Available online at www.sciencedirect.com
SciVerse ScienceDirect
journal homepage: www.elsevier.com/locate/ijrefrig

Thermal and hydraulic performance of sinusoidal corrugated plate heat exchanger for low temperature lift heat pump

Hoseong Lee [a], *Yunho Hwang* [a,*], *Reinhard Radermacher* [a], *Ho-Hwan Chun* [b]

[a] *Center for Environmental Energy Engineering, University of Maryland, 4164 Glenn L. Martin Hall Bldg., College Park, MD 20742, USA*
[b] *Advanced Ship Engineering Research Center (ASERC), Dept. of Naval Architecture & Ocean Engineering, Pusan National University, 30 Changjeon-dong, Kumjeng-ku, Busan 609-735, Republic of Korea*

ARTICLE INFO

Article history:
Received 26 March 2012
Received in revised form
10 October 2012
Accepted 6 November 2012
Available online 10 November 2012

Keywords:
Low temperature lift
Heat pump
Plate heat exchanger
Corrugated

ABSTRACT

Thermal and hydraulic performance of a sinusoidal corrugated plate heat exchanger (PHX) was investigated for the application of a low temperature lift heat pump (LTLHP), which requires unique operating conditions. The water-side heat transfer coefficient and pressure drop of the PHX were obtained through the experimental test. The refrigerant-side heat transfer performance was investigated by varying several parameters. The PHX performance was poor due to low refrigerant mass flux. The PHX needs better balance in two fluids for the LTLHP application. From the current study, it is concluded that the conventional PHX applied for the LTLHP application is limited by two main factors: a large pressure drop on the water-side due to corrugated shape, and a low heat transfer performance due to the low refrigerant-side heat transfer performance. In order to address these drawbacks, heat exchanger designs must be improved by optimizing its geometry and flow area asymmetrically for each fluid.

Performance thermique et hydraulique de l'un échangeur à plaques sinusoïdal d'une pompe à chaleur à faible élévation de température

Mots clés : low temperature lift ; pompe à chaleur ; échangeur à plaques ; ondulées

1. Introduction

As energy cost is projected to increase due to population and income growth, enhancing energy efficiency of the energy conversion systems becomes more important than ever. One of such approaches is reducing the temperature lift of the vapor compression cycle, which is used for air conditioning and heat pump applications (Lee et al., 2012). A schematic diagram of the typical heat pump vapor compression cycle (VCC) is shown in Fig. 1. A working fluid (refrigerant) absorbs heat from the evaporator, and discharges the heat to the heat sink through the condenser. Fig. 2 shows the heat pump cycle in a temperature–entropy (T–s) diagram. As shown in the figure, the heat source temperature is higher than evaporating

* *Corresponding author*. Tel.: +1 301 405 5286.
E-mail address: yhhwang@umd.edu (Y. Hwang).
0140-7007/$ – see front matter
http://dx.doi.org/10.1016/j.ijrefrig.2012.11.008

Nomenclature

Symbols

A	heat transfer area, m^2
b	flow channel gap, mm
Bo	boiling number, $q\ G^{-1}\ h_{fg}^{-1}$, dimensionless
c1	constant number
c2	constant number
C_p	specific heat, $J\ kg^{-1}\ K^{-1}$
d_e	equivalent diameter, 2b, m
d_h	hydraulic diameter, $2b\ \Phi^{-1}$, m
f	friction factor, dimensionless
G	mass flux, $kg\ m^{-2}\ s^{-1}$
h	heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
k	thermal conductivity, $W\ K^{-1}\ m^{-1}$
L	length, m
L_p	plate length, m
$\dot{m}$	mass flow rate, $kg\ s^{-1}$
Nu	Nusselt number, dimensionless
P	corrugation pitch, m
p	pressure, Pa
Pr	Prandtl number, dimensionless
$\dot{Q}$	total heat transfer, W
Re	Reynolds number, $G\ d\ \mu^{-1}$, dimensionless
Re_e	Reynolds number with equivalent diameter, $G\ d_e\ \mu^{-1}$, dimensionless
Re_h	Reynolds number with hydraulic diameter, $G\ dh\ \mu^{-1}$, dimensionless
Re_l	liquid Reynolds number, $G\ (1-x)\ d\ \mu^{-1}$, dimensionless
ΔT	temperature difference, K
T	temperature, K
t	thickness of wall, mm
U	overall heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
U_m	fluid mean velocity, $m\ s^{-1}$
w	plate width, m
X_{tt}	Martinelli parameter, $((1-x)\cdot x^{-1})^{0.9}\cdot(\rho_g\cdot\rho_f^{-1})^{0.5}\cdot(\mu_f\cdot\mu_g^{-1})^{0.1}$, dimensionless
β	corrugation angle, degree
ε	acceptable error, dimensionless
λ	corrugation pitch, mm
μ	dynamic viscosity, Pa s
ρ	density, $kg\ m^{-3}$
Φ	enlargement factor, dimensionless

Acronyms

DP L^{-1}	pressure drop per unit length, $Pa\ m^{-1}$
HTC	heat transfer coefficient, $W\ K^{-1}\ m^{-1}$
MFR	mass flow rate, $kg\ s^{-1}$
LMTD	logarithmic mean temperature difference, K
LTLHP	low temperature lift heat pump
PHX	plate heat exchanger
VCC	vapor compression cycle

Subscript

core	distance between inlet and outlet ports of PHX
e	equivalent
f1	fluid 1
f2	fluid 2
h	hydraulic
in	inlet
l	laminar
out	outlet
r	refrigerant
s	plate surface
t	turbulent
tp	two-phase
w	water

temperature, and the heat sink temperature is lower than the condensing temperature of the VCC. Work input to the VCC is mainly determined by two saturation temperatures: evaporating and condensing temperatures. As the temperature difference (ΔT) between two saturation temperatures decreases, or, as the cycle is operated at the low temperature lift, the system work also decreases. Therefore, reducing the temperature lift can increase the system efficiency. Heat pump system utilizing a small temperature difference between a heat source and a heat sink is referred as a low temperature lift heat pump (LTLHP) hereafter. The system

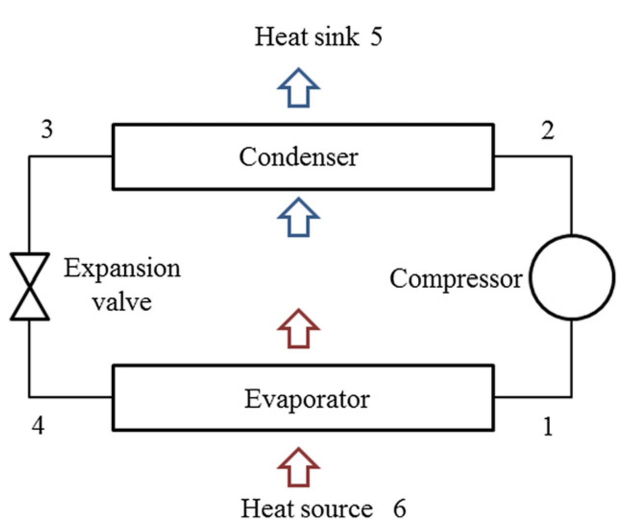

Fig. 1 – Typical heat pump VCC.

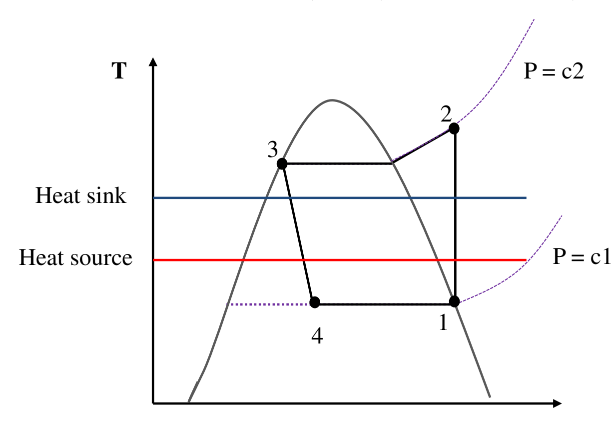

Fig. 2 – T-s diagram of heat pump VCC.