International Journal of
Scientific Engineering and Technology Research

ISSN 2319-8885
Vol.06,Issue.04
February-2017,
Pages:0638-0644

www.ijsetr.com

Comparative Analysis of Parallel and Counter Flow Heat Exchangers

KOLEPAKA SRIDHAR[1], K. BICHA[2]

[1]PG Scholar, Dept of Mechanical Engineering (Thermal Engineering), Mallareddy College of Engineering and Technology, Secunderabad, TS, India, E-mail: Kolepakasridhar@gmail.com.

[2]Associate Professor, Dept of Mechanical Engineering(Thermal Engineering), Mallareddy College of Engineering and Technology, Secunderabad, TS, India, E-mail: bichamech@gmail.com.

Abstract: The flow pattern through a heat exchanger affects the required heat exchanger surface. A counter flow heat exchanger needs the lowest heat transfer surface area. It gives a higher value for log mean TEMPERATURE difference than either a parallel flow heat exchanger or a cross flow heat exchanger. A heat exchanger can have several different flow patterns. Counter flow, parallel flow, and cross flow are common heat exchanger types. A counter flow heat exchanger is the most efficient flow pattern of the three. It leads to the lowest required heat exchanger surface area because the log mean TEMPERATURE drop is the highest for a counter flow heat exchanger. In this thesis analysis is done to compare the heat transfer rates between the two basic flow arrangements: (i) the unidirectional parallel flow (parallel flow or co-current flow), and (ii) bidirectional flow (counter flow or counter-current flow). CFD analysis and thermal analysis is done on the heat exchanger for different fluids, by taking hot water and refrigerants R134A, R22, R600A and different materials of heat exchangers. 3D models are done in Pro/Engineer and analysis is done in Ansys.

Keywords: CFD, Temperature, R134A, R22, R600A, Heat Exchangers.

I. INTRODUCTION

The technology of heating and cooling of systems is one of the most basic areas of mechanical engineering. Wherever steam is used, or wherever hot or cold fluids are required we will find a heat exchanger. They are used to heat and cool homes, offices, markets, shopping malls, cars, trucks, trailers, aero planes, and other transportation systems. They are used to process foods, paper, petroleum, and in many other industrial processes. They are found in superconductors, fusion power labs, spacecrafts, and advanced computer systems. The list of applications, in both low and high tech industries, is practically endless. Heat exchangers are typically classified according to flow arrangement and type of construction. In this introductory treatment, we will consider three types that are representative of a wide variety of exchangers used in industrial practice. The simplest heat exchanger is one for which the hot and cold fluids flow in the same or opposite directions in a concentric-tube (or double-pipe) construction. In the parallel-flow arrangement of Fig. 1.1a, the hot and cold fluids enter at the same end, flow in the same direction, and leave at the same end. In the counter flow arrangement, Fig. 1.1b, the fluids enter at opposite ends, flow in opposite directions, and leave at opposite ends. A common configuration for power plant and large industrial applications is the shell-and tube heat exchanger, shown in Fig. 1.1c.

This exchanger has one shell with multiple tubes, but the flow makes one pass through the shell. Baffles are usually installed to increase the convection coefficient of the shell side by inducing turbulence and a cross flow velocity component. The cross-flow heat exchanger, Fig. 1.1d, is constructed with a stack of thin plates bonded to a series of parallel tubes. The plates function as fins to enhance convection heat transfer and to ensure cross-flow over the tubes. Usually it is a gas that flows over the fin surfaces and the tubes, while a liquid fluid flows in the tube. Such exchangers are used for air-conditioner and refrigeration heat rejection applications. The concept of overall heat transfer resistance or coefficient that you were introduced earlier in your heat transfer course, if we apply this concept to a, for example, doublepipe heat exchanger, the total resistance is the sum of the individual components; i.e., resistance of the inside flow, the conduction resistance in the tube material, and the outside convective resistance, given by

$$R_{total} = \frac{1}{A_i h_i} + \frac{t}{kA_{\ln}} + \frac{1}{A_o h_o} \tag{1}$$

II. LITERATURE SURVEY

A. Graetz Problem

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.

B. Flux Boundary Condition

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 termswould 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.

II. INTRODUCTION TO SOFTARES

A. Abbreviations and Acronyms

- CAD Computer Aided Design
- FEM Finite Element Method
- CFD Computational Fluid Dynamics
- FEA Finite Element Analysis
- PRO.E Pro/Engineering
- ANSYS American Computer Aided Engineering Software

B. Units

- Use either SI (MKS) or CGS as primary units. (SI units are encouraged.) English units may be used as secondary units (in parentheses). An exception would be the use of English units as identifiers in trade, such as "3.5-inch disk drive."
- Avoid combining SI and CGS units, such as current in amperes and magnetic field in oersteds. This often leads to confusion because equations do not balance dimensionally. If you must use mixed units, clearly state the units for each quantity that you use in an equation.
- Do not mix complete spellings and abbreviations of units: "Wb/m2" or "webers per square meter," not "webers/m2." Spell units when they appear in text: "...a few henries," not "...a few H."
- Use a zero before decimal points: "0.25," not ".25." Use "cm3," not "cc." (bullet list)

C. Equations

Heat Transfer Considerations: The energy flow between hot and cold streams, with hot stream in the bigger diameter tube, is as shown in Fig.1. Heat transfer mode is by convection on the inside as well as outside of the inner tube and by conduction across the tube. Since the heat transfer occurs across the smaller tube, it is this internal surface which controls the heat transfer process. By convention, it is the outer surface, termed Ao, of this central tube which is referred to in describing heat exchanger area. Applying the principles of thermal resistance,

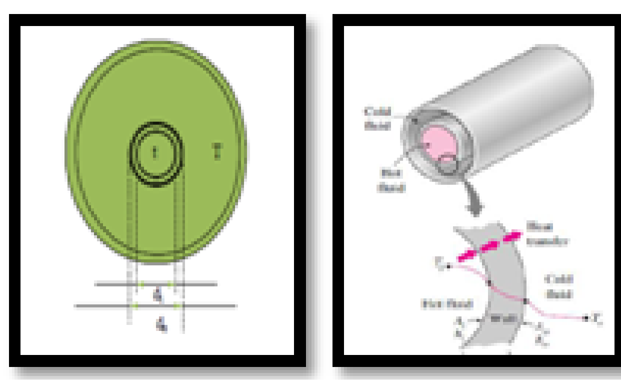

Fig.1.

End view of a tubular heat exchanger

$$R = \frac{1}{h_o A_o} + \frac{\ln\left(r_o / r_i\right)}{2\pi \cdot kl} + \frac{1}{h_i A_i} \quad \textbf{(1)}$$

If we define overall the heat transfer coefficient, Uc, as:

$$U_c \equiv \frac{1}{RA_o} \quad \textbf{(2)}$$

Substituting the value of the thermal resistance R yields:

$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{r_o \ln\left(r_o / r_i\right)}{k} + \frac{A_o}{h_i A_i} \quad \textbf{(3)}$$

Standard convective correlations are available in text books and handbooks for the convective coefficients, ho and hi. The thermal conductivity, k, corresponds to that for the material of the internal tube. To evaluate the thermal resistances, geometrical quantities (areas and radii) are determined from the internal tube dimensions available. Fouling Material deposits on the surfaces of the heat exchanger tubes may add more thermal resistances to heat transfer. Such deposits, which are detrimental to the heat exchange process, are known as fouling. Fouling can be caused by a variety of reasons and may significantly affect heat exchanger performance. With the addition of fouling resistance, the overall heat transfer coefficient, Uc, may be modified as:

$$\frac{1}{U_d} = \frac{1}{U_c} + R'' \quad \textbf{(4)}$$

where R" is the fouling resistance.

Fouling can be caused by the following sources:

- Scaling is the most common form of fouling and is associated with inverse solubility salts. Examples of such salts are $CaCO_3$, $CaSO_4$, $Ca_3(PO_4)_2$, $CaSiO_3$, $Ca(OH)_2$, $Mg(OH)_2$, $MgSiO_3$, Na_2SO_4, $LiSO_4$, and Li_2CO_3.
- Corrosion fouling is caused by chemical reaction of some fluid constituents with the heat exchanger tube material.
- Chemical reaction fouling involves chemical reactions in the process stream which results in deposition of material on the heat exchanger tubes. This commonly occurs in food processing industries.
- Freezing fouling is occurs when a portion of the hot stream is cooled to near the freezing point for one of its components. This commonly occurs in refineries where paraffin frequently solidifies from petroleum products at various stages in the refining process. , obstructing both flow and heat transfer.
- Biological fouling is common where untreated water from natural resources such as rivers and lakes is used as a coolant. Biological micro-organisms such as algae or other microbes can grow inside the heat exchanger and hinder heat transfer.
- Particulate fouling results from the presence of microscale sized particles in solution. When such particles accumulate on a heat exchanger surface they sometimes fuse and harden. Like scale these deposits are difficult to remove. With fouling, the expression for overall heat transfer coefficient becomes:

$$\boxed{\frac{1}{U_d} = \frac{1}{h_i \cdot \left(r_i / r_o\right)} + \frac{\ln\left(r_o / r_i\right)}{k} + \frac{1}{h_o} + R''} \quad \textbf{(5)}$$

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. This is especially beneficial in cases where the tubes are sensitive to fouling effects which are aggravated by high Temperature.

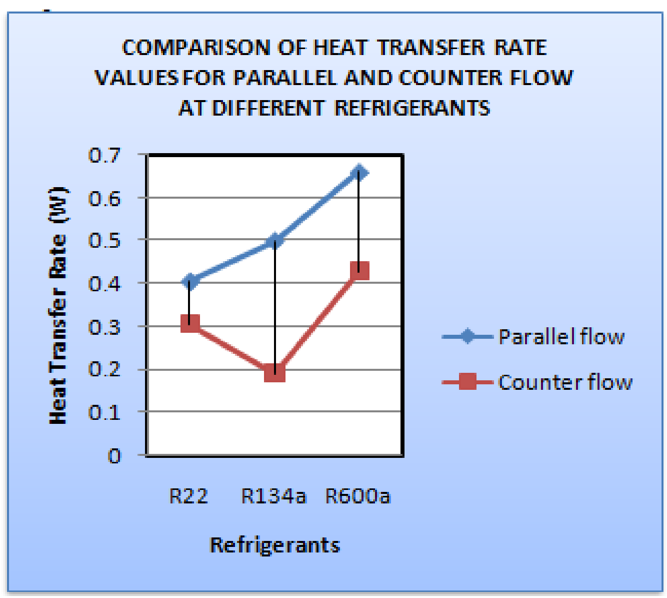

Fig.2.

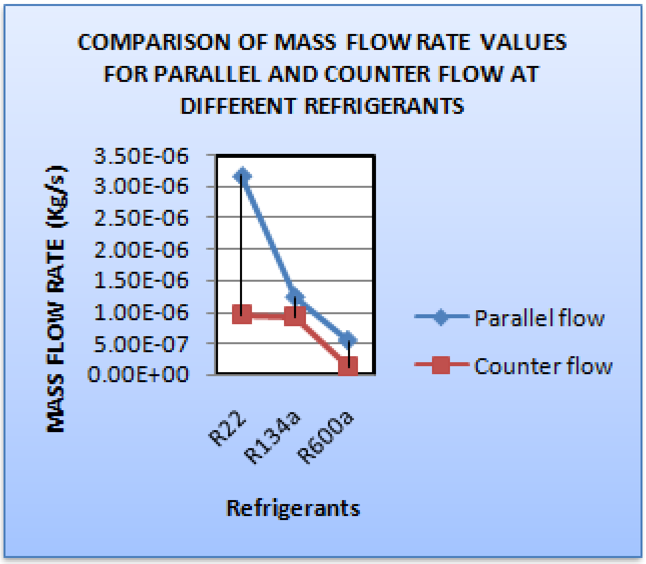

Fig.3.

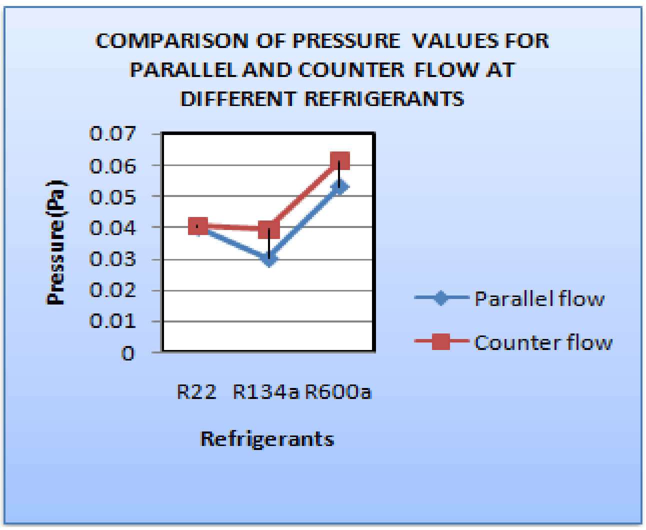

Fig.4.

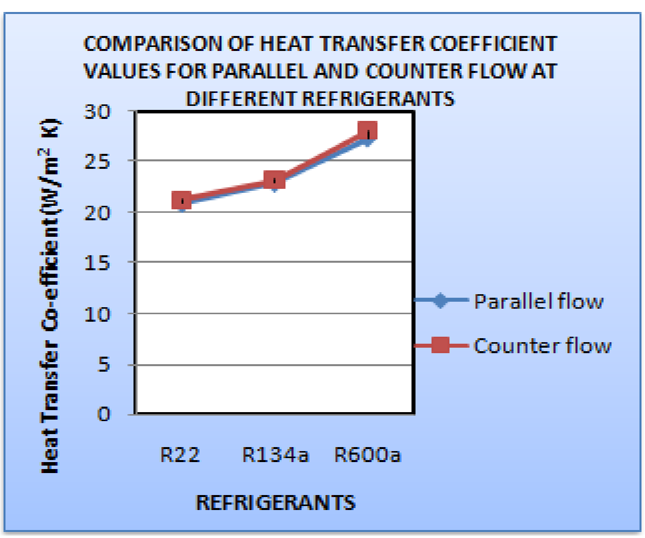

Fig.5.

III. CFD ANALYSIS PARALLEL FLOW HEAT EXCHANGER WITH HOT WATER AND R22 REFRIGERENT

Ansys → workbench→ select analysis system → fluid flow fluent → double click

→→Select geometry → right click → import geometry → select browse →open part → ok

→→ Select mesh on work bench → right click →edit → select mesh on left side part tree → right click → generate mesh →

Imported Geometry:

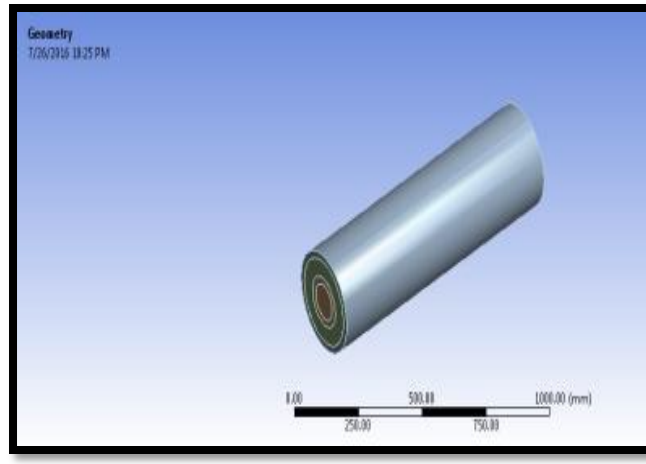

Fig.6.

Meshed Model:

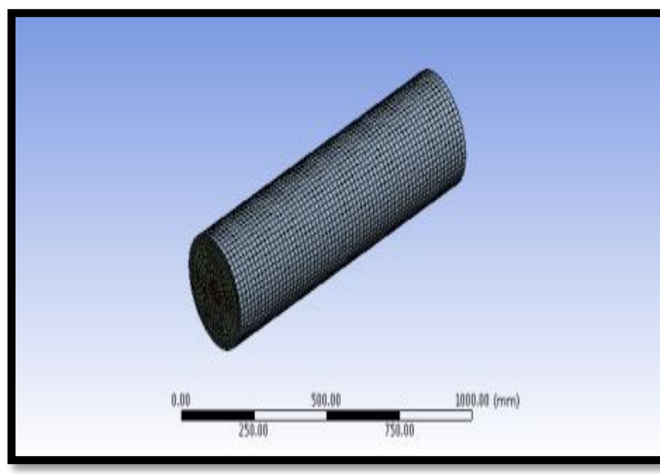

Fig.7.

Select faces → right click → create named section → enter name → cold fluid inlet

Select faces → right click → create named section → enter name → cold fluid outlet

Select faces → right click → create named section → enter name → hot water inlet

Select faces → right click → create named section → enter name →hot water outlet

Hot and Cold Inlet and Out Lets:

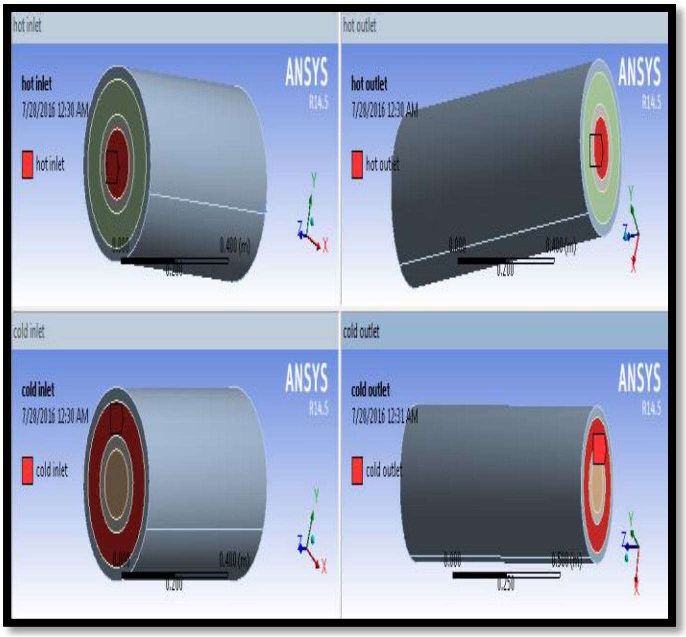

Fig.8.

Update project>setup>edit>model>select>energy equation (on)>ok

Materials> Materials > new >create or edit >specify fluid material or specify properties > ok

Select fluid.

R22 at 0^0 C:

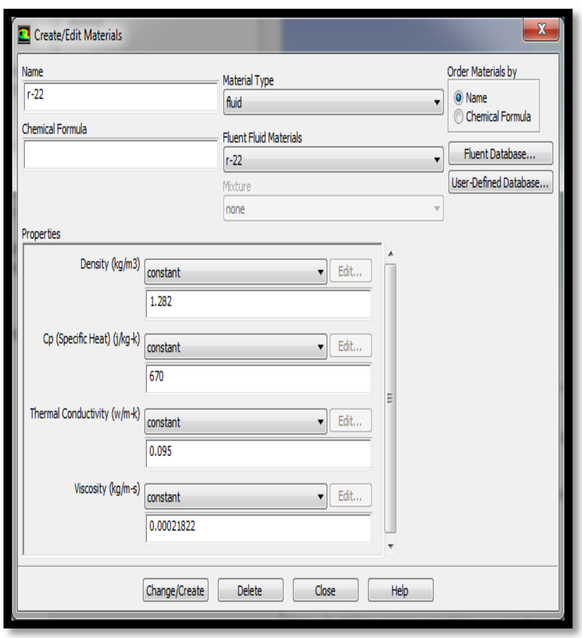

Fig.9.

TABLE I: Boundary Conditions

Inlet Temperatures(T)	273 K
Inlet pressure(P)	101325 Pa
Inlet velocity(V)	0.01 m/s

B. Hot Water at 35^0C

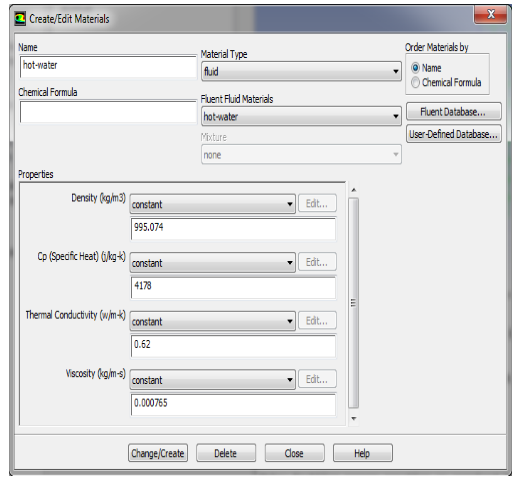

Fig.10.

TABLE II: Boundary Conditions

Inlet Temperatures(T)	305 K
Inlet pressure(P)	101325 Pa
Inlet velocity(V)	0.01 m/s

Solution > Solution Initialization > Hybrid Initialization >done

Run calculations > no of iterations = 10> calculate > calculation complete>ok

Results>edit>select contours>ok>select location (inlet, outlet, wall.etc)>select pressure>apply

A. Headings

Types of Heat Exchangers – concentric tube:

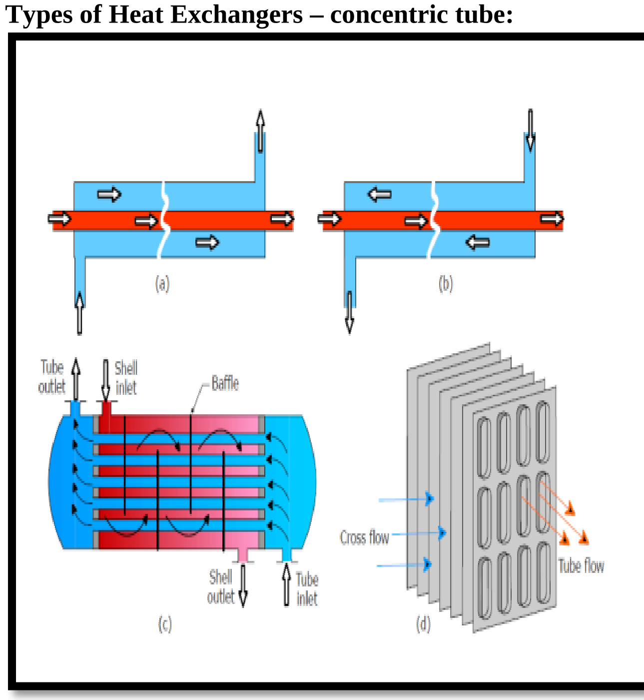

Fig.11.

Basic Heat Exchanger Flow Arrangements:

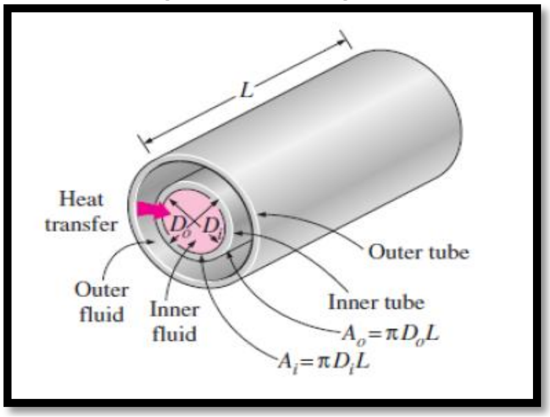

Fig.12.

- Log Mean Temperature Differences
- Comparison of Heat Exchangers

B. Component Headings

Cfd Analysis Parallel Flow Heat Exchangerwith Hot Water And R22 Refrigerent:

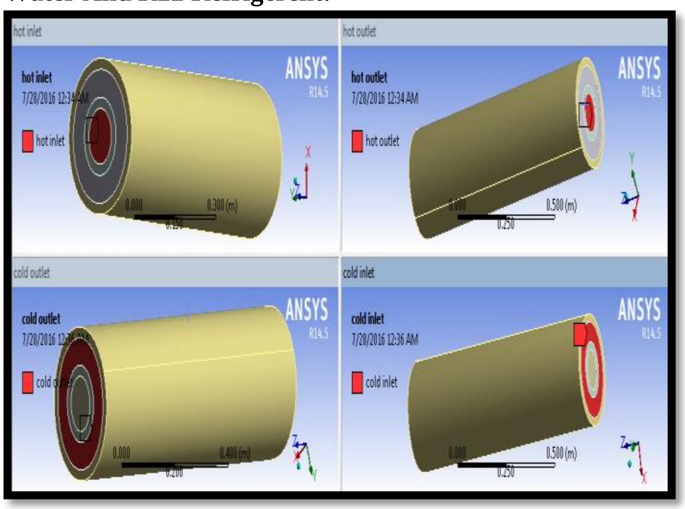

Fig.13.

With Hot Water And R134A Refrigerant:

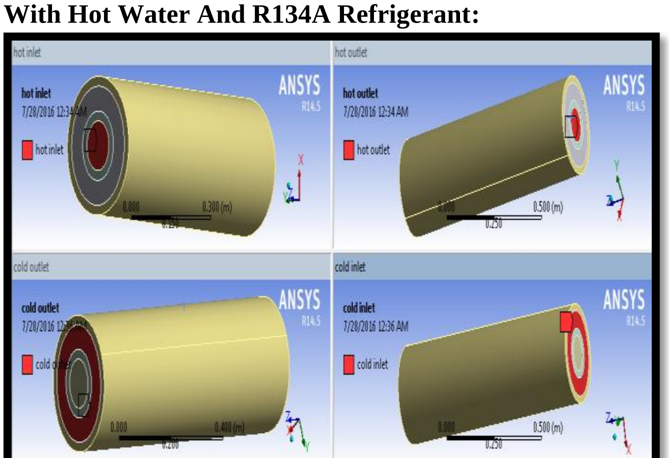

Fig.14.

- With Hot Water and R600A Refrigerant.

C. Figures and Tables

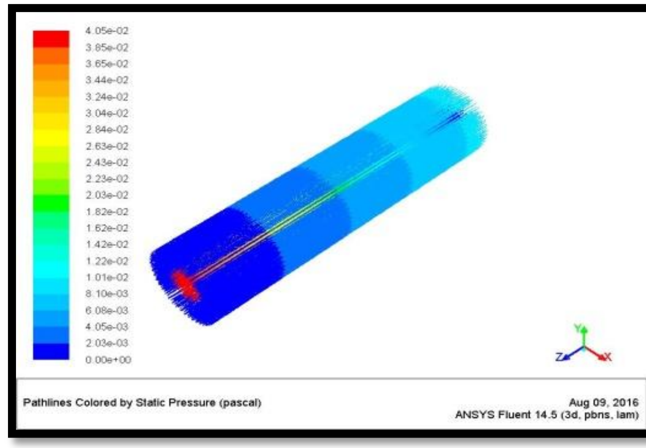

Fig. 15. With hot water and r22 refrigerent Pressure.

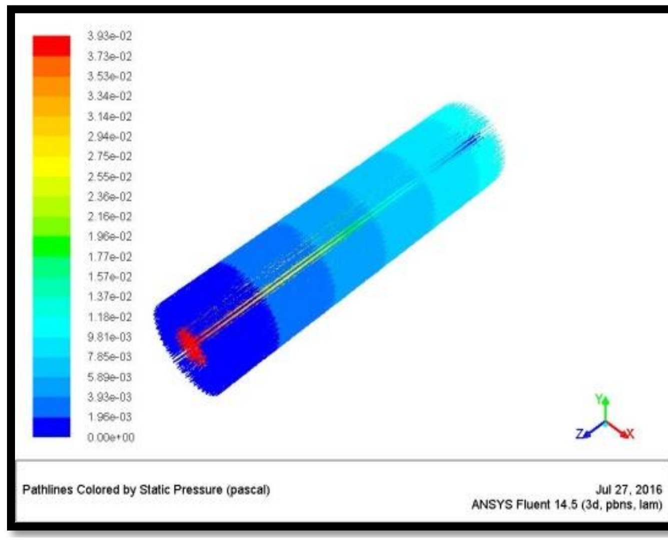

Fig. 16 .with hot water and r134a refrigerant Pressure.

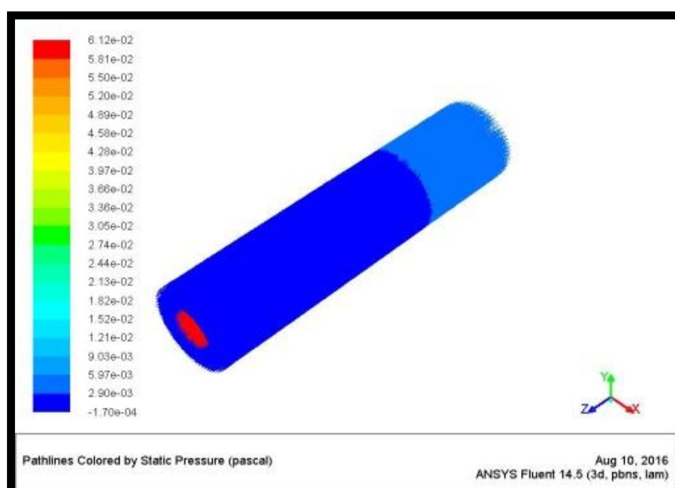

Fig.17. With Hot Water And R600a Refrigerant Pressure.

TABLE III: THERMAL ANALYSIS
Boundary Conditions: Convection (W/m^2-K)

Parallel flow			Counter flow		
R22	R134a	R600a	R22	R134a	R600a
20.9	22.9	27.3	21.3	23.2	28.1

TABLE IV: CFD Analysis Hot Water With Refrigerants Parallel Flow

Refrigerants	Pressure (Pa)	Velocity (m/s)	H.T.Co (W/m2 K)	Heat transfer rate (w)	Mass flow rate(kg/s)
R22	0.04	0.0170	20.9	0.406	3.169e^{-6}
R134A	0.03	0.0173	22.9	0.5	1.24e^{-6}
R600A	0.053	0.0184	27.32	0.66	0.545e^{-6}

TABLE V: Counter Flow

Refrigerants	Pressure (Pa)	Velocity (m/s)	H.T.Co (W/m2 K)	Heat transfer rate (w)	Mass flow rate(kg/s)
R22	0.0143	0.0112	21.3	0.306	0.933e^{-3}
R134A	0.0393	0.0134	23.2	0.192	0.916e^{-3}
R600A	0.0612	0.0348	28.1	0.43	0.125e^{-3}

IV. CONCLUSION

In this thesis analysis is done to compare the heat transfer rates between the two basic flow arrangements: (i) the unidirectional parallel flow (parallel flow or co-current flow), and (ii) bidirectional flow (counter flow or counter-current flow). CFD analysis is done on two types of heat exchangers by taking hot water and R134A, R22, R600A as refrigerants. By observing CFD analysis results, the heat transfer coefficient is more for counter flow and heat transfer rate is less for counter flow heat exchanger due to more area than parallel flow. By comparing the results between fluids, the heat transfer coefficient and heat transfer rate are more for R600A. Thermal analysis is done on the heat exchanger for different fluids, R134A, R22, RaA nd different materials Aluminum and Copper. By observing the results, the HEAT FLUX is more for counter flow heat exchanger than parallel flow heat exchanger. The maximum HEAT FLUX value is obtained for counter flow heat exchanger when R600A is used as refrigerant and Copper is used as material.

V. ACKNOWLEDGMENT

The authors would like to thank the Principal and Management of MallaReddy college of Engineering and technology, Hyderabad for their constant support during this project. I would also like to thank all the supporting staff of the Dept. of Mechanical Engineering and all other departments who have been helpful directly or indirectly in making the project a success.

VI. REFERENCES

[1] Concentric tube heat exchanger: operating principle with parallel flow. Art.
[2] White, Frank. Viscous Fluid Flow. 3rd ed. New York: The McGraw-Hill Companies, Inc., 2006.
[3] Conley, Nancy, Adeniyi Lawal, and Arun B. Mujumdar. ―An Assessment of the Accuracy of Numerical Solutions to the Graetz Problem.‖ Int. Comm. Heat Mass Transfer. Vol.12. Pergamon Press Ltd. 1985.
[4] Kays, William, Michael Crawford, and Bernhard Weigand. Convective Heat and Mass Transfer. 4th ed. New York: The McGraw-Hill Companies, Inc.,2005.
[5] Sellars J., M. Tribus, and J. Klein. ―Heat Transfer to Laminar Flow in a Round Tube or Flat Conduit—The Graetz Problem Extended.‖ The American Society of Mechanical Engineers. Paper No. 55-SA-66 AD-A280 848. New York. 1955.
[6] Subramanian, Shankar R. ―The Graetz Problem.‖ Web. 12 Apr. 2012.
[7] K.S.Hwang, S.K.Jang, S.U.S.Chio, Flow and convective heat transfer characteristics of water-based Al2O3 nanofluids in fully developed laminar flow regime, International Journal of Heat and Mass Transfer, 52 (2009) 193-199.
[8] V. Bianco, F. Chiacchio, O. Manca, S. Nardini, Numerical investigationof nanofluids forced convection in circular tubes, Applied Thermal Engineering, 29 (2009) 3632 – 3642.
[9] M. Akbari, N. Galanis, A. Behzadmehr, Comparative analysis of singleand two-phase models for CFD studies of nanofluid heat transfer, International Journal of Thermal Sciences, 50 (2011) 1343 – 1354.
[10] Valko, Peter P. ―Solution of the Graetz-Brinkman Problem with the Laplace Transform Galerkin Method. International Journal of Heat and Mass Transfer 48. 2005.
[11] Blackwell, B.F. ―Numerical Results for the Solution of the Graetz Problem for a Bingham Plastic in Laminar Tube Flow with Constant Wall Temperature.‖ Sandia Report. Aug. 1984. Encyclopædia Britannica Online. Web. 12 Apr. 2012.
[12] White, F.M., Fluid Mechanics, 3rd edition. Mc- Graw Hill, 1994.