Heat Transfer Coefficient Correlation for Lance & Offset Fin of a Compact Heat Exchanger for R1234yf Using CFD Analysis

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ABSTRACT

The refrigerants with low global warming potential (GWP) and zero ozone depletion potential (ODP) are gaining importance due to environmental concerns. Refrigerant R1234yf has a 4-50 GWP and zero ODP compared to 1430 GWP and zero ODP for R134a and has thermodynamic properties similar to R134a. This paper describes the numerical analysis to study the heat transfer coefficient of R1234yf in Lance &Offset fin of compact heat exchangers. The performance of R1234yf is compared with R134a at different Reynolds number. Computational fluid dynamics (CFD) results of R134a are validated with experimental results. The second order polynomial equations with R^2 value of 0.99 generated for all the thermophysical properties of R1234yf and used for CFD analysis. CFD methodology has been used to develop the Single phase R1234yf heat transfer coefficient correlation for the selected lance & offset fin using ANSYS Fluent 2019R1. Also, CFD analysis carried out for the same fin using R134a. R1234yf CFD analysis results shows that the performance is slightly less compared to R134a. These investigations also include the study of flow pattern for laminar regions. Finally, the generalized heat transfer coefficient correlation is developed for the selected Lance and Offset fin for the laminar flow region. This numerical estimation can reduce the number of tests/experiments to a minimum for similar applications.

Keywords:Compact Plate Fin Heat Exchangers, R1234yf, L&O fins, CFD Analysis, Colburn factor '*j*'.

1. INTRODUCTION

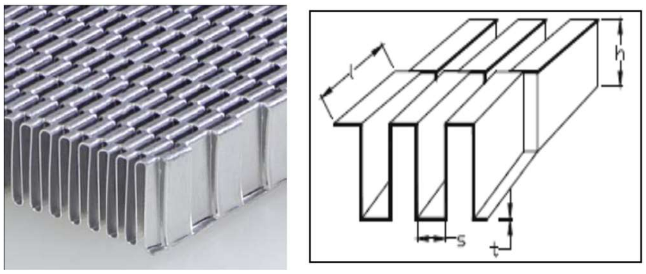
The most popularly used fin surfaces in compact heat exchangers are the lance & offset (L&O) fins, wavy fins, louvered fins and plain fins. Amongst these fin types, the lance & offset fins assume lot of importance due to its enhanced thermo-hydraulic performance. Thermo-hydraulic design of Compact Heat Exchangers (CHEs) is strongly dependent upon the predicted/measured dimensionless performance (Colburn factor '*j*' and Fanning friction '*f*' vs. Reynolds number (Re) of heat transfer surfaces. These are widely used in Aerospace, automobile industry and various process plants. Typically, a heat exchanger is called compact if the surface area density is greater than 700 m^2/m^3 in either one or more channels of a two stream or multi-stream heat exchangers as defined by Shah R.K et al., [1].

Refrigerant R134a is a commonly used refrigerant in vapor compression refrigeration cycles in automobile and aerospace industries after the refrigerant R12 was phased out due to higher ODP. Now, it is the time to use the low GWP refrigerants due environmental concerns. The refrigerants with low global warming potential gaining importance due to environmental concerns. R1234yf has a significant potential to be a drop in replacement for R134a [2]. Gaurav and Kumar [3] research summarizes that eco-friendly R1234yf refrigerant is the replacement of R134a in air-conditioning with temperature set for the indoors is between 20^0C and 24^0C. Two phase CFD flow analysis of R134a and R1234yfrefrigerants shows that R134a properties were better towards cooling performance of an evaporator core [4]. Kays and London [5] made one of the attempts at analytical modeling of the heat transfer and friction loss in L&O fins and proposed a laminar boundary layer solution that includes the finite drag contribution of the blunt fin edges. Joshi and Webb [6] developed elaborate analytical models to predict *f* and *j*. Weiting [7] proposed empirical correlations for *j* and *f* performing experiments on 22 rectangular Offset strip fin configurations over two Reynolds number ranges: $Re \leq 1000$ which is primarily laminar and $Re \geq 2000$ which is primarily turbulent. The experimental data used have been obtained from experiments using air as working fluid and hence the applicability of the correlations to fluids outside the gas Prandtl number range may be open to question.

Among the very few numerical analysis for generation of heat transfer and pressure drop characteristics of an Offset strip fin heat exchanger is a study by Bhowmik and Kwan-Soo Lee [8]. They recalculated the Nusselt number using the correlated *j* and the effects of Prandtl numbers were studied. They also indicated that an air model should not be used to predict the fluid flow and heat transfer of Offset strip fin heat exchangers operated with other media.

Ke Li et al [9] studied the multi parameter optimization of serrated fins and the analysis results show the effect of fin thickness and solid material is negligible and interrupted length is most significant on *j* factor with Re.

It is important to know the heat transfer and flow friction characteristics (`*j*' and '*f*' factors) of the enhanced surfaces for a proper selection and rating of the equipment. Lance & Offset fins have a high degree of surface compactness and substantial heat transfer enhancement obtained as a result of the periodic starting and development of laminar boundary layers over interrupted channels formed by the fins and their dissipation in the fin wakes. There is, of course, an associated increase in the pressure drop due to increased friction and form-drag contribution from the finite thickness of the interrupted fins. The isometric view of a Lance & Offset fin is shown in Fig. 1(a).The geometrical features of the three-dimensional offset fin flow channel are described by the fin height(h), fin spacing(s), fin thickness(t), lance length(l) as shown in Fig 1(b).



(a) photograph of L & O fin (b) L&O fin dimensional notations

Figure 1: Schematic of fin geometry

Pallavi and Ranganayakulu [10] compared and summarized the various research aspects relating to internal single phase flow studies using air in Lance & Offset fins. In open literature, single phase heat transfer coefficient for L&O fins with R1234yf is not available. This paper describes the numerical analysis to study the heat transfer coefficient of R1234yf using L&O fin of compact heat exchangers. The results are compared with previous air-cooled models.This paper focuses on generation of Colburn factor *j for* L&O fins with R1234yf using ANSYS Fluent.

2. CFD APPROACH

In this work, the CFD package ANSYS FLUENT 2019R1 is usedfor simulation. R1234yf fluid properties library not available in fluent and Thermo-physical properties are added in the form of second order polynomial equations. In FLUENT, the conservation equations of mass, momentum and energy are solved using the finite volume method. There are several turbulence models available in the code. The turbulent flow is calculated by the semi-implicit SIMPLER as mentioned in Versteeg and Malalasekera[11]. A standard k-ε model with enhanced wall treatment is used to predict turbulent flow in the fin geometry. The Reynolds transport equations can be written in a generalized form as given by John Anderson [12] and Patankar [13].

$$\text{div}\,(\rho u\varphi) = \text{div}\,(\Gamma\, grad\ \varphi) + S_\varphi \qquad (1)$$

Where φstands for a generalized transport variable, which is used for all conserved variables in a fluid flow problem, including, mass, momentum and the turbulence variables k and ε. Γ represents the effective diffusivity (sum of the eddy diffusivity and the molecular diffusivity). $S_\varphi$is the source term for the respective dependent variable. The solution of the above set of equations is applied to the prediction of velocity and turbulence levels throughout the domain.

3. METHODOLOGY

The CFD analysis is carried out using ANSYS Fluent 2019R1 for an estimation of $j$factors for L&O fin geometry for different fluids. In this model, a single layer of actual offset strip fin is modeled and meshed. The three-dimensional computational domain of fin model is shown in Fig.2. The offset strip fin is characterized for the laminar range of Reynolds number to determine the corresponding j values. In order to overcome the entrance effect, the concept of periodic fully developed flow as suggested by Patankar et al [14] is implemented for this part of analysis. After the analysis using ANSYS Fluent we get the pressure drop for unit length, which is used for estimation of the friction factor f as per Kays and London [5] procedure. In this model, the “velocity inlet” and “outflow (pressure outlet)” boundary conditions are used at the inlet and outlet of the fin geometry, respectively. In Present analysis conjugate heat transfer effect taken into account. The constant wall temperature boundary condition is employed for the walls as used by few authors earlier. The temperature difference between inlet and outlet of the core, in turn, is used for calculating the j factor using the Kays and London [5] methodology.

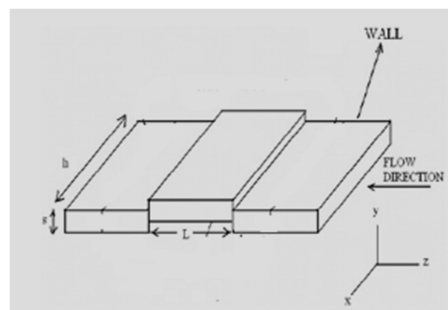


Figure 2: Computational domain for an offset fin

3.1 L&O Fin details

The details of the selected L&O fin geometry are given in Table 1.

Table 1: L&O Fin Details

Fin height, h (mm)	3.05
Lance length, *l* (mm)	3.175
Fin spacing, s (mm)	0.847
Fin thickness, t (mm)	0.1016
Hydraulic diameter, D_h (mm)	1.1894
Fins per inch FPI	30
Material	Aluminium Alloy (AA3003)

3.2 Grid Independency Test

Grid independence test is carried out in the first step for each and every analysis. This analysis is performed to determine the minimum size required to capture the co-efficient of discharge approximately. This is essential to strike a proper balance between the conflicting requirements of lower simulation times and the desired accuracy levels. Different mesh sizes ranging from very coarse to very fine mesh is taken first and solved with the same boundary conditions. Then the mesh size is finalized based upon the consistency of the desired value attained. The number of elements of 633447 is used for the further analysis. A Graph plotted between mesh size and Colburn *j* factor for grid sensitivity analysis is shown in Fig.3.

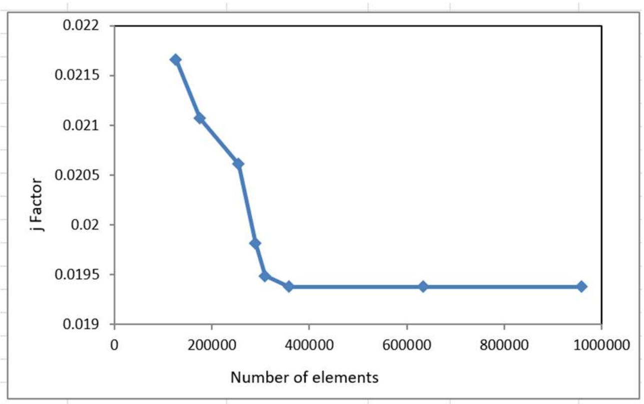


Figure 3: Grid Independence Analysis

3.3 Convergence Criteria

Convergence criteria test to show the level of convergence and beyond which the results are not much varied. The convergence criterion of 10^{-4} for the continuity equation, momentum equation and convergence criterion of 10^{-6} for the energy equation is adopted for the entire analysis. The convergence criterion with respect to the number of iterations is shown in the Fig.4.

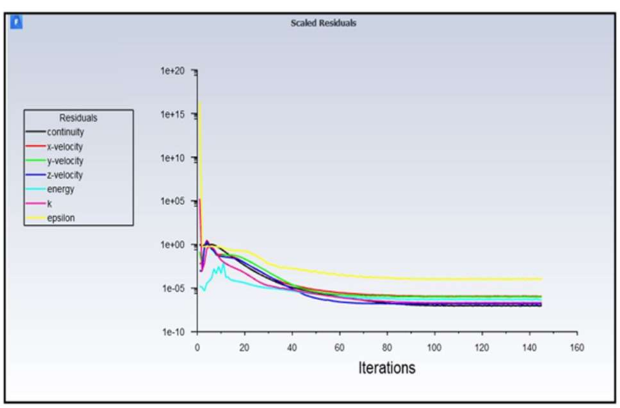


Figure 4: Convergence residuals

3.4 Thermodynamic properties of fluids

Refrigerants R134a and R1234yf are selected for analysis and comparison of performance at different Reynolds number using ANSYS Fluent CFD. The material propertiesof R134a and R1234yf are provided in Table 2.

Table 2: Thermodynamic properties of Refrigerants

Thermodynamic Property	R1234yf	R134a
Chemical Name	2,3,3,3-Tetrafluropropane	1,1,1,2-Tetrafluroethane
Chemical Formula	$C_3F_4H_2$	CF_3CH_2F
molar Mass (kg/kmol)	114.04	102.03
Boiling Point at 1atm (K)	243.7	247.08
Freezing Point (K)	169.85	162.3
Critical Temperature (K)	367.85	374.21
Critical Pressure (MPa)	3.38	4.06
Critical Density (kg/m³)	478.01	511.9
Vapor Density (kg/m³)	12.07	9.81

Thermodynamic properties of the R1234yf and R134a have very similar values for critical temperature and molar mass and R1234yf is a good choice with respect to environmental concern. There is a need to generate good correlations for proper design of vapor compression system for automobile and aerospace industries.

4. Heat Transfer coefficient

Generally, the single-phase heat transfer coefficient can be expressed with the Colburn *j* factor,

$$j = \frac{h}{G\,C_p} Pr^{2/3} \quad (2)$$

For the present fin geometry, the Reynolds number for water flow is defined as

$$Re = \frac{\rho V D_h}{\mu} = \frac{G D_h}{\mu} \quad (3)$$

Where $G = \frac{m}{A_f}$ (4)

and

$$D_h = [2(s-t)h] / \left[(s+h) + \frac{th}{l}\right] \quad (5)$$

An extensive literature survey has been carried out to find a single-phase heat transfer coefficient correlation of the refrigerant R1234yf for lance & offset fin. The water side heat transfer coefficient and friction factor correlations presented by Ranganayakulu et al [15] by comparing with Weiting[7], Manglik and Bergles[16] and Joshi and Webb[6], which were obtained from the experiments with air flow. Hu et al.[17-18] have published the effects of the Prandtl number on Colburn *j* factor for water and polyalphaolefin (PAO) fluids. They claimed that air models over-predict the *j* factor for other fluids Hence, a detailed analysis has been carried out using ANSYS Fluent tool to estimate the '*j*' factors for Lance & offset fins for R1234yf in the following section similar to Ranganayakulu et.al, [15].

5. VALIDATION OF CFD RESULTS

CFD analysis is carried out for the selected L&O fin at different Reynolds number using air. The CFD results are compared with open literature correlations for laminar flows.

The comparison of CFD results with literature correlations are plotted Re vs j-factor. The CFD results are closure to the correlations of Manglik and Bergles. The variations may be due to manufacturing and other measurement errors. The CFD estimated j-factor vs Re and comparison with other authors are shown in Fig.5.

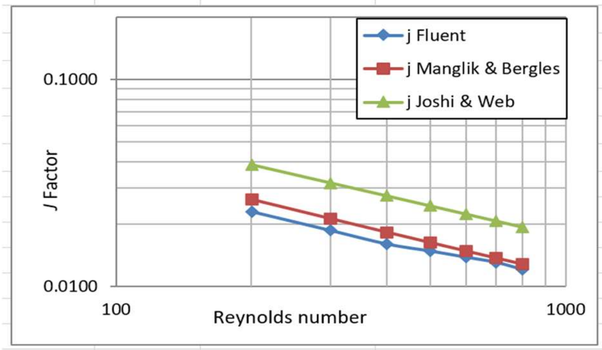


Figure 5: Validation of CFD results

6. RESULTS AND DISCUSSION

For the validation of the numerical analysis conducted in the present study, a grid independence test is carried out for the same fin. CFD results are plotted as the number of elements versus *j* factor in Fig. 3. This figure shows that, the variation in the *j* factor becomes negligible after 4,00,000 cells. Hence the optimized size of cells used in this analysis is about 6,33,447. Using optimized size elements CFD analysis carried out using air for the laminar flow. The results are compared and are good agreement with open literature published data. The validation results are presented in fig.5.

The velocity contour, Pressure streamlines and temperature streamlines for the Reynolds number 300 using R1234yf in vapor is shown in Fig.6, Fig.7 and Fig.8 respectively. From the velocity contours it is quite clear that the flow is more laminar at Reynolds number 300. One more interesting feature is the shape of the vectors, which are pulled towards the interrupted part. This is basically because of the formation of the recirculation zone that is pulling the velocity contours towards it in order to compensate for the excess pressure drop due to the recirculation pocket. The pressure and temperature streamlines are shown in Fig. 7 and Fig.8 respectively for the same fin surface. It is clear that the boundary layer interrupts and fresh boundary layer starts from the interruption. Moreover, high pressure drop occurs at the interruptions and the velocity reaches its maximum value at the same place. In addition, the results obtained from Fluent in the form of Colburn *j* factors are compared with air, water vapor and R134a vapor in Fig.9 for laminar region. According to Hu et al. [15-16], there will be a considerably large deviation in *j* values of air with other selected fluids due to difference in Prandtl number. Hence, CFD analysis is carried out for exact *j* value, which gives the single phase heat transfer coefficient hfor R1234yf and R134a. Fig. 9 provides the *j* values for the L&O fin compact heat exchanger used in this study for air and water vapor, R1234yf and R134a for laminar regions.

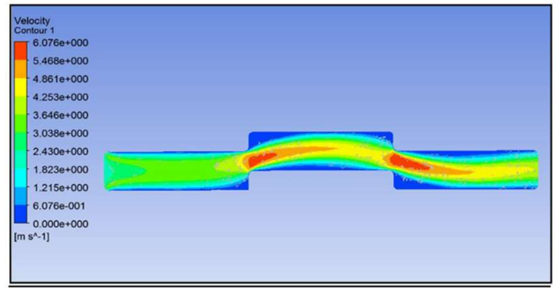


Figure 6: Velocity Contour at Re=300

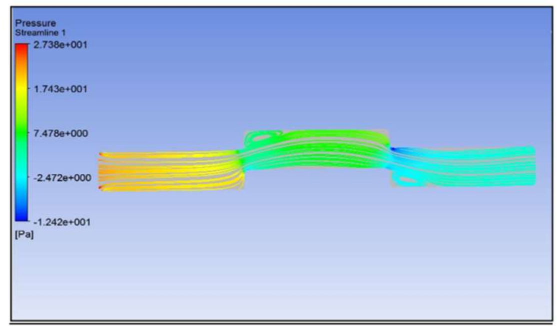


Figure 7: Pressure Streamlines at Re=300

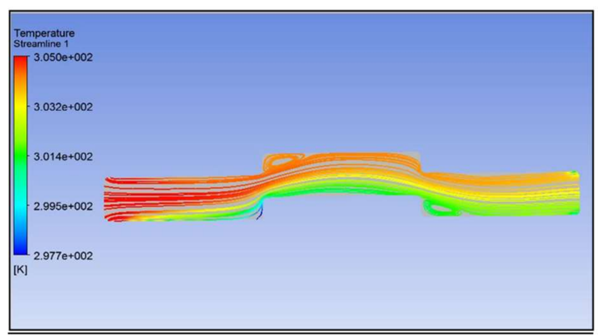


Figure 8: Temperature Streamlines at Re=300

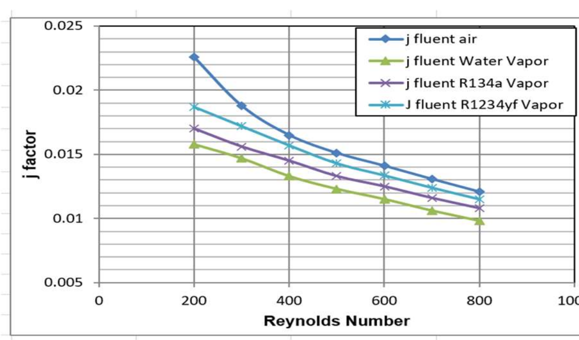


Figure 9: Re vs. *j* factor for different fluids

7. CONCLUSIONS

It is found that the *j* of R134a is lower by about 6-9%% when compared with R1234yf. CFD analysis is not carried out for f factor as there is no significant deviation in *f* values as observed by Hu et al. [15-16]. Even though *j* is lower for R134a, the heat transfer coefficient, h_{R134a} is higher when compared to R1234yf. As Reynolds number increases the Colburn j factor reduces for both the fluids as expected.As explained in section 6.0 the refrigerant side heat transfer coefficient is calculated using a CFD analysis in accordance with Ranganayakulu et. al,[15]. An alternative dimensionless heat transfer coefficient often used for offset fin studies is the Colburn factor *j,* defined as

$$j = {Nu}\Big/\left(Re * Pr^{\frac{1}{3}}\right) \qquad (6)$$

The single phase vapor formrefrigerant R1234yfand R134a side heat transfer coefficient correlation for laminar range (100<Re<800)is as follows:

$$Nu_{R1234} = 0.1256 {Re_{R1234yf}}^{0.6477} Pr^{\frac{1}{3}} \qquad (7)$$

$$Nu_{R134a} = 0.0984 {Re_{R134a}}^{0.675} Pr^{\frac{1}{3}} \qquad (8)$$

Then,

$$h_{R1234} = 0.1256 \left(\frac{\lambda}{D_h}\right) {Re_{R1234yf}}^{0.6477} Pr^{\frac{1}{3}} \qquad (9)$$

$$h_{R134a} = 0.0984 \left(\frac{\lambda}{D_h}\right) {Re_{R134a}}^{0.675} Pr^{\frac{1}{3}} \qquad (10)$$

The values obtained from these expressions are in good agreement with the literature data. The variations are found to be 5-10% in *j* values. The above correlations can be used for the given lance & offset fin for R1234yf. The generated correlations by using CFD analysis can helps to reduce the number of tests and also help to calculate the heat transfer coefficient for different Reynolds number.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A_f exchanger total fin area on one side, m^2
C_p specific heat at constant pressure, J/kg.K
D_h hydraulic diameter, m
f fanning friction factor, dimensionless
G mass flow rate per unit area, kg/sec m^2
h fin height, mm
h heat transfer coefficient (W/m^2k)
j Colburn factor, dimensionless
l lance length, mm
m mass flow rate (kg s^{-1})
Nu Nusselt number (hD_h / λ), dimensionless
Pr Prandtl number ($\mu C_p/\lambda$), dimensionless
Re Reynolds number, dimensionless
s fin spacing, mm
t fin thickness, mm
u velocity component in x direction

Greek Symbols

ε turbulence dissipation rate,m^2/s^3
k turbulent kinetic energy, m^2/s^2
μ dynamic viscosity, Ns/m^2
φ generalized transport variable
ρ density of the water, kg/m^3
Γ effective diffusivity, m^2/s^2
υ specific volume ,m^3/kg

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