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Single phase flow heat transfer and pressure drop measurements in doubly enhanced tubes

Ratnesh Raj, Narender Shankar Lakshman, Yagnavalkya Mukkamala*

School of Mechanical and Building Sciences, VIT University, Vellore, India

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ABSTRACT

Global in-tube flow heat transfer and pressure drop measurements were conducted with Water ($Pr = 4.42$) and Ethylene Glycol ($Pr = 80.5–135.7$), in the laminar-transition-turbulent flow regime ($Re = 400–39{,}750$) in double pipe heat exchangers fabricated from doubly enhanced tubes. The internal fin height to effective diameter ratio $e_{f,i}/d_{eff}$ varied between 0.0289 and 0.0314. The internal groove helix angles varied between 35 and 40°, while the internal fin pitch to height $p/e_{f,i}$ ratios varied between 3.4 and 5.36. The external fin density was 40 fins per inch (25.4 mm). In comparison to a smooth tube, tube-side (internal) heat transfer enhancements ($h_{deff,enhanced}/h_{deff,smooth}$) of up to 6% were measured with Ethylene Glycol in the laminar regime, while tube-side enhancement of up to 34% was measured with water in the turbulent flow regime. The overall heat transfer enhancement $(UA_{eff})_{enhanced}/(UA_{eff})_{smooth}$ based on the overall heat transfer coefficients was measured as 116% with water in the turbulent flow regime. Transition to fully developed rough pipe flow was detected in the enhanced tubes after a critical Reynolds number and the corresponding Prandtl number exponent was estimated to be 0.47. At constant pumping power, the maximum tube-side heat transfer enhancement was 6% with Ethylene Glycol in the laminar regime, and 18% with water in the turbulent regime. On a constant heat duty basis, the maximum reduction in pumping power was 23% with Ethylene Glycol in the laminar regime and 54% with water in the turbulent regime. Although the inner (tube-side) heat transfer enhancement is lower than that measured in conventional micro-fin tubes (inner grooves only), the overall heat transfer enhancement suggests that these tubes will be useful for designing heat exchangers deploying fluids of similar thermal conductivity in the tube and annulus. Typical applications include, but are not limited to, design of marine oil coolers, lubricating oil coolers, and sub-cooled sections of condensers.

1. Introduction

Single phase convective heat transfer enhancement in tubes finds applications, for example, in desalination plants, lubricating oil or engine oil coolers, and condensers of power plants. Vapor phase convective heat transfer augmentation may also be required in the superheated zone of refrigerant evaporators, steam boilers, air intercoolers for multistage compressors, etc. The objective of heat transfer augmentation is the reduction in the heat exchanger surface area for a specific heat duty or increased heat transfer rate for a given pump duty and exchanger surface area. Other objectives include, but are not limited to, lower mean temperature differentials, lower tube side pressure drops for a given heat duty, lower entropy increase, etc. The performance of an enhanced tube is evaluated by comparison with an equivalent smooth, or un-enhanced, tube.

Finned surfaces enhance the heat transfer rate by disturbing the thermal boundary layer and also by providing additional surface area. Spiral grooves provide additional heat transfer enhancement from the swirl effect that increases fluid transit time and tube wall–fluid interaction. Fins are deployed with low thermal conductivity fluids. Consequently, tubes with both internal and external fins (doubly enhanced tubes) are suitable for designing heat exchangers with fluids of similar thermal conductivity on the tube and shell side. Doubly enhanced tubes are frequently used in condensers and evaporators. However, their use in single phase applications is quite limited. A brief summary of state of the art in heat transfer enhancement with doubly enhanced tubes will now be presented.

Evraam Gorgy and Steven Eckels [1] reported local heat transfer coefficients for nucleate pool boiling of R-134a and R-123 on doubly

* Corresponding author. Tel.: +91 9345310732.
E-mail addresses: ratnesh_raj@hotmail.com (R. Raj), shankar.narender@gmail.com (N.S. Lakshman), yagnasmukkamala@vit.ac.in (Y. Mukkamala).

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Nomenclature

A	area (m^2)
A_i	nominal inner tube inner surface area
A_{eff}	effective inner surface area of inner tube (m^2)
A_{ff}	free flow area of inner tube (m^2)
c_p	constant pressure specific heat (J/kg K)
d, D	diameter (m)
d_{eff}	effective diameter of inner tube (m)
d_h	hydraulic diameter (m)
d_i	nominal inner diameter (m)
D_o	outer diameter
e	roughness (fin) height (m)
exp.	experimental value
f	Darcy friction factor
G	mass velocity (kg/m^2 s)
ΔH	pressure head (m)
h	heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
L	heat exchanger tube length (m)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number
Δp	measured differential pressure (Pa)
Pr	Prandtl number
p/e_f	fin pitch to height ratio
$\dot{Q}$	heat transfer rate (W)
R_1	constant pump duty efficiency index
R_2	constant heat duty efficiency index
Re	Reynolds number
T	temperature (K)
U	overall heat transfer coefficient (W/m^2 K)
u_{deff}	mean tube-side velocity (m/s)
$\dot{V}$	volume flow rate (m^3/s)

Subscripts

avg	average
b	bulk
hb	hot fluid bulk value
bottom	bottom parameter
c	coolant
cb	coolant bulk
c.p	constant property
corr	corrected value
deff	effective diameter
eff	effective
enhanced	enhanced tube parameter
eq	equivalent
exp.	experimental (measured)
f	fin
ff	free flow
h	hot fluid
i	inlet
m	manometer
o	outlet
smooth	smooth tube parameter
top	top
w	wall parameter

Superscripts

exponentexponent value in equation

Greek

α	helix angle (°)
β	fin tip angle (°)
μ	absolute viscosity (kg/m s)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
τ	time (s)

Abbreviations

HX	heat exchanger

enhanced tubes (with water and refrigerant side enhancements). Their findings indicate that the pool boiling structure on doubly enhanced tubes was significantly different from that observed on smooth or un-enhanced tubes.

Wen-Tao Ji et al. [2] reported turbulent heat transfer and friction factors for boiling and condensation of R-134a in doubly enhanced tubes. Predictive correlations developed from experimental data in 16 doubly enhanced tubes compared well with published data with 99% of the available data agreeing with the correlation with a relative deviation of ±40%.

Ayub et al. [3] designed a double enhanced carbon steel tube based, low charge ammonia spray evaporator. For a 1760 kW cooling capacity, there was a 83% reduction in charge, 14% decrease in overall pressure drop, while the tube length, number of tube passes and total cost decreased by 59%, 65%, and 71% respectively in comparison to the plain tube flooded condenser. Ooi [4] et al. reported heat transfer coefficients for low pressure steam condensing on the external integral fins (rectangular fins) of a copper–nickel tube. The internal fins were elongated helical ribs. Due to the low fin height, the integral external fins were flooded with the condensate, and hence no significant shell side enhancement was detected.

Similarly, Park and Jung [5] presented experimental heat transfer coefficients for data for condensation on a plain tube and four low fin, integral fin tubes of 19 mm diameter. In order to determine the optimum fin density, heat transfer experiments were carried out at a vapor temperature of 39 °C, with the wall subcooling of 3–8 °C. Low fin tubes yielded heat transfer coefficients that were 5.8 times the corresponding plain tube coefficients. Single-phase investigations in doubly enhanced tubes are obviously limited. This section will now present a summary of single-phase works in a related field, i.e., experimental studies in micro-fin tubes.

Single phase studies were reported extensively in micro-fin tubes with inner, spiral grooves. Micro-fin tubes were first developed by Fuji et al. [6] of Hitachi cable corporation, and later tested by Tatsumi et al. [7] for their use in air-conditioners, while Shinohara et al. [8] further improved the tube. Increasingly, micro-fins are also being used for enhancing single-phase convective heat transfer in tubes.

Al-Fahed et al. [9] reported heat transfer and pressure drop data for laminar flow in a horizontal micro-fin tube at Re $\leq$ 2300. The documented micro-fin tube heat transfer coefficients were on par with the corresponding smooth tube values, indicating the ineffectiveness of the micro-fins in enhancing heat transfer at such low mass velocities.

Ito and Kimura [10] obtained isothermal friction factors in a horizontal micro-fin tube in the transition regime. Koyama et al. [11] reported heat transfer and isothermal friction data with Ethylene Glycol in the transition regime in a horizontal micro-fin