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Performance Enhancement of Shell and Helical Coil Water Coolers Using Different Geometric and Fins Conditions

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Experimental investigations of heat transfer characteristics and performance enhancement of shell and helical coil water coolers using external radial fins and different shells diameters were conducted. The study aims to enhance the water coolers performance in a trial to improve coil compactness. Two helical coils; one with a plain tube and the other with external radial fins, were tested in four shells of different tube diameters. Refrigerant passing inside the helical coils was used to cool water that enclose/passes in the space between the helical coil and the shell. Tests were conducted under mixed convection heat transfer regimes. Results showed performance and compactness enhancement with the insertion of external radial fins and increasing the shell diameter to helical coil diameter ratio. For nonfinned and finned coils, Nusselt number increased with increasing Reynolds number, Grashof number, and shell diameter. Correlations were predicted to give the Nusselt number in terms of Reynolds number, Grashof number, and shell diameters for finned and nonfinned helical coils. Correlations predictions were compared with present and previous experimental results and good agreements were obtained. © 2015 Wiley Periodicals, Inc. Heat Trans Asian Res, 45(7): 631–647, 2016; Published online in Wiley Online Library (wileyonlinelibrary.com/journal/htj). DOI 10.1002/htj.21180

Key words: shell and helical coil, compact heat exchanger, water cooler, radial fins

1. Introduction

Due to their compact structure and high heat transfer coefficient compared to straight tubes, helically coiled tubes have been considered as one passive heat transfer enhancement and compactness technique. Moreover, heat transfer enhancement in helical coils using internal or external fins enables the size of the heat exchanger to be considerably reduced and compact. Helical coils are widely used in various industrial applications including power plants, nuclear reactors, automotive industries, heat recovery systems, chemical processing, food industries, commercial and household refrigeration, ventilating and air-conditioning systems, as well as in environmental engineering. Review of the literature indicated an enormous amount of experimental and analytical investigations of internal heat transfer inside helical coils while the data on external heat transfer investigation and enhancement are very limited.

For internal heat transfer, several studies [1–12] have indicated that helically coiled tubes are superior to straight tubes when employed in heat transfer applications. This is attributed to the centrifugal force due to tube curvature resulting in secondary flow development that enhances heat transfer rate. Secondary flow profile in helical tube was initially investigated and described by Dean [1] showing the occurrence of swirling flow patterns separated by a horizontal center line. Dean [1] presented the Dean number (De), ratio of inertial to viscous forces, to characterize the secondary flow profile. Yang and colleagues [2] numerically showed that the overall Nusselt number slightly decreases with increasing helical coil torsion for low Prandtl numbers, but significantly decreases with larger Prandtl numbers. Petrakis and Karahalios [3] showed that the presence of a coaxial core in a curved pipe affects the flow properties, especially at high Dean numbers. Xin and colleagues [4] experimentally determined pressure drop correlations for both single- and two-phase flow in helicoidally annular pipes. Petrakis and Karahalios [5] numerically examined the effect of core size, showing that for small core sizes changes in the Dean number significantly affect the flow properties and this was not the case for large core sizes. Guo and colleagues [7] experimentally showed the dependence of the pressure drop on coil inclination angle especially for two-phase flow. Zhao and colleagues [8] experimentally correlated pressure drop and boiling heat transfer characteristics for boiling of steam inside helical coils. Naphon and Wongwises [9] provided a literature review on heat transfer and flow characteristics of single- and two-phase flow in curved tubes including helically coiled tubes and spirally coiled tubes. Shokouhmand and Salimpour [10] and Salimpour [11] investigated fully developed laminar flow and heat transfer in a helically coiled tube with uniform wall temperature. A correlation for predicting optimal Reynolds number was proposed. Jamshidi and colleagues [12] studied the effect of the insertion of Al2O3 nanofluid in water passing in helical tubes and it was shown that nanofluids improve the thermal–hydraulic performance of helical tubes. In a trial to increase compactness of heat exchangers, Nada and colleagues [13, 14] correlated pressure drop and Nusselt number for tube in tube and multitubes in a tube helical heat exchanger, respectively.

A little work was conducted to investigate heat transfer characteristics outside helical coils [15–22]. Most of these works were conducted on unfinned tubes. Prabhanjan and colleagues [15] studied heat transfer from a helical coil immersed in a water bath under natural convection. Moawed [16] presented experimental results of natural convection from uniformly heated helicoidal pipes oriented vertically and horizontally in the surrounding air. The overall average Nusselt number was observed to depend on pitch to pipe diameter ratio, coil diameter to pipe diameter ratio, and length to pipe diameter ratio and coil orientation. Thongwik and colleagues [17] studied the heat transfer phenomenon of melting slurry ice on the external surface of a copper helical coil. Heat transfer models of slurry ice during melting and postmelting in terms of Nusselt number with the pertinent parameters were developed. Shokouhmand and colleagues [18] and Slaimpour [19] performed experimental studies of shell and helically coiled tube heat exchangers. Different coils geometric parameters were tested under parallel and counter flow conditions. Later, an experimental study of thermal performance of shell-and-coil heat exchangers was undertaken by Ghorbani and colleagues [20, 21]. It was found that the mass flow rate of the tube- to shell-side ratio was effective on the axial temperature profiles of a heat exchanger. The results were obtained for various Reynolds and Rayleigh numbers, various tube-to-coil diameter ratios, and dimensionless coil pitch and different flow regimes. Genic and colleagues [22] presented results of experimental research on shell-side heat transfer coefficient concerning three heat exchangers with helical coils to correlate Nu in terms of Re Pr and the coil's geometric parameters. Zhang and colleagues [23] experimentally compared the performance of shell-and-tube oil coolers with overlapped helical baffles and segmental baffles. The results showed that the oil cooler with helical baffles has a lower shell side pressure drop and higher heat transfer coefficient per unit pressure drop at fixed volume flow rate than the oil cooler with segmented baffles.

Moawed [24] studied the forced convection from outside surfaces of helical coiled tubes experimentally for different geometric parameters of helical coils. The experimental results indicated that coil geometries have important effects on the heat transfer coefficient. Jayakumar and colleagues [25] numerically analyzed helically coiled heat exchangers, where one of the working fluids is flowing through the helical coil and the other is outside. Results have been established that correlations for heat transfer and pressure drop should take into account the pitch circle diameter and coil pipe diameter. Naphon and Wongwises [26] and Huzzayien and colleagues [27] investigated the average tube- and air-side heat transfer coefficients in a spirally coiled finned tube and straight tubes heat exchangers under dry and wet surface conditions experimentally, respectively. The effects of the inlet conditions of both working fluids flowing through the heat exchanger on the heat transfer coefficients are discussed. Correlations predicting heat transfer coefficients under dry and wet conditions were proposed. Naphon [28] studied the thermal performance and pressure drop of the helical coil heat exchanger with and without helical crimped fins. The effects of the inlet conditions of both working fluids flowing through the test section on the heat transfer characteristics are discussed.

The above literature shows that although a significant amount of research has been performed on flow patterns and heat transfer in curved pipes and helically coiled pipes, information on the outer surface of helically coiled tubes with and without fins in shell heat exchangers are very limited. The present study aims to experimentally investigate heat transfer characteristics of helically coiled tubes, with and without external radial fins, and shell heat exchangers. The effects of shell tube diameter and existence of radial fins as well as inlet fluid conditions on the performance of the heat exchanger are investigated. Experiments have been carried out under mixed convection conditions. This study aims to enhance heat transfer characteristics to lead to more compact heat exchangers.

Nomenclature

b: Coil pitch, m
C_{cw}: Specific heat capacity of cooling water, J/kg.K
D_c: Curvature diameter of helical coil, m
D_h: Hydraulic diameter of helical coil, m
D: Shell diameter, m
D^*: Dimensionless shell diameter (D_s /D_c)
d: Coil tube diameter, m
Gr: Grashof number
g: Acceleration of gravity, m/s^2
h: Average convection heat transfer coefficient, W/m^2 .K
h_{fg}: Latent heat of vaporization, J/kg
k_w: Water thermal conductivity, W/m.K
L: Length of coil, m
$\dot{m}_c$: Refrigerant condensate flow rate, kg/s
$\dot{m}_{cw}$: Cooling water mass flow rate, kg/s

Nu: Nusselt number
n: Number of coil turns
q: Heat transfer rate, W
R: Curvature radius, m
Ro: Coil outside diameter, m
T_{cwi}: Inlet cooling water temperature, K
T_{cwo}: Outlet cooling water temperature, K
T_s: Average tube surface temperature, K
T_w: Average cooling water temperature, K
V: Average water velocity, m/s

Greek Symbols

β: Thermal expansion coefficient of water, K^{-1}
γ: Dimensionless pitch ($b/\pi Dc$)
μ_w: Water dynamic viscosity, kg/m.s
ρ_w: Density of water, kg/m^3

2. Experimental Setup and Procedure

2.1 Experimental setup

An experimental setup was designed to study heat transfer characteristics for cooling water flow in shell and helical coil heat exchanger. The fluid which passes in the helical coil tube is refrigerant and the fluid that passes in the shell is water. The experimental set up was designed to enable varying and controlling cooling water flow rates and inlet water temperatures as well as changing the geometric parameters of the heat exchanger; namely, shell outside diameters and the existence of external radial fins on the outer surface of the helical coil.

A schematic diagram of the experimental setup is shown in Fig. 1. It consists of three main sections: a refrigerant circuit, cooling water circuit, and test section. The refrigeration circuit consists of a condensing unit and helical coil evaporator. The condensing unit consists of compressor, forced convection air cooled condenser, filter, and expansion device. The compressor is of 1/3 hp hermetic reciprocating type. The helical coil evaporator (part of the test section) is made from aluminum. Two evaporators were tested one with an unfinned helical coil and the other with external radial fins.

The water circuit is an open type circuit. It consists of an overhead water tank, heating section, the shell tube of the test section, piping network and fittings, and measuring instruments. The overhead tank is a 15 L water storage tank. The water level in the tank is maintained constant by a float valve to insure constant flow rate and pressure at the inlet of the test section. City water flows to the constant head water tank. Water from the overhead water tank passes through a 4 kW electric heating coil to control and maintain its temperature at the desired value. The heater is thermally insulated with glass wool and connected with step power variac to control the power according to the required exit water temperature. A control valve upstream of the test section is used to control the required water flow rate to the test section. The water exit from the test section is collected in a scaled vessel to measure the water flow rate.

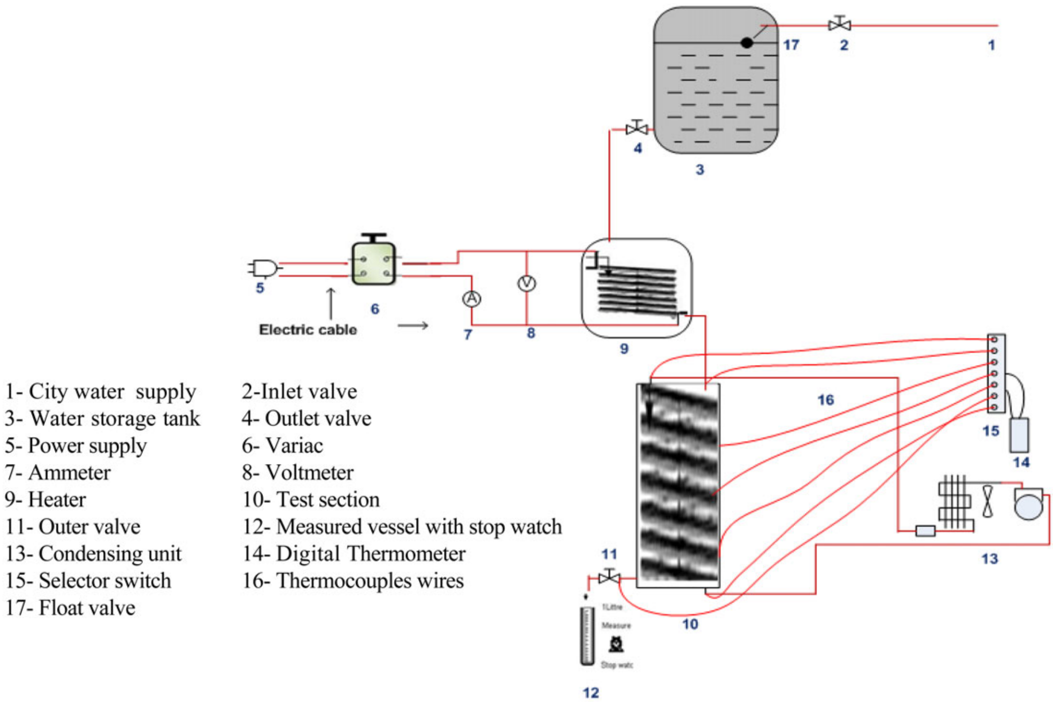


Fig. 1. Experimental set up.

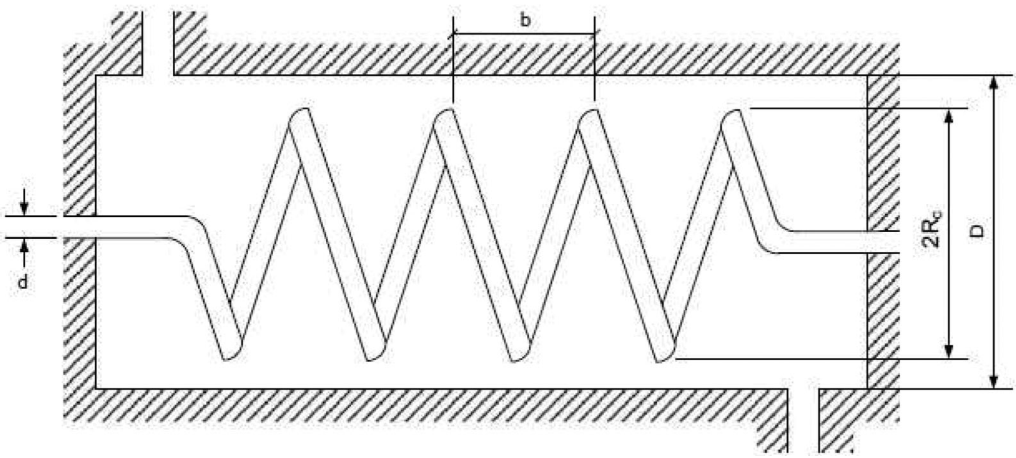
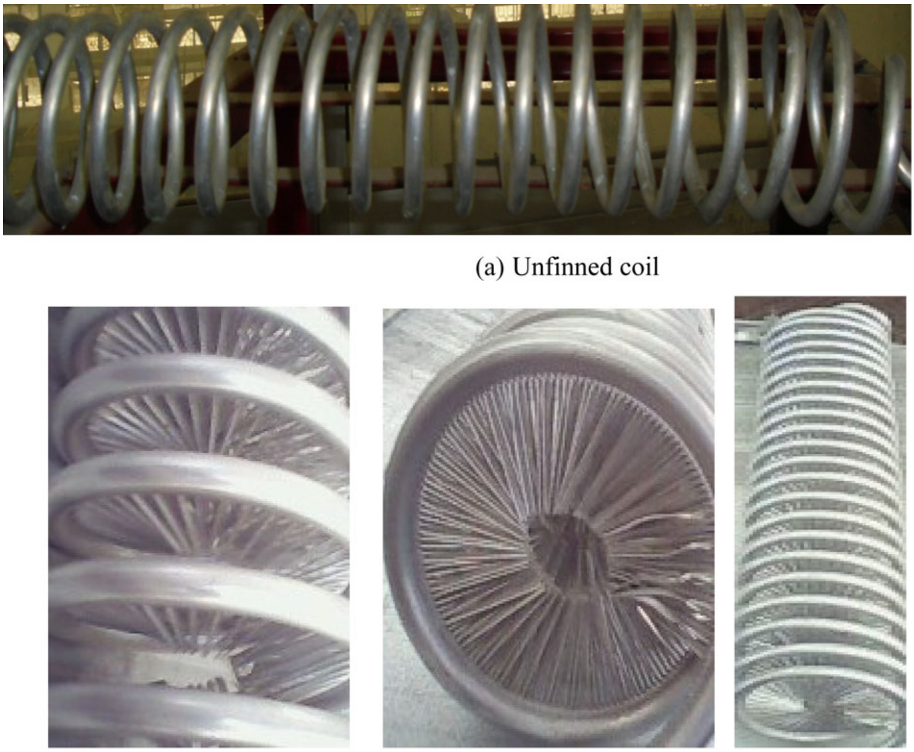


Fig. 2. Test section arrangement.

The test section is a shell and helical coil heat exchanger as shown in Fig. 2. The refrigerant passes inside the helical coil and the water passes outside the helical coil. The shell is a transparent cylindrical tube of length 70 cm. Four shells of different diameters 10, 15, and 20 and 25 cm are used. The shell was thermally insulated by 5 cm glass wool. The helical coils are made from thin aluminum tubes of 8 mm inside diameter and 0.5 mm thickness. Two coils are tested, one without fins and the other with external radial fins. The fins are wire strips fins. The dimensions of the wire strips are 30 mm × 2 mm × 0.5 mm (length × width × thickness). The spacing between each wire



(a) Unfinned coil

(b) Helical coil with external radial fins (different views)

Fig. 3. Photographs of tested helical coils.

and the other is 0.7 mm. Figure 3 shows the geometry of the two coils. The geometric parameters of the coils are: $d = 0.008$ m, $D_c = 2$, $R_c = 0.08$ m, length of straight part of the coil tube at ends = 0.06 m, $b = 0.02$ m, and the number of turns = 24, as shown in Fig. 2.

Measuring instruments were used to measure the physical quantities which are necessary to investigate and quantify the heat transfer in the test section. Surface temperature of the helical coil is measured by five thermocouples embedded on the outside surface of the coil tube. Two thermocouples are used for measuring the coil temperatures at the inlet and outlet of the test section. The other three thermocouples are equally distributed along the length of the coil. Two thermocouples props are used to measure the water temperature at the inlet and exit of the test section. All thermocouples are 0.8 mm K-type. The thermocouple wires are connected to data logger reading device for temperature recording. All thermocouples were calibrated by using standard thermometer and the error measurements within ±0.2 °C. To measure water flow rate, a graduated vessel having a maximum capacity of 10 L and a minimum division of 0.01 L is used to collect water at the exit of the test section. A stop watch of 0.01 s resolution is used for measuring the time of water collection.

The experiments were carried out at Reynolds number (Re) between 271 and 674 and Grashof number (Gr) in the range $5.6 \times 10^{10} < \mathrm{Gr} < 3.5 \times 10^{11}$. These ranges are identical to those in the actual use of a water cooler. The system was allowed to come to steady state conditions before data collection. Steady-state condition was considered to be achieved when the variation of temperature reading is within 0.2 °C for 30 min.

2.2 Data reduction

Heat transfer rate q was calculated by making heat balance of the cooling water at the inlet and exit of the test section and the refrigerant vapor inlet to the test section and the condensate exit from the test section as follows; neglecting the heat losses from the test section due to its excellent thermal insulation:

$$q = \dot{m}_{cw} C_p (T_{cwo} - T_{cwi}), \tag{1}$$

$$q = \dot{m}_c h_{fg}, \tag{2}$$

where $\dot{m}_{cw}$ and $\dot{m}_c$ are the cooling water flow and condensate flow rates, respectively. Here, T_{cwi} and T_{cwo} are the cooling water temperatures at inlet and exit of test section while C_p and h_{fg} are the specific heat of the cooling water and the heat of vaporization of the refrigerant, respectively. In our calculations, the heat transfer rate is calculated from the water side as it was difficult to measure the refrigerant condensate rate and the refrigerant conditions across the coil. An estimation of the heat losses from the insulated outer shell was within 1.5%.

The mean outside heat transfer coefficient was calculated from

$$h = \frac{q}{\pi\ d\ L\ (T_s - T_w)}, \tag{3}$$

where T_s is the coil outside average surface temperature, $T_w = (T_{cwi} + T_{cwo})/2$ is the average cooling water temperature, d is the tube outer diameter, and L is the coil tube length. The average tube surface temperature is calculated as the arithmetic mean of the local tube wall temperatures at the thermocouple locations. The average Nusselt number Nu is calculated from

$$Nu = \frac{h\ D_h}{k_w}, \tag{4}$$

where k_w is cooling water thermal conductivity and D_h is the hydraulic diameter calculated from [19]:

$$D_h = \frac{D^2 - \pi\ D_c d^2 \gamma^{-1}}{D + \pi\ D_c d \gamma^{-1}},\ \gamma = \frac{b}{\pi D_c}, \tag{5}$$

where D is the shell diameter, D_c is the coil diameter, d is the coil tube outside diameter, and γ^{-1} is the dimensionless turns of the tube and b is the coil pitch.

Cooling water Reynolds number Re based on the hydraulic diameter defined by Eq. (5) is calculated from:

$$\mathrm{Re} = \frac{\rho_w \ V_w \ D_h}{\mu_w}, \tag{6}$$

where $\rho_w, \ \mu_w$ are the density and viscosity of cooling water and $V_w = \frac{\dot{m}_{cw}}{\rho_w \pi \ D^2/4}$ is the average cooling water velocity in the shell.

Grashof number based on the average temperature difference between the coil surface and cooling water is calculated from

$$Gr = \frac{{\rho_w}^2 g \ \beta \ {D_h}^3 (T_s - T_w)}{{\mu_w}^2}, \tag{7}$$

where β is the thermal expansion coefficient of water and g is the acceleration of gravity.

Combining Eqs. (1) to (4), the expression of Nu can be put in the form:

$$Nu = f(x_1, x_2, ...x_n), \tag{8}$$

where x_1 to x_n are all the variables ($T_s \ T_w, \ T_{cwi}, \ T_{cwo}, \ \dot{m}_{cw}, ...$) that affect the experimental determination of *Nu*. The uncertaintyΔNu in the value of *Nu* was estimated based on the procedure of Holman and Gajda [29] and is expressed as follows

$$\Delta Nu = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial NU}{\partial x_i} \Delta x_i \right)^2}, \tag{9}$$

where Δx_i is the uncertainty in the x_i variable. It was found that the uncertainty for all data of *Nu* ranges was from 4% to 7%. The same was done for *Re* and *Gr* and it was found that their uncertainty lies in the range 4% to 6% and 6% to 8%, respectively.

3. Results and Discussion

The experimental work was performed to study the effect of the existence of external radial fins on coil performance and heat transfer characteristics. Studying the effects of shell diameter and flow characteristics (*Re* and *Gr*) were also an aim of this study. The experiments were carried out for different inlet cooling water temperatures, inlet cooling water flow rates, and shell diameters for finned and unfinned helical coils. The average Nusselt number was found to be dependent on coil surface fins conditions, shell diameter, and flow parameters *Re* and *Gr*.

3.1 Effect of flow parameters

Varying cooling water flow rate and cooling water inlet temperature will affect coil surface temperature and flow velocity on the coil which lead to changes in Reynolds number and Grashof numbers of the flow around the coil. Increasing water flow rate will increase *Re* and raising the temperature of the coil surface by increasing inlet water temperature and decreasing water flow rate will increase Gr. Figures 4 to 6 show the variation of *Nu* with *Re* and *Gr*, respectively, for finned and unfinned coils at different shell diameters. The figures show that both *Re* and *Gr* have an effect on *Nu* and this indicates that the flow regime lies in the mixed convection region. Figures 4 to 6 also show the increase of Nusselt number with increasing Reynolds and Grashof numbers. The trend is the same for finned and unfinned coils at any shell diameter. Increasing *Nu* with *Re* can be attributed to the increase of the convection current with increasing the velocity of cooling water (i.e., increasing *Re*). The increase of *Nu* with *Gr* is attributed to the increase of buoyancy forces with increasing tube surface temperature by increasing cooling water inlet temperature and decreasing cooling water flow rate. Increasing buoyancy force produces more convection currents which improves heat transfer.

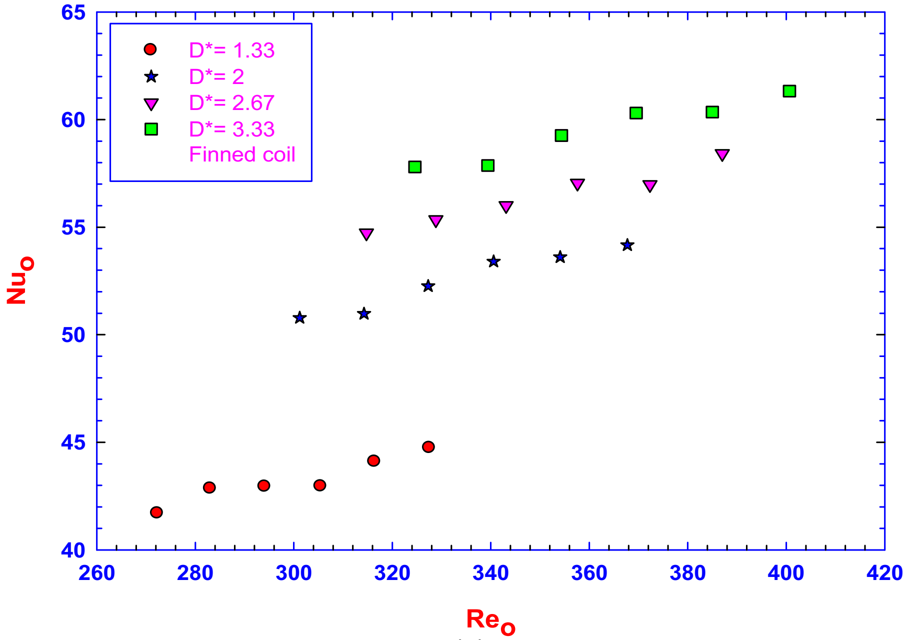
3.2 Effect of shell diameter

Figures 4 to 6 also show the effect of shell diameter on Nusselt numbers for finned and unfinned coils. The figures show the increase of Nu with the increase of the shell diameter for the same *Re* and *Gr*. The trend is the same for finned and unfinned coils. The figures also show that the rate of increase of Nu ($\Delta Nu/\Delta D$) with increasing shell diameter decreases with increasing shell diameter. For example, Fig. 4(a) shows that increasing shell diameter (D^*) from 1.33 to 2 increases *Nu* from 45 to 53 (i.e., by 18%), however, increasing D^* from 2.67 to 3 increases *Nu* from 55 to 57.5 (i.e., by 4.5%). This reveals that it is worthless to increase the shell diameter beyond a certain limit as the enhancement in Nu will be negligible.

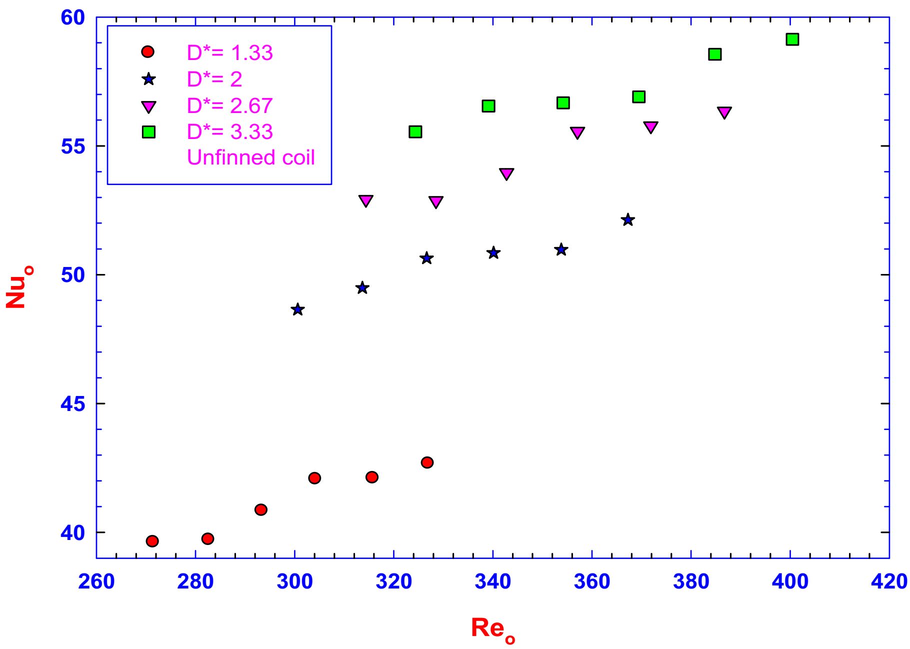
Figures 5 and 6 show that at small shell diameter $D^* = 1.33$, the effect of Gr on NU vanishes, however, the effect of *Re* on *Nu* is the dominant. This can be attributed to the increase of the flow velocity and convection current with decreasing shell diameter. On the other hand, increasing shell diameter decreases flow velocity which leads to a higher coil surface temperature and accordingly higher buoyancy force and the free convection heat transfer having considerable effect.

3.3 Effect of using external radial fins

Figure 7 shows the effect of using external radial fins on the outside surface of the coil. As shown in the figure, using external radial fins enhances heat transfer and Nusselt number. Table 1 gives the amount of enhancement in *Nu* numbers due to using fins. As shown in the table, the enhancement of *Nu* is about 5% to 6%. The enhancement of *Nu* can be attributed to the increase of heat transfer surface area by using fins. The percentage of *Nu* enhancement due to using external fins is relatively small versus what was expected. This can be attributed to the decrease of the temperature difference between coil surface and cooling water due to heat transfer rate enhancement. Decreasing the temperature difference decreases the buoyancy force and consequential decreases the free convection heat transfer share. On the other hand, this small enhancement in Nu will have a more valuable effect on the overall heat transfer coefficient between refrigerant and cooling water and will lead to smaller and more compact heat exchangers.

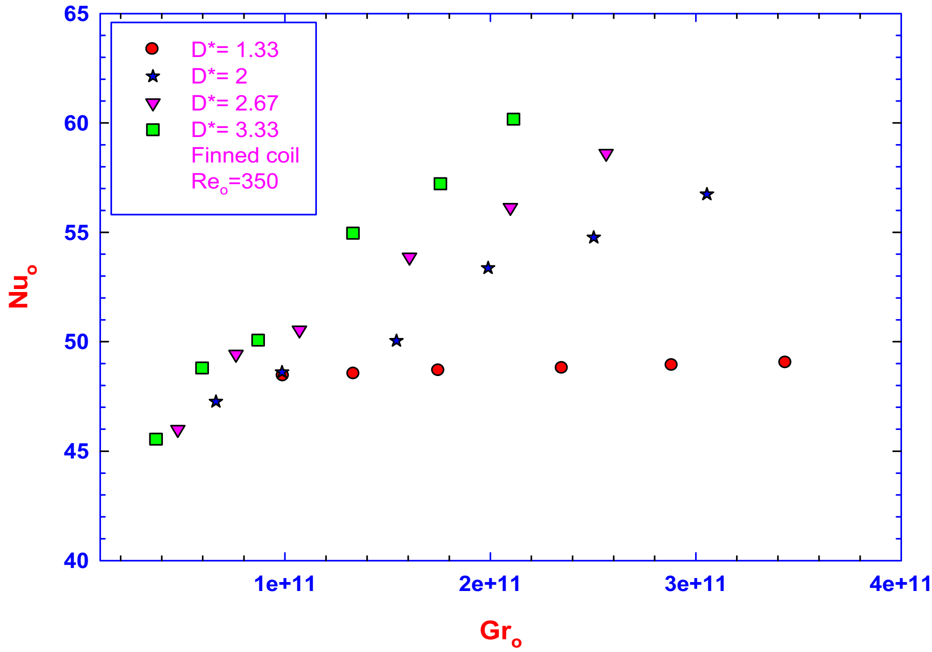


(a) Finned Coil

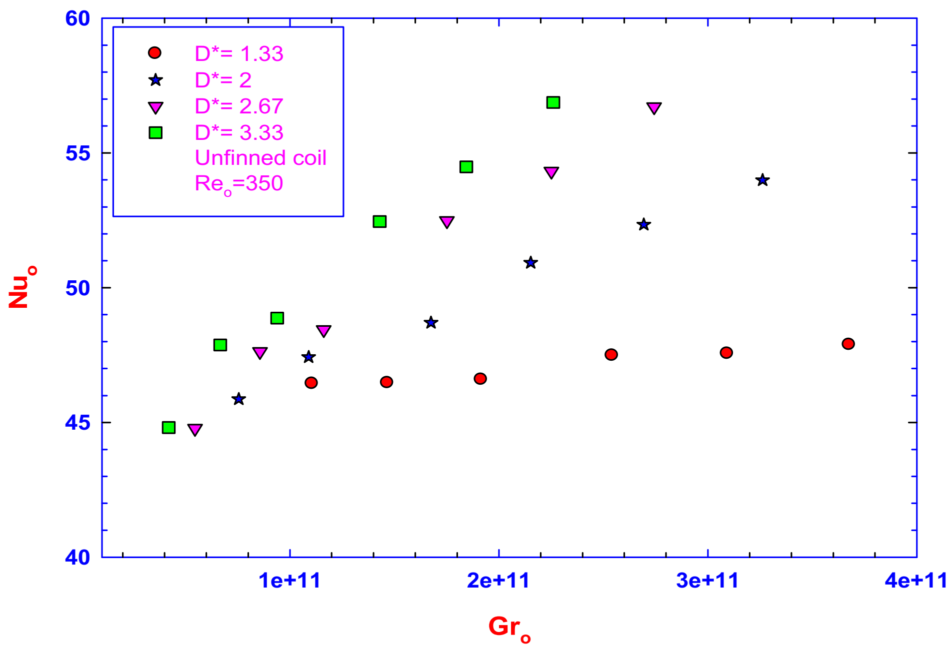


(b) Un Finned coil

Fig. 4. Variation of Nusselt number with Reynolds number.

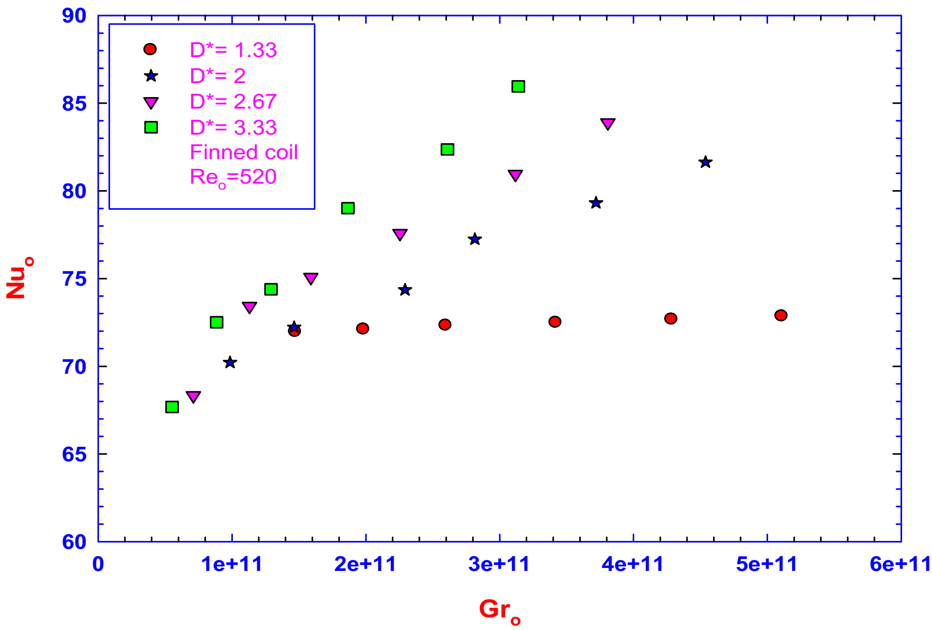


(a) Finned Coil

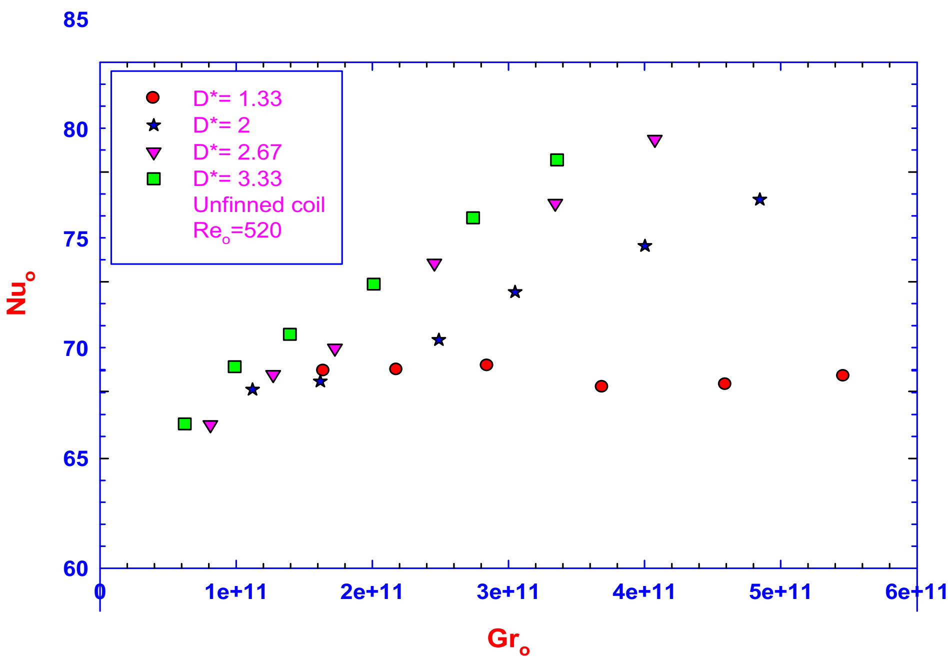


(b) Un Finned Coil

Fig. 5. Variation of Nusselt number with Grashof number at Re = 350.

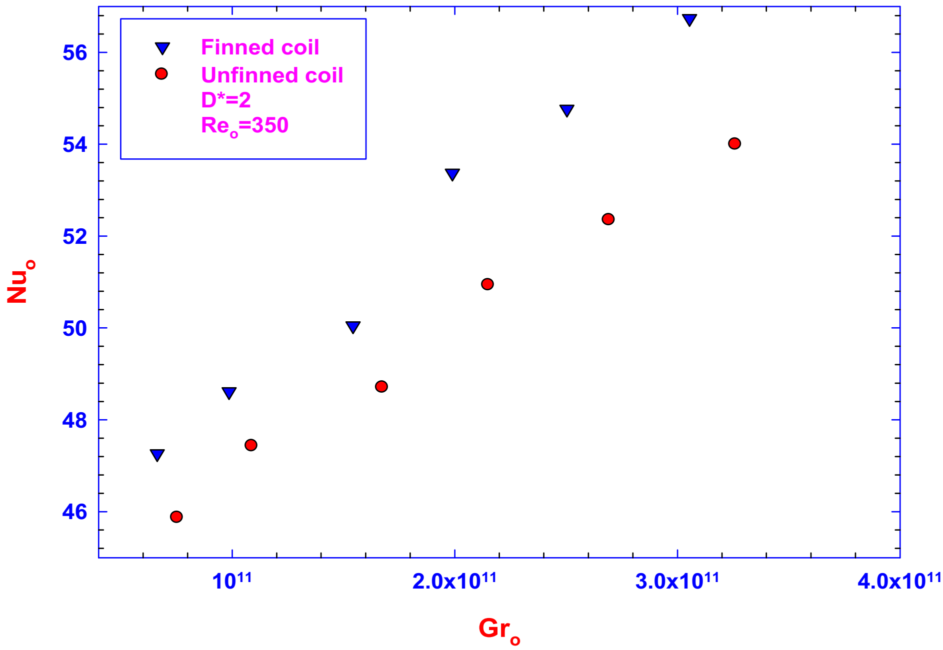


(a) Finned Coil

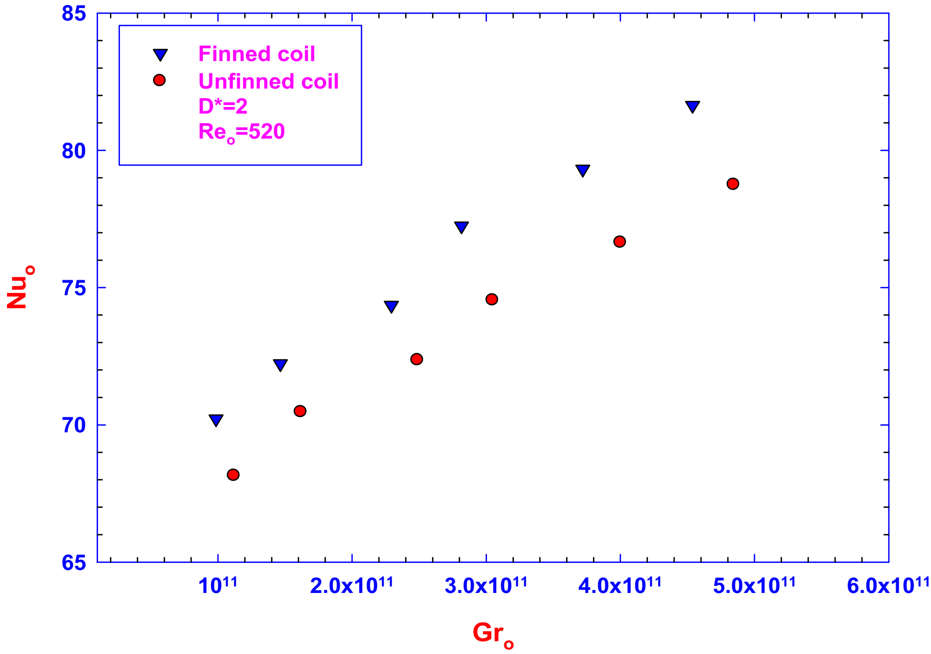


(b) UnFinned Coil

Fig. 6. Variation of Nusselt number with Grashof number at Re = 520.



(a) *Re* = 350



(b) *Re* = 520

Fig. 7. Effect of existence of external radial fins on Nusselt number.

Table 1. Nu for different shell diameters (Gr) = 2×10^{11}

D_s / Re	10.16	15.24	20.32	25.4	Case
520	72	73.5	76.5	79	Finned
350	48.5	53	55	63.5	
520	69	71	73.5	75	Unfinned
350	46.5	50	53	55	

4. Experimental Correlation and Comparison with Previous Work

New correlations were developed to predict Nusselt number for a shell and helical coil heat exchanger with and without external radial fins in terms of *Re*, *Gr*, and D^*. Based on the experimental results and regression analysis, the following functional relationships were obtained to predict Nusselt number for each coil with the smallest uncertainty:

$$Nu = 0.5\ \mathrm{Re}^{0.506}\ Gr^{0.057} D^{*0.249}\ \text{(for unfinned coil)} \tag{10}$$

$$Nu = 0.5\ \mathrm{Re}^{0.492}\ Gr^{0.062} D^{*0.244}\ \text{(for finned coil)}\,. \tag{11}$$

These equations were derived from the experimental results of data in the ranges $271 < Re < 674$ and $5.6 \times 10^{10} < Gr < 3.5 \times 10^{11}$. The calculated average Nusselt numbers ($Nu_{\mathrm{correlated}}$) from Eqs. (10) and (11) are plotted versus experimental average Nusselt number ($Nu_{\mathrm{experimental}}$) in Fig. 8. As shown in the figure, the maximum deviation between the experimental data and correlation prediction is ±8.85% revealing that the proposed correlations can predict experimental data with fair agreement. Figure 9 presents a comparison between the present experimental results and those obtained by Salimpour [19] for shell and helical coil heat exchangers without external fins. Figure 9 also shows the trend of the empirical correlation proposed by Salimpour [19] for $271 < \mathrm{Re}_o < 370$ and $D^* = 1.67$. No data are available in the literature for coils with radial external fins. As shown in the figure, the results obtained by Salimpour [19] for $D^* = 1.67$ lies between the present results for $D^* = 1.33$ and $D^* = 2$ proving satisfactory agreement.

5. Summary and Conclusion

A mixed convection heat transfer investigation of a shell and helical coil water cooler with and without external radial fins were conducted for different shells diameters and flow parameters. Two helical coils; one with a plain tube and the other with external radial fins, were tested in four shells of different diameters. The study's aim was to enhance water cooler performance in a trial to improve its compactness. Results shows Nu enhancement with insertion of external radial fins and with increasing shell diameter to helical coil diameter ratio up to a certain limit. For unfinned and finned coils of different shell diameters, results show the increase of Nusselt number with increasing Reynolds and Grashof numbers. Correlations were predicted to give Nusselt number in terms of *Re*,

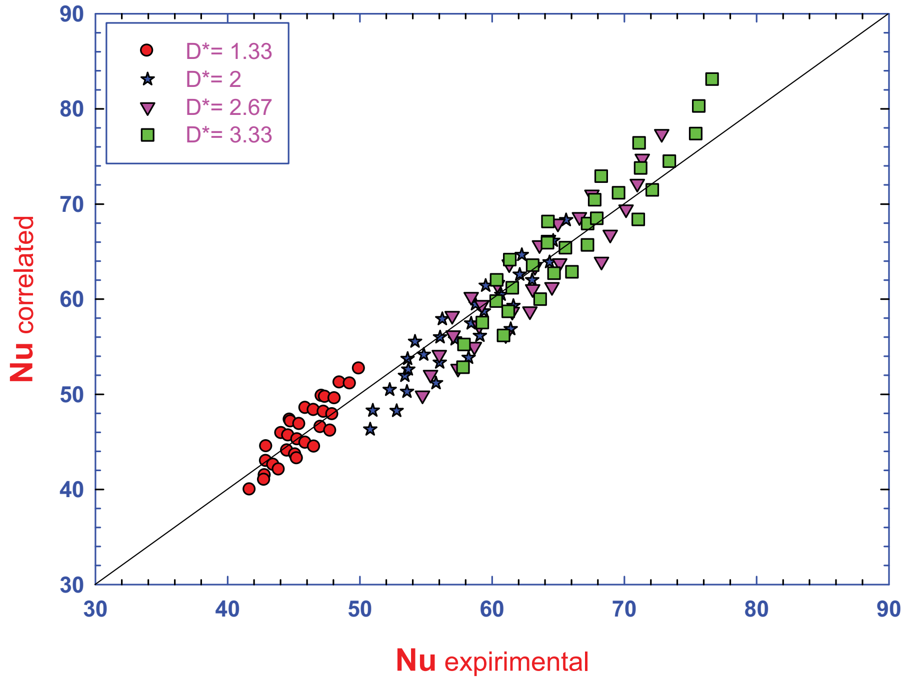


Fig. 8. Comparison of correlation prediction with experimental data.

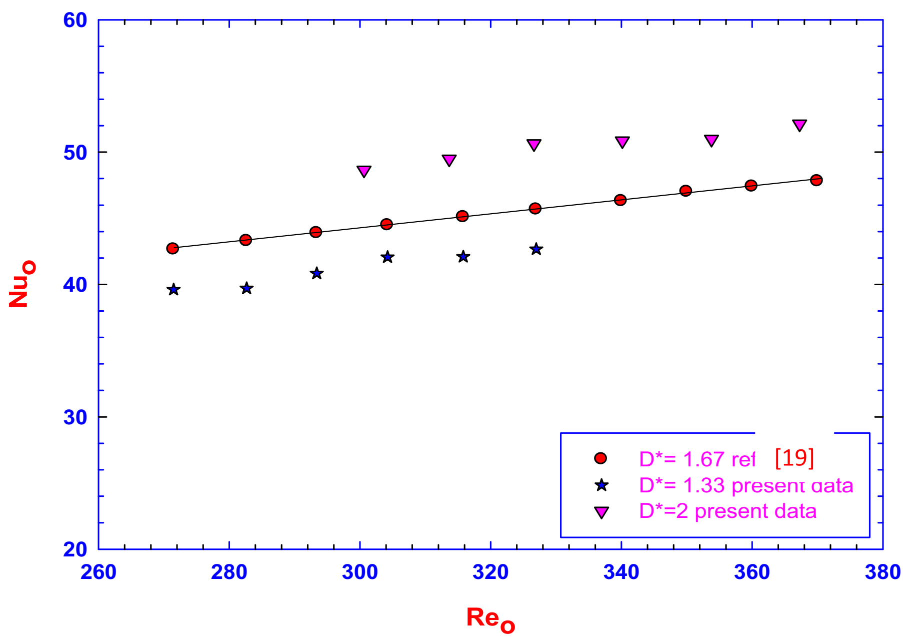


Fig. 9. Comparison of correlation prediction with experimental data.

Gr, and D^* for finned and unfinned helical coils. Correlation predictions were compared with present and previous experimental results and good agreement was obtained.

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