International Journal of Thermal Sciences 48 (2009) 2300–2308

Contents lists available at ScienceDirect

International Journal of Thermal Sciences

journal homepage: www.elsevier.com/locate/ijts

Development of heat transfer coefficient correlation for concentric helical coil heat exchanger

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ARTICLE INFO

Article history:
Received 14 October 2008
Received in revised form
24 March 2009
Accepted 18 April 2009
Available online 31 May 2009

Keywords:
Heat transfer coefficient correlation
Mathematical model
CFD model
Optimization

ABSTRACT

The present study deals with developing a Correlation for heat transfer coefficient for flow between concentric helical coils. Existing Correlation is found to result in large discrepancies with the increase in gap between the concentric coils when compared with the experimental results. In the present study experimental data and CFD simulations using Fluent 6.3.26 are used to develop improved heat transfer coefficient correlation for the flue gas side of heat exchanger. Mathematical model is developed to analyze the data obtained from CFD and experimental results to account for the effects of different functional dependent variables such as gap between the concentric coil, tube diameter and coil diameter which affects the heat transfer. Optimization is done using Numerical Technique and it is found that the new correlation for heat transfer coefficient developed in this investigation provides an accurate fit to the experimental results within an error band of 3–4%.

1. Introduction

Helical coil tubes are used in a variety of applications, e.g. thermal oil heating, steam generation, thermal processing plants, food and dairy processing, refrigeration and air conditioning and heat recovery processes. Helical coil tubes are advantageous due to their high heat transfer coefficient and compactness compared to straight tubes. The developments in process industry is mainly driven by the cost and efficiency of heat exchangers, which requires precise and accurate equation for the heat transfer estimation.

Somchai et al. [1] has studied the heat transfer characteristics and performance of a spirally coiled heat exchanger. The correlation used in his paper simulates flow over a coil and tube diameter was considered as hydraulic diameter. The present study focus on the closely spaced helical coil with no pitch or pitch equal to tube diameter, whereas the helical coil configuration studied by Somchai et al. [1] is not closely spaced and has pitch higher than tube diameter. Prabhanjan et al. [2] has studied the heat transfer rates in helically coiled tube for the fluid flowing inside the tube. Paisarn et al. [3] has also reviewed the various published heat transfer coefficient correlation for the fluid flowing inside the tube of helical coil heat exchanger. Rahul et al. [4] obtained experimental results for estimation of the heat transfer coefficient for coiled tube surface in cross flow air. Bharuka et al. [5] has studied the flow through a helically coiled annulus. The flow and heat transfer behaviour between two concentric helical coils has not been documented in open literature. Avina [6] has suggested in his thesis that the flow and heat transfer behaviour for flow over the helical coil can be approximated as flow over the tube bank and the Zukauskas correlation can be used for the heat transfer estimation for flow over the helical coil.

Most studies on helical tubes have been carried out on the heat transfer characteristics of the fluid flowing inside the helical tubes. The objective of this work is to study heat transfer characteristics of the fluid flowing outside (flue gas side) the helical coil and therefore developing a correlation for heat transfer coefficient for the flow between concentric helical coils (helical annulus).

As lot of variations are possible in coil geometry, it requires a lot of experimental data to capture the effect of different physical parameters like tube diameter, coil diameter, coil gap and makes it very expensive, time consuming and difficult. In this work, a Computational Fluid Dynamics (CFD) model has been validated with experimental data and the same has been used to generate data for the various combination of geometrical parameter to reduce time and effort. A comparative study of heat transfer by considering flow over a tube bank and flow in annular space is described and validated in this work. This comparison is used as basis for the development and refinement of the heat transfer equation. The sensitivity analysis is carried out to understand the effect of various design parameters like tube diameter and coil gap.

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1290-0729/$ – see front matter
doi:10.1016/j.ijthermalsci.2009.04.008

Nomenclature

Nu	Nusselt Number
Re	Reynolds Number
P	Prandtl Number
D	Pitch Circle Diameter (m)
d	Tube Diameter (m)

Greek letters

k	Turbulent Kinetic Energy (m^2/s^2)
ε	Turbulent Dissipation Rate (m^2/s^3)

Subscripts

i	inside coil
o	outside coil
h	hydraulic

2. Experimental setup and procedure

Experiment was conducted on a working installation of thermic fluid heater. A schematic of the thermic fluid heater is shown in Fig. 1(a). Thermic fluid heater primarily consists of three major parts namely, furnace, radiant coil and convective coil. Fuel is burnt in furnace and the produced hot flue gas transfers heat to thermic fluid in radiant coil and then enters into convective coil via a connecting duct. After passing over convective coil, it flows through air pre heater, dust collector and finally is exhausted to atmosphere via chimney. In this paper only convective coil heat exchanger is chosen for the analysis and CFD model validation. A detailed sectional view of convective coil heat exchanger is shown in Fig. 1(b). The dimensions of the helical coil heat exchanger are listed in Table 1. Also, a cross section of the three dimensional model of convective coil heat exchanger representing flue gas domain is shown in Fig. 1(c). In first pass of convective coil which is mostly dominated by radiation heat transfer, flue gas travels vertically downward and then takes U turn to enter into the annulus space (second pass) between two helical coils and goes upward. This is the region where radiation as well convection heat transfer takes place and is an area of interest for the present study. Flue gas will again take U turn to enter third pass and travel vertically downwards through the space between jacket or shell and outer coil. Temperature readings for flue gas side were taken using temperature probes located at furnace, inlet and outlet of convective coil.

Fuel firing rate is varied depending on the process load of the thermic fluid heater. Flue gas temperature at the inlet and outlet of the convective coils is recorded using temperature probes for different load conditions. Fuel quantity and flue gas composition are recorded at different load conditions to calculate flue gas quantity. Flue gas quantity and the flue gas inlet temperature are used to provide inlet boundary condition for CFD model. The experimental data obtained from working installation is listed in Table 2.

3. Computational resource description

Three dimensional Model of the fluid domain used for analysis is built using Solid Edge V20. Grid is generated in Gambit 2.4.6. Meshed model is then exported to Fluent 6.3.26 [8] for analysis. A HP workstation XW6400 having the following configuration is used – Processor – 2x Intel(R) Xeon(R) dual core CPU, RAM – 8 GB, Hard Disk – 160 GB, Graphic Card – NVIDIA QUADRO FX 1500.

4. CFD model of a working heater for validation of approach

The basic objective of this analysis is to validate CFD model with experimental results. CFD analysis has been performed for a working installation for the validation of approach. The selected heat exchanger is a three pass thermal oil heater as shown in Fig. 1(b), where flue gas is used to heat thermal oil. This heat exchanger consists of two helical coils and a jacket or shell. The flue gas enters in the inner helical coils, where radiation is the dominant mode of heat transfer, and then it passes between two helical coils and finally exits from the annular section bounded by outer coil and the jacket. Initially k–ε standard turbulence model is used and after some iterations, switched to k–ε realizable turbulence model as the geometry is having the flow features that include strong streamline curvature, vortices, and rotation. Distance between wall and the nearest cell centroid (wall y^+) is found in the range of 80–120. Standard Wall function is used to capture boundary layer. In the first pass radiation will be the dominant mode of heat transfer and hence accounted for the same by using Discrete Ordinate (DO) model for radiation heat transfer. For calculation of absorption coefficient Weighted Sum of Grey Gas Model (WSGGM) is used. For fluid material a mixture material comprising of CO_2, H_2O, SO_2, N_2, O_2 is used based on experimental data. Analytically, metal temperatures are calculated and used as wall boundary condition.

Following boundary conditions are used for CFD analysis.

1) Inlet boundary condition – Mass flow rate type is chosen. Temperature, mass fractions, turbulence intensity and hydraulic diameter for backflow parameters are specified.
2) Outlet boundary condition – Pressure outlet boundary condition is used. Atmospheric pressure, turbulence intensity and hydraulic diameter at the outlet are specified.
3) Wall Boundary Condition – Wall boundary condition is used. For momentum no slip condition at wall and for thermal boundary condition constant temperature is specified. Thermic fluid temperature at the inlet and outlet of the coil and thermal oil side heat transfer coefficient are used to estimate wall temperature to provide isothermal wall boundary condition for CFD model.

Thermal fluid heater is simulated using CFD. Figs. 2 and 3 represent the temperature and velocity profile of the thermal oil heater respectively. This simulation is done at the various load conditions and compared with data collected from the working installation. Complete set of data is recorded when the working installation has reached steady state conditions. Fig. 4 represents the comparison of CFD results and the experimental results. CFD model for this working installation has been found fairly accurate with maximum error of 5%.

5. CFD model for flow between two helical coils

As the above CFD model for a working heater has been validated against experimental results, it acts as a benchmark for further development of heat transfer coefficient correlation for the flow between two helical coils. The objective of the model as shown in Fig. 5(a) is to estimate the heat transfer coefficient for the flow between two helical coils. The second pass of flue gas which is the region of present study is shown in Fig. 5(b). The approach assumed in the previous section is exactly followed but the effect of radiation is not considered so as to study the effect of convection only.

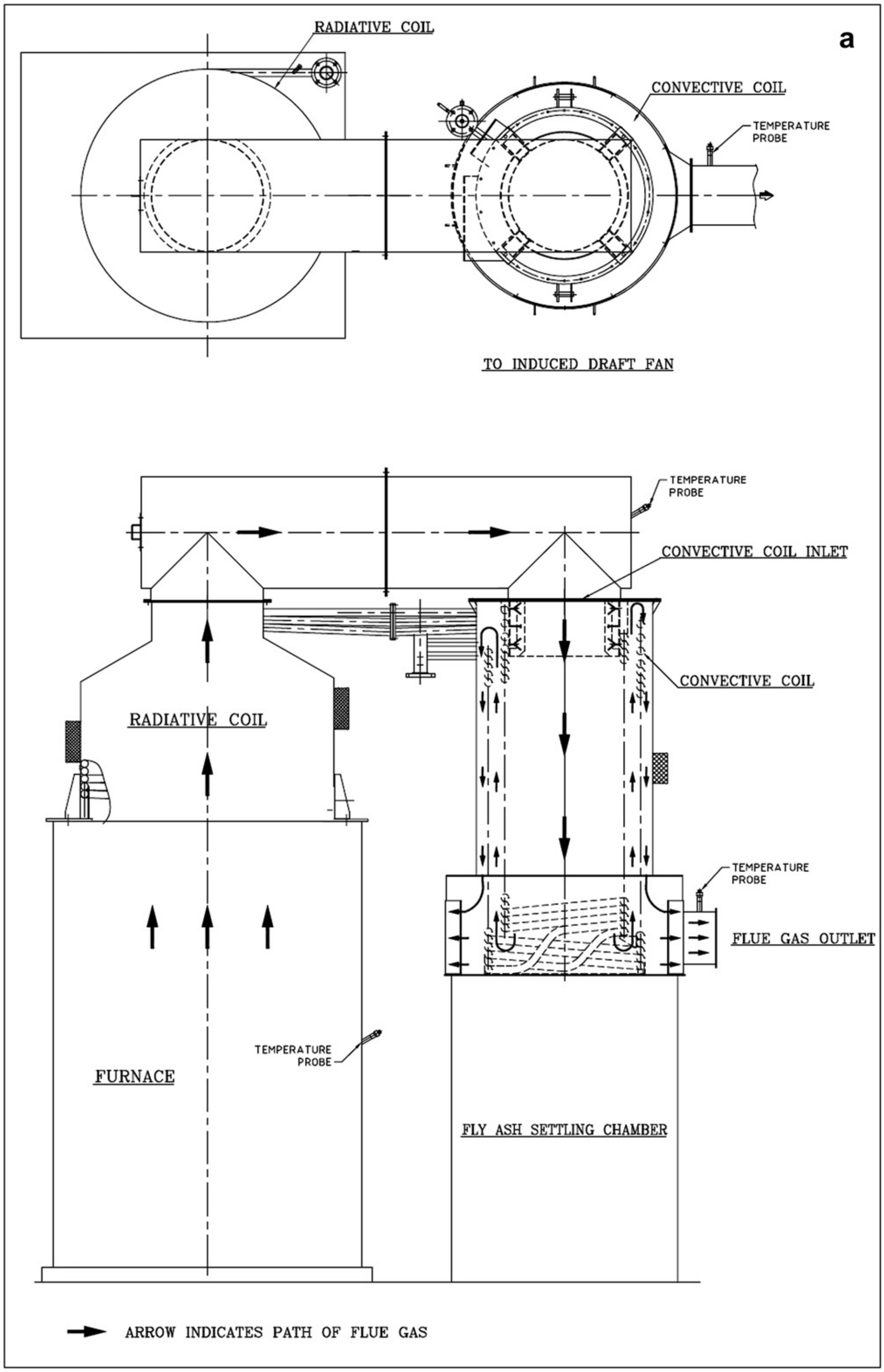
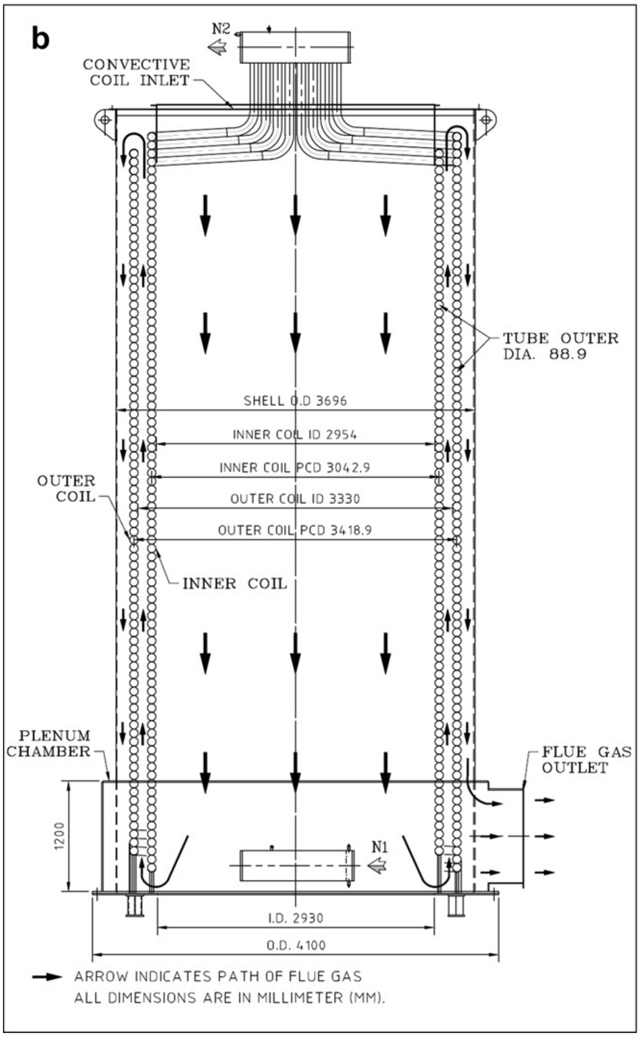


Fig. 1. (a) A schematic of the working model of a thermic fluid heater. (b) Detailed View of the convective coil heat exchanger. (c). A section of the convective coil heat exchanger representing flue gas domain.

6. Methodology

Initially the CFD model is developed for a working thermal oil heater and the result is validated with available experimental data. The objective of this exercise is to develop a heat transfer equation applicable for wide range of velocity. A set of geometry is developed to capture the effect of various design parameter like coil diameter, tube diameter and coil gap. The diameter of inner coil is fixed and the coil gap is varied to understand the effect of coil gap and the same is repeated for different tube diameter. The similar exercise is repeated for varying inner coil diameter. In this way a set of coil geometry configurations has been generated for CFD analysis. Selected tube diameter varies from 38.1 mm to 88.9 mm with coil gap from 30 mm to 140 mm. These coil geometry



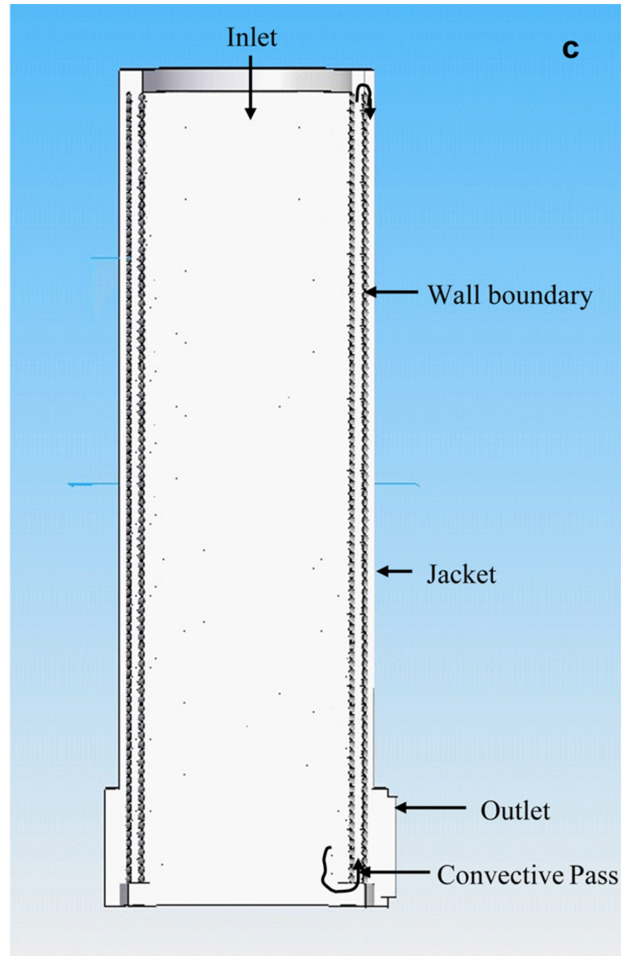


Fig. 1. (*continued*).

configurations are analyzed using CFD and the temperature profiles are noted.

Initially flow between two helical coils is approximated as, flow over the tube bank and annular flow respectively. Then it is compared for the complete range of Reynolds number, to develop the basis of heat transfer coefficient correlation. This correlation is then refined and evaluated for the complete range of Reynolds number. The final correlation is developed taking into account the effect of various design parameters on the heat transfer performance.

Table 1
Dimensions of the helically coiled heat exchanger.

Inner coil tube outside diameter, mm	88.9
Inner coil tube inside diameter, mm	88.84
Outer coil tube outside diameter, mm	88.9
Outer coil tube inside diameter, mm	88.84
Pitch circle diameter of inner coil, mm	3042.9
Pitch circle diameter of outer coil, mm	3418.9
Innermost diameter of inner coil, mm	2954
Innermost diameter of outer coil, mm	3330
Number of turns of inner coil	88
Number of turns of outer coil	88

Table 2
Experimental data.

Load %	Mass flow rate of flue gas (kg/s)	Inlet Temperature of flue gas (°C)	Outlet temperature of flue gas (°C)
First set of reading			
100	6.25	1007	392
80	5	985	375
60	3.75	963	358
40	2.5	939	339
Second set of reading			
100	6.25	1013	398
80	5	978	368
60	3.75	955	350
40	2.5	945	345
Third set of reading			
100	6.25	1010	395
80	5	987	377
60	3.75	970	365
40	2.5	942	342
Fourth set of reading			
100	6.25	1005	390
80	5	983	373
60	3.75	964	359
40	2.5	942	342

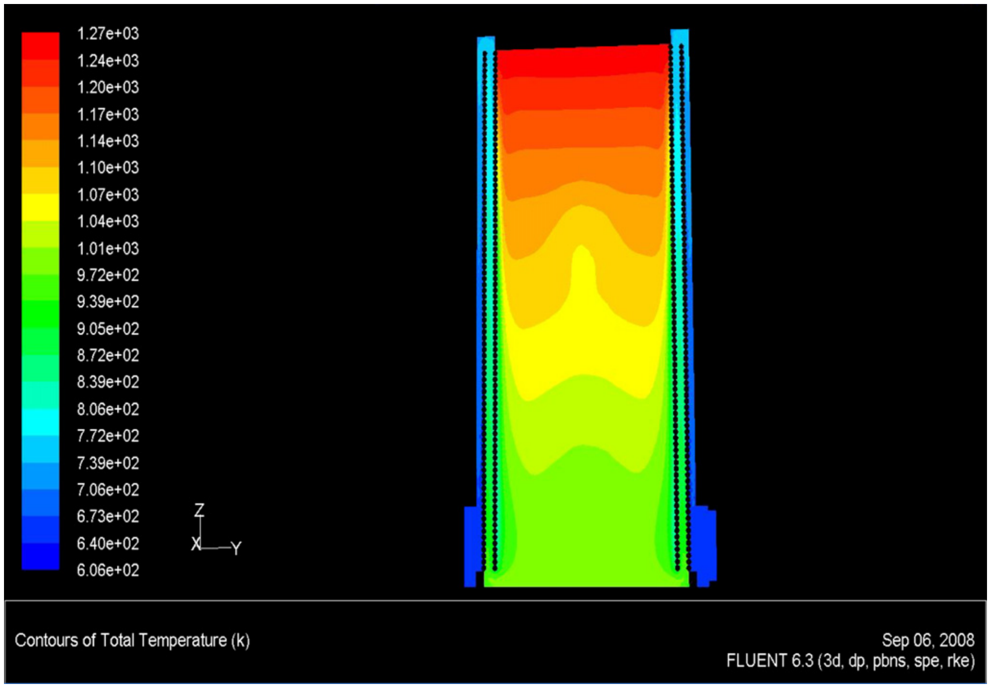


Fig. 2. Temperature.

7. Analysis

The flow profile between two coils can be either approximated as flow over tube banks or flow in annular channel. The CFD results are compared with the analytical result using these two flow profiles. The basic objective of this analysis is to understand the proximity of actual flow profile with the suggested two flow profiles.

To evaluate the tube bank correlation for the analysis of heat transfer between two helical coils, a comparison has been done between CFD results and tube bank correlation results. Nusselt number is plotted against the Reynolds number as shown in Fig. 6. The following correlations are used for the Nusselt number calculation, approximating flow over the tube bank[7].

$$Nu = 0.27\, Re^{0.63} Pr^{0.36} \text{for } Re < 200000 \tag{1}$$

$$Nu = 0.033\, Re^{0.8} Pr^{0.36} \text{for } Re > 200000 \tag{2}$$

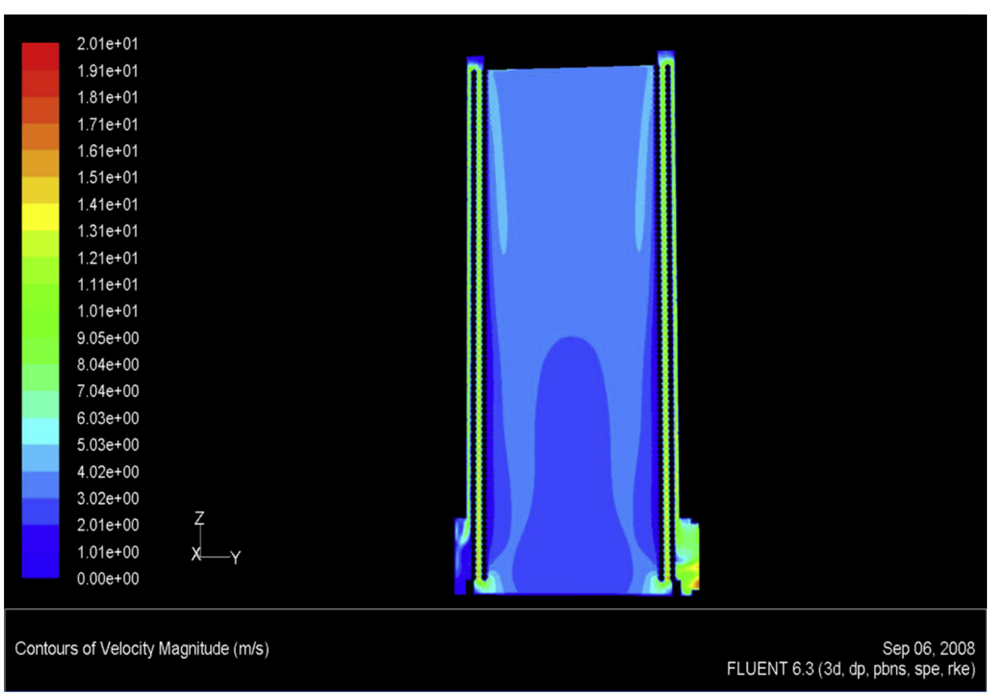


Fig. 3. Velocity Profile.

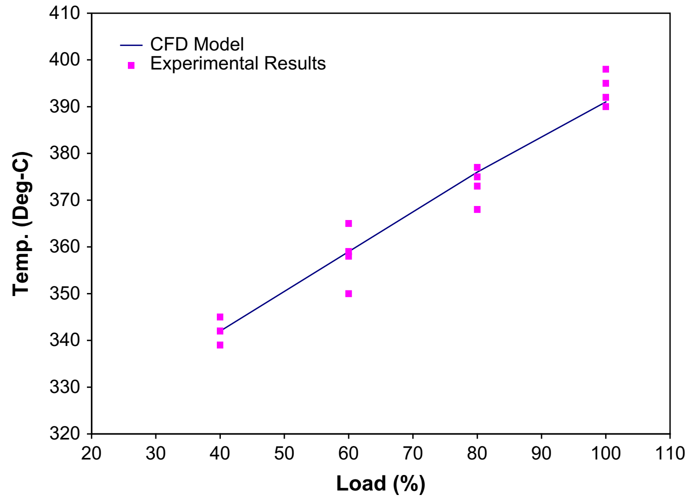


Fig. 4. Plot of Temperature vs Load.

Fig. 6 indicates a significant difference between analytical results (tube bank correlation [7]) and CFD results and the absolute error is estimated as 117.95%. This also indicates that tube bank equation over estimates heat transfer performance. It is also found that the correlation of Nusselt number calculated by using CFD results against Reynolds number is not consistent. The circled portion on the graph indicates very sharp change in Nusselt number in comparison with analytical results. This encircled set of data represents constant pitch circle diameter of inner coil, constant tube diameter with variation in coil gap. Coil gap is varied by varying pitch circle diameter of outer coil. This indicates that Nusselt number is quite sensitive with coil gap, which is ignored in tube bank equation.

The correlation between Nusselt and Reynolds number can be represented as follows.

$$Nu = a(Re)^b \tag{3}$$

To understand this relationship, logarithmic value of Nusselt number and Reynolds number is plotted in Fig. 7 and the slope of the graph indicates the power exponent. It indicates that different power exponent can give fairly correct correlations between Reynolds number and Nusselt number.

This indicates the possibility of refinement in correlations by changing power exponent of Reynolds number. This refinement will not completely overcome the drawback of this approximation, as it still ignores the effect of coil gap on the Nusselt number.

A similar analysis is then conducted to evaluate the annular flow correlation for the analysis of heat transfer between two helical coils. A comparison has been done between CFD results and annular flow correlation results. Nusselt number is plotted against the Reynolds number as shown in Fig. 8. Following equation is used for heat transfer calculation for annular flow [7].

$$Nu = 0.023\, Re^{0.8} Pr^{0.3} \tag{4}$$

$D_h = D_o - D_i$

D_h = Hydraulic diameter

D_o = Pitch circle diameter of outer coil

D_i = Pitch circle diameter of inner coil

Fig. 8 indicates comparatively less difference between analytical (annular flow equation (7)) results and CFD results, with an absolute value of 29.8% error. This also indicates fairly consistent correlation between Nusselt number and Reynolds number as per

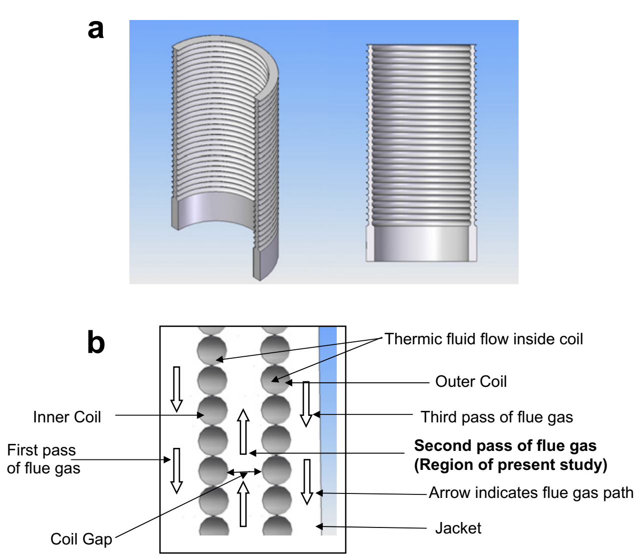


Fig. 5. (a). Heat Exchanger Model for CFD simulations. (b). Closed view of working heat Exchanger model.

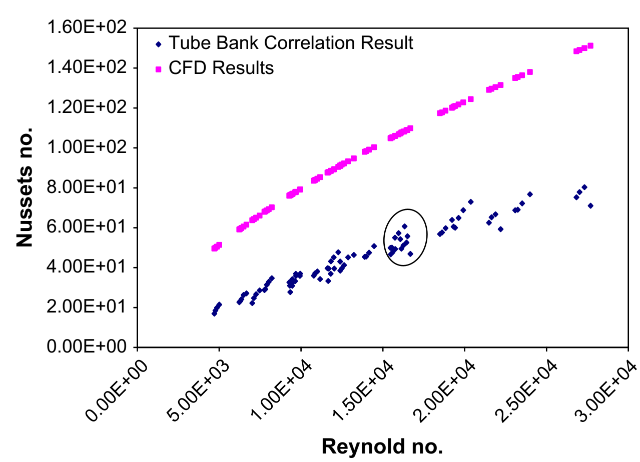


Fig. 6. Comparative Analysis of CFD and analytical results based on tube bank correlations.

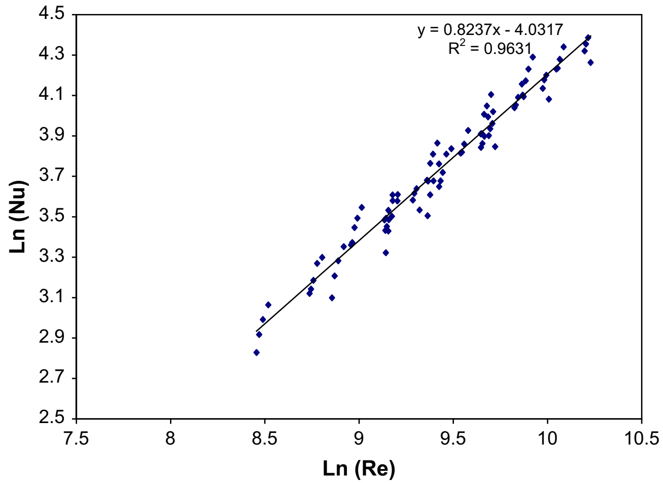


Fig. 7. Logarithmic plot of Nusselt Number Vs Reynolds Number with tube bank approximation.

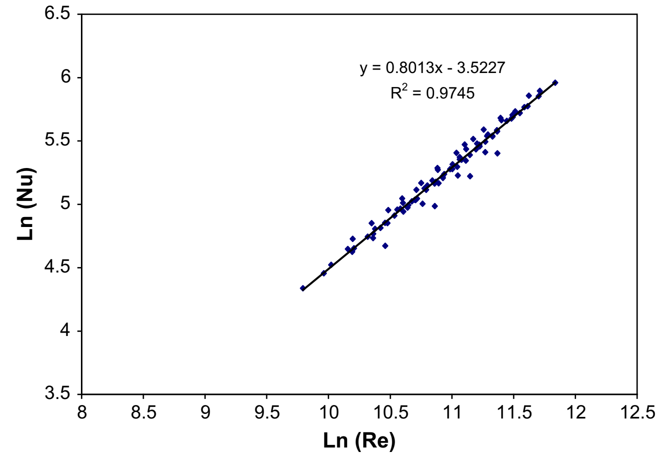


Fig. 9. Logarithmic plot of Nusselt number vs Reynolds Number.

both CFD results and annular flow correlation. In this case hydraulic diameter captures the effect of the gap. The difference between annular flow correlation results and CFD results are very less in the case of higher tube diameter and lower velocity. In these cases flow between two coils can be fairly approximated as annular flow, as the higher diameter tube approached toward a flat annular surface.

The correlation between logarithmic value of Nusselt number and Reynolds number is plotted, as shown in Fig. 9. This approves the annular flow equation results, as the power exponent in this case has been found equal to the power exponent considered in analytical correlation with fairly good regression coefficient. This graph also indicates better regression coefficient in the case of annular flow approximation in comparison with tube bank correlations.

All the three heat transfer coefficient values are plotted on the same graph to understand their comparative standing as shown in Fig. 10. This indicates that the tube bank correlations over estimates the heat transfer performance and the error is significantly high. The annular flow approximation is quite close with CFD results and slightly under estimates the heat transfer performance. This can be primarily attributed to the turbulence induced due to the circularity on the annular surface.

This suggests that the nature of the heat transfer coefficient equation for annular flow should be taken as basis for the development of new equation, as the analytical result with this assumption is quite close to CFD results and the Nusselt number calculated from CFD analysis has a definite and consistent correlation with Reynolds number defined for annular flow. As the error is approximately 29%, the equation needs a refinement. The least square technique can be used for the calculation of the various power exponent and coefficient.

The hydraulic diameter should be calculated similar to annular flow and the nature of equation can be as follows.

$$Nu = CRe^{m}Pr^{n} \tag{5}$$

The value of C, m and n are calculated using least square method and the results are plotted against Reynolds number for comparison with CFD results as shown in Fig. 11.

The equation seems quite accurate in the complete range of the Reynolds number, but the error at few points is quite significant and cause of concern. The average value of absolute error is 3.8% with maximum error of 21.3%. To understand the effect of coil

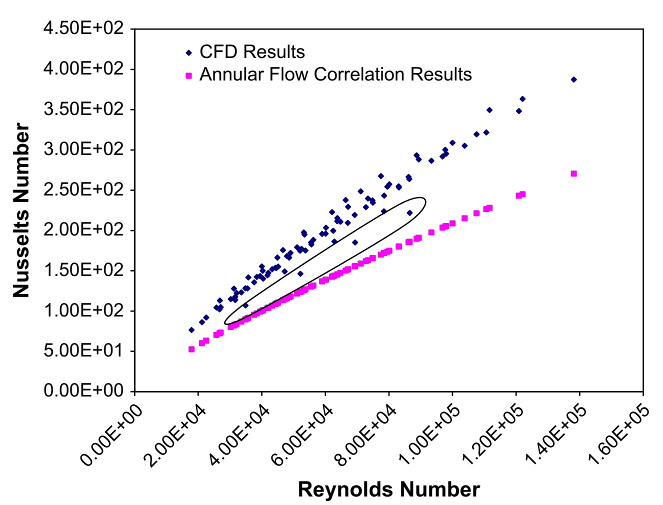


Fig. 8. Comparative analysis of CFD and analytical results based on annular flow correlation.

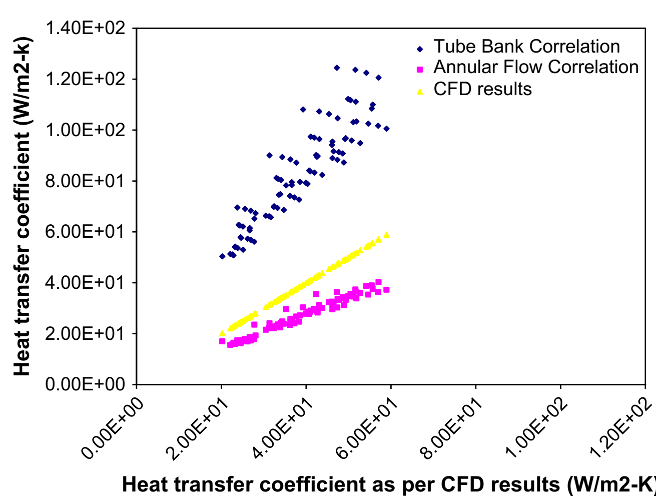


Fig. 10. Comparison of Heat Transfer Correlation and CFD Results.

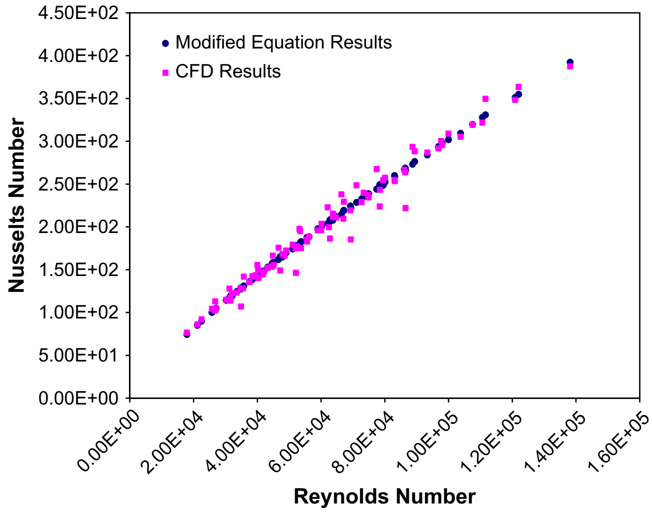


Fig. 11. Comparison of Nusselt Number with Refined Correlation and CFD results.

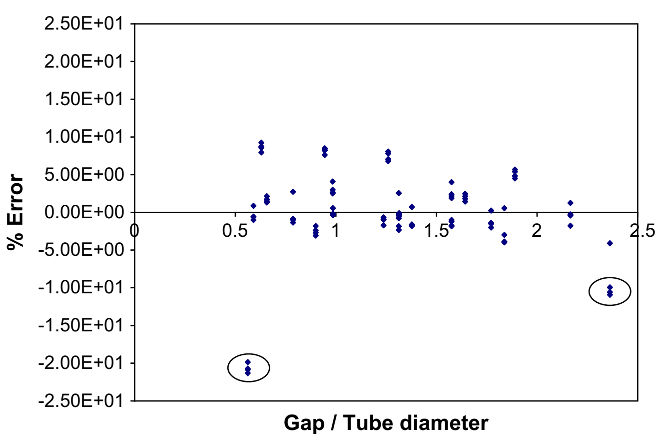


Fig. 12. Error Analysis of Modified Equation.

geometry on the error, it is plotted against the ratio of coil gap and tube diameter as shown in Fig. 12. The encircled set of data indicates very high error and lies at the highest and lowest point of ratio. These two sets of the data is for the highest tube diameter

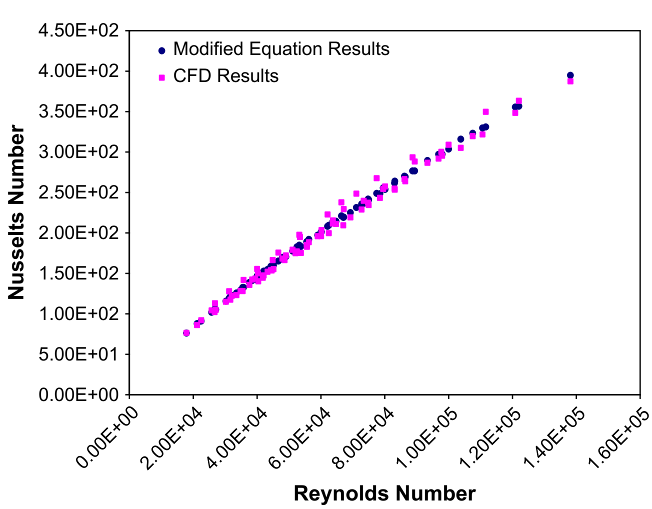


Fig. 13. Comparison of Nusselt Number from Modified Equation with CFD results.

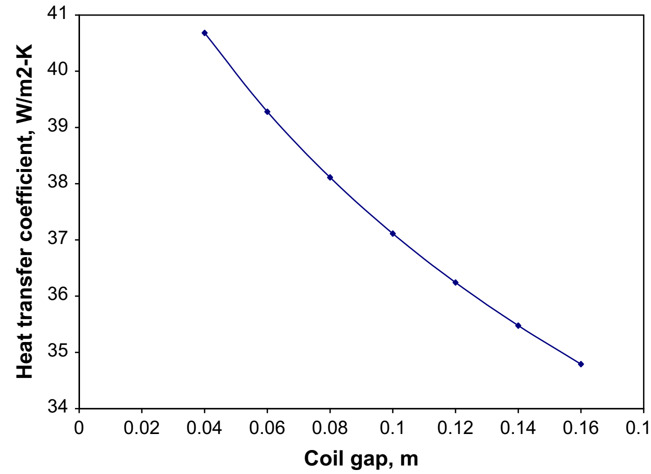


Fig. 14. Effect of Coil Gap on Heat Transfer Coefficient.

with the lowest gap and the lowest tube diameter with the highest gap stretching the specified ratio to extreme ends.

To achieve the better equation, these extreme data can be filtered and one more dimensionless parameter can be introduced to capture the effect of coil gap and tube diameter ratio. This new dimensionless parameter can be named as gap ratio and can be defined as follows.

$$\text{Gap ratio} = (D_0 - D_i)/d \tag{6}$$

where,

D_0 Pitch circle diameter of outer coil.
D_i Pitch circle diameter of inner coil.
d Tube diameter.

The new modified equation can be expressed as follows.

$$Nu = 0.02652604\, Re^{0.834694285} Pr^{0.3} (\text{Gap ratio})^{-0.096856199} \tag{7}$$

The value of these coefficients and power exponents are calculated by using least square technique. New set of Nusselt numbers is plotted against Reynolds number to compare it with the CFD results as shown in Fig. 13.

The new modified equation seems fairly accurate in the complete range of the Reynolds number. The average value of absolute error is reduced to 2.57% and maximum error is reduced to 7.2%.

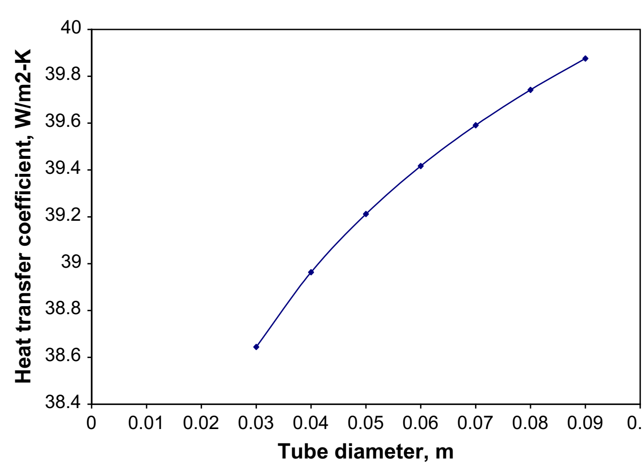


Fig. 15. Effect of tube diameter on Heat Transfer Coefficient.

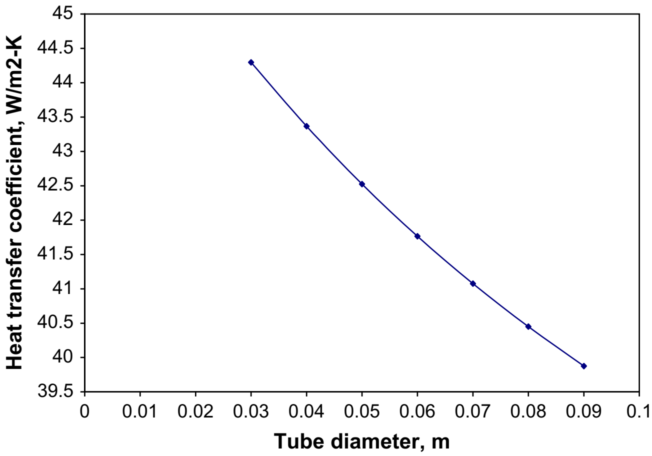


Fig. 16. Effect of Tube Diameter with constant coil gap on Heat Transfer Coefficient.

8. Results and discussion

The modified equation can be used to analyze the effect of different design parameter on the heat transfer coefficient. Two most important design parameters are coil gap and tube diameter for the study of their effect on heat transfer performance. To analyze the effect of coil gap on the heat transfer coefficient, the value of heat transfer coefficients are evaluated with different coil gap keeping inner coil pitch circle diameter, tube diameter and velocity constant. These heat transfer coefficients are plotted against coil gaps as shown in Fig. 14. This indicates that the heat transfer coefficient decreases with the increase in gap. This seems logical as the higher turbulence can be expected at lower coil gap, which will finally result in higher heat transfer coefficient.

To analyze the effect of tube diameter on the heat transfer coefficients, the heat transfer coefficients are evaluated for the different tube diameter keeping inner coil pitch circle diameter, outer coil pitch circle diameter and velocity constant. Inner and outer coil diameters are kept constant to neutralize the effect of hydraulic diameter on the heat transfer coefficient. These results are plotted against tube diameter as shown in Fig. 15. This reveals that the heat transfer coefficient increases with the increase in tube diameter. This result is quite interesting and does not agree with the expected flow behaviour. This is primarily due to reduction in coil gap with increasing tube diameter and the effect of tube diameter is not dissociated with the effect of coil gap. The effect of coil gap seems dominating over the effect of the tube diameter.

To dissociate the effect of the coil gap from the effect of the tube diameter, a set of data is generated using the same inner coil diameter but the outer circle diameter is varied with tube diameter to keep coil gap constant. These results are again plotted against tube diameter as shown in Fig. 16.

This plot indicates that the heat transfer coefficient decreases with the increase in tube diameter. This is as per expectation, as the higher tube diameter approaches towards a flatter annular surface with lower degree of turbulence.

9. Conclusions

One extra parameter has been introduced to capture the strong correlations between coil gap and heat transfer coefficient. A wide range of data has been analyzed, which covers a wide range of the Reynolds number from 20 000 to 150 000. It is found that the extreme range of data identified by the ratio of coil gap and tube diameter can introduce significant error in the equation. These extreme data is filtered to develop a better equation. As these data is filtered, the equation is not valid in this extreme range. The developed equation is only valid, if the specified ratio (Coil gap/Tube diameter) is from 0.55 to 2.25. This covers the most of the practical range of the helical coil heat exchanger application.

Acknowledgement

We are thankful to G.S. Deshpande and A. Krishnakumar for their continuous support and encouragement. Also the support from Abhay Mane and C.P. Prajeendran during our project tenure is gratefully acknowledged.

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