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NUMERICAL AND EXPERIMENTAL ANALYSIS OF THERMAL PERFORMANCE IN A PARALLEL FLOW AND COUNTER FLOW DOUBLE PIPE HEAT EXCHANGE

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Keywords:

Heat transfer, Double pipe heat exchanger, CFD simulation software, Organic fluids, Copper, Aluminium,Laminar flow, Turbulent flow, Parallel flow and Counterflow

Abstract

Heat transfer is taken into consideration as transfer of thermal energy from bodily frame to another. It is the most essential parameter to be measured because the overall performance and efficiency of the double pipe heat exchanger. By the use of CFD simulation software, it reduces the time and operation cost in comparison to experimental calculations, The goal of this paper is to evaluate the influence of the Double pipe heat exchanger to get better heat transfer. The purpose of this examine is to apply CFD software program and Experimental setup to analyze the Temperature drop, Pressure drop and Friction factors, by varying under different inlet conditions like Temperature and Flow rate as a function of both inlet velocity and temperature variations and converting heat exchanger tube material properties like copper and aluminium.

In this experiment, heat transfer from hot fluid to cold fluid with the aid of Double pipe heat exchanger is experimentally by using different organic fluids like Benzene, Glycol, Transformer oil, Acetone and Water to get better heat transfer, the identical thing is established in CFD analysis. The test is accomplished at Laminar flow beneath different flow preparations like Parallel flow and Counterflow.

The test is done ondouble pipe heat exchanger under various working fluids with different operating conditions, it is predicted that Counterflow exhibits better heat transfer than Parallel flow.

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1. Introduction

The heat exchanger is a device, along with an automobile radiator, used to transfer heat from a fluid on one facet of a barrier to a fluid on the opposite side without bringing the fluid into direct contact (Fogiel, 1999). Typically, this barrier is crafted from metal which has excellent thermal conductivity so as to transfer heat efficiently from one fluid to another fluid.

Whilst heat exchanger is directly fired via a combustion technique, it becomes furnace, boiler, heater, tube-still heater and engine. Vice versa, while heat exchanger make a alternate in phase in one of flowing fluid including condensation of steam to water, it turns into a chiller, evaporator, sublimator, distillation-column reboiler, still, condenser or cooler-condenser.

1.1 Basic Heat Exchanger Flow Arrangements

Two basic flow arrangements are as shown in Figure 1.8. Parallel and counter flow provides alternative arrangements for certain specialized applications. In parallel flow both the hot and cold streams enter the heat exchanger at the same end and travel to the opposite end in parallel streams. Energy is transferred along the length from the hot to the cold fluid so the outlet temperatures asymptotically approach each other. In a counter flow arrangement, the two streams enter at opposite ends of the heat exchanger and flow in parallel but opposite directions. Temperatures within the two streams tend to approach one another in a nearly linearly fashion resulting in a much more uniform heating pattern. Shown below the heat exchangers are representations of the axial temperature profiles for each. Parallel flow results in rapid initial rates of heat exchange near the entrance, but heat transfer rates rapidly decrease as the temperatures of the two streams approach one another. This leads to higher energy loss during heat exchange. Counter flow provides for relatively uniform temperature differences and, consequently, lead toward relatively uniform heat rates throughout the length of the unit.

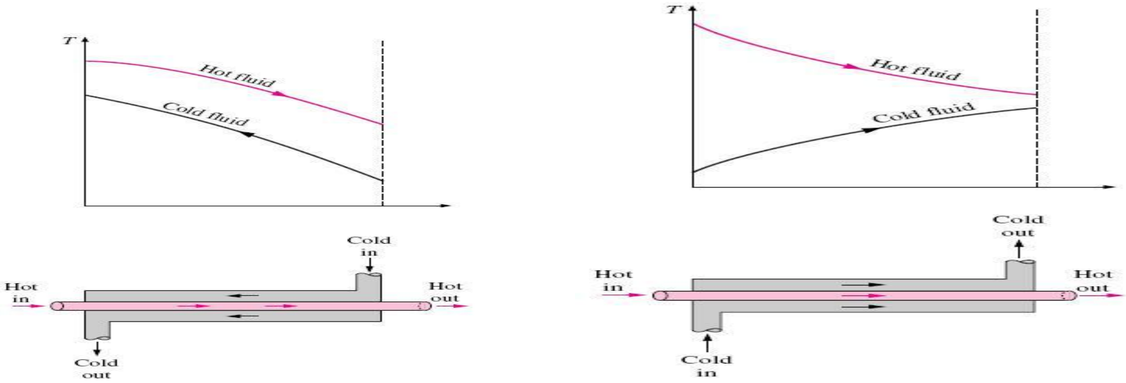


Fig. 1.1 Basic Flow Arrangements for Tubular Heat Exchangers.

1.2 Applications for Counter and Parallel Flows

We have seen two advantages for counter flow, (a) larger effective LMTD and (b) greater potential energy recovery. The advantage of the larger LMTD, as seen from the heat exchanger equation, is that a larger LMTD permits a smaller heat exchanger area, Ao, for a given heat transfer, Q. This would normally be expected to result in smaller, less expensive equipment for a given application. Sometimes, however, parallel flows are desirable (a) where the high initial heating rate may be used to advantage and (b) where it is required the temperatures developed at the tube walls are moderate. In heating very viscous fluids, parallel flow provides for rapid initial heating and consequent decrease in fluid viscosity and reduction in pumping requirement. In applications where moderation of tube wall temperatures is required, parallel flow results in cooler walls.

1.3 Heat Exchanger Effectiveness (ε)

The effectiveness ε is the ratio of the actual heat transfer rate to the maximum possible heat transfer rate:

$$\varepsilon = \frac{q_{actual}}{q_{\max}}, 0 \le \varepsilon \le 1 \text{..........(2)}$$

Where Q_{max} is for an infinitely long Hex

The effectiveness equation is usually defined by the type of heat exchanger. The equations for effectiveness include the value of NTU (number of transfer units) and Cr(ratio of heat capacities). These values are arranged into different equations depending upon the type of heat exchanger.

There are many uses for heat exchangers from car radiators, to air conditioners, to large condensers in power plants. But for all applications the effectiveness of these heat exchangers are dependent on many factors. Not only does the viscosity and density of the fluids affect the heat transfer due to being a factor of the Reynolds number and therefore Nusselt number, but the inlet velocity (mass flow rate) and temperatures of the fluids are proportional to the heat transfer rate.

$$\dot{q} = \dot{m} \times c \times (T_h - T_c) \quad \text{...........(3)}$$

This paper looks at the heat exchange between fluids in concentric tube heat exchangers. In this type of heat exchanger, forced convection is caused by fluid flow of different temperatures passing parallel to each other separated by a boundary, pipe wall. Several assumptions will have to be made to make it easier to focus on the inlet velocity and temperature dependence on heat exchanger temperature drop. Not only will the viscosity and density remain constant for the calculations, but specific heat and overall heat transfer coefficients will be assumed constant. Any effects from potential and kinetic energy are assumed negligible.

1.4Heat Exchanger Analysis Theory

Two types of analysis for parallel flow heat exchangers to determine temperature drops are the log mean temperature difference and the effectiveness-NTU method. Both methods will be attempted to be used for the project. The equation for heat transfer using the log mean temperature difference becomes:

$$q = UA\Delta T_{lm} = UA \times \frac{\Delta T_2 - \Delta T_1}{\ln\left({}^{\Delta T_2}\!/_{\Delta T_1} \right)} \quad \text{...........(4)}$$

Where the only change for parallel and countercurrent flow is how the delta-T are defined. The equation for friction factor using the pressure drop becomes:

F= **ΔP / (L/di)*(ρv²/2)** ....(5)

2 CATIA MODELING:

In the process of the Catia modeling of Double Pipe Heat Exchanger we have to design the following Parts.

2.1 Inner pipe:

Dimensions:

Pipe outer Diameter	= 25mm
Pipe inner Diameter	= 21mm
Thickness	= 4mm
Tube Length	= 1400mm

2.2 Outer pipe:

Dimensions:

Pipe outer Diameter	= 38mm
Pipe inner Diameter	= 32mm
Thickness	= 6mm
Tube Length	= 1100mm

Fig.2.1 Designed Catia model of Double pipe Heat Exchanger

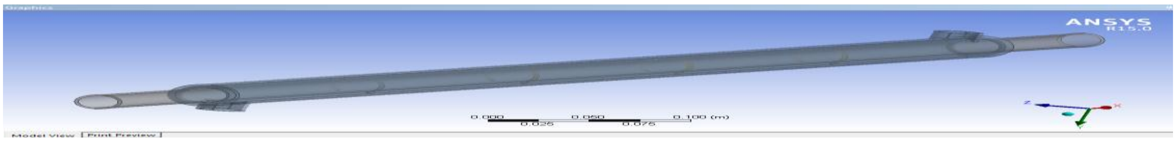
3 CFD:

Computational fluid dynamics (CFD) study of the system starts with the construction of desired geometrythe geometry (physical bounds) of the problem is defined.The volume occupied by the fluid is divided into discrete cells (the mesh). The mesh may be uniform or non-uniform. Generally, geometry is simplified for the CFD studies

3.1 Geometry:

Heat exchanger is built in the ANSYS workbench design module. It is a counter-flow & parallel flow heat exchanger. First, the fluid flow (fluent) module from the workbench is selected. The design modeler opens as a new window as the geometry is double clicked.

Fig. 3.1 Imported model in geometry

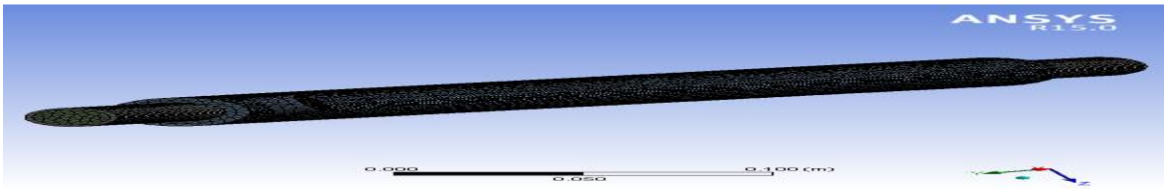


3.1.2 Meshing:

Initially a relatively coarser mesh is generated. This mesh contains mixed cells (Tetra and Hexahedral cells) having both triangular and quadrilateral faces at the boundaries. Care is taken to use structured hexahedral cells as much as possible. It is meant to reduce numerical diffusion as much as possible by structuring the mesh in a well manner, particularly near the wall region. Later on, a fine mesh is generated. For this fine mesh, the edges and regions of high temperature and pressure gradients are finely meshed.

The different surfaces of the solid are named as per required inlets and outlets for inner and outer fluids.

Fig. 3.2: Double pipe model after Meshing



Save project again at this point and close the window. Refresh and update project on the workbench. Now open the setup. The ANSYS Fluent Launcher will open in a window. Set dimension as 3D, processing as Serial type and hit OK. The Fluent window will open.

3.1.3 Setup: The mesh is checked and quality is obtained.

3.1.4 Materials: The create/edit option is clicked to add required fluids among glycerin, glycol, acetone, benzene, water liquid, aluminum and copper from the list of fluid and solid respectively from the fluent database.

3.1.5 Cell Zone conditions: In cell zone conditions, we have to assign the conditions of the liquid and solid.

Table 3.1 cell zone conditions

Sno.	PART/BODY	MATERIAL
1.	INNER FLUID	WATER-LIQUID
2.	OUTER FLUID	WATER-LIQUID
3.	INNER PIPE	COPPER
4.	OUTER PIPE	COPPER
5.	BAFFLES	COPPER

3.1.6 Boundary conditions: Boundary conditions are used according to the need of the model. The inlet and outlet conditions are defined as mass flow inlet. The direction specification method is defined as normal to boundary. The details about all boundary conditions can be seen in the table as given below.

Table 3.23 Boundary conditions

	BOUNDARY CONDITION TYPE	MASS FLOW RATE(kg/s)	TEMPERATURE (k)
INNER INLET	Mass flow inlet	0.02	312
OUTER INLET	Mass flow inlet	0.01	300

3.1.7 Run calculation:

After giving the boundary conditions to the inner and outer fluid, finally we have to run the calculations.The number of iteration is set to 500 and the solution is calculated and various contours, vectors and plots are obtained.

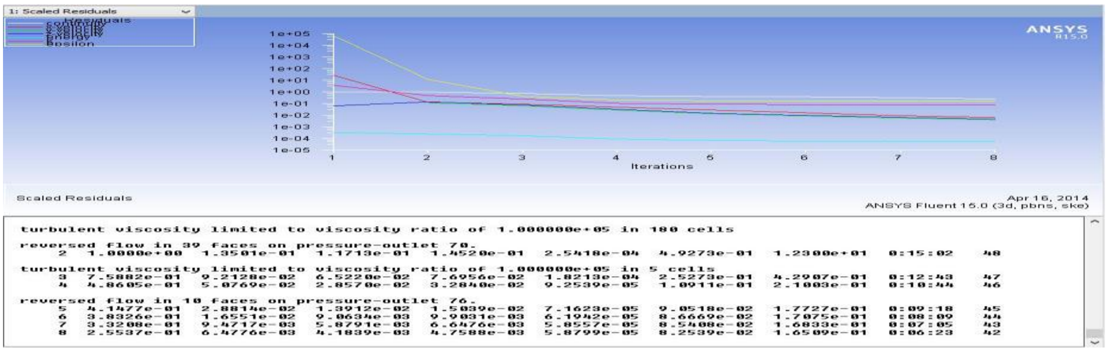
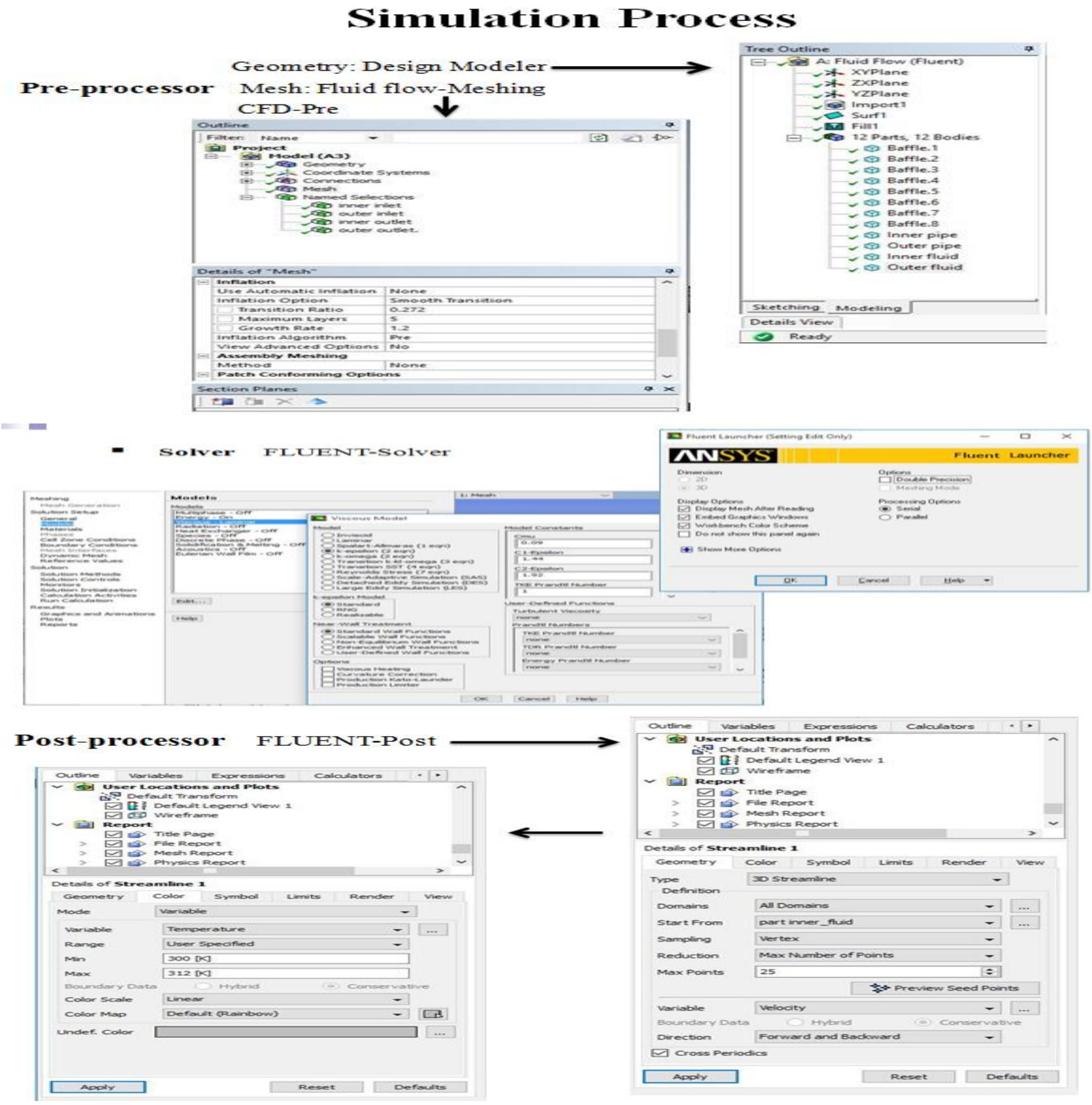


Fig 3.3 Calculations are running

3.1.8 POST PROCESSOR PRE PROCESSOR SCREEN SHOTS:



4. Methodology and Approach:

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 double pipe heat exchanger. By using CFD simulation software, it can reduce the time and operation cost compared to experimental calculations, in order to measure the optimum parameter and the behavior of this type of heat exchanger.

The experiment is carried under different flow arrangements i.e. turbulent flow and laminar flow which are further carried out in parallel and counter flow using different organic fluids like glycerin, glycol, acetone, benzene, transformer oil, water keeping water as base fluid and by varying only hot fluids. The inlet flow conditions like mass flow rate are varied maintaining the temperature constant i.e. at inner inlet 312^{o}K and at outer inlet 300^{o}K, the mass flow rates in laminar flow for inner inlet & outer inlet is taken as 0.0133, 0.0166, 0.02 kg/s and 0.01, 0.0133, 0.0166 kg/s similarly in turbulent flow the mass flow rates for inner inlet & outer inlet are taken as 1.2, 1.8, 2.4 kg/s & 1.166, 1.66 & 2.166 kg/s.

The Physical parameters like Materials properties and baffle segmental were introduced into the double pipe heat exchanger fluent model as the properties cannot be varied in experimental setup heat exchanger.

Materials with good thermal conductivities like copper and aluminum where opted for analyzing the heat exchanger. The baffle segmental are introduced to the heat exchanger on the surface of inner pipe at different angular orientations i.e. 0^{o}, 45^{o}, 90^{o} and 135^{o} to predict that at which angular orientation better pressure drop, and heat transfer co-efficient are building up

4.1 Experimental setup:

Fig: 4.1: Double pipe heat exchanger

5 Defining Material Properties

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.

Table 5.1: material properties

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

5.1 Defining fluid Properties

Water was used as the base fluid flowing through outer pipe. Fluids like glycerin, glycol, acetone, benzene & water are used as hot fluids which are allowed to flow through inner pipe by maintaining constant inner temperature.The fluids were defined in FLUENT using its fluids 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, viscosity, specific heat.

FLUIDS	DENSITY (Kg/m^3)	VISCOSITY (Ns/m^2)	SP.HEAT (Kj/kgk)	THERMALCONDUC TIVITY (W/mk)
ACETONE	791	0.00033	2160	0.180
BENZENE	876.5	0.00058	1821	0.167
GLYCERIN	1261	0.799	2813	0.285
GLYCOL	1116	0.0157	2200	0.258
WATER	998.2	0.001003	4174	0.6

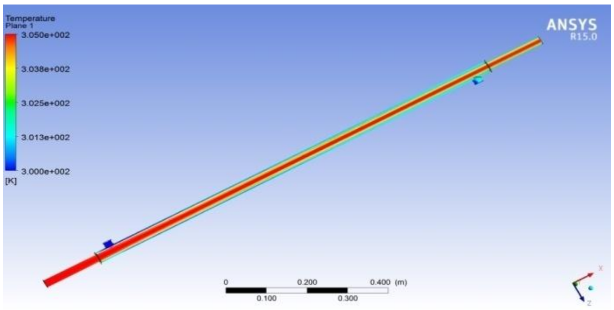
Table 5.2: Fluid properties

5.2 Experimental validation for parallel flow:

The temperature, pressure & velocity variation in a parallel flow double pipe heat exchanger of copper material performed for laminar flow is as shown in below profiles.

5.2.1Temperature, pressure & Velocity Profile for parallel flow Heat Exchanger:

At mass flow rate 0.02 (Plane representation)



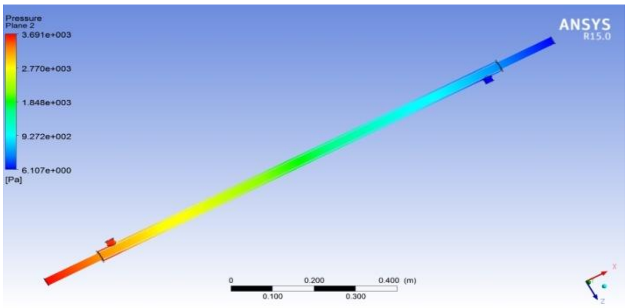


Fig 5.2.1Temperature variation **Fig 5.2.2Pressure variation**

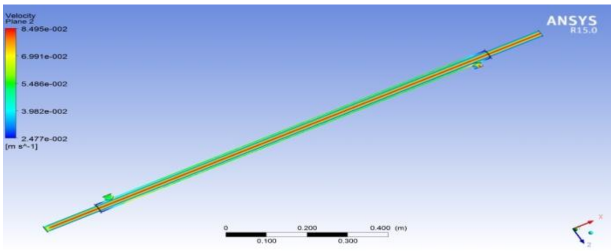
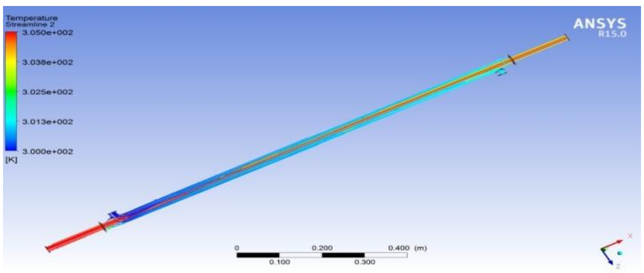


Fig 5.2.3Velocity variation

-At mass flow rate 0.02 (Stream line representation)



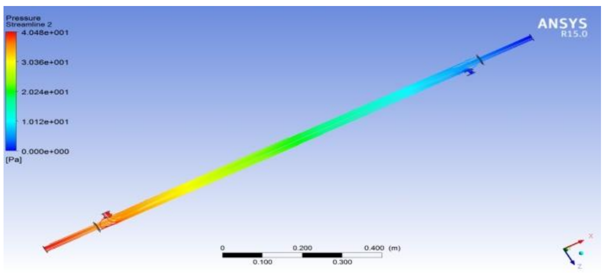


Fig 5.2.4 Temperature variation

Fig 5.2.5Pressure variation

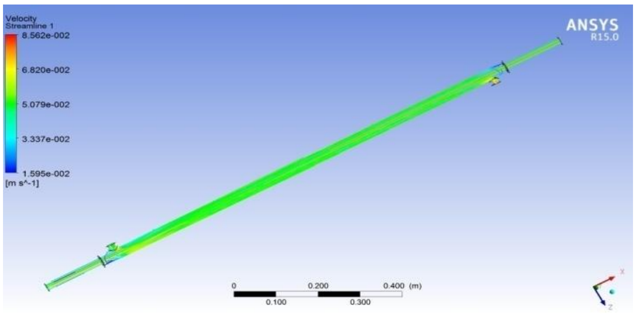


Fig 5.2.6 Velocity variation

5.2.2 Sample Calculation for Parallel Flow:

Thi= 312 K Tci= 300 K

Tho=307.138 K Tco= 303.597 K

Mh = (1.2)/60 = **0.02 liters/s**

Mc = (1)/60 = **0.016667 liters/s**

Cph = 2813 J/kgk Cpc = 4174 J/kgk

Qh= Mh Cph (Thi –Tho) watts

= **273.53612 Watts**

Qc= Mc Cpc (Tco– Tci) watts

= **250.245213 Watts**

Qa = (Qh + Qc)/2

= **261.8906667Watts**

LMTD = (θ2 – θ1)/ Ln (θ2/θ1)

θ2 = Thi– Tci = 12 K

θ1 = Tho – Tco = 3.5408 K

LMTD = **6.93062357 K**

Overall heat transfer coefficient:

Uo=Qa/ (Ao* LMTD) = (261.890667) / (0.0863*6.93062357) =**391.4060045 W/m2K**

5.2.3Parallel flow study results:

For parallel flow heat exchangers the hotter fluid will lower in temperature as it loses heat to the cooler fluid which will then rise in temperature due to the heat transfer Figure 5.7 shows this gradual temperature change in both flow paths. This is the correct curve form already proven for co-current flow heat exchangers.

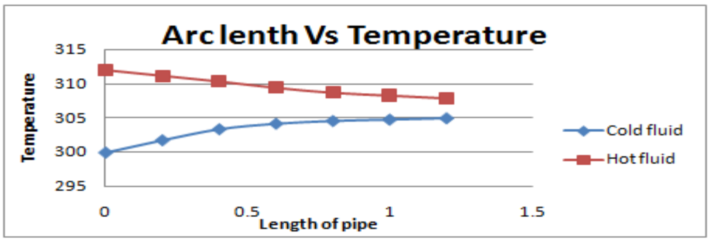


Fig 5.2.7 Parallel Flow profile curve

5.2.4 Hot fluid temperature variation:

From the below figure the hot fluid inlet and outlet temperatures is directly proportional to mass flow rate i.e. on increasing the mass flow rate the temperature of hot fluid increases

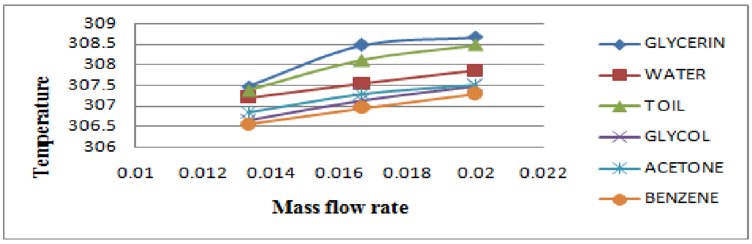


Fig 5.2.8 Temperature vs. Mass flow rate

Above graph illustrates that glycerin has acquired highest hot inlet & outlet temperatures compared to remaining fluids

5.2.5 Cold fluid temperature variation:

From the below figurethe cold fluid inlet and outlet temperatures is inversely proportional to mass flow rate i.e. on increasing the mass flow rate the temperature of cold fluid decreases.

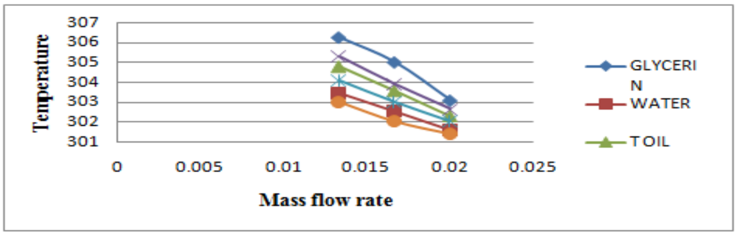


Fig 5.2.9 Temperature vs. Mass flow rate

5.2.6 Fluent Values vs. Experimental values for parallel flow:

(a) Co-efficient of heat transfer (Q_A)

The below figure 5.10 shows that co-efficient of heat transfer directly proportional to mass flow rate i.e. by increasing mass flow rate, the co-efficient of heat transfer increases.

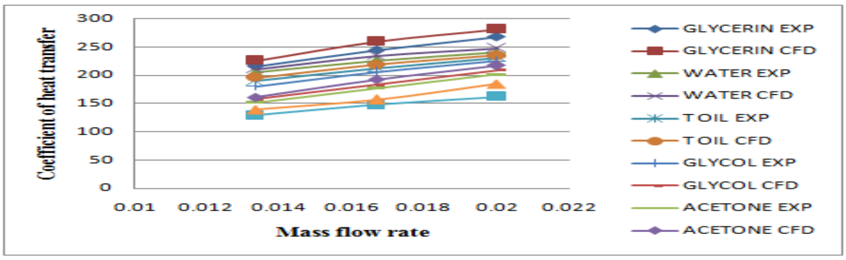


Fig5.2.9.1 Co-efficient of heat transfer vs. Mass flow rate

As shown in figure, Glycerin CFD acquired better co-efficient of heat transfer compared to other organic fluids on increasing mass flow rate.

(b) Overall heat transfer co-efficient (U_o):

The below figure 5.11 shows that co-efficient of heat transfer directly proportional to mass flow rate i.e. by increasing mass flow rate, the Overall heat transfer co-efficientincreases

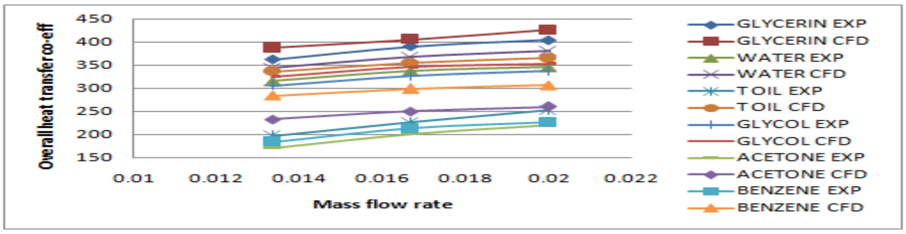


Fig 5.2.9.2 Overall heat transfer Co-efficient vs. Mass flow rate

As shown in figure, Glycerin CFD acquired better overall heat transfer co-efficient compared to other organic fluids on increasing mass flow rate.

(c) Log Mean Temperature Difference (LMTD):

The below figure 5.12 shows that logarithmic mean temperature difference directly proportional to mass flow rate i.e. by increasing mass flow rate, LMTD increases.As shown in figure, Glycerin CFD acquired better logarithmic mean temperature difference compared to other organic fluids on increasing mass flow rate.

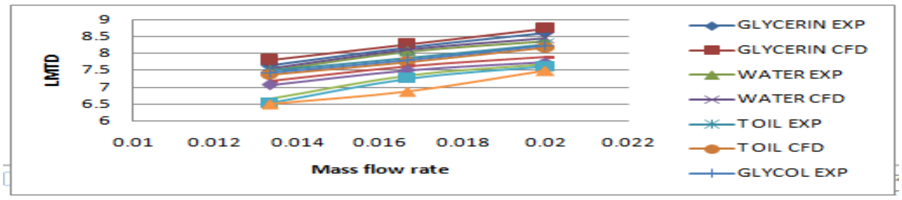


Fig 5.2.9.3 LMTD vs. Mass flow rate

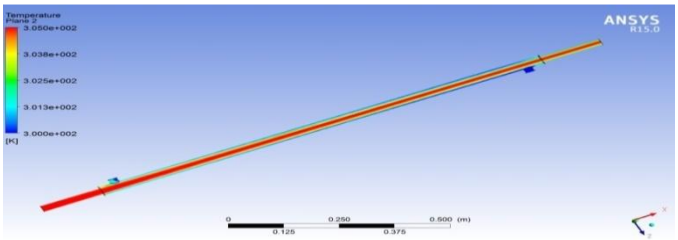
From the above graphical representation, it is resolved that the obtained values like Coefficient of heat transfer (Q_A), Overall heat transfer coefficient (U_O) and Logarithmic mean temperature difference (LMTD) are proved to be within 6% of error for laminar parallel flow validation. So the analysis is terminated successfully.

5.3 Experimental validation for counter flow:

5.3.1 Laminar Flow in a Counter Heat Exchanger FLUENT model:

Temperature, pressure & Velocity Profile for counter flow Heat Exchanger:

At mass flow rate 0.02 (Plane representation)



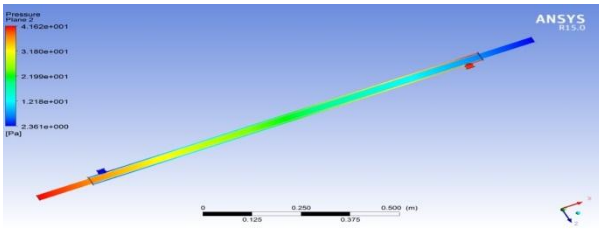


Fig.5.3.1Temperature variation **Fig.5.3.2 Pressure variation**

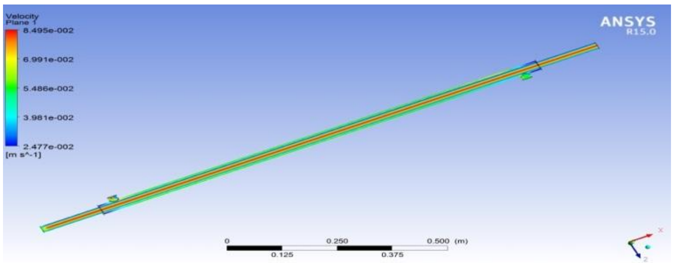
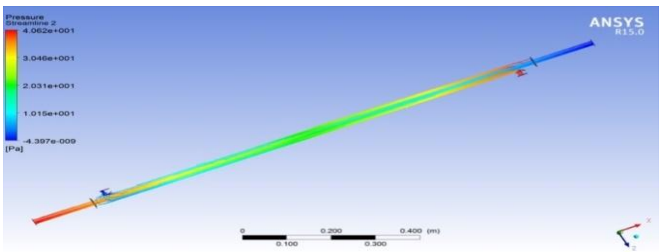


Fig.5.3.3 Velocity variation

-At mass flow rate 0.02 (Stream line representation)



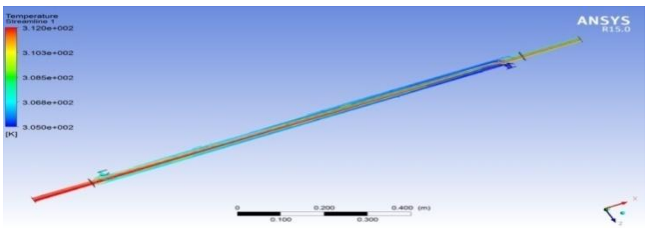


Fig.5.3.4Temperature variation **Fig.5.3.5 Pressure variation**

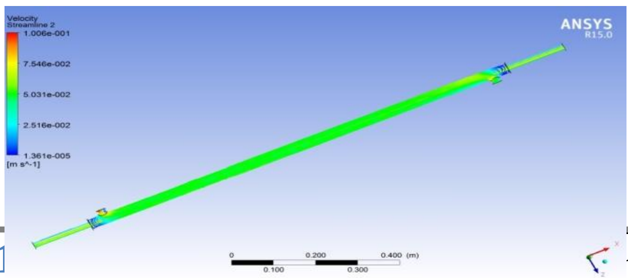


Fig 5.3.6 Velocity variation

5.3.2 Sample Calculation for Counter Flow:

Thi = 312 K   Tci = 300 K

Tho =306.80013 K   Tco = 303.991287 K

Mh = (1.2)/60 = **0.02 liters/s**Mc = (1)/60 = **0.016667 liters/s**

Cph = 2813 J/kgk   Cpc = 4174 J/kgk

Qh = Mh Cph (Thi –Tho) watts

Qh =**292.5442361 Watts**.

Qc = Mc Cpc (Tco– Tci) watts

Qc = **277.6605323 Watts**

Qa = (Qh + Qc)/2= **285.102384 Watts**

LMTD = (θ2 – θ1)/ Log (θ2/θ1)

θ2 = Tho – Tci =306.80013 – 300 = 6.800138 K

θ1 = Thi – Tco = 312 - 303.991287 = 8.0088 K

LMTD = (θ2 – θ1)/ Log (θ2/θ1) = **7.38795723 K**

Overall heat transfer coefficient:

Uo = Qa/ (Ao* LMTD)

= (285.102384) / (0.0863= **447.1628144 W /m2K**

5.3.3Counter flow study results:

For counter flow heat exchangers the hotter fluid will lower in temperature as it loses heat to the cooler fluid which will then rise in temperature due to the heat transfer Figure 5.19 shows this gradual temperature change in both flow paths. This is the correct curve form already proven for counter-current flow heat exchangers.

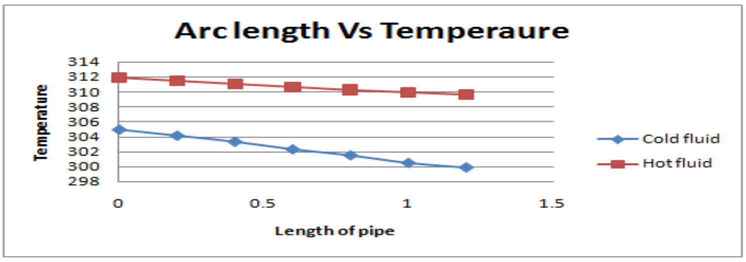


Fig 5.3.7 Counter Flow profile curve

5.3.4 Fluent Values vs. Experimental values for counter flow:

(a) Co-efficient of heat transfer (Q_A)

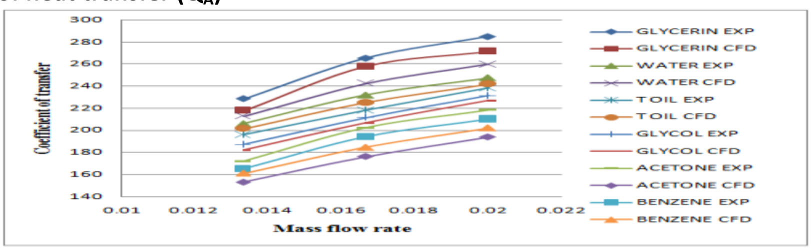


Fig 5.3.8 Co-efficient of heat transfer vs. Mass flow rate

The below figure 5.20 shows that co-efficient of heat transfer directly proportional to mass flow rate i.e. by increasing mass flow rate, the co-efficient of heat transfer increases.As shown in figure, Glycerin EXP acquired better co-efficient of heat transfer compared to other organic fluids on increasing mass flow rate.

(b) Overall heat transfer co-efficient (U_o):

The below figure 5.21 shows that heat transfer co-efficient is directly proportional to mass flow rate i.e. by increasing mass flow rate, the Overall heat transfer co-efficientincreases.

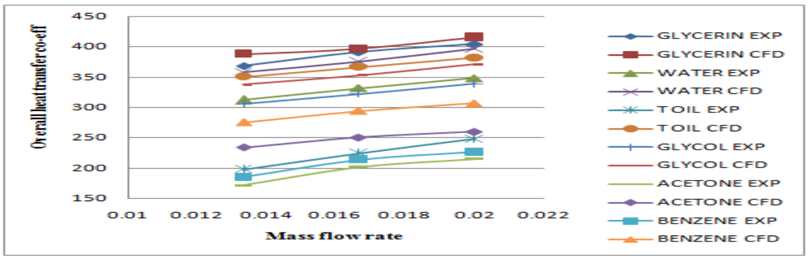


Fig 5.3.9 Overall heat transfer Co-efficient vs. Mass flow rate

As shown in figure, Glycerin CFD acquired better heat transfer co-efficient compared to other organic fluids on increasing mass flow rate.

(c)Log Mean Temperature Difference (LMTD):

The below figure 5.22 shows that LMTD is directly proportional to mass flow rate i.e. by increasing mass flow rate, LMTDincreases.

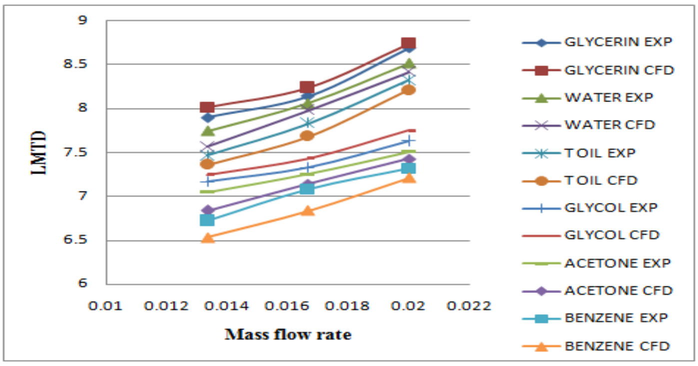


Fig 5.3.9.1 LMTD vs. Mass flow rate

As shown in figure, Glycerin CFD acquired better LMTD compared to other organic fluids on increasing mass flow rate.

CONCLUSION

At first, the main objective of this project is to create a validation between Experimental&CFD results which was performed using different organic solutions like Glycerine, Ethylene glycol, Acetone, Benzene and Transformer oil to check the percentage error, in order to affirm the experimental setup. As the percentage of error is within 6% the experimental setup is validated.

In Parallel flow arrangement, considering laminar flow. At a flow rate of 0.02 m/s by using Glycerine fluid acquired better heat transfer of 267.4689661 watts.

In counter flow arrangement, considering laminar flow. At a flow rate of 0.02 m/s by using Glycerine fluid acquired better heat transfer of 285.1023842 watts. Hence, Glycerine fluid in counter flow arrangement is better effective.

References

[1] Arya, K.S., "Analysis and Simplification of Three-Dimensional Space Vector PWM for Three-Phase Four-Leg Inverters," *IEEE Transactions on Industrial Electronics,* vol. 58, pp. 450-464, Feb 2011.

[1] Muhammad Mahmoud Salam Butta, Nasir Hayat et.al,CFD Applications In Various Heat Exchangers Design: A Review, Department Of Mechanical Engineering, University Of Engineering & Technology, Applied Thermal Engineering, 2011.

[2] J.S.Jayakumar, Helically Coiled Heat Exchangers Chemical Engineering Research and Design 221-232.

[3] S.M. Mahajani, J.C. Mandal, Experimental and CFD estimation of heat transfer in helically coiled heat exchangers. 2008, Chemical Engineering Research and Design 221-232.

[4] M. Chandra Sekhara Reddy and Veeredhi Vasudeva Rao, Experimental investigation of heat transfer coefficient and friction factor of ethylene glycol water based TiO2 Nano fluid in double pipe heat exchanger with and without helical coil inserts, International Communications in Heat and Mass Transfer, Volume 50, January 2014, Pages 68-76.

[5] Topakoglu, Steady laminar flows of an incompressible viscous fluid in curved pipes. Journal of Mathematics and Mechanics, Vol. 16(12):1321- 1337.

[6] Ender Ozden, Ilker Tari, Shell Side CFD Analysis of a Small Shell and Tube Heat Exchanger, Middle East Technical University, 2010.

[7] Prof. Naresh B. Dhamane et al,Heat Transfer Analysis of Helical Strip Insert with Regularly Spaced Cut Sections Placed inside a Circular Pipe. International Journal of Modern Engineering Research (IJMER) Vol. 2,

Issue. 5, Sep.-Oct. 2012 pp-3711- 3716.
[8] W. A. Aly, Numerical study on turbulent heat transfer and pressure drop of Nano fluid in coiled tube in-tube heat exchangers. Energy Conversion and Management, Volume 79, March 2014, Pages 304-316.
[9] Liu. L and et. al, The Potential Impact of Nano fluid Enhancements on the Performance of Heat Exchangers. Heat Transfer Engineering, Volume 33, Issue 1, 2012, pages 31-41.
[10] G. E. Kondhalkar & V. N. Kapatkat, Performance Analysis ofSpiral tube heat exchanger used in oil extraction system. International Journal of Modern Engineering Research, Vol.2, Issue.3, PP – 930 – 936, ISSN No.2249 – 6645, May – June2012.
[11] Su Thet Mon Than, Khin Aung Lin, Mi Sandar Mon, Heat Exchangerdesign. World Academy of Science, Engineering and Technology 462008.
[12] Lilijana Markovska et al, Optimal Design of Shell and Tube Heatexchanger. Bulletin of the chemist and technology of Macedonia, 1996,15, 39-44.