

Experimental Investigation of Tube Configuration for a Surface Condenser

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ABSTRACT: This paper presents the analyses of the effect of the arrangement of tubes in a tube bundle in a horizontal, two-pass condenser on the amount of heat transferred to the circulating water in the tubes. The tube bundle is assumed to act as a staggered tube bank in cross-flow with downward superheated steam flow. The saturated circulating water is assumed to be turbulent flow. The relationships for heat transfer through tube banks, including condensate inundation, vapor shear, and the effect of tube surface geometry are used in analyzing six tube configurations to determine the largest change in temperature of the circulating water. The heat flux in the system is defined as a function of the condenser and tube material properties, tube geometry, tube spacing, condensate inundation and steam velocity. Numerical modeling of the six tube configurations using a Reynolds-averaged Navier-Stokes (RANS) approach is presented to confirm the analytical results. An attempt is made to analyses of the analytical and numerical results from the six configurations examined provide the optimal tube arrangement for maximum heat transfer to the circulating water. It is found that the circulating water temperature is dependent on the steam-air mixture velocity profile used. The most heat transfer occurs in the tubes rows where the steam-air mixture velocities are the highest. Furthermore, the magnitude of the velocity profile is proportional to the magnitude of the change in circulating water temperature.

KEYWORDS: Tube Configuration, Flow3d, Two Pass, Heat Transfer.

I. INTRODUCTION

The steam cycle that occurs most commonly in thermal power plants can be ideally represented by the Rankine Cycle. The Rankine Cycle is a thermodynamic cycle involving liquid water, steam and a mixture of the two as the working fluids. A condenser is used to condense the expanded steam into liquid and to maintain vacuum on the turbine outlet to increase the available energy across the turbine.

Horizontal, two-pass condensers are commonly used in steam cycles. Two-pass condensers has an inlet/outlet water box with the inlet and outlet of the circulating water on the same side of the condenser, and a return water box on the opposite side of the condenser to direct the circulating water from the first-pass into the second-pass tubes. The circulating water moves through tubes that are arranged together to form a tube bundle. A conventional horizontal, two-pass condenser has the top half of the tubes as the “cold” first-pass and the bottom half of the tubes as the “warmer” second-pass. This tube configuration allows the coldest circulating water to first come into direct contact with the condensing steam. Condensers are typically kept under vacuum conditions to increase the pressure drop across the turbine, thereby maximizing the power output. The vacuum system in the condenser removes non-condensable in the condenser. The non-condensable decrease the heat transfer in the tubes and increase condenser pressure. A heat-transfer device used for condensing steam to water by removal of the latent heat of steam and its subsequent absorption in a heat-receiving fluid, usually water, but on occasion air or a process fluid.

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II. LITERATURE SURVEY

Several papers has been written providing heat transfer, vapor velocity, film condensation and pressure drop correlations over horizontal tube banks based on experimental results and detailed simulations using computational fluid dynamics (CFD) models.

An analysis of a two-pass condenser is performed by M.W.Malin [9] using a CFD model simulating flow and heat transfer. In Malin's work, a single-phase approach for the steam-air mixture flow within the condenser is used to calculate the performance of a condenser with a superheated steam supply. The simulated condenser employs the use of two tube bundles of parallel staggered tubes with the first-pass entering the lower bundle and exiting the condenser through the upper tube bundle.

M.W.Browne et.al.[10] examined variations in experimental observations made in over 70 papers to provide an overview of condensation heat transfer on horizontal tube bundles for downward flowing condensing vapor. The effects of surface geometry, condensate inundation, vapor shear and gravity are studied.

Wilson et.al.[11] provided results for laminar and turbulent flow of air across a single tube row as well as staggered and in-line tube banks. The effects of flow and tube geometry on the Nusselt number, friction factor, velocity and turbulence kinetic energy profiles are presented therein.

Mehrabian [12] evaluated the heat transfer and pressure drop of air over a single, circular tube and over a tube bank based on experimental results. Additionally, a relationship between the velocity distribution of air in cross flow and pressure drop over horizontal tubes is provided.

III. PROBLEM DESCRIPTION

The main objective of this project is to analyze different tube configurations in a tube bundle to determine the best arrangement for the maximum amount of heat transferred to the circulating water in a horizontal, two-pass condenser. The six configurations shown below will be examined.

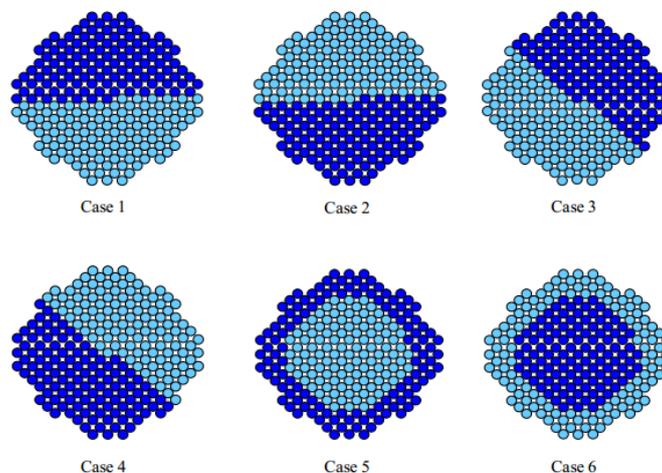


Fig.1 Cross-Sectional View of Tube Configurations

The dark blue and light blue portions of the cross-sectional views in Fig. 1 represent the cold first-pass and warmer second-pass in the tube bundle, respectively.

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IV. METHODOLOGY AND ASSUMPTIONS

The heat transferred from turbulent superheated steam-air mixture in downward cross flow over horizontal tubes to turbulent flow within the tubes is predicted using RANS equations. The mathematical model presented by M.R.Malin [9] is adapted for each of the six tube configurations analyzed in this study.

To determine the maximum outlet temperature of the circulating water, the following assumptions are made to simplify the problem in order to solve it using the mathematical model.

1. A symmetrical condenser with a horizontal, two-pass tube bundle is examined, where the steam inlet opening to the condenser is in the fore/aft and athwart ship center of the condenser.
2. The algorithm evaluates the condenser in two dimensions, assuming that the heat transfer is constant along the length of the tube bundle, but varies through and around the circumference of the tube bundle.
3. The condenser dimensions and material are constant as well as the steam inlet velocity for all six cases.
4. For the six cases that are analyzed in the study, the inlet temperature of the circulating water is kept constant, along with the individual tube size, spacing and number of tubes, thereby keeping the overall tube bundle size constant.
5. The circulating water is turbulent flow and complete mixing of the water is assumed at the end of the first-pass to ensure a uniform circulating water inlet temperature for the second-pass tubes.
6. The circulating water temperature does not become a function of the number of tubes in each pass, since the number of first and second-pass tubes is kept as equal as possible.
7. In the FLOW3D models, the steam-air mixture flow is assumed laminar in order to accelerate the simulations so that a comparison to the Mehrabian velocity approach may be made.

V. INITIAL CONDITIONS

A. Condenser Dimensions

A two-pass condenser with a rectangular upper shell is used in the analyses. The upper shell is 3.048 meters long, 0.6096 meters high and 0.6096 meters wide. The tube bundle is approximately 0.3048 meters in diameter and is centred in the upper shell of the condenser. The steam inlet opening is centered on the top of the condenser and is 0.3048 meters wide and 0.6096 meters long. The opening between the upper shell and the hotwell that collects the condensate formed around the tubes is 0.1524 meters wide and runs the length of the condenser. The condenser is assumed to be made of alloy 316L stainless steel.

B. Number of Tubes

There are 217 titanium tubes in the tube bundle, each with a 0.0143-meter outer diameter. The tubes are 20 Birmingham Wire Gage (BWG), resulting in an inner diameter of 0.0125-meters. Two alternating rows of tubes are used to create the bundle. One row, row "A," is centred with a tube on the vertical centreline of the tube bundle. The alternating row, row "B," is offset by half of one tube diameter from the tube bundle vertical centreline. The tube pitch, or centreline-to-centreline distance between tubes, Fig. 2, is the same in the vertical and horizontal direction.

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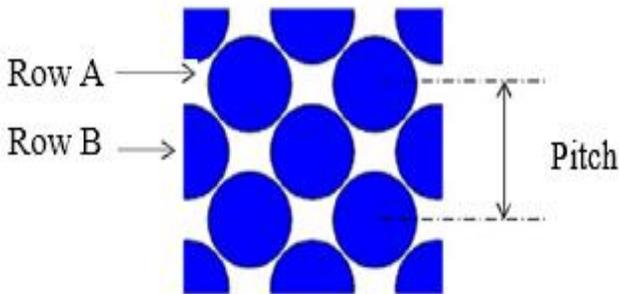


Fig. 2 Tube Row Orientation

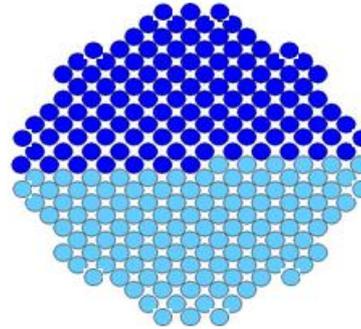


Fig. 3 Case 1 Tube Configuration

CASE-1

The Case 1 tube bundle is split horizontally, with the first-pass tubes comprised of the top 12 rows of tubes and half of the tubes in row 13. The second-pass is comprised of half of the tubes in row 13 and all of the tubes in the remaining bottom 12 rows. The tube configuration is shown below in Fig. 3.

This configuration is chosen to demonstrate the effect on the circulating water temperature of a horizontal tube configuration with the first-pass on the top half of the tube bundle. The number of tubes in the first-pass totals 109 and the number of tubes in the second-pass totals 108.

A. Algorithm Results Using Mehrabian-Based Velocities

The results from Case 1 show that for both the first and second-passes, the highest heat transfer occurs in the lower the tube rows. This results in a higher temperature for the circulating water. This correlation is seen in Fig.4 and Fig.5, and presented quantitatively in Table I and Table II. The topmost row in each pass, row 1 and row 13, has the coolest circulating water temperatures in the first and second pass, respectively.

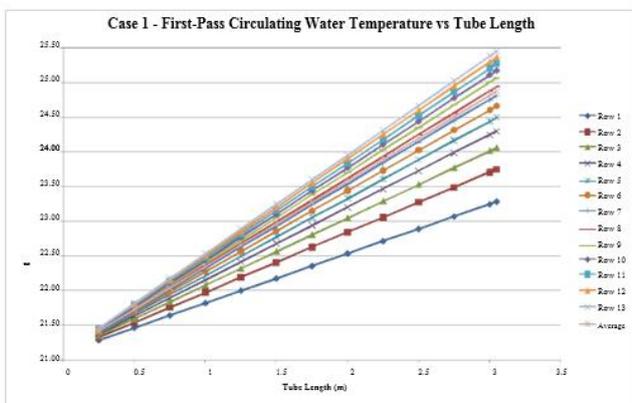


Fig. 4 Case 1 First-Pass Circulating Water Temperature vs Tube Length

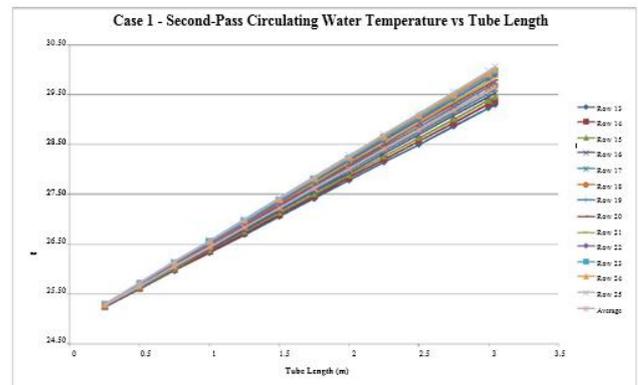


Fig.5 Case 1 Second-Pass Circulating Water Temperature vs Tube Length

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Table. I Case 1 first and second-pass heat flux and outlet circulating water temperature

First-Pass			Second-Pass		
Row	q" (W/m ² K)	T _{l,out} (°C)	Row	q" (W/m ² K)	T _{l,out} (°C)
1	1.274E+04	23.284	13	2.597E+04	29.302
2	1.547E+04	23.749	14	2.646E+04	29.385
3	1.729E+04	24.059	15	2.692E+04	29.463
4	1.870E+04	24.299	16	2.735E+04	29.537
5	1.985E+04	24.496	17	2.776E+04	29.607
6	2.083E+04	24.664	18	2.815E+04	29.674
7	2.169E+04	24.810	19	2.852E+04	29.737
8	2.246E+04	24.941	20	2.888E+04	29.798
9	2.316E+04	25.061	21	2.922E+04	29.857
10	2.380E+04	25.170	22	2.955E+04	29.913
11	2.439E+04	25.271	23	2.986E+04	29.966
12	2.494E+04	25.364	24	3.016E+04	30.016
13	2.546E+04	25.452	25	3.045E+04	30.066

Table. II Case 1 Calculated Temperatures

Case 1 Temperatures (°C)	
Inlet	21.11
First-Pass Outlet	24.872
Second-Pass Outlet	29.657
T Between Passes	4.786
Overall ΔT	8.547

Table.III Case 1 Energy Balance

	E _{out} (kJ/s)	E _{in} (kJ/s)	ΔE (kJ/s)
Circulating Water (m c _p ΔT)	242.12	236.09	6.02
Air (Y _a H _a m _a)	0.03	0.02	0.01
Steam (Y _s H _s m _s)	0.00	1909.64	-1909.64
Tube Wall (kΔT)	6591.90	6809.32	-217.42
Condensation (m ³ LA)	1907.25		1907.25
Total	8741.30	8955.06	-213.76
Percent Change in Energy			-2.45%

The larger change in circulating water temperature in the second-pass can be attributed to the velocity profile. The cumulative mass condensation rate is larger as more condensate continues to form moving from the top of the tube bundle to the bottom. This could lead the higher heat flux rates, and consequently, the higher circulating water temperatures.

B. FLOW3D Results

The values used to create the condenser tube configuration, initial conditions and operating parameters for Case 1 are used in a FLOW3D model to provide velocity profiles, from which new values for the circulating water temperature are calculated. The resulting velocity profile is shown in the Fig.6 velocity magnitude contours.

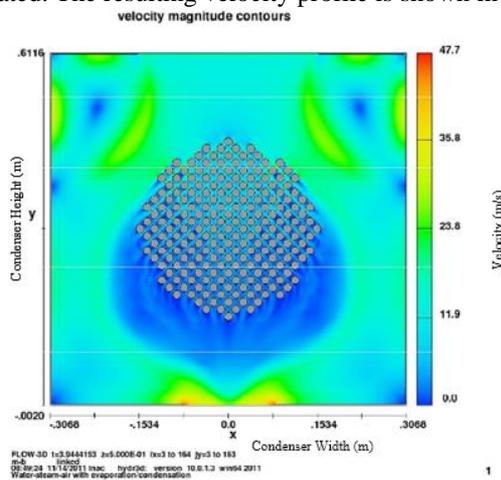


Fig.6 Case 1 FLOW3D Velocity Magnitude Contours

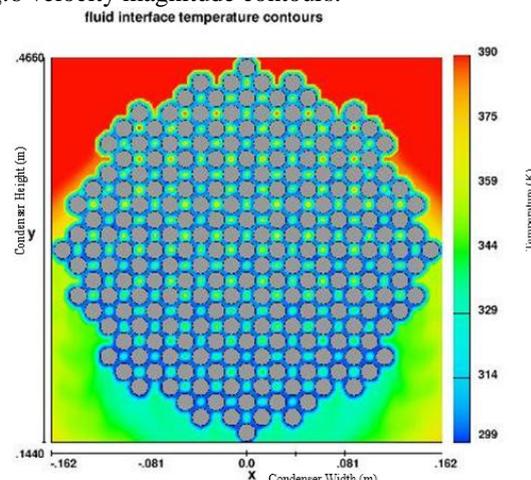


Fig.7 Case 1 FLOW3D Mixture Temperature Contours

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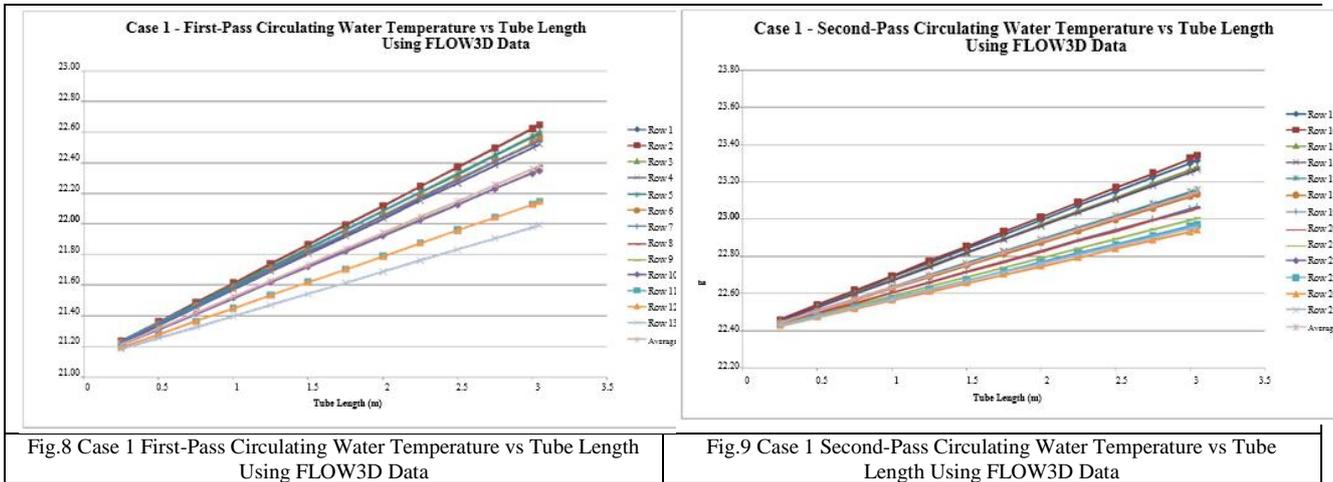


Fig.8 Case 1 First-Pass Circulating Water Temperature vs Tube Length Using FLOW3D Data

Fig.9 Case 1 Second-Pass Circulating Water Temperature vs Tube Length Using FLOW3D Data

Table. IV Case 1 first and second-pass heat flux and outlet circulating water temperature using flow3d data

First-Pass			Second-Pass		
Row	q" (W/m ² K)	T _{l,out} (°C)	Row	q" (W/m ² K)	T _{l,out} (°C)
1	8.988E+03	22.644	13	5.497E+03	23.317
2	9.016E+03	22.649	14	5.645E+03	23.342
3	8.740E+03	22.601	15	5.277E+03	23.280
4	8.269E+03	22.521	16	5.204E+03	23.267
5	8.677E+03	22.591	17	4.564E+03	23.158
6	8.471E+03	22.556	18	4.405E+03	23.131
7	8.399E+03	22.543	19	4.036E+03	23.068
8	7.424E+03	22.377	20	3.975E+03	23.058
9	7.311E+03	22.358	21	3.672E+03	23.006
10	7.260E+03	22.349	22	3.400E+03	22.960
11	6.061E+03	22.145	23	3.458E+03	22.969
12	6.072E+03	22.147	24	3.297E+03	22.942
13	5.182E+03	21.995	25	3.403E+03	22.960

Table. V Case.1 Comparison of Mehrabian And Flow3d Calculated Temperatures

	Temperature (°C)	
	Mehrabian	FLOW3D
First-Pass Outlet	24.872	22.380
Second-Pass Outlet	29.657	23.147
T Between Passes	4.786	0.767
Overall ΔT	8.547	2.037

The significantly lower velocities obtained from FLOW3D result in very little heat transfer to the circulating water compared to Mehrabian. This shows a proportional relationship between the magnitude of the steam-air mixture velocity and the circulating water temperature.

Same mathematical (Mehrabian) and numerical (Flow 3D) models used for getting the results for other 5 tube configurations. The results of the same are concluded in conclusion section.

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VI. CONCLUSIONS

Six unique tube configurations in a horizontal, two-pass condenser are analyzed in an iterative heat and mass transfer algorithm to determine the outlet circulating water temperature through the tubes. The algorithm considers the heat transferred from the steam-air mixture to the interface between the mixture and condensate, through the condensate, through the tube wall and into the circulating water. The algorithm also takes into account the latent heat produced by the condensate forming around the tubes. A steam-air mixture velocity profile is established using the approach taken by Mehrabian. The outlet circulating water temperature for each first-pass row is calculated and using a weighted average, a new inlet circulating water temperature is created for the second-pass tubes, from which an exit circulating water temperature is calculated.

Table.VI Steam-Air Mixture Velocity Profiles

Steam-Air Mixture Velocity Profiles (m/s)								Steam-Air Mixture Velocity Profiles (m/s)							
	Mehrabian	FLOW3D							Mehrabian	FLOW3D					
Row	Cases 1-6	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Row	Cases 1-6	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
1	48.20	26.74	26.74	26.77	26.72	26.82	26.68	14	180.34	10.98	11.05	10.96	10.96	10.94	10.87
2	68.16	26.93	26.80	26.79	26.81	26.76	26.85	15	186.67	9.79	9.90	9.72	9.74	9.75	9.62
3	83.48	25.58	25.84	25.84	25.84	25.79	25.87	16	192.80	9.56	9.64	9.53	9.51	9.34	9.27
4	96.40	23.26	23.57	23.58	23.57	23.52	23.59	17	198.73	7.64	7.80	7.60	7.61	8.23	8.23
5	107.78	25.33	30.17	30.14	30.12	30.18	30.24	18	204.49	7.19	7.93	7.82	7.80	8.35	8.34
6	118.06	24.33	24.37	24.31	24.32	24.36	24.31	19	210.09	6.19	6.77	6.54	6.55	6.87	6.81
7	127.52	24.01	24.04	24.01	24.01	24.07	24.10	20	215.55	6.04	6.16	6.01	6.03	6.63	6.27
8	136.33	19.41	19.62	19.61	19.64	19.64	19.72	21	220.87	5.28	5.85	5.53	5.49	5.97	5.93
9	144.60	18.92	17.43	17.34	17.38	17.33	17.08	22	226.07	4.63	5.37	5.16	5.14	5.73	5.37
10	152.42	18.71	18.76	18.82	18.75	18.87	18.84	23	231.15	4.77	5.12	4.72	4.67	5.18	5.12
11	159.86	13.71	13.93	13.98	13.91	14.03	13.98	24	236.13	4.39	4.72	4.37	4.33	5.05	4.65
12	166.97	13.77	13.75	13.78	13.70	14.00	13.94	25	240.99	4.64	5.08	4.49	4.48	5.18	4.90
13	173.78	10.49	10.57	10.46	10.48	10.72	10.69								

The results using the assumed Mehrabian-based steam-air mixture velocity profile in the heat and mass transfer algorithm show that all six cases has higher outlet circulating water temperatures for the second-pass tubes than for the first-pass tubes. More heat is transferred to the first-pass tubes when these tubes are located on the bottom of the tube bundle, such as in Cases 2 and 4. Case 2 resulted in the most heat transferred to the first-pass tubes, resulting in the warmest circulating water at the outlet of the first-pass, with a temperature of 25.996°C. Case 1 resulted in the largest change in circulating water temperature from the first-pass to the second-pass with a change of 4.786°C. Overall, Case 3 resulted in the most heat transferred to the circulating water, with an average second-pass outlet temperature of 29.758°C.

The Mehrabian steam-air mixture velocity profile that is used in the algorithm could be more accurate and closer to the actual velocity in the condenser, similar to the velocity profile seen in the FLOW3D simulations, by using a different correction factor. The Mehrabian [12] approach directly increases the pressure drop, and consequently, the row velocity with each successive row, which significantly increases the velocity at higher rows. The steam-air mixture velocity profiles obtained from FLOW3D simulations of the six cases decreases as the steam-air mixture moves downward through the tube bundle, due to the tubes obstructing the mixture flow path. This velocity profile is opposite from the assumed profile based on Mehrabian. The FLOW3D velocity profiles obtained for each of the six cases are relatively similar and exhibit symmetry. Comparable to the results using the Mehrabian-based velocity profile, the results using FLOW3D data shows an increase in circulating water temperature in both the first and second-passes. Case 1 has the hottest circulating water temperature of 22.380°C at the outlet of the first-pass tubes. Case 2 has the largest change in temperature between the first and second-passes of 1.322°C. Case 5 has the hottest second-pass circulating water temperature of 23.216°C.

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In comparing results calculated from the Mehrabian and FLOW3D steam-air mixture velocity profiles in the algorithm, the heat flux and circulating water temperature are found to be proportional to the velocity. As the velocity increases, the heat flux and circulating water temperature increases, consistent with thermodynamic principles. The first-pass tubes that experience the highest velocity, which are the lowest tube rows with a Mehrabian-based velocity profile (Cases 2 and 4) and the highest tube rows with a FLOW3D-based velocity profile (Cases 1 and 3), has the most heat transfer to the tubes. The tubes where the highest velocities result in the highest outlet circulating water temperature. Therefore, the velocity is proportional to the circulating water temperature.

The FLOW3D models may be refined to more accurately compare the results of the algorithm with those employing the Mehrabian approach to the steam-air mixture velocity. The FLOW3D grid that is generated is relatively coarse and the flow is assumed laminar in order to expedite simulating all six cases. A higher grid resolution and assuming a turbulent steam-air mixture flow through the bundles would each increase the predicted maximum velocity through the tubes. A grid sensitivity and/or closure model sensitivity analysis could be performed to further validate the results obtained herein.

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