Pergamon

Int. J. Heat Mass Transfer. Vol. 37, No. 4, pp. 665–671, 1994

0017–9310/94 $6.00+0.00

Experimental investigation of natural convection from vertical helical coiled tubes

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(*Received* 10 *August* 1992 *and in final form* 8 *June* 1993)

Abstract—An experimental study has been made on steady state natural convection heat transfer from vertical helical coiled tubes. Average heat transfer coefficients were obtained for turbulent natural convection to water. The experiments have been carried out for four coil diameter to tube diameter ratios, for five and ten coil turns, and for five pitch to outer diameter ratios. The data are correlated with the Rayleigh number for two different coil sets. The heat transfer coefficient decreases with coil length for tube diameter $d_0 = 0.012$ m, but increases with coil length for $d_0 = 0.008$ m. A critical D/d_0 is obtained for a maximum heat transfer coefficient for tube diameter of 0.012 m with either five or ten coil turns.

1. INTRODUCTION

HELICALLY coiled tubes are used frequently in heating, refrigerating and HVAC applications, and in steam generator and condenser designs in power plants because of their large surface area per unit volume. In spite of their widespread use, there is no information available on natural convection from such coils. However, correlations in the literature for natural convection from vertical and horizontal plates and cylinders are available. The foregoing considerations provided motivation for the present research to fill the gap in the literature.

The discussion of prior work in turbulent forced and natural convection will be divided into two categories: forced flow inside helically coiled circular tubes and natural convection from vertical cylinders and plates. All cited equations will be written using the parameter definitions in this paper.

The criterion for transition from laminar to turbulent flow in curved pipes was established experimentally by Ito [1] and can be represented by

$$Re_t = 2\times10^4 (d_i/D)^{0.32}. \qquad (1)$$

Turbulent heat transfer inside helically coiled tubes has been studied by Rogers and Mayhew [2] who used water flowing through steam-heated tubes for diameter ratios $D/d_i = 10.8$, 13.3, and 20.1. The physical properties in their correlation

$$Nu = 0.023\,(Re)^{0.85}\,(Pr)^{0.4}(d_i/D)^{0.1} \qquad (2)$$

were evaluated at the arithmetic mean of the bulk temperature of the fluid at inlet and outlet. Similar studies were conducted by Seban and McLaughlin [3] using two different tube coils having ratios $D/d_i = 17$ and 104. In their experiment the fluid was heated by electrical dissipation through the tube wall. Most recently, Garimella *et al.* [4] have investigated the heat transfer in coiled annular ducts using equation (2) to estimate the heat transfer coefficient inside the inner duct to map the shell-side heat transfer coefficient. They reported that coiling augments the heat transfer coefficient above the values for a straight annulus in the laminar region. However, this augmentation is less than that for a coiled circular tube and it decreases at the transition region.

Laminar and turbulent natural convection to air, water, and other liquids from vertical flat plates and cylinders were reported by many authors over different ranges of Ra. In the range $10^{11} \leqslant Ra \leqslant 10^{15}$ of interest here for turbulent natural convection to water correlations have been reported by Fujii *et al.* [5] for a vertical circular cylinder and by Vliet and Liu [6] for a vertical plate.

Fujii *et al.* [5] obtained local heat transfer coefficient correlations for water, spindle oil, and Mobiltherm oil on the outside surface of a vertical cylinder. For water in turbulent flow, they reported the following results for uniform wall temperature and uniform heat flux, respectively:

$$Nu_x = 0.13\,(Ra_x)^{0.33}, \quad 4\times10^{10} \leqslant Ra_x \leqslant 8\times10^{10}$$

$$Nu_x = 0.129\,(Ra_x)^{0.33},$$

$$3.6\times10^9 \leqslant Ra_x \leqslant 1.33\times10^{10}.$$

The following correlation in terms of the present definition of Ra_x

$$Nu_x = 0.484\,(Ra_x)^{0.282},$$

$$4.17\times10^{10} \leqslant Ra_x \leqslant 5.3\times10^{12} \qquad (3)$$

has been reported by Vliet and Liu [6] to fit their experimental data for turbulent natural convection

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NOMENCLATURE

A	surface area [m^2]
c	specific heat [kJ kg^{-1} K^{-1}]
D	helix coil diameter [m]
d	tube diameter [m]
g	acceleration due to gravity [m s^{-2}]
H	coil height [m]
h	heat transfer coefficient [kW m^{-2} K^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
L	coil length [m]
$\dot{m}$	mass flow rate [kg s^{-1}]
N	number of coil turns
Nu	Nusselt number, hL/k or hH/k
Pr	Prandtl number, ν/α
p	coil pitch [m]
Q	heat transfer [kW]
Ra	Rayleigh number, $g\beta(T_1-T_2)L^3/\nu\alpha$ or $g\beta(T_1-T_2)H^3/\nu\alpha$
Re	Reynolds number, Vd_i/ν
T	temperature [K]
T_1	average hot water temperature, $[(T_c)_{in}+(T_c)_{out}]/2$
T_2	average ambient water temperature, $[(T_w)_b+(T_w)_a]/2$
U	overall heat transfer coefficient
V	fluid velocity inside coil.

Greek symbols

α	thermal diffusivity [m^2 s^{-1}]
β	coefficient for thermal expansion [K^{-1}]
θ	bulk arithmetic mean temperature [°C]
ν	kinematic viscosity [m^2 s^{-1}].

Subscripts

a	after the run
b	before the run
c	coil
H	coil characteristic length
i	inner
in	inlet
L	coil characteristic length
lm	logarithmic mean
o	outer
out	outlet
s	slanted
t	transition
w	water.

boundary layers on a vertical plate of constant heat flux. Notice that the above equation results in a slight decrease in the heat transfer coefficient along the vertical plate. The analytical expression for turbulent natural convection from a vertical flat plate of constant temperature using the 1/7th power law is reported in ref. [7] as

$$Nu_x = 0.0248\frac{Pr^{1/15}}{(1+0.494\,Pr^{2/3})^{0.4}}(Ra_x)^{0.4} \quad (4)$$

for $Ra_x \geqslant 10^9$. For air ($Pr = 0.72$), equation (4) is only 1% different from the experimental correlation reported by Kreith [8] for natural convection from a vertical cylinder to air for $10^9 \leqslant Ra \leqslant 10^{12}$ and also by McAdams [9] for convection from short vertical plates to air.

This paper presents the results of an experimental study of turbulent natural convection heat transfer from vertical helical coiled tubes to water. The investigation focuses on the determination of average heat transfer coefficients. Nusselt numbers have been correlated as a function of the Rayleigh number and are compared to other available correlations in the same range for vertical plates and cylinders, respectively.

Section 2 describes the experimental apparatus and test procedure. This is followed by the analysis of experiments in Section 3 with results and discussion in Section 4. Summary and conclusions are given in Section 5.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

A schematic diagram of the experimental apparatus is sketched in Fig. 1. The testing coil (8) is fixed in an ambient temperature bath (1) which serves as the coolant. The hot temperature bath (2) contains a stirrer (3), a 10 kW heater (4), and a temperature controller. Flow of hot turbulent water is pumped through the coil with a 1.5 hp centrifugal pump (5). The hot water loop circuit is completed when water from the pump returns to the hot water tank (2). Tanks (1) and (2) are cylindrical vessels 0.90 m deep, 0.6 m diameter, and they were filled to approximately 0.025 m from the top with tap water. The volume flow rate through the coil is measured by a flowmeter (6) and temperatures are measured with calibrated iron–constantan (type J) thermocouples (7) at the inlet and outlet of the first and last turn of the coil, respectively. A thermocouple housing shown in Fig. 2(a) has circuitry designed to directly measure the inlet temperature, the outlet temperature, and their difference. Thermocouples are rated to meet the ANSI special limits of error 0.5°C or 0.4% whichever is greater. The thermocouples are separated from the tube bottom housing by a two mil layer of mercury. A total of 6–11 temperature difference measurements are made for every p/d_o. The coils are formed from initially straight tubing (brass 70% CU and 30% ZN with thermal

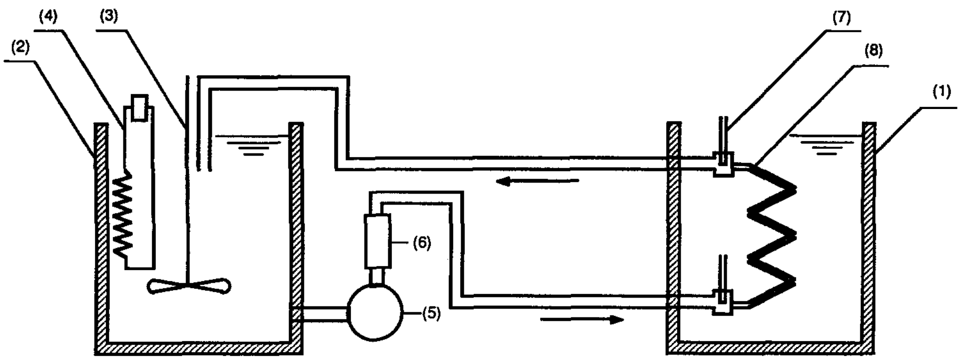

FIG. 1. Schematic diagram of the experimental apparatus. (1) Constant temperature bath, (2) test reservoir, (3) stirrer, (4) heater, (5) centrifugal pump, (6) flowmeter, (7) thermocouples, and (8) testing coil.

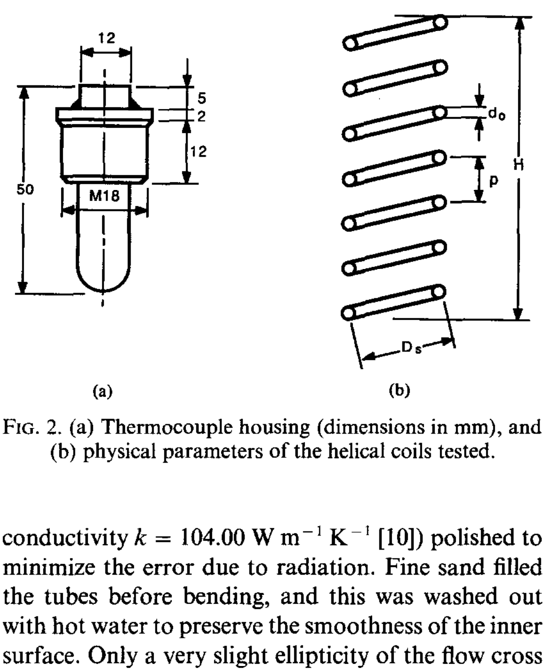

FIG. 2. (a) Thermocouple housing (dimensions in mm), and (b) physical parameters of the helical coils tested.

conductivity $k = 104.00\ \mathrm{W\ m^{-1}\ K^{-1}}$ [10]) polished to minimize the error due to radiation. Fine sand filled the tubes before bending, and this was washed out with hot water to preserve the smoothness of the inner surface. Only a very slight ellipticity of the flow cross section and distortion of wall thickness are introduced by the bending process. The inner and outer diameter, and the wall thickness of every coil was measured by using a sample cut of that coil. Table 1 lists the dimensions and parameters of the coils used in this investigation. To allow for the obliquity of the helix, the slanted outer turn diameter D_s (Fig. 2(b)) for each turn is measured using a vernier caliper and the helix coil diameter is obtained from the following equation

$$D = \frac{\sum_{i=1}^{N} \sqrt{\left(D_s^2 - \left(\frac{p}{2}\right)^2\right)}}{N}. \tag{5}$$

The coil length is calculated from $L = \pi DN$ and the coil turns are separated from each other using Plexiglas spacers with a specific length to fix the pitch of the coil. The water in tank (2) was heated and well mixed using a 10 kW heater (4) and a stirrer (3). Data were collected for one pitch to coil diameter ratio for hot temperatures ranging from 35 to 80°C. The temperature was increased by 5°C for each run and the procedure was repeated for the other test coils. After every run the ambient water in tank (1) was drained and filled again with fresh tap water to keep the ambient water temperature nearly the same before each run. The experiments are designed so that the outer (external tube-side) heat transfer coefficients could be calculated using measured overall heat transfer coefficients and predicted inner (internal tube-side) heat transfer coefficients. The mass flow rate is maintained constant for each coil but it has a higher value for the large diameter tube than that of the small diameter one. The procedure outlined above is used to generate natural convection heat transfer data over the Prandtl number range $3.44 \leqslant Pr$

Table 1. Physical dimensions of the test coils

Coil no.	d_i (m)	d_o (m)	D (m)	L (m)	p/d_o	N	D/d_o
1	0.011	0.012	0.2495	7.8383	1.5, 2, 3, 3.5	10	20.792
2	0.011	0.012	0.2495	3.9191	1.5, 2, 3, 3.5	5	20.792
3	0.011	0.012	0.1671	5.2490	1.5, 2, 3.5, 4	10	13.923
4	0.011	0.012	0.1671	2.6245	1.5, 3.5, 4	5	13.923
5	0.011	0.012	0.1190	3.7375	1.5, 2, 3, 3.5	10	9.914
6	0.011	0.012	0.1190	1.8687	1.5, 2, 3, 3.5, 4	5	9.914
7	0.007	0.008	0.1597	5.0157	1.5, 2, 3, 3.5	10	19.957
8	0.007	0.008	0.1597	2.5078	1.5, 2, 3, 3.5	5	19.957
9	0.007	0.008	0.0793	2.4917	1.5, 2, 3, 3.5	10	9.914
10	0.007	0.008	0.0793	1.2458	1.5, 2, 3, 3.5	5	9.914

$\leqslant 5.30$. The variation in Prandtl number is due to the difference in temperature between individual runs.

3. ANALYSIS OF EXPERIMENTS

In the following discussion, the physical properties of the hot water inside the tube coil are assumed constant along the coil length and evaluated at the average bulk temperature for each run. Heat loss from the coil can be calculated from

$$Q = \dot{m}c[(T_c)_{in} - (T_c)_{out}], \quad (6)$$

where Q is the heat loss from the coil, $\dot{m}$ is the hot water flow rate, $(T_c)_{in}$ is the coil inlet temperature, $(T_c)_{out}$ is the coil outlet temperature, and c is the hot water specific heat. Having obtained a value for the rate of heat transfer Q, the overall heat transfer coefficient U is calculated from

$$Q = U\Delta T_{lm} \quad (7)$$

where ΔT_{lm} is the log-mean temperature difference

$$\Delta T_{lm} = \frac{\{[(T_c)_{in} - (T_w)_b] - [(T_c)_{out} - (T_w)_a]\}}{\ln\left\{\dfrac{(T_c)_{in} - (T_w)_b}{(T_c)_{out} - (T_w)_a}\right\}} \quad (8)$$

and U is defined as

$$U = \left[\frac{1}{h_i A_i} + \frac{\ln(d_o/d_i)}{2\pi k L} + \frac{1}{h_o A_o}\right]. \quad (9)$$

Once U is calculated from equation (7), h_i is evaluated from equation (2), k for the coil material is given, and then the outer heat transfer coefficient h_o can be calculated from equation (9). Direct temperature measurements of the ambient water temperature in tank (1) were made before and after each run. In calculating the physical properties for the nondimensional groups the bulk arithmetic mean temperature is used

$$\theta = 0.5\left[\frac{(T_c)_{in} - (T_c)_{out}}{2} + \frac{(T_w)_b - (T_w)_a}{2}\right]. \quad (10)$$

All temperature measurements in this experiment are accurate to $\pm 0.01\,^{\circ}\mathrm{C}$.

The coil length L has been adopted as the characteristic length in the nondimensional groups. However, the coil height H is also used to determine the range of Rayleigh number and Nusselt number in both cases and its relation to the region of natural convection. The equation for the coil height is $H = Np$, where N is the number of coil turns and p is the coil pitch.

4. RESULTS AND DISCUSSION

The experimental data points of all coil sets are presented in Fig. 3 by plotting Nu_L against Ra_L on a logarithmic scale. As seen in this figure there are two groups of data, the upper group corresponding to tube diameter $d_o = 0.012$ m and a lower group having

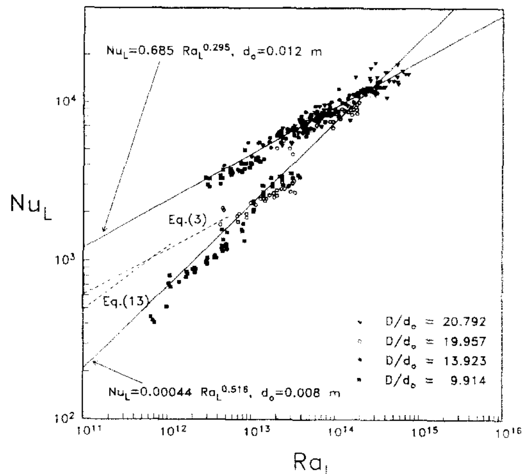

FIG. 3. Average turbulent heat transfer correlation for vertical coils in water with coil length as a characteristic length.

$d_o = 0.008$ m. The upper data are for three D/d_o diameter ratios 20.797, 13.923, and 9.914 while the lower data are for D/d_o of 19.957 and 9.914. In both cases data have been taken for both five and ten coil turns. A least-squares power law fit through the two data sets yields the following correlations

$$Nu_L = 0.685\,(Ra_L)^{0.295},$$
$$d_o = 0.012\,\mathrm{m}, \quad 3\times10^{12} \leqslant Ra_L \leqslant 8\times10^{14} \quad (11)$$
$$Nu_L = 0.00044\,(Ra_L)^{0.516},$$
$$d_o = 0.008\,\mathrm{m}, \quad 6\times10^{11} \leqslant Ra_L \leqslant 1\times10^{14}. \quad (12)$$

Equation (4) was evaluated at $Pr = 4.37$ corresponding to the average Prandtl number for water used in the present experiment and the resulting correlation is

$$Nu_L = 0.0195\,(Ra_L)^{0.4}. \quad (13)$$

Correlations (3) and (13) are plotted as dashed lines in Fig. 3 for comparison with the present results.

Although the exponent in equations (11) and (12) is sensitive to experimental scatter in the data, the results are clearly adequate to show that the exponent is less than 1/3 in equation (11) and is greater than 1/3 in equation (12). It is clear that the heat transfer coefficient h_o decreases slightly with boundary-layer length for equation (11) as

$$h_o \propto L^{-0.115}. \quad (14)$$

However, h_o increases rapidly with boundary-layer length for equation (12) as

$$h_o \propto L^{0.548}. \quad (15)$$

Equations (11) and (12) suggest that increasing the tube diameter for fixed L and for the same value of Ra_L enhances h_o.

In an attempt to ascertain the effect of the coil height H as a characteristic length in Ra and Nu, Figs.

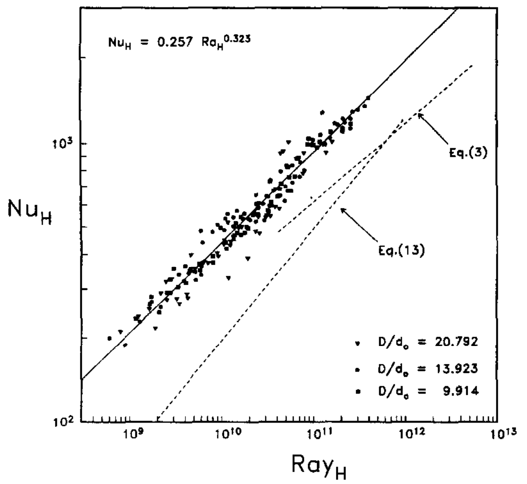

FIG. 4. Average turbulent heat transfer correlation for a vertical set of coils with $d_o = 0.012$ m in water compared with the previous results on flat plates.

4 and 5 were constructed. The data are presented by plotting Nu_H against Ra_H for $d_o = 0.012$ m and for D/d_o of 20.792, 13.923, and 9.914 in Fig. 4 and for $d_o = 0.008$ m and for D/d_o of 19.957 and 9.914 in Fig. 5. The best power law fit through the data in Fig. 4 is given by

$$Nu_H = 0.257\,(Ra_H)^{0.323}, \quad 6\times10^8 \leqslant Ra_H \leqslant 3\times10^{11}, \tag{16}$$

indicating that h_o is decreasing very slightly or it could be constant along the coil height because the exponent in equation (16) may vary due to experimental uncertainty to reach the value 1/3. In other words, increasing the coil height results in a very slight decrease in h_o or has no influence on h_o for tube diameter $d_o = 0.012$ m. Figure 5 shows two regions, one laminar and the other turbulent, since turbulent nature convection is distinguished by $Ra > 10^9$. The range $10^8 \leqslant Ra_H \leqslant 10^{10}$ could be defined as a transition region from laminar to turbulent flow since the heat transfer coefficients increase abruptly as for a vertical wall in water (Fujii *et al.* [5]). This behavior of abruptly increasing h_o can be seen from the following best-fit correlations for two different D/d_o

$$Nu_H = 0.016\,(Ra_H)^{0.433},$$

$$\frac{D}{d_o} = 19.957, \quad 2\times10^8 \leqslant Ra_H \leqslant 5\times10^{10} \tag{17}$$

$$Nu_H = 0.0023\,(Ra_H)^{0.494},$$

$$\frac{D}{d_o} = 9.914, \quad 3.5\times10^8 \leqslant Ra_H \leqslant 7\times10^{10}. \tag{18}$$

These equations suggest that h_o is increasing with H for this set of data as

$$h_o \propto H^{0.299} \quad \text{and} \quad h_o \propto H^{0.482} \tag{19}$$

which agree with the assumption that h_o increases rapidly with H in the transion region from laminar to turbulent flow for both diameter ratios. However, the rate of increase of h_o for $D/d_o = 9.914$ is higher than that for $D/d_o = 19.957$.

Figures 6 and 7 were constructed to see the effect of the number of coil turns on the heat transfer coefficient h_o. In Fig. 6 the data are presented for two diameter ratios 13.923, 9.914, at $d_o = 0.012$ m by plotting Nu_L against Ra_L. The best-fit lines to this data are

$$Nu_L = 5.11\,(Ra_L)^{0.214}N^{0.296},$$

$$\frac{D}{d_o} = 13.923, \quad 4.8\times10^{12} \leqslant (Ra_L) \leqslant 4\times10^{14} \tag{20}$$

$$Nu_L = 2.64\,(Ra_L)^{0.228}N^{0.324},$$

$$\frac{D}{d_o} = 9.914, \quad 2.8\times10^{12} \leqslant (Ra_L) \leqslant 1\times10^{14}. \tag{21}$$

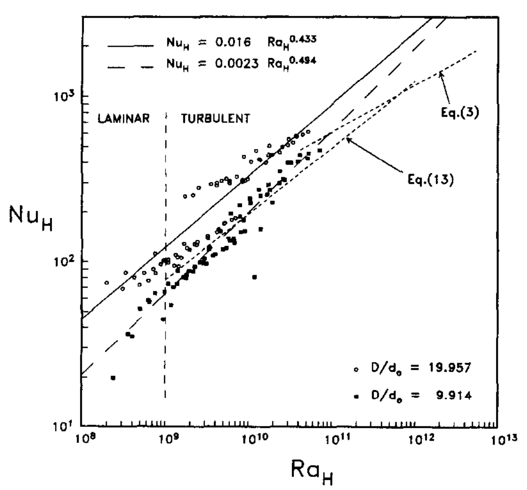

FIG. 5. Average heat transfer correlation for a vertical set of coils with $d_o = 0.008$ m in water.

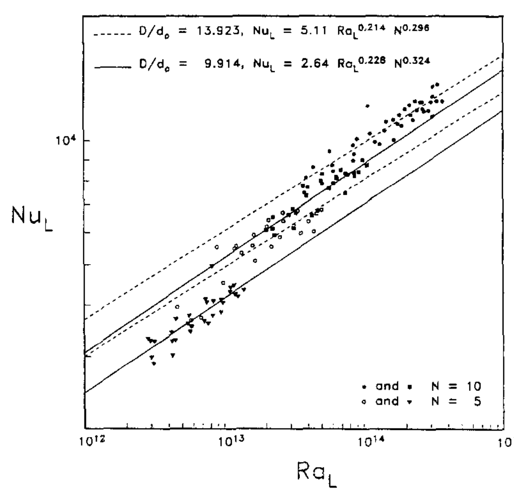

FIG. 6. Average turbulent heat transfer correlation for two coils with $d_o = 0.012$ m; $N = 5$ and 10 for each diameter ratio D/d_o.

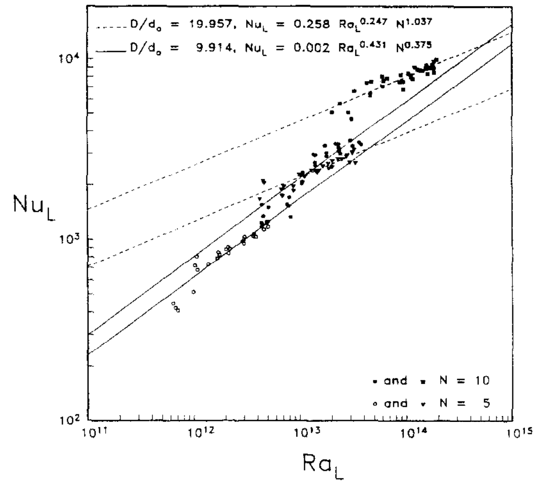

FIG. 7. Turbulent average heat transfer correlation for two coils having $d_o = 0.008$ m; $N = 5$ and 10 for each diameter ratio D/d_o.

Equations (20) and (21) suggest that h_o slightly decreases as N increases for $D/d_o = 13.923$ and is almost constant for $D/d_o = 9.914$. Furthermore, since the decreasing rate of h_o is very low or zero

$$h_o \propto N^{-0.062} \quad \text{and} \quad h_o \propto N^{0.008} \qquad (22)$$

credence is given to the idea mentioned above that the boundary layer flow is indeed turbulent.

Figure 7 shows the experimental data for $D/d_o = 19.975$, 9.914 and for five and ten coil turns in a logarithmic scale for $d_o = 0.008$ m. The least-squares fits to the measurements are given by

$$Nu_L = 0.258\,(Ra_L)^{0.247} N^{1.037},$$

$$\frac{D}{d_o} = 19.957, \quad 3\times10^{12} \leqslant (Ra_L) \leqslant 2\times10^{12} \quad (23)$$

$$Nu_L = 0.002\,(Ra_L)^{0.431} N^{0.375},$$

$$\frac{D}{d_o} = 9.914, \quad 5\times10^{11} \leqslant (Ra_L) \leqslant 3\times10^{13}. \quad (24)$$

The above equations show that

$$h_o \propto N^{0.778} \quad \text{and} \quad h_o \propto N^{0.668} \qquad (25)$$

for equations (23) and (24), respectively, indicating that h_o increases abruptly as N increases for the same diameter ratio, and the rate of increase is very high compared to the rate of decrease for the set of data in Fig. 6. The explanation for the different heat transfer rate is that, as mentioned before, the data in Fig. 7 for $d_o = 0.008$ m are in a region of transition from laminar to turbulent flow, while the data in Fig. 6 are entirely in the turbulent region.

Figure 8 was constructed to see the effect of the diameter ratio D/d_o on the h_o by plotting the measured values of h_o for different D/d_o on a dimensional linear scale against the total coil surface area. These values of h_o for each coil are the average value for that coil: thus there are two values of h_o, one for five and the other for ten coil turns. In Fig. 8 each group of data points for $N = 5$ and 10 having $d_o = 0.012$ m are connected by dashed lines. One can see that h_o exhibits a maximum for $N = 5$ and 10. In other words, there exists a critical diameter ratio for five and ten coil turns corresponding to a maximum h_o. The critical D/d_o was found to be 14.4 for five and 13.0 for ten coil turns, respectively. Unfortunately, there are insufficient data to determine the critical diameter ratio for the data having $d_o = 0.008$ m.

5. SUMMARY AND CONCLUSIONS

Experimental studies on turbulent natural convection heat transfer from vertical helical coils to water were performed. Average heat transfer measurements were made for Rayleigh numbers up to 10^{15} and for two sets of coils. The first set has $d_o = 0.012$ m and three D/d_o of 20.792, 13.923, and 9.941 and the second set has $d_o = 0.008$ m and two D/d_o of 19.957 and 9.941. Using L as a characteristic length it was found that the transition region to turbulence is marked by an abrupt increase in h_o for $d_o = 0.008$ m while the turbulent region for $d_o = 0.012$ m is marked by a slightly decreasing or nearly constant value of h_o. The same behavior holds when H is used as the characteristic length. However, the range of Ra_H is less than the range of Ra_L. The correlation covering the first set of coils is

$$Nu_L = 0.685\,(Ra_L)^{0.295}$$

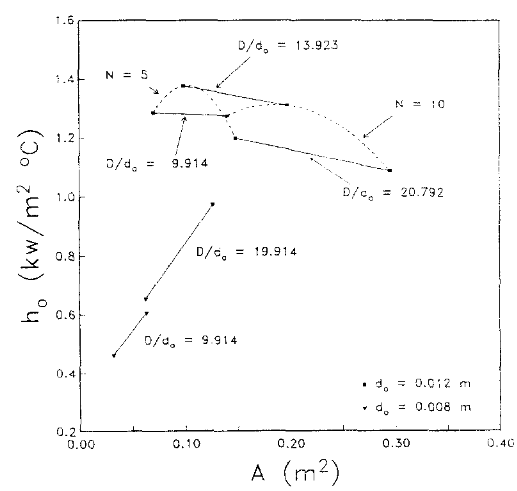

FIG. 8. Variation of average heat transfer coefficient h_o with coil outer surface area for all D/d_o. Dashed lines connect the average data points for the five and ten turn coils.

which results in a slight decrease in the heat transfer coefficient with coil length for the same diameter ratio. The equation covering the second set of coils is

$$Nu_L = 0.00044\,(Ra_L)^{0.516}$$

which shows an abrupt increase in the heat transfer coefficient. It was also found that increasing the number of turns for the same diameter ratio results in a decreasing heat transfer coefficient for the first set of coils ($d_o = 0.012$ m) but an increasing heat transfer coefficient for the second set of coils ($d_o = 0.008$ m). Finally, a maximum heat transfer coefficient for the first set of coils ($d_o = 0.12$ m) was obtained at the critical diameter ratio of 14.4 when $N = 5$ and 13.0 when $N = 10$.

Acknowledgements—The author would like to acknowledge Professor Kamal El-Din Hassan for introducing the problem. The author also acknowledges the gracious hospitality of the Mechanical Engineering Department at the University of Colorado at Boulder (U.S.A.) and especially that of his host Professor Patrick D. Weidman for interesting discussion on the subject of this paper.

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