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Condensation of R-134a inside a helically coiled tube-in-shell heat exchanger

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ABSTRACT

This article presents an experimental investigation of heat transfer and pressure drop characteristics of R-134a condensing inside a horizontal helical coil tube with the cooling water flowing in the shell in counter flow direction. The test runs are performed at vapor saturation temperature 35 ± 0.5 and 40 ± 0.5 °C for the mass flux varying from 100 to 350 kg m^{-2} s^{-1} and vapor quality ranging from 0.1 to 0.9. The flow regimes observed during the experiment have been plotted on Taitel and Dukler and mass flux versus vapor quality flow map. The transitions between different flow regimes have also been discussed. The effect of mass flux, vapor quality and saturation temperature on the heat transfer coefficient and pressure drop have been investigated. The experimental results of the helical coil tube are compared to straight tube. The thermodynamic advantage of helical coil over straight tube is evaluated in terms of enhancement parameter. The enhancement parameter is higher than one for mass fluxes lower than 200 kg m^{-2} s^{-1}. The correlations have been developed to predict two-phase Nusselt number and frictional pressure drop multiplier during condensation of R-134a inside horizontal helical coil tube.

1. Introduction

The condenser is an integral part of any refrigeration and air-conditioning system and the development of high performance condensers with the least pressure drop has always been a primary design consideration for engineers. The film resistance during condensation of refrigerants inside a tube often governs the overall heat transfer coefficient of any water cooled condenser. Therefore, the improvement of refrigerant side heat transfer coefficient shall improve the overall heat transfer coefficient of the condenser. The passive techniques to improve the refrigerant side heat transfer coefficient such as rough surfaces, swirl flow devices, coiled tubes do not require any external energy. Coiled tubes are essentially swirl flow devices, which facilitate forced convection heat transfer by creating secondary flows inside the tube [1]. The two-phase condensation phenomenon in helical tube is more complex than in the straight tube attributable to the centrifugal force due to its curvature. The vapor phase of the fluid that flows at a high velocity in the tube experiences high centrifugal force than the liquid phase at the tube wall. Under the influence of centrifugal force, the vapor is drawn along the surface of the liquid film. The liquid at the top of the film is directed to the inner wall of the tube and then back to inner core [2]. This circulatory flow normal to main axial flow is termed as secondary flow, which exists all along the length of the coil.

An intensive literature review on condensation of refrigerant vapor inside and outside smooth and enhanced tubes has been reported by Cavallini [3]. Jung et al. [4] measured flow condensation heat transfer coefficients of R-22, R-134a, R-407C, and R-410A inside horizontal straight and micro-fin tubes of 9.52 mm outside diameter and 1 m length at saturation temperature of 40 °C with mass fluxes of 100, 200, and 300 kg m^{-2} s^{-1} and a heat flux of 7.7 to 7.9 kW m^{-2}. Heat transfer coefficients of the micro-fin tube were 2–3 times higher than those of a straight tube and the heat transfer enhancement factor decreased as the mass flux increased for all refrigerants tested. Sapali and Patil [5] measured condensation heat transfer coefficient of R-134a and R-404A in smooth and micro-fin tubes for different mass flux between 90 and 800 kg m^{-2} s^{-1} and condensing temperature ranging from 35 to 60 °C. The heat transfer coefficient increases with increasing mass flux and decreases with increasing condensing temperature. The heat transfer coefficients of R-134a are greater than that of R-404A at a similar mass flux and condensing temperature. The enhancement factors for R-134a and R-404A vary from 1.5 to 2.5 and 1.3 to 2.01 respectively. Akhavan-Behabad et al. [6] carried out an experimental investigation to find the heat transfer coefficient during condensation of R-134a vapor inside a horizontal plain tube and tubes with twisted tape with different twisted ratios of 6, 9, 12 and 15. Test runs were carried out for the mass flux of 92, 110, 128 and 147 kg m^{-2} s^{-1}. An empirical correlation was developed to

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Nomenclature

A	heat transfer area (m^2) ($=\pi d_i l$)
Bo	boiling number
C_P	specific heat at constant pressure ($J\,kg^{-1}\,K^{-1}$)
d	tube diameter (mm)
D	coil mean diameter (mm)
d_i/D	curvature ratio (–)
Fr_v	vapor Froud number (–)
f	friction factor (–)
G	mass flux ($kg\,m^{-2}\,s^{-1}$)
h	heat transfer coefficient ($kW\,m^{-2}\,K^{-1}$)
i	enthalpy ($kJ\,kg^{-1}$)
k	thermal conductivity ($W\,K^{-1}\,m^{-1}$)
l	length (m)
m	mass flow rate ($kg\,s^{-1}$)
Nu	Nusselt number (–)
p	pressure (kPa)
Pr	Prandtl number (-)
p_r	reduced pressure (–)
Q	heat transfer rate (W)
Re	Reynolds number (–)
T	temperature (°C)
x	vapor quality (–)
$pred$	predicted
ph	pre-heater
R	refrigerant
s	saturation
st	straight
tp	two-phase
ts	test-section
tt	turbulent–turbulent
v	vapor
w	wall
W	water

Greek symbols

α	void fraction (–)
χ	Martinelli parameter (–)
ρ	density ($kg\,m^{-3}$)
Δ	difference (–)
ϕ	two-phase multiplier (–)
σ	surface tension ($N\,m^{-1}$)
μ	viscosity (Pa s)

Subscripts

a	accelerational
$crit$	critical
exp	experimental
f	frictional
g	gravitational
h	homogeneous
he	helical
i	inner
in	inlet
l	liquid
lv	latent heat of vaporization
o	outer
out	outlet

predict the heat transfer coefficient. Olivier et al. [7] presented an experimental study of heat transfer, pressure drop, and flow pattern during condensation of refrigerants R-22, R-407C, and R-134a inside smooth, helical micro-fin, and herringbone tubes. The refrigerant condensed at an average saturation temperature of 40 °C with mass fluxes ranging from 400 to 800 $kg\,m^{-2}\,s^{-1}$. The results illustrated that the herringbone tube had an average heat transfer enhancement factor of 1.7 for the three refrigerants against the smooth tube. The enhancement factor was in the order of 1.4 for the herringbone tube against the helical micro-fin tube.

A few papers are also published on the condensation inside helically coiled concentric tube-in-tube heat exchanger reporting heat transfer and pressure drop characteristics of R-134a. Condensation heat transfer and pressure drop characteristics of R-134a for the mass fluxes from 100 to 400 $kg\,m^{-2}\,s^{-1}$ and cooling water Reynolds number between 1500 and 9000 inside a helicoidal tube at refrigerant saturation temperature 33 °C and the tube wall temperature at 12 and 22 °C has been experimentally investigated by Kang et al. [8]. They observed that the refrigerant-side heat transfer coefficients decreased with the increase of cooling water mass flux, although the overall heat transfer coefficient increased. When the cooling tube wall temperature increased from 12 to 22 °C, heat transfer and pressure drop decreased by nearly 30%. Yu et al. [9] presented an experimental investigation of condensation heat transfer of R-134a flowing inside a helical pipe with cooling water through the concentric annular passage in counter-flow direction with the helix axis of the test-section in vertical, inclined and horizontal directions with refrigerant mass flux in the range of 100–400 $kg\,m^{-2}\,s^{-1}$ and cooling water Reynolds number in the range from 1500 to 10000. The results revealed that the orientation of the test-section had a significant effect on heat transfer coefficients. The refrigerant side heat transfer coefficient for an inclined tube was found 6–7 times greater than the vertical position. Han et al. [10] conducted an experimental investigation into the condensation heat transfer and pressure drop characteristics of R-134a in the annular helical tube at three different saturated temperatures 35, 40, and 46 °C with the mass flux of R-134a ranging from 100 to 420 $kg\,m^{-2}\,s^{-1}$. The results showed that the refrigerant-side condensation heat transfer coefficients and pressure drops of R-134a increased with the refrigerant mass flux. The condensation heat transfer coefficients of R-134a in the annular tube decreased with increase in saturated temperature. Wongwises and Polsongkram [11] have experimentally investigated two-phase condensation heat transfer and pressure drop of R-134a in a helically coiled concentric tube-in-tube heat exchanger. The test runs were carried out at saturation temperatures 40 and 50 °C. The mass flux was between 400 and 800 $kg\,m^{-2}\,s^{-1}$ and the heat fluxes were between 5 and 10 $kW\,m^{-2}$. It was found that the percentage increase of the average heat transfer coefficient and the pressure drop of the helically coiled concentric tube-in-tube heat exchanger, compared with that of the straight tube-in-tube heat exchanger, were in the range of 33–53% and 29–46%, respectively. The correlations to predict the condensation heat transfer coefficient and pressure drop were also proposed. Lin and Ebadian [12] conducted experimental investigations to determine the condensation heat transfer and pressure drop of refrigerant R-134a in annular helicoidal pipe at three inclination angles viz. horizontal (0°), vertical (90°) and inclined (45°) with the mass flux of R-134a ranging from 60 to 200 $kg\,m^{-2}\,s^{-1}$, and Reynolds number of cooling water from 3600 to 22,000; condensation temperatures of R-134a at 30 and 35 °C, and cooling water at 16, 20 and 24 °C. The Nusselt number was higher at lower refrigerant saturation temperature, and would increase with the increase of mass flow rates of both refrigerant and cooling water. When the orientation of the helicoidal pipe changed from 0 to 90°, the Nusselt number of the refrigerant-side for 0 to 45° orientation accounted for more than two times of that