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Effect of Fan Speed and Ambient Temperature on the Performance of Air Cooled Condenser- A Numerical Study

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Abstract
In present trends of power generation in power plants, Air cooled heat exchanger has got a wide acceptance due to reduced number of components and low cost. Further Air cooled Condensers has occupied a major place in power plants whether it is a closed or open cycle unit Presently the Air cooled condensers are of great importance which is taken for m study. The flow of the steam and condensation of steam is done by maintaining the pressure to a certain level. In this paper focus is given to understand the effect of Fan speed and ambient temperature on the effective performance of Air cooled Condensers. Geometry of the tubes and fins is constructed using CATIA V 5. Numerical simulation using CFD ANSYS Fluent is done to see the effects at various velocities of air coming from the fan and various ambient conditions. Eulerian model is used with Realizable k – epsilon model. A 6.5 MW power plant is chosen for study where the focus is on Air cooled Condenser [ACC]. A single tube is chosen for study, the first tube in the bundles of ACC. A sample simulation is shown, and finally results in form of tables are shown. The objective is to guide the manufacturers to take these things into consideration before getting the right thing installed.
Keywords*:Air cooled heat exchangers [ACHE]; Air cooled condensers [ACC]*

1, Introduction

An Air Cooled Heat Exchanger (ACHE) is a heat exchanger in which air is used to cool the fluid either for Condensation or bringing it to the desired temperature as per application, which isin contrast to other types of heat exchanger. The main advantage of this exchanger is that it requires less amount of water amount of waterthough the plant requires large cooling capacity. The cooling is provided by using Axial-flow fan which provides high velocity air to cool the hot fluid. It can be as small as an automobile radiator or large like big ACHEs in power plant industry or Petrochemical industry. Due to several problems like water shortage and its cost with water pollution, the use of air cooled heat exchangers has increased especially for process cooling in refineries and chemical plants. There is an increased use of Air Cooled Condensers for power stations. The basic Condensers are normally configured as an A-frame or "roof type". .Though ACHE looks simple the design of an ACHE is more complex than for a Shell and Tube Heat Exchanger, as there are many more components which affect its performance. Air-cooled heat exchangers are mainly used in the oil and gas industry, from upstream production to refineries and petrochemical plants, under extreme conditions like high pressure and high temperature conditions, as well as corrosive fluids and environments. The most used is Air cooled Condenser which is of high use in power plant industry. These Condensers run in Vacuum operated conditions. Though lot of research has taken place, the industries dealing with manufacturing of Air cooled Condensers are still looking for a practical solution to problem of condensation in extreme conditions.

1.1.Literature Survey

Effect of water spay cooling is proposed by presenting the influence of water spay rate, spray direction and nozzle distance on cooling performance of ACC and proposed an improved water spray arrangement which can cause effective performance of ACC with reduction in back pressure and concluded that upward spray direction and nozzledistance of 0.8 m is the best [1]. Thermal energy storage system is used to achieve sub ambient cooling in one of ACC and presented daily power output, annual power output and payback period for ACC and showed that all advantages of dry cooling can be achieved though this method at high temperature [2]. To present optimized values for effective performance of ACC gave an Analytical model by presenting two different models that can be used as a tool for cooling across tubes in ACC [3]. ACC model is developed to test the power to reduce the negative effect caused due to the use of techniques that lowers air side thermal resistance and increase mass flow rate of the air. It was found that 68 % increase in air flow rate decrease in air side thermal resistanceand air side pressure losses can improve the performance of ACC [4]. By taking into account various parameters like Ambient temperature, wind direction and speed, height of surrounding buildings, flow field and temperature field, standard configurations for effective performance of ACC was proposed focusing on the main cause of Hot air recirculation [5].The heat transfer process between the turbine exhaust and cooling unit of ACC is examined. Model of cooling air with circulated water were developed which included flow and heat transfer models and found that most freezing rate takes place at base of the fins due to high cooling capacity[6]].Use of porous metals like Aluminum as an extended surface was investigated to improve heat transfer in ACC. Found that by using transverse pitch in tubes he pressure drop can be adjusted and use of Metal foams is better than finned surface [7].In ACC the effect of counter flow due to non condensable gases was examined and found that hat capacity to reject heat is proportional to mass flow rate of the air and difference of temperature between the ambient air and finned tubes [8].In one of A frame ACC thermo flow characteristics are compared with normal ACC and found that below 9 m/sec wind speed the heat transfer increases by 21% and above 9 m/sec the vertical arrangement shows better performance. Also found that for low speed winds 15 m height is optimal and for high speed winds 40 m height is best. [9].The effect of inclination in tubes of ACC was investigated to find the effect on pressure drop and flow areas and found that increasing the inclination angle of the tubes can improve the performance of ACC due to improvement in drain of condensed steam assisted by gravity [10].Square array of ACC is presented to overcome the adverse effects of ambient winds and found that both reverse flow and recirculation are weakened using square arrays [11]. Use of Thermo vacuum pump is investigated to maximize the work delivered by the turbine in ACC and found that thermal vacuum pump controls and increases the vacuum which reduces the loading capacity of the turbine in summer [12].The effect of inlet flow distortions by the fans is investigated which gave rise to Actuator disc model which is most used in wind turbines [13].

2. Challenges and Simulation

A 6.5 MW power plant working on Rankine cycle is taken for analysis. The most important feature of this plan is 000that right from the exit of the turbine to the entire It is found that under extreme conditions like very high ambient temperature, Condensation is a problem which affects the working of the entire plant and the plant runs on lower load capacity. To take a step towards achieving solution to it Numerical Simulation is done to show the effect of various parameters like Fan velocity and Ambient temperature. An effort is done towards by taking data from a running unit in one of the installations where the objective is to give a solution to the problem of complete Condensation. A single tube is taken for simulation of Condensation at various ambient conditions and various velocities of Air coming from the Fan. For this Eulerian model is chosen under which Realizable k-epsilon model is chosen for simulation.

The Eulerian model focuses on specific locations of the flow in the flow field. Eulerian simulation model generally employs a fixed mass for the fluid. The field is represented as a function of position x and time t, u = f{x,t}, where u is the flow field. The governing equations used are Navirer stokes equation. Further Realizable k- epsilon model is used to simulate further analysis.

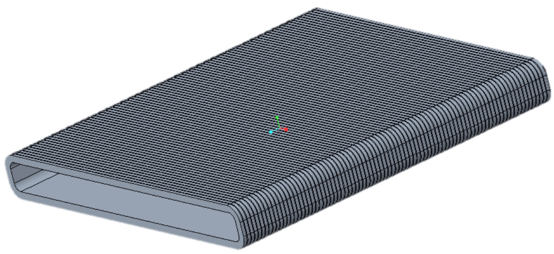

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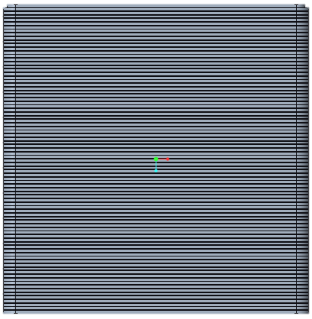
Fig 1- 3 D view of single tube

Table 1-Tube specifications

Tube Dimensions in mm	219 x 19 x 1.5
Tube Length in mm	9600 mm
Tube material	Aluminium

Table 2- Fin specifications

Fin Dimensions in mm	200 x 0.25
Fin Pitch in mm	2.3
Fin material	Aluminium



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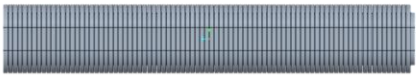

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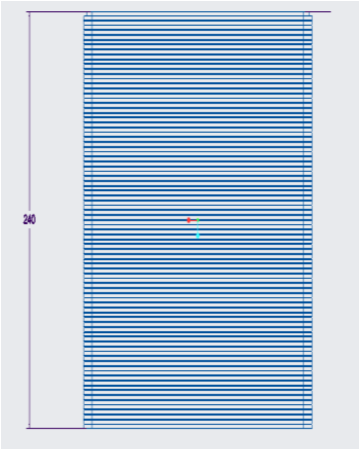

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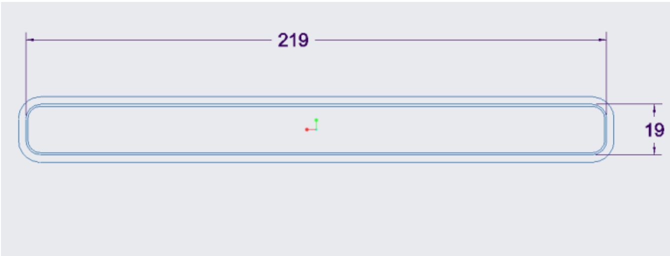

[d]

Fig 2--a-Top view of finned tube , b-left side view of finned tube, c-tube for simulation, d-tube dimensions

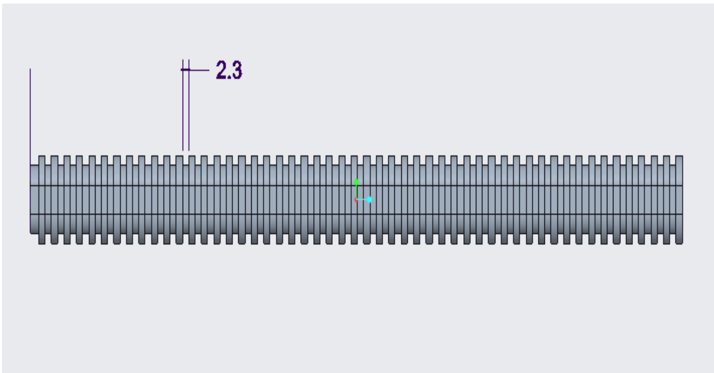

Fig 3- geometry showing dimensions of Fin

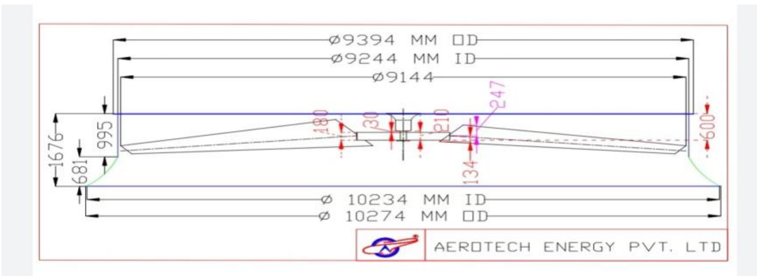

Fig 4 Fan geometry [17]

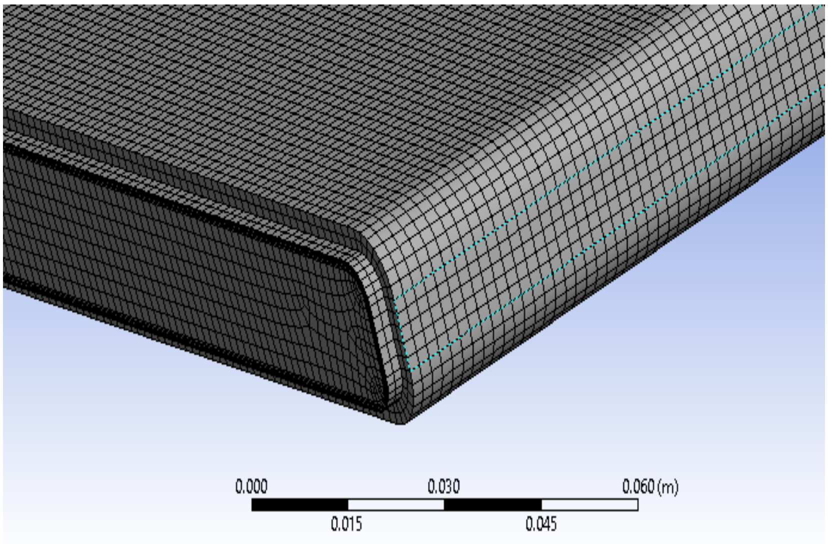

Fig 5- Meshing

GRID INDEPENDENCE TEST

	P1 - Edge Sizing Element Size [m]	P2 - Mesh Nodes	P3 - Mesh Elements	P4 - Velocity-out [m s^-1]	P5 - Pressure-out [Pa]	P6 - Pressure-out-total [Pa]	P7 - Maximum-velocity [m s^-1]
Name	P1	P2	P3	P4	P5	P6	P7
DP 0	0.005	170746	154463	0.000952323	-4.31E-07	2.13E-07	0.000137868
DP 1	0.004	170136	153858	0.00095436	-4.21E-07	2.25E-07	0.000135194
DP 2	0.003	170136	153858	0.00095436	-4.21E-07	2.25E-07	0.000135194
DP 3	0.002	170380	154100	0.000951537	-4.18E-07	2.26E-07	0.000122056
DP 4	0.001	398118	374693	0.000798064	-1.24E-06	-7.38E-07	8.22E-05
DP 5	0.0009	454511	429668	0.000773619	-1.34E-06	-8.61E-07	8.57E-05
DP 6	0.0008	560567	533080	0.000740865	-1.57E-06	-1.12E-06	6.77E-05
DP 7	0.0007	652001	622183	0.000710143	-1.75E-06	-1.33E-06	5.36E-05

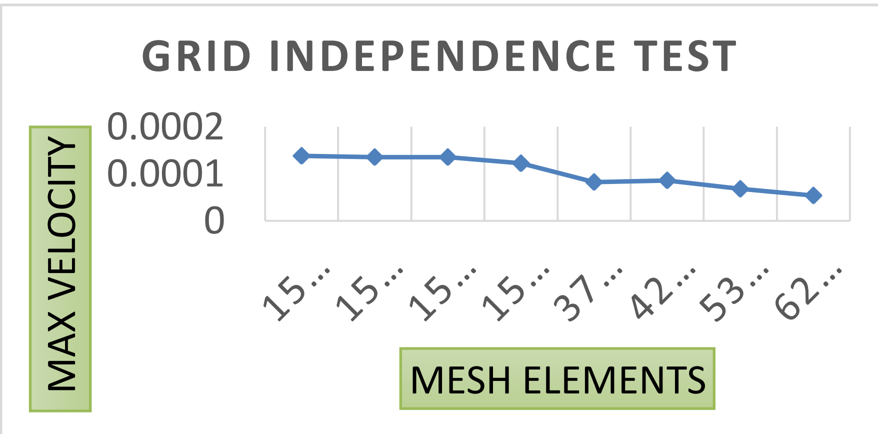

Fig 6-Graph showing Grid independence test

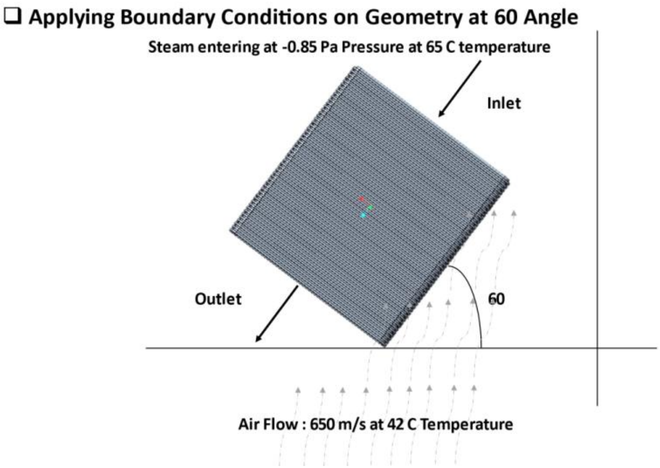

Fig 7- -View showing the inclination of the tube at 6o deg

Table 3 – Heat transfer coefficient values of Air on the outside wall of the tube on Fan side

Ambient temperature in deg K	Velocity of the Air from the Fan in m/sec	Heat transfer coefficient on the outer wall of the finned tube on the fan side in W/ m^2 deg K
315	650	580
315	700	614
315	750	630
316.5	650	574.5
316.5	700	608
316.5	750	639
318	650	571
318	700	602
318	750	648

2.1- Calculations

The value of heat transfer coefficient on the wall of the tube on the fan side is calculated by using the formula,

Nu = [h{a} x La] / k{a}

Re = [ρ{a} x v{a} x D] / μ{a}

Nu = 0.023 $[Re]^{0.8} [Pr]^{0.6}$

The values of thermal conductivity, specific heat, dynamic viscosity, density and Prandtl number for air at a particular temperature can be obtained from the table showing properties of ait.

2.2- Simulation sample

Table 4 -Conditions taken for simulation

Temperature at the inlet of he Condenser tube	338 deg K
Pressure at the inlet of the Condenser tube	-0.833 bar
Mass flow rate of the steam at the tube inlet of Condenser	4.76 kg/dec
Ambient temperature	315 deg K
Heat transfer coefficient on the wall on the air side or fan side	630 W/m^2 deg K
Velocity of air on the fan side	750 m/sec

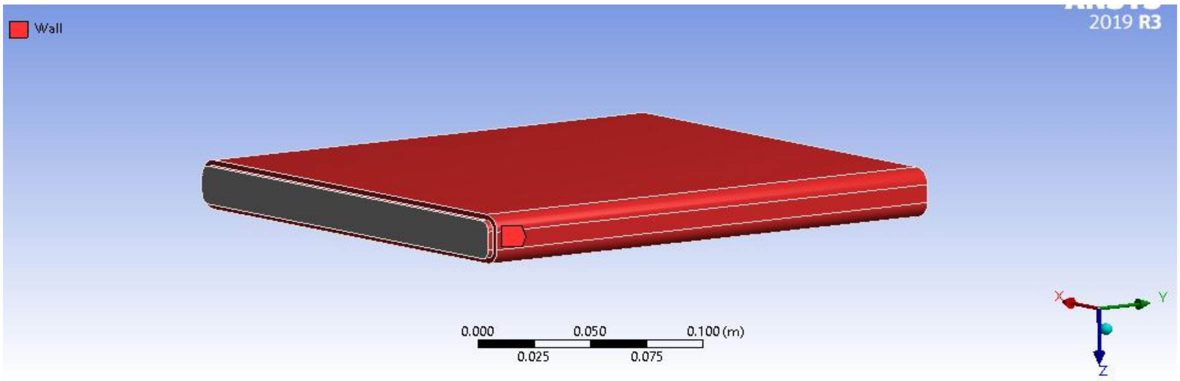

Fig 8-Tube taking boundary conditions

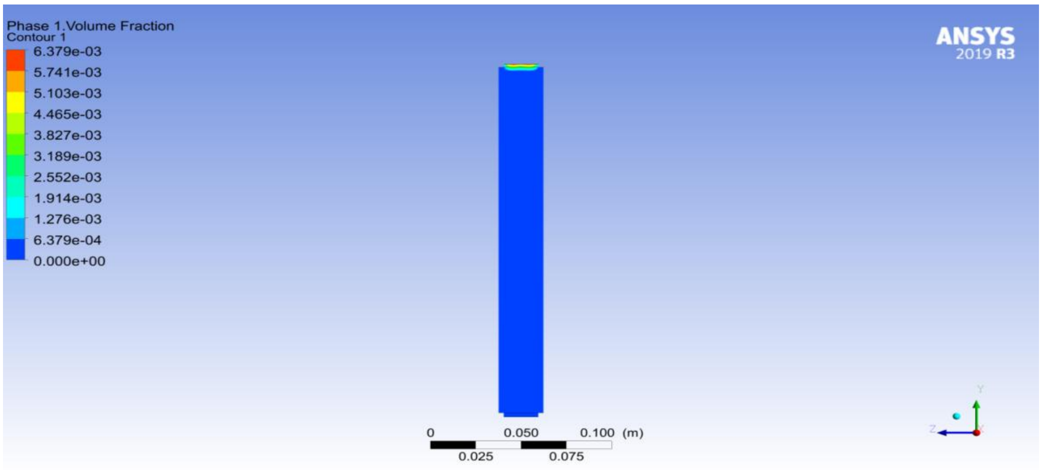

Fig 9--Volume fraction showing condensation

Steam is condensed at the wall at volume fraction of 0.00639912

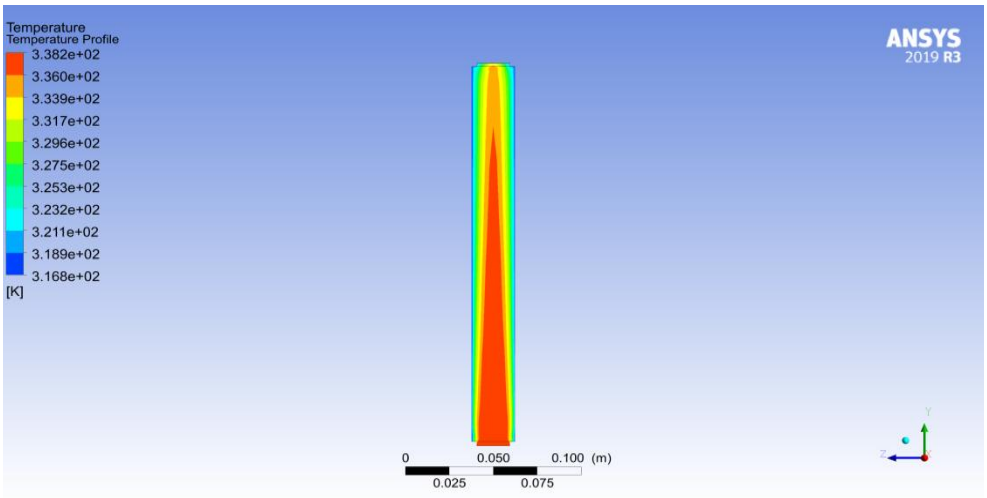

Fig10--Temperature profile during Condensation

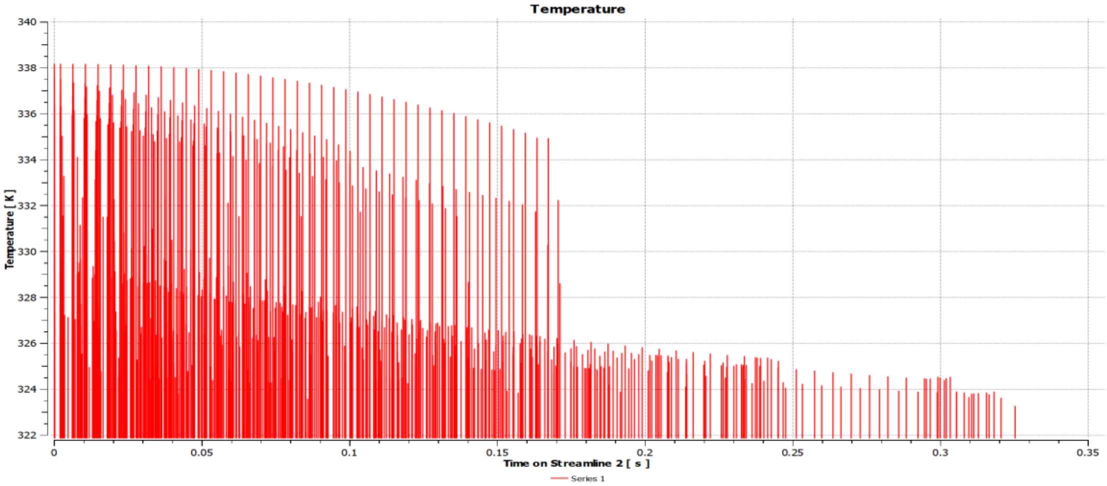

Fig 11-Graph of Temperature profile

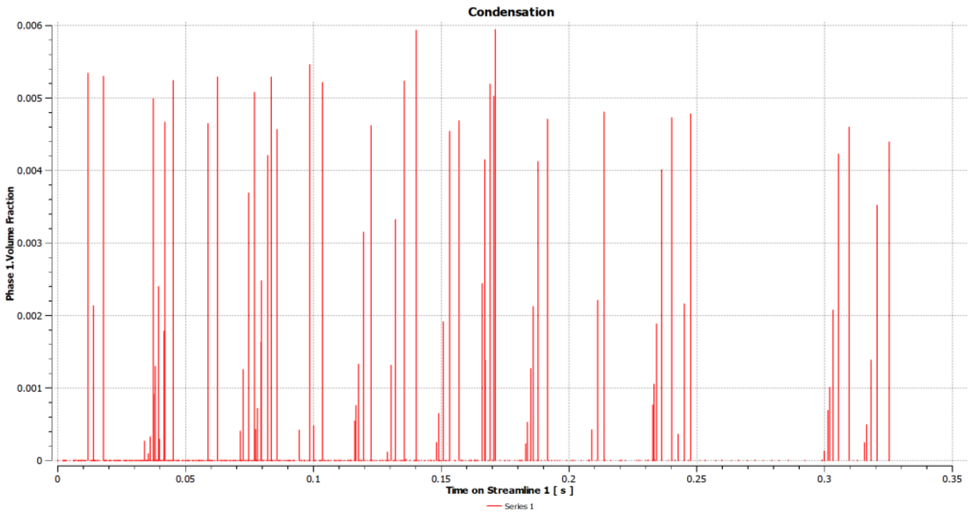

Fig 12-Graph of volume fraction of Condensation

3. Results

Table 5- Effect of change in velocity of air from the fan at a particular ambient temperature

Ambient temperature ,deg K	Velocity of air from the Fan, m/sec	Wall heat transfer coefficient on steam side,W/m^2K	Temperature at which condensation takes place, deg K	Reynolds number	Prandtl number
313	650	780.1	314.2	112.58	1.827
313	700	779.7	314.1	112.57	1.829
313	750	779.5	314	112.55	1.831
315	650	772.5	315.31	112.615	1.774
35	700	772.1	315.3	112.611	1.779
315	750	772	315.29	112.59	1.791
317	650	731.56	316.8	112.632	1.7181
317	700	731.17	316.77	112.631	1.7185
317	750	731	316.75	112.63	.72

Table 6- Effect of change in ambient temperature at a particular velocity of air from the fan

Velocity of air coming from Fan, m/sec	Ambient temperature, deg K	Wall side heat transfer coefficient on steam side, W/m^2deg K	Temperature at which condensation takes place, deg K	Reynolds number	Prandtl number
650	313	780.1	314.2	112.56	1.827
650	315	772.5	315.31	112.61	1.774
650	317	731.56	316.8	112.63	1.7181
700	313	774.7	314.1	112.57	1.829
700	315	772.1	35.3	112.61	1.779
700	317	731.17	316.79	112.62	1.7189
750	313	779.5	314	112.55	1,831
750	315	772	315.29	112.59	1.791
750	317	731.17	316.75	112.63	1.72

4. Conclusion

1. At a particular temperature with increase in velocity of air from the fan the wall heat transfer coefficient is decreasing, the temperature at which condensation takes place is decreasing,- Reynolds number is decreasing but Prandtl number is increasing

2. With increase in ambient temperature at a particular velocity of air- On the steam side the Wall heat transfer coefficient is decreasing, the temperature at which condensation takes place is increasing, Reynolds number is increasing and Prandtl number is decreasing

Appendix

Re = Reynolds number
Pr = Prandtl number
Nu = Nusselt number
h{a} = heat transfer coefficient on the outside of the tube on the fan side = W/[m^2 K]
la = effective length of the tube , m
D = Hydraulic diameter of the fan. m
k{a} = thermal conductivity of air , W/[m K]
ρ {a}= Density of air, kg/m^3
μ{a} = Dynamic viscosity of air . kg/[m sec

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