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Experimental investigation on Al_2O_3-R123 nanorefrigerant heat transfer performances in evaporator based on organic Rankine cycle

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ABSTRACT

This paper presents an experimental investigation of variation tendency of heat transfer coefficient, log-mean temperature difference and differential pressure of pure R123 and 20 nm Al_2O_3-R123 nanorefrigerants with four various volume concentrations, 0.03%, 0.13%, 0.18%, 0.23% flowing inside the evaporator of organic Rankine cycle system under the conditions of various heat source temperatures and flow rates. Heat source temperatures are in the range of 50–90 °C at an interval of 10 °C, and heat source flow rates are 0.7 m^3/h, 1.3 m^3/h and 1.8 m^3/h. Results show an increment of heat transfer coefficient along flowing direction for both pure R123 and four Al_2O_3-R123 nanorefrigerants with the increment of heat source temperature and flow rate, and that of four nanorefrigerants are higher than that of pure R123. There is no optimum value of heat transfer coefficient when operation condition is changed for the loading carrying capacity increasing with the increasing intensity of operation condition. Meanwhile, suspending nanoparticles and increasing heat source temperature can change the variation tendency of heat transfer coefficient along flowing direction except pure R123 and 0.18 vol% nanorefrigerant. In addition, 0.13 vol% as a whole is the optimum volume concentration for both log-mean temperature difference and differential pressure.

1. Introduction

In the early studies, researchers adopted the suspended particles of micrometer or even millimeter order to enhance heat transfer which result in clogging, pipe eroding and aggregating in operation systems [1,2]. To solve those problems, Choi [3] firstly used nanofluid, which consists of nanoparticles and base fluid, as an enhancing technology of heat transfer. Subsequently, numerous researches have been conducted to study nanofluid thermophysical properties.

Experimental studies were conducted and showed that the thermal conductivity enhancement of tested nanofluids can be found with the increasing particle loading, increasing particle aspect ratio and decreasing base fluid thermal conductivity [4,5]. Meanwhile, nanoparticles (with CuO, Al_2O_3, TiO_2, SiO_2) can enhance the heat transfer coefficient of fluids [6–11], and have a wide application, such as jet impingement cooling [12].

Nanorefrigerant is one kind of promising nanofluid, and its base fluid is refrigerant [13–15]. In addition to focusing on its particles aggregation behavior [16] and dispersion stability [17], researches also have been conducted on its thermophysical properties and applications. Bi [18] used TiO_2-R600a in domestic refrigerator and found that adopting TiO_2-R600a can save energy consumption and accelerate freezing velocity of nanorefrigerant system. I.M. Mahbubul adopted Al_2O_3/R141b in horizontal smooth circular tubes to investigate the thermophysical properties and heat transfer performance. They found that volume fractions have positive effects on heat transfer coefficients, viscosity and pressure drop [19,20], and thermal conductivity of nanorefrigerant increases with the increasing volume concentration and temperature of mixture [21]. Meanwhile, the viscosity and density increase accordingly with the increasing volume concentration and decreases with the increasing temperature [22]. Kedzierski [23,24] used polyolester/Al_2O_3 as nanolubricant boiling on a reentrant cavity surface and found that nanolubricants can promote R134a boiling when the nanoparticles remain dispersed well and are at sufficiently large concentration. Henderson [25] adopted R134a as based refrigerant in a horizontal tube, and found that the heat transfer coefficient decreases with the direct dispersion of SiO_2 nanoparticles in R134a.

Although prior researchers have studied heat transfer performances of nanorefrigerants, it can be found out that these

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Nomenclature

A	surface area (m^2)
B	constant
C	constant
c_p	specific heat ($J \cdot kg^{-1} \cdot K^{-1}$)
D	diameter over fins of inner tube (mm)
d	hydraulic diameter (mm)
d_1	outside diameter of plain section (mm)
d_2	inside diameter of finned section (mm)
d_3	root diameter (mm)
d_4	diameter over fins (mm)
J	polynomial order
K	heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
L	axial length of evaporator (mm)
p	pressure (MPa)
q''	heat flux ($W \cdot m^{-2}$)
q	heat transfer rate (W)
s_1	wall thickness of plain end and plain lands (mm)
s_2	root wall thickness (mm)
T	temperature (K)
u	velocity of working fluid in evaporator inlet (m/s)
$\dot{V}$	volume flow rate ($m^3 \cdot h^{-1}$)
x	non-dimensional parameter of length
X	independent variable
Y	dependent variable

Greek letters

μ	viscosity (Pa·s)
φ	volume concentration of nanoparticles
ρ	density ($kg \cdot m^{-3}$)
Δ	differential value

Subscripts

acc	accelerating
ave	average
below	under exterior surface of inner tube
bf	base fluid
f	frictional
g	gravitational
hw	hot water/heat source
i	temperature measure point (i = 0.5 m, 0.75 m, 1.0 m, 1.25 m, 1.5 m, 1.75 m)
in	inlet
left	left exterior surface of inner tube
local	local position
nf	nanofluid
np	nanoparticle
out	outlet
right	right exterior surface of inner tube
total	total value
up	upper exterior surface of inner tube
wall	exterior surface of inner tube
wf	working fluid

conclusions were drawn mostly on the perspective of average value and grant total of heat transfer properties. There is no research done to investigate the variation tendency of heat transfer performances along flowing direction, which is an important research field of nanorefrigerant heat transfer performance. In this paper, the variation tendencies of local heat transfer coefficient along the flowing direction in evaporator of five working fluids will be studied, including four Al_2O_3-R123 nanorefrigerants with 0.03%, 0.13%, 0.18%, 0.23% volume concentration nanoparticles and pure R123, under the conditions of different temperatures from 50 °C to 90 °C at an interval of 10 °C and volume flow rates of heat source including 0.7 m^3/h, 1.3 m^3/h and 1.8 m^3/h. Meanwhile, the experimental heat transfer coefficient, differential pressure of inlet and outlet working fluid and log-mean temperature difference in evaporator also will be presented.

2. Experimental apparatus and procedures

Fig. 1 shows the basic schematic diagram of experimental apparatus. This system is composed of three circuits: working fluid circuit, heat source circuit and cooling water circuit. Working fluid circuit can be divided into four processes: working fluid is supplied from the storage tank by a pump, and is heated to two-phase state from single-phase state by heat source in evaporator. Then the two-phase flow is condensed into liquid, and flows into storage tank. The whole cycle is completed.

In this system, hydraulic diaphragm metering pump is adopted. Its flow range, highest working pressure and measuring accuracy are 0–0.4 m^3/h, 0.8 MPa and 0.2%, respectively. The evaporator, covered by an outside protective case which is made of stainless steel, is a 2000 mm long double-tube counter flow heat exchanger. To observe the state of working fluid, there exist eight visualization windows on the protective case. Working fluid flows in the outer tube, and heat source (hot water) flows in the inner tube. For outer tube, Silica is adopted as the material, and its inner and outer diameters are 32 mm and 45 mm, respectively. Thermocouples distribution of each measuring point in evaporator is shown in Fig. 2. To draw the thermocouple wires out feasibly, the sectorial pipe-coupling method is adopted in outer tube, and diameters of connections and silica tube are the same. The code No. of inner tube is Turbo-EHP which is made by Wieland Thermal Solutions

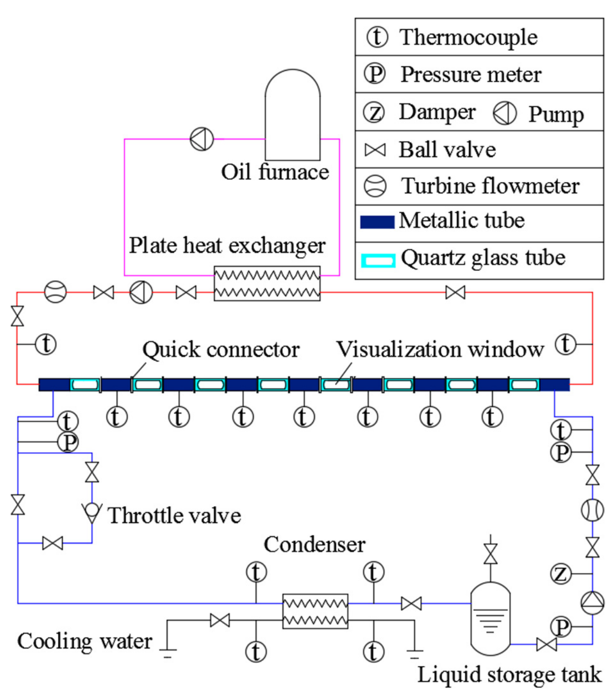

Fig. 1. Basic schematic diagram of experimental apparatus.