captured by the weather station in UTP, as reported by [13].

B. Thermal Analysis

Two indicators, the mean air temperature and the air outlet temperature are adopted in the recent investigation to analyze the convective heat transfer in the passage of the solar heater. The measurements have been carried out at various inclination angles and the results are presented in Figs. 4 (a)-(c) for the cases of 2.0m, 1.5m, and 1.0m, respectively.

Fig. 4 shows the variation of mean air temperature at various inclination of the absorber. The inlet air temperature is found to be stable and vary negligibly and it is almost $26 \pm 0.5^\circ\text{C}$. The trend of the air mean temperature is not following the solar radiation value, I only. Precise observation in the

results shows that the inclination angle influences the natural convective heat transfer in the passage.

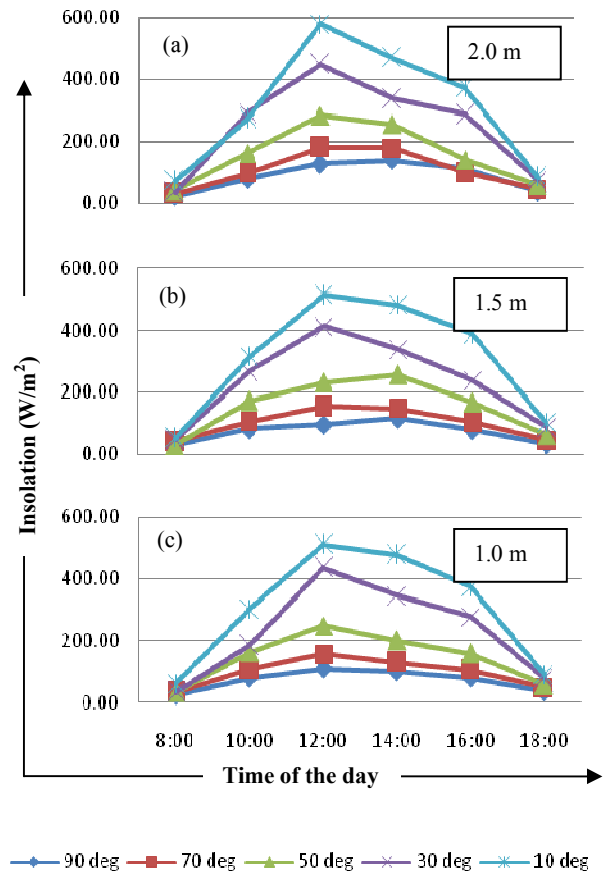


Fig. 3 The measured insulations during the day time at various inclinations of three heating passages

At 70° inclination, the heat transfer to air is dropped noticeably after 2:00pm. In spite that the highest mean temperature is achieved at 10° inclination, this is not the optimum indication for optimum convective heat transfer operation, where the mass flow rate is another factor that should be considered in the decision of optimum inclination. As the passage tilt is reduced towards the horizontal position, the mean and outlet temperatures are reduced noticeably due to the effect of ambient temperature which blown into the passage and pushing up the air in the passage to move out faster.

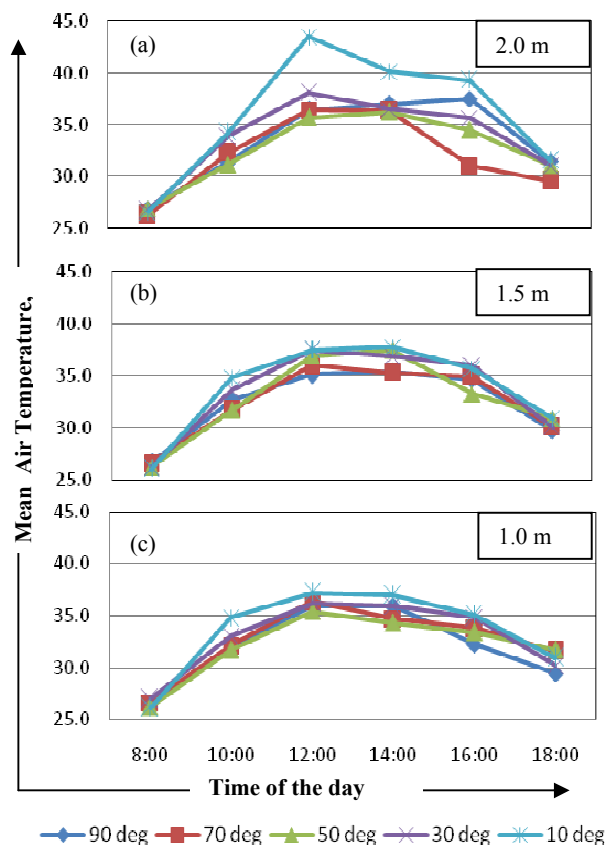


Fig. 4 The mean air temperature in the passage at various inclinations

The influence of the passage length on the mean air temperature is illustrated in Table II. Three different times during the sunny measurement period are selected, namely; 8:00am, 12:00noon, and 4:00pm.

TABLE II
THE MEAN AIR TEMPERATURE AT VARIOUS PASSAGE LENGTHS; A. AT 30° INCLINATION, B. AT 50° INCLINATION, C. AT 70° INCLINATION

a. Inclination at 30°				
	Length of passage			
	1.0 m	1.5 m	2.0 m	
8.00am	27.0	26.2	26.7	
12:00noon	36.2	37.5	38.1	
4:00pm	34.8	36.0	35.7	
b. Inclination at 50°				
	Length of passage			
	1.0 m	1.5 m	2.0 m	
8.00am	26.2	26.2	26.9	
12:00noon	35.4	36.9	35.7	
4:00pm	33.4	33.3	34.5	
c. Inclination at 70°				
	Length of passage			
	1.0 m	1.5 m	2.0 m	
8.00am	26.6	26.6	26.3	
12:00noon	36.4	36.0	36.4	
4:00pm	33.8	35.0	31.5	

In the morning, the mean air temperature in the passage is almost same for all cases of lengths and inclinations. That is because the solar intensities still low and the amount of thermal energy transferred to air stream is not significant. At 12:00noon, the mean air temperature is the highest in the day. It can be noticed that the air stream at 30° inclination is heated higher than 50° and 70° inclinations. That is because the incidence of the solar beam produces higher insolation at this angle than the other two angles, and consequently, higher absorber temperature and larger amount of thermal energy transfer to the air particles in the stream. The values of the mean air temperature are not highly consistent since the insolation is not considered in the evaluation criteria. It is recommended to evaluate the heat transfer performance in terms of insolation and useful gained heat. However, even by elementary evaluation based on the mean air temperature, it can be realized that the air stream gain more thermal energy as the absorber is longer as highlighted in Table II.

C. Mathematical Results

The predicted results from the mathematical model are validated by comparison with the experimental results. To perform the validation, L , θ , T_{G2} , T_A , and T_{amb} input to the program as they are obtained from the experimental measurements. Then, the computed output temperature, T_{fo} is compared with the experimentally measured outlet air temperature from the experimental data.

Table III shows the comparison between the theoretically predicted results and experimental results for all channels configurations at 12:00. In addition, the individual percentages of errors between the experimental and theoretical results are estimated at each inclination at the three investigated lengths, as:

$$\varepsilon\% = \left| \frac{T_{Exp} - T_{Theor}}{T_{exp}} \right| \times 100 \quad (16)$$

TABLE III
COMPARISON BETWEEN THEORETICAL AND EXPERIMENTAL RESULT OF THE OUTLET TEMPERATURE

L (m)	θ	Outlet air temperature, T_o (°C)		
		Theo.	Exp.	Error%
2.0	90.0	40.9	38.0	7.09
	70.0	43.1	37.9	12.06
	50.0	44.2	38.5	12.90
	30.0	46.6	40.5	13.09
	10.0	52.8	45.0	14.77
1.5	90.0	39.1	36.0	7.93
	70.0	41.5	37.4	9.88
	50.0	42.7	38.1	10.77
	30.0	45.3	39.4	13.02
	10.0	49.1	42.5	13.44
1.0	90.0	39.1	36.6	6.39
	70.0	39.7	37.5	5.54
	50.0	42.1	37.1	11.88
	30.0	43.8	37.8	13.70
	10.0	45.6	39.6	13.16
Mean % Error				11.0

The error values become higher as the inclination angle reduced toward the horizontal. This, in fact, is caused by wind traveling through the air passage. In most cases, the wind has greater impact on lower inclination angle configuration. This is because when the air passage is more parallel to the direction of the wind itself, the wind is able to travel through the passage more freely, thus displacing more hot air with cooler ambient air. Therefore, when the inclination angle of the system is decreasing, the percentage of error is naturally increasing. The error value of 14.8 is the maximum at 10° tilt of the 2.0 m passage length. The overall % of error for the case of 12.00noon is 11%.

Meanwhile, Table IV shows the individual mean percentage error from 08:00 to 18:00, respectively. The mean percentage of error calculated varied from 6.21% (at 08:00) to 11.0% at 12:00.

Time	08:00	10:00	12:00	14:00	16:00	18:00	Mean
Mean % Error	6.2	8.6	11.0	10.7	10.0	10.1	9.4

In addition, the variation of outlet air temperature against the inclination angles at different length for both theoretical and experimental results were plotted. Results measured experimentally and predicted mathematically at 12:00, are displayed in Fig. 5. From both Table III and Fig. 5, it can be seen that theoretical results of each configuration is noticeably higher than experimental results. It is believed that the losses to the atmosphere become higher as the system inclination angle is reduced. Since the mathematical model does not account for the losses, then the error increases accordingly. The results are indicating that the outlet air temperature is increasing as the passage is brought towards the horizontal.

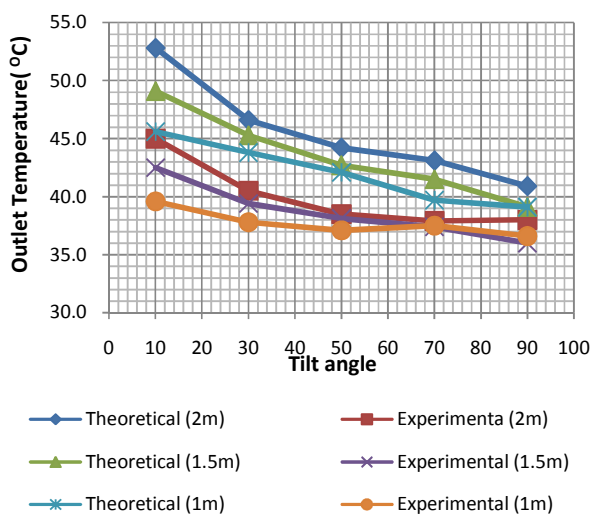


Fig. 5 Outlet air temperatures versus inclination angles

V. CONCLUSIONS

Experimental measurements and theoretical calculations were performed to study the convective heat transfer in the

passage of solar air heater. The investigations were carried out at 1.0m, 1.5m, and 2.0m passage lengths. Also, the air passage tilt angles of 10°, 30°, 50°, 70°, and 90° were considered.

Convection heat transfer is directly proportional to the solar insolation. The length of collector influences the amount of the heat transfer. The air outlet temperature is rising as the absorber plate is longer.

As the passage tilt is reduced towards the horizontal position, the mean and outlet temperatures are increased noticeably. This is demonstrated experimentally and theoretically.

It is recommended to use another indication parameter to evaluate the heating performance by including the solar intensity as input and the gained heat by air, as output.

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