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COMPUTATIONAL FLUID DYNAMICS ANALYSIS OF FINNED TUBE DEHUMIDIFIER FOR SOLAR HUMIDIFICATION DEHUMIDIFCATION DESALINATION PLANT

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ABSTRACT

Desalination by air Humidification Dehumidification (HDH) is one of the upcoming technologies of converting brackish water into freshwater. It works by evaporating the brackish water and humidifying air passing over it, leaving the salts behind. The humidified air is then passed through the condensers to recover the water it carries by condensation of water vapour. This condensed water is free from salts and can be directly used or can be further treated for use. The use of a compact finned tube heat exchanger for the purpose of the 2nd dehumidifier in a two stage HDH process is carried out by using computational fluid dynamics (CFD) analysis. The boundary condition for the computational work is obtained from experimental setup in a pilot plant. As CFD models can take up very large computational power for solving fin structures, the model is solved by modelling the fins side of the dehumidifier as a porous medium. The water yield obtained from condensation, thermal characteristics of the fluid model are obtained is compared with experimental results. The yield of water obtained is 3848 ml/hour for the first set of experimental observations.

Key words: CFD, humidification, dehumidification, compact heat exchanger, condensation

INTRODUCTION

Availability of fresh water on the surface of earth is very less. It is not only used for drinking, but also for agricultural and industrial purpose. The availability of drinking water is drastically reducing due to population growth, atmospheric changes due to industrialization, decrease in forestry rate and unsustainable consumption of water. Researchers are working to find solutions for potable water from brackish or saline water available plenty but cannot be used directly. Methods like Reverse Osmosis (RO) and electric desalination require a lot of capital, research and have a higher energy usage. Humidification Dehumidification (HDH) is a novel technique, which uses low temperature energy for water desalination. HDH process works on the principle of water vapour mixing with air, it is also observed that the water vapour carrying capability increases with increase in air temperature. Approximately 0.5 kg of water vapour can be carried by 1 kg of dry air when its temperature increased from 30°C to 80°C (Kabeel, et al., 2013). Use of computational analysis for heat and mass transfer studies instead of experimental setup is done to reduce the cost associated with the setup and labor. CFD is a tool used to predict the thermal characteristics of heat exchangers. However, CFD models take considerable amount of computational power and time. The study of the 2nd dehumidifier is done using ANSYS® CFX® 15.07 software and compared with the experimental result to validate the simulation.

Saravanan and Vijaya Lakshmi (2017) presented the drinking water pollution and its severity in major cities of India with pollution index value. They also found that many of the urban population is facing this problem due to flushing of industrial waste from leather, oil, chemical and process industries. Ezhilarasi et al., (2017) analysed the economic feasibility of photo voltaic, solar thermal and hybrid photo voltaic thermal systems in comparison with conventional system for a hostel building. Priyanka et al., (2017) studied the feasibility of photo voltaic, solar thermal and hybrid photo voltaic thermal systems for rural row houses. Economic and emission analysis gives hybrid photo voltaic solar thermal system gives an optimal solution compared to conventional or other individual solar systems. Bourouni et al., (2001) conducted a study on the process of desalination using HDH. They concluded that at the moment, it is suitable for small scale localized demand for desalination. Ettouney (2005) studied four different configuretions humidification dehumidification, humidification dehumidification- vapour compression, desiccant and membrane drying for desalination. They found that the main drawback in all the configurations is the large quantity of air that needs to be circulated. This alone hampered the efficiency of their setups. The solution for this was suggested as the increase in condenser size. Ettouney and Al-Sahali (2008) conducted experimental studies as well as mathematical modelling of the HDH process. They worked out a correlation for the heat transfer coefficient. They concluded that heat transfer coefficient for humid air is approximately 10 times that of dry air. Also, they suggested that for the best possible efficiency, finned heat exchangers need to be used. Hatamipour and Eslamimanesh (2009) worked exclusively on the mathematical modelling of the HDH process. They studied the effect of different process parameters on the production rate/yield by applying principles of mass and energy balance on the humidifier and dehumidifier. They found that by increasing the air recycle flow rate and the inlet air temperature, coefficient of heat transfer increased and hence the yield of fresh water also increased.

Yamali and Solmus (2008) conducted experimental and numerical analysis on a solar HDH setup with double pass solar heater, fin and tube heat exchanger with air cooling. They found that by incorporating a double pass solar heater, the yield was increased by 15% when compared to single pass solar heater. The yield also increased when it is integrated with evacuated tubular solar water heater. Their experiments also confirmed previous findings regarding the correlation of yield and parameters like mass flow rate, inlet temperatures. Amer et al., (2009) conducted a theoretical and experimental study for a CAWO HDH setup. They used three different packing materials in their experiments and compared the results using with those obtained from empirical relations. They found that increasing the water temperature at the humidifier, the production rate increases. Also, increasing the flow rate of water increases all parameters except the temperature at the humidifier. This showed that a balance has to be worked out between the flow rate and the water temperature at the humidifier. Chiranjeevi and Srinivas (2014, 2015, 2016) carried out thermodynamic studies, and conducted experimental and simulation studies on a two stage HDH desalination plant. They found that by adding a second stage of desalination, the yield of fresh water improved significantly. It was observed that low hot water inlet temperature did not generate much desalination at the first stage plant, but no such issues were observed for the second stage of desalination. Kang et al., (2015) conducted performance evaluation on a 3-stage regenerative HDH desalination process. They found that the decreased temperature range of the circulation improved yields enormously. GOR of up to 5.13 was attained as compared to a GOR of 2.34 for a two stage HDH desalination plant operating under the same conditions. Chiranjeevi and Srinivas (2016) studied the influence of vapor absorption refrigeration (VAR) operating parameters on two-stage integrated HDH desalination and cooling plant to get optimum conditions. Chiranjeevi and Srinivas (2017) theoretically evaluated the performance of a single stage conventional HDH desalination system, conventional HDH desalination with VAR cooling and an integrated two stage HDH desalination with conventional and VAR cooling in first and second stage respectively. The study reveals two-stage integrated HDH desalination plant with VAR cooling in second stage give better energy utilization compared to other options.

Compact heat exchangers are best for high heat transfer rates due to high surface area suffer from a characteristic called maldistribution which causes improper utilisation of exchanger surface which is discussed by Saad et al., (2014). Problems of using CFD studies for multiphase simulation in compact heat exchanger are because of the complexity of solving. The improvements for such problems are brought in direct numerical simulation in multiphase flow by Tryggvason et al., (2005). Lee et al., (2012) emphasised the use of different CFD software for direct numerical simulation finite volume method. Hooman and Gurgenci (2010) carried out simulation of a fin-tube heat exchanger as a porous medium approach. It concludes that if the macroscopic properties of the porous medium (porosity, permeability and form drag coefficient) are determined, the results don't fluctuate to the internal structure of the porous matrix. Rohit and Gosai (2013) use the porous medium approach to find out the thermal characteristics of the heat exchanger. Barak et al., (2014) worked on CFD simulation of condensation of moist air in heat exchangers using comercial CFD code and not using User Defined Functions (UDF) for the condensation process. Recently Marale et al., (2017) carried out CFD studies on a two-stage integrated cooling plant to evaluate the performance of shell and tube heat exchanger as dehumidifier using chilled water. They validated the CFD model by comparing the model results with experimental observations.

METHODOLOGY

Humidification Dehumidification (HDH) desalination works on the principle of evaporating water into air, leaving the salts behind and then condensing the air-water vapour mixture to obtain desalinated water. The process setup consists of multiple air humidifiers and dehumidifiers, air-preheater and water heater. The process starts with preheating the air and heating the water in the water heater. These two are then mixed in the humidifier to evaporate the water. This air-water vapour mixture is then passed into multiple dehumidifiers, depending on the number of stages in the HDH process.

Different types of HDH desalination arrangements are discussed in the following section. Open Air Open Water (OAOW) type has both the air and cooling water circuit open. The air is taking in from the atmosphere, run through the plant and exited to the atmosphere. This type can also be used to achieve district cooling as the air obtained after dehumidification is cooler. The cooling water is obtained from a large water body present near the desalination plant. Open Air Closed Water (OACW) type uses a cooling water loop which is closed. The water used in the condenser is routed through a cooling pass such that the coolant temperature drops down. Such systems are used in pilot plants as cooling water in such are used from a refrigeration loop. Closed Air Open Water (CAOW) type of loop recycles the air used from the last stage of condensation. It is generally used in areas where purification of air for the humidification process is costly. In the present study as shown in Fig.1 OACW two stage HDH desalination system is considered and the dehumidification process considered in the second dehumidifier. As shown in the Fig.1 the atmospheric air sent into the 1st air preheater for heating with hot water and then it enters the 1st humidifier for further heating and humidification follows air cooling and dehumidification in 1st dehumidifier resulting in desalinated water which completes the 1st stage HDH desalination. The air will undergo same processes for 2nd stage HDH desalination in the respective components. The dehumidification takes place in an air-cooled dehumidifier followed by a chilled water dehumidifier. The hot water is supplied from solar water heaters for air preheaters and humidifiers.

The modelling of the heat exchanger is carried out in Solidworks® 2014 Computer Aided Drawing (CAD) software. The modelling of each component of the heat exchanger is explained. Due to computational power constraints, the model is divided into 4 equal flow channels. The initial coolant water supply and outlet channel are studied separately to ease computational difficulties. In the later part of the chapter, the modelling of the fin side of the heat exchanger is done as a porous medium and the determination of the porous domain coefficients is carried out.

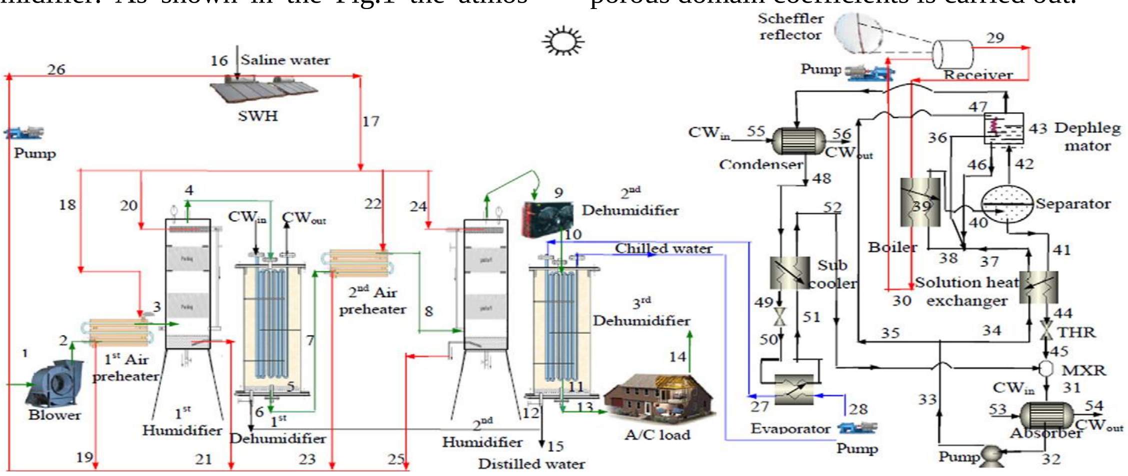

Fig. 1: Two-Stage Humidification Dehumidification Plant schematic (Chiranjeevi and Srinivas, 2014)

The heat exchanger in study is a compact heat exchanger with corrugated fins which is available commercially. As the computation power required for such a heat exchanger is very high, the fins are modelled as porous domain. The computational power for even such a model is very high as phase change modelling requires very fine mesh sizes. Thus, the total model of the heat exchanger is divided into 4 passes containing 2 outer and inner loops. The velocity of inlet of the loops is found out using inlet feed tube velocity simulations. The outlet velocity is also simulated from the outlet velocities of each loop. The porous medium coefficients are obtained from the domain model. To configure the porous domain the porosity coefficient of the domain is required, which are mentioned in Table 1. The inlet and outlet channel are a 350 mm tube of 10 mm inner diameter and 12 mm outer diameter. The coolant is fed through the single hole on one side and the outlet to the four loops of the heat exchanger using the four outlets.

Table 1: Porosity data for simulation.

Porosity Characteristics	Value	Unit
Porous Volume	0.012306	m^3
Fins Volume (for 323 fins)	0.003068	m^3
Interfacial Surface Area	13.68971	m^2
Interfacial Area Density (IAD)	1112.442	1/m
Porosity	0.750726	

As the inlet to the initial tube is only known from the experimental data, the velocities at each outlet is to be determined using CFD. The mesh generation is done is ANSYS® Mechanical solver. An unstructured mesh is generated for the model. The water flow rate through the heat exchanger is 128.1 liters per hour (LPH) is obtained from the experimental setup in the lab. The boundary conditions for inlet channel of the heat exchanger are represented in Table 2.

Table 2: Boundary Conditions for Inlet Channel.

Boundary Condition	**Type of Boundary**	**Value**	**Unit**
Inlet	Velocity Inlet	$V_{in} = 0.451$	m/s
Outlet 1	Static Pressure	$P_{gauge} = 0$	Pa
Outlet 2	Static Pressure	$P_{gauge} = 0$	Pa
Outlet 3	Static Pressure	$P_{gauge} = 0$	Pa
Outlet 4	Static Pressure	$P_{gauge} = 0$	Pa

For simulation the default values of relaxation factors are set and a minimum value of $1x10^5$ was obtained for all residual values. The simulation is run at isothermal as the tubing is insulated and no heat simulation of such is conducted. The turbulence model is set as k-ε turbulence and at low intensity due to very low flow rate.

RESULTS AND DISCUSSION

The flow characteristics of the finned tube compact heat exchanger are developed by giving the boundary conditions from the experimental data. The variation in fluid temperatures, velocities of the fluid from entry to exit are obtained and compared with the experimental results for validation of the model. The velocity streamlines and outlet velocities of the inlet channel are shown by the Fig. 2a. Outlet 1 and 4 are the ones situated at the ends of the inlet channel and Outlet 2 and 3 in the middle. The velocities of the two-outlet channel in the outer side of the heat exchanger and the two in the middle are same due to symmetry. The outlet velocity on the outer outlets is higher than on the inner outlets. Average inlet channel velocities are shown in Fig. 2b. The Table 5 gives the average velocity of the fluid in different loops of the heat exchanger. The simulation of the heat exchanger is divided into 2 outer and 2 inner loops. The outlet velocity of water from the channel is found out using the simulation. The boundary conditions used for simulation are listed in Table 3.

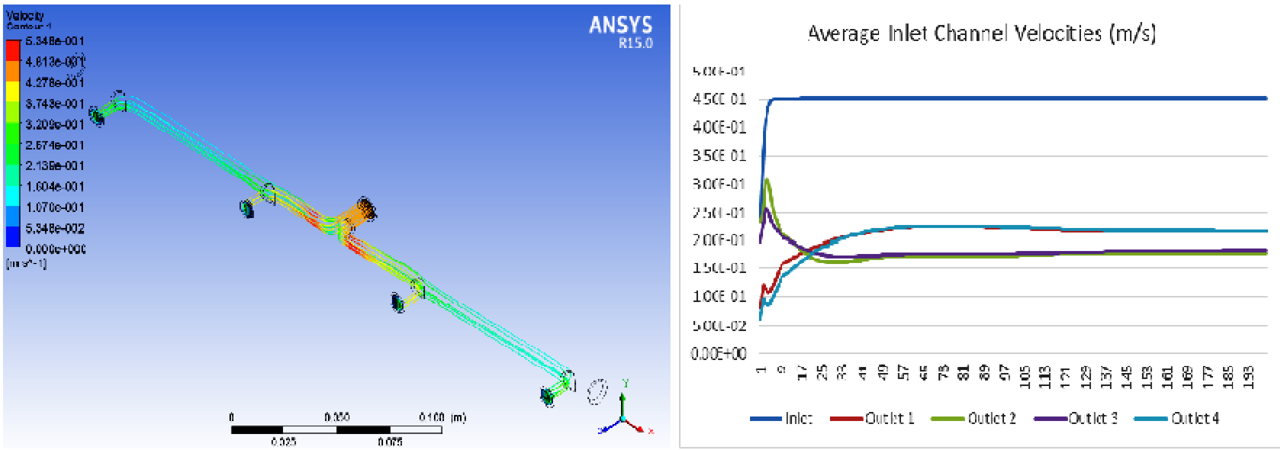

Fig. 2 (a): Velocity Streamlines and outlet contours of the inlet channel, (b) Comparison of average inlet channel velocities.

Table 3: Boundary Conditions for Outlet Channel.

Boundary Condition	**Type of Boundary**	**Value (V_{in})**	**Unit**
Inlet 1	Velocity Inlet	0.2138	m/s
Inlet 2	Velocity Inlet	0.175	m/s
Inlet 3	Velocity Inlet	0.175	m/s
Inlet 4	Velocity Inlet	0.2138	m/s
Outlet	Static Pressure	$P_{gauge} = 0$	Pa

For simulation the default values of relaxation factors are set and a minimum value of $1x10^5$ was obtained for all residual values. The simulation is run at isothermal as the tubing is insulated and no heat simulation of such is conducted. The turbulence model is set as k-ε turbulence and at low intensity due to very low flow rate. The outlet velocity stream lines of the outlet channel are shown in Fig. 3(a) and the average outlet channel velocity at different points are shown in Fig. 3(b).

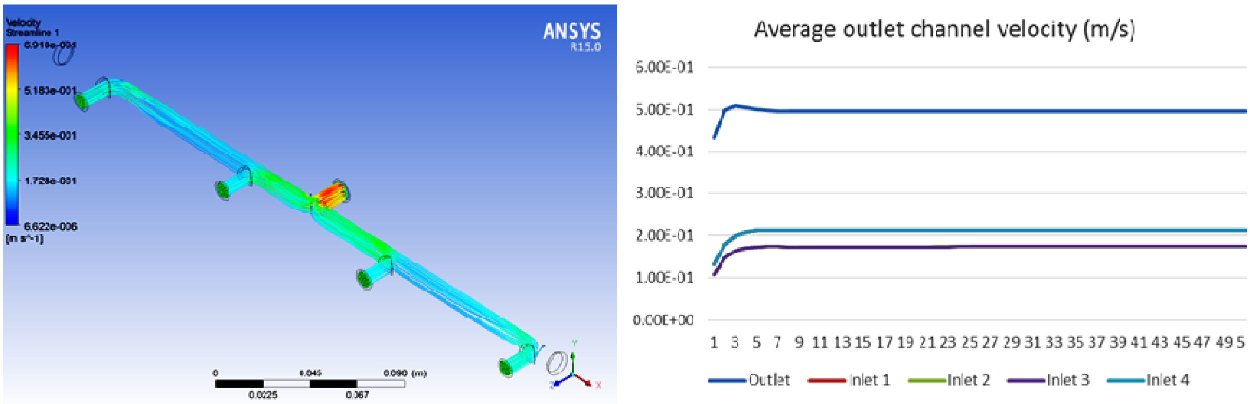

Fig. 3(a): Velocity Streamlines and outlet contours of the outlet channel, (b) Comparison of average outlet channel velocities

Experiments were carried out on a pilot plant is setup in the lab by varying key operating parameters to obtain the fresh water yield are shown in Table 4. The experimental values obtained are used for comparison with CFD simulation results to validate the model. The simulated model is tested for with different test data taken from the experimentation for validation. The CFD simulation results are compared with experimental results. The variations in the results are tabulated for the 5 sets of reading.

The percentage error between simulation and experimental observations of cooling water temperature at outlet, exit temperature of the cooled air and desalinated water generated in the dehumidifier is shown in the Table 5. Fig. 4a, Fig. 4b and Fig. 5a shows the variation fresh water generation result by CFD with experiment for different set of observations taken and Fig. 5b represents the percentage error for the respective readings. The initial volume fraction of air and water vapor in humid air are taken as 0.8 and 0.2 respectively. The result shows the volume fraction of water vapor is reduced due to the cooling and dehumidification i.e. some amount of desalinated water is generated.

Table 4: Experimental observation from the pilot plant.

Particulars	**Unit**	**Set of Observations**				
Inlet Air Temperature to Dehumidifier	°C	45	45.5	42.5	39	37
Outlet Air Temperature from Dehumidifier	°C	25.8	26	26.9	27.2	27.6
Inlet Water Temperature to Dehumidifier	°C	7.75	8.24	10.3	10.9	12
Outlet Water Temperature from Dehumidifier	°C	22.6	22.8	23.7	24	24.6
Air Flow rate in Dehumidifier	m^3/h	15	15	15	15	15
Cooling Water flow rate in Dehumidifier	LPH	128.1	128.1	128.1	128.1	128.1
Desalinated Water yield	LPH	3.4	3.33	3.085	3.03	2.925

Table 5: Percentage error between simulation experimental observations.

Cooling water temperature at outlet	**Cooled air temperature at the outlet**	**Fresh water generation**
0.6275	0.3882	13.1817
0.6592	0.3929	14.1211
0.6993	0.3967	16.0092
0.6835	0.3881	16.1776
0.6804	0.4141	16.7746

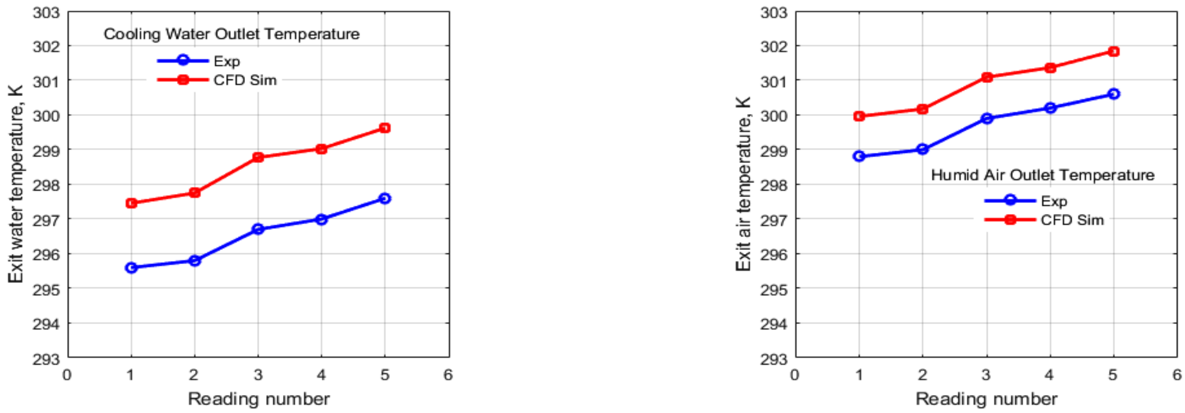

Fig. 4a: Comparison of coolant water outlet temperature, (b) Comparison of humid air outlet temperature.

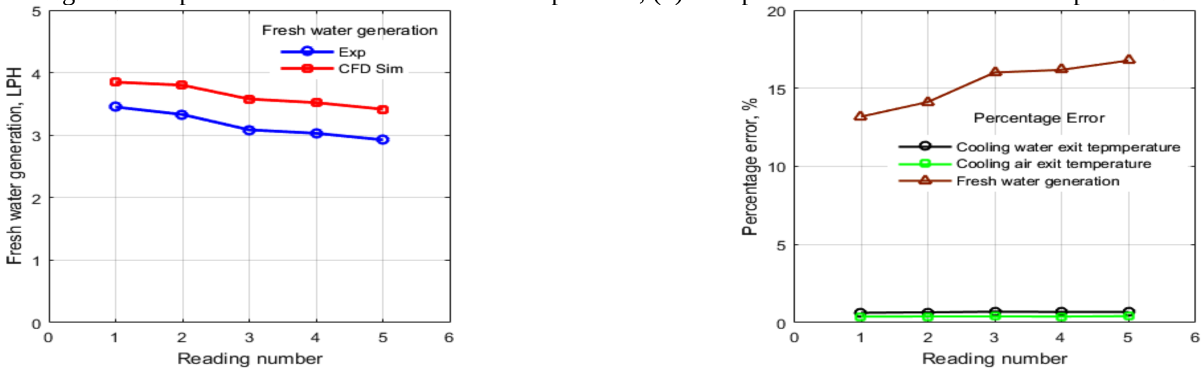

Fig. 5a: Comparison of fresh water generation, (b) Percentage error variation of CFD result

CONCLUSIONS

The CFD analysis of the compact heat exchanger used for air dehumidification in a two stage HDH desalination plant. Yield of water condensation is the matter of study in the project. The analysis is carried out for 15 m^3/hour of humid air flow and 128.1 LPH of cooling water flow. To reduce the complexity of the model for simulation, use of porous material modelling is carried out. The model is developed from the commercial off the shelf heat exchanger which is used in the experimentation process. The model analysis is carried out as 4 separate loops which are fed by an inlet channel and completed with an outlet channel. The dual benefits, desalinated water and air cooling from the plant depend on the dehumidifier performance. So, to study the same a CFD simulation model is developed for the dehumidifier used in the plant and validated the model with the experimental operating conditions. The CFD results showed the temperature of humid air is reduced from 318 K to 299.96 K. The volume fraction of the water vapour is reduced from 0.2 to 0.174 as some of the water vapour condenses into liquid water which is desalinated water. The average percentage deviation in the temperatures is 0.396 % for humid air temperatures, 0.669% for cooling water and 15.25328% for yield of condensation.

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