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Heat Transfer in the Evaporators of a Double-Evaporator Refrigerating System

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This paper presents the heat transfer characteristics of the evaporators in a double-evaporator refrigerating system with an environment-friendly refrigerant, propane (R-290). Based on the Buckingham Pi theorem, dimensionless correlations are developed to predict the heat transfer coefficients of the refrigerant in high- and low-temperature evaporators (h_H and h_L) and the ratio of the cooling capacity of the high-temperature evaporator to the total capacity (α) in the system. The results show that h_H is affected mainly by the condensing pressure, the length of the low-temperature capillary tube, and the logarithmic-mean temperature difference of the high-temperature evaporator, while h_H is affected mainly by the length of the high-temperature capillary tube and the logarithmic-mean temperature difference of the low-temperature evaporator. Note, though, that the condensing pressure and the logarithmic-mean temperature difference of the high-temperature evaporator are the main factors affecting α. Some of the correlations result in good predictions, though the required numbers of variables of these correlations are much more than those presented in this work.

A comparison of the experimental measurements of h_H and h_H to the values calculated by the correlations from the literature is made. Some of the correlations result in good predictions, though the required variables of these correlations are much more than those presented in this work.

INTRODUCTION

The high ozone-depleting potential (ODP) and global warming potential (GWP) has led to the restriction in the use of chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC), such as R-12 and R-22 [1]. Propane (R-290), with zero ODP and extremely low GWP characteristics, is very attractive in this respect and has been recommended as an alternative to R-22 due to the similarities in their refrigerating properties.

The two-phase heat transfer coefficient of the refrigerant affects the performance of the evaporator in a refrigerating system. For the two-phase flow in the evaporator, the heat transfer involves both nucleate boiling and forced-convection. Several authors [2–5] have proposed correlations for the two-phase heat transfer coefficient in the evaporator. These correlations have been developed through an extensive database of fluids, including water, R11, R12, R13B1, R22, R113, R114, R134a, R152, R22/R124/R152a, benzene, n-pentane, n-heptane, cyclohexane, methanol, ethanol, n-butanol, hydrogen, helium, neon, nitrogen, ammonia, and ethylene glycol. However, propane (R-290) is not included in this list and warrants further study.

From a thermodynamic viewpoint, a refrigerator with two evaporators in series and one capillary tube performs better than that with only one evaporator. A two-evaporator refrigerator with the zeotropic refrigerant R22/R11 proposed by Lorenz-Meutzner [6] showed a power savings of up to 20% compared to a conventional refrigerator with R-12. With two capillary tubes and two evaporators connected in series, different evaporating temperatures can be obtained by a refrigerating system with pure azeotropic or zeotropic refrigerants [7]. For such a system, the distribution of the cooling load between the evaporators may be an important characteristic.

The design of the distribution of the cooling load between the evaporators in a refrigerating system with two evaporators connected in series is an important study. Based on the present experimental results and the Buckingham Pi theorem, dimensionless correlations for the heat transfer coefficients of refrigerant in the high- and low-temperature evaporators and the ratio of the cooling capacity of the high-temperature evaporator to the total capacity are developed. The experimental measurements of the heat transfer coefficients of the refrigerant are then compared

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Table 1 Test conditions

P_c (kPa)	f (Hz)	d_H (mm)	d_L (mm)	L_e (m)	$D_{i,i}$ (mm)	$D_{o,i}$ (mm)	$D_{i,o}$ (mm)	$D_{o,o}$ (mm)	L_H (m)	L_L (m)
1764	40	1.0	1.4	2	8.7	9.525	14.85	15.875	1.6	0.3
1666	50								1.9	0.6
1568	60								2.2	0.9
1470	70								2.5	1.2
1372	80								2.8	

with the results calculated by the correlations in the literature [2–5].

EXPERIMENTAL METHOD

Facility

Figure 1 shows the experimental facility of a series-connected two-evaporator refrigerating system with R-290. The test apparatus is composed of a refrigerant loop and two heat-exchange fluid loops. The states of the working fluids are monitored with T-type thermocouples and pressure gauges.

The reciprocating compressor in the refrigerant loop is controlled by the frequency converter. The output of the converter for stable operation can be adjusted to 40–80 Hz. The rotating speed of the fan controlled through a voltage transformer affects the condensing pressure of the system. The dimensions of these components are listed in Table 1. The two evaporators are double-tube with the same dimensions. $D_{i,i}$ and $D_{o,i}$ represent

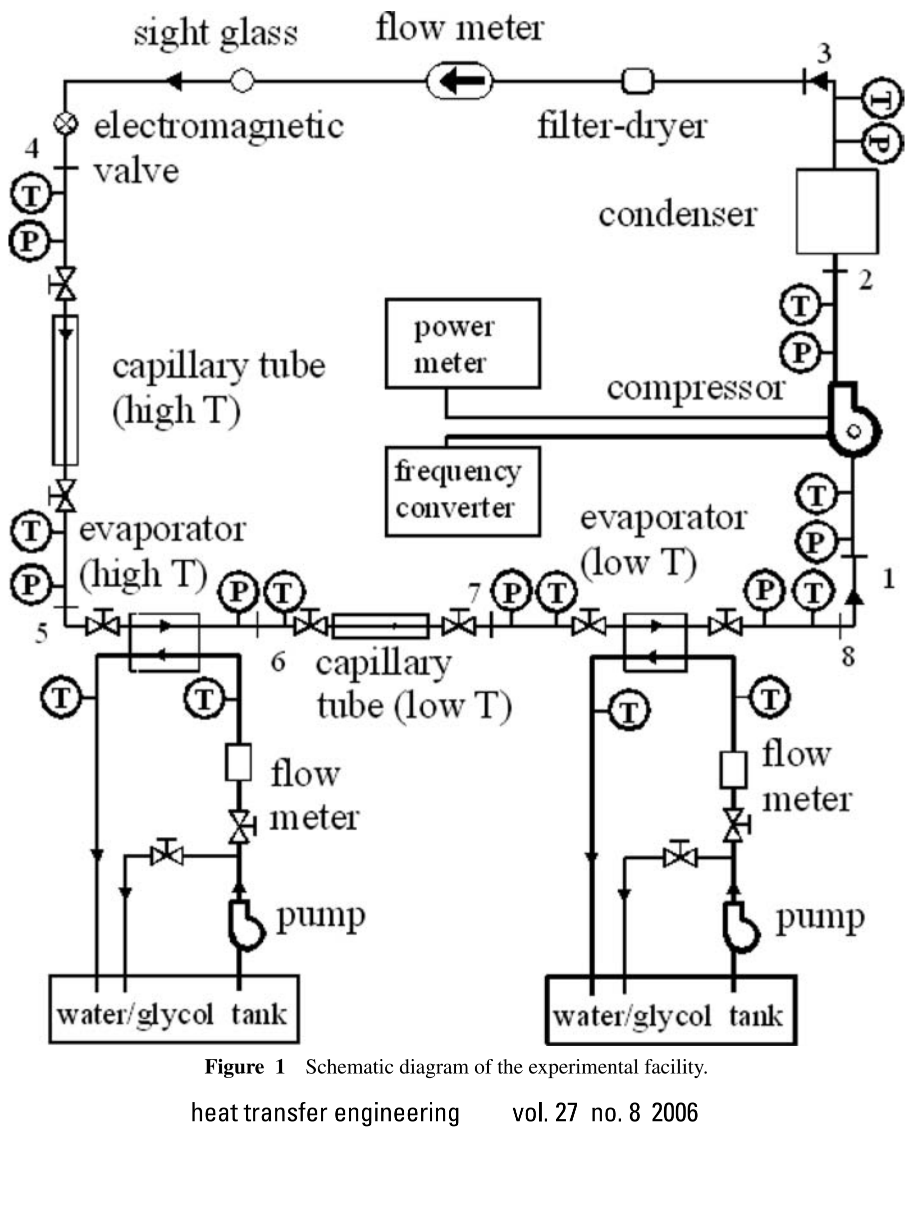

Figure 1 Schematic diagram of the experimental facility.

the inside and outside diameter of the inner tube, respectively, while $D_{i,o}$ and $D_{o,o}$ represent the inside and outside diameter of the outer tube, respectively. The refrigerant flows in one direction through the inner tube, while the heating medium flows in the opposite direction through the annular space between the inner and outer tubes. Note that the evaporators and capillary tubes are all made of copper and heat-insulated.

Both heat-exchange loops are composed of a refrigerator, a pump, a flow meter, a thermometer controller, and an electrically heated unit. The heating media of both evaporators are water/glycol (50/50 wt.%) mixture. The temperatures of water/glycol entering the high- and low-temperature evaporators are set at around 25°C and −9.5°C, respectively. The condensing pressure, the lengths of the capillary tubes for the high- and low-temperature evaporator, and the compressor frequency are the variables of the experiment. Table 1 lists the details of the test conditions.

Data Reduction and Dimensionless Correlation

The Heat Transfer Coefficient of the Refrigerant

To calculate the heat transfer coefficient of the refrigerant in both evaporators, the heat transferred to the refrigerant should be known. For a given evaporator, the cooling capacity (Q) may be expressed as

$$Q = (U \cdot A) \cdot (LMTD) \tag{1}$$

where

$$(U \cdot A) = \left(\frac{1}{h_r \cdot A_i} + R + \frac{1}{h_{\mathrm{hm}} \cdot A_o}\right)^{-1} \tag{2}$$

$$(LMTD) = \frac{(T_{\mathrm{hm,out}} - T_{r,\mathrm{in}}) - (T_{\mathrm{hm,in}} - T_{r,\mathrm{out}})}{\ln\left(\frac{(T_{\mathrm{hm,out}} - T_{r,\mathrm{in}})}{(T_{\mathrm{hm,in}} - T_{r,\mathrm{out}})}\right)} \tag{3}$$

Assuming that the variations in $T_{\mathrm{hm,in}}$ are negligible and keeping m_{hm} constant, the heat transfer coefficients of the heating media (h_{hm}) are more or less fixed. The conduction resistances (R) are more or less constant, while the inside and outside heat transfer areas, A_i and A_o, are fixed. With all temperatures in Eq. (3) and the cooling capacity ($Q = \dot{m}_{\mathrm{hm}} \cdot C_{p_{\mathrm{hm}}} \cdot (T_{\mathrm{hm,in}} - T_{\mathrm{hm,out}})$) measured, the experimental values of the heat transfer coefficient of refrigerant (h_r) can then be obtained through Eqs. (1–3).

Dimensionless Correlations

The quality of the refrigerant in the high- and low-temperature evaporators is different, and the analyses for the heat transfer coefficient of refrigerant in the evaporator (h_r) are thus divided into two categories: the heat transfer coefficients of refrigerant in the high- and low-temperature evaporators, h_H and h_L. The analytic process used to develop the dimensionless correlations for predicting h_r in the system is based on the Buckingham Pi theorem. The first step is to determine the variables that may influence h_r: both the diameters and lengths of the high-temperature (d_H and L_H) and low-temperature (d_L and L_L) capillary tube, the inlet conditions of the refrigerant (P_c and ΔT_{sc}), the frequency of compressor (f), the saturated properties of R-290 (liquid density ρ_f, liquid viscosity μ_f, and liquid-specific heat C_{pf}), and the logarithmic-mean temperature difference of the specific evaporator ($LMTD_H$, $LMTD_L$). Note that the properties of refrigerant are obtained by using REFPROP [8]. L_H and L_L are non-dimensionalized by d_H and d_L, respectively. However, in order to non-dimensionalize h_r, P_c, ΔT_{sc}, and f, a new repeating variable $\bar{d}$, based on the definition of hydraulic diameter, is defined as $(d_H^2 + d_L^2)(d_H + d_L)^{-1}$. h_r can then be expressed as

$$h_r = f_1(P_c, L_H, L_L, d_H, d_L, \bar{d}, \Delta T_{\mathrm{sc}}, \rho_f, C_{p_f}, \mu_f, f, LMTD) \tag{4}$$

Similarly, the ratio of the cooling capacity of the high-temperature evaporator to the total capacity (α) can be expressed as

$$\alpha = f_2(P_c, L_H, L_L, d_H, d_L, \bar{d}, \Delta T_{\mathrm{sc}}, \rho_f, C_{p_f}, \mu_f, f, LMTD) \tag{5}$$

where d_H, d_L, $\bar{d}$, ρ_f, μ_f, and C_{pf} are the repeating variables. The dimensionless correlations for h_H, h_L, or α can now be expressed as

$$\Pi_{8,9,\mathrm{or}10} = A \cdot \Pi_1^B \cdot \Pi_2^C \cdot \Pi_3^D \cdot \Pi_4^E \cdot \Pi_5^F \cdot \Pi_6^G \cdot \Pi_7^H \tag{6}$$

where Π_1, Π_2, Π_3, Π_4, Π_5, Π_6, Π_7, Π_8, Π_9, and Π_{10} in Eq. (6) represent P_c, L_H, L_L, ΔT_{sc}, f, $LMTD_H$, $LMTD_L$, h_H, h_L, and α, respectively, as shown in Table 2. By substituting the experimental values into the statistical software STATISTICA, the values of constant A and exponents of the Π parameters are obtained.

Table 2 Dimensionless parameters Π group

Pi Groups	Definition	Effect
Π_1	$\frac{\bar{d}^2 \cdot \rho_f \cdot P_c}{\mu_f^2}$	Condensing pressure
Π_2	$(\frac{L_H}{d_H})$	Geometry
Π_3	$(\frac{L_L}{d_L})$	Geometry
Π_4	$(\frac{\bar{d}^2 \cdot \rho_f^2 \cdot C_{p_f} \cdot \Delta T_{\mathrm{sc}}}{\mu_f^2})$	Subcooling
Π_5	$\frac{\bar{d}^2 \cdot f \cdot \rho_f}{\mu_f}$	Compressor frequency
Π_6	$(\frac{\bar{d}^2 \cdot \rho_f^2 \cdot C_{p_f} \cdot LMTD_H}{\mu_f^2})$	$LMTD_H$
Π_7	$(\frac{\bar{d}^2 \cdot \rho_f^2 \cdot C_{p_f} \cdot LMTD_L}{\mu_f^2})$	$LMTD_L$
Π_8	$\frac{h_H \cdot \bar{d}}{C_{p_f} \cdot \mu_f}$	h_H
Π_9	$\frac{h_L \cdot \bar{d}}{C_{p_f} \cdot \mu_f}$	h_L
Π_{10}	$\frac{Q_H}{Q_H + Q_L}$	Ratio of Q_H

RESULTS

The Dimensionless Correlation for h_H

Note that $LMTD_L$ is not one of the main factors affecting h_H. The dimensionless correlation for h_H is thus

$$\Pi_8 = 10^{-4.52} \cdot \Pi_1^{-0.343} \cdot \Pi_2^{0.097} \cdot \Pi_3^{-0.209} \cdot \Pi_4^{-0.045} \cdot \Pi_5^{-0.021} \cdot \Pi_6^{0.888} \quad (7)$$

From the exponents of the Π parameters in Eq. (7), it is clear that h_H is affected mainly by P_c, L_L, and $LMTD_H$. The effect of P_c on h_H seems more significant than that of L_L. Increasing P_c will increase the mass flow rate of refrigerant ($\dot{m}_r$) but decrease $LMTD_H$. For the two-phase flow in the evaporator, the heat transfer involves both nucleate boiling and forced convection. The nucleate boiling heat transfer increases with $LMTD$ [9], while forced-convection heat transfer increases with Reynolds number, which is in proportion to $\dot{m}_r$. It seems that the effect of $LMTD_H$ induced by P_c on h_H dominates that of $\dot{m}_r$. In addition, both $\dot{m}_r$ and $LMTD_H$ decreases with L_L [7], which induce h_H to decrease with L_L. Figure 2 shows that the range of the maximum error (β) for h_H using Eq. (7) is between –16% and +16%. Note that β represents the maximum deviation of the experimental data from the predicted values.

The Dimensionless Correlation for h_L

Again, $LMTD_H$ is not a main factor affecting h_L. The dimensionless correlation for h_L is

$$\Pi_9 = 10^{-6.095} \cdot \Pi_1^{-0.038} \cdot \Pi_2^{-0.185} \cdot \Pi_3^{-0.08} \cdot \Pi_4^{0.023} \cdot \Pi_5^{0.00002} \cdot \Pi_7^{0.688} \quad (8)$$

Equation (8) shows that L_H and $LMTD_L$ are the main factors affecting h_L. h_L decreases with L_H but increases with $LMTD_L$; $\dot{m}_r$ decreases but $LMTD_L$ increases with L_H [7]. It seems that h_L decreases with L_H, as the effect of $\dot{m}_r$ on h_L dominates that of $LMTD_L$ induced by L_H. Figure 3 shows that β for h_L using Eq. (8) is between -12% and $+16\%$.

The Dimensionless Correlation for α

The ratio of the cooling capacity of the high-temperature evaporator to the total capacity (α) is an important characteristic of the system because the relative cooling capacities of the two evaporators in the system may vary in various applications. The analysis shows that α can be expressed as

$$\Pi_{10} = 10^{-1.416} \cdot \Pi_1^{-0.253} \cdot \Pi_2^{-0.0815} \cdot \Pi_3^{-0.02} \cdot \Pi_4^{-0.052} \cdot \Pi_5^{-0.027} \cdot \Pi_6^{1.563} \cdot \Pi_7^{-0.999} \quad (9)$$

From the relative values of the exponents of Π parameters in Eq. (9), α is mainly affected by P_c, $LMTD_H$, and $LMTD_L$. α decreases with P_c and $LMTD_L$ but increases with $LMTD_H$. However, the effect of $LMTD_L$ on α is more significant than that of P_c. Figure 4 shows that β for α using Eq. (9) is between -10% and $+10\%$.

Comparison of Experimental Results with Existing Correlations

For the two-phase heat transfer coefficients of a refrigerant in the two evaporators, h_H and h_L, some correlations may be

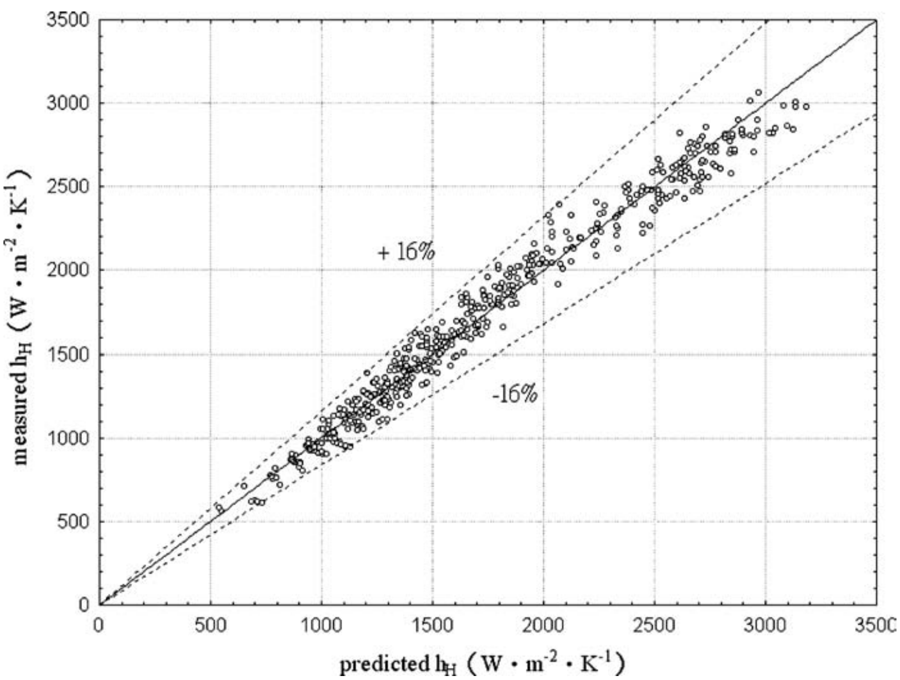

Figure 2 Measurement of h_H versus predicted h_H using Eq. (7).

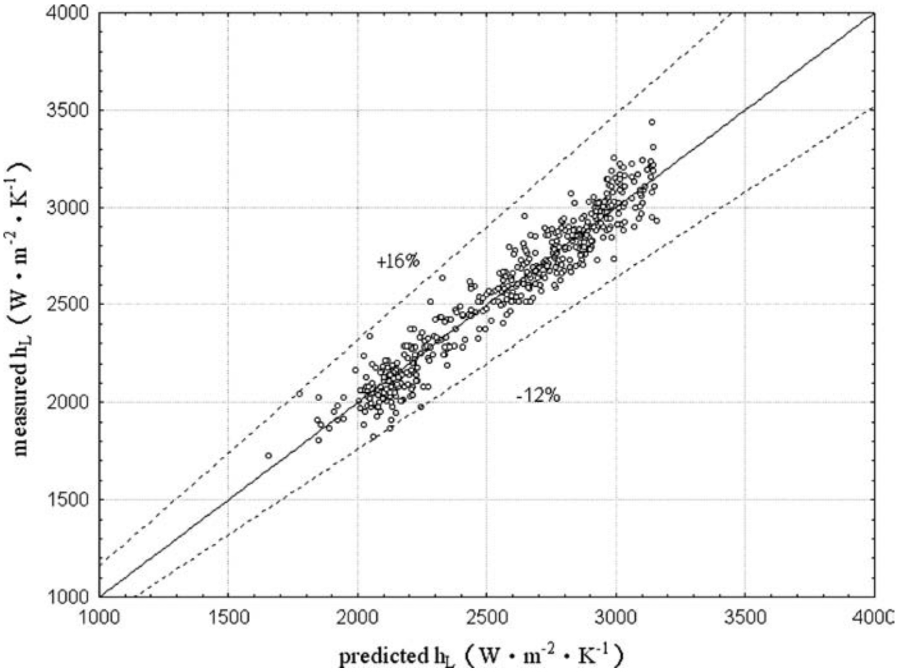

Figure 3 Measurement of h_L versus predicted h_L using Eq. (8).

found from various sources (e.g., [2–5]). These correlations for the two-phase heat transfer coefficient cover both the nucleate boiling and forced convection regions. A comparison is thus made for these correlations with the test values measured in this work.

Correlation from Gungor-Winterton [2]

A large database for water, R11, R12, R22, R113, R114, and ethylene glycol was used by Gungor-Winterton [2]. The empirical equation obtained for the two-phase heat transfer coefficient

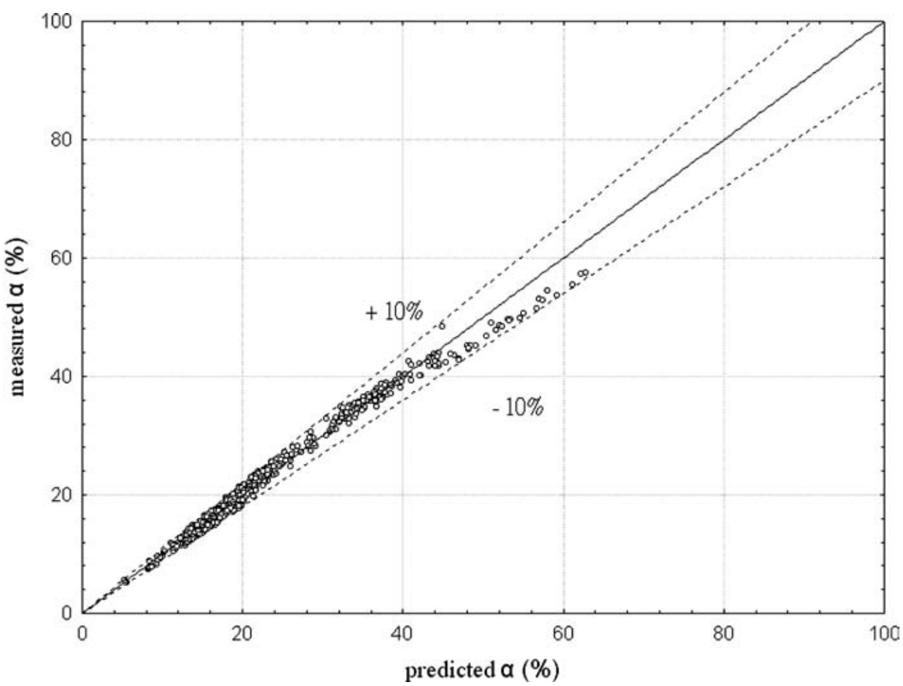

Figure 4 Measurement α versus predicted α using Eq. (9).

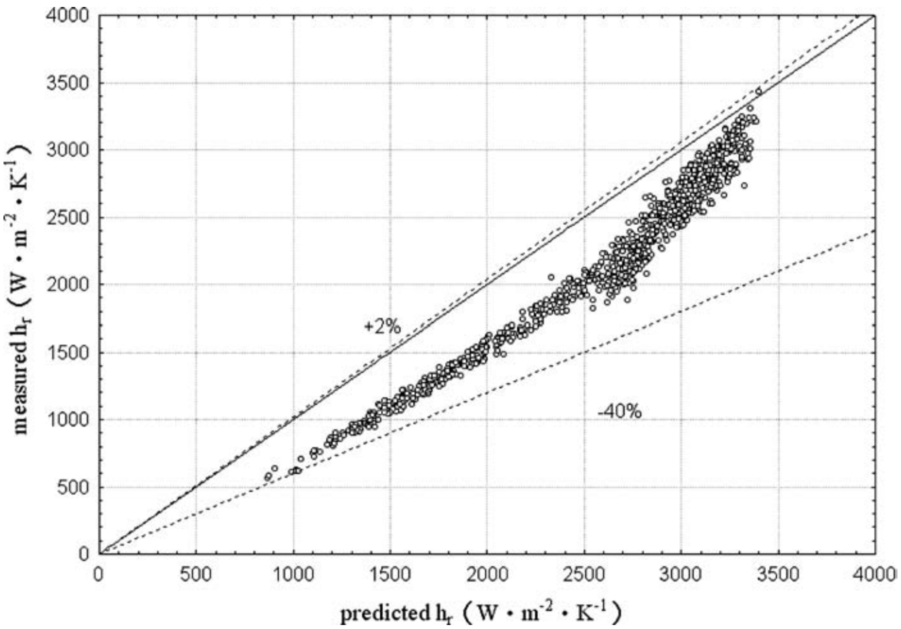

Figure 5 Measurements versus predictions in h_r with correlation [2].

in evaporator is

$$h_r = E \cdot h_f + S \cdot h_{\text{pool}} \tag{10}$$

The expressions for h_f, h_{pool}, E, and S in Eq. (10) are

$$h_f = 0.023 \cdot \text{Re}_f^{\ 0.8} \cdot \text{Pr}_f^{0.4} \cdot k_f / D_{i,i} \tag{11}$$

$$h_{\text{pool}} = 55 \cdot \text{Pr}^{0.12} \cdot (-\log_{10} \text{Pr})^{-0.55} \cdot M^{-0.5} \cdot q^{0.67} \tag{12}$$

$$E = 1 + 24000 \cdot Bo^{1.16} + 1.37(1/X_{tt})^{0.86} \tag{13}$$

$$S = \frac{1}{1 + 1.15 \cdot 10^{-6} E^2 \cdot \text{Re}_f^{1.17}} \tag{14}$$

where Bo and X_{tt} in Eq. (13) are

$$Bo = \frac{q}{\lambda \cdot G} \tag{15}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_f}\right)^{0.5} \left(\frac{\mu_f}{\mu_v}\right)^{0.1} \tag{16}$$

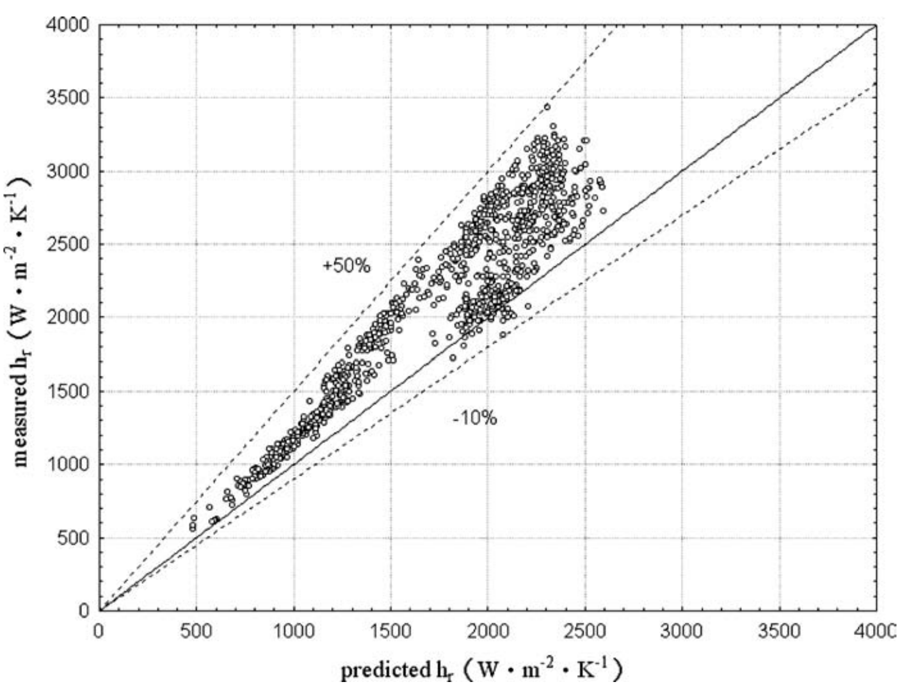

Figure 6 Measurements versus predictions in h_r with correlation [3].

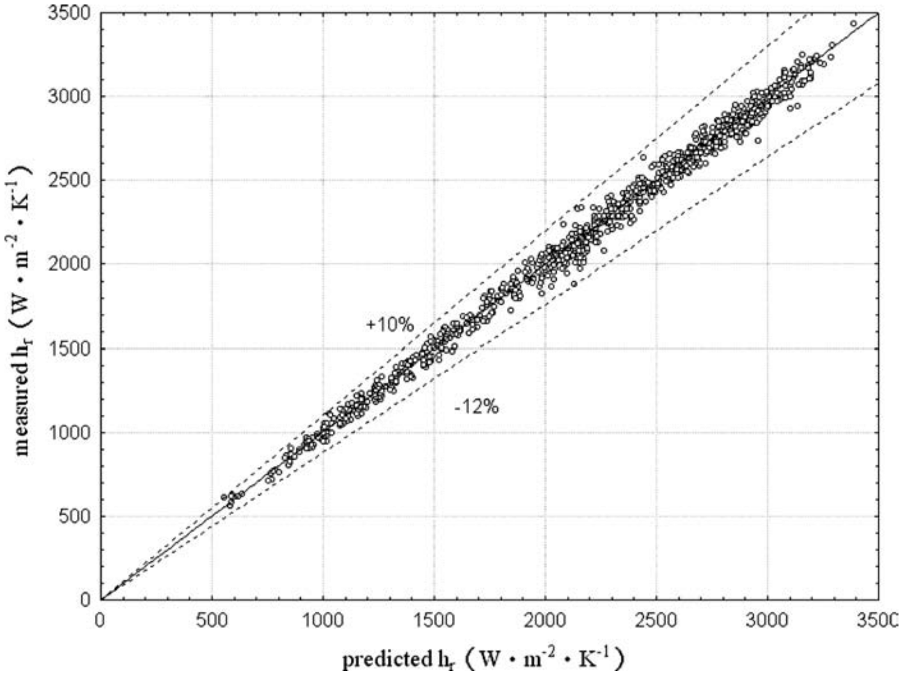

Figure 7 Measurements versus predictions in h_r with correlation [4].

Figure 5 shows that, compared with the experimental results in the present work, both h_H and h_L calculated by the correlation from [2] are over-predicted by about 0–40%.

Correlation from Kandlikar [3]

A large database for water, R11, R12, R13B1, R22, R113, R114, R-152, neon, and nitrogen was also used by Kandlikar [3]. The correlation for the two-phase heat transfer coefficient in evaporator is

$$h_r = h_f \cdot (C_1 \cdot Co^{C_2} \cdot (25 \cdot Fr_{lo})^{C_5} + C_3 \cdot Bo^{C_4} \cdot F_{fl}) \quad (17)$$

where Co is a convection number expressed as

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \left(\frac{\rho_v}{\rho_f}\right)^{0.5} \quad (18)$$

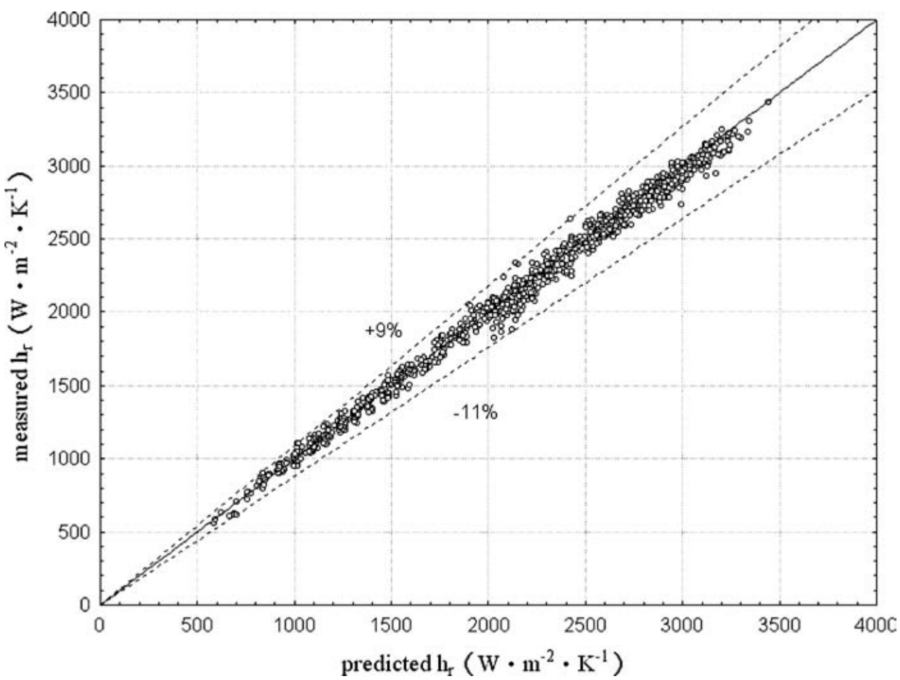

Figure 8 Measurements versus predictions in h_r with correlation [5].

The constants C_1 through C_5 and the fluid-dependent parameter F_{fl} are given in the literature [3]. Figure 6 shows that both h_H and h_L calculated by the correlation from [3] are underpredicted by about 0–35%.

Correlation from Steiner-Taborek [4]

Again, a large database for water, R11, R12, R13B1, R22, R113, benzene, n-pentane, n-heptane, cyclohexane, methanol, ethanol, n-butanol, hydrogen, helium, nitrogen, and ammonia were used by Steiner-Taborek [4]. The correlation for the two-phase heat transfer coefficients in evaporator is

$$h_r = ((h_{\text{pool}})^3 + (h_{cb})^3)^{1/3} \tag{19}$$

where h_{cb} is the forced convection coefficient, which is a function of μ_f, μ_v, k_f, k_v, ρ_f, ρ_v, G, the Fanning friction factor, quality, and the local liquid- and gas-phase forced convection coefficients. The expression of h_{cb} is very complex; therefore, the details are not listed here. Figure 7 shows that β for both h_H and h_L from the correlations of [4] is between -12% and $+10\%$.

Correlation from Wattelet et al. [5]

The database for R-12, R134a, and a mixture of R22/R-124/R-152a were used by Wattelet et al. [5]. The correlation for the two-phase heat transfer coefficient in evaporator is

$$h_r = ((h_{\text{pool}})^{2.5} + (h_{cb})^{2.5})^{0.4} \tag{20}$$

where h_{cb} can be expressed as

$$h_{cb} = \left(1 + 1.925 \cdot X_{tt}^{-0.83}\right) \cdot h_l \cdot \delta \tag{21}$$

The reduction parameter δ in Eq. (21) is

$$\begin{aligned} \delta &= 1.32 \cdot Fr_f^{0.2} && \text{if} \quad Fr_f < 0.25 \\ \delta &= 1 && \text{if} \quad Fr_f \geq 0.25 \end{aligned}$$

Figure 8 shows that β for both h_H and h_L from the correlations of [5] is between -11% and $+9\%$.

Note that β for both h_H and h_L from the correlations of [4, 5] are smaller than those from the present correlations, Eqs. (7) and (8). However, the required numbers of variables of the current correlations are much fewer than those of the correlations from [4, 5].

CONCLUSIONS

The dimensionless correlations for analyzing the characteristics of the series-connected two-evaporator refrigerating system with R-290 as the refrigerant, h_H, h_L, and α, are developed in this paper. In addition, a comparison of the experimental measurements of h_H and h_L to those calculated by the correlations from literature is made. Some conclusions may thus be drawn:

1. Based on the present experimental results and the Buckingham Pi theorem, the dimensionless correlations for h_H, h_L, and α can be developed. The accuracy of those is acceptable.
2. P_c, L_L, and $LMTD_H$ are the dominant factors of h_H, while L_H and $LMTD_L$ are those for h_L. On the other hand, the dominant factors for α are P_c, $LMTD_H$, and $LMTD_L$.
3. β for both h_H and h_L from the correlations of [2, 3] are greater than those from the current correlations, Eqs. (7) and (8), while that from the correlations of [4, 5] are smaller.
4. In this study, the dimensionless correlations for R-290 may be applied to other refrigerants with refrigerating properties close to R-290, such as R-22.
5. For the two-phase heat transfer coefficient of a refrigerant in an evaporator, the accuracy in prediction with some correlations from the literature [4, 5] is good. However, the specific correlations are much more complicated than the dimensionless correlations developed in this paper.

NOMENCLATURE

Bo	boiling number
C_p	specific heat, $\text{J} \cdot \text{kg}^{-1}\text{K}^{-1}$
C_1–C_5	constants in Eq. (17)
d_H	diameter of the high-temperature capillary tube, mm
d_L	diameter of the low-temperature capillary tube, mm
$D_{i,i}$	inside diameter of the inner tube of evaporator, mm
$D_{i,o}$	inside diameter of the outer tube of evaporator, mm
$D_{o,i}$	outside diameter of the inner tube of evaporator, mm
$D_{o,o}$	outside diameter of the outer tube of evaporator, mm
E	enhancement factor
Fr_{lo}	Froude number with all flow as liquid
F	frequency of the compressor, s^{-1}
G	mass flux, $\text{k} \cdot \text{gm}^{-2}\text{s}^{-1}$
h_{cb}	heat transfer coefficient of convective boiling, $\text{W} \cdot \text{m}^{-2}\text{K}^{-1}$
h_f	heat transfer coefficient of saturated liquid, $\text{W} \cdot \text{m}^{-2}\text{K}^{-1}$
h_H	heat transfer coefficient of refrigerant in the high-temperature evaporator, $\text{W} \cdot \text{m}^{-2}\text{K}^{-1}$
h_L	heat transfer coefficient of refrigerant in the low-temperature evaporator, $\text{W} \cdot \text{m}^{-2}\text{K}^{-1}$
h_{pool}	heat transfer coefficient of pool boiling, $\text{W} \cdot \text{m}^{-2}\text{K}^{-1}$
k_f	thermal conductivity of saturated liquid, $\text{W} \cdot \text{m}^{-1}\text{K}^{-1}$
L_H	length of the high-temperature capillary tube, m
L_L	length of the low-temperature capillary tube, m
$LMTD_H$	logarithmic-mean temperature difference of the high-temperature evaporator, K
$LMTD_L$	logarithmic-mean temperature difference of the low-temperature evaporator, K
M	molecular weight
$\dot{\text{M}}$	mass flow rate, kg s^{-1}
P_c	condensing pressure, Pa
Pr_f	Prandtl number of saturated liquid
Q	heat flux, $\text{W} \cdot \text{m}^{-2}$

Q_H	cooling capacity of the high-temperature evaporator, W
Q_L	cooling capacity of the low-temperature evaporator, W
Re_f	Reynolds number of saturated liquid
S	suppression factor
T	Temperature, K
U	overall heat transfer coefficient, $\mathrm{W \cdot m^{-2}K^{-1}}$
X	Quality

Greek Symbols

ΔT_{sc}	Subcooling, K
ρ_f	density of saturated liquid, $\mathrm{k \cdot gm^{-3}}$
ρ_v	density of saturated vapor, $\mathrm{k \cdot gm^{-3}}$
μ_f	viscosity of saturated liquid, Pas
μ_v	viscosity of saturated vapor, Pas
A	ratio of cooling capacity
B	range of maximum error
Π	dimensionless parameter
λ	latent heat ($\mathrm{J \cdot m^{-3}}$)

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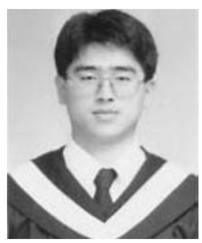

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