T H E A R C H I V E O F M E C H A N I C A L E N G I N E E R I N G

VOL. LIX 2012 Number 3

10.2478/v10180-012-0019-9

Key words: numerical, helical, heat exchanger, heat transfer

MOHAMED ABO ELAZM *, *AHMED RAGHEB* *, *AHMED ELSAFTY* **,
MOHAMED TEAMAH ***

COMPUTATIONAL ANALYSIS FOR THE EFFECT OF THE TAPER ANGLE AND HELICAL PITCH ON THE HEAT TRANSFER CHARACTERISTICS OF THE HELICAL CONE COILS

This numerical research is devoted to introducing the concept of helical cone coils and comparing the performance of helical cone coils as heat exchangers to the ordinary helical coils. Helical and spiral coils are known to have better heat and mass transfer than straight tubes, which is attributed to the generation of a vortex at the helical coil. This vortex, known as the Dean Vortex, is a secondary flow superimposed on the primary flow. The Dean number, which is a dimensionless number used in describing the Dean Vortex, is a function of Reynolds Number and the square root of the curvature ratio, so varying the curvature ratio for the same coil would vary the Dean Number. Numerical investigation based on the commercial CFD software fluent is used to study the effect of changing the structural parameters (taper angle of the helical coil, pitch and the base radius of curvature changes while the height is kept constant) on the Nusselt Number, heat transfer coefficient and coil outlet temperature. Six main coils having pipe diameters of 10 and 12.5 mm and base radius of curvature of 70, 80 and 90 mm were used in the investigation. It was found that, as the taper angle increases, both Nusselt Number and the heat transfer coefficient increase, also the pitch at the various taper angles was found to have an influence on Nusselt Number and the heat transfer coefficient. A MATLAB code was built to calculate the Nusselt Number at each coil turn, then to calculate the average Nusselt number for all of the coil turns. The MATLAB code was based on empirical correlation of Manlapaz and Churchill for ordinary helical coils. The CFD simulation results were found acceptable when compared with the MATLAB results.

1. Introduction

Helical coils have been long and widely used as heat exchangers in power, petrochemical, HVAC, chemical and many other industrial processes.

* *Arab Academy for Science, Technology and Maritime Transport, Alexandria, Egypt; e-mail: a.ragheb.9@gmail.com*

** *American University of the Middle East, Kuwait.*

*** *Faculty of Engineering, Alexandria University, Alexandria, Egypt.*

Helical and spiral coils are known to have better heat and mass transfer compared to straight tubes, the reason for that is the formation of a secondary flow superimposed on the primary flow, known as the Dean Vortex [1]. The Dean Vortex was first observed by Eustice; then numerous studies have been reported on the flow fields that arise in curved pipes (Dean, White, Hawthorne, Horlock, Barua, Austin and Seader) [2]. The first attempt to mathematically describe the flow in a coiled tube was made by Dean. He found that the secondary flow induced in curved pipes (Dean Vortex) is a function of Reynolds Number and the curvature ratio. The Dean Number is widely used to characterize the flow in curved tubes:

$$\mathrm{De} = \mathrm{Re} * \sqrt{\frac{a}{R}} \tag{1}$$

It has been widely observed that the flow inside coiled tubes remains in the viscous regime up to a much higher Reynolds Number than that for straight tubes Srinivasan et al. [1]. The curvature-induced helical vortices (Dean Vortex) tend to suppress the onset of turbulence and delay transition. The critical Reynolds Number, which describes the transition from laminar to turbulent flow, is given by some correlations; the following correlation is given by Srinivasan et al.[1]:

$$\mathrm{Re}_{\mathrm{cr}} = 2100 * \left(1 + 12\sqrt{\frac{a}{R}}\right) \tag{2}$$

Dennis and Ng [3] numerically studied laminar flow through a curved tube using a finite difference method with emphasis on two versus four vortex flow conditions. They ran simulations in the Dean range of 96 to 5000. The four vortex solutions would only appear for a Dean number greater than 956. Dennis and Riley [4] developed an analytical solution for the fully developed laminar flow for high Dean Numbers. Though they could not find a complete solution to the problem, they stated that there is strong evidence that at high Dean Numbers the flow develops into an inviscid core with a viscous boundary layer at the pipe wall.

The effect of pitch on heat transfer and pressure drop was studied by Austin and Soliman [5] for the case of uniform wall heat flux. The results showed significant pitch effects on both the friction factor and the Nusselt Number at low Reynolds Numbers, though these effects weakened as the Reynolds number increased. The authors suggested that these pitch effects are due to free convection, and thus decrease as the forced convection becomes more dominant at higher Reynolds Numbers. The effect of the pitch on the Nusselt Number in the laminar flow of helicoidal pipes was also investigated by Yang et al [6] Numerical results for fully developed flow with a finite pitch showed that the temperature gradient on one side of the pipe will increase with increasing torsion; however, the temperature gradient on the opposite side will decrease. Overall, the Nusselt Number slightly decreases with increasing torsion for low Prandtl Numbers, but significantly decreases with larger Prandtl Numbers. On the other hand, Germano [7] introduced an orthogonal coordinate system to study the effect of torsion and curvature on the flow in a helical pipe. In the results of the perturbation, the method indicated that the torsion had a second order effect and curvature had a first order effect on the flow. Further studies by Tuttle [8] indicated that the frame of reference (coordinate system) determines if the torsion effect is first or second order.

Kalb and Seader [9] numerically studied the heat transfer in helical coils in the case of uniform heat flux using an orthogonal toroidal coordinate system. They found that, for Prandtl Numbers greater than 0.7, the local Nusselt Number in the area of the inner wall was always less than that of a straight tube, and increasing less as the Dean Number is increased till it reached a limiting value. The local Nusselt Numbers on the outer wall continued to increase with increasing Dean Number. Fully developed laminar flow and heat transfer was studied numerically by Zapryanov et al. [10] using a method of fractional steps for a wide range of Dean (10 to 7000) and Prandtl (0.005 to 2000) numbers. Their work focused on the case of constant wall temperature and showed that the Nusselt number increased with increasing Prandtl numbers, even for cases of the same Dean number.

Spiral coils have received little attention compared to helical coils, though the reported results of spiral coils show better performance than the helical ones. Figueiredo and Raimundo [12] experimentally investigated the thermal response of a hot-water store and the thermal discharge characteristics from heat exchanger coils placed inside. The classical cylindrical coil and the flat spiral coil were investigated. The results indicated that the efficiency of the flat spiral coil was higher than that of a cylindrical one. The results of the comparison have shown good agreement between the model and the experiments. Naphon and Suwagrai [13] studied the Effect of curvature ratios on the heat transfer in the horizontal spirally coiled tubes, both experimentally and numerically, they have found that due to the centrifugal force, the Nusselt number and pressure drop obtained from the spirally coiled tube are, respectively, 1.49 and 1.50 times higher than those from the straight tube.

Helical cone coils have received a lower attention than spiral coils. Only very few researchers have investigated the capabilities of these coils because, due to the complexity of the structure, it was hard to investigate them both numerically and experimentally. Yan Ke et al. [14] have investigated the helical cone tube bundles both numerically and experimentally. With some still foregoing experiments, the authors have found that the cone angle has a significant effect on enhancing the heat transfer coefficient, and also that the pitch has nearly no effect on the heat transfer.

The aim of this paper is to further investigate numerically the effect of the taper angle on Nusselt Number and the heat transfer coefficient for helical cone coils, also to further investigate the effect of the pitch on the heat transfer for these coils. Finally, we will try to optimize the helical cone coils and provide principle formulation for it.

2. Numerical simulation

2.1. Helical cone coil geometry

The geometry of the helical cone tube is shown in Fig. 1; both the curvature and torsion are variable along the tube. The bottom radius of curvature is denoted (R), the pipe diameter (a), the helical pitch as (P), the straight height (H) and finally the inclined height (I). For a straight helical coil the height (H) will be equal to (I) but when changing the inclination angle (θ), the height of the coil (I) will change in accordance to that angle, while keeping (H) constant.

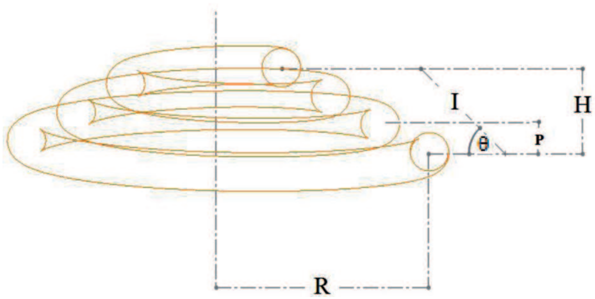

Fig. 1. Helical coil geometry

Three bottom radii of curvatures (R) were used 50, 75 and 100 mm, also two pipe diameters (a) were used 10 and 12.5 mm. So as to keep the height of the coil (H) constant, the height (I) which changes with respect to the taper angle (θ) was proposed. It should be noted that the helical spiral coil is mainly optimized to be used as a condenser (dehumidifier) for a solar HDH desalination unit.

2.2. Simulation model

The laminar flow in the helical spiral coil was simulated using the commercial CFD software Fluent. In the simulation of the laminar fluid flow, the flow and pressure equations were solved with SIMPLEC algorithm, which is one of the three widely, used velocity pressure coupling algorithm in Fluent. The Second Order Upwind algorithm was employed in the discretization of the equations because of its accuracy and iterating efficiency. The parameters of laminar fluid flow model were in accordance with the default values of the CFD software:

$$P_{urf} = 0.3 M_{urf} = 0.7 \tag{3}$$

Where, the P_{urf} and M_{urf}, respectively denote the Under Relaxation Factor of pressure and momentum of the fluid flow inside the tube during the iterating of the calculation. The commercial software Fluent uses both Navier – Stocks equation, continuity equation and the energy equation in the solution, and the equations are solved for laminar, steady and 3D flow.

2.3. Model verification

In order to verify the accuracy of the mathematical model being investigated, a preliminary mathematical model for a helical cone coil was built and its results were compared to the experimental results of that coil. Table 1 shows the coil data for verification of the agreement between the numerical and the experimental model. During the experiment, vapor at various temperatures was admitted to heat water inside the coil The vapor temperature and the water temperature at the coil inlet from the experimental testing were used in the numerical model to see the difference between the results of the numerical model and the experimental results. Fig. 2 shows both the numerical and the experimental results. It can be clearly seen that the difference between the coil exit temperature obtained from both the numerical and that from experimental results is less than 10%.

Table 1.

Verification helical cone coil data

P (mm)	R (mm)	a (mm)	Re	u (m/s)	t (mm)	H (mm)	I (mm)
20	95	11	6360	0.58	0.7	40	85

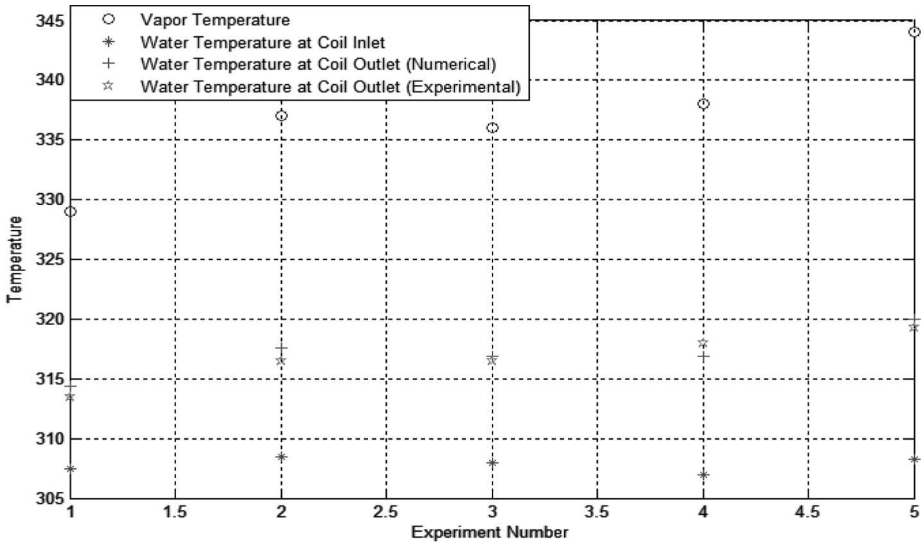

Fig. 2. Model verification with experrimantal results

3. Results and discussion

3.1. Taper angle

Twenty two cases were used to study the effect of the taper angle on the heat transfer coefficient and Nusselt Number. Table 2 shows the details of these cases. To facilitate understanding the results, each bottom radius of curvature (R) and pipe diameter (a) will be discussed separately, then a comparison between them will be made to have a complete understanding for the effect of the taper angle on each case, also to know how to optimize each case.

Table 2.

Helical coil details

R (mm)	a (mm)	Re	u (m/s)	T_{wall} (k)	θ	I (mm)
70	8	1595	0.1	360	0, 20, 40	25, 26.6, 32.64
	12	2392			0, 25, 40	25, 27.58, 32.64
80	8	1595			0, 20, 50, 60	25, 26.6, 38.9, 50
	12	2392			0, 20, 50, 70	25, 26.6, 38.9, 73.1
90	8	1595			0, 25, 45, 60	25, 27.58, 35.4, 50
	12	2392			0, 20, 40, 70	25, 26.6, 32.64, 73.1

For R = 70, 80, 90 mm and a = 8 mm, Figs 3a, 4a and 5a show the increase in the outlet temperature of the coil with increasing the taper angle (θ). This increase is attributed to the increase in the heat transfer surface area increasing with the taper angle. Figures 3b, 4b and 5b show that Nusselt Number increases with increasing taper angle too. Finally, Figs 3c, 4c and 5c shows the increase in the heat transfer coefficient when increasing the taper angle. It is worth noting that the heat transfer coefficient, Nusselt number and coil outlet temperature increases with increasing taper angle while utilizing lower space (similar diameter and height as the ordinary helical coil, but the taper angle decrease overall space). It has a significantly positive effect on the space utilization in the industrial applications. It should be noted that the increase in the Nusselt Number and heat transfer coefficient is logical, as the Nusselt Number varies directly with the Dean Number (De), which varies directly with the curvature ratio (a/R). So, as (R) decreases when increasing the taper angle, the curvature ration increases too.

Finally, Fig. 6a, 6b, 6c represents the angles (0, 20, 40), respectively, for R = 70. These figures show the change in the velocity due to the change in the taper angle. As it can be seen, they indicate that the velocity increases, which means that Reynolds Number increases and thus the Dean Number increases as well. Also, it could be noted that the center of the main flow is shifted towards the outwards of the pipe (Dean Vortex).

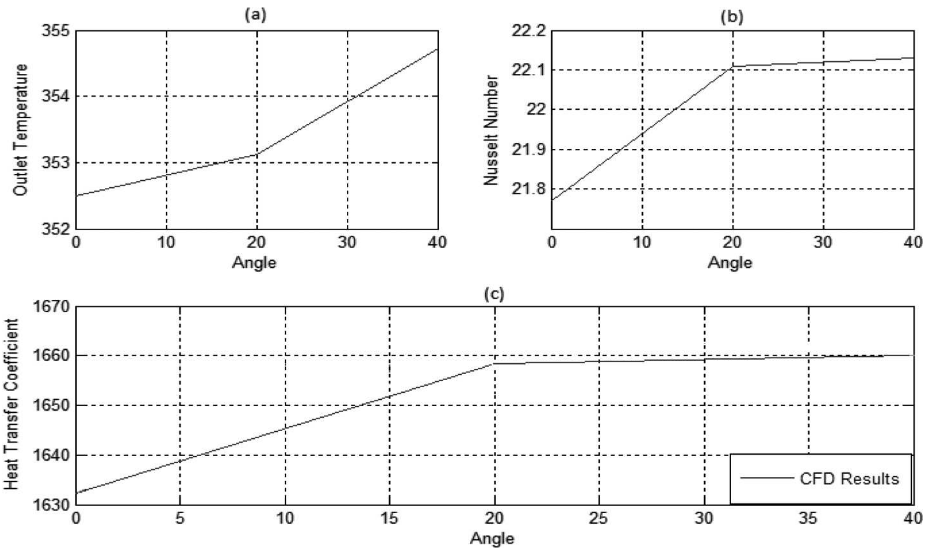

Fig. 3. The effect of the taper angle on the temperature, heat transfer coefficient and Nusselt number (R = 70)

For R = 70, 80, 90 mm and a = 12 mm, Figs 7a, 8a and 9a show the increase in the outlet temperature of the coil with increasing taper angle (θ). This increase is attributed to the increase in the heat transfer surface area with increasing taper angle. Figures 7b, 8b and 9b show that Nusselt

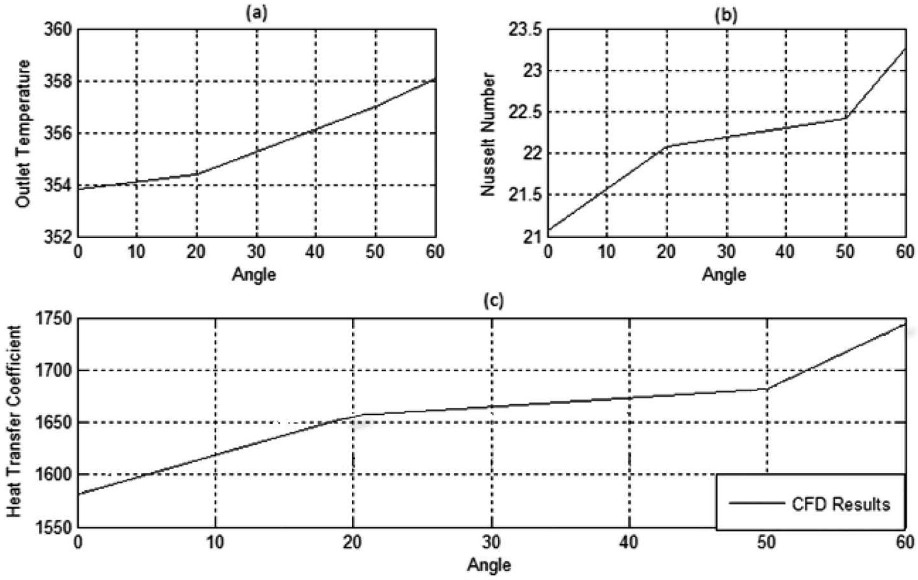

Fig. 4. The effect of the taper angle on the temperature, heat transfer coefficient and Nusselt number (R = 80)

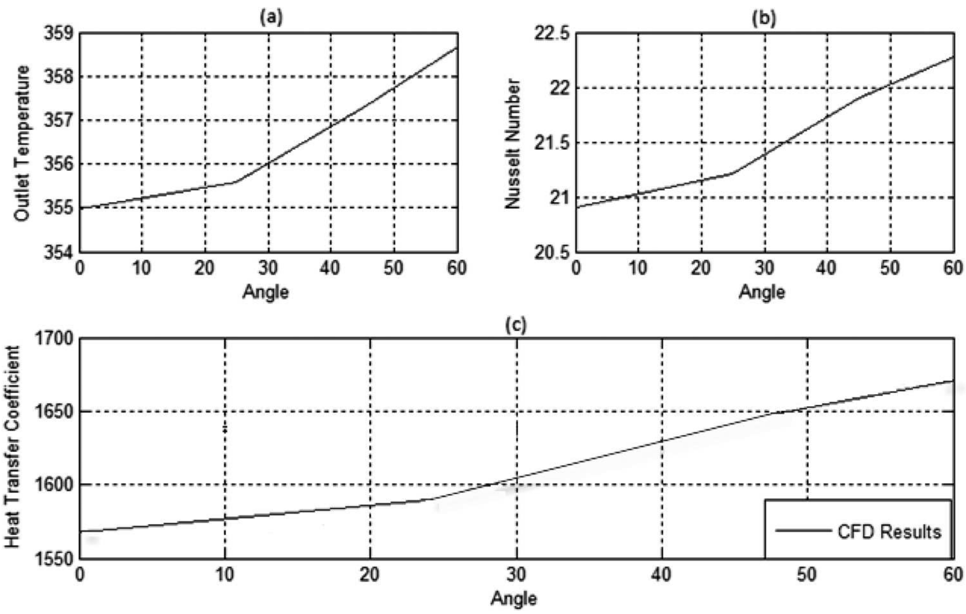

Fig. 5. The effect of the taper angle on the temperature, heat transfer coefficient and Nusselt number (R = 90)

Number increases with increasing taper angle, too. Finally, Figs 7c, 8c and 9c show the increase in the heat transfer coefficient when increasing the taper angle. Figures 10a, 10b, 10c represent the angles (0, 25, 40), respectively,

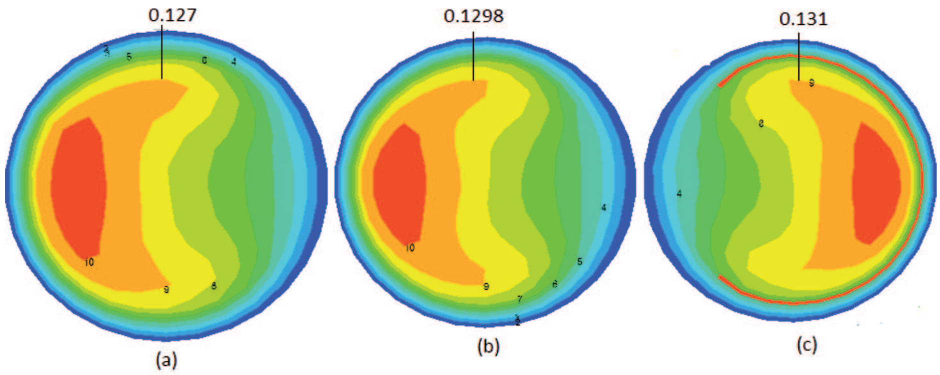

Fig. 6. The effect of the taper angle on the velocity profile

for R = 70. These figures show that the center of the main flow is shifted towards the outwards of the pipe (Dean Vortex).

It can be clearly seen from the following curves that both the heat transfer coefficient and Nusselt Number increases when increasing the taper angle for any bottom radius of curvature (R) and any pipe diameter (a).

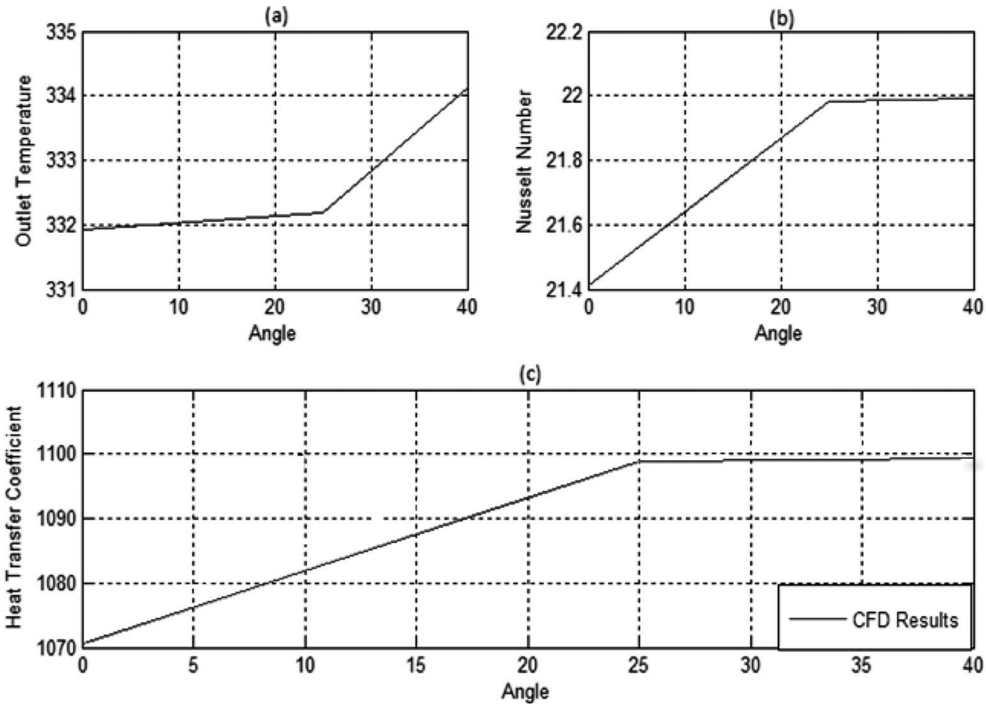

Fig. 7. The effect of the taper angle on the temperature, heat transfer coefficient and Nusselt number (R = 70)

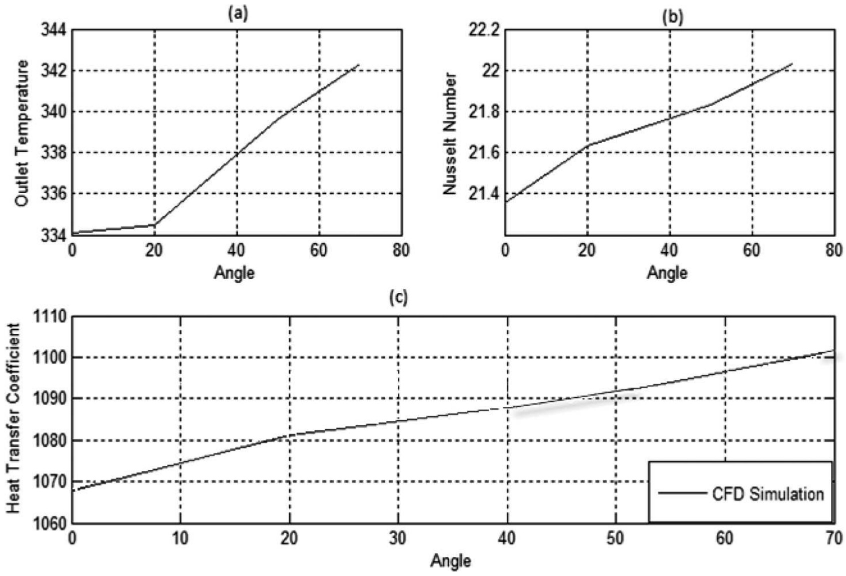

Fig. 8. The effect of the taper angle on the temperature, heat transfer coefficient and Nusselt number (R = 80)

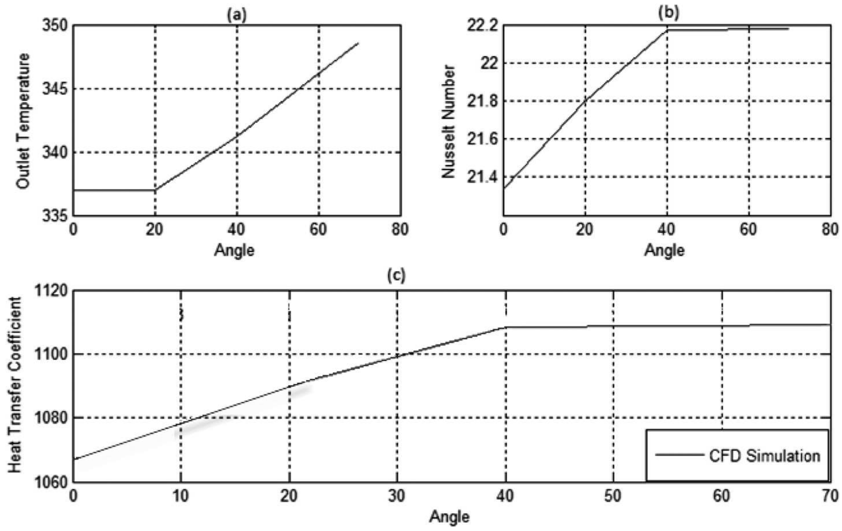

Fig. 9. The effect of the taper angle on the temperature, heat transfer coefficient and Nusselt number (R = 90)

3.2. Pitch

The effect of the pitch variation on the heat transfer coefficient and Nusselt Number will be studied in this section using eight mathematical cases. It has been discussed in the previous section that, as the taper angle increases, the heat transfer coefficient and the Nusselt Number increase, so

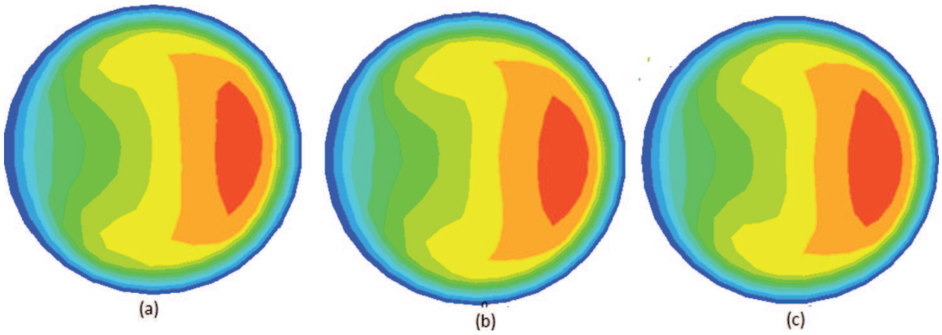

Fig. 10. The effect of the taper angle on the velocity profile

the two coils used in the simulation will have taper angle equals to sixty. The construction parameters of the two coils could be found in Table 3. The height is kept constant, so the number of turns changes when changing the pitch.

It can be clearly seen in Fig. 11 and Fig. 12 that both the heat transfer coefficient and the Nusselt Number increase when increasing the pitch. That also seems to confirm the fact that, as the pitch increases while keeping the height constant for helical cone coils, the curvature ratio (a/R) increases, which leads to an increases in Dean Number and so in Nusselt Number.

Table 3.

Construction parameters for the two coils

R (mm)	a (mm)	H (mm)	u (m/s)	Re	P (mm)	θ
100	10	11	0.1	997	50,60,70,80	60
	12.5			1246		

3.3. Results comparison with the MATLAB code

In this section, we will discuss comparison between the results of the MATLAB code, which was built based on the experimental equation of Manlapaz and Churchill for ordinary helical coils subjected to constant wall temperature, and the CFD results The reason for this comparison is to see whether these equations could be used for the helical cone coils or not. To apply the equations to the helical cone coil, the equation will be calculated for every coil turn, then an average value for the Nusselt Number will be evaluated. The comparison will be made on one of the previous results only, and the model will be for R = 70 and a = 8. It can be concluded from Fig. 13 that Manlapaz and Churchill equation could be used till taper angle reaches

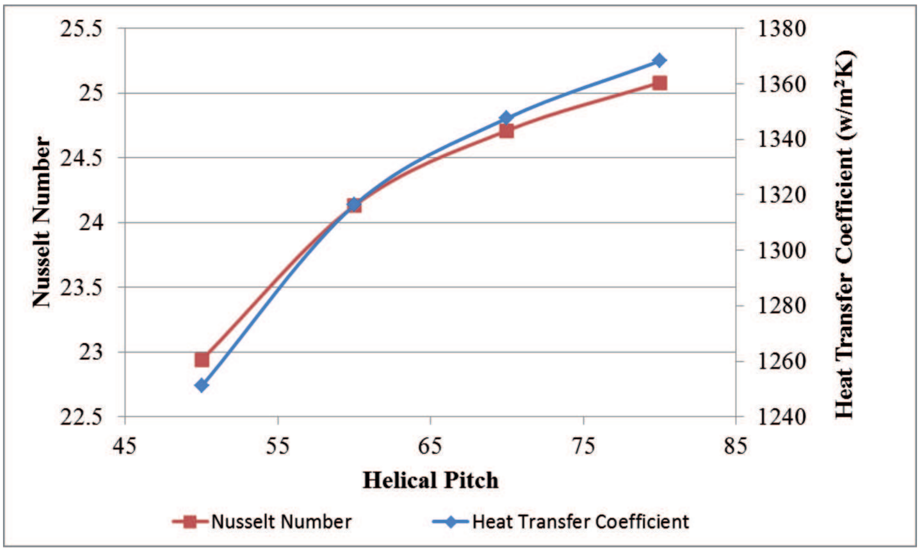

Fig. 11. The effect of pitch variation on Nusselt Number and heat transfer coefficient (a = 10)

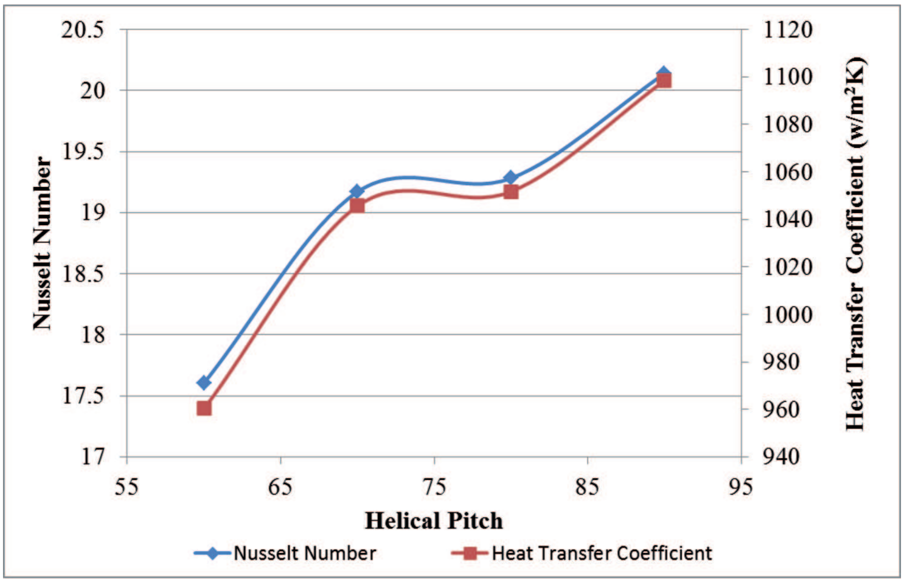

Fig. 12. The effect of pitch variation on Nusselt Number and heat transfer coefficient (a = 12.5)

forty degrees, but after that error increases significantly. The Manlapaz and Churchill equation that resulted from their experimental work is as follow:

$$\mathrm{Nu} = \left[\left(3.675 + \frac{4.343}{x_1}\right)^3 + 1,158\left(\frac{\mathrm{De}}{x_2}\right)^{3/2}\right]^{1/3} \tag{4}$$

$$x_1 = \left(1 + \frac{957}{De^2\,Pr}\right)^2 \tag{5}$$

$$x_2 = \left(1 + \frac{0.477}{Pr}\right) \tag{6}$$

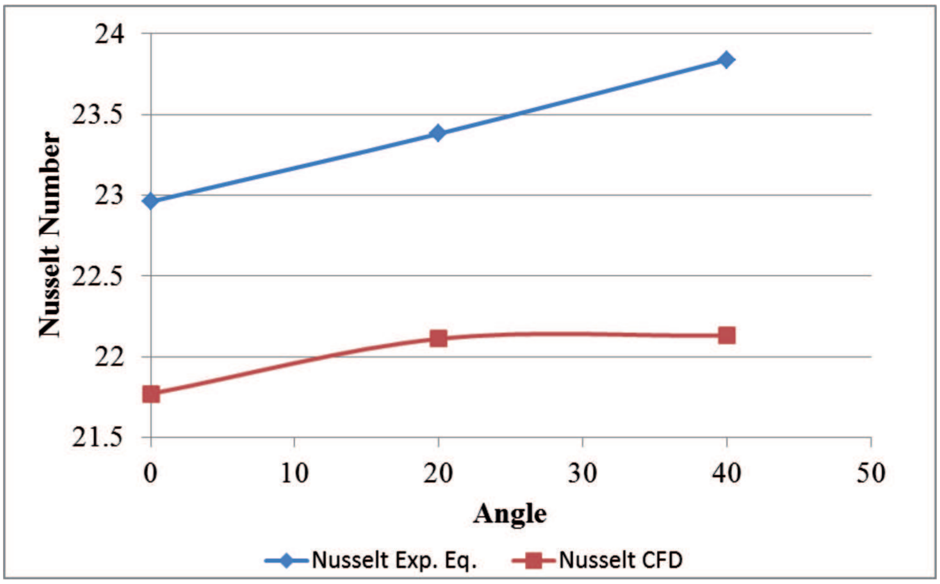

Fig. 13. Nusselt Number from the experimental equation vs. Nusselt Number from the CFD simulation

4. Conclusion

From the results presented in this paper it can be clearly concluded that the taper angle has a significant effect on the heat transfer characteristics of the helical coil. The taper angle was found to be directly proportional to the heat transfer coefficient and Nusselt number (as the taper angle increases both the heat transfer coefficient and Nusselt number increases, and vice versa. The coil exit temperature was also found to be directly proportional to the taper angle (as the taper angle increases the coil exit temperature increases).

The increase in the coil taper angle also leads to a lower space requirement for the installation, which is an advantage of the helical cone coil over the ordinary helical coil. Both the enhancement in heat transfer and the lower space requirements for the helical cone coil makes it a more viable option over the ordinary helical coils.

Nomenclature

a	–	Pipe radius (mm).
H	–	Helical coil height (mm).
h	–	Heat transfer coefficient (w/m^2 K).
I	–	Inclined height (mm).
M_{urf}	–	Relaxation factor of momentum.
P	–	Helical Pitch (mm).
P_{urf}	–	Relaxation factor of pressure.
R	–	Coil radius of curvature (mm).
T	–	Temperature (k).
T_{wall}	–	Wall temperature (k).
u	–	Inlet velocity (m/s).
De	–	Dean Number.
Nu	–	Nusselt Number.
Re_{cr}	–	Critical Reynolds number.
Re	–	Reynolds number.
Greek Symbol		
Θ	–	Taper angle.
ρ	–	Density.

Manuscript received by Editorial Board, January 29, 2012;
final version, September 14, 2012.

REFERENCES

[1] Rohsenow W.M., James R.H., Cho Y.I.: 1998, Handbook of Heat Transfer, United States of America: McGraw Hill.
[2] Bejan A., Kraus A.D.: 2002, Heat Transfer Handbook. United States of America: WILEY.
[3] Dennis S.C.R., Ng M.: 1982, Dual solutions for steady laminar flow through a curved tube. Quarterly Journal of Mechanics and Applied Mathematics, Vol. 35 (3):305-324.
[4] Dennis S.C.R., Riley N.: 1991, On the fully developed flow in a curved pipe at large Dean number. Proc. R. Soc. London Ser. A 434 (1891), 473-478.
[5] Austen D.S., Soliman H.M.: 1988, Laminar flow and heat transfer in helically coiled tubes with substantial pitch. Experimental Thermal and Fluid Science, Vol. 1, 183-194.
[6] Yang G., Dong F., Ebadian M.A.: 1995, Laminar forced convection in a helicoidal pipe with finite pitch. International Journal of Heat and Mass Transfer, Vol.38(5):853-862.
[7] Germano M.: 1982, On the effect of torsion on a helical pipe flow. Journal of Fluid Mechanics, Vol. 125:1-8.
[8] Tuttle E.R.: 1990, Laminar flow in twisted pipes. Journal of Fluid Mechanics, Vol. 219:545-570.

[9] Kalb C.E., Seader J.D.: 1972, Heat and mass transfer phenomena for viscous flow in curved circular tubes. International Journal of Heat and Mass Transfer, Vol. 15:801-817.
[10] Zapryanov Z., Christov C., Toshev E.: 1980, Fully developed laminar flow and heat transfer in curved tubes. International Journal of Heat and Mass Transfer, Vol. 23:873-880.
[11] Figueiredo A.R., Raimundo A.M.: 1996, Analysis of the performances of heat exchangers used in hot-water stores. Applied Thermal Engineering, 16:605-11.
[12] Paisarn Naphon, Jamnean Suwagrai: 2007, Effect of curvature ratios on the heat transfer and flow developments in the horizontal spirally coiled tubes, International Journal of Heat and Mass Transfer, 50: 444-451.
[13] Yan Ke, Ge Pei-qi, Sue Yan-cai, Meng Hai-tao: 2011, Numerical simulation on heat transfer characteristic of conical spiral tube bundle, Applied Thermal Engineering, 31: 284-292.

Analiza numeryczna różnic między charakterystykami działania stożkowych wężownic spiralnych i zwykłych wężownic spiralnych stosowanych jako osuszacze w zespołach osuszania i nawilżania w instalacjach odsalania wody

Streszczenie

W pracy przedstawiono badania numeryczne mające na celu prezentację koncepcji stożkowych wężownic spiralnych i porównanie charakterystyk ich działania jako wymienników ciepła do charakterystyk zwykłych wężownic spiralnych. Jak wiadomo, wężownice spiralne i stożkowe charakteryzują się lepszym przenoszeniem ciepła i masy niż proste rury, co jest związane z powstawaniem wiru w wężownicy spiralnej. Ten tzw. wir Deana (Dean Vortex) jest przepływem wtórnym, nałożonym na przepływ pierwotny. Bezwymiarowy współczynnik Deana, stosowany do opisu wiru Deana, jest funkcją liczby Reynoldsa i pierwiastka kwadratowego ze współczynnika krzywizny, toteż liczba Deana zmienia się dla danej wężownicy wraz z jej krzywizną. Obliczenia numeryczne wykonano przy użyciu komercyjnego oprogramowania CFD w celu zbadania wpływu zmian parametrów strukturalnych wężownicy spiralnej (kąt zbieżności, skok i promień bazowy krzywizny zmieniały się, podczas gdy wysokość pozostawała stała) na liczbę Nusselta, współczynnik wymiany ciepła i temperaturę na wyjściu wężownicy. W badaniach wykorzystano sześć głównych wężownic, o średnicach rury 10 i 12,5 mm i promieniach bazowych krzywizny 70, 80 i 90 mm. W wyniku badań stwierdzono, że zarówno liczba Nusselta jak współczynnik wymiany ciepła rosną wraz ze wzrostem kąta zbieżności. Stwierdzono również, że przy różnych kątach zbieżności skok spirali ma wpływ na liczbę Nusselta i współczynnik wymiany ciepła. Opracowano program w środowisku MATLAB przy pomocy którego obliczono liczby Nusselta dla każdego zwoju wężownicy; na tej podstawie obliczono następnie wartość średnią liczby Nusselta dla całej wężownicy. Program obliczeniowy był oparty na równaniu empirycznym Manlapaza i Churchilla dla zwykłych wężownic spiralnych. Wyniki symulacji uzyskane przy użyciu oprogramowania CFD okazały się możliwe do przyjęcia w zestawieniu z wynikami obliczeń w programie MATLAB.