Pressure drop analysis of Condenser using refrigerants R-134a, R-407C

[#1]Sachin N.Jadhav, [#2]Dr. Pradeep A.Patil

[#12]Department of Mechanical Engineering, Savitribai Phule University of Pune, Pune, India.

ABSTRACT— This paper presents experimental study of effect of Pressure Drop on a shell side in a Condenser. Their are Two different Refrigerants are used i.e.R-134a and R-407C. Then all this test are conducted in a Test rig set-up. In this paper are performed for various condensing temperatures and evaporating temperatures. The test runs are performed for each average saturated condensing temperatures ranging from 38.6 °C to 45.6 °C for which evaporating temperatures are varied. The experimental results were studied for effect of mass flow rate on shell side pressure drop and friction factor. It indicates that the frictional pressure drop increases whereas friction factor decreases with increase in mass flow rate. It is observed that frictional pressure drop for R-134a is more than R-407C.If allowable pressure drop range is 0.8 Pa.

Keywords— STHE, Pressure drop, Frictional factors, Mass flow rate, condensing temperatures and evaporating temperatures.

I. INTRODUCTION

Heat exchanger is a device to transfer the thermal energy ie. enthalpy between two or more fluids, between a solid surface and solid particles and a fluid at different temperatures'
In thermal contact. In a heat exchangers there are no external heat and work interactions .Heat exchanger is a wide variety of applications such as Air-conditioning, Petrochemical industries, space applications, manufacturing industries, milk pasteurization industries, waste heat recovery, power generations etc. In most heat exchangers the heat transfer between fluid takes place through a separating wall. Such heat exchangers rate the fluids are by heat transfer area , and they do not mix or leak such heat exchangers are direct transfer type. Exchangers in which there is a intermittent heat exchangers between the hot and cold fluids via thermal energy storage and release through the exchanger surface.
The heat exchanger are broadly classified as : according to transfer process ,according to number of fluids, surface compactness, according to construction, flow arrangements, heat transfer mechanism. But in this paper only studied the shell and tube heat exchanger. It is a type of tubular heat exchanger. Tubular heat exchangers can be designed for high pressure Tubular exchangers can be designed
for high pressures relative to the environment and high-pressure differences between the fluids. Tubular exchangers are used primarily for liquid-to-liquid and liquid-to-phase change (condensing or evaporating) heat transfer applications. They are used for gas-to-liquid and gas-to-gas heat transfer applications primarily when the operating temperature and pressure is very high.

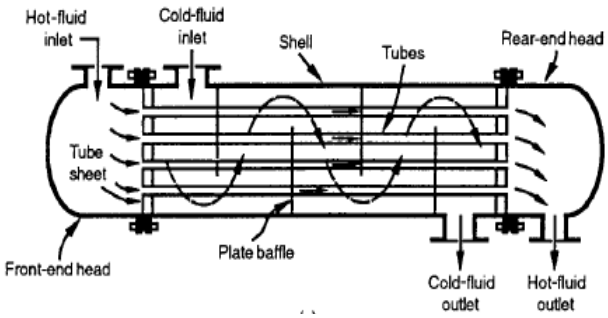

Fig.No.1 Shell and tube Heat Exchanger.

In above fig. shows that a Tubular type shell and tube heat exchanger. In this exchanger no. of tubes mounted in a cylindrical shell. One fluid flows inside the tubes and other flows across the means in shell side. the major components of this exchangers are tube sheets , plate baffle spacing, tubes or tubes bundle , shell, front end head, rear end head. The internal constructions is depending on the pressure drop and heat transfer ,thermal stresses, to control corrosion, to containing operating temperatures and pressures, to prevent leakages.
In shell and tube heat exchangers classified widely used in a TEMA standards.(Tubular Exchanger Manufacturers association). TEMA standards specify the manufacturing tolerances for various mechanical classes ,the range of tube sizes and pitches, baffle space and plates, tube sheet thickness formula, pressure classification and so on. widely used in industries because operating

condition such as from high vacuum to ultra high pressure(100Mpa).It can be designed for special operating conditions :erosion, corrosion, toxicity, vibration, highly viscous fluid, multi component mixtures, density of fluid, flow of fluid.

II. LITERATURE REVIEW

Heat transfer coefficient and pressure drop on the shell side of a shell-and-tube heat exchanger] have been experimentally obtained for three different types of copper tubes (smooth, corrugated and with micro-fins). Also, experimental data has been compared with theoretical data available. Correlations have been suggested for both pressure drop and Nusselt number for the three tube types. A shell-and-tube heat exchanger of an oil cooler used in a power transformer has been modeled and built for this experimental work in order to investigate the effect of surface configuration on the shell side heat transfer as well as the pressure drop of the three types of tube bundles.[R. Hosseini , A. Hosseini-Ghaffar and M. Soltan Vol.27,April 2007]

They have presented a compact formulation to relate the shell-side pressure drop with the exchanger area and the film coefficient based on the full Bell–Delaware method. In addition to the derivation of the shell side compact expression, they have developed a compact pressure drop equation for the tube-side stream, which accounts for both straight pressure drops and return losses. They have shown how the compact formulations can be used within an efficient design algorithm. They have found a satisfactory performance of the proposed algorithms over the entire geometry range of single phase, shell and tube heat exchangers. 3.Saunders in his book proposed practical method and simple design factors are provided and the method is used rapidly for fixed set of geometrical parameters. In his work the correction factors are for heat transfer and pressure drop correlations.[M.Serna and Hydro-fluorocarbon (HFC) refrigerants, like R134a and zero-tropic refrigerant mixtures like R407C have been introduced in order to replace R12 and R22, respectively. As per the agreement done in The Montreal Protocol, efforts towards the replacement of these refrigerants are under progress. For the optimum performance, an accurate design technique is essential for the prediction of refrigerant pressure-drops and flow-patterns through the evaporator, condenser and other heat exchangers (Smith et al., 2001).

Experimental setup :

Fig.1 Experimental test rig

The actual photo of set up is shown in above fig. The test rig was vapor compression cycle along with some additional components and instruments to maintain desired operating condition to measure the parameters at various location for study . The setup used for the experiments is a Compressor-Calorimeter test rig, as shown in fig. 1, which can also be used to analyze Vapor Compression Cycle (VCC). The refrigerants are charged at the suction port of the rig. Once the compressor starts, it pumps up the refrigerant to the higher pressure. Oil Separator is provided to separates the oil that is probably carried along with refrigerants during the compression stroke. The outlet of the compressor has two thermocouples to measure the compressor top shell temperature and discharge temperature of the compressor From the oil separator the refrigerant passes to the shell and tube condenser, where in the cooling water supplied by the cooling tower at the ambient condition exchanges heat in the shell and tube frame work of the condenser. The refrigerant coming out of the condenser expected to be in the saturated state is made to flow in the sub cooler via liquid temperature controller where the liquid is further condensed to ensure the liquid state before it is expensed in the expansion valve.

The liquid (sub cooled) phase refrigerant passes on to the drier, to get wiped out of any moisture content present within. The receiver following next to the drier is just a storage unit where in it stores the refrigerant. Mass flow meter provided measures the amount of refrigerant that is been used by the compressor for particular given condition.

Then the refrigerant is passed to the evaporator, where actually the cycle is theoretically believed as the starting point. The evaporator cabin (Sealed tank) as shown in fig.2 incorporates evaporator coil, stirrer and glycol heater. The glycol mixture is used as anti-freezing agent in the cabin to see to that the temperature does fall beyond the set value during the experimentation.

There are 7 temperature sensors fixed in the circuit lines to measure the temperature condition of the refrigerant and the cooling water that runs in the system during cycle operation. The suction pressure and the discharge pressure of the compressor is been regulated by the pressure switch, the magnitudes of the same is measured by the transducer fixed at the inlet and outlet of the compressor. There are two more pressure transducer to sense the pressure in water supply lines to the condenser and sub cooler.

A proportional-integral-derivative controller (PID controller) is a control loop feedback mechanism (controller) used in this system 3 in numbers that is been used to set the discharge pressure of the compressor, desired sub cooling temperature of the refrigerant and the suction temperature of the refrigerant before it enters the compressor.R134a,R407C are charged in the system one by one. for each condensing temperature, evaporating temperature is varied.

Water cooled condenser specification:

Cosmic Refrigeration:

Sr No.	Parts Name	Specification
1	Length of tube	566mm
2	Length of shell	570mm
3	Shell inner diameter	105mm
4	Shell outer diameter	115mm
5	Shell wall thickness	5mm
6	Number of tubes	10No's
7	Tube inner diameter	16mm
8	Tube outer diameter	19.05mm

Methodology:

Analysis:

1.In a Test-Rig panel take a observation for temperature and pressure for using refrigerant R-134a,R22,R404A,R407C(Hydrocarbon Blends).

2.Take readings for Compressor suction temperature, discharge temperature ,Suction pressure, Evaporator in pressure, Condenser in temp. ,Refrigerant mass flow rate, Glycol temp.

3.Using Refrigerant chart(p-h),REFPROP and cool-pack software for refrigerant R22,R134a,R404A,R407C obtain the values of enthalpies, density and viscosity.

4.Calucalation COP, Work done, and Refrigerant effect

5.Determination of Pressure drop, Mass flow rate, friction factor for different loads for R22,R134a,R404A,R407C.

The pressure drop experienced by the shell-side fluid is calculated by;

$$\Delta p_s = \frac{f_s G_s^2 (N_b + 1) D_s}{2 \rho_s D_e \varphi_s} \quad (1)$$

Where ,

Gs=Shell side mass velocity(M_s/A_s)

ϕ_s=Variable correction factor (μ_b/μ_w)

μ_b=Is the viscosity of the shell-side fluid at . bulk temperature .

μ_w= is the viscosity of the tube-side fluid at wall temperature.

Fs=Shell side friction factor

$$f_s = \exp(0.576 - 0.19 \ln \mathrm{Re}_s)$$

Re_s=Shell side Reynolds No.

$=G_s D_e/\mu$ $400<Re_s = G_s D_e/\mu<1*10^6$

Reynolds number for the shell-side is based on the equivalent diameter and the velocity based on a reference flow:

$$\mathrm{Re}_s = \frac{\dot{m}_s}{A_s} \times \frac{D_e}{\mu_s}$$

Shell Side Area(A_s):

$$A_s = \frac{D_s}{P_T} \times (P_T - d_o) \times B$$

Equivalent diameter for Triangular layout:

$$D_{e-triangular} = \frac{4A_{flow}}{P_e} = \frac{4\left\{\frac{\sqrt{3}P_T^2}{4} - \frac{\pi}{8}d_O^2\right\}}{\pi d_O / 2}$$

Free Flow Area(A_{flow}):

$$A_{flow} = P_T^2 - 4\left(\frac{1}{4}\left\{\frac{\pi}{4}d_O^2\right\}\right) = P_T^2 - \frac{\pi}{4}d_O^2$$

6.Ploting of graphs:

a)Pressure drop Vs Mass flow rate
b)Friction factor Vs Mass flow rate
c)Temp. ratio Vs Mass flow rate
d) Temp. ratio Vs Pressure drop.

7.Then Comparison for the performance by the results.

Results and Discussion :

Present experiment was performed for various condensing temperatures ranging from 38.6 °C to 45.6 °C on the test apparatus shown in fig. 1. For each condensing temperature evaporating temperature was varied from 2 °C to 10 °C and corresponding readings of total pressure drop, mass flow rate and refrigerant temperature after sub-cooling were noted down. The values of the $\Delta Ptot$ observed from the experiment were used to calculate frictional pressure drop, momentum pressure drop and friction factor for refrigerants R-134a, R-407C using equations (1). The graphs were plotted to check the trend of variation of pressure drop and friction factor with respect to the variation of mass flow rate. From calculations done it was observed that frictional pressure drop values were much higher than Total shell side pressure drop values. Hence the frictional pressure drop plays a vital role in design of condenser. Physical properties such as density, surface tension, kinematic viscosity and pressure have significant effect on shell side pressure drop.

1)Variation in different mass flow rate Vs Pressure drop for R134a:

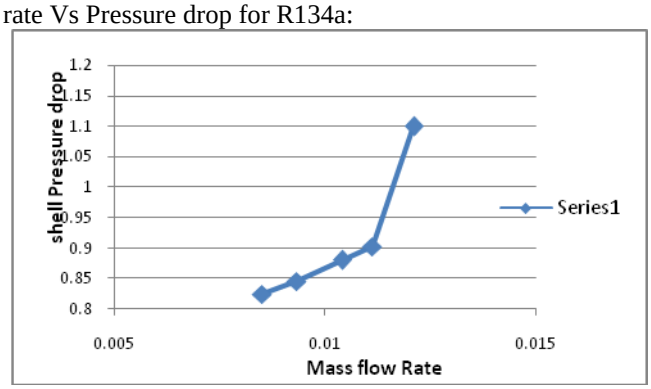

Fig.2 Variation in different mass flow rate Vs Pressure drop for R134a

Fig. 2 shows variation of shell side pressure drop with respect to mass flow rate for R-134a refrigerant. It can be clearly observed that the pressure drop is strongly affected by mass flow rate. As the flow velocity increases with increase in mass flow rate, there is increase in shell side pressure drop .

2)Variation in different mass flow rate Vs frictional factor for R134a:

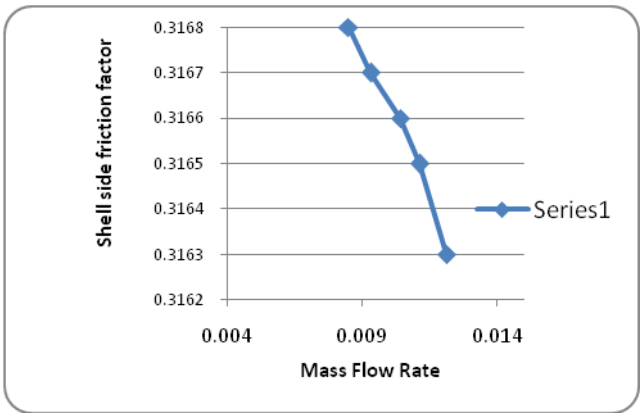

Fig.3 Variation in different mass flow rate Vs frictional factor for R134a

Fig. 3 shows variation of shell side friction factor with respect to mass flow rate for R-134a refrigerant. It can be clearly observed that the friction factor is strongly affected by mass flow rate. As the flow velocity increases with increase in mass flow rate, there is decrease in mass flow rate.

3)Variation in different mass flow rate Vs Pressure drop for R407C**:**

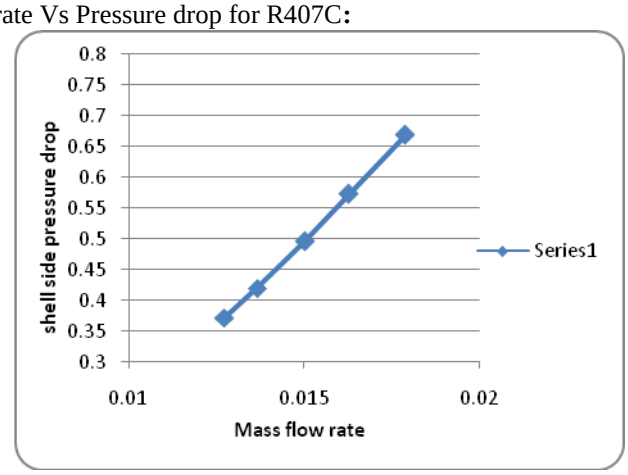

Fig.4 Variation in different mass flow rate Vs Pressure drop for R407C

Fig. 4 shows variation of shell side pressure drop with respect to mass flow rate for R-407C refrigerant. It can be clearly observed that the pressure drop is strongly affected by mass flow rate. As the flow velocity increases with increase in mass flow rate, there is increase in shell side pressure drop

4)Variation in different mass flow rate Vs frictional factor for R407C:

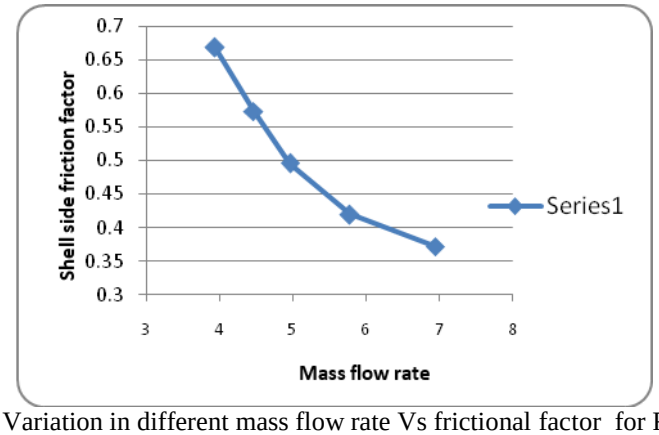

Fig.5 Variation in different mass flow rate Vs frictional factor for R407C

Fig. 5 shows variation of shell side friction factor with respect to mass flow rate for R-407C refrigerant. It can be clearly observed that the friction factor is strongly affected by mass flow rate. As the flow velocity increases with increase in mass flow rate, there is decrease in mass flow rate.

5)Variation in different mass flow rate Vs Temp. .Ratio(T_d/T_s) for R407C.

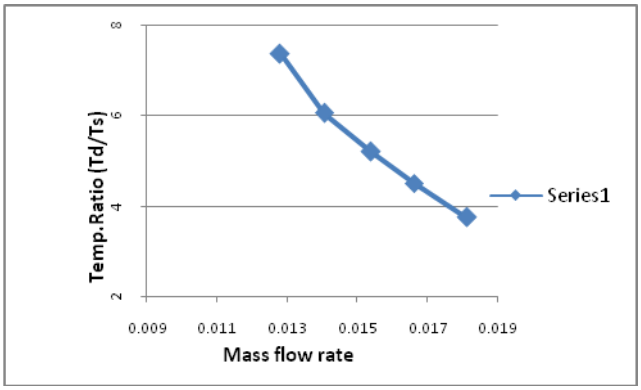

Fig.6 Variation in different mass flow rate Vs Temp. .Ratio(T_d/T_s) for R407C.

Fig. 6 shows variation of Temperatures with respect to mass flow rate for R-407C refrigerant. It can be clearly observed that the Temperature is strongly affected by mass flow rate. As the flow velocity increases with increase in mass flow rate, there is decrease in Temperature.

6)Variation in different mass flow rate Vs Temp.Ratio(Td/Ts) for R134a:

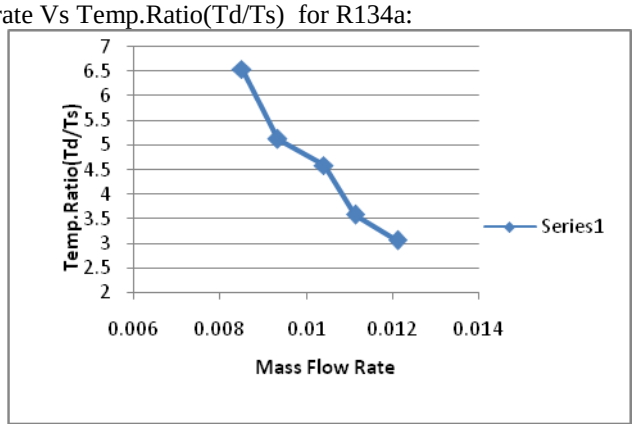

Fig.7 Variation in different mass flow rate Vs Temp.Ratio(Td/Ts) for R134a.

Fig. 7 shows variation of Temperatures with respect to mass flow rate for R-134a refrigerant. It can be clearly observed that the Temperature is strongly affected by mass flow rate. As the flow velocity increases with increase in mass flow rate, there is decrease in Temperature.

7)Comparative analysis for variation of Temperature with mass flow rate.

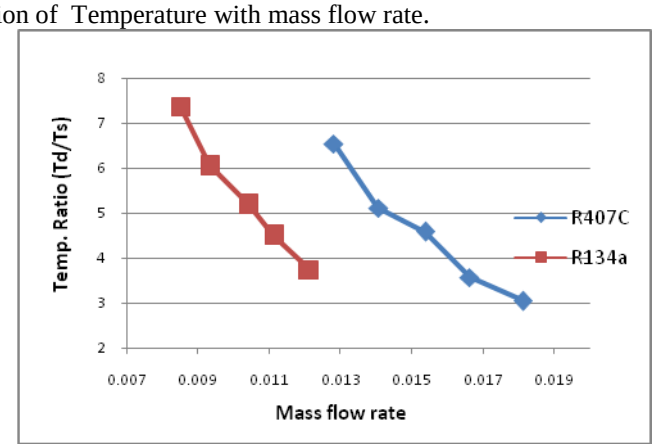

Fig.8 Comparative analysis for variation of Temperature with mass flow rate.

Fig. 6 and fig. 7 a show the variation of Temperature with mass flow rate for R-134a and R-407C respectively. As the mass flow rate increases there is decrease in Temperature. The same trend is observed for all the refrigerants. This decrease is 0.2 % to 3.5 % , and 0.5 % to 1.8% with respect to mass flow rate for R-134a, and R-407C respectively

8) Comparative analysis for variation of shell side friction factor with mass flow rate.

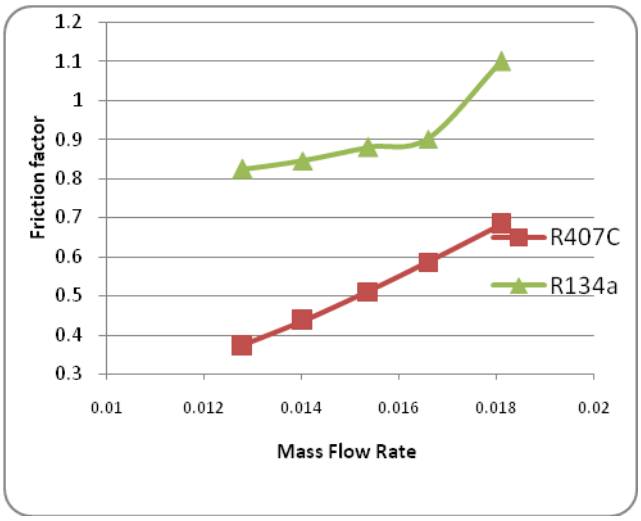

Fig.6 Comparative analysis for variation of shell side friction factor with mass flow rate.

Fig. 2 and fig. 5 a show the variation of friction factor with mass flow rate for R-134a and R-407C respectively. As the mass flow increases there is 0.2 % to 3.5 % , and 0.5 % to 1.8% decrease in friction factor. The same trend is observed for all the refrigerants. This decrease is with respect to
mass flow rate for R-134a, and R-407C respectively.

9)Comparative analysis for variation of shell side pressure drop with mass flow rate

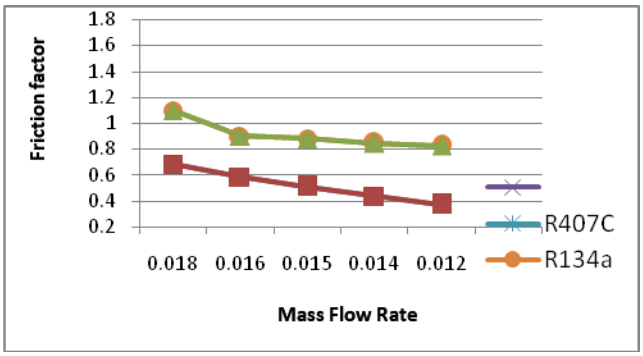

Fig.10 Comparative analysis for variation of shell side pressure drop with mass flow rate.

Fig.10shows the comparative frictional pressure drop analysis with respect to mass floe rate for all the three refrigerants. Only frictional pressure drop is considered for analysis as it accounts major part of total pressure drop. It can be seen that R-134a, R-407C is the increasing order of frictional pressure drop with respect to mass flow rate.

III. CONCLUSION

Pressure drop and friction factor for refrigerants R-134a and R-407C were investigated across Condenser. Effect of mass flow rate on frictional pressure drop and friction factor was found out for each refrigerant and even comparative analysis is also done. Pressure drop is higher for higher mass flow rate where as friction factor is lowest for higher mass flow rate. And lower temperature where as higher mass flow rate.Frictional pressure drop values play significant role for design purpose as it accounts major part of total pressure drop. The allowable pressure drop is 0.3-0.8 bar.
Frictional pressure drop for R-407c is larger than R-134a .

Friction factor R-134a is larger than R-407C s. but R-134a is not environment friendly refrigerant. Hence R-407c can be considered as best amongst these two refrigerants from the experiment performed.
Friction factor correlation can also be developed from current data and can exist as future scope of the same experiment.

IV. REFERENCES

1.R. Hosseini, A. Hosseini-Ghaffar and M. Soltan " April 2007 Experimental determination of shell side heat transfer coefficient and pressure drop for an oil cooler shell-and-tube heat exchanger with three different tube bundles", Applied thermal Engineering ,Volume 27.
2.M.Serna and A.Jimenez, “A compact formulation of the Bell Delaware method for Heat Exchanger design and optimization”, Chemical Engineering Research and Design.83(A5): 539–550.
3.Ajit kumar M.S[1],Ganesha T[2],M.C.Math[3]vol.2,No7, “ CFD Analysis to study the effects of inclined Baffles on fluid flow in a shell and tube heat exchangers” Dep. Of thermal power engineering VTU PG Centre,Mysore.

4.K Anand, Jan2014 Experimental Investigation of shell and tube heat exchanger using Bell Delaware method. Thermal power Engg. Dept.of Mech.Engg.PDACE Gulbarga vol-2 jan2014.
5. Saunders, 2008. Heat Exchangers, John Wiley & Sons, New York. Vol. 12 .
6**.** S. Noie Baghban, M. Moghiman and E. Salehi, 1990
"Thermal Analysis of shell side flow of shell and tube
type heat exchanger using experimental and theoretical methods ", Institute for Scientific Research, University of Guanajuato Lascurain de Retana No. 5 Guanajuato, Gto, México rence, and 36 pp. 256-262
7.C. Aprea, A. Greco, A. Rosato, Comparison of R407C and R417A heat transfer coefficients and pressure drops during flow boiling in a horizontal smooth tube, *Energy Conversion and Management*, 1629–1639.

NOMENCLATURE

P_S=Shell side pressure drop.(P_a)

F_S=Shell side friction factor.

N_b=Number of baffles.

ϕ_s=Variable property correction

$F_{s=}$Shell side friction factor.

ρ_s=Shell side density(K_g/m^3)

$\mu_{b=}$viscosity of the shell-side fluid at bulk temperature .(K)

$\mu_{w=}$ viscosity of the tube-side fluid at wall temperature.(K)

A_s= Shell Side Area(m)

A_{flow}=Area of flow.(m)

P_e=Equivalent Perimeter.(m)

P_T=Tube Pitch.(m)

M_s=shell side mass flow rate($K_g/m^2.s$)

T=Temperature (k)

