EXPERIMENTAL INVESTIGATION ON THERMAL PERFORMANCE OF NANOCOATED SURFACES FOR AIR-CONDITIONING APPLICATIONS

by

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Extended surfaces are frequently used in heat exchanging devices for increasing the heat transfer between a primary surface and the surrounding fluid. Heat transfer augmentation by providing an extended surface to air side, increases weight of the system, obstruct air-flow, increases pressure drop and pumping power to keep the fluid stream flowing. These major problems will be minimized with the help of recent developments in the field of nanotechnology that provides an avenue to augment heat transfer by providing a coating of nanometer thickness on the heat exchanger surface. The present work aims to investigate the effect of the nanocoating on the heat exchanger surface with the experiments. The experimental results show that the presence of nanocoating does not have any effect on drag change, where the actual pressure drop of the coil for air velocity of 0.6 m/s, 1.6 m/s is 20 Pa and 60 Pa, respectively. For nanocoated surface, 24.2% improvement in heat transfer rate was observed and transient temperature response of the nanocoated heat exchanger tube is higher than uncoated one for a given experimental conditions.

Key words: *heat exchanger, heat transfer, nanocoating, pressure drop*

Introduction

In the past years, heat transfer enhancement technology has been developed and widely applied to heat exchanger applications like, refrigeration, automotives, process industry, solar water heater, *etc*. The usual goals are to reduce the size of a heat exchanger required for a specific heat duty and to upgrade the capacity of existing heat exchange equipment [1-3]. The present trend is moving toward components with augmentation, enhancement or intensification. However, the heat transfer augmentation by providing an extended surface to air side increases pressure drop and obstruct airflow [4]. The heat transfer enhancement in heat-exchange equipment becomes a challenge for engineers in industry. Worldwide demand for efficient, reliable and economical heat exchange equipment is accelerating rapidly, particularly in large-scale power and process industry, refrigeration and air-conditioning systems [5-7]. For better air side heat transfer performances, great improvements in energy conservation and the protection of the environment are possible. Many research works has been carried to improve the corrosion resistance of metals in environment conditions, in order to expand their applications by nanocoating [8-11]. Movaen *et al*. [12] reported the effects of super hydrophobic nanocoating on frictional drag force produced by sol-gel coating of the aluminum sample using TiO_2 nanoparticles. They suggested developing of super hydropho-

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bic coating might also be utilized for anti-fouling applications owing to its self-cleaning effects. Enright *et al*. [13] has experimentally demonstrated that a 25% higher overall heat flux and 30% higher heat transfer coefficient can be achieved using nanocoated copper oxide (CuO). They found properly designed super hydrophobic non-structural surfaces cannot only allow for easy droplet removal at micrometre length scales during condensation but also promise to enhance heat transfer performance. Wang *et al*. [14] has developed good super hydrophobic coating on the copper sample to study anti icing performance. In the icing test, it was observed that the starting time for icing is completely different from the plain and coated sample. The results prove that the coated surface with a high contact angle and low contact angle hysteresis cannot only be effective in delaying the starting icing time but also increase the whole icing process time compared with the plain copper surface. The results revealed that the nanofluorocarbon coating over the copper surface has a better anti icing performance. Forrest *et al.* [15] investigated the thin-film coatings prepared by layer-by-layer assembly and its effects on the increase of critical heat flux and nucleate boiling heat transfer. For a known heat flux, the hydrophobic surface has the greatest enhancement in the nucleate boiling heat transfer coefficient up to about 100%, whereas the hydrophilic surface suffers a slight deprivation of 10%. The super hydrophilic surface conversely shows substantial degradation of about 50% in the heat transfer coefficient, only recovering at very high-heat fluxes. They concluded that the boiling heat transfer coefficient increases with reduced wettability.

The following observations were made, based on the previous review: nanocoating enhances heat transfer capabilities, through its high area to volume ratio, nanocoating prevents adhering of water droplets on the surface that eliminates the formation of thin film on the heat exchanger surface [16-18]. Nanocoating results in low energy consumption and requires less maintenance. Among the nanoparticle, the multiwalled carbon nanotube (MWCNT) is more suitable for nanocoating applications due to their high heat transport properties. Considering the pressing need to enhance the heat transfer performance on airside in developing an energy efficient thermal system and recent developments in the field of nanotechnology, the specific objective of the present work is to explore the effectiveness of nanocoating MWCNT on the heat exchanger surface through thermal performance analysis.

Experimental set-up

The experimental trials were carried out to determine the cooling performance of the nanocoated heat exchanger.

Design calculations

The duct is used to deliver and remove air from the heat exchanger; the dimensions of the duct used in the experiment are calculated [19-21]. Dimensions of the duct, frictional loss due to the presence of duct and minimum pumping power required to pump the fluid through the heat exchanger are calculated as follows.

Laminar-boundary layer thickness:

$$\delta = \frac{5x}{\mathrm{Re}_x^{1/2}} \tag{1}$$

where $\mathrm{Re}_x < 5 \cdot 10^5$.

Where volume and flow rate value is $9.16 \cdot 10^{-5}$ m^3 and $9.16 \cdot 10^{-5}$ m^3/s at a velocity of 2.5 m/s. So the mafximum velocity should be in the range of 2.5 m/s. Given this velocity, Reynolds number can be computed to indicate whether the inner flow is laminar or turbulent. This will most likely to be fully-developed, laminar flow for velocity 0.1 m/s with Reynolds number 1092 and turbulent flow for 0.5-2.5 m/s with Reynolds number ranging from 5463-27318 at different temperatures like 343 K, 338 K, and 333 K, respectively.

Frictional pressure drop encountered in the duct is calculated using Darcy-Weisbach equation:

$$\Delta p = \frac{\lambda \rho l u^2}{2 d_h} \tag{2}$$

where Δp [Pa, N/m^2] is the pressure loss, λ – the Darcy-Weisbach friction coefficient, l = 1.1 [m] – the length of duct or pipe, d_h = 0.19 [m] – the hydraulic diameter, and ρ = 1.2 [kg/m^3] – the density.

Frictional factor for Darcy Weisbach equation is calculated using Colebrook equation:

$$\frac{1}{\sqrt{\lambda}} = -2\log\left[\frac{2.51}{\mathrm{Re}\sqrt{\lambda}} + \frac{\dfrac{k}{d_h}}{3.72}\right] \tag{3}$$

where k is the roughness of duct tube surface. For acrylic material, the absolute roughness coefficient is 2 μm. Range of air velocity in the set-up – u = 0.6-1.6 m/s, kinematic viscosity – $\nu = 16 \cdot 10^{-6}$ m^2/s, λ = 0.0262, and $\Delta p_{\text{friction}}$ = 0.254 Pa.

Experimental details

The schematic of the experimental set-up is shown in fig 1. It consists of a heat exchanger, blower, pump, temperature regulator, anemometer, and digital micro manometer. Temperature on the surface of the coil is measured by resistance temperature detector (RTD) (Class A) and the temperature measurements are made on five different locations 500, 1000, 1500, 2000, and 2500 mm, respectively, from top of the coil. An anemometer is used to measure the velocity of air-flowing through the duct and differential pressure transmitter is used to measure the differential pressure across the heat exchanger. Hot water through the coil is circulated with the help of pump. During the experiments, temperature on the surface of the coil is recorded continuously with the help of data logger. Temperature of hot water at the inlet of the coil is maintained with the help of constant temperature bath controller.

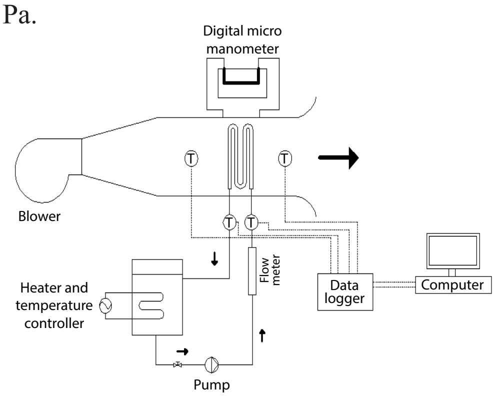

Figure 1. Schematic of experimental set-up

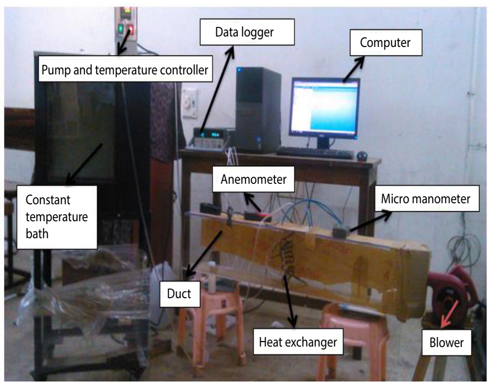

Figure 2. Photographic view of experimental set-up

Results and discussion

The results primarily demonstrate the influence of air velocity, hot water temperature, and coating of nanomaterial on the temperature distribution along the different locations on the heat exchanger. The morphology of the CNT was used in the work is obtained from MWCNT, Cheap Tubes Inc. According to the specification on condition by the manufacturer, the MWCNT have an average diameter of 30-50 nm, length of 10-20 nm, and specific surface area of 60 m^2/g, density 2.1 g/cm^3, and purity of 95%. The photographic image of Cu tube heat exchanger is shown in fig. 3 which having the iner diametar (ID) of 7 mm, outher diametar (OD) of 10 mm and length of 2600 m and nanoparticle-coated heat exchanger is also shown in fig. 4. The MWCNT nanocoating was produced by trial and error method to repetitively, test the best amount of solvent (MWCNT/water nanofluid) to dilute the bulk material foreasy spray [22]. For microstructural analysis, SEM (Tescon vega-3) was performed. From the SEM image, figs. 5 and 6, CNT were visually observed to be non-uniform along the surface with the thickness of 2 μm.

Figure 3. Uncoated heat exchanger coil

Figure 4. Coated heat exchanger coil

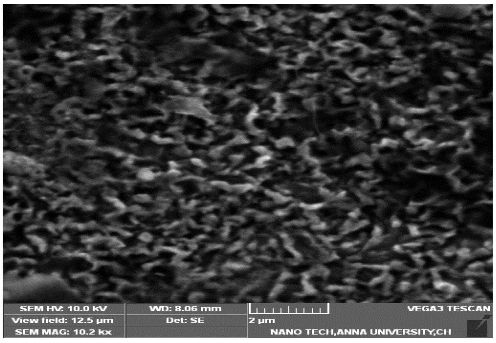

Figure 5. The SEM image of the MWCNT on the surface of Cu tube

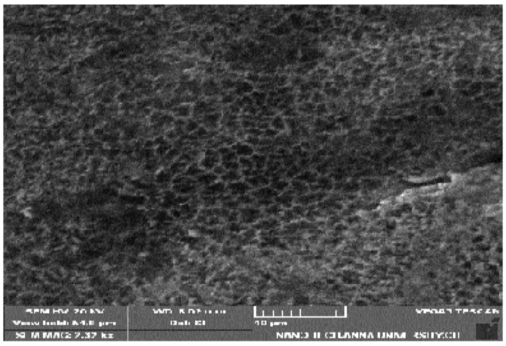

Figure 6. The SEM image of coated tube

Influence of nanoparticle on wall temperature

The collected data showed the wall temperature for the untreated and CNT treated heat exchanger surfaces. The results of the experimental study are shown below.

Figures 7-9 shows the temperature distribution on the surface of the heat exchanger in the measurements location 500, 1000, 1500, 2000, and 2500 mm, respectively, for air velocity 0.6 and 1.6 m/s, and for various water temperatures. The temperature decreases gradually for for non-coated heat exchanger along the vertcal length of the exchanger and reaches minimum at the bottom of the coil. The temperature of the coated coil shows lower temperature value than the uncoated one. This is due to the fact coated material increases the surface area of coil by act as pin fin.

Figures 7-9 illustrates the results, indicate irrespective of the hot water, temperature on the surface of heat exchanger decreases from the top of the coil. However, the magnitude of temperature drop depends on coating material, geometry, and thickness. It is also observed that the decrease in temperature for higher water inlet temperature is more compared with that of low water inlet temperature. This can be explained by the higher degrees of convection at higher capacity of heat exchanger, which in turn facilitate a more efficient heat removal.

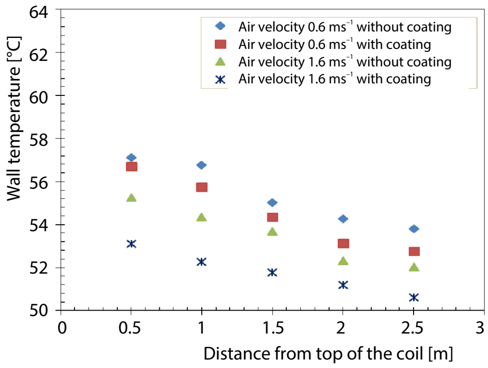

Figure 7. Influence of nanoparticle on wall temperature at water inlet temperature of 60 °C

Augmentation in heat transfer rate

The flow and heat transfer have been analyzed to study the influence of heat power input and boundary cooling. The heat transfer rate of nanocoated heat exchanger increased with increase in hot water inlet temperature, heat transfer rate observed from figure 10 to be 139 W at air velocity of 1.6 m/s for uncoated tube and for coated it be 163 W at 60 °C water inlet temperature, fig. 10. The maximum improvement in heat transfer rate by 24.2% for water inlet temperature of 70 °C was noticed.

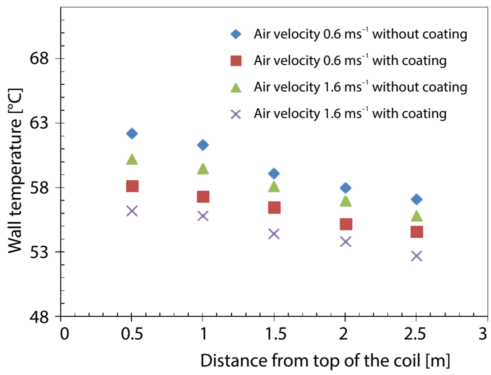

Figure 8. Influence of nanoparticle on wall temperature at water inlet temperature of 65 °C

The effective convective heat transfer coefficient was extracted from the temperature profiles and the effective convective coefficient was increased by 12%, indicating the coating of CNT indeed increased the performance. Further improvements in cooling performance are likely to occur with higher CNT densities and longer tube lengths. Nanotubes, which are realized as nanotube bundles, can be made of any suitable material with high thermal conductivity includ-

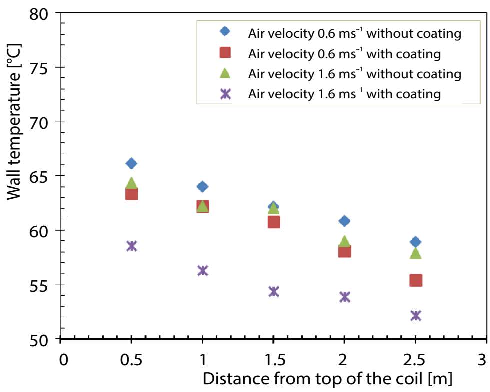

Figure 9. Influence of nanoparticle on wall temperature at water inlet temperature of 70 °C

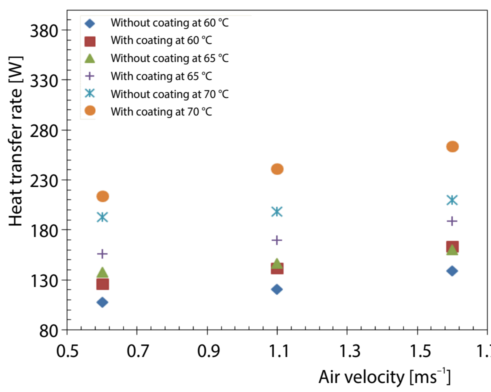

Figure 10. Augmentation in heat transfer rate

ing carbon or boron nitride. Nanotubes are advantageously spaced apart by some distance up to 1mm so that air or another cooling fluid can circulate between the nanotubes.If the nanotube spacing is on the order of 100 nm, surface area can be increased by a factor of 10000. Accordingly, heat exchanger can dissipate considerably more heat than its conventional counterparts. The nanocoatings can be applied to any material, which require efficient heat removal by convective heat transfer from or to the system.

Table 1. Pressure drop cross heat exchanger

Air velocity [ms^{-1}]	0.6	1	1.6
Pressure drop across the bare tube heat exchanger [kPa]	0.02	0.03	0.06
Pressure drop across the nanoparticle coated heat exchanger [kPa]	0.02	0.03	0.06

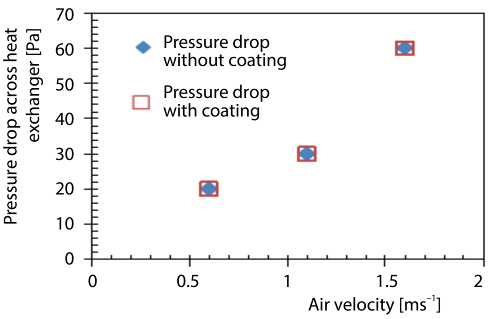

Figure 11. Pressure drop across heat exchanger

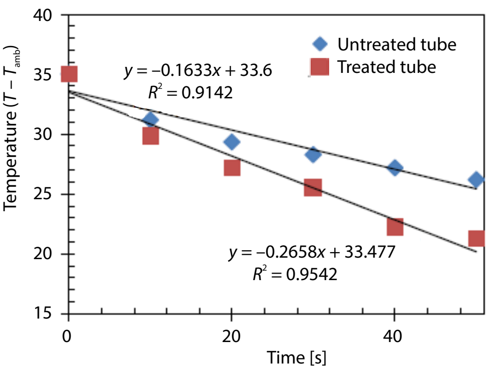

Figure 12. Cooling profiles of uncoated and MWCNT coated tube

Pressure drop across heat exchanger with and without nanocoating

From tab. 1 and fig. 11 it is observed that there is no change in pressure drop occurred for the heat exchanger coil, where the actual pressure drop of the coil for air velocity of 0.6 m/s and 1.6 m/s are 60 Pa, and 20 Pa, respectively.

Average slope of treated tube is –0.2658, which is higher than untreated tube slope –0.1633, this in turn indicates the quicker temperature drop of treated tube with untreated one. From the fig.12, it is observed that the same temperature drop of 25 °C reaches 20 seconds before for MWCNT treated tube.

Conclusion

Experimental investigation were carried out in a heat exchanger with nanocoating under forced convection. The experiments were performed to reveal the effect nanoparticle on thermal hydraulic performance of the heat exchanger. The following conclusions are made based on the outcome of the present work.

The increase in surface area of 46% is achieved by providing the spacing of 2 μm compared to that of bare tube area. The results also showed that the increase in surface area tends to decrease with respect to increase in spacing between pin fins. The presence of nanocoating does not have any effect on drag change, as the thickness of viscous sublayer is much higher than the thickness of the nanocoating. The actual pressure drop of the coil for air velocity of 0.6 m/s, 1.6 m/s is 20 Pa and 60 Pa, respectively. For the nanocoated surface there is an improvement in heat transfer rate of 24.2% is achieved at air velocity of 1.6 m/s , water inlet temperature of 70 °C and also there is significant drop in wall temperature for the similar given conditions. Transient temperature response of nano treated heat exchanger tube is higher than untreated one, for given initial conditions temperature drop of 25 °C reaches 20 seconds before for the MWCNT treated tube. It is possible that size of the heat exchanger for a given specific heat duty can be reduced considerably through the application of nanocoating.

Nomenclature

d_h – hydraulic diameter, [m]
h – heat transfer coefficient, [$Wm^{-2} K^{-1}$]
K – thermal conductivity, [$Wm^{-1}K^{-1}$]
k – roughness of duct tube surface
l – length of duct or pipe, [m]
Δp – pressure loss, [Pa]
Re – Reynolds number
T – temperature, [K]
u – velocity of air, [ms^{-1}]

Greek symbols

λ – Darcy-Weisbach friction coefficient
ν – kinematic viscosity, [m^2s^{-1}]
ρ – density, [kgm^{-3}]

References

[1] Pathaka, S, *et al*., Carbon Nanotube (CNT) Fins for the Enhanced Cooling of Shape Memory Alloy Wire, *Proceedings*, SPIE 6926:6969292K91K-2, 2009
[2] Senthilkumar, R., *et al*., Experimental Investigation on Carbon Nanotubes Coated Brass Rectangular Extended Surfaces*, Applied Thermal Engineering*, *50* (2013), 1, pp. 1361-1368
[3] Senthilkumar, R., *et al*., Analysis of Natural Convective Heat Transfer of Nanocoated Aluminium Fins Using Taguchi Method, *Heat and Mass Transfer*, *49*, (2013), 1, pp. 55-64
[4] Chi, Y. L., *et al*., Pool Boiling Heat Transfer with Nano-Porous Surface, *International Journal of Heat and Mass Transfer*, *53* (2010), 19-20, pp. 4274
[5] Sang, M. k., *et al*., Effects of Pressure, Orientation, and Heater Size on Pool Boiling of Water with Nan-Coated Heaters*, International Journal of Heat and Mass Transfer*, *53* (2012), 23-24, pp. 5199-5208
[6] Zhenyu, W., *et al*., Fire and Corrosion Resistances of Intumescent Nanocoating Containing Nano-SiO_2 in Salt Spray Condition*, Journal of Materials Science & Technology*, *26* (2010), 1, pp. 75-81
[7] Aliofkhazraei, M., *Nanocoatings: Size Effect in Nanostructured Films*, Springer, New York, USA, 2011
[8] Tun-Ping, T., Tun-Chien, T., Enhance Heat Dissipation for Projection Lamps by MWCNTs Nanocoating, *Applied Thermal Engineering, 51* (2013), 1-2, pp. 1098-1106
[9] Xu, Z., Buehler, M. J., Nanoengineering Heat Transfer Performance at Carbon Nanotube Interfaces, *ACS Nano*, *3* (2009), 9, pp. 2767-2775
[10] Shah, K. R., Sekulic, P. D., *Fundamentals of Heat Exchanger Design*, John Wiley and Sons Inc., New York, USA, 2003
[11] Cavaleiro, A., Hosson de J. T. M. , *Nanostructured Coatings*, Springer, New York, USA, 2006
[12] Moavena, Kh., *et al*., Experimental Investigation of Viscous Drag Reduction of Superhydrophobic Nano-coating in Laminar and Turbulent Flows, *Experimental Thermal and Fluid Science*, *51* (2013), Nov., pp. 239-243
[13] Enright, R., *et al*., Condensation on Superhydrophobic Copper Oxide Nanostructures, In ASME 2012, *Proceedings*, 3rd International Conference on Micro/Nanoscale Heat and Mass Transfer, Atlanta, Geo., USA, 419, ASME International, 2012
[14] Wang, H., *et al*., Effects of Nano-Fluorocarbon Coating on Icing, *Applied Surface Science*, *258* Pages (2012), 18, pp. 7219-7224
[15] Forresta, E., *et al*., Augmentation of Nucleate Boiling Heat Transfer and Critical Heat Flux Using Nanoparticle Thin-Film Coatings, *International Journal of Heat and Mass Transfer*, *53* (2010), 1-3, pp. 58-67
[16] Zhang, S., *Handbook of Nanostructured Thin Films and Coatings*, Taylor & Francis Group, Oxford, UK, 2010
[17] Bejan, A., *Convection Heat Transfer*, John Wiley & Sons Inc., 3ed, New York, USA, 2004
[18] Kakac, S., *Boilers, Evaporators, and Condensers*, John Wiley & Sons Inc., New York, USA, 1991
[19] Schlichting, H., Gersten, K., *Boundary-Layer Theory*, Springer, New York, USA, 8th ed., 2000
[20] Ozisik, M. N., *Heat Transfer – A Basic Approach*, McGraw-Hill, New York, USA, 1985
[21] Lipsman, L. R., Rosenberg, J., *A Guide to MATLAB: For Beginners and Experi-Enced Users*, Cambridge University Press, Cambridge, UK, 2006
[22] Tun-Ping, T., Tun-Chien, T., "Enhance Heat Dissipation for Projection Lamps by MWCNTs Nanocoating, *Applied Thermal Engineering*, *51* (2013), 1-2, pp. 1098-1106

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