CHAPTER FOUR

RESULTS

This chapter presents the experimental results of the developed CST using PTC system on selected days during the months of December 2015, January and February 2016. The developed system were tested for its thermal performance under the tropical climatic environment of Malaysia where the degree of solar insolation is constantly varying due to formation of clouds, rain fall and wind. However, this chapter investigates the effects of various parameters on the performance of the PTC. Parameters investigated are the design of collector and liquid system, material as well as the size of tube receiver.

4.1 Collector Design

There are two type of collector design used; automatic tracking system and manual system. In the automatic system, PTC will move following the sun track automatically based on the sensor in the system. On the other hand, manual system will track the sun based on observation every 15 minutes. For manual system approach presented in this research, it was observed that the earth travels 360^{o} in 24×60 minutes. In each 1 minute, the earth travels at an angle of $0.25\ ^{o}/$ min. Therefore, the position of the PTC reported here was set to be changing by rotating it manually after each 15-minute interval at an angle of 3.75^{o}.

Results for both are plotted in Figure 19. The trend of the outlet water temperatures obtained with PTC automatic tracking was observed to be higher than the manual system. This is as expected because the automatic tracking system was able to track the solar insolation on a biaxial path every minute while the manual system only moves every interval period of measurement. Consequently, it is less efficient in absorbing a significant amount of solar insolation as it deviates away from solar collector when it is operated manually.

The increase of water outlet temperature could be not only due to collector tracking system but other factor such as humidity and ambient temperature as well. Relationship between these factors is also studied. Figure 20 shows effect of humidity and ambient temperature with solar insolation measured each for a period of 30 minutes. It clearly shows that as the solar irradiance increase, the higher will be the ambient temperature and lower percentage of humidity.

The parabolic solar collector usually collects a little amount of energy in fixed position or less when the collector is operated manually. For maximum solar energy collection, tracking system should be used. Since the higher efficiency of PTC with automatic tracking system compared to manual system has been proven, this automatic tracking system will be adopted throughout the experiment. Meanwhile due to dependence of humidity and ambient temperature with solar irradiance, solar insolation will be used as independent axis.

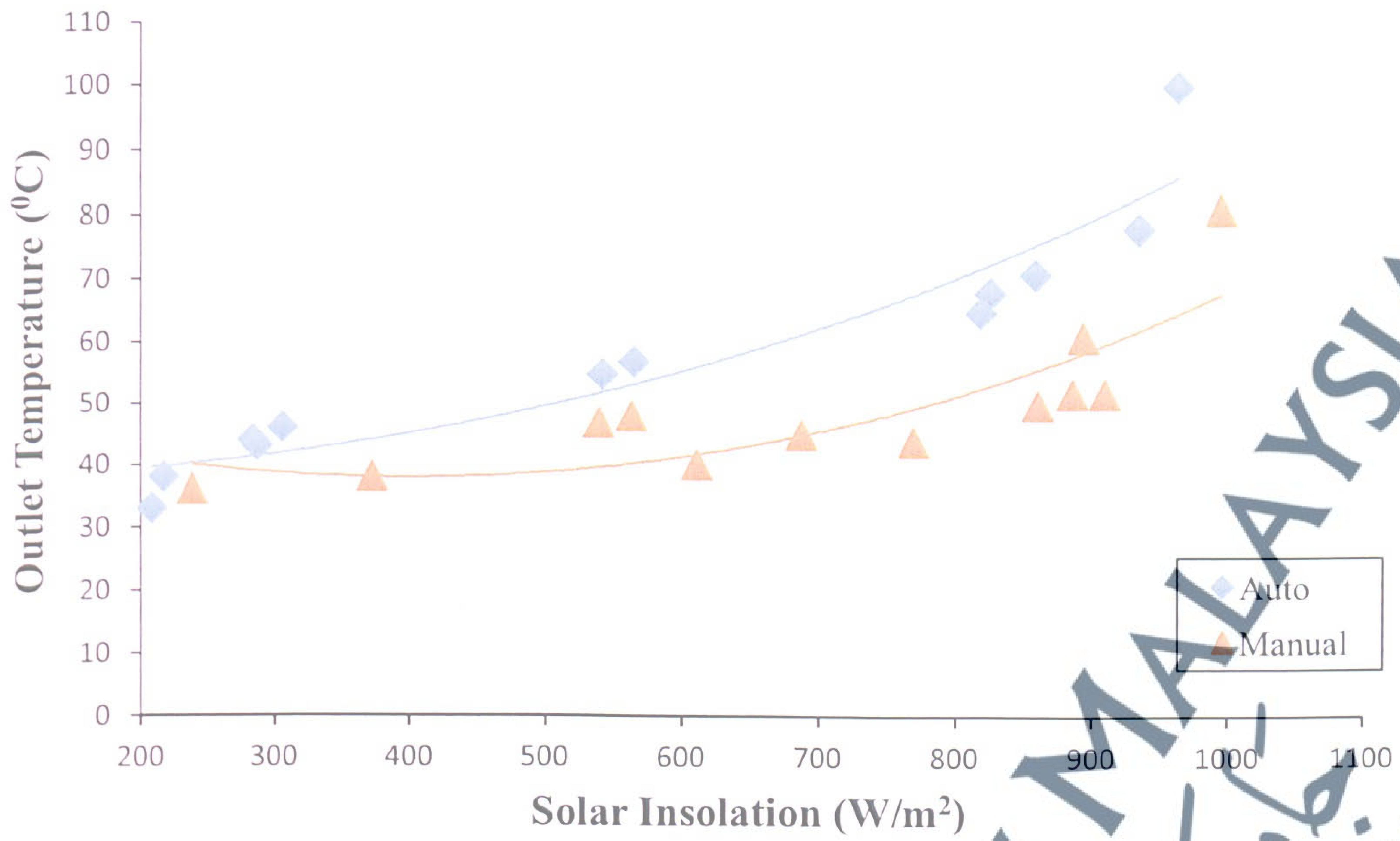

Figure 19: Comparison of Water Outlet Temperature for Automatic PTC Tracking System and Manual PTC System with Varying of Solar insolation.

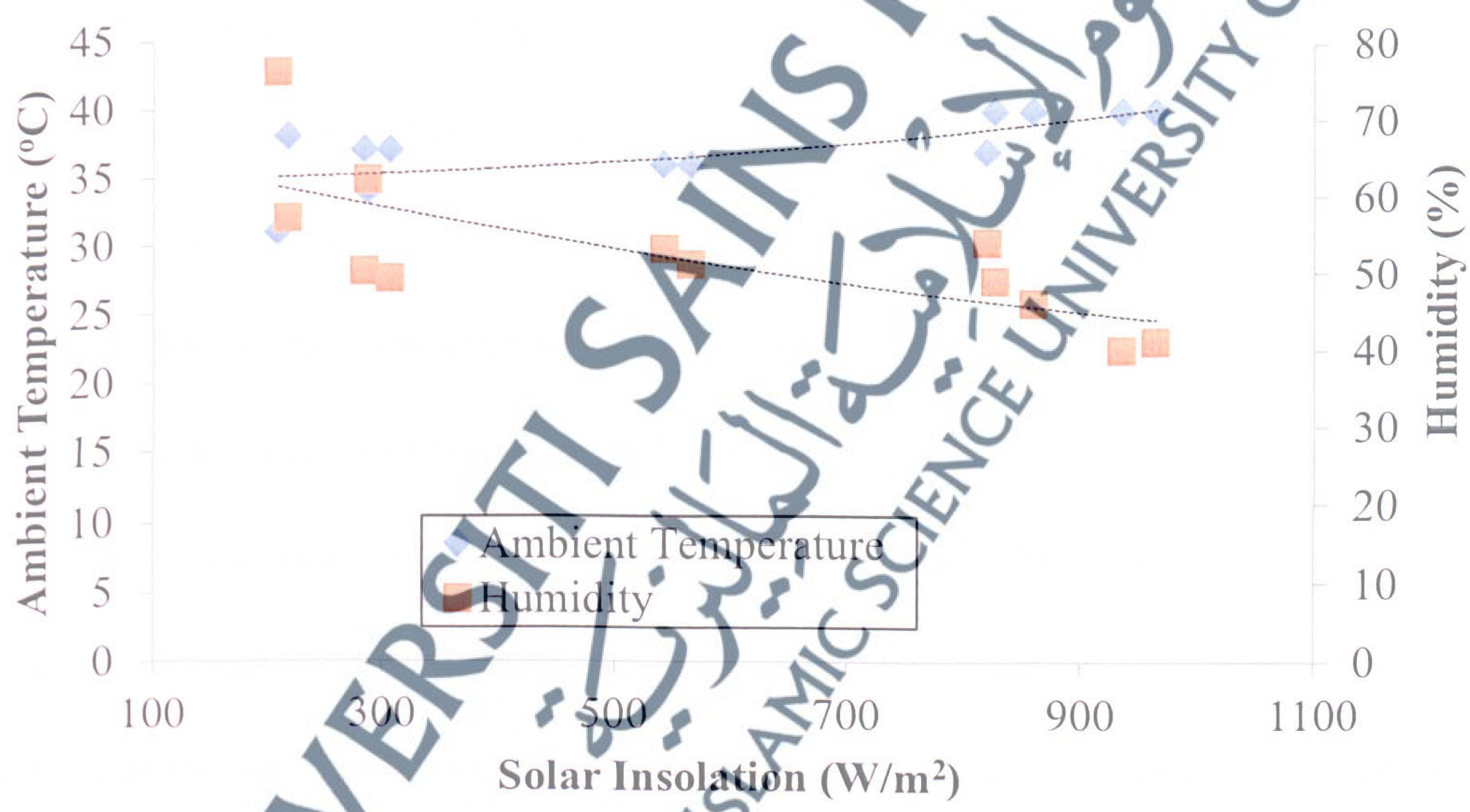

Figure 20: Variation of Ambient Temperature and Humidity with Solar Insolation

4.2 System Design

There are 2 different design options for the liquid heat transfer namely closed system and open system. Closed system does not allow the water to flow out of the system in contra with open system where water is allowed to flow out of the system. Both systems are discussed below.

4.2.1 Closed System (Static Water)

To investigate the ability of the fabricated CST design, the increase in water was determined in the closed system. Using the 2cm copper tube receiver and automatic tracking PTC as collector, water through the receiver was kept static (water not flowing). Inlet valve at one end of tube receiver is opened to allow water flow while closing the valve at the outlet end. After interval period of 30 minutes, the water temperature rise, ΔT (= T_o - T_i) were measured and recorded upon releasing the valve. Prior to that, initial temperature T_i and the volume of the water were determined.

Figure 21 shows the water temperature rise and direct solar insolation obtained throughout the experiment time for closed system. The graph shows that heat absorption by water was not constant throughout the time from 10:30 am until 3:30 pm. Its showing peak temperature rise obtains at 2.30 pm and low in the morning. Clearly, the low absorption of heat in the morning was due to low solar insolation. Formation of clouds oscillates in and out and covers the intensity of solar insolation during the experimental test. The small increase in temperature of water is justified with the closed system condition, where the water cannot absorb more heat when the temperature started to increase and become significant with ambient temperature.

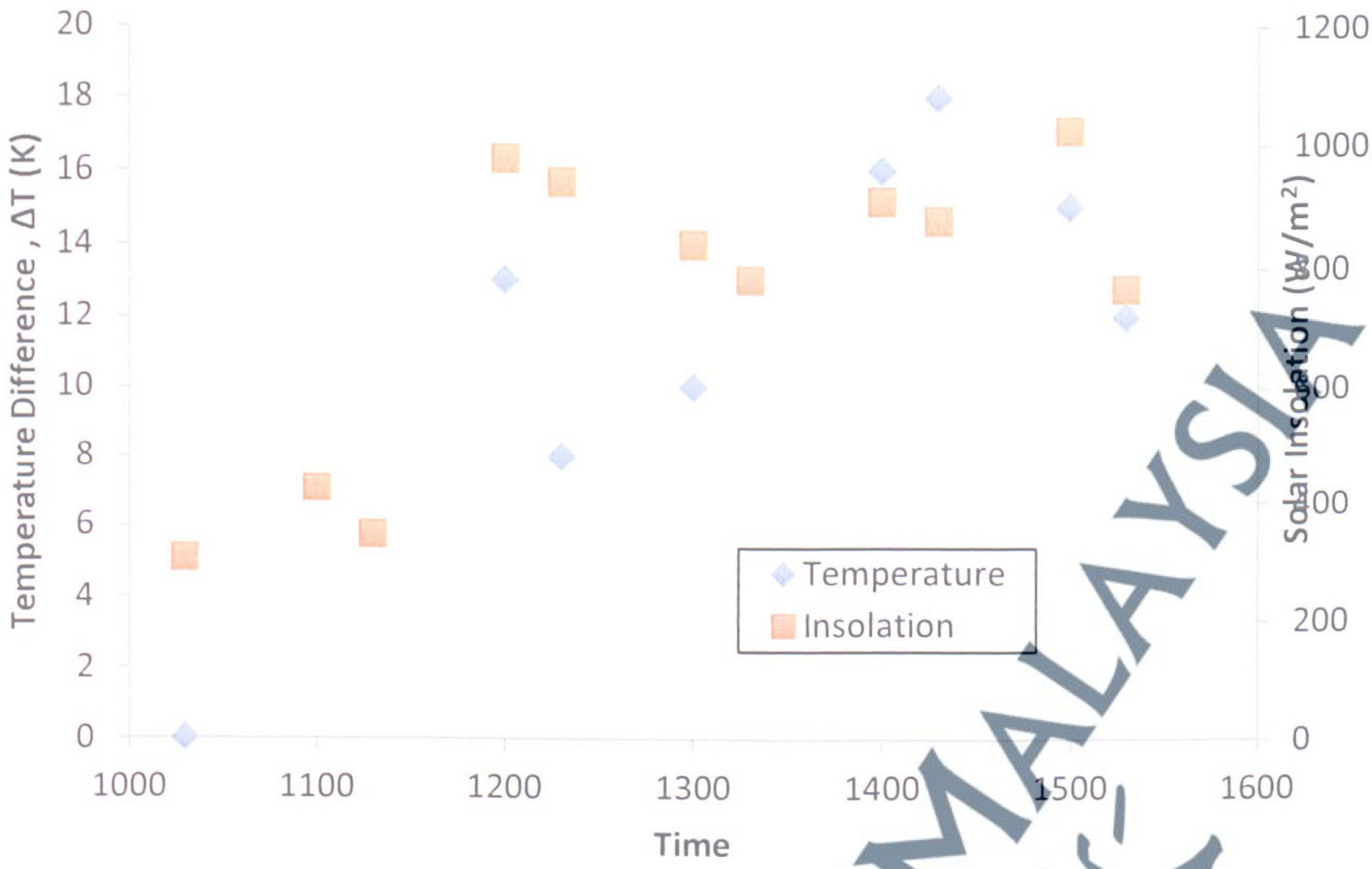

Figure 21: Variation of Water Temperature Rise and Solar Insolation throughout the Time Taken during the Experiment for Closed System

4.2.2 Open System (Flowing Water)

The CST was then tested under open system, which represents the condition of real hot water heating system. The same 2 cm diameter tube receiver was used with the PTC oriented along N-S direction. To ensure that the water is flowing from inlet to outlet receiver tube, a container that is responsible for supplying water to the inlet was located 1 m above the solar concentrator while outlet container was located on the ground below the PTC system. This simple gravity-based system ensured the water flows during the experiment. The mass flow rate was measured to be 0.0025 kg/s and was kept constant throughout the experiment.

Figure 22 and 23 shows temperature rise in water flowing system and as compared to the static water system. As expected, temperature increase will be higher in open system because heat accumulated in open system will be driven away through water flow. This can be shown by rate of water heat absorption as shown in Figure 24. Rate of water heat absorption for closed system is calculated from rate of heat transfer by radiation, $\dot{Q}_s = \varepsilon\sigma\,(\Delta T^4)$. Thus if the temperature difference between ambient and water become closer, rate of heat transfer become smaller, as observed. This is in contra with open system where rate of heat absorbed keep increase due to increase in temperature difference between outlet and inlet.

Figure 22: Variation of Water Temperature Rise and Solar Insolation throughout the Time Taken during the Experiment for Open System.

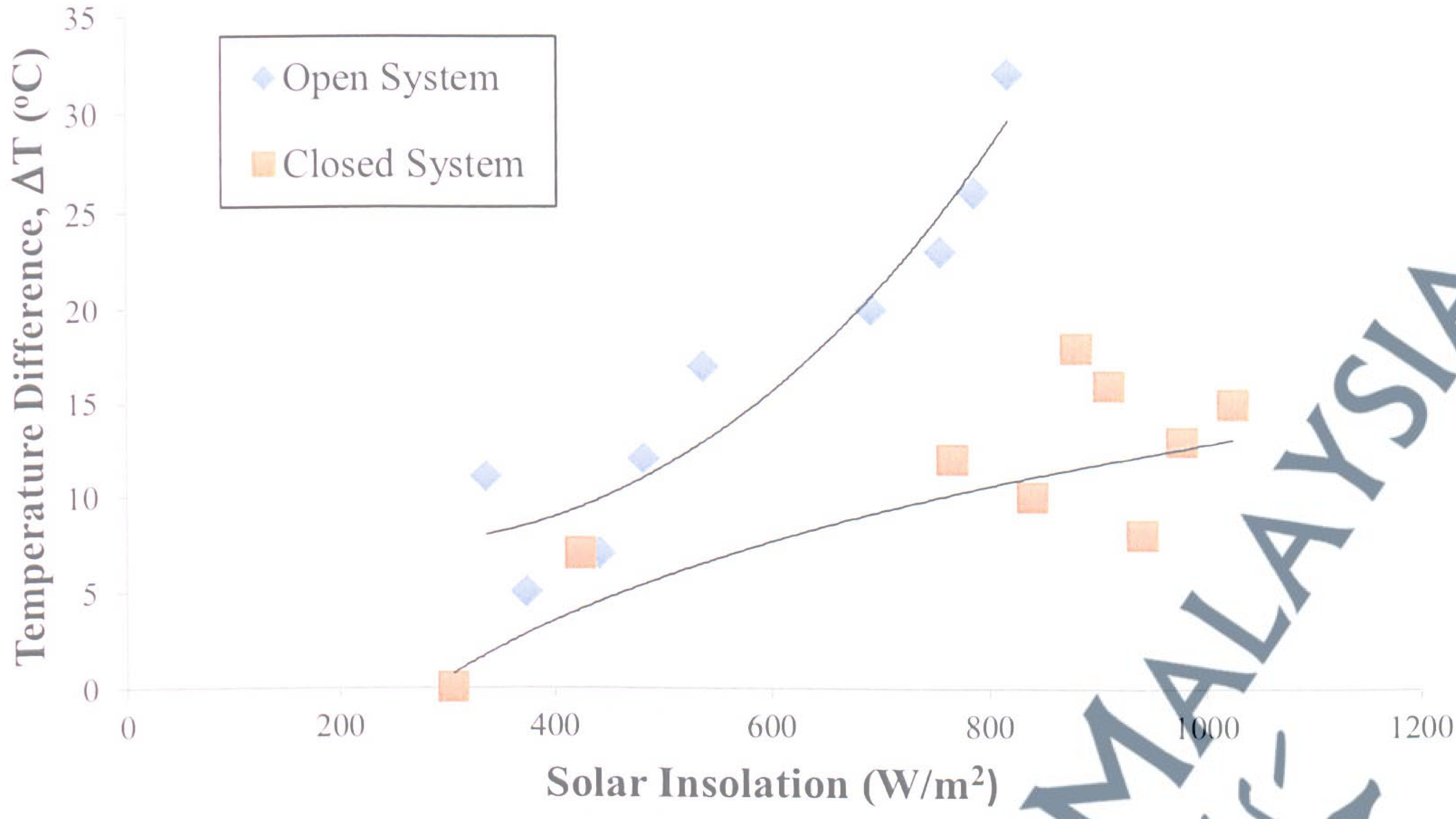

Figure 23: Comparison between Water Temperature Rise with Solar Insolation for Open and Close System.

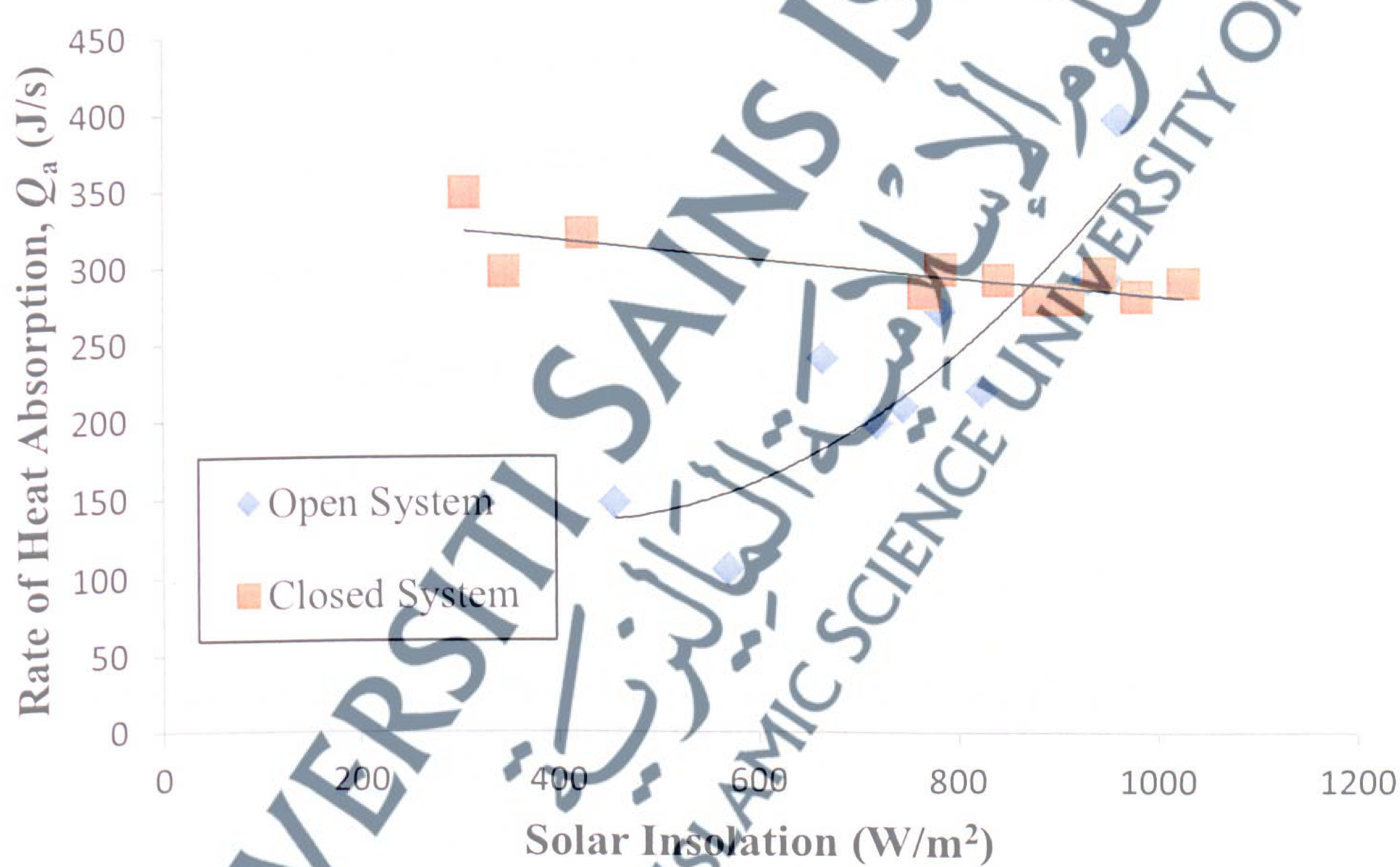

Figure 24: Comparison between Rates of Water Heat Absorption between Open and Closed System as a Function of Solar Insolation.

4.3 Tube Receiver

The tube receiver plays an important role in transferring all the received heat into the water. In this study, effect of varying size, material and design of the tube receiver will be investigated.

4.3.1 Size

Two different sizes of tube receivers are investigated. A 2 cm diameter and a 1 cm diameter of copper tube. Different sizes of tube receivers are tested to determine which receiver performs better at high temperature. The most common way of reducing receiver heat loss at elevated temperatures is by reducing the size of the receiver because heat loss is directly proportional to the area of the hot surface (Kumar et al., 2013). But on the other hand, the larger the diameter of tube receiver, more water can flow through the tube thus cooling the temperature at the tube receiver to absorb more heat.

The thermal efficiency obtained from both size is plotted in Figure 25. From the figure, it was observed that the efficiency of the 2 cm copper tube is higher than the 1 cm tube for rate of water flow 0.0025 kg/s. The peak efficiency achieved for 2 cm tube receiver is 43 % with an outlet temperature of 69 ^{o}C while 1 cm tube receiver has a maximum efficiency of 27 % at an outlet temperature of 61^{o}C even though received higher solar insolation of 958 W/m^2 compared to 817 W/m^2 in 2 cm tube receiver.

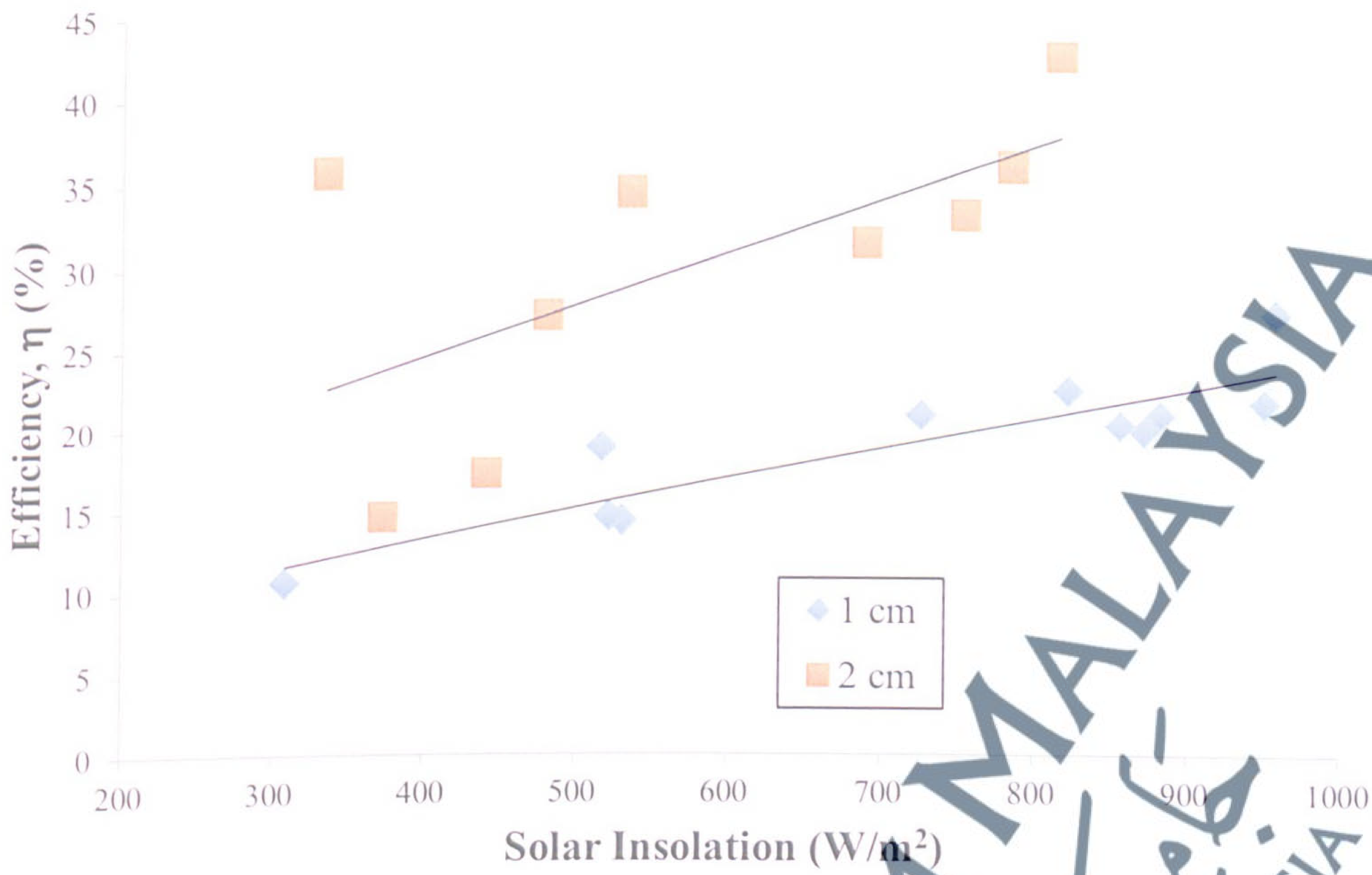

Figure 25: Efficiency of 1 cm and 2 cm Tube Receivers with Solar Insolation Obtained by the Open, Uncovered Automatic Tracking System.

Temperature difference produce in 2 cm diameter copper tube receiver are higher than 1 cm tube, thus producing higher efficiency in 2 cm tube receiver. This result indicates that more heat can be carried away from the tube receiver therefore cooling the water compared to heat loss through the surface of the larger diameter tube receiver. This allows more heat can be absorbed, thus increase the efficiency.

4.3.2 Design

The tube receiver is responsible to receive solar irradiance from the collector to heat up water that flows through it. Therefore, the design of tube receiver is important as well. To reduce heat loss, tube receiver is enclosed inside a box cover. The bottom of the box made out of glass (facing downward) to reduce the amount of heat losses by convection because glass have low thermal conductivity that is 0.96 W/mK compared to copper that is 385 W/mK. The rest of the box's sides are painted black to increase area of heat absorption from the sun and collector. Figures 26 and 27 shows efficiency of 2 cm diameter copper tube receiver encapsulated inside box cover and without encapsulation respectively from two different sets of data. The solid line represents the average values. Figure 28 show the comparison between both of tube receiver design with an error bar. The PTC with covered tube receiver presented here gives no difference in efficiency with non-covered tube receiver. Encapsulated receiver inside box cover did not give influence on the 2 cm diameter tube receiver.

Figure 26: Efficiency of 2 cm Diameter of Copper Tube Receiver Encapsulate Inside Box Cover with Solar Insolation Obtained by the Open, Automatic **Tracking System.**

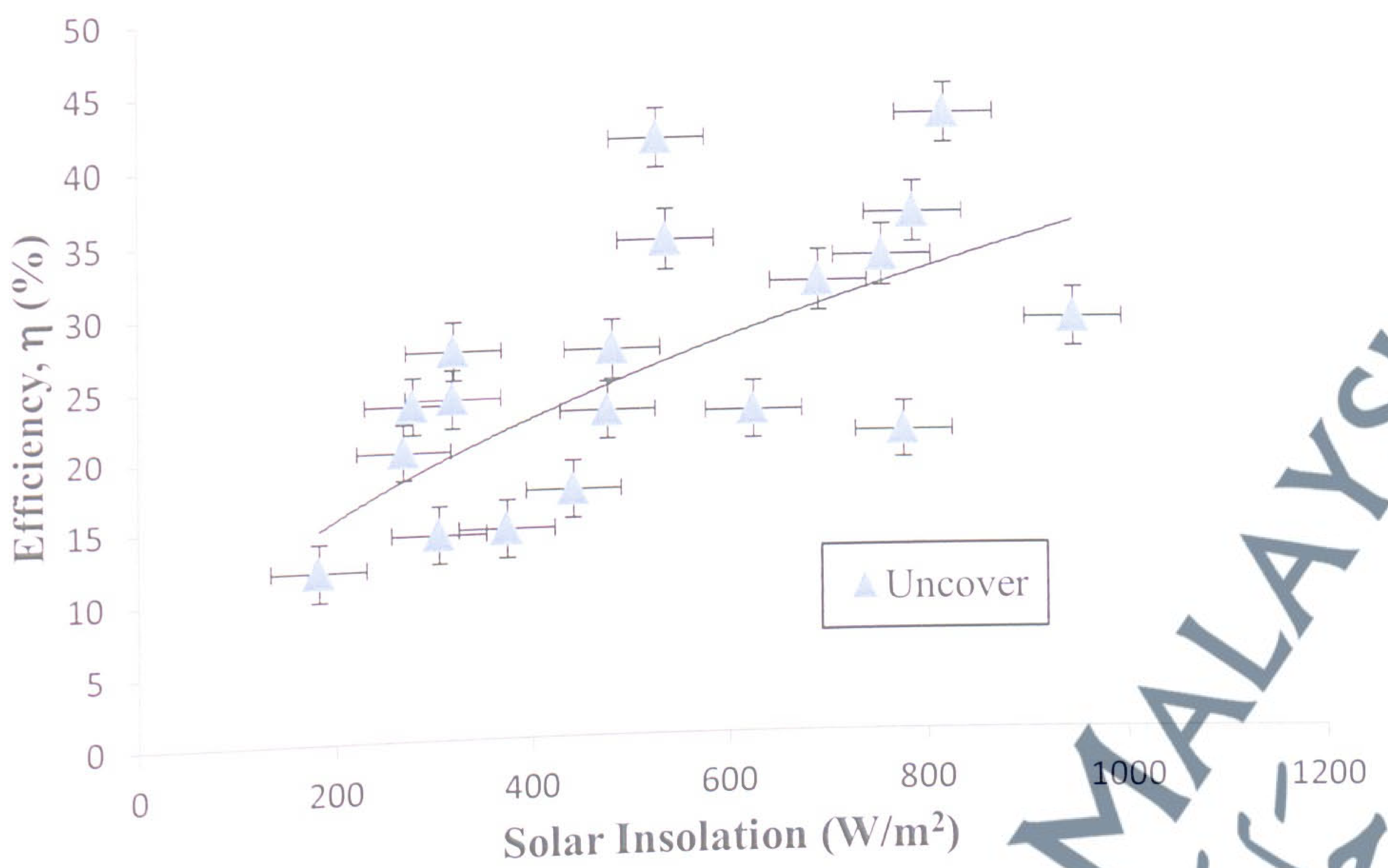

Figure 27: Efficiency of 2 cm Diameter of Copper Tube Receiver without Box Cover with Solar Insolation Obtained by the Open, Automatic Tracking System.

Figure 28: Comparison between Efficiency Of 2 cm Diameter of Copper Tube Receiver Covered and Uncovered with Solar Insolation Obtained by the Open, Automatic Tracking System.

Further investigation on influence of wind speed and humidity to the thermal performance of CST were made. Results for both covered and uncovered 2 cm copper tube receiver with wind speed and humidity variation are shown in Figures 29 and 30 respectively. Clearly from the results, wind speed did not influence the system since both conditions have similar wind speed during measurement. On the other hand, during the measurement of cover tube receiver, the humidity is low around 40 % while uncovered tube receiver was measured with high humidity level. High level of humidity could help in reducing heat loss from the uncovered tube receiver, thus increase the efficiency. That is why we can really see any difference in efficiency of covered and uncover tube due to the high humidity under uncovered measurements.

Figure 29: Influence of Wind Speed of Covered and Uncovered 2 cm Diameter Copper Tube Receiver with Time Obtained by the Open, Automatic Tracking System.

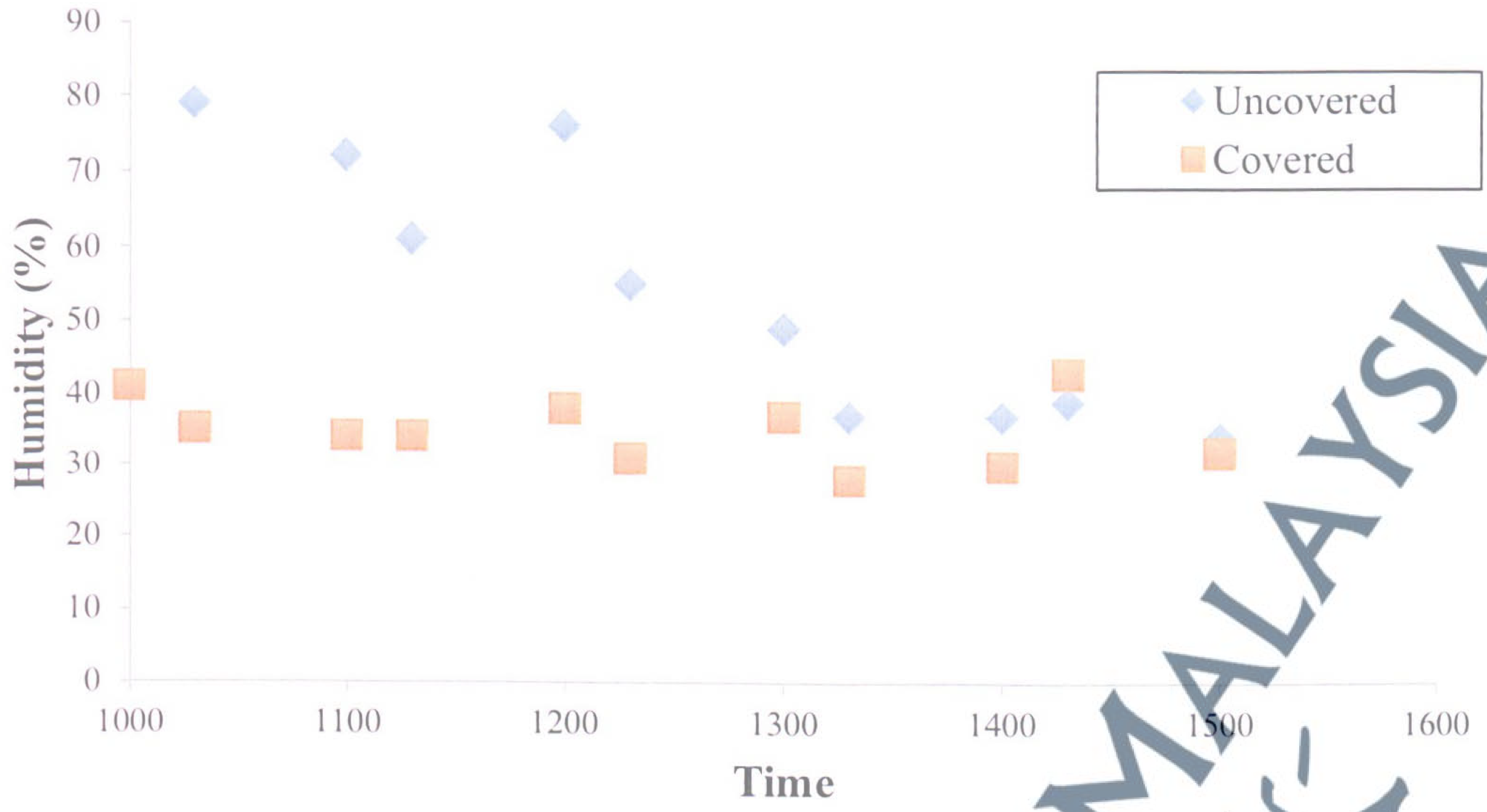

Figure 30: Influence of Humidity of Covered and Uncovered 2 cm Diameter Copper Tube Receiver with Time Obtained by the Open, Automatic Tracking System.

Further investigation was carried out to see if the effect could be clearly observed on much smaller diameter tube receiver. 1 cm diameter tube receiver made of the same material was used. The efficiency obtained by this tube from 2 sets of measurements under encapsulated box receiver and uncovered tube is plotted in Figures 31 and 32 respectively. Comparison of both shows significant difference under high solar insolation (Figure 33). As expected, the enclosed tube receiver produces higher efficiency than uncovered one. This can be observed under much smaller diameter tube because smaller tube produces less heat absorption than larger one. Thus, box receiver would help assist the heat absorption on the tube receiver by reducing the amount of heat losses by convection and increases heat absorption by radiation. The higher the rate of heat lost by the receiver the lower temperature output would be.

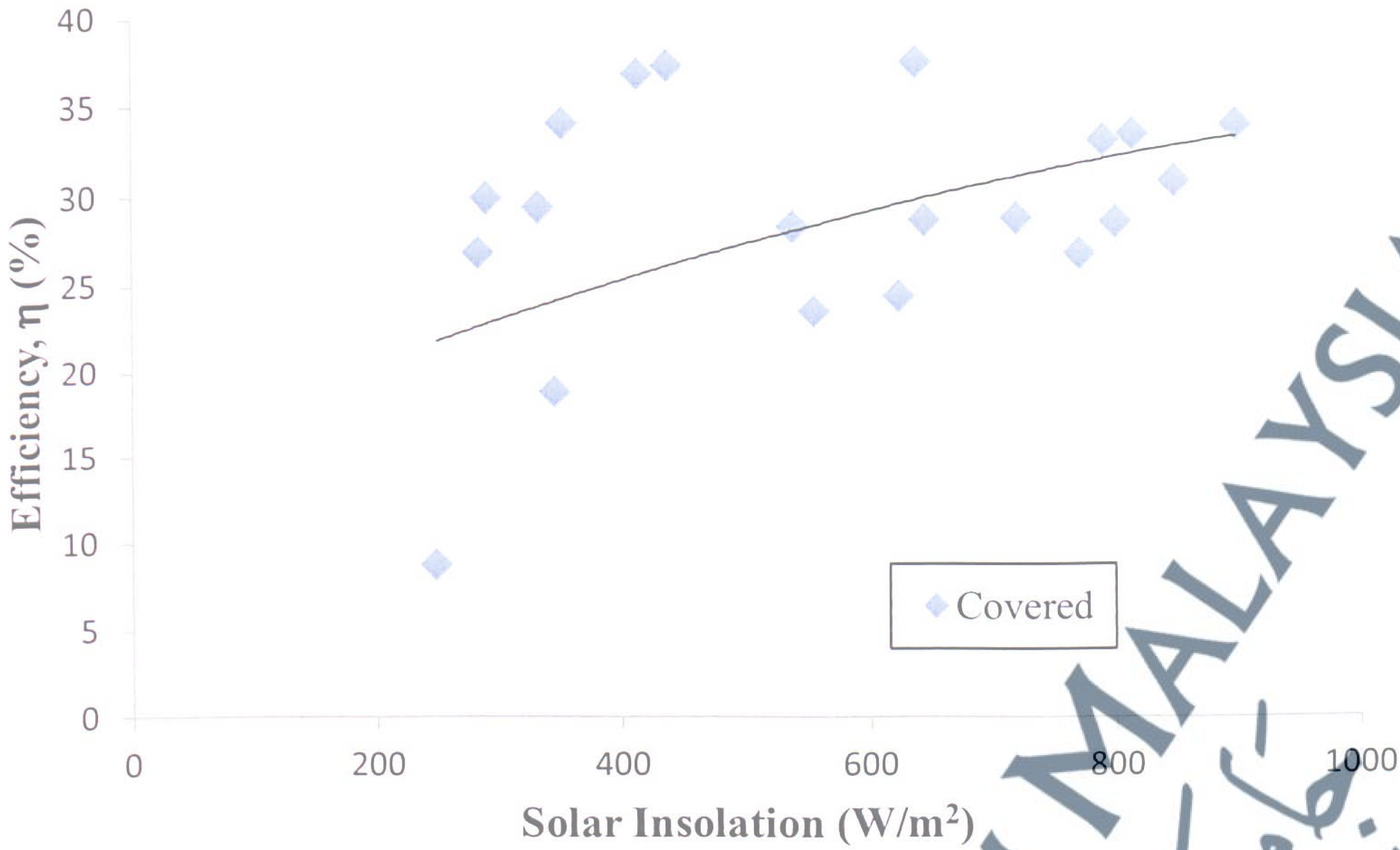

Figure 31: Efficiency of 1 cm Diameter of Covered Copper Tube Receiver with Solar Insolation Obtained by the Open, Automatic Tracking System.

Figure 32: Efficiency of 1 cm Diameter of Uncovered Copper Tube Receiver with Solar Insolation Obtained by the Open, Automatic Tracking System.

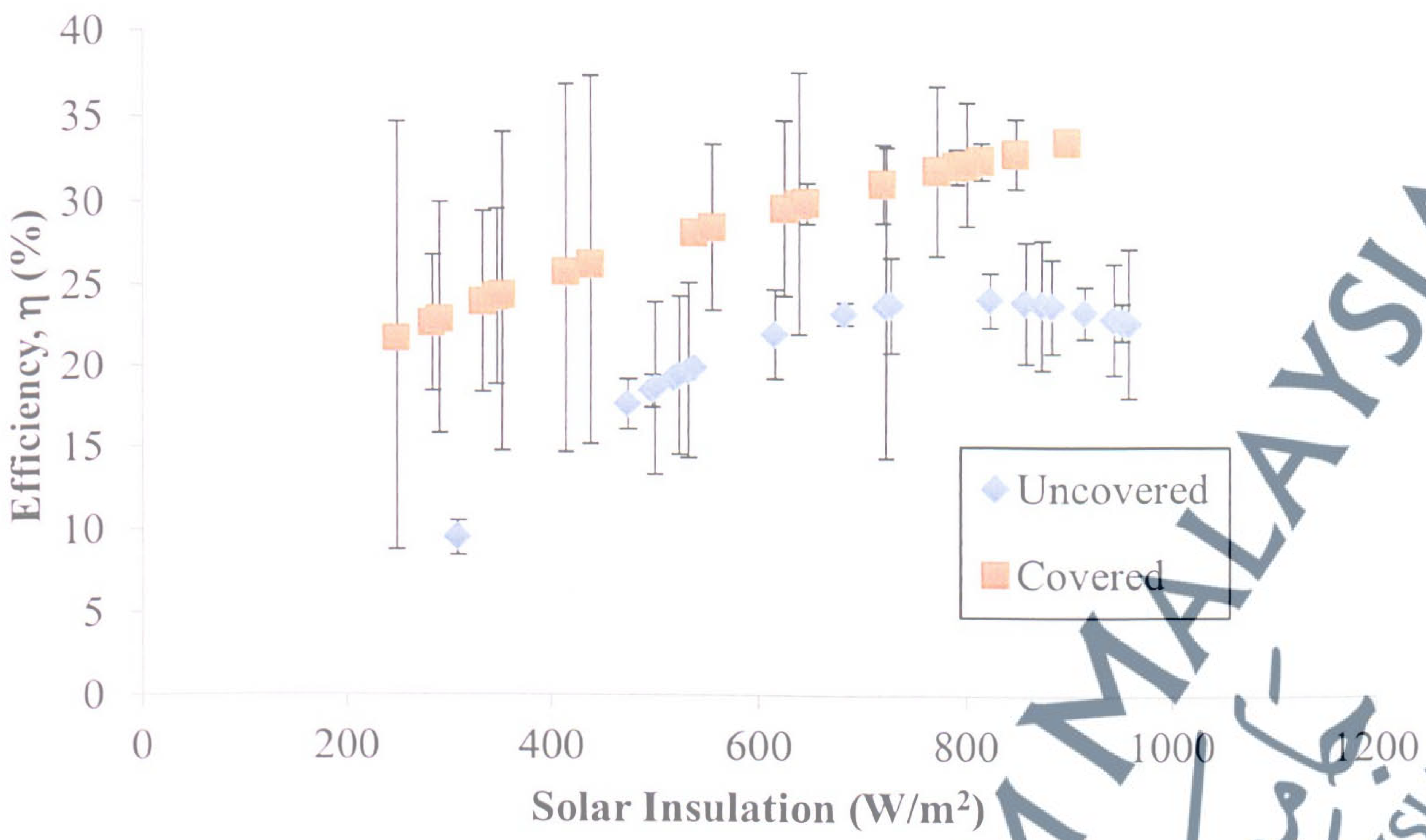

Figure 33: Comparison between Efficiency of 1 cm Diameter of Copper Tube Receive Covered and Uncovered with Solar Insolation Obtained by the Open, Automatic Tracking System.

To confirm that the increase efficiency for cover tube receiver 1 cm diameter copper was not due to wind speed and humidity during the measurement, relationship of the data with solar insolation was plotted in Figure 34. It was observed that humidity for both covered and uncovered tube receiver are almost similar, thus did not give effect on the increase efficiency of covered tube receiver. On the other hand, wind speed for uncovered tube receiver is much larger than during measurement of covered tube receiver. Theoretically, high wind speed will assist in heat loss from tube receiver, thus decreasing the rate of heat absorption of tube receiver as stated by **Newton's law of cooling** which states that the rate of heat loss of a body is directly proportional to the difference in the temperatures between the body and its surroundings provided the temperature difference is small and the nature of radiating surface remains same. In this case, we can conclude that the lower efficiency of uncovered tube was due to the design itself or due to the wind speed because of the variety of the parameter and it is not easy to fix all the nature parameters. Nevertheless, we can conclude that the efficiency of the tube receiver to absorb heat is not totally dependent on the design (covered or uncovered) only but also dependent on both humidity and wind speed as well.

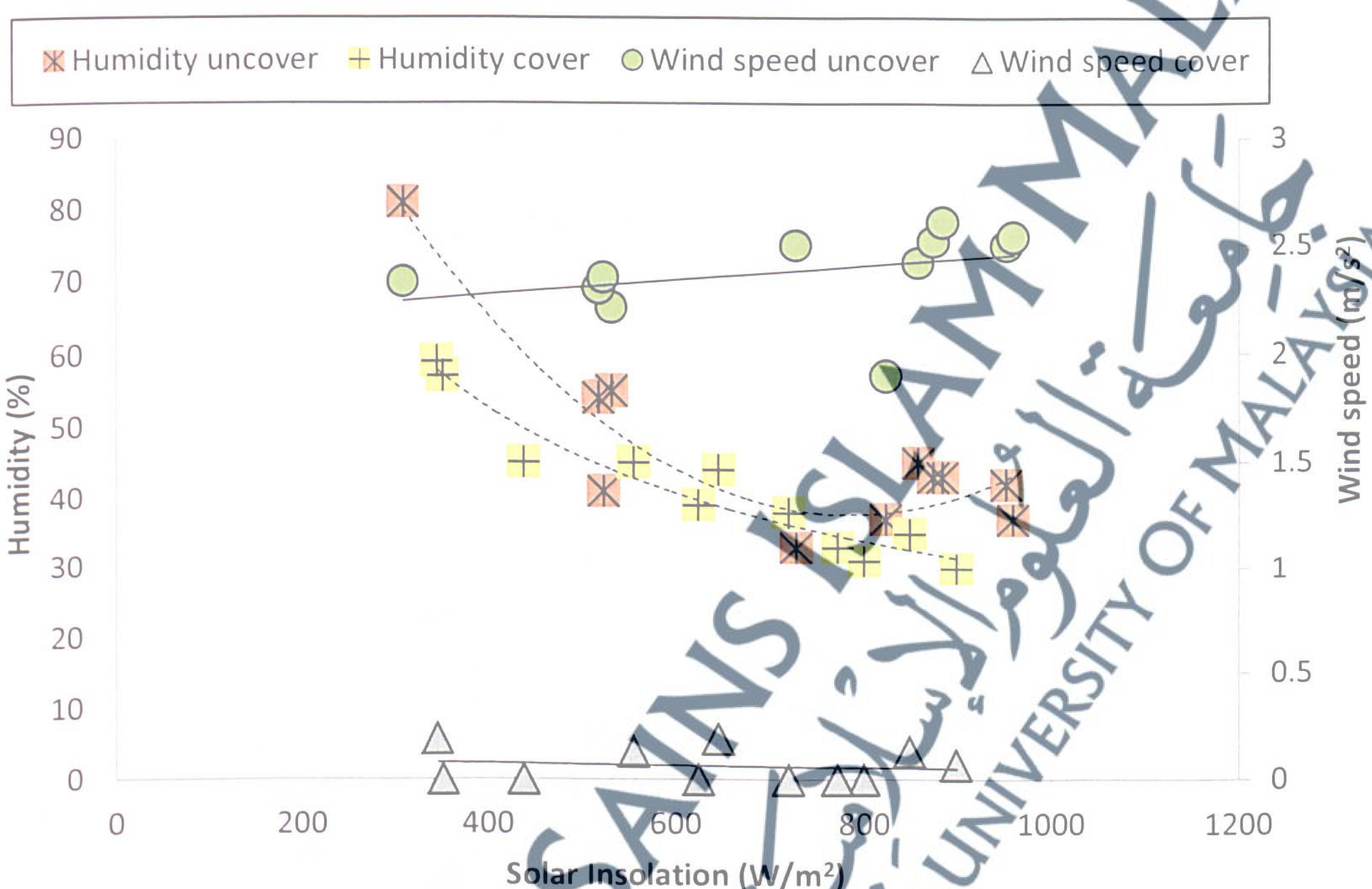

Figure 34: Influence of Wind Speed and Efficiency to Covered and Uncovered 1 cm Diameter Copper Tube Receiver with Solar Insolation Obtained by the Open, Automatic Tracking System

4.3.3 Material

Two different materials of tube receiver are investigated; aluminum and copper. To observe the influence of material to the CST tube receiver design, 2 cm diameter of tube receiver made of copper and 2 cm diameter aluminum tube receiver, under covered condition were tested. It is expected that the higher thermal conductivity of copper which is 385 W/mK should be able to transfer more heat to water thus increase efficiency compared to aluminum with thermal conductivity of 205 W/mK. However, if the copper tube receiver is exposed directly to environment, tendency of heat loss will be higher. The thermal efficiency obtained from the experiments for both materials is plotted in Figure 35.

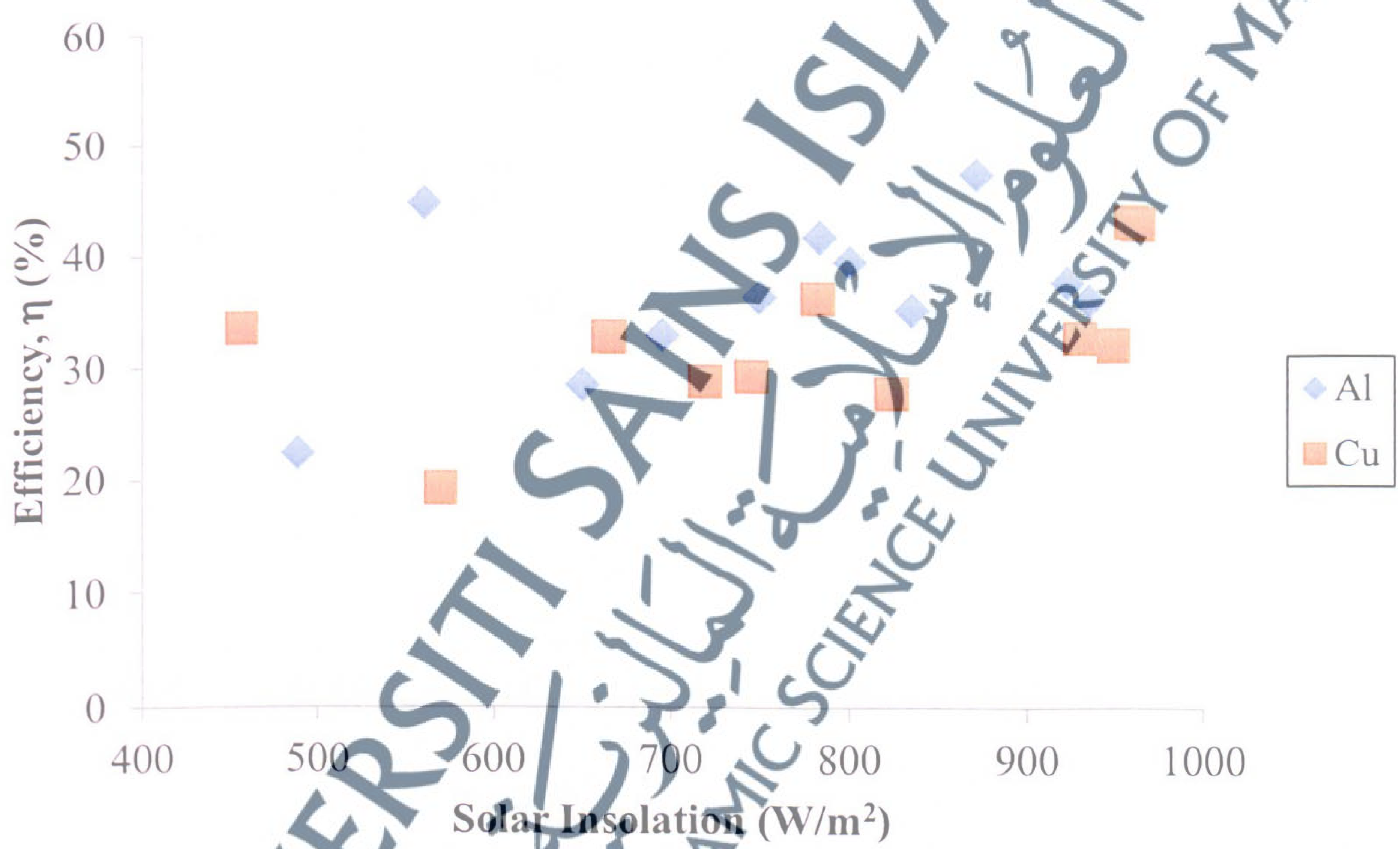

Figure 35: Efficiency of 2 cm Diameter of Aluminum and Copper Tube Receivers with Solar Insolation Obtained by the Open, Covered, Automatic Tracking System.

The graph shows not much difference in the efficiency, which is unexpected even though the solar insolation throughout the experiment for both sets of data is almost the same (Figure 37). Investigation on the wind speed and humidity on both experiments observed that humidity is similar (30-50 %) for both materials. However, wind speed during measurement of copper tube receiver is much higher than wind speed during measurement using aluminum tube receiver as shown in Figure 37. Wind speed increases the amount of heat loss thus decreasing copper efficiency. Therefore, we cannot see the high efficiency in copper in Figure 36 even though copper has high thermal conductivity than aluminum.

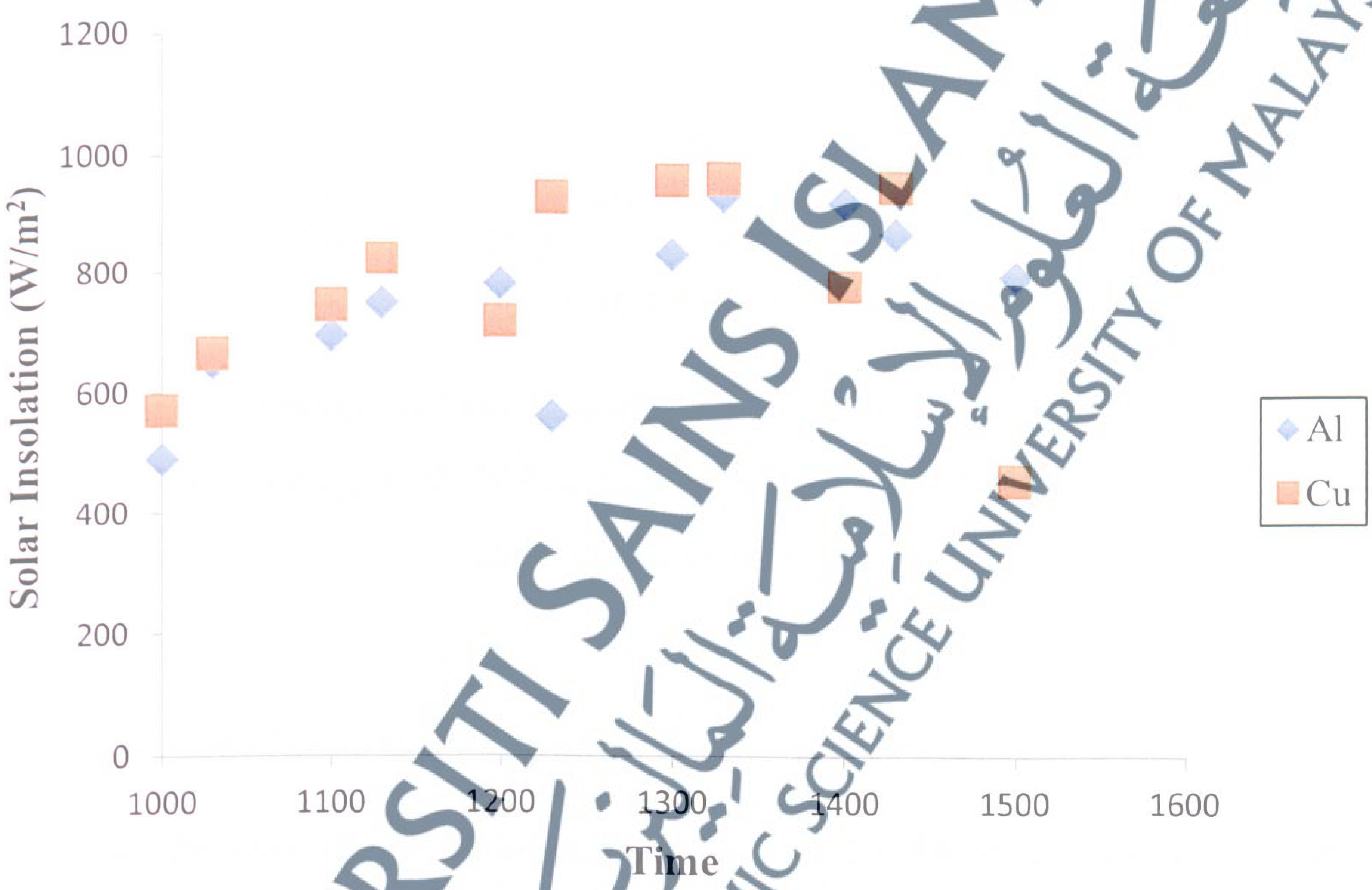

Figure 36: Solar Insolation Change with Time Recorded For 2 m Diameter of Aluminum and Copper Tube Receiver Obtained by the Open, Covered, Automatic Tracking System.

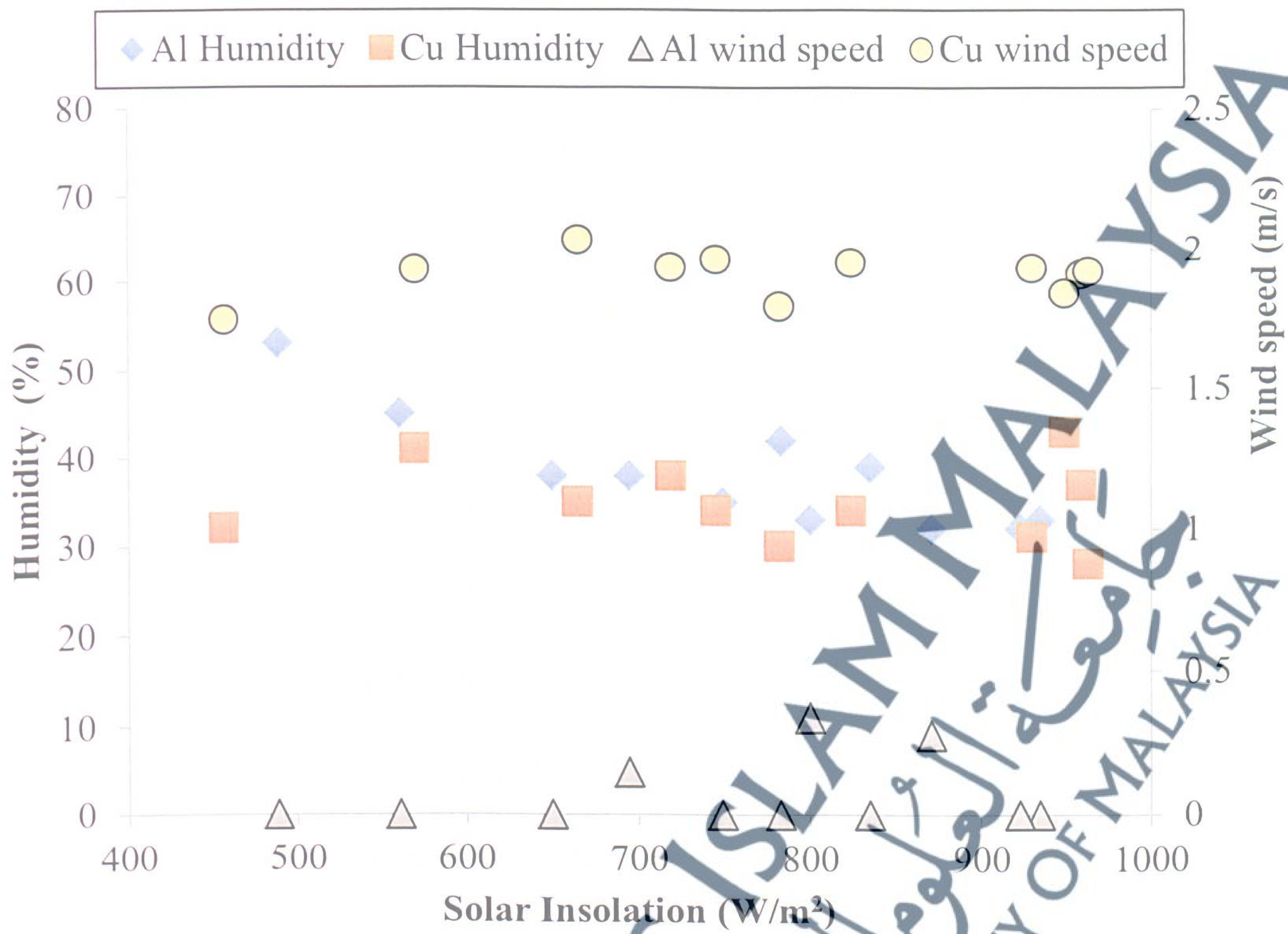

Figure 37: Variation of Humidity and Wind Speed Measured on 2 cm Diameter of Aluminum and Copper Tube Receiver with Solar Insolation Obtained by the Open, Automatic Tracking System.

4.4 Thermal Efficiency Performance Curve of the PTC

The thermal efficiency of a PTC can be described by ASHRAE Standard 93 (1986) can now be re-written as:

$$\eta_{th} = F_R\left(\eta_0 - \left[\frac{U_L(T_i - T_a)}{I_b\, C}\right]\right) \quad 4.1$$

$$\eta_0 = -\left(\frac{A_r U_L F_R}{A_a}\right)\left(\frac{Dt}{I_b}\right) + \left(\frac{Aa}{A_r}\right) F_R \eta_0 \quad 4.2$$

$$\eta_{th} = F_R\eta_0 - \frac{F_R\, U_L}{\mathrm{C}}\left(\frac{T_i - T_a}{I_b}\right) \quad 4.3$$

The thermal efficiency depends on the concentrator design parameters and the parameters characterizing the operating conditions. The optical efficiency, heat loss coefficient and heat removal factor are the design dependent parameters while the solar insolation, inlet fluid temperature and the ambient temperature define the operating conditions (Brooks, 2006). If the thermal efficiency from equation 4.3 is plotted against $(T_i - T_a)/I_b$ straight line will result provided U_L is constant. The intercept is $F_R\eta o$ and the slope is$F_R U_L/C$. The performance curve of the PTC obtained from a series of tests conducted is given in Figures 38, 39, 40, 41 and 42 respectively. An equation of the curve is obtained using the standard technique of a least squares fit.

The optical efficiency, η_o is the fraction of solar radiation incident on the aperture of the collector, which is absorbed at the surface of the receiver tube represented by:

$$\eta_o = \frac{S}{I_b} \quad 4.4$$

The absorbed radiation, *S,* or the actual amount of radiation on the receiver is calculated by:

$$S = I_b(\alpha\,\tau\;\psi\,\rho) \quad 4.5$$

The optical efficiency is given as:

$$\eta_0 = \alpha\,\tau\,\psi\,\rho \quad 4.6$$

where η_0 is the optical efficiency, α absoptance of the absorber, ψ is the intercept factor, ρ reflectance of the reflecting material and τ glass cover transmittance respectively.

The best fit curve displayed in Figure 38 was obtained and shown as a solid line. This leads to the thermal performance equation for the collector with aluminum receiver tube.

$$\eta = 0.48 - 0.39 \left(\frac{T_I - T_a}{I_b}\right) \qquad 4.7$$

From equation 4.3, ($A_r U_L F_R / A_a$) = 0.39 W/Km^2 and $F_R \eta_o$= 0.48. For a geometric concentration ratio (A_a/A_r) of 10.2 the gradient of above equation gives $U_L F_R$ = 3.978 W/Km^{-2}. The optical efficiency was calculated from equation 4.6 which gives η_o = 0.59, this result gives in a heat removal factor F_R of 0.81. The heat removal factor represents the ratio of actual useful energy gain of the collector to the useful gain if the whole receiver were at the fluid inlet temperature. This in turn yields an overall heat loss coefficient U_L of 4.91W/Km^{-2}. However, for copper tube receiver presented in Figure 39 the best fit curve obtained shown and is given in equation 4.8 when using standard technique of the best fit. The intercept is equal to 0.42 and the slope is 0.27. Therefore, the collector thermal efficiency equation for PTC with can be written as:

$$\eta = 0.42 - 0.27 \left(\frac{T_i - T_a}{I_b}\right) \qquad 4.8$$

The optical efficiency, η_o of the collector was determined to be 0.59 for a concentration ratio of 10.2. The heat removal factor and overall heat loss was determined to be 0.705 and 3.47 W/Km^{-2} respectively. Similarly, for the case of a 2 cm uncovered copper tube receiver given its best fit curve in Figure 40. The intercept is equal to 0.44 and the slope is 0.34. Therefore, the collector thermal efficiency equation for PTC with the best fit curve can be written as:

$$\eta = 0.44 - 0.34 \left(\frac{T_i - T_a}{I_b}\right) \qquad 4.9$$

The optical efficiency, η_o of the collector was determined to be 0.7 for a concentration ratio of 10.2. The heat removal factor and overall heat loss was determined to be 0.63 and 5.50 W/Km^{-2} respectively.

For the case of 1 cm copper tube receiver, the best fit curve displayed in Figure 41 leads to the thermal performance equation given in equation 4.10 below for the collector with 1 cm covered copper tube receiver.

$$\eta = 0.37 - 0.25 \left(\frac{T_i - T_a}{I_b}\right) \quad 4.10$$

The optical efficiency, η_o of the collector was determined to be 0.55 for a concentration ratio of 16.84. The heat removal factor calculated to be 0.67 where the overall heat loss determined to be 6.28 W/Km^{-2}. Similarly, for uncovered copper tube receiver, the best fit curve obtained is given Figure 42 with it is written in equation 4.11. The intercept is equal to 0.45 and the slope is 0.34. The optical efficiency, η_o of the collector was determined to be 0.64 for a concentration ratio of 16.84. The heat removal factor and overall heat loss was determined to be 0.7 and 8.42 W/Km^{-2} respectively. Therefore, the collector thermal efficiency equation for PTC can be written as:

$$\eta = 0.45 - 0.35 \left(\frac{T_i - T_a}{I_b}\right) \quad 4.11$$

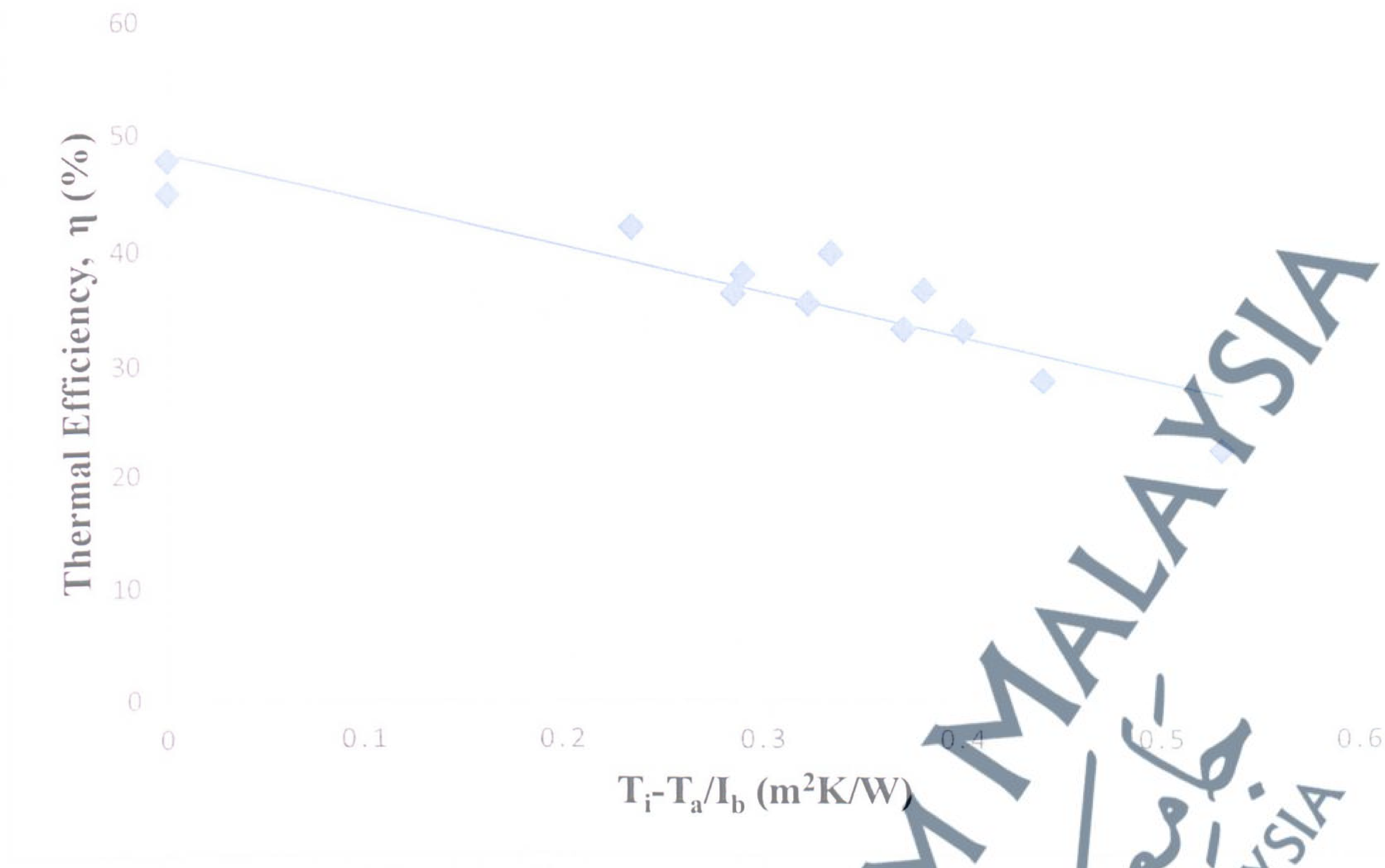

Figure 38: Thermal Efficiency Curve for 2 cm Aluminum Tube.

Figure 39: Thermal efficiency curve for 2 cm covered copper tube.

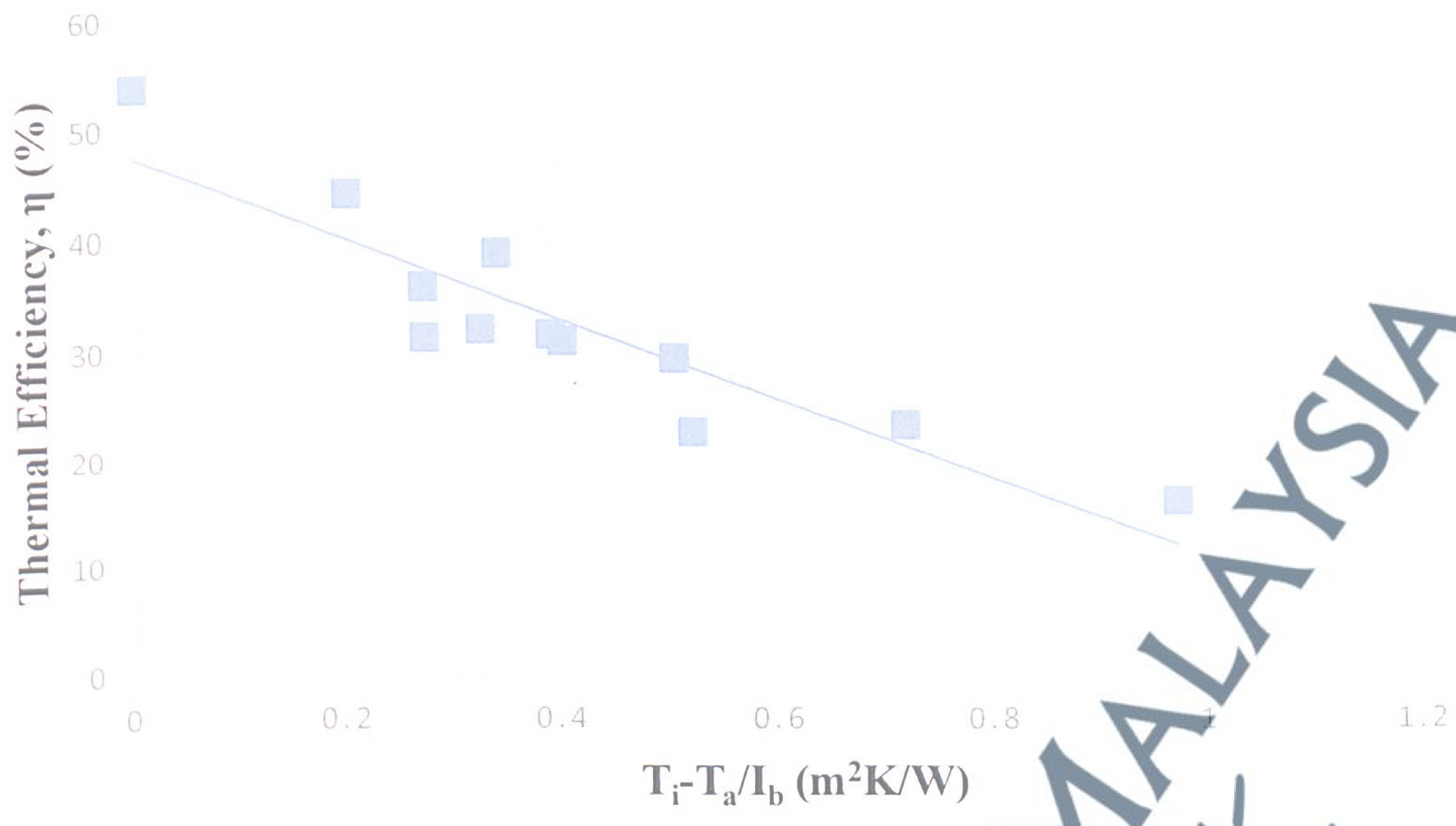

Figure 40: Thermal Efficiency Curve for 2 cm Un-Covered Copper Tube

Figure 41: Thermal Efficiency Curve for 1 cm Un-Covered Copper Tube.

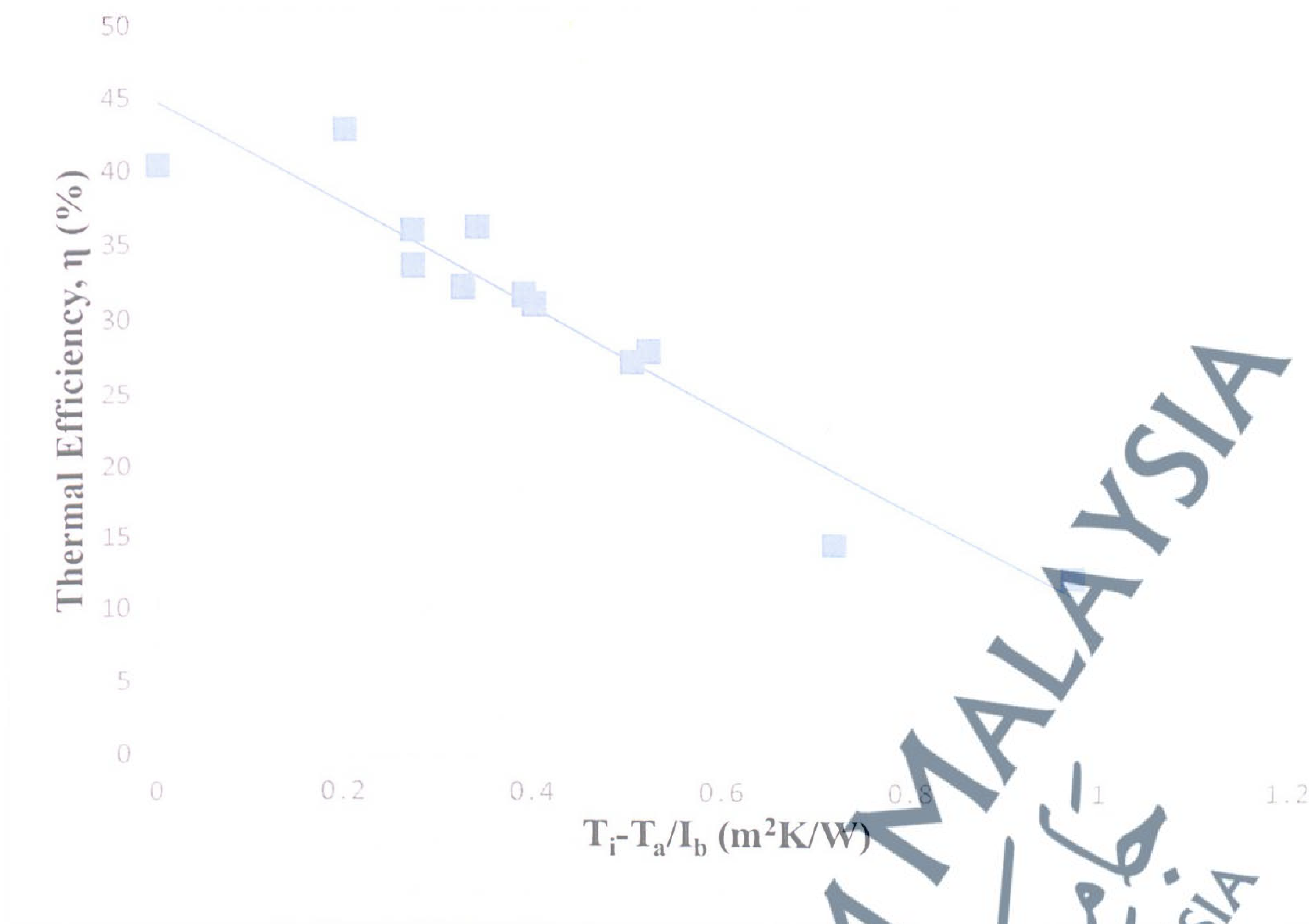

Figure 42: Thermal Efficiency Curve for 1 Cm Covered Copper Tube.

From the above findings, the scatter observed in the data was caused by variations of solar insolation due to the formation of clouds, varying temperature, wind effects and variations in angle of incidence. The key data obtained from the analysis of the thermal efficiency of the PTC was summarized in Table 4 given below.

Table 4: Summary of the Key Data from the Thermal Efficiency Test

Receiver	Peak η (%)	Ave. η (%)	Ave.Dt /I_b(W/m^2)	Ave.I_b (W/m^2)	Ave. T_i(^{0}C)	Ave. T_a(^{0}C)	Ave. Humidity (%)	Ave. Wind (m/s)	F_R	U_L (W/K m^2)
Covered Aluminum (2 cm)	47.7	36	0.2986	753.73	36	42.55	38.18	0.07	0.81	4.91
Covered Copper (2 cm)	43.3	30	0.439	778.45	35.09	41.45	34.81	1.43	0.71	3.47
Uncovered Copper (2 cm)	42.8	30	0.4089	599.35	31.85	38.6	53.6	-	0.63	5.5
Covered Copper (1 cm)	37.5	27	0.45	624.78	34.32	40.32	41.41	0.065	0.67	6.28
Uncovered Copper (1 cm)	26.8	19	0.3832	747.87	33.91	38.69	47.82	2.38	0.7	8.42

From the table above, it was observed that the covered fabricated PTC receiver heat lost was observed to be less when compared to the non-covered in almost all the cases presented above. This is because, the covered receiver experienced lower speed convectional currents around its enclosure which resulted to the thermal efficiency obtained by the covered PTC receiver to be more significant than that of the non-covered receiver since the non-covered receiver suffered high heat losses by convection from the collector to the surrounding environment. Similarly, the thermal efficiency obtained with 2 cm covered copper tube receiver increasing steadily compared with those previous results. This indicates that the fabricated custom box improves the overall performance of the PTC. The peak efficiency value and the thermal efficiency equation obtained by covered aluminum tube receiver and the result in general is close to other research works reported in the literature. Peak values of 45 % (Bakos et al., 1999), 63.8 % (ValanArasu&, Sornakumar, 2007), 37 % (Eltahir Ahmed Mohamed, 2013), 50.57 %

(Macedo et al, 2014), 18.23 %, 20.25 % (Santosh et al., 2014), and 50 % (Baha et al., 2014) are described in the literature. Overall, the efficiencies are lower than those of commercial PTC's the collector tested by sandia laboratory (Dudley et al. 1995) holds a maximum efficiency between 70.8 % and 76.3 %, while the newest Eurotrough collector is about 75 % efficient (Geyer et al., 2002). This wide difference in terms of the performance results is mainly due to highly reflecting surfaces, higher intercept factors, receiver's absorption due to the selective coatings of the receiver and good environmental conditions.