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Condenser Designs for Greenhouse Desalination

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Abstract

Innovative condenser designs were created in this study. The plate channeled condenser and the vibrating plastic surface condenser, first condenser design consisting of a cooled plate surface upon which water vapor, carried by saturated air, condenses at its outer surface. The plates are made of plastic and are constantly cooled by coolant flowing through internal passages within the plate. The second condenser design which is the vibrating plastic surface condenser consists of a very thin plastic sheet mounted on a frame. The plastic sheet vibrates by the impingements action of pressurized water sprayed from the rear side. Experimental tests were done to investigate condensate production rate per unit area. These results were then compared to conventional vertical fin-tube bank condenser designs used by other investigators. The comparison showed that a higher rate of condensate production (an increase of 16.4%) is achieved using the plate channeled which is simpler and less expensive condenser than other designs. On the other hand the vibrating plastic surface condenser did not meet the expected performance.

1. Introduction

In many parts of the world fresh water shortages have led to almost total dependence on desalination. The desalination techniques most widely used are based on thermal and reverse osmosis technologies. Recently, technologies based on air humidification-dehumidification processes in greenhouses have been proposed and investigated (Paton and Davies [1], Sablani [2] and Dawoud et al.[3]). See fig. 1.

The hot and dry air entering the green house is humidified using seawater. While in the green house this humidified air is then heated even further by the green house effect. The water vapor content in this air is then increased further as it passes through a second humidifier upon exiting the greenhouse. Finally fresh water is extracted from this air as it passes through a condenser. One major component vital to greenhouse desalination process is an efficient condenser that can deal with moist air with a high concentration of noncondensable gas (the air).

1.1. Literature review and theoretical back ground:

The presence of noncondensable gas (NG) leads to a significant reduction in heat transfer during condensation. The effect of NG may be understood by looking at the condensation boundary layer shown in fig. 2 which shows a wall at temperature T_w on which a film of condensation occurs.

The vapor condenses on the wall leaving the noncondensable gas (NG) in the free stream. The highest concentration of noncondensable gases, x_{ai}, occurs at the condensate-mixture interface relative to its concentration in the bulk (free stream) mixture (x_{ab}). The partial pressures of water vapor p_v and NG p_a, vary across the condensation boundary layer while their sum is constant and equal to the bulk mixture pressure.

In the condensation of pure water vapor (condensation without NG) only thermal resistance between the interface and wall will affect heat transfer between bulk flow and the wall. In the case in which the bulk flow contains a high concentration of noncondensable gases (condenstation with NG) at the wall, three resistances occur compared with one for pure vapor. These resistances are: thermal resistance between the interface and wall, forced convective heat transfer resistance between bulk flow and interface plus mass transfer resistance due to the diffusion of vapor to the interface.

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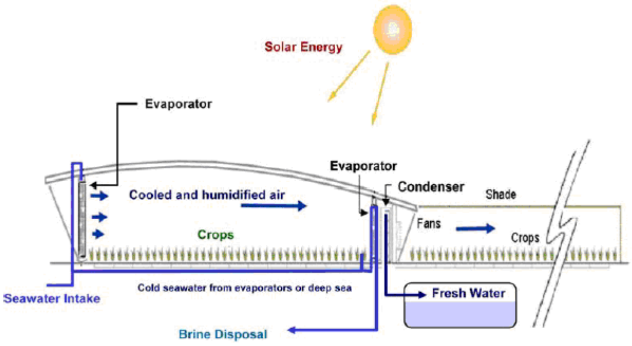

Figure 1:Schematics of green house for sea water desalination [4].

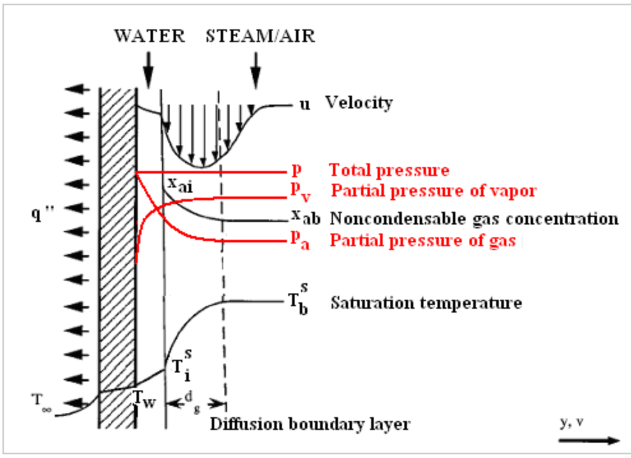

Figure 2: Schematic illustration of condensation with non-condensable gas.

The NG effect can be seen clearly in the experimental results in fig. 3. From this figure an 80% reduction in condensation heat transfer coefficient can be observed for just 8% volume fraction of air in the steam.

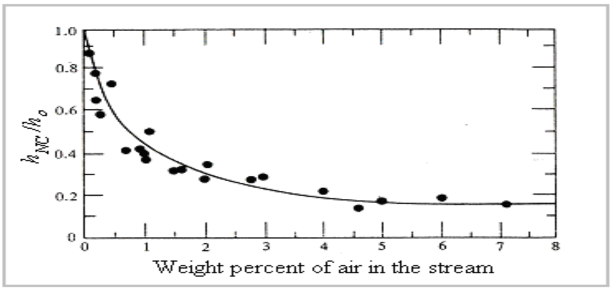

Figure 3: Effect of air on condensation heat transfer [5].

Eslamimanesh andHatamipour (2007), made a Mathematical modeling of a direct contact humidification– dehumidification desalination process. In this model a computer program was written using mass and energy balances for modeling the process behavior. The parameters considered in this work were inlet air and fresh water recycle temperatures, inlet air flow rate, saline water and fresh water recycle flow rates, and saline water to air flow ratio. Results of simulation showed that increasing inlet air and fresh water recycle flow rate increases fresh water production [6].

Sablani, et al. (2002), presented simulation of fresh water production using a humidification and dehumidification seawater greenhouse. A thermodynamic simulation study was performed to investigate the influence of different parameters affecting the greenhouse desalination process [2]. The performance of the seawater greenhouse for various climate conditions was investigated by Paton and Davis for a greenhouse of 10^4 m^2 area. Table 1 shows the performance of seawater greenhouse in terms of fresh water production and the power consumption of the fans and pumps per cubic meter of fresh water.

Table 1: Performance of seawater greenhouse for three climate conditions [4].

Climate condition	Total fresh water produced, m^3/year-hectare	Power consumption, kWh/m^3
Temperate	20,370	1.9
Tropical	11,574	1.6
Oasis	23,529	2.3

Dawoud et al. (2006) studied the possible techniques to cool the condenser of greenhouse desalination plant. Based on their study the recommended technique for cooling the condenser is the use of the deep sea water as a coolant. Different condenser inlet and outlet temperatures have been estimated and compared with cooling machines capacitates. Cooling capacity in the order of 1 MW is required for a typical greenhouse air flow rate of 15 m^3/s. This translates into a 100 tons/h of deep seawater flow rate to adequately remove the same cooling capacity [3].

Davies and Paton (2004) reviewed theories of the condensation process and described experiments conducted by the authors using two types of condenser with the same surface area of 1.25 m^2. The first type is a *standard tube-and-fin condenser* made of metal. The second type is called watermaker condenser which was developed specifically for greenhouse desalination. This design consisted of a 5×5 square array of vertical polythene tubes each 32 mm in diameter and 500 mm in height with a pitch of 38 mm between centers. The air flow direction was perpendicular to the axes of the tubes. The results are presented in Table 2.

Table 2: Condensation results for two types of condensers [1].

Type of Condenser	Fluid Temperature °C				Condensate Flow ml/hour	
	Air		Water		Observed	Corrected
	In	Out	In	Out		
Tube-and-fin	23.0	17.6	11.8	14.2	187	193
Tube-and-fin	23.0	21.2	14.2	15.6	194	186
Water-maker	25.5	21.4	12.0	14.5	367	363
Water-maker	29.5	27.3	17.2	20.2	457	423
Water-maker	29.6	23.9	16.1	19.2	477	450

The objective of the present work was to develop and test simple condensers for saturated air moisture condensation. A plastic plate channel condenser was developed and tested. The experimental setup and tests for this condenser are reported in this paper.

1.2. Methodology:

The experimental set up shown in fig. 4 consists of an air conditioning unit which provides the moist air at the desired conditions, water-to-water refrigeration unit which supplies cooling water at the required constant temperature and a test section that houses the condenser modules. The test section is equipped with condensate collection trays, one for each module. The air conditioning unit shown consists of a pre-heater, a humidification section, a refrigeration coil and a re-heater. The unit controls the air flow rate, humidity and temperature.

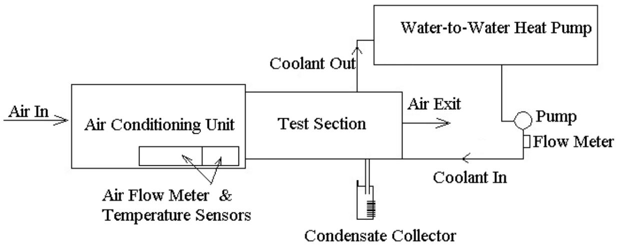

Figure 4: Experimental set up schematic diagram

The water-to-water refrigeration unit consists of a refrigeration cycle connected to two brazed plate heat exchangers one on the hot side and the other on the cold side. These two heat exchangers are connected to two 1 m^3 water tanks from which the constant-temperature coolant (water) is obtained either from the hot side or from the cold side or by mixing both.

The test section is made from transparent Plexi glass box. The first part of the test section is the calming section where the flow becomes uniform before entering the condenser box where the condenser modules are mounted. The second part is the condenser box where the condenser modules are fitted to the condenser frame. At the bottom of the condenser box a corrugated channel is located to collect the water condensate from individual condenser modules. Flow resistance baffles are installed underneath the condenser to prevent flow by-pass, i.e., force the air to pass through the channels between the condenser modules.

Based on the working conditions of the greenhouse condenser, the parameters affecting the condensation process along with their range were selected. These are: air inlet temperature range (30-50ºC), air inlet humidity range (50%-100%), air inlet velocity range (0.2-0.4m/s), cooling water temperature range (15-30ºC), and cooling water flow rate range 2-5 l/min.

Calibration of sensors such as air flow meter, water flow meter and other available devices was carried out. Although the air conditioning unit has an orifice flow meter the air volume flow rate was measured by measuring the local air velocity at the exit from the test section using a pitot tube and integrating the velocity profile. The repeatability of the measurements was tested via two experiments at two different days at the same conditions.

1.2.1. Design of Plate Channel Condenser

The plate channel condenser was made of plastic. The condensation process occurs on the outer side surface of the channel with coolant flowing vertically inside the small passages. The plate channel is connected to a header and a return pipe made from aluminum which has been slotted to accommodate the plate channel. Coolant flow visualization using dye was carried out and showed uniform flow distribution among the small vertical channels of the unit. See fig.5.

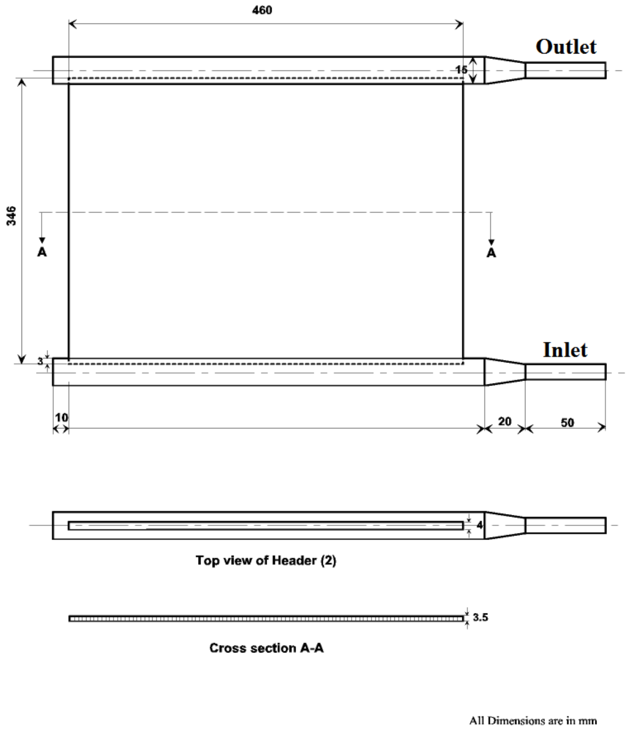

Figure 5: Plate channel condenser module

Figures 5 show the dimensions of the condenser unit manufactured at the workshop of the Institute of Technical thermodynamics University of Aachen, Germany.

1.2.2. Vibrating Plastic Surface Condenser

One of the challenging areas in condensation is to maintain drop-wise condensation. Drop wise condensation results in heat transfer ten times more than of film condensation [7]. Based on this a new design was built with vibrating heat transfer surface as it may produce conditions similar to drop wise condensation. It also can minimize the condensate layer thickness to reduce the resistance to heat flow.

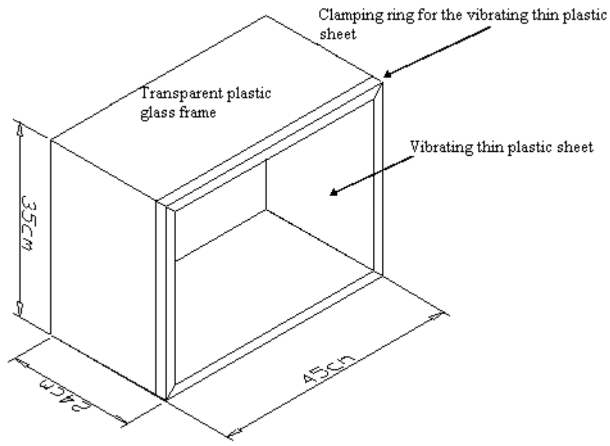

Figure 6: Vibrating plastic surface

This design presented in fig. 6 is made of plexiglass box with thin plastic sheet in front and three nozzles spraying water on the sheet. The water nozzles provide a very good coverage of surface area and vibrate the sheet.

1.2.3. Experimental Setup and Procedure

The experimental results both condensers were obtained for different parameters: the air inlet temperature, humidity, velocity, flow rate and coolant (water) temperature. Prior to each run the air conditioning unit was set at the conditions required and let run for one hour to achieve a steady state air condition. This is to ensure constant parameters of the air (temperature, humidity and flow rate) during the experiment. Also, in the case of channel plate condenser three plate modules were installed side by side at certain spacing and the condensate yield from the middle unit is reported as it has identical (symmetric) conditions on both sides. See fig. 7.

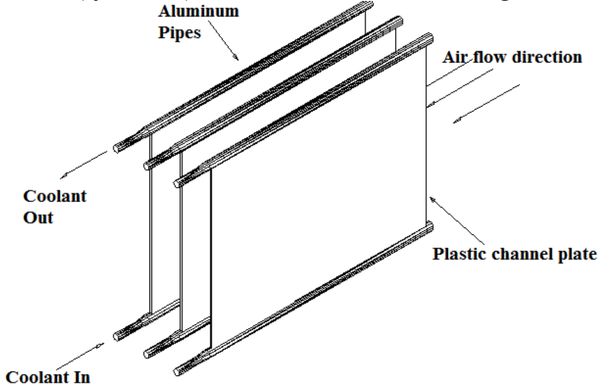

Figure 7: Plate channel condenser arrangement**.**

This is to eliminate the effect of the test section side walls. Some of the experimental results obtained in this work incorporated a variation of spacing between the condenser units.

2. RESULTS AND DISCUSSIONS

Figure 8 presents the effect of air inlet dry bulb temperature, relative humidity, coolant temperature and coolant flow rate on condensate yield at constant air velocity of 0.2 m/s. It is seen that the higher the air dry bulb temperature (T_{db}) and humidity the higher the condensation rate (compare the curves for a 100% relative humidity at 44 and 36 ^{o}C dry bulb temperatures).

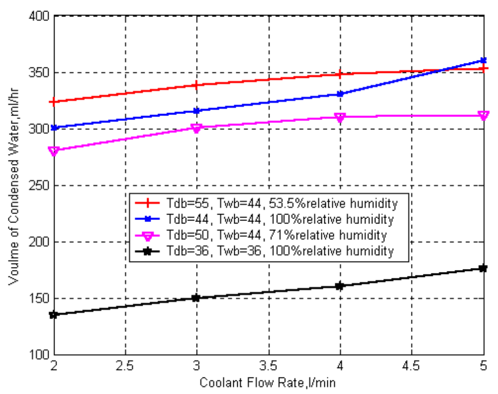

Figure 8:Water condensate production at constant coolant temperature of 15 ^{o}C and constant air velocity of 0.2 m/s

The test at 55^{o}C dry bulb temperature produces more condensate yield than the one at 44^{o}C despite the higher relative humidity for the latter. This is because the former has a higher difference between the air and coolant temperatures (11^{o}C higher). The same behavior can be observed with coolant temperature set at 20^{o}C while under the same air conditions.

This result is quite motivating for introducing design improvements on the greenhouse which has been recommended by Dawoud et al. A particular design incorporates a secondary channel in the greenhouse roof serving as a heater for a portion of the air flowing into the greenhouse. At the exit from the green house, mixing of hot air exiting the secondary channel with that flowing below through the greenhouse results in hotter exit air temperature before entering the second evaporator. The net effect is a higher temperature of an almost saturated air at the inlet to the condenser, yielding higher rate of water condensate [3].

The above results are consistent with the condensation theory. That is, one of the major factors affecting the condensation process is the temperature difference between the bulk air-vapor mixture and the cooling surface temperature. While the surface temperature was not measured (a very thin plastic plate of 50 micron thickness) it is assumed that the surface temperature is very close to the coolant temperature, i.e., 15 ^{o}C in fig. 8. By looking at the temperature difference we can see that the curves for the dry bulb temperatures of 44 ^{o}C and 36 ^{o}C have temperature differences (T_{db}-$T_{coolant}$) of 24 and 16 ^{o}C, respectively. By increasing the coolant flow rate, more heat can be extracted by the coolant resulting in higher condensation rate.

Figure 9 shows the effect of coolant temperature on condensation rate at different air inlet conditions.

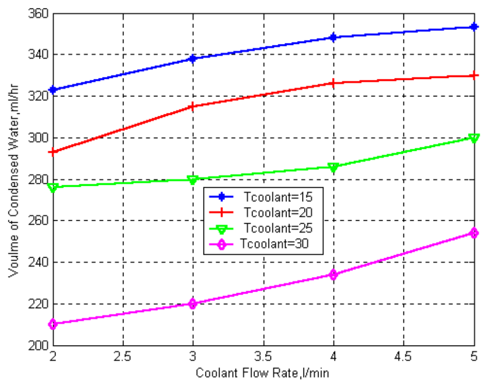

Figure 9:Effect of coolant temperature on condensate yield at air inlet conditions of T_{db}=55 ^{o}C, 53% relative humidity at constant air velocity 0.2 m/s.

The general trend is that for all coolant temperatures and flow rates the lower the coolant temperature the higher the condensation rate. However, the variation trend with coolant flow rate varies especially at low coolant temperature. That is, in fig. 9 the curves for coolant temperatures of 15 ^{o}C and 20 ^{o}C level off contrary to the case shown in fig. 10. In an attempt to explain this behavior it was initially thought that experimental errors are responsible. However, the pattern persisted despite the repetition of the experiments. The only thing noticeable here is that for the case of fig. 10 the air is fully saturated.

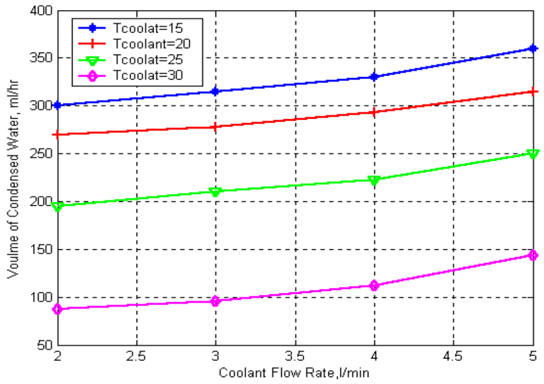

Figure10: Effect of coolant temperature on condensate yield at air inlet conditions of T_{db}=44 ^{o}C, 100% relative humidity at constant air velocity 0.2 m/s.

To study the effect of air velocity on the condensation with noncondensable gas, two cases were tested each with different velocities and coolant temperatures. Figure 11 shows the effect air velocity has on the condensation rate for a coolant temperature of 15 ^{o}C. Higher air velocity produces more condensate yield than lower air velocity. This could be attributed to two factors: an increase in condensation heat transfer coefficient and the increase level of turbulence which is more effective in counteracting the adverse effect of noncondensable gas. The increase reaches as high as 140 ml/h (or 45%). Similar results have been arrived at for the second case tested. That is, for the coolant temperature of 25 ^{o}C the behavior is quantitatively similar (i.e., 140 ml/h) but with higher percentage increase, i.e., 65%.

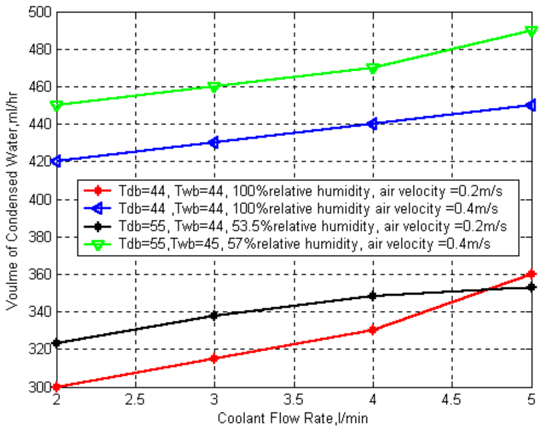

Figure 11:Variation of water condensate by varying the air speed at constant coolant temperature of 15 ^{o}C

To study the effect of spacing between the three condenser units experiments were conducted at two different spacing, each at two different velocities. Results are presented in Table 3, which also show high repeatability in the experimental results. Note that each run had been made three times at the same conditions to make sure of repeatability of data.

By looking at Table 3 it is seen that although small, the effect of a larger space between plates increases the condensation rate. This slight increase could be attributed to the increase in the volume of air and thus the amount of vapor that is brought into contact with the condenser surface. That is, at constant air velocity more humid air passes through the channel between the condenser units. As a result, it may be stated that the slight increase seen as a result of the spacing is bought at the expense of compactness which is much more effective in boosting the condensate yield of the condenser in question.

Table 3: Effect of air velocity on water condensate yield**.**

Test #	Water condensate (ml/hr) at air speed of 0.2m/s 5 cm spacing	Water condensate (ml/hr) at air speed of 0.2m/s 10 cm spacing
1	80	86
2	80	84
3	82	86
Avg.	81.0	85.0
	Water condensate (ml/hr) at air speed of 0.4m/s 5 cm spacing	Water condensate (ml/hr) at air speed of 0.4m/s 10 cm spacing
1	112	126
2	112	128
3	114	126
Avg.	113.0	127.0

2.1 Heat Transfer Enhancement

In order to boost the condensation rate it was thought that promoting turbulence and mixing of bulk stream will reduce the concentration of the noncondensable gas near the cooled surface. Hence, two kinds of fins (0.5-cm and 1-cm high fins) were mounted on the condenser wall surface. See fig. 12 for dimensions and table 4 for results.

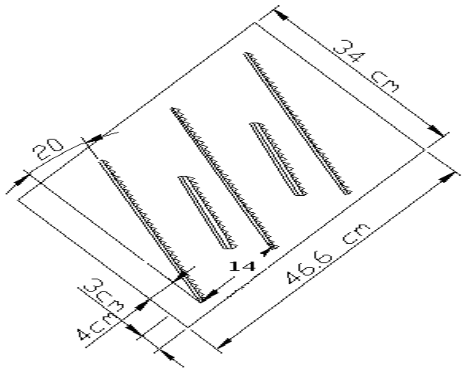

Figure 12: Fin distribution on condenser surface

Table 4: Effect of fins on water condensate (ml/hr)

Air velocity	5cm spacing without fins	5cm spacing, with three 0.5cm fins	5cm spacing, with three 0.5cm and two 1cm fins
0.4 m/s	112	116	124
	10cm spacing, without fins	10cm spacing with three 0.5cm and two 1cm fins	
0.4 m/s	126	132	

Table 4 gives the results for the case considered. It is shown that with the addition of fins there is a slight increase in water condensate. This increase is not significant to warrant a concrete conclusion. Higher fins or different fin arrangements may lead to more conclusive evidence. This is recommended for future works.

2.2 Vibrating Plastic Surface Condenser

Vibrating plastic surface condenser was tested to evaluate it is performance. Some problem was observed such as nozzles blockage occurred because small solid particles in the water. Furthermore, for effective spraying a distance between the nozzle and the surface should be large enough which makes the condenser bulky. Also the nozzles used are expensive which defeats the purpose of obtaining inexpensive condenser.
The results of water production for this vibarting surface can be shown figs. 13 these results are obtained at air velocity of 4 m/s and coolant flow rate of 2.78 l/min.

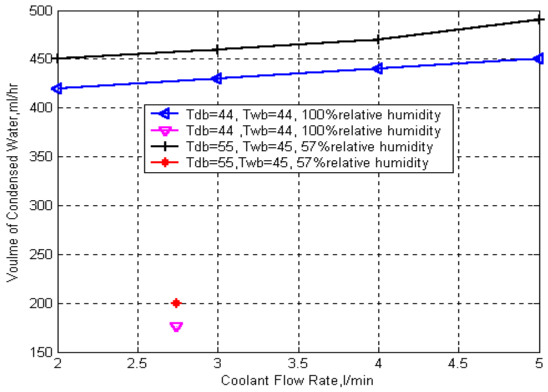

Figure 13. Comparison of produced condensate water (ml/hr) for the vibrating plastic surface condenser-II (water flow rate of 2.74 ml/min) and flat plate channel condenser at air velocity of 0.4 m/s and constant coolant temperature of 15^{o} C

Two experiments (indicated by a * and a ∇) were conducted with the vibrating surface condenser and compared with the results from the plate channel condenser. The results show a large different as the condensate yield from the vibrating surface condenser constitutes about 40% of that from the plate channel condenser, i.e., 260 ml/hr as opposed to 455 ml/hr.

This could be attributed to spray pattern as the coolant did not have enough time to cool off the surface due to water droplet bouncing back from the surface. Furthermore sprayed water did not cover 100% of surface area as in the chandelled plate condenser which considered a fall down for this condenser. More work in this area is needed.

3. CONCLUSIONS

The comparison showed that higher rate of condensate production is achieved using the plate channel condenser which is simpler and less expensive than the more elaborate tube bank design. An increase in fresh water production rate of 16.5% is achieved using the plate channel condenser developed in this work compared with water maker. The air temperature and humidity have significant effect on the condenser condensate yield. Higher temperature and humidity produce more condensate for the same flow rate and coolant temperature. The results presented in this work support the modification to greenhouse design proposed by Dawoud et al. [3]. The tests also showed that the effect of air velocity is significant (increase the water condensate) while the spacing between the condenser modules has a marginal effect. Condensation enhancement using fins mounted on the surface of the condenser has been attempted. Although the results show a marginal enhancement they may be considered preliminary.

The vibrating surface condenser results show a large different in the condensate yield compared to the channeled plate condenser about 40% less, i.e., 260 ml/hr as opposed to 455 ml/hr. this can be justified by the water spraying pattern .

4. FUTURE WORK

Vibrating surface condenser concept should be combined with channeled plate condenser to achieve the highest condensate yield. Surface conditions for both condensers should be investigated. Finally fin design and spacing should be investigated.

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NOMENCLATURE

NG noncondensable gas

P total pressure

P_V partial pressure of vapor

P_a air partial pressure of gas

$T^{s}_{b/i}$ saturation temperature

T_w wall temperature

T_{db} dry bulb temperature

T_{wb} wet bulb temperature

$X_{ab/i}$ noncondensable gas concentration

Subscripts

b bulk

i interface