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Design and Analysis of Air-Cooled Fin and Tube Heat Exchanger with Smaller Diameter Micro Finned Tubes using R32 in Replacement of R410A

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Abstract: ***Difluoromethane (HFC32) is the perfect replacement of R410A due to its zero ozone depletion Potential and lower Global warming potential (GWP as 675) that is much less than R410A (2088) and Zero Ozone Depletion Potential. R32 refrigerant can achieve higher heat transfer coefficients with less quantity of refrigerant charge when compared to R410A. Fin and Tube heat exchangers (FTHE) are widely used in the refrigeration, air conditioning industries and in many other applications to exchange or transfer the heat from refrigerant or working fluid and to the sink. The aim of this paper is to calculate the Heat transfer coefficient, pressure drop and heat load of refrigerants in Air-cooled Fin and Tube heat exchanger. Here FTHE is used as a condenser in one TR residential air conditioning application and their comparison using R32 and R410A refrigerants. To study the behaviour of two refrigerants in liquid phase, two phase (liquid and vapor phase) and vapor phases inside the condenser. Here the airflow to the condenser is counter flow. Materials used were Aluminium for fins and copper for tubes to achieve greater heat transfer coefficient. Here fin and tube material combination is very important because of their material properties. Optimizing the design of FTHE, i.e. selecting the micro finned tubes to generate turbulence in refrigerant flow, which results in enhancement of heat transfer coefficient. Slit type fin is selected for fins. The micro finned copper tubes with smaller inner diameter can save the material cost. Coil Designer a simulation software used for the design and analysis of FTHE.***

Keywords: Tube fin heat exchanger, Micro finned Copper tube, Airflow rate, Mass flow rate, Coil Designer, R32 and R410A.

I. INTRODUCTION

In basic refrigeration cycle (VCRS), it has compressor, condenser, expansion device and evaporator. Condenser is one of the most important component of the cycle. In this paper, the study is on condenser. R32 is a refrigerant based on its properties better than R410A. R32 is a pure refrigerant and its performance is excellent in achieving greater heat transfer and pressure drop than R410A. R410A is zeotropic (50/50 mass %) mixture of R32 and R125 [1]. The Refrigerant charge amount will be more (100% for example) in case of R410A in air conditioning applications to achieve greater heat transfer coefficient and pressure drop but R32 can achieve better results than R410A with Less Charge amount (65%) that is we can save over 35% of refrigerant charge amount [2]. In the present air conditioning application, Condenser is air-cooled FTHE where air is drawn by motorized propeller fan as Counter flow. In VCRS Compressor compresses, the refrigerant from low pressure, low temperature vapour to high pressure, high temperature vapour then refrigerant enters the Condenser and condensation happens where vapour loses heat to ambient air and turns to liquid. Then that high pressure liquid enters Expansion device where refrigerant pressure drops due to expansion. Then low-pressure liquid Refrigerant enters the evaporator where refrigerant boils and turns to low pressure vapour [3]. The focus is on comparison of the two refrigerants (R32 and R410A) behaviour and parameters inside the condenser and Coil Designer Software used for the design and analysis of FTHE. In the present competitive world considerations like material, material cost, design, fabrication according to its application of FTHE plays an important role. Material used are aluminium fins and micro finned copper tubes to improve the Heat transfer coefficient and pressure drop of refrigerant.

II. DESIGN OF FIN AND TUBE HEAT EXCHANGER

Tube-fin heat exchanger is designed by using Coil Designer software. Fast Solver is selected to calculate the heat transfer coefficient and pressure drop using mass flow rate and inlet state of refrigerant. Coil designer software uses finite volume approach to simulate the heat exchanger. Tubes in Heat exchanger are arranged in staggered convergent configuration. H is the height of the condenser; L is the Length of the coil; W is the width of the condenser. X_h is the horizontal spacing between tubes and X_v is the vertical spacing between tubes.

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N_t is the number of tubes; Nr is the number of rows; δ is the fin thickness; Fin density that is number of Fins per inch (FPI); d_i is the inner diameter of tube; d_o is the outer diameter of tube. δ_t is the tube thickness.

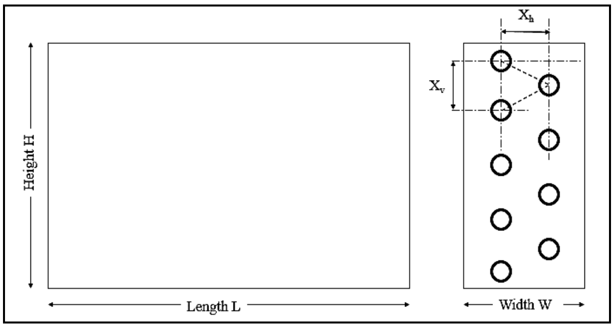

Figure 1: Terminology of fin and tube heat exchanger

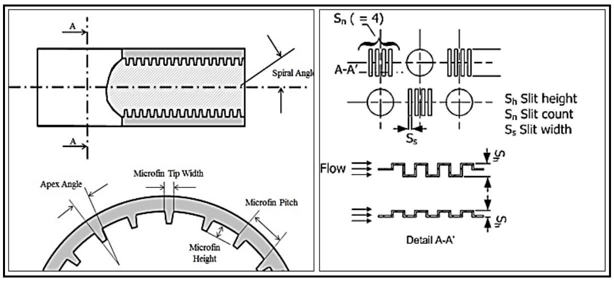

Figure 2: Terminology of micro fin

Since the coil tubes are micro finned which are enhanced to generate the turbulence of refrigerant, in order to achieve the turbulence refrigerant needs surface area. Micro fin arrangement generate the surface area and turbulence, which increases the heat transfer coefficient. From the above figure, spiral angle decides the velocity of the flow [4]. These micro fin increase the coil heat transfer area and reduces the material quantity and material cost. Fins selected are slit type fins arranged on the tube with Fin density that is 22 Fins per inch with the thickness of 0.12 mm and fin spacing as 1.03mm. Slit dimensions plays crucial role in increasing the thermal contact resistance between the fins and there will be less chance of fouling on or inside of the tubes because the refrigerant flow is turbulent [5]. The circuitry of the condenser has two inlets and one outlet. Coming to the material properties of copper tube has tube density as 8900 kg/m^3 and thermal conductivity as 380 W/mk. Aluminium Fin has density as 2700 kg/m^3 and thermal conductivity as 237 W/mk. Design input dimensions for the tube configuration, micro fin parameters and slit fin are given in table 1 and table 2.

Table 1: Design inputs for tube Configuration.

Tube configuration			
Tubes per bank	22	Tube ID	6.5 mm
No of tubes	2	Tube thickness	0.25 mm
Tube length	700 mm	Tube horizontal spacing	12.5 mm
Coil height	450 mm	Tube vertical spacing	21 mm
Tube width	300 mm		
Tube OD	7 mm		

Table 2: Design inputs for Tube Micro fin and Slit type fin.

Micro Fin parameters		Slit Type fin dimensions	
Micro Fin height	0.20 mm	FPI	22
Micro fin width	0.20 mm	Fin thickness	0.12 mm
Micro fin pitch	0.35 mm	Fin spacing	1.03 mm
Apex angle	18 °	Slit height	10 mm
Helix angle	30 °	Slit width	1 mm
Micro Fin count	60	Slit count	4

Air and refrigerant inlet conditions:

The inlet air temperature and humidity conditions are as per IS 1391 part 2 standards. Inlet airflow rate is calculated by using the expression $\dot{v}$ = A x V where $\dot{v}$ is the airflow rate in m3/s; A is cross sectional area and V is the velocity of air entering the condenser. Velocity of inlet air was calculated by using anemometer. Refrigerant inlet conditions are calculated from respective p-h charts for both R32 and R410A are given in table 3 and table 4.

Table 3: Air and refrigerant inlet conditions for R32.

Air inlet Conditions		Refrigerant inlet Conditions	
Parameter	**Value**	**Parameter**	**value**
Dry bulb temperature	35 °C	Inlet Temperature	78.9 °C
Wet bub temperature	24 °C	Inlet pressure	2748 kPa
Air Flow rate	1180 m^3/hr	Mass flow rate	0.014319 4 kg/sec

Table 4: Air and refrigerant inlet conditions for R410A.

Air inlet Conditions		Refrigerant inlet Conditions	
Parameter	**Value**	**Parameter**	**Value**
Dry bulb temperature	35 °C	Inlet Temperature	80.2 °C
Wet bub temperature	24 °C	Inlet pressure	2774 kPa
Air Flow rate	1180 m^3/hr	Mass flow rate	0.0167946 kg/sec

III. HEAT TRANSFER COEFFICIENT AND PRESSURE DROP CALCULATIONS

Nomenclature:

A	Surface area, m^2
T_2	fluid temperature, k
T_1	Surface temperature, k
h	Heat transfer coefficient, W/m^2 K
q	heat flux, W/m^2
ΔT	Temperature gradient, k
$P_1 – P_2$	Pressure Drop, N/m^2
f	Darcy friction factor
ρ	Density of fluid, kg/m^3
V	Velocity of fluid, m/s
L	Length of the tube, m
D	Hydraulic Diameter, m

Heat transfer coefficient:

The heat transfer coefficient h can be calculated from heat transfer rate equation i.e.

$$\dot{Q} = hA(T_2\text{-}T_1)$$

Heat transfer coefficient in W/m^2 K is

$$h = \frac{q}{\Delta T}$$

Pressure Drop:

Pressure drop can be calculated using Darcy-Weisbach friction loss equation

$$P_1 – P_2 = \frac{f\rho Lv^2}{2D}$$

IV. ANALYSIS AND RESULTS OF FIN AND TUBE HEAT EXCHANGER USING R32.

The heat load capacity of condenser calculated should be greater than 3516 W for one TR capacity air conditioning system and the refrigerant pressure drop should be less than 50kPa. The obtained capacity of the Condenser is 3678 W, which is acceptable. The pressure drop calculated is 16.954 kPa. The average heat transfer coefficient of refrigerant in each phase inside the condenser is

Phase type	Heat transfer coefficient
Vapour phase	1741.537 W/m^2 K
Two-phase	3468.621 W/m^2 K.
Air side	344.9742 W/m^2 K
Liquid phase	2335.392 W/m^2 K

Coming to the refrigerant phase wise distribution inside the condenser.

Refrigerant phase distribution	HX length (%)	Total charge (%)	Heat load (%)
Liquid phase	3.863	15.62	2.64
Two phase	83.18	79.58	74.71
Vapour phase	12.95	4.80	22.66

The 3-D representation of refrigerant Condensation phase change inside evaporator is shown in figure 3.

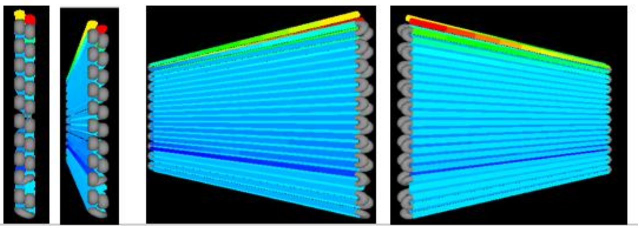

Figure 3: 3-D representation of refrigerant phase change along Condenser using R32.

V. ANALYSIS AND RESULTS OF FIN AND TUBE HEAT EXCHANGER USING R410A.

The obtained capacity of the Condenser using R410A is 3518.7 W and the pressure drop calculated is 14.345 kPa. The average heat transfer coefficient of refrigerant in each phase inside the condenser

Phase type	Heat transfer coefficient
Vapour phase	1903.362 W/m^2 K
Two-phase	2666.060 W/m^2 K.
Air side	342.306 W/m^2 K
Liquid phase	2627.702 W/m^2 K

Coming to the refrigerant phase wise distribution inside the condenser.

Refrigerant phase distribution	HX length (%)	Total charge (%)	Heat load (%)

Liquid phase	10.45	27.94	4.58
Two phase	76.14	67.98	69.76
Vapour phase	13.41	4.08	25.66

The 3-D representation of refrigerant Condensation phase change inside evaporator is shown in figure 4.

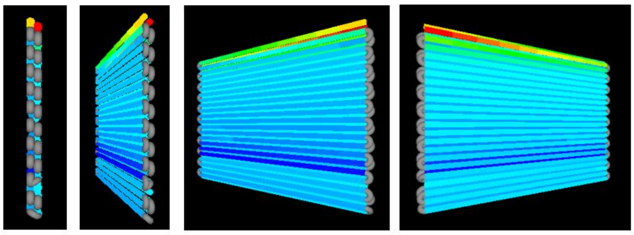

Figure 4: 3-D representation of refrigerant phase change along Condenser using R410A.

Comparison of the average heat transfer coefficient for both the refrigerants is given in below figure 5.

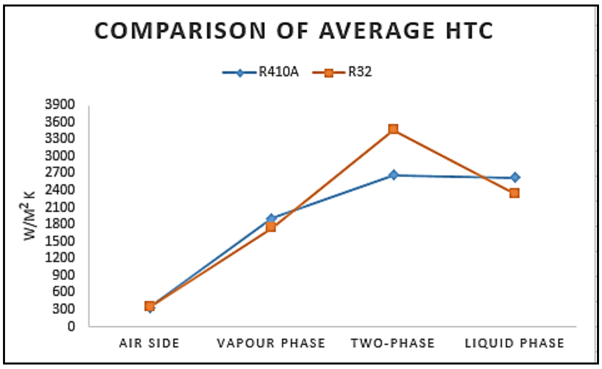

Figure 5: Comparison of average heat transfer Coefficient using R32 and R410A.

From the figure 5 it is observed that heat transfer coefficient of R32 is higher than R410A only in air side and two phase state than other phases. From Darcy-Weisbach friction loss equation, we can see that pressure drop is less in R410A case compared to R32 because the density of R32 is less than R410A. However, this pressure drop in R32 does not affect the heat transfer because of thremo physical properties of R32. If the pressure drop of R32 is more but heat transfer coefficient of R32 will be greater than R410A.

VI. CONCLUSION

Coil Designer software is used to design the FTHE. The main objective is to study and compare the refrigerant behaviour inside the condenser and to calculate the heat transfer coefficient, pressure drop and heat load of refrigerant inside the condenser using refrigerants R32 and R410A. The obtained capacity of the condenser for one TR system using R32 was 3678W and pressure drop was 16.954 kPa which is under the limit and acceptable. Using R410A, the obtained capacity was 3518.7 W and pressure drop was 14.345 kPa. On comparing the obtained results, using both the refrigerants the capacity and the pressure drop using R32 is higher than R410A. According to the properties of R32, pressure drop may be more but heat transfer coefficient will higher than R410A. The heat transfer coefficient observed is gradually increasing when refrigerant is converting from vapour phase to two-phase and then it is decreasing in liquid phase due to pressure drop in refrigerant. Refrigerant phase wise distribution observed is more in refrigerant two-phase form than vapour and liquid phase.

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