Numerical Simulation of Fin and Tube Gas Cooler For Transcritical CO_2 Air Conditioning System

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Abstract

This paper focuses on the development of plain fin and tube gas cooler for CO_2 transcritical air conditioning system operating in subtropical conditions. The numerical simulation is carried out for identification of variation of heat transfer and pressure drop. The thermo-physical and transport properties for CO_2 are taken from NIST database. The simulation study has been carried out for 0.26 kg/sec air mass flow rate and 0.006944 kg/sec refrigerant mass flow rate. The gas cooler pressure of 90 bar at 40^oC dry bulb temperature has been considered. Air side and refrigerant side heat transfer coefficients are estimated and compared with available correlations developed by various researchers viz. Rich, McQuiston, Webb and Gray. Further it is observed that simulation results are having a close matching with correlations from literature.

1. Introduction

Global energy and environmental issues have compelled the researchers and manufactures to develop an environmental friendly and energy efficient refrigeration and air conditioning systems. There are two ways to deal with this problem; one is retrofitting of existing system for an energy efficient working fluid or development of new system. A refrigerant is ideal when is chemically stable and inert, has excellent thermal and fluid flow characteristics. It is compatible with common materials, soluble in lubricating oils, nontoxic and nonflammable, environmentally acceptable and has low cost. Since no single fluid meets all these attributes, a variety of refrigerants have been developed and used in HVAC&R systems.

Existing chemical refrigerants; CFC, HCFCs and HFCs have disadvantages of high cost and unresolved issues of environmental impact compared with natural refrigerants. Among natural refrigerants, CO_2 has low toxicity, non-flammability and low cost. This makes it preferred sustainable refrigerant as a permanent replacement candidate to man-made refrigerants. CO_2 offers the thermo-physical and transport properties that are unique and substantially different than other conventional refrigerants. Due to low critical temperature (31^oC) of CO_2, whenever ambient temperature exceeds the critical temperature, the heat rejection is by single phase gascooling process. High vapour pressure and volumetric heat capacity of CO_2 causes to freshly design the air conditioning system instead of retrofitting the existing system.

This paper focuses on simulation of plain fin and tube gascooler for transcritical CO_2 air conditioning system for high ambient Indian subtropical climatic conditions. This study has oriented towards the development of the plain fin and tube gascooler, which will be possible to manufacture locally at low cost.

2. Literature review

Lorentzen et al. [1] reintroduced natural refrigerant CO_2 as substitute to chemical refrigerants since it has competitive thermodynamic and transport properties. Many researchers have reported simulation studies and findings on fin and tube gascooler for low ambient climatic conditions [5, 6, 7, 13, 17, 18, 19]. The efforts are lacking in the thermal design of plain fin and tube gascooler for transcritical CO_2 air conditioning system in the open literature for subtropical conditions. Few researchers have focused on the development of microchannel gascooler since they offer highest compact ratio and capacity [8, 9, 10, 11, 12]. However manufacturing and use of micro channel gascooler requires huge investment in design, tooling, etc., which may not be cost effective. Many studies are available on prediction of heat transfer coefficient and pressure drop for micro and mini channel tubes [2, 3, 9, 13, 14, 20, 21]. Pettersen et al. [8] have notified that because of smaller tube and manifold dimensions the size of heat exchanger gets reduced as compared with

HFC134a system and the heat exchangers give best overall efficiency.
Yin et al. [9] developed gas cooler model by using a finite element method. The experimental results have shown that the heat transfer coefficient increased as temperature of supercritical CO_2 decreased along the gas cooler flow length. The authors concluded that the performance of the gas cooler improved by using multiple slabs instead of multiple passes and proved that the three pass gascooler was the best slab design.
Kim et al. [13] suggested effectiveness and NTU method for obtaining air side thermal performance for cross flow heat exchanger with both fluids unmixed. The authors observed that for increased flow depth from 16 mm to 24 mm and face velocity 0.75 m/sec, the heat transfer coefficient decreased from 100 to 65 W/m^2K. For change in flow velocity from 0.75 to 3 m/sec and flow depth of 16 mm, the heat transfer coefficient enhanced was from 100 W/m^2K to 130 W/m^2K. They concluded that for 16 mm flow depth and 23^o louver angle, the heat transfer coefficient (140 W/m^2K) does not change with flow velocity.For 16 mm flow depth and 23o louver angle, the heat transfer coefficient obtained was 100 W/m^2K. For 27^o louver angle from 15^o to 23^o, the pressure drop increased from 40 Pa to 60 Pa for 16 mm flow depth.
Yoon et al. [14] proved that for copper tube diameter of 7.73 mm, increasing mass flux from 241 kg/m^2sec to 464 kg/m^2sec influences the heat transfer coefficient. For calculating heat transfer coefficient, the authors have used Krasnoshchekov and Protopopov, Baskov, Petrov and Popov and Pitla correlations [23, 24, 25, 26] with RMS deviations 29.7, 29.6, 47.9 and 38% respectively. For calculating refrigerant properties at bulk temperature, the authors have suggested Dittus-Boelter's correlation with average deviation 12.7%. The authors also observed the drastic variation in specific heat near the critical region. The heat transfer coefficient has the maximum value at its pseudo critical temperature. Pettersen [15] reported that the heat transfer coefficient increases with varying heat flux from 5 to 20 kW/m^2 and mass flux from 190 to 570 kg/m^2sec.Son and Park [16] presented that because of maximum specific heat of CO_2 near the pseudo critical temperature, the heat transfer coefficient increases slowly in the entrance region of gascooler and decreases at the exit of gas cooler. The authors concluded that the Bringer-Smith's correlation showed the best agreement for heat transfer coefficient with mean deviation of 23.6%.
Simulation model has been developed for plain fin and tube gas cooler by Chang and Kim [7]. The authors presented that for increased fin pitch and transverse tube pitch, the air side heat transfer coefficient decreases, while longitudinal tube pitch has no influence on the air side heat transfer coefficient. The fin efficiency is not much affected by the fin pitch or the longitudinal tube pitch but decreases largely when the transverse tube pitch increases. Ge and Cropper [5] discussed the simulated performance of the air cooled gascooler for different circuitry arrangements.

3. Numerical simulation

The spreadsheet has been developed to simulate the cross flow unmixed – unmixed plain fin and tube gascooler. The thermo-physical and transport properties of CO_2 are taken from REFPROP, refrigerant properties database developed by NIST [27]. The operating inlet and outlet conditions of the gascooler are worked out by transcritical CO_2 cycle simulation considering subtropical conditions and are given in Table 1.

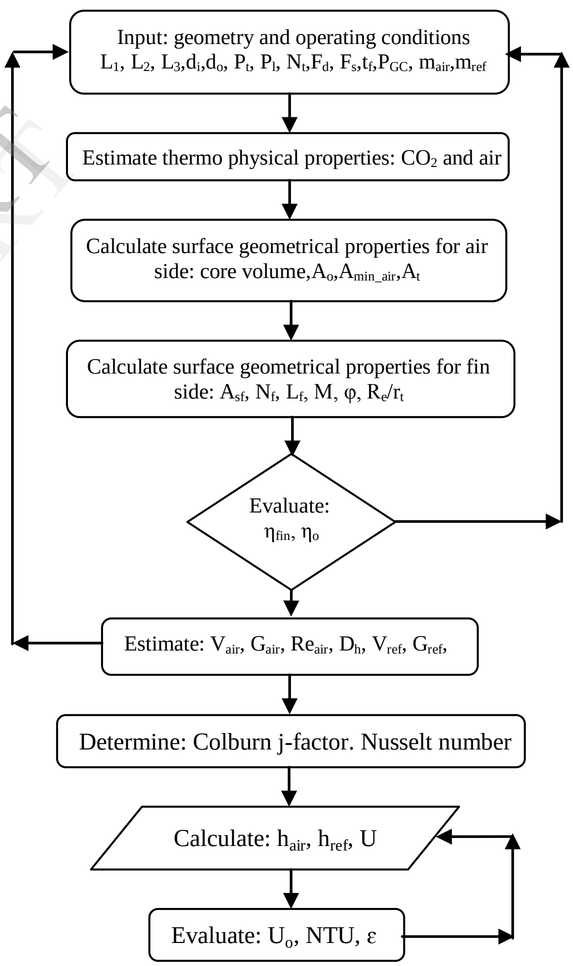

Figure 1. Flow chart indicating steps in the simulation

It has been reported that three to four rows gives better compact design of gascooler than two rows geometry [19]. Hence, the counter cross flow, four row single pass staggered tube arrangement has been considered for the gascooler as shown in Figure 2. Figure 1 shows the methodology used for the simulation and Table 2 provides the geometry of the gascooler.

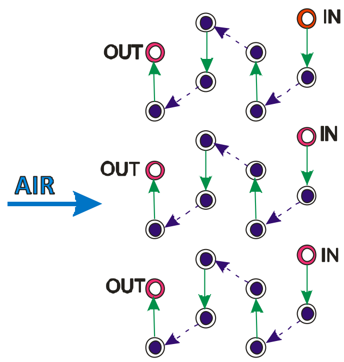

Figure 2. Four pass - three circuit staggered tube gascooler

The geometrical size of the base case gascooler is finalized by parametric evaluation considering the optimum gascooler pressure of 90 bars and 40°C dry bulb temperature to avoid an extensive loss of capacity at higher temperatures.

Table 1. Operating conditions at gascooler

Air inlet temperature [°C]	40	Mass flow rate of air [kg/sec]	0.26
Refrigerant inlet temperature [°C]	106	Velocity of air [m/sec]	2.55
Refrigerant outlet temperature [°C]	42	Mass flow rate of refrigerant [kg/sec]	0.0069
Refrigerant inlet pressure [bar]	90	Velocity of refrigerant [m/sec]	0.9254

Table 2. Geometry of base case gascooler model

Tube inner diameter [mm]	4.75	Tube material	Cu
Tube length [mm]	550	Fin material	Al
Transverse tube spacing [mm]	25	Fin spacing [mm]	1.37
Longitudinal tube spacing [mm]	18	Fin density [fpi]	16
Number of tube rows	4	Fin thickness [mm]	1.27

4. Simulations

The heat transfer rate of air side and refrigerant side is calculated using equations 1 and 2.

$$Q_a = \dot{m}_a . C_{p_a} \Delta T_a \qquad (1)$$

$$Q_r = \dot{m}_r . C_{pr} \Delta T_r \qquad (2)$$

$$Q_{max} = {Q_a}/{\varepsilon} \qquad (3)$$

For both fluids unmixed effectiveness is calculated by equation 4.

$$\varepsilon = 1 - exp\left[\frac{NTU^{0.22}}{C_r}\left\{exp\left(-C_r.NTU^{0.78}\right)-1\right\}\right] \qquad (4)$$

Where,

$$C_r = \frac{\left(\dot{m}C_p\right)_{min}}{\left(\dot{m}C_p\right)_{max}} \qquad (5)$$

Overall heat transfer coefficient for the heat exchanger is obtained from equation 6.

$$U_o A_o = \left(\dot{m}C_p\right)_{min} NTU \qquad (6)$$

The air side heat transfer coefficient is calculated using equation 7.

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{\delta_w}{k_w A_w} + \frac{1}{\eta_o h_o A_o} \qquad (7)$$

Colburn j factor is calculated by using available correlations suggested in literature review and found that Rich correlation shows best agreement with maximum deviation of 23%, and equation 8 calculates j factor.

$$j_R = 0.195 * R_{e_{air}}^{-0.35} \qquad (8)$$

Based on the experimental data on gas cooling of supercritical carbon dioxide, Yoon et al. [14] suggested an empirical correlation using the form of Dittus-Bolter's correlation and multiplying the density ratio, shown in equation 9. The Yoon correlation has an average deviation of 1.6%, the absolute average deviation of 12.7% and the root mean square deviation of 20.2%.

$$N_u = a.R_e^{\,b}.P_r^{\,c}.\left(\frac{\rho_{pc}}{\rho}\right)^n \qquad (9)$$

Where,

a=0.14; b=0.69; c=0.66; n=0 ... for $T > T_{pc}$

a=0.013; b=1.0; c=-0.05; n=1.6 … for $T < T_{pc}$

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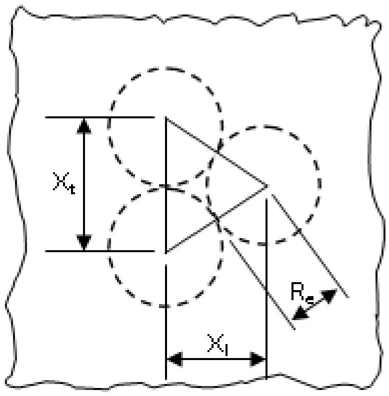

Figure 3. Continuous circular fins

Taylor has symmetrically divided the continuous plain fin area around the staggered tubes into hexagonal shaped section to calculate the overall air side fin surface efficiency [18]. The authors modified this method and considered circular fin sectional area to avoid iterations involved in calculating geometry parameters of hexagonal fin surface area shown in Figure 3.

The air side fin efficiency is calculated by equation 10.

$$\eta_o = 1 - \frac{A_{fin}}{A_o}\left(1-\eta_{fin}\right) \qquad (10)$$

The fin efficiency of a circular fin is calculated from equation 11.

$$\eta_{fin} = \frac{tanh\left(mr_t\varphi\right)}{mr_t\varphi} \qquad (11)$$

Where, m is the standard extended surface parameter, which is defined as,

$$m = \sqrt{\frac{2h_o}{k_{fin}.t_f}} \qquad (12)$$

The fin efficiency parameter for a circular fin, φ is calculated using equation 13.

$$\varphi = \left(\frac{R_e}{r_t} - 1\right)\left(1 + 0.35ln\left(\frac{R_e}{r_t}\right)\right) \qquad (13)$$

where, the equivalent circular fin radius, R_e, is defined from equation 14

$$\frac{R_e}{r_t} = 1.27\left(\frac{X_t/2}{r_t}\right)\sqrt{\left(\frac{X_l}{X_t/2}\right) - 0.3} \qquad (14)$$

5. Result and discussion

5.1 Fin density

Colburn *j* factor is the dimensionless heat transfer coefficient. For air flow rate 830 m^3/hr and inlet air temperature 40^{o}C, it is observed from Figure 4 that, as fin density increases from 8 fpi to 24 fpi through 33.33% change, air side minimum free flow area decreases by 5.04%, Reynolds number increase by 10.53% and j factor decreases by 0.9%. This increase in the fin density has marginally increased Nusselt number through 1.58% shown in Figure 5.

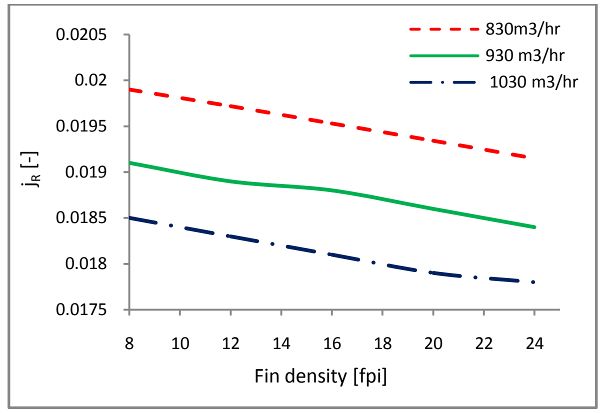

Figure 4. Effect of fin density on j factor

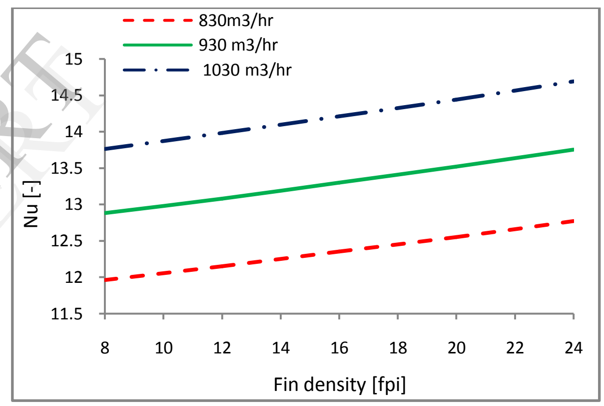

Figure 5. Effect of fin density on Nusselt number

5.2 Fin thickness

As shown in Figures 6 to 9, for 830 m^3/hr air flow rate, 26.78% increase in fin thickness from 0.112 mm to 0.142 mm, the minimum free flow area reduces by 2.57% results in 2.62% increase in the Reynolds number, 1.70% increase in Nusselt number, 5.18% increase in number of transfer units and 7.84% increase in fin efficiency. This change in the fin thickness increases an effectiveness of the gascooler by 2.88% as shown in Figure 9.

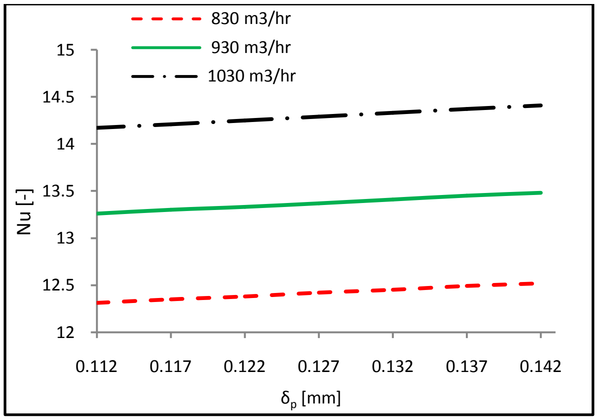

Figure 6. Effect of fin thickness on Nusselt number

As shown in Figures 7 and 9, for the base case gascooler model when air flow rate increases from 930 m^3/hr to 1030 m^3/hr Reynolds number and Nusselt number increase by 10.75% and 6.88% respectively and fin efficiency decreases by 2.32%. The effectiveness of gas cooler decreases by 4.89% and the capacity of the gas cooler increases by 2.36% for the same change in the air flow rate.

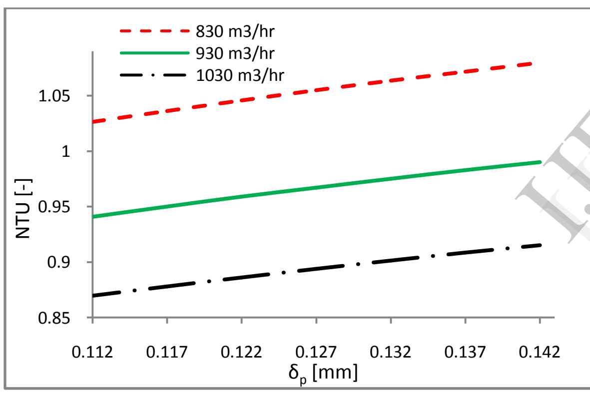

Figure 7. Effect of fin thickness on NTU

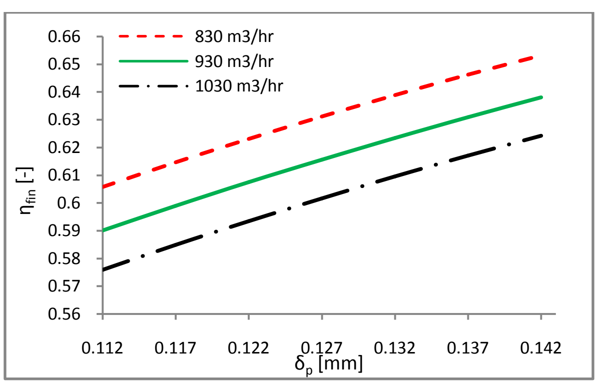

Figure 8. Effect of fin thickness on fin efficiency

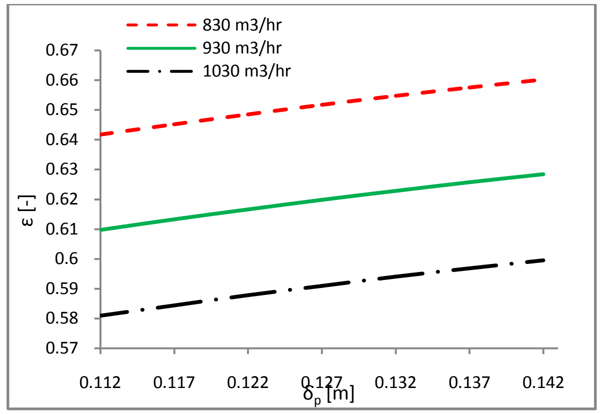

Figure 9. Effect of fin thickness on effectiveness

5.3 Transverse tube spacing

The effect of transverse tube spacing and longitudinal tube spacing on the performance of gas cooler is also studied. It has been notified that the performance of gas cooler mainly depends upon the transverse tube spacing rather than on the longitudinal tube spacing. If the transverse tube spacing is very less, the average air velocity and the Reynolds number in the core are very high. This also causes un-necessary turbulence effect in the core of the gascooler. Air side pressure drop tends to increase in inverse proportion with the transverse tube pitch and definitely affects the performance of gas cooler.

The effect of 66.66% increase in transverse tube spacing from 15 mm to 25 mm on the Nusselt number for varying air flow rate is represented in Figure 10.

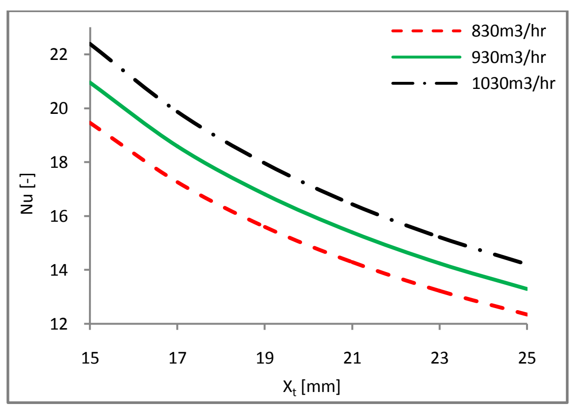

Figure 10. Effect of transverse tube spacing on Nusselt number

An increase in transverse tube spacing at air flow rate 830 m^3/hr, minimum free flow area increases by 101.67% as a result the Reynolds number reduces by 101.67%. Therefore j factor decreases by 27.82% and the mass flux decreases by 101.66%, which results in 57.57% decrease in Nusselt number as shown in Figure 10.

5.4 Effect of gas cooler width

An increase of 24.48% gascooler width from 490 mm to 610 mm for constant air flow rate 830 m^3/hr, it is observed that minimum free flow area increases by 28.57% and j factor increases by 7.96%. This result in 24.48% and 15.33% drop in Reynolds and Nusselt numbers respectively as shown in Figure 11.

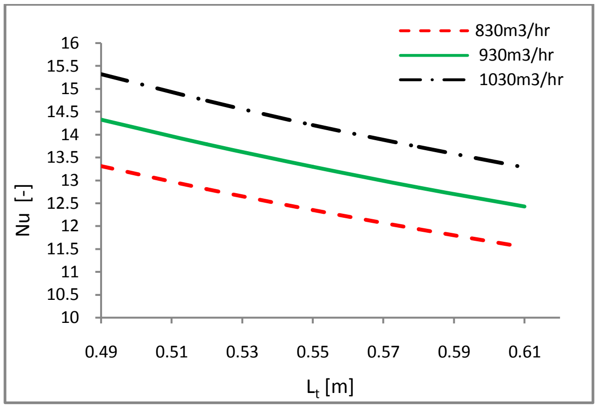

Figure 11. Effect of gas cooler width on Nusselt number

This change in gascooler width has also an effect on the number of transfer units and the overall heat transfer coefficient of the gascooler. The number of transfer units increased by 17.7% for the same change in the width of the gascooler as given in Figures 12. The effectiveness of the gascooler increases by 9.83% as shown in Figure 13.

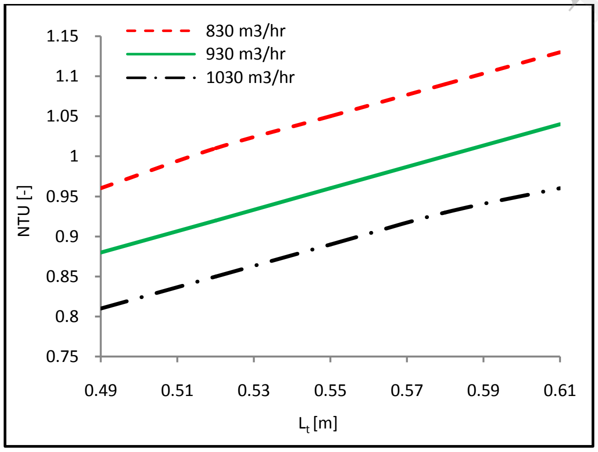

Figure 12. Effect of gas cooler length on NTU

5.4 Effect of inlet temperature

The effect of 53.33% increase in inlet temperature of air from 30°C to 46°C on the overall heat transfer coefficient is depicted in Figure 14 by using different air side correlations: Rich, McQuiston and Webb and Gray correlations [19]. The overall heat transfer coefficients increase by 4.89% by Rich correlation, 3.65% by McQuiston and 3.93% by Webb and Gray correlations as shown in Figure 14.

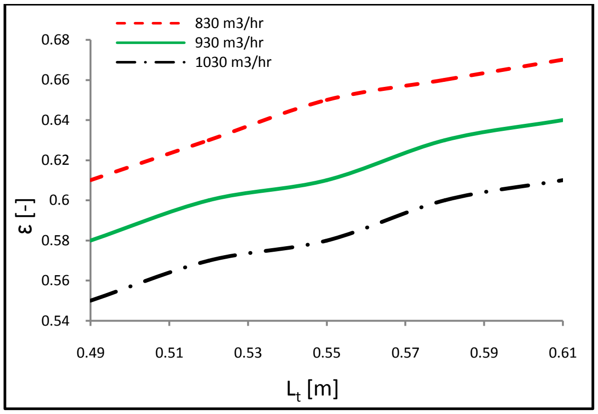

Figure 13. Effect of gas cooler length in effectiveness

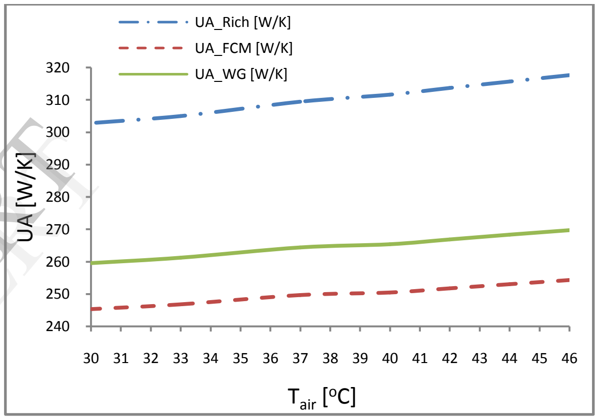

Figure 14. Effect of inlet temperature of air on overall heat transfer coefficient

The effect of change in inlet temperature on effectiveness for 830 m^3/hr flow rate is graphically represented in Figure 15. From Figure 15 it has been notified that the effectiveness calculated by Rich correlation is 13.77% more compared with McQuiston correlation and 9.77% more compared with Webb and Gray correlation.

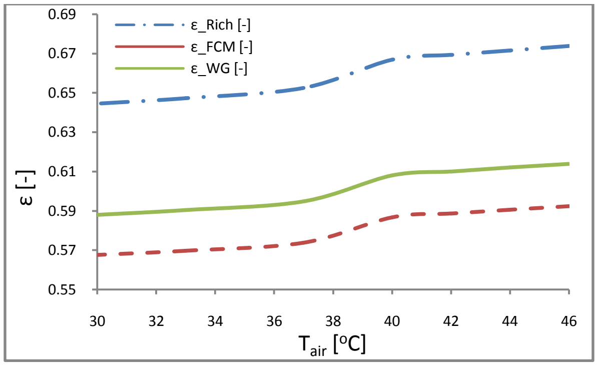

Figure 15. Effect of inlet temperature of air on effectiveness of gas cooler

6. Conclusion

From this simulation work following conclusions can be drawn.1) For increase in fin density, Reynolds number increases by 10.53%, Nusselt number by 1.58% and heat transfer coefficient by 7.39 for 830 m^3/hr air flow rate. While Colburn *j* factor decrease by 0.9%. For increase air flow rate, the air side minimum free flow area decreases by 5.04% and Reynolds number increases by 4.08% which results 4.18% decreases in *j* factor.

2) Effect of fin thickness on various factors are monitored and observed that, increased fin thickness increases Reynolds number by 2.62%, Nusselt number by 1.70% hence increases the heat transfer coefficient by 1.69%. It has been also notified that increased fin thickness reduces minimum free flow area by 2.57%, this results increase in NTU by 5.18%, overall heat transfer area by 1.73%, effectiveness by 2.88% and capacity of gas cooler by 3.29%. While the fin efficiency decrease by 2.38%.

4) As the transverse tube spacing increases, minimum free flow area decrease by 108.5%, Reynolds number by 101.68%, mass flux by 101.66%. Hence this parameters decreases Nusselt number by 57.57% and heat transfer coefficient by 57.76%, while *j* factor increase by 28.28% for 830 m^3/hr air flow rate.

5) Effect of increase in inlet temperature of air is monitored and observed that the overall heat transfer coefficient increase by 4.89% calculated by Rich correlation, which shows well agreement as compared with McQuiston and Webb and Gray correlations. Also as the inlet temperature of air increases, the effectiveness of gas cooler increases by 13.77%. The capacity of the gas cooler is more calculated by Rich correlation and shows maximum value up to 6.18 kW for 31°C.

6) As the gas cooling process progresses, the capacity of gas cooler reaches it's maximum and then decreases. Since the specific heat increases drastically near the critical region, the capacity of the gas cooler increases greatly and reaches the maximum value nearly at the pseudo critical temperature.

7) Increase in gas cooler length decreases minimum free flow area by 24.37%, Colburn *j* factor by 8.02%, Reynolds number by 24.48%, mass flux by 48.16%, Nusselt number by 15.33% and heat transfer area by 15.30% for 830 m^3/hr air flow rate. While increase gas cooler length increases NTU by 17.7%, overall heat transfer coefficient by 18%, effectiveness of gas cooler by 9.83% and capacity of gas cooler by 1.96% for 830 m^3/hr air flow rate.

It has been observed that for increase in gas cooler length and air flow rate, increases Nusselt number by 6.90%, heat transfer coefficient by 6.87%, Reynolds number by 10.75% and capacity of gas cooler by 1.96%, while it decreases NTU by 8.64%, overall heat transfer coefficient by 2.27% and effectiveness of gas cooler by 5.45%.

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