CFD Analysis And Performance Of Parallel And Counter Flow In Concentric Tube Heat Exchangers

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Abstract

A Heat Exchanger is a device built for the efficient heat transfer from one fluid to another, whether the fluids are separated by a solid wall so that they never mix, or the fluids are directly in contact. Every year research in Heat exchanger technology is growing to develop efficient, compact and economical heat exchangers, all over the world. Updating the community for this development needs an interaction. In last five years Concentric tube heat exchangers utilize forced convection to lower the temperature of a working fluid while raising the temperature of the cooling medium. The purpose of this paper is to use ANSYS FLUENT12.1 software and hand calculations to analyze the temperature drops as a function of both inlet velocity and inlet temperature and how each varies with the other. Each heat exchanger model was built in steps and analyzed along the way until both parallel flow and counter flow heat exchanger models were developed. The results were compared between each model and between parallel and counter flow with fouled piping. Turbulent flow was also analyzed during the development of the heat exchangers to determine its effect on heat transfer. While as expected the fouled heat exchanger had a lower performance and therefore cooled the working fluid less, the performance of the counter heat exchanger unexpectedly of the parallel heat exchanger.

1 INTRODUCTION

1.1 PROJECT BACKGROUND

The heat exchanger is a device which transferred the heat from hot medium to cold medium without mixed both of medium since both mediums are separated with a solid wall generally. There are many types of heat exchanger that used based on the application. For example, double pipe heat exchanger is used in chemical process like condensing the vapor to the liquid. When to construct this type of heat exchanger, the size of material that want to uses must be considered since it affected the overall heat transfer coefficient. For this type of heat exchanger, the outlet temperature for both hot and cold fluids that produced is estimated by using the best design of this type of heat exchanger.

1.2 LITERATURE SURVEY

The development of fluid flow and temperature profiles of a fluid after undergoing a sudden change in wall temperature is dependent on the fluid properties as well as the temperature of the wall. This thermal entrance problem is well known as the Graetz Problem. From reference [1] for incompressible Newtonian fluid flow with constant ρ and k,

The velocity profile can also be developing and can be used for any Prandtl number material assuming the velocity and temperature profiles are starting at the same point [2]. For the original Graetz problem, Poiseuille flow was assumed and equation was used to describe the fully developed velocity field of the fluid flowing through the constant wall temperature tubing.

Analyzing the paper from Sellars [3] where he extends the Graetz problem, this equation for velocity is also used. For the purposes of this paper and the use of the finite element program, a constant value for the inlet velocity was used. This means a modified Graetz problem was introduced and analyzed.

In the cases studied, engine oil was assumed to be flowing through the inner pipe which was made of copper and cooled by the outer concentric pipe in which water was flowing. Material properties such as dynamic viscosity, density, Prandtl number, and thermal conductivity were obtained from reference [4].

Graetz found a solution in the form of an infinite series in which the eigenvalues and functions satisfied the Sturm-Louiville system. While Graetz himself only determined the first two terms, Sellars, Tribus, and Klein [5] were able to extend the problem and determine the first ten eigenvalues in 1956. Even though this further developed the original solution, at the entrance of the tubing the series solution had extremely poor convergence where up to 121 terms would Mnot make the series converge.

Schmidt and Zeldin[6] in 1970 extended the Graetz problem to include axial heat conduction and found that for very high Peclet numbers (Reynolds number multiplied by the Prandtl number) the problem solution is essentially the original Graetz problem.

Hwang et al [7] measured pressure drop and heat transfer coefficient in fully developed laminar pipe flow using constant heat flux conditions. Based on the experimental results they showed that the experimental friction factor was in good agreement with the theoretical predictions using the Darcy equation.

Bianco et al [8] observed only amaximum of 11% difference between single and two phase results for the laminar regime.

Akbari et al [9] for the first time compared three different two phase models and the single phase model in the laminar regime. Single and two phase models were found to be predicting identical hydrodynamic fields but very different thermal ones.

The expression defining the velocity distribution in a pipe flow across turbulent flow is derived and demonstrated in **Bejan, "Convective heat transfer coefficient",1994** . Hydro dynamically developed flow is achieved in a pipe after a certain length i.e. entrance length *Le* , where the effect of viscosity reaches the centre of pipe. At this point the velocity assumes some average profile across the pipe which is no longer influenced by any edge effects arising from the entrance region. The flow of real fluids exhibit viscous effects in pipe flow. Here this effect is identified for turbulent flow conditions.

A closer look at all the experimental and numerical works reveals that most of the forced convective heat transfer studies in pipe flow have been done with constant wall flux boundary condition. So in this work, a systematic computational fluid dynamic investigation with constant wall temperature Boundary condition has been carried out adopting the single phase approach in the turbulent regime and the results are compared with the analytical and numerical results available in the literature.

2 PROBLEM STATEMENT

The double pipe heat exchanger is used in industry such as condenser for Chemical process and cooling fluid process. This double pipe heat exchanger is designed in a large size for large application in industry. To make this small double pipe heat exchanger type become practicality, the best design for this small double pipe heat exchanger is choose.

Heat transfer is considered as transfer of thermal energy from physical body to another. Heat transfer is the most important parameter to be measured as the performance and efficiency of the concentric tube heat exchanger. By using CFD simulation software, it can reduces the time and operation cost compared by Analytical calculations in order to measure the optimum parameter and the behaviour of this type of heat exchanger.

2.1 OBJECTIVES OF RESEARCH:

The objective of the present study is to provide more complete understanding Flow maldistribution in tubular heat exchanger by studying area weighted and mass weighted temperature profiles for maldistribution without back flow and maldistribution with back flow. And comparison of average temperature profiles of flow maldistribution with the average temperature profiles of uniform mass flow distribution.

This numerical investigation was carried out for the concentric tube arrangement with different diameter of tubes. A finite volume numerical scheme is used to predict the conjugate heat transfer and fluid flow characteristics with the aid of the computational fluid dynamics (CFD) commercial code, FLUENT. The governing equations for the energy and momentum conservation were solved numerically with the assumption of three-dimensional steady flow. An effective model, the standard based k-ε turbulence model was applied in this investigation.

As described in the section 1.2, the available relevant literature is quite limited With respect to the analytical and it is still difficult to predict the physics of the flow maldistribution within the circular tube banks. Therefore, temperature distributions within the bundle were studied numerically. The objective of this study is to develop a CFD simulation to predict heat Transfer in concentric tube heat exchanger by using different fluids.

2.2 SCOPES OF RESEARCH:

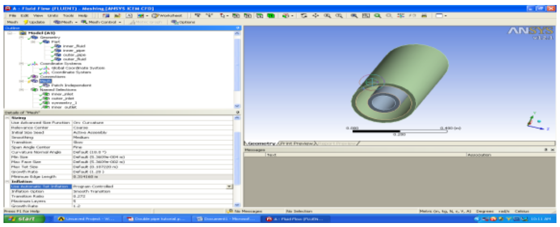
The scopes of this research are as follows:

i. Study on heat transfer for heat exchanger specific to double pipe heat exchanger types.
ii. Design the double pipe heat exchanger by using ANSYS WORKBENCH.
iii. Simulation in double pipe heat exchanger by using FLUENT software.
iv. Analysis the heat exchanger specific to flow rate of hot and cold fluid.
v. To simulate heat transfer in concentric tube heat exchanger by using CFD-Fluent software.
vi. To analyze the heat transfer in concentric tube heat exchanger by comparing the simulation result to the Analytical calculations. Validate simulation results to the Analytical calculations within 5% error.

2.3 Problem description and modelling:

Geometry Modelling:

The geometry made in ANSYS workbench. This geometry imported to ansys fluent and repairs the geometry. The geometry consists of a length of 1m. Concentric tube of inner tube inner diameter 0.1m and outer tube inner diameter 0.24m.



For this project, fully developed laminar and turbulent incompressible fluid flow will be analyzed in three heat exchanger cases: parallel flow, counter flow, and flow in a fouled heat exchanger. The resulting temperature difference will be compared and be determined as a function of the inlet velocity and inlet temperatures. The overall objective is to determine the max temperature difference in these cases for both laminar and turbulent flow for a variety of flow rates and inlet temperatures. To simplify the number of variables, water and oil will be chosen as the fluids to maintain viscosities and densities of the fluids constant. The type of heat exchanger used will be of concentric tube design. Water will be the cooling medium and oil the working fluid.

2.4 Defining Material Properties:

Water was used as the base fluid flowing through tubing or piping. Its material properties were derived from tables based on the temperature which was being calculated in the model. The material was defined in FLUENT using its material browser. For the different flow arrangement problem model certain properties were defined by the user prior to computing the model, these properties were: thermal conductivity, density, heat capacity at constant pressure, ratio of specific heats, and dynamic viscosity. For the modified Graetz problem with pipe wall conduction as well as for the heat exchanger models the material library properties in FLUENT were used.

Different fluids properties	Density (ρ) kg/m^3	P_r	Thermal conductivity(K) W/mk	Specific heat C_P j/kgK	Dynamic viscosity (μ) kg/m-s
Transformer oil	826	159	0.134	2328	0.0091
Toluene	866	6.5	0.151	1675	0.0005
Benzene	875	6.51	0.159	1759	0.00058
Gas oil	830	50.4	0.135	2050	0.00332
Ethylene glycol	1111.4	150.4	0.252	2415	0.0157
Glycerin	1259.9	6780.3	0.286	2427	0.799
Water	998.2	6.99	0.6	4182	0.001003

Different material properties	Density (ρ) kg/m^3	Thermal conductivity(K) W/mk	Specific heat C_P j/kgK
Copper	8978	387.6	381
Aluminum	2719	203.2	871

3 SIMULATION

3.1 FINITE VOLUME METHOD:

The mass, momentum, and scalar transport equations are integrated over all the fluid elements in a computational domain using CFD. The finite volume method is a particular finite differencing numerical technique, and is the most common method for calculating flow in CFD codes. This section describes the basic procedures involved in finite volume calculations.

The finite volume method involves first creating a system of algebraic equations through the process of discretising the governing equations for mass, momentum, and scalar transport. To account for flow fluctuations due to turbulence in this project, the RANS equations are discretised instead when the cases are run using the *k-epsilon* turbulence model. When the equations have been discretised using the appropriate differencing scheme for expressing the differential expressions in the integral equation (i.e. central, upwind, hybrid, or power-law, or other higher-order differencing schemes), the resulting algebraic equations are solved at each node of each cell.

3.2 Numerical procedure and computational methodology:

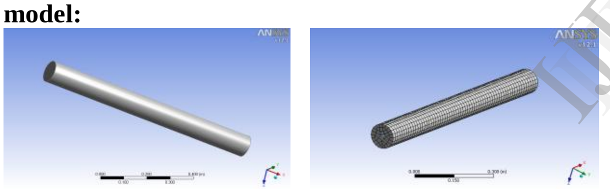
The governing differential transport equations were converted to algebraic equations before being solved numerically. After the specification of the boundary condition, the solution control and the initialization of the solution have to be given before the iteration starts. The solution controls like the pressure velocity coupling and the discrimination of the different variables and the relaxation factors have to be specified. The solutions sequential algorithm (called the segregated solver) used in the numerical computation requires less memory that the coupled solver. Since we are using the segregated solver for our problem, the default under relaxation factors are used and the SIMPLE scheme for the pressure velocity coupling is used and the second discrimination is used for the momentum and the standard scheme is used for the pressure.

4 RESULT AND DISCUSSIONS

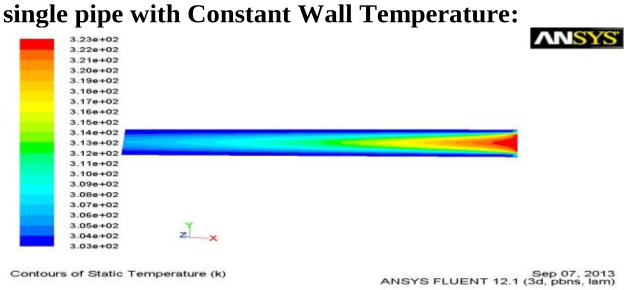
The observations of CFD simulation results are discussed below for the given boundary conditions. To determine the best design for double pipe heat exchanger type.

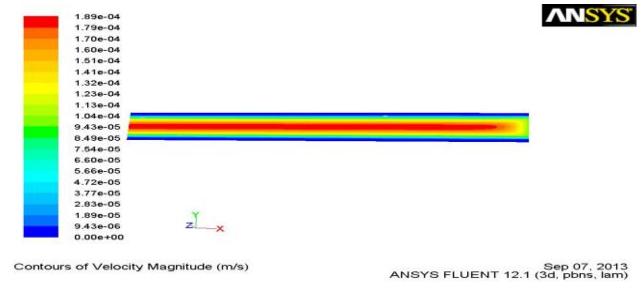
i. Results will be analysed using ANSYS FLUENT.
ii. Hand calculations will be done to use as a cross check
iii. ANSYS FLUENT is a useful tool for modelling laminar flow heat transfer.

The modified gratez problem ANSYS workbench model:

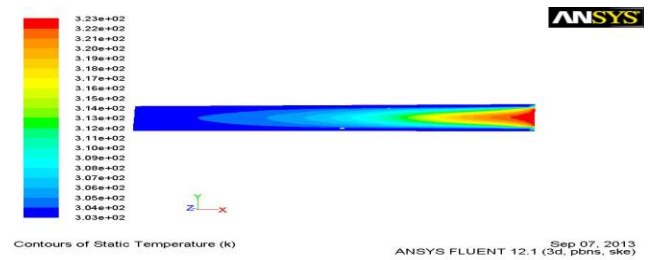


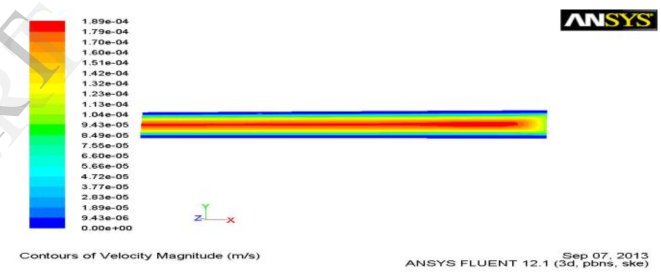
4.1 The Modified Graetz Problem Results:
Laminar Flow temperature and velocity profile of single pipe with Constant Wall Temperature:





Turbulence Flow temperature and velocity profile of single pipe with Constant Wall Temperature:





Graetz problem calculations:

At velocity **V=0.0001m/s**

Reynolds number

$R_e = \rho vD/ \mu$

$= (988x0.0001x0.1) / (5.47x10^{-4}) =$ **18.062**

Now $P_r = Cp\ \mu/K$

$= (4181X5.47X10^{-4})/0.64 =$ **3.57**

Dimensionless length value

$L^* = L/DR_eP_r$

$= 1/ (0.1x18.062x3.57) =$ **0.1556**

Nusselt number

$Nu_m = 3.66 + \{(0.075/ L^*)/ (1+ (0.05/ (L^*) ^{\wedge} (2/3)\}$

$=$ **4.072**

$Nu_m = (-1/4\ L^*)\ x\ (\ln (T_m^*(L)$

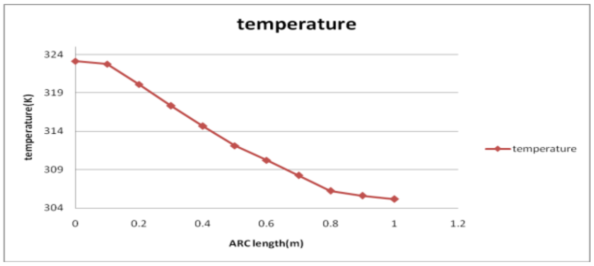
$T_m^*(L) = e^{-4\ L^*\ Num}$ =**0.0793**

$T_m\ (L) = T_w-(Tw-To)\ X\ T_m^*(L)$

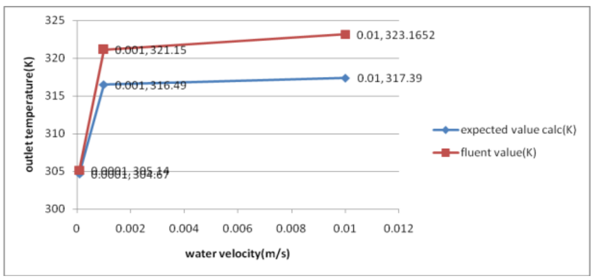
$= 30-(30-50)\ x0.0793$

=**31.6°c =304.67 k**

Modified Graetz Problem Centreline Temperature for single pipe:



Different velocity vs. out let temperature of single pi[e:

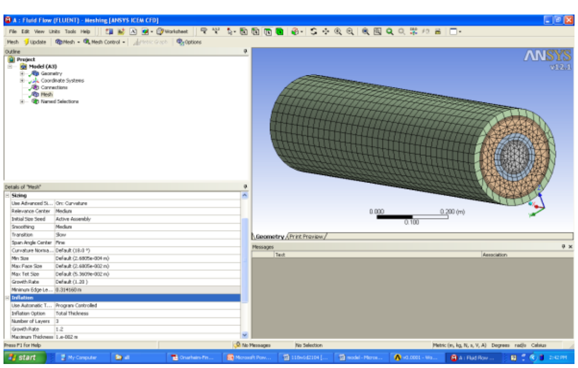


inlet Velocity (m/s)	Inlet temp. (K)	Wall temp. (K)	Expected value by calculation (K)	FLUENT VALUE (K)	% Error
0.0001	50	30	304.67	305.142	0.1572
0.001	50	30	316.49	321.151	1.4510
0.01	50	30	317.39	323.165	1.7836

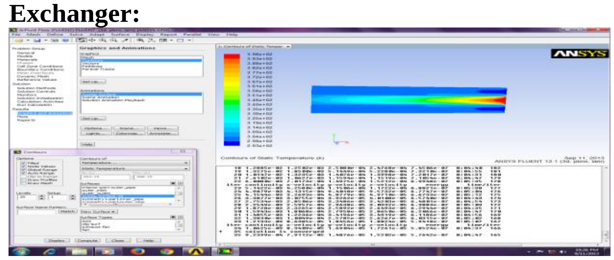
The Modified Graetz Problem Calculations: Comparison of fluent value vs. expected value:

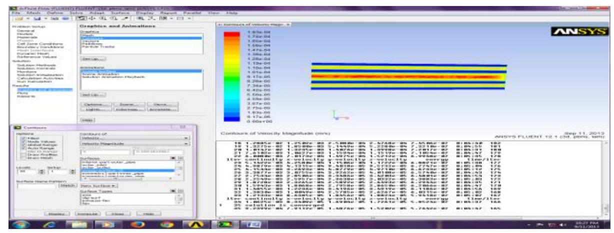
4.2 Concentric Tube parallel Flow Heat Exchanger: Concentric tube Heat Exchanger ansys12.1 workbench Model:

ANSYS12.1 workbench Mesh model:

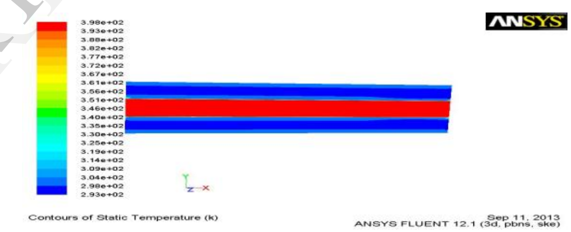


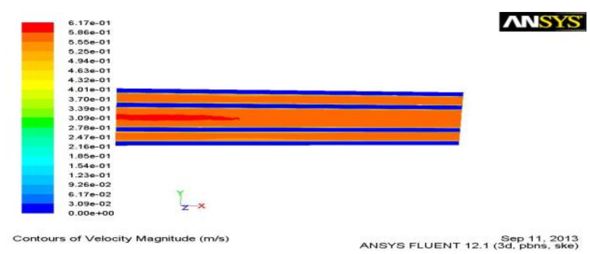
Laminar Flow in a parallel Heat Exchanger: Temperature Velocity Profile for parallel Heat Exchanger:





Turbulent Flow in a parallel Heat Exchanger: Temperature and Velocity Profile for a turbulent parallel Flow Heat Exchanger:





Laminar Flow in parallel Heat Exchanger Problem Calculations:

At velocity **V=0.0001m/s**

$D_O = 0.14m$ $D_i = 0.1m$

$A_O = \mathbf{0.4398\ m^2}$ $A_i = \mathbf{0.3142\ m^2}$

Cross sectional area of each fluid flow

$A_{oil} = \pi r^2 = \mathbf{0.00785 m^2}$

$A_{water} = \pi (r_o^2 - r_i^2) = \mathbf{0.02985\ m^2}$

Mass flow rate

$M_{oil=}$ ρAv= (826). (0.007854).(0.0001)=**0.0006487 kg/s**
$M_{water=}$ ρAv= (998.2). (0.02984).(0.0001)=**0.00297 kg/s**

Heat capacity rates.
$C_{oil} = Cp_{oil} \times M_{oil}$ = 2328×0.0006487=**1.51W/K**
$C_{water}=C_{p,\ water}\times M_{water}$ =4182×0.002979 =**12.45 W/K**
Ratio of heat capacity:
$C_r=C_{min}/C_{max}= C_{oil/}\ C_{water}$=**0.1212**

Reynolds number
$R_e = \rho vD/\mu$
$= (4 \times M_{oil})/(\pi D_i\ \mu oil)$
$= (4\text{x}0.0006487)/(\pi \times 0.1\ \text{x}0.00915)$ = **0.902**

Nusselt Number
$\mathbf{N_u} = 3.66 + (0.0668A/(1+(0.4 \text{ X } A^{0.667}))$
= **4.435**
$A = R_e P_r (D_{i/L})$ =**14.35 m^2**

Heat transfer coefficient
The inner pipe wall is:
$h_i = K_{oil} \times N_u / D_i$
$= (0.134 \times 4.435)/0.1 = 5.942\ w/m^2k$

Design overall heat transfer coefficient
$UA = 1/\{(1/h_i A_i)+(1/h_o A_o)+(\ln(D_O/D_i)/2\pi LK_{copper})\}$
= **0.7687 w/k**

Number of transfer units
$NTU = UA/C_{min}$ = 0.7687/1.5102 = **0.509**

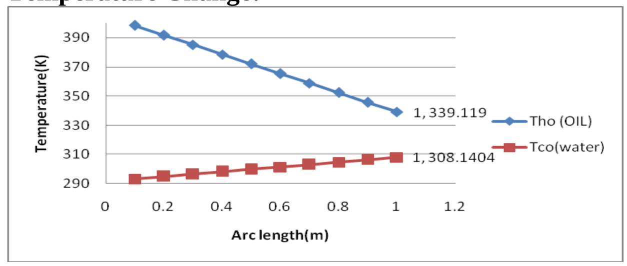
Effectiveness
$\acute{\varepsilon} = 1- e^{\{-NTU\ (1+Cr)\}/\ (1+Cr)}$
$= 1- e^{\{-0.59\ (1+0.12121)\}/\ (1+0.12121)}$
= **0.3879**

Heat capacity
$q = \acute{\varepsilon} \times C_{min} \times (T_{hi}-T_{ci})$
= **61.507 watts**

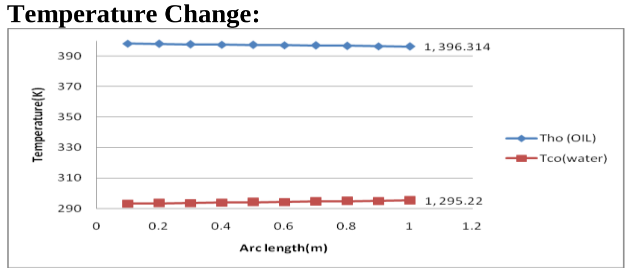
Cold and hot fluid out let temperatures
$T_{co} = T_{ci} + (q/\ C_{max)}$ = 293.15 + (61.507/12.45)
=**298.09 k**
$T_{ho} = T_{hi} - (q/\ C_{min)}$ =398.15 – (61.507/1.510)
= **357.42 k**

Log mean temperature difference
$q = UA\Delta T_{LM}$
ΔT_{LM}=61.507/0.7687=**80.017 k**
$\Delta T_{LM} = (\Delta T_2 - \Delta T_{1)}/\ \ln(\Delta T_2/\Delta T_{1)}$
=(357.42-298.08)-(398.15-293.15)/ln((357.42-298.08)/(398.15-293.15))
ΔT_{LM} =**80.17k**

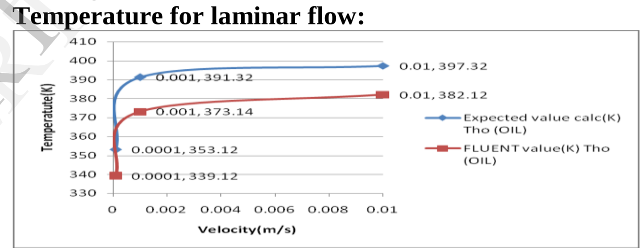
Laminar parallel Flow Heat Exchanger Temperature Change:



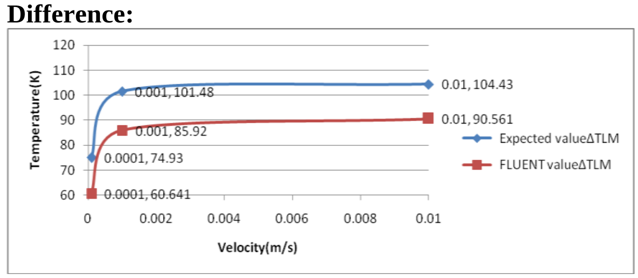
Turbulent parallel Flow Heat Exchanger Temperature Change:



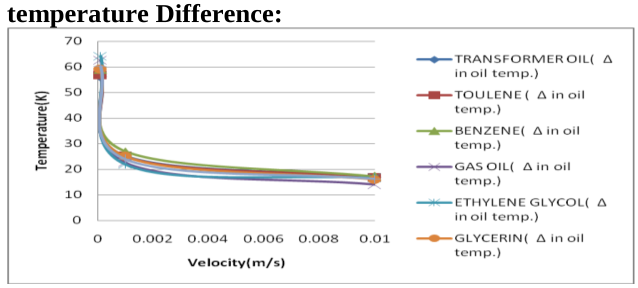
Cooling Water Flow Rate Effect on Oil Outlet Temperature for laminar flow:



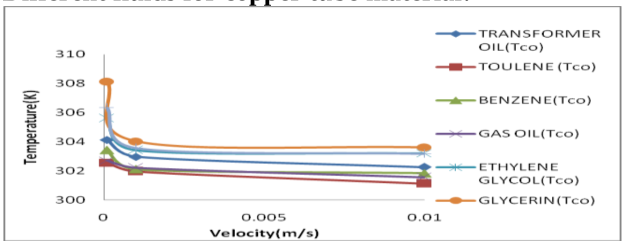
Cooling water velocity vs. the log mean temperature Difference:



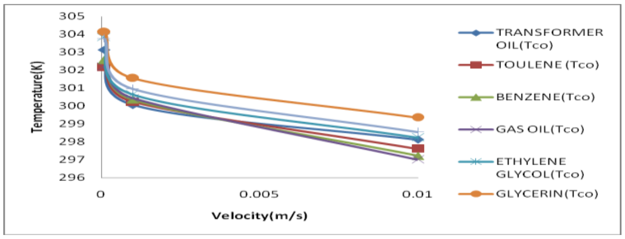
Cooling water velocity vs. Change in oil temperature Difference:



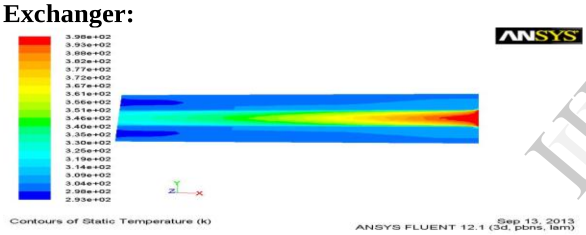
Cooling water velocity vs. the outlet temperature of Different fluids for copper tube material:

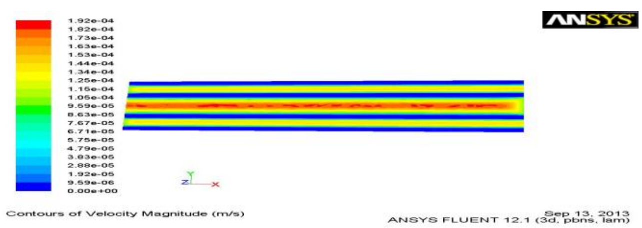


Cooling water velocity vs. the outlet temperature of Different fluids for aluminium tube material:

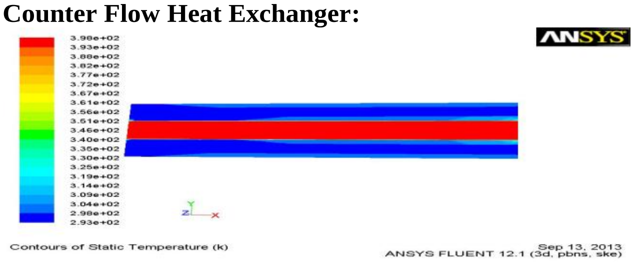


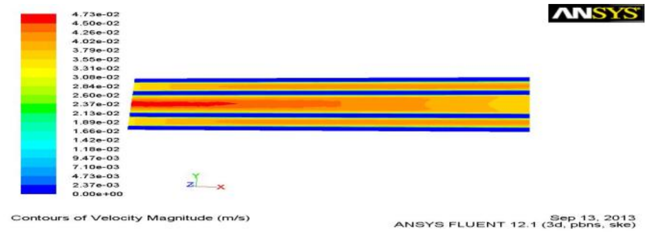
4.3 Concentric Tube Counter Flow Heat Exchanger:
Laminar Flow in a Counter Heat Exchanger:
Temperature and Velocity Profile for Counter Heat Exchanger:





Turbulent Flow in a Counter Heat Exchanger:
Temperature and Velocity Profile for a Turbulent Counter Flow Heat Exchanger:





Laminar Flow in Counter Heat Exchanger Problem Calculations:

Effectiveness

$\acute{\varepsilon} = 1- e^{\{-NTU\ (1-Cr)\}/\ \{1-\ Cr\ e\ \{-NTU\ (1-Cr)\}\}}$

$= 1- e^{\{-0.59\ (1+0.12121)\}/\ 1-\ 0.12121\ x\ e\ \{-0.59\ (1+0.12121)\}}$

$=$ **0.391**

Heat capacity

q = $\acute{\varepsilon}$ x C_{min} x $(T_{hi}-T_{ci})$ **= 61.988 watts**

Cold and hot fluid out let temperatures

$T_{co} = T_{ci} + (q/\ C_{max)} = 293.15 + (61.507/12.45)$ =**298.09 k**

$T_{ho} = T_{hi} + (q/\ C_{min)}$ =398.15 – (61.507/1.510) = **357.42 k**

Log mean temperature difference

q = UAΔT_{LM}

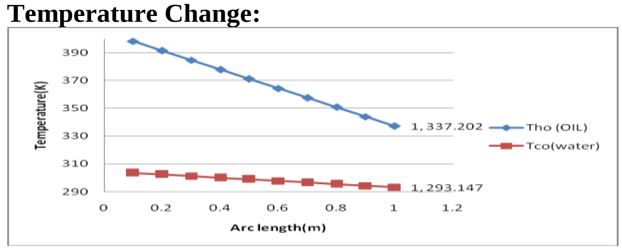
ΔT_{LM} = 61.998/0.7687 **= 80.645 k**

$\Delta T_{LM} = (T_{ho}- T_{ci)\ -}\ (T_{hi}- T_{co)}/\ \ln\ (T_{ho}\ -T_{ci}/T_{hi} - T_{co})$

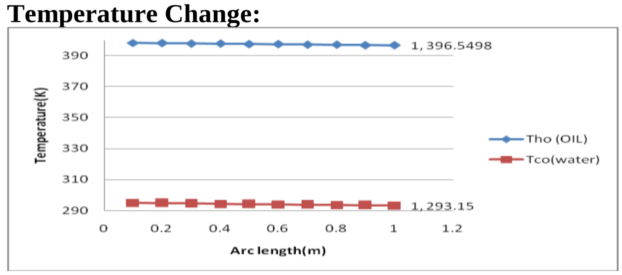
=(357.09-293.15)-(398.15-298.15)/ln((357.09-293.15)/(398.15-298.12))

ΔT_{LM} =**80.64k**

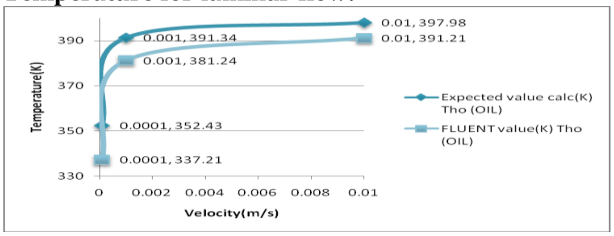
Laminar Counter- Flow Heat Exchanger Temperature Change:



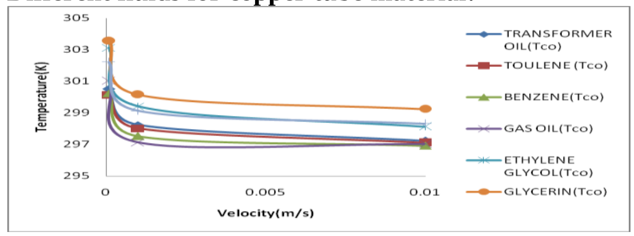
Turbulent Counter- Flow Heat Exchanger Temperature Change:



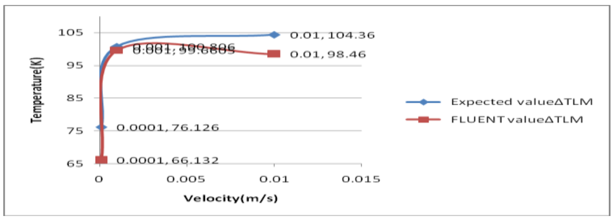
Cooling Water Flow Rate Effect on Oil Outlet Temperature for laminar flow:



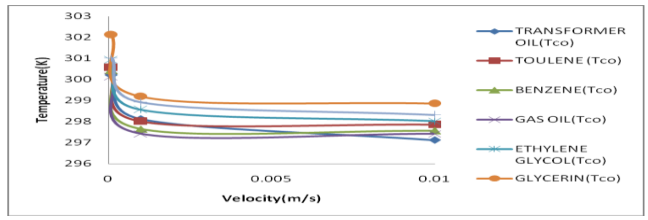
Cooling water velocity vs. the outlet temperature of Different fluids for copper tube material:



Cooling water velocity vs. the log mean temperature Difference:



Cooling water velocity vs. the outlet temperature of Different fluids for aluminium tube material:



Laminar parallel flow heat exchanger for copper material:

		Expected value for calculations							Fluent value		
Different fluids	Velocity (m/s)	Reynolds number	Heat transfer co-efficient of inner pipe (w/m2k)	Overall heat transfer co-efficient (w/k)	Heat capacity (watts)	Cold fluid out let temp (k).	Hot fluid outlet temp (k).	Log mean temp. Difference (k).	Cold fluid out let temp (k).	Hot fluid outlet temp (k).	Log mean temp. difference (k).
Transformer oil	0.0001	0.902	5.492	0.7687	61.507	298.09	357.42	80.17	304.12	342.2	65.92
	0.001	9.0273	11.02	1.426	142.07	294.29	388.74	99.63	302.94	373.1	88.83
	0.01	90.17	25.56	3.3062	342.91	293.42	395.08	103.70	302.21	381.24	90.96
toulene	0.0001	14.77	3.85	0.756	56.47	297.68	348.52	80.32	302.56	341.3	66.46
	0.001	147.78	6.59	1.172	116.92	294.68	387.55	99.29	301.96	371.24	85.90
	0.01	1477.8	7.94	0.976	123.92	293.29	397.06	126.94	301.12	380.31	91.9
benzene	0.0001	14.85	6.184	0.797	59.6	297.73	348.79	79.69	303.41	339.2	64.82
	0.001	148.55	6.96	0.91	91.30	293.89	390.57	97.57	302.11	375.61	88.31
	0.01	1488.5	8.38	1.08	112.36	293.25	397.21	104.23	301.82	380.92	91.43
Gas oil	0.0001	2.5	5.373	0.693	55.28	297.58	356.87	79.76	302.71	335.4	62.01
	0.001	25	5.967	0.774	78.64	293.71	392.34	101.46	302.21	377.31	89.25
	0.01	250	7.317	0.946	99.51	293.22	397.38	105.12	301.56	384.21	93.37
Ethylene glycol	0.0001	0.7078	9.83	1.275	95.9	300.8	352.57	75.4	305.61	334.1	59.06
	0.001	7.078	11.01	1.432	143.1	294.24	391.33	99.86	303.11	376.32	88.14
	0.01	70.78	13.42	1.72	180.8	293.29	397.32	105.21	303.12	381.56	91.07
glycerine	0.0001	0.0158	11.15	1.44	107.86	301.76	353.12	74.93	308.14	339.2	60.64
	0.001	0.158	12.49	1.625	163.41	294.81	391.32	101.48	303.82	373.14	85.92
	0.01	1.58	15.23	1.96	204.69	293.31	397.12	104.43	304.61	382.12	90.56
water	0.0001	37.816	9.015	1.034	115.93	299.56	351.83	76.53	306.32	337.2	60.68
	0.001	378.16	9.69	1.235	136.26	294.29	387.24	97.68	303.12	374.42	87.12
	0.01	3781.6	11.56	1.486	198.26	293.46	390.98	104.82	303.56	381.92	91.03

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Laminar flow Counter heat exchanger for copper material:

		Expected value for calculations							Fluent value		
Different fluids	Velocity (m/s)	Reynolds number	Heat transfer co-efficient of inner pipe (w/m2k)	Overall heat transfer co-efficient(w/k)	Heat capacity (watts)	Cold fluid out let temp (k).	Hot fluid outlet temp (k).	Log mean temp. differen ce(k).	Cold fluid out let temp (k).	Hot fluid outlet temp (k).	Log mean temp. differen ce (k).
Transfor mer oil	0.0001	0.902	5.492	0.7687	61.507	298.12	357.09	80.64	300.51	338.56	68.25
	0.001	9.0273	11.02	1.426	142.07	294.42	387.64	99.03	298.24	378.25	92.32
	0.01	90.17	25.56	3.3062	342.91	293.42	395.82	104.23	297.24	386.34	96.9
toulene	0.0001	14.77	3.85	0.756	56.47	297.72	347.95	77.32	300.14	335.11	66.06
	0.001	147.78	6.59	1.172	116.92	294.03	387.8	99.27	298.01	380.22	93.45
	0.01	1477.8	7.94	0.976	123.92	293.25	397.31	104.5	297.11	390.62	99.24
benzene	0.0001	14.85	6.184	0.797	59.6	297.96	348.46	75.19	300.24	338.24	68.12
	0.001	148.55	6.96	0.91	91.30	293.88	390.51	101.6	297.52	382.56	94.9
	0.01	1488.5	8.38	1.08	112.36	293.23	397.13	104.51	296.91	390.81	102.24
Gas oil	0.0001	2.5	5.373	0.693	55.28	297.60	356.55	80.56	301.02	339.21	69.104
	0.001	25	5.967	0.774	78.64	293.7	392.31	102.12	297.12	383.21	92.81
	0.01	250	7.317	0.946	99.51	293.29	397.4	104.76	297.01	390.42	99.14
Ethylene glycol	0.0001	0.7078	9.83	1.275	95.9	301.42	351.42	80.98	303.12	336.24	65.64
	0.001	7.078	11.01	1.432	143.1	294.3	391.28	101.29	299.41	386.24	96.01
	0.01	70.78	13.42	1.72	180.8	293.29	398.01	104.39	298.12	391.24	99.05
glycerine	0.0001	0.0158	11.15	1.44	107.86	301.94	352.43	76.12	303.56	337.21	66.138
	0.001	0.158	12.49	1.625	163.41	294.46	391.34	100.8	300.16	381.24	99.67
	0.01	1.58	15.23	1.96	204.69	294.04	397.98	104.36	299.24	391.21	98.49
water	0.0001	37.816	9.015	1.034	115.93	301.56	357.76	78.79	302.22	338.12	67.24
	0.001	378.16	9.69	1.235	136.26	294.38	391.21	100.85	299.31	383.92	94.74
	0.01	3781.6	11.56	1.486	198.26	293.56	398.21	104.82	298.32	391.56	99.11

Turbulent parallel and counter flow heat exchanger for copper material:

Different fluids	Velocit y (m/s)	Parallel flow			Counter flow		
		Cold fluid out let temp. (K)	Hot fluid outlet temp. (K)	Log mean temp. differe nce(K)	Cold fluid out let temp. (K)	Hot fluid outlet temp. (K)	Log mean temp. differenc e(K)
Transforme r oil	0.56	294.2	396.52	103.65	294.01	395.92	103.45
toulene	0.035	294.1	397.21	104.04	293.91	396.21	103.919
benzene	0.56	294.3	397.56	104.21	293.61	396.56	103.97
Gas oil	0.22	293.2	396.92	104.35	294.12	396.32	103.59
Ethylene glycol	0.72	293.5	397.21	104.32	294.32	397.11	103.89
glycerine	0.56	294.1	396.32	103.57	294.92	396.56	103.319
water	0.002	104.2	397.11	104.23	294.21	397.12	103.95

Laminar parallel and counter flow heat exchanger for Aluminium material:

Different fluids	Velocity (m/s)	Parallel flow			Counter flow		
		Cold fluid out let temp. (K)	Hot fluid outlet temp. (K)	Log mean temp. difference (K)	Cold fluid out let temp. (K)	Hot fluid outlet temp. (K)	Log mean temp. difference(K)
Transformer oil	0.0001	303.16	332.1	57.78	300.21	336.26	66.78
	0.001	302.56	384.16	92.8	298.12	372.16	89.24
	0.01	298.18	391.26	98.94	297.22	390.56	99.204
toulene	0.0001	302.16	334.21	61.47	300.56	335.61	66.24
	0.001	301.62	381.01	91.59	298.01	378.56	92.59
	0.01	297.61	390.56	98.85	297.12	390.96	99.04
benzene	0.0001	302.56	336.21	62.71	300.42	333.21	64.66
	0.001	301.86	381.26	91.63	297.68	380.26	93.94
	0.01	297.41	392.56	100.82	297.56	392.52	99.78
Gas oil	0.0001	302.18	337.21	63.73	300.12	336.79	67.206
	0.001	300.92	382.42	92.75	297.42	382.56	94.95
	0.01	297.01	393.42	100.64	297.42	391.56	99.56
Ethylene glycol	0.0001	303.86	334.18	60.12	300.92	336.12	66.48
	0.001	302.14	385.1	93.41	298.56	383.21	94.74
	0.01	298.24	391.56	99.04	298.02	391.92	99.44
glycerine	0.0001	304.16	338.21	63.03	302.25	337.21	66.704
	0.001	303.56	384.56	92.48	299.19	381.56	93.58
	0.01	299.36	391.92	98.64	298.82	392.56	99.36
water	0.0001	303.75	337.42	62.71	301.02	336.81	66.79
	0.001	302.96	383.21	92.07	298.92	383.41	94.62
	0.01	298.56	391.02	98.59	298.32	391.56	98.71

Turbulent parallel and counter flow heat exchanger for Aluminium material:

Different fluids	Velocity (m/s)	Parallel flow			Counter flow		
		Cold fluid outlet temp. (K)	Hot fluid outlet temp. (K)	Log mean temp. Difference (K)	Cold fluid out let temp. (K)	Hot fluid outlet temp. (K)	Log mean temp. Difference (K)
Transformer oil	0.56	294.12	397.01	103.94	294.01	396.21	103.86
toulene	0.035	294.04	392.31	101.59	293.86	394.87	102.98
benzene	0.56	294.36	389.25	99.85	293.61	396.21	102.96
Gas oil	0.22	293.96	394.56	102.78	294.01	394.24	102.45
Ethylene glycol	0.72	294.12	397.12	98.403	294.12	397.56	104.21
glycerine	0.56	294.93	397.61	103.83	294.56	397.21	103.62
water	0.002	294.23	396.13	103.58	294.12	397.45	104.06

5. Conclusions

The performance, CFD analysis of different fluids and different pipe materials were investigated on parallel and counter flow in concentric tube heat exchanger. The conclusions of this investigating at are as follows.

- The main objective of this project was to analyse the fluid flow in double pipe heat exchangers and the subsequent performance of these heat exchangers.
- To facilitate this analysis, the ANSYS FLUENT 12.1 finite volume analysis program was used to perform the modelling and calculations. In order to verify the development of each model, the models were built in stages and each stage analysed and verified.
- The first stage was a modified Graetz problem model where velocity and temperature profiles were analysed for fluid flow in a tube of 1.0 m in length. This basic model was verified by hand calculations and the percent difference was seen to be less than 2%.
- It should be noted, that for the lower velocity of .0001 m/s, the laminar model actually cooled the outlet flow to 305.148 K in the modified Gratez problem while the turbulent model only cooled the flow to 322.25503 K. It was also discovered that the centreline temperature from start to finish (0 to 1 m) had a more parabolic, gradual slope lowering from the inlet temperature of 323.15 K to 305.93648 K while the turbulent flow had an almost linear relationship with the centreline temperature against the arc length.
- The parallel and counter flow models first were verified to be providing the same cooling relationship as expected from these types of heat exchangers. All models with and without fouling showed that as the cooling water flow increased, so did the amount of heat transfer by a decreasing oil outlet temperature and increasing oil temperature difference.
- The ANSYS FLUENT results were found to be fairly consistent with hand calculations with most of the values within 5% of each other.
- The material properties which were temperature dependent were considered constant based on the inlet temperature of the fluid flow for the hand calculations instead of using the average temperature whereas the ANSYS FLUENT values were from the material library.
- The fouled heat exchangers performance was much lower than the non-fouled heat exchanger. There was an approximate 2 K rise in outlet temperature for a fouled concurrent heat exchanger and an approximate 1 K rise in outlet temperature for a fouled countercurrent heat exchanger vice a fraction of a Kelvin change between the concurrent and countercurrent heat exchangers.
- This finding proves it is more important for engineers and developers to focus on the method of preventing damage to the heat transfer surfaces and the type of material chosen than it is to focus on the type of flow. The more time that is spent researching how a material will perform over time and the type of corrosion that occurs with the material or ways to prevent corrosion and deterioration in the system will be more efficient.

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