PERFORMANCE OF COMPACT BRAZED PLATE HEAT EXCHANGER OPERATING AS CONDENSER IN DOMESTIC HEAT PUMP SYSTEM—AN EXPERIMENTAL INVESTIGATION

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ABSTRACT

Brazed plate heat exchangers have become the preferred choice as heat exchangers in domestic heat pumps by Swedish manufacturers, both as evaporators and condensers. The reason is the high heat transfer rates at small temperature differences obtainable with this type of heat exchanger. At the Royal Institute of Technology, a large number of measurements have been conducted on one specific condenser under varying conditions. This paper reports area averaged condensation heat transfer coefficients under operating conditions encountered in ground source heat pump applications. Two distinct regimes of heat transfer mechanism are observed. For low overall heat transfer rates (many plates), the classical theory of Nusselt yields reasonable values. For high heat transfer rates (few plates), the resulting condensation heat transfer coefficient is significant higher than predicted by the Nusselt theory and the all-liquid convective heat transfer coefficient yields reasonable agreement.

Key Words: ***plate heat exchangers, condensation, domestic heat pump.***

1 INTRODUCTION

Plate heat exchanger has emerged as the most promising and effective heat exchanger for condensation and evaporation in ground source heat pumps. Swedish heat pump manufacturers use compact brazed plate heat exchangers exclusively for these applications. Plate heat exchangers have several features making them suitable for condensers and evaporators in domestic heat pumps; e.g. small internal and external volume and effective heat transfer. As the internal volumes are small, the amount of refrigerant used in the system may be kept low. In addition, as the external volume of the heat exchanger is small, compact heat pumps may be manufactured which make installations at the costumer easier.

Compact brazed plate heat exchangers are developed from the gasketed plate heat exchangers. Plate heat exchangers consist of several plates stacked to form multiple parallel channels. Most common is to use identical plates; however, to better meet heat transfer demand and available pressure drop, mixed plates may also be used. The plate heat exchanger was originally used in the dairy industry due to the ease at which the plates could be cleaned. In addition, due to high heat transfer coefficients, small temperature difference was obtained, yielding small wall temperature differences, which also was beneficial in the dairy industry. In refrigerating applications, the gaketed plate heat exchangers are not common; instead, compact brazed plate heat exchangers are used. The major reason for this is that brazed plate heat exchangers are tight to the surrounding and are able to withstand high operating pressures. This has become very important in light of the green house effect and the impact of chlorinated refrigerants on the ozone layer.

The most common plate is the corrugated plated, also called chevron type or herringbone type. In principal, the plates are pressed with a sinusoid like pattern, where the corrugations are oriented at some angle (denoted chevron angle) with respect to the main flow direction. The corrugation pattern provides many contact points between two adjoining plates, at which the plates are brazed together, and offer therefore a strong unit. The brazed plate heat exchanger has typical maximum operating pressure above 30 bar(a).

A compact brazed plate heat exchanger is normally manufactured from stainless steel plates, pressed to the desired pattern. Every second plate is rotated and put on top on the previous one, see **Fig. 1**. In between the plates, a copper (normally) sheet is placed. The number of plates depends on the desired load the heat exchanger should have. Simply by adding more plates, larger heat loads may be transferred.

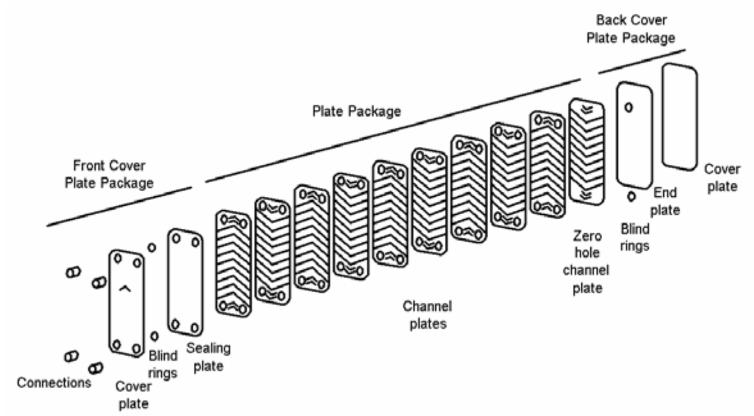

Fig. 1. Assembly of a brazed plate heat exchanger
(Adopted from SWEP International AB).

1.1 Geometry and Important Definitions of Plate Heat Exchanger

In **Fig. 2** the important geometrical parameters for a plate heat exchanger are defined, the chevron angle (φ), corrugation (pressing) depth[1] (b), and the corrugation pitch (Λ). It has also been found convenient to define a parameter "Surface enlargement factor" (ϕ), the ratio between the heat transfer area and the projected area.

In the treatment of plate heat exchangers in the literature, at least two different definitions of the hydraulic diameter are used. The perhaps most common definition used is similar to the definition of two wide parallel plates, with a distance of b between the plates, hence

$$d_e = 2 \cdot b \tag{1}$$

The other definition, perhaps more "physically" correct as it is defined according to the non-circular tube definition of the hydraulic diameter, is

$$d_h = \frac{4 \cdot V}{A} = \frac{2 \cdot b}{\phi} \tag{2}$$

In the following, we distinguish between these two by the use of different subscript, e for effective diameter and h for hydraulic diameter, as suggested by Shah and Focke (1988).

[1] Sometimes also referred to as the average plate spacing.

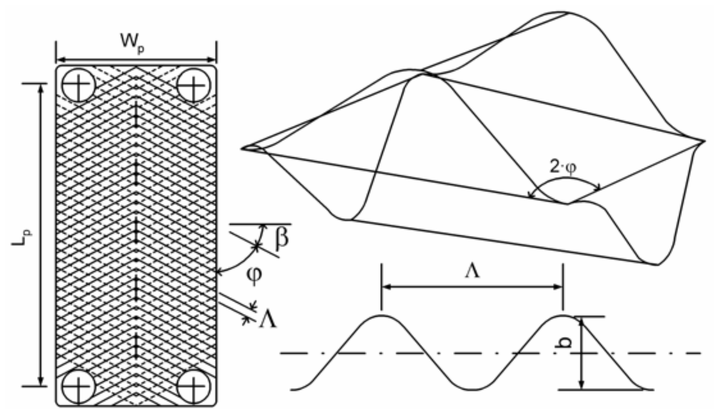

Fig. 2. Plate heat exchanger geometry.

The Reynolds and the Nusselt numbers may be defined as

$$Re = \frac{\rho \cdot u_m \cdot d_e}{\mu} = \frac{G \cdot d_e}{\mu} \tag{3}$$

and

$$Nu = \frac{\alpha \cdot d_e}{\lambda} \tag{4}$$

where the effective diameter is used. When using correlations from the literature it is very important to adhere to the definitions used in the original texts. Especially confusing is the definition of the friction factor, where an extra geometrical parameter, the flow length, is added. Two different definitions of flow length can be found in the literature, developed flow length and the length between the inlet and outlet ports. These have been mixed freely with different definitions of hydraulic diameter. The friction factor of a plate heat exchanger in this work is defined based on the effective diameter and the projected length between the inlet and outlet ports, as

$$f = \frac{\rho \cdot \Delta p \cdot d_e}{2 \cdot L_p \cdot G^2} \tag{5}$$

However, several different definitions may be found in the literature and it is important to adhere to the definitions used during the calculations.

1.2 Single Phase Correlation

The heat transfer and pressure drop in plate heat exchangers have been investigated for several years, and the number of reports has become rather extensive. On the other hand, the possible combinations of geometric parameters are almost infinite. Hence, there does not exist any general theory or correlation covering all geometrical combinations. Each investigation should therefore rather be regarded as a special case and the results only applicable for the specific geometry tested.

In the present investigation the single phase film heat transfer coefficient is calculated using (Bogaert and Bölcs 1995)

$$\mathrm{Nu} = B_1 \cdot \mathrm{Re}^{B_2} \cdot \mathrm{Pr}^{\frac{1}{3} e^{\frac{6.4}{\mathrm{Pr}+30}}} \cdot \left(\frac{\mu}{\mu_w}\right)^{\frac{0.3}{(\mathrm{Re}+6)^{0.125}}} \tag{6}$$

where the specific values of the constants B_1 and B_2 were given to the present author by the manufacturer and depend on the specific heat exchanger and Reynolds number.

1.3 Condensation (in Plate Heat Exchangers)

Nusselt suggested that gravity controlled condensation heat transfer coefficient may be calculated as (Collier and Thome 1996)

$$\alpha_r \cdot \frac{1}{\lambda_l} \cdot \left[\frac{\mu_l^2}{\rho_l \cdot (\rho_l - \rho_g) \cdot g}\right]^{1/3} = 1.47 \cdot \mathrm{Re}^{-1/3} \tag{7}$$

The classical Nusselt theory is valid for low Reynolds numbers on a plain wall. Plate heat exchangers have a rather different geometry. However, condensation in plate heat exchangers has also been reported in the literature.

For instance, Cooper (1987) discussed the application of plate heat exchangers as condensers for steam. He used the correlation by Lockhart-Martinelli for pressure drop and a simple condensate heat transfer correlation by Ananiev, shown to be successful to predict local heat transfer coefficients during condensation in plate heat exchangers.

$$\alpha_r = \alpha_l \cdot \sqrt{\frac{\rho_l}{\rho_{tp}}} \tag{8}$$

Baskin (1991) investigated the literature for plate heat exchanger in heat pumps. He stated that heat transfer for condensation should be calculated as

$$\overline{\alpha_r} = 13.8 \frac{\lambda_l}{d_e} \left(\frac{c_p \mu_l}{\lambda_l}\right)^{1/3} \left(\frac{\Delta h_{lg}}{c_p \Delta t}\right)^{1/6} \left[\frac{d_e G_g}{\mu_l} \left(\frac{\rho_l}{\rho_g}\right)^{0.5}\right]^{0.2} \tag{9}$$

if $1\,000 < d_e\, G_g / \mu_l\, (\rho_l/\rho_g)^{0.5} < 20\,00$ and for $20\,000 < d_e\, G_g / \mu_l\, (\rho_l/\rho_g)^{0.5} < 100\,00$ as

$$\overline{\alpha_r} = 0.1 \frac{\lambda_l}{d_e} \left(\frac{c_p \mu_l}{\lambda_l}\right)^{1/3} \left(\frac{\Delta h_{lg}}{c_p \Delta t}\right)^{1/6} \left[\frac{d_e G_g}{\mu_l} \left(\frac{\rho_l}{\rho_g}\right)^{0.5}\right]^{2/3} \tag{10}$$

Thonon and Chopard (1995) studied condensation in plate heat exchangers and reviewed prediction methods available in literature. They suggests that condensation in plate heat exchangers should be modeled as

$$\alpha_{gr} = \left(\frac{\alpha_{corrugated}}{\alpha_{plain}}\right)_{liquid} \cdot 1.1 \cdot \lambda_l \cdot \left[\frac{\mu_l^2}{\rho_l \cdot (\rho_l - \rho_g) \cdot g}\right]^{-1/3} \cdot \mathrm{Re}_l^{-1/3} \tag{11}$$

and

$$\alpha_{cv} = a \cdot Re_{LO}^{b} \cdot Pr_{l}^{c} \cdot \left(1 + x \cdot \left(\frac{\rho_l}{\rho_g} - 1\right)\right)^{0.5} \tag{12}$$

where a, b, and c the constants from the single phase heat transfer coefficient. The transition between these regions occurred smoothly and thence

$$\alpha_r = \sqrt{\alpha_{gr}^2 + \alpha_{cv}^2} \tag{13}$$

Yan et al. (1999) investigated R134a condensating in a plate heat exchanger. At higher vapor quality, the heat transfer and pressure drop were also higher. Higher heat flux does not significantly increase heat transfer. Increasing system pressure slightly decreases the heat transfer, however the effect was rather small. The heat transfer was correlated as

$$\alpha_r = \frac{\lambda_l}{d_h} \cdot 4.118 \cdot Re_{eq}^{0.4}\, Pr_l^{1/3} \tag{14}$$

where

$$Re_{eq} = \frac{G \cdot \left[(1 - x) + x \cdot \left(\frac{\rho_l}{\rho_g}\right)^{0.5}\right] \cdot d_h}{\mu_l} \tag{15}$$

Wang et al. (2000) studied steam condensing in a plate heat exchanger. A literature review of available condensation models for PHE was conducted. All of the correlations were stated to be cast in an inappropriate form. In the analysis of their experiments, the void fraction by Zivi was used. They stated that the heat transfer should be calculated in a step-wise manner due to non-linearity. The data were curve fitted to the equation

$$\alpha_r = \alpha_l \cdot \left(\frac{\rho_l}{\rho_{tp}}\right)^{a + b \cdot Re_l^c} \tag{16}$$

where the constants were determined as a ≈ 0.3 to 0.37, b ≈ 5.0 to 6.0 and c ≈ -0.6 to -0.64.

Thonon and Bontemps (2002) studied condensation in a welded plate heat exchanger using pure and mixtures of hydrocarbons. A new correlation was proposed, based on film condensation and used an corrective term accounting for the geometry. Equivalent Reynolds number was used, due to better agreement with experiments.

$$\alpha_r = \alpha_{LO} \cdot 1564 \cdot Re_{eq}^{-0.76} \tag{17}$$

where Re_{eq} is defined in eq. (15).

2 EXPERIMENTAL SETUP

The experimental equipment used during the tests consists of a well-instrumented lab rig; see **Fig. 3**, emulating a domestic heat pump. R134a is used as refrigerant and a mixture of Ethanol (~24% by mass) and water is used as secondary refrigerant (brine). Water is used as heat carrier. An electric expansion valve from Siemens and a PID-controller from Eurotherm are used to control the superheat.

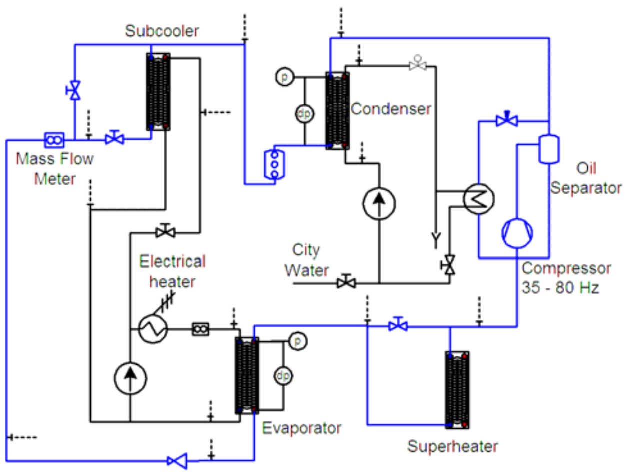

Fig. 3. Experimental test rig.

A hermetic scroll compressor from Copeland is used, and a frequency inverter, allowing for testing different heat loads, adjusts its speed. Brazed plate heat exchangers from SWEP are used as evaporator and condenser.

The mass flow rate of the refrigerant is measured using a Micromotion Coriolis Mass Flow Meter and the brine volumetric flow rate is measured using a MagnetoFlow Primo from BadgerMeter. All temperatures are measured with T-type thermocouples. The cold junctions of each and every one of the thermocouples are connected to an isothermal block. The temperature of the isothermal block is measured by the logger and used as reference temperature. Pressure transducers from Druck limited are used for measuring absolute pressures (PDCR 960 & 961) and differential pressures (PDCR 2110 & 2160). Campbell Scientific Instrument type CSI 21X is used as data logger.

A computer program written in HP VEE communicates with the logger and retrieves the data. A reading is taken every 10^{th} second and every measured point consists of 120 sequential readings from the logger. A stability criterion is used where 8 pre-chosen temperatures have to be within certain limits.

For all these 8 temperatures, HP VEE calculates the standard deviation. If the standard deviation is too large, the measured point is rejected and then a new trial is initiated. If the standard deviation is within the limit[2] then eight straight lines, one for each temperature, are produced using regression analysis. The difference between the first point (corresponding to the oldest reading) and the last point (corresponding to the newest reading) should not be greater than a preset value. If it is, the measurement point is rejected and a new trial is initiated. Having a too large difference between the most recent reading and the oldest reading indicates that the system has not reached steady state.

[2] This was usually set to 0.1 K except for the temperature of the refrigerant leaving the evaporator.

If all criteria are within the given range, HP VEE transfers all measured data for the last 120 readings into MS Excel. Then the set point for the next point (e.g. a new brine flow rate) is set to the auxiliary equipment and the computer waits for the next steady state.

2.1 Data Reduction

It is difficult to measure and evaluate local heat transfer coefficients in compact brazed plate heat exchangers since the actual heat transfer surface is inaccessible. In this investigation, the plate area averaged condensing heat transfer coefficient is determined. As the condenser may consist of one to three zones, depending on the operating conditions, different averaging methods may be applied. Here the condensing heat transfer coefficient is evaluated as (Palmer et al. 2000)

$$(UA)_{tot} = (UA)_{sub} + (UA)_{cond} + (UA)_{desup} \tag{18}$$

where

$$(UA)_i = \frac{\dot{Q}_i}{\vartheta_{LMTD_i}} \tag{19}$$

The energy transferred in each section of the condenser is calculated using the enthalpies of the saturation line of the refrigerant. Thus, the transferred heat in the subcooled section of the condenser is

$$\dot{Q}_{sub} = \dot{m}_r \cdot \left(h_r^{'} - h_{r,out}\right) \tag{20}$$

and the transferred heat in the condensing section of the condenser is

$$\dot{Q}_{cond} = \dot{m}_r \cdot \left(h_r^{''} - h_r^{'}\right) \tag{21}$$

and finally the transferred heat in the desuperheated section is

$$\dot{Q}_{desup} = \dot{m}_r \cdot \left(h_{r,in} - h_r^{''}\right) \tag{22}$$

The temperature difference used in eq. (19) is the classical logarithmic mean temperature difference, found in any textbook concerning heat transfer. In order to being able to calculate the logarithmic mean temperature difference, the two water temperatures at the intersections between the three sections are required. These are calculated using energy balances of the subcooled and desuperheating section. Hence, the water temperature corresponding to saturated liquid refrigerant is calculated as

$$t_w^{'} = t_{w,in} + \frac{\dot{Q}_{sub}}{\dot{m}_w \cdot c_{p_w}} \tag{23}$$

and the water temperature corresponding to saturated vapor refrigerant is calculated as

$$t_w^{''} = t_{w,out} - \frac{\dot{Q}_{desup}}{\dot{m}_w \cdot c_{p_w}} \tag{24}$$

The refrigerant condensing heat transfer coefficient is calculated as

$$\alpha_r = \frac{1}{\frac{1}{U_{tot}} - \frac{1}{\alpha_w}} \tag{25}$$

where the heat transfer resistance in the heat transfer wall is neglected. As the heat transfer area of the two fluid sides is equal in a plate heat exchanger, the heat transfer area is not included in eq. (25). One should perhaps comment on the magnitudes of the individual film heat transfer coefficients; typical values of water film heat transfer coefficient are 8 000 to 10 000 whilst the refrigerant condensation heat transfer coefficient is less than 5 000. Thus, the main heat transfer resistance is on the refrigerant side.

2.2 Operating Conditions

The operating conditions during experiments are those typical of domestic ground source heat pump. Thus, evaporating temperature is varied between -10 °C and +5 °C. The water temperature to the condenser is controlled in such a way that the refrigerant liquid temperature into the expansion device is between 20 °C to 50 °C. The experimental obtained reduced pressure of the refrigerant (R134a) in the condenser is from 0.14 to 0.56.

The superheat from the evaporator is controlled to be only a few degrees, yielding the hot gas temperature depended on the condenser and evaporating pressure. Typical values of the hot gas temperature are between 60 °C to 120 °C.

Two different sizes of the same heat exchanger model (a B25[3] from SWEP International AB) are used as condenser. The system is equipped with a liquid line receiver, hence the amount of subcool in the condenser are limited. A separate subcooler is installed in the system. The water flow rate in the condenser is not controlled. The pump is running at constant speed, giving similar water flow rate for many of the measured points. However, different water pumps are used due to mechanical failure. 919 points are collected in the present investigation.

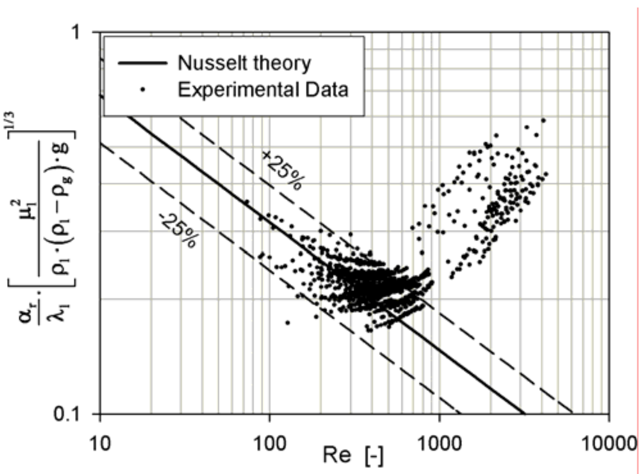

Fig. 4. Experimentally obtained dimensionless condensation heat transfer coefficients.

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[3] One of the most common heat exchanger used for condenser in domestic ground source heat pumps in Sweden.

3 EXPERIMENTAL RESULTS AND DISCUSSION

The resulting condensating heat transfer coefficient is represented in this investigated as suggested by Nusselt, i.e. the left hand side of eq. (7). All measurements are plotted in **Fig. 4**.

It may be seen from **Fig. 4** that three different regions exists, one in which the experimental data are well represented by the classical theory by Nusselt (within ±25%), and one where significant enhancement over classical Nusselt theory is obtained, and finally one transition region connecting these two regions.

These two regions have been referred to as gravity controlled and shear controlled regions (Thonon and Chopard 1995). As the reported heat transfer coefficients in the present investigation is total area averaged heat transfer coefficients, these labels may not be entire correct. In addition, the data reduction procedure used does not consider the fact that condensation also occurs even if the refrigerant vapor is not saturated. Condensation will occur if the wall temperature is below the saturation temperature of the refrigerant. A typical simplified temperature profile is indicated in **Fig. 5**. The “condensation” section of the condenser is assumed to be the “saturated condensation” section. The “desuperheating” section in this investigation contains both the “truly desuperheating” section and the “superheated condensation” section.

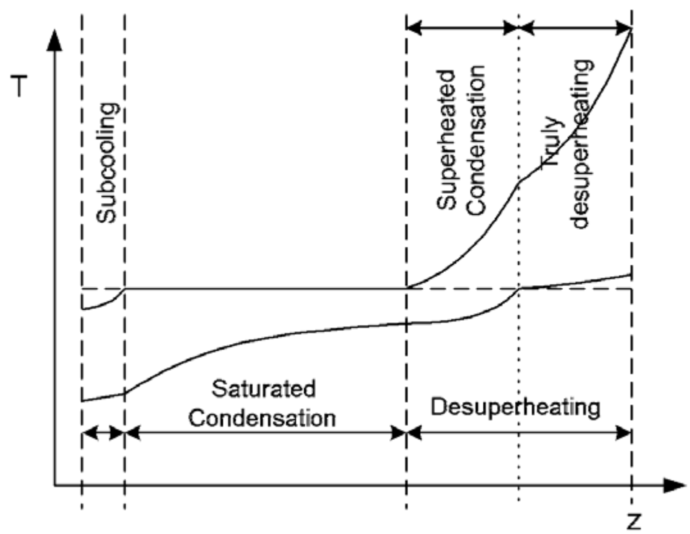

Fig. 5. Schematic temperature profiles in condenser. The different condenser sections are indicated in the figure.

This could be one reason for the significant scatter seen in **Fig. 4**. A detailed numerical stepwise investigation is required in order to determine the local condensation heat transfer coefficient. However, this is beyond the scope of the present investigation.

It may be interesting to note the resemblance between **Fig. 4** and Fig. 10.6 in Collier and Thome (1996), where area averaged condensating heat transfer coefficients are plotted for three different investigations. Qualitatively agreement are obtained between the experimental data and the values in Fig. 10.6 in Collier and Thome (1996) and as reported by Thonon and Chopard (1995) in their Fig. 3.

Thonon and Chopard (1995) suggests to calculate the high Reynolds range (shear controlled) as a multiplier of the all liquid heat transfer coefficient. The multiplier was suggested to be a function of thermodynamic quality, in addition to vapor and liquid densities. As the reported heat transfer coefficient in the present investigation is area averaged values, the use of the quality is not applicable. However, evaluating the all liquid heat transfer coefficient for all points, using an asymptotic model, as suggested by Thonon and Chopard (1995), provides an rough design criterion. Thus, it may be suggested that the performance of the condenser may be modeled as

$$\alpha_r = \sqrt{(\alpha_{LO})^2 + (\alpha_{Nusselt})^2} \tag{26}$$

where the all liquid heat transfer coefficient is calculated using eq. (6) using liquid properties and the entire mass flow as liquid. $\alpha_{Nusselt}$ is calculated using the theory by Nusselt, eq. (7). This approach predicts 93.5% of the data within ±25% and 80% of the data within ±13.5%, see **Fig. 6**.

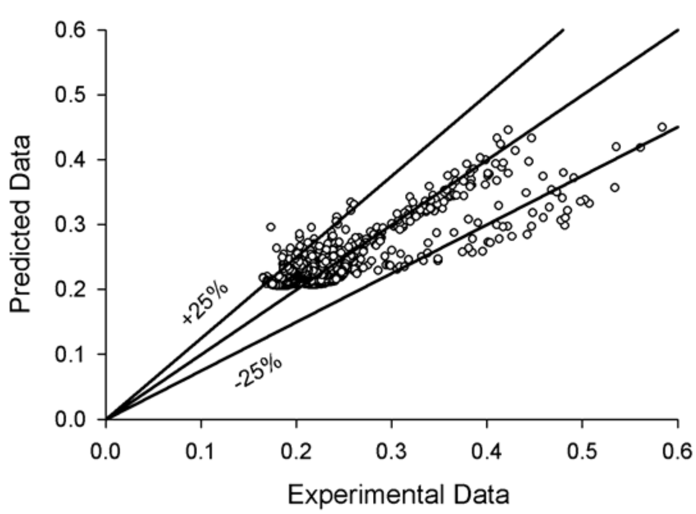

Fig. 6. Predicted heat transfer coefficient, eq. (26), against experimental data.

4 CONCLUSIONS

Condensation of refrigerant R134a in compact brazed plate heat exchangers in a domestic heat pump has been investigated. Typical operating conditions for a ground source domestic heat pump were employed. The area averaged condensating heat transfer coefficient was determined, including the desuperheating and the subcooling section of the condenser. Two distinct regions were observed, gravity controlled at low Reynolds numbers and shear controlled at high Reynolds numbers.

Consequently, the gravity-controlled region was correlated using classical Nusselt theory. The shear-controlled region was correlated using the all-liquid single phase heat transfer coefficient without any enhancement factor. These two regions were put together using an asymptotic model. The predicted heat transfer coefficients were correlated within ±25% for most of the data. It was suggested that the outlined model might be used as initial design model for compact brazed plate heat exchangers operating as condensers in domestic ground source heat pumps.

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NOMENCLATURE

A	Wall area	m^2
a, b, c	Regression constants	-
b	Pressing depth	m
B_1	Constant	-
B_2	Constant	-
c_p	Specific heat capacity	J/(kg·K)
d_e	Effective diameter	m
d_h	Hydraulic diameter	m
f	Friction factor	-
G	Mass flux	kg/(s·m²)
g	Gravitational acceleration	m/s²
h	Specific enthalpy	J/kg
L_p	Plate length	m
m	Mass flow rate	kg/s
Nu	Nusselt number	-
Pr	Prandtl number	-
Q	Heat load	W
Re	Reynolds number	-
t	Temperature	°C
U	Overall heat transfer coefficient	W/(m²·K)
u_m	Plug velocity	m/s
V	Volume	m³
W_p	Plate width	m
α	Heat transfer coefficient	W/(m²·K)
$\beta = 90 - \varphi$	Chevron angle	°

Δh_{lg}	Latent heat of vaporization	J/kg
Δp	Pressure drop	Pa
ϑ_{LMTD}	Logarithmic mean temperature difference	K
Λ	Corrugation pitch	m
ϕ	Surface enlargement factor	-
φ	Chevron angle	°
λ	Thermal conductivity	W/(m·K)
μ	Dynamic viscosity	Pa·s
ρ	Density	kg/m³

<u>Index</u>

cond	Condensating section of condenser
corrugated	Corrugated channel
cv	Shear-controlled (convective)
desup	Desuperheating section of condenser
e	Effective
eq	Equivalent
freeze	Freezing point of mixture (brine)
g	Gas (vapor)
gr	Gravity-controlled
h	Hydraulic
in	Entering condenser
l	Liquid
liquid	All flow as liquid
LO	All flow as liquid
m	Mean
Nusselt	According to theory of Nusselt, eq. (7)
out	Leaving condenser
p	Plate
plain	straight tube
r	Refrigerant
sub	Subcooled section of condenser
tot	Entire condenser (heat exchanger)
tp	Two phase
w	Wall

ACKNOWLEDGEMENTS

The funding for this investigation from The Swedish Energy Agency, SWEP International AB and Thermia Värme AB is greatly acknowledged.