CPI, Edificio 8E, Cubo F, 5ª planta, Camino de Vera s/n, 46022 Valencia, Spain Tel. (34) 963877270, Fax (34) 963877272, Email: energeti@upvnet.upv.es

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Assessment of condensation heat transfer correlations in the modelling of fin and tube heat exchangers

F. Vera-García[a], J.R. García-Cascales[a,*], J.M. Corberán-Salvador[b], J. Gonzálvez-Maciá[b], David Fuentes-Díaz[b]

[a]Thermal and Fluid Engineering Department, Universidad Politécnica de Cartagena, 30202 Cartagena, Murcia, Spain
[b]Applied Thermodynamics Department, Universidad Politécnica de Valencia, Spain

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Abstract

This is the second paper of a series that assesses the performance of a refrigeration system model by means of cycle parameters. In this case, the condensation temperature is the parameter to study and it is focused on fin and tube condensers. It also studies the influence of the heat transfer models on the estimation of this refrigeration cycle parameter and different correlations for the heat transfer coefficients have been implemented in order to characterise the heat transfer in the heat exchangers. The flow inside the heat exchangers is considered one-dimensional as in previous works. In the cycle definition, other submodels for all the cycle component have been taken into account to complete the system of equations that characterises the behaviour of the refrigeration cycle. This global system is solved by means of a Newton–Raphson algorithm and a known technique called SEWTLE is used to model the heat exchangers. Some experimental results are employed to compare the condensation temperatures provided by the numerical procedure and to evaluate the performance of each heat transfer coefficient. These experimental results correspond to an air-to-water heat pump and are obtained by using R-22 and R-290 as refrigerants.

Keywords: Refrigeration; Air conditioning; Condenser; Finned tube; Modelling; Correlation; Heat transfer

Evaluation des corrélations de transfert de chaleur lors de la condensation dans la modélisation des échangeurs de chaleur du type aileté

Mots clés : Réfrigération ; Conditionnement d'air ; Condenseur ; Tube aileté ; Modélisation ; Corrélation ; Transfert de chaleur

1. Introduction

This paper assesses the performance of a refrigeration system model by means of the condensation temperature estimation. It is the second paper of a series and is focused

* Corresponding author. Tel.: +34 968 325 991; fax: +34 968 325 999.
E-mail address: jr.garcia@upct.es (J.R. García-Cascales).

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Nomenclature

A	cross-section area (m^2)
c_1, c_2	coefficients in the heat transfer multipliers
c_p	specific heat (J/kg K)
D_h	hydraulic diameter (m)
e	wall thickness (m)
f	friction factor
f_i	interfacial roughness correction factor
Fr	Froude number
g	gravity (m/s^2)
G	mass velocity (kg/s m^2)
Ga	Galileo number
G_W	transition flow rate
i	enthalpy (J/kg)
Ja	Jacobi number
J_G	dimensionless vapour velocity
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number
p	pressure (Pa)
p^*	reduced pressure
P	perimeter (m)
Pr	Prandtl number
q	heat flux (W/m^2)
Q	heat (W)
Re	Reynolds number
S	slip ratio
T	temperature (K)
u	velocity (m/s)
U	global heat transfer coefficient (W/m^2 K)
W	humidity (kg vap/kg dry air)
x	vapour quality
X_{tt}	Martinelli parameter
z, y	spatial co-ordinates (m)

Greeks

α	heat transfer coefficient (W/m^2 K)
ϵ	void fraction
η	dynamic viscosity (N s/m^2)
Δ	Increment
λ	thermal conductivity (W/m K)
∇^2	Laplacian operator
Φ_f^2	two-phase friction multiplier
ϕ	two-phase friction multiplier
ϕ, φ	generic variable
ρ	density (kg/m^3)
σ	surface tension
θ	angle with horizontal
τ	two-phase friction factor

Subscripts and superscripts

a	air
an	annular
c	convective
eq	equivalent
f	saturated liquid
g	saturated vapour
h	homogeneous
i	inlet, cell index
j	cell index
l	liquid
lo	liquid only
o	outlet, cell index
r	refrigerant
s	saturation
so	Soliman
strat	stratified
trans	transition
tt	turbulent–turbulent
v	vapour
w	wall
wat	water
wavy	wavy

on fin and tube condensers. It also studies the influence of heat transfer models on the estimation of refrigeration cycle parameters. As in the evaporation case, the thermal resistances in the coils are balanced and the correct characterisation of the heat transfer coefficients of both sides is fundamental. The flow inside the heat exchangers is considered one-dimensional as in previous works [1]. Different models are also considered for characterising the other component behaviour. The resulting global system of equations is solved by means of a variant of the Newton–Raphson algorithm. An existing technique called SEWTLE [2] is used to model heat exchangers. Some experimental results have been used to compare condensation temperatures provided by the numerical procedure and to evaluate the performance of each heat transfer coefficient. These experimental results correspond to an air-to-water heat pump and are obtained by using R-22 and R-290 as refrigerants.

In the following sections, a brief overview of the global model is carried out. Some aspects related to the condenser model and its analysis are also considered. However, for further information the interested reader is referred to previous authors' work for a wide description of the models and the iteration procedure used in the modelling of heat exchanger. The heat transfer coefficient correlations used in the code to characterise the behaviour of heat exchangers are then introduced. Firstly, a general comparison of the most common correlations is carried out. Among them those proposed by Cavallini and Zecchin, Shah, Dobson and Chato, Cavallini et al. and Thome et al. stand out. The experimental facility used in the modelling procedure is schematically described, emphasising the test heat exchanger specifications and the measurement procedures followed. As in other cases, the ART© code [3] is also used in the modelling procedure to obtain the numerical results which afterwards are compared with the experimental ones. The experiments have been modelled numerically so the performance of the cycle is compared on the basis of the condensation temperature by using this to evaluate the goodness of the numerical method and the heat transfer coefficient correlations used. This work is concluded by comparing the experimental results with the numerical ones so lastly the final conclusions can be drawn.

2. Global model solution and condenser governing equations

The behaviour of the refrigeration cycle is modelled as in [1] and after considering each model component, a global system of equations is obtained. The inputs of the problem are pressure, enthalpy, and mass flow rate at the inlet and outlet of each element and the system is solved by using a variant of the Newton–Raphson method.

The governing equations considered in the analysis of the condenser depend on the process which takes place in each part of it, as was also seen in the case of the evaporator. Firstly, after the compressor discharge, a de-superheating process takes place changing the refrigerant state to gas. When saturation temperature is reached, phase change occurs and different two-phase regimes are encountered inside the tubes. Finally, the refrigerant is subcooled if the heat exchanger size allows it. Then, from the point of view of the analysis, three different zones are considered (Fig. 1) in the heat exchanger calculations, of which the two-phase flow is characterised by a separate fluid model as in [1].

SEWTLE is the evaluation procedure employed in the heat exchanger analysis, in which special attention is paid to the heat transfer coefficient correlations utilised in each heat exchanger model. This is because a proper coefficient characterisation leads to more accurate results and therefore to a better approach to the refrigeration system behaviour.

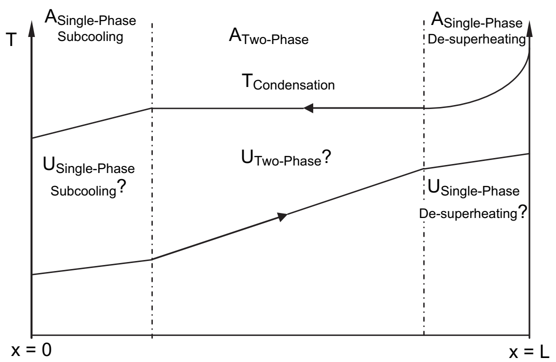

Fig. 1. Condenser modelling. Three zone model.

3. Heat transfer coefficients and friction factor correlations

This section focuses on the study of several condensation heat transfer coefficients in the refrigerant side, whilst also highlighting some of their characteristics. A brief reference is made to the friction factor coefficient and the heat transfer coefficient in the secondary fluid side. Other important correlations for heat transfer coefficients and friction factor in plain horizontal pipes can be found in [2].

3.1. Heat transfer coefficient correlations inside horizontal plain tubes

Heat transfer during condensation inside horizontal plain tubes is only considered. Regarding the estimation of the heat transfer coefficient, many semi-empirical correlations are available in the existing literature and their number is constantly growing due to the continuous evolution of the air-conditioning and refrigeration industry. The CFCs prohibition, the introduction of HCFCs, their HFC substitutes, the zeotropic and azeotropic mixtures, and other organic and inorganic refrigerants have contributed to the development of a large number of new correlations which attempt, in a certain way, to be as universal as possible.

In the case of condensation inside horizontal pipes, the phase-change process is dominated by vapour shear or gravity forces in such a way that the regime encountered depends on the force controlling the condensation process. Thus, if annular flow is the dominant process, it is because of the high vapour shear but if stratified, wavy or slug flows is found, it is due to gravity forces. It has been observed that in the annular flow regime, the heat transfer coefficient varies with mass velocity, G, vapour quality, x, and saturation temperature during condensation of pure fluids and nearly azeotropic mixtures. Only in the stratified regimes, the measured heat transfer coefficient was affected by temperature difference between saturation and tube wall ($T_s - T_w$) [4]. First, semi-empirical developments were derived for high mass flow rates and covered mostly the annular flow regime. Some examples of this case are the correlations developed by Cavallini and Zecchin [5] or Shah [6]. They modified single phase correlations by introducing an equivalent Reynolds number for the two-phase mixture or by means of adapted two-phase multipliers. Only a few correlations take into account the occurring flow pattern. Among them, those proposed by Dobson and Chato [7], Cavallini et al. [8] or Thome et al. [9] stand out.

3.1.1. Cavallini and Zecchin correlation

This correlation was obtained considering experimental data for R-113, R-12 and R-22 and is the result of the study of the condensation of saturated vapours inside tubes [5]. The expression proposed for the heat transfer coefficient was

$$\alpha_{\mathrm{tp}} = \frac{\lambda_{\mathrm{f}}}{D_{\mathrm{h}}} 0.05 Pr_{\mathrm{f}}^{1/3} Re_{\mathrm{eq}}^{0.8}, \tag{1}$$

where the equivalent Reynolds number, Re_{eq}, is defined by

$$Re_{\mathrm{eq}} = Re_{\mathrm{g}} \left(\frac{\eta_{\mathrm{g}}}{\eta_{\mathrm{f}}}\right) \left(\frac{\rho_{\mathrm{f}}}{\rho_{\mathrm{g}}}\right)^{0.5} + Re_{\mathrm{f}}. \tag{2}$$

3.1.2. Shah correlation

This was developed using experimental data from the condensation of water, R-11, R-12, R-22, R-113, methanol, ethanol, benzene, toluene and trichloroethylene inside horizontal, vertical, and inclined pipes of different diameters [6]. The correlation proposed defines the heat transfer coefficient as

$$\alpha_{\mathrm{tp}} = \alpha_{\mathrm{f}} \left[(1-x)^{0.8} + \frac{3.8 x^{0.76} (1-x)^{0.04}}{p^{*0.38}} \right], \tag{3}$$

where the liquid heat transfer coefficient is calculated by using the Dittus–Boelter equation

$$\alpha_{\mathrm{f}} = 0.023 Re_{\mathrm{f}}^{0.8} Pr_{\mathrm{f}}^{0.4} \frac{\lambda_{\mathrm{f}}}{D_{\mathrm{f}}}. \tag{4}$$

This correlates the experimental data with a mean deviation of 15%.

3.1.3. Dobson and Chato correlation

Dobson and Chato derived their correlation using R-22, R-134a, R-410A, and a mixture R-32/R-125 (60%/40%). They observed that the factors which control the flow in the case of condensation inside smooth horizontal tubes are gravity and vapour shear. At low vapour velocities, the effect of gravity prevails and condensate is formed in the upper part of the tube and flows downwards into a liquid pool which advances throughout the tube (which is driven axially) due to vapour push (or flow) and to gravitational forces (or head).

They divided the flow regimes into two groups in terms of void fraction [7].

- High void fraction regimes, divided into:
  - gravity dominated: stratified and wavy flow;
  - influenced equally by gravity and shear: wavy-annular flow;
  - shear dominated: annular flow and annular-mist flow.
- Low void fraction regimes that include slug, plug, and bubbly flow. In this case, gravity controls all the processes.

From their point of view, the flow regime must be known at each point in order to calculate the heat transfer coefficient properly. They only consider the transition between wavy and annular flow based on Soliman's work [10]. This takes into account the Froude number, which depends on the liquid Reynolds number, and is given by

$$Fr_{\mathrm{so}} = 0.025 Re_{\mathrm{f}}^{1.59} \left(\frac{1+1.09 X_{\mathrm{tt}}^{0.039}}{X_{\mathrm{tt}}}\right)^{1.5} \frac{1}{Ga^{0.5}} \quad \text{for } Re \le 1250, \tag{5}$$

$$Fr_{\mathrm{so}} = 1.26 Re_{\mathrm{f}}^{1.04} \left(\frac{1+1.09 X_{\mathrm{tt}}^{0.039}}{X_{\mathrm{tt}}}\right)^{1.5} \frac{1}{Ga^{0.5}} \quad \text{for } Re > 1250. \tag{6}$$

Thus, they consider that gravity-driven condensation phenomena which include stratified, wavy, and slug flow regions are lumped together because the dominant heat transfer mechanism is conduction across the film at the top of the tube. The force-convective and the film-wise components are weighed so the first one is important at the bottom and the second at the top. Then the final expression is

$$Nu = \frac{0.23 Re_{\mathrm{vo}}^{0.12}}{1+1.11 X_{\mathrm{tt}}^{0.58}} \left[\frac{Ga\, Pr_{\mathrm{f}}}{Ja_{\mathrm{f}}}\right]^{0.25} + (1-\theta_{\mathrm{f}}/\pi) Nu_{\mathrm{forced}}, \tag{7}$$

where θ_{f} is the angle subtended from the top of the tube to the liquid level,

$$Nu_{\mathrm{forced}} = 0.0195 Re_{\mathrm{f}}^{0.8} Pr_{\mathrm{f}}^{0.4} \phi_{\mathrm{f}}(X_{\mathrm{tt}}), \tag{8}$$

$$\phi_{\mathrm{f}}(X_{\mathrm{tt}}) = \sqrt{1.376 + \frac{c_1}{X_{\mathrm{tt}}^{c_2}}}. \tag{9}$$

For $0 < Fr_{\mathrm{f}} \le 0.7$

$$c_1 = 4.172 + 5.48 Fr_{\mathrm{f}} - 1.564 Fr_{\mathrm{f}}^2, \quad c_2 = 1.773 - 0.169 Fr_{\mathrm{f}}. \tag{10}$$

For $Fr_{\mathrm{f}} > 0.7$

$$c_1 = 7.242, \quad c_2 = 1.655. \tag{11}$$

Fr_{f} is the liquid Froude number and θ_{f} is related to the void fraction by

$$\epsilon = \frac{\theta_{\mathrm{f}}}{\pi} - \frac{\sin(2\theta_{\mathrm{f}})}{2\pi}. \tag{12}$$

Then, by assuming a model for the void fraction, such as Zivi's model, they propose a simplification for the term as follows

$$1 - \frac{\theta_{\mathrm{f}}}{\pi} \cong \frac{\arccos(2\epsilon - 1)}{\pi}. \tag{13}$$

Under annular flow regime, Dobson and Chato derived a correlation by considering the Travis expression and assuming that the Reynolds number is usually greater than 1125. So the Nusselt number is given by a two-phase multiplier correlation

$$Nu = 0.023 Re_f^{0.8} Pr_f^{0.4}\left[1+\frac{2.22}{X_{tt}^{0.89}}\right]. \quad (14)$$

3.1.4. Cavallini et al. correlation

The model suggested by Cavallini et al. [11] is based on a predictive study of the flow patterns occurring during condensation inside horizontal smooth tubes operating with pure or blended halogenated refrigerants. It was derived by considering experimental data corresponding to R-22, R-134a, R-125, R-32, R-236ea, R-407C and R-410A which also includes Dobson and Chato values. The correlation depends on the regime considered each time. The transitions from one to another are defined as a function of the dimensionless vapour velocity, $J_G = xG/[gd\rho_g(\rho_f-\rho_g)]^{0.5}$ and the Martinelli parameter, $X_{tt} = (\eta_f/\eta_g)^{0.1}(\rho_g/\rho_f)^{0.5}((1-x)/x)^{0.9}$.

The flow is annular if $J_G \geq 2.5$ and $X_{tt} < 1.6$ and then the heat transfer coefficient is calculated by means of

$$\alpha = \rho_f c_{p_f}\left(\frac{\tau}{\rho_f}\right)^{0.5}\frac{1}{T^+}, \quad (15)$$

where T^+ is a parameter which is a function of the liquid Prandtl and Reynolds numbers, and τ is a two-phase friction factor given by $\tau = (dp/dz)_f\, d/4$.

The flow is in the transition region between annular and stratified flows or is fully stratified if $J_G < 2.5$ and $X_{tt} \leq 1.6$. In this case the transition heat transfer coefficient α_{trans} is given by a linear interpolation between the heat transfer coefficient for annular flow at $J_G = 2.5$ and that for stratified flow α_{strat}

$$\alpha_{trans} = \left(\alpha_{an,J_G} - \alpha_{strat}\right)\left(\frac{J_G}{2.5}\right) + \alpha_{strat}, \quad (16)$$

where α_{strat} is

$$\alpha_{strat} = 0.725\left\{1+0.82[(1-x)/x]^{0.268}\right\}^{-1}\left[k_f^3\rho_f(\rho_f-\rho_g)\, g i_{fg}/(\eta_f D\Delta T)\right]^{0.25} + \alpha_f(1-\theta/\pi), \quad (17)$$

where α_f is given by $\alpha_f = \alpha_{fo}(1-x)^{0.8}$. α_{lo} is a liquid-only heat transfer coefficient (the Dittus–Boelter equation) and $(\pi-\theta)$ half the angle subtended in the centre of the tube by the chord formed by the vapour–liquid interface. In this case, void fraction is determined by using Zivi's expression $\epsilon = x/[x+(1+x)(\rho_g/\rho_f)^{0.66}]$.

Finally, if $X_{tt} > 1.6$ with $J_G < 2.5$, the flow is in the transition from stratified to slug or it is slug. In this case the correlation suggested is

$$\alpha_{st-sl} = \alpha_{lo} + \frac{x}{x_{1.6}}(\alpha_{1.6} - \alpha_{lo}), \quad (18)$$

where α_{lo} is determined by using the Dittus–Boelter correlation, and the vapour quality corresponding to $X_{tt} = 1.6$, $x_{1.6}$, by means of

$$x_{1.6} = \frac{\left(\frac{\eta_f}{\eta_g}\right)^{1/9}\left(\frac{\rho_g}{\rho_f}\right)^{5/9}}{1.686 + \left(\frac{\eta_f}{\eta_g}\right)^{1/9}\left(\frac{\rho_g}{\rho_f}\right)^{5/9}}, \quad (19)$$

and the heat transfer coefficient, $\alpha_{1.6}$, by Expression (16) at $X_{tt} = 1.6$.

Bubbly flow may be encountered at low vapour qualities when J_G has values higher than $J_G > 2.5$. Due to a lack of data, annular flow correlations are also recommended. The interested reader can complete the previous definitions by Cavallini et al. [11].

3.1.5. Thome et al. correlation

This correlation considers a simplified flow structure of the flow regimes in order to predict the local condensation heat transfer coefficient. It is an improved version of the previous one and was derived employing the Cavallini et al. database along with data from other laboratories obtained with single component refrigerants, binary azeotropic and very near azeotropic mixtures. In this case: annular, intermittent, bubbly, stratified-wavy, fully stratified and mist flows are the regimes considered. The general expression for the local heat transfer coefficient is

$$\alpha_{tp} = \frac{\alpha_f\theta + (2\pi-\theta)\alpha_c}{2\pi}, \quad (20)$$

where α_f is the film condensation heat transfer coefficient, α_c is the convective contribution to the coefficient and θ is the upper angle of the tube not wetted by stratified liquid. Its value depends on the regime and is 0 if the flow is annular, intermittent or mist. If it is stratified $\theta = \theta_{strat}$, and in the case of stratified-wavy flow

$$\theta = \theta_{strat}\left[\frac{G_{wavy} - G}{G_{wavy} - G_{strat}}\right]^{0.5}. \quad (21)$$

The expression of θ_{strat} coincides with that defined for the evaporation correlation developed by Wojtan et al. [12]. In this case, the principles used in intube boiling are adapted to intube condensation heat transfer.

The convective condensation coefficient, α_c, is given by

$$\alpha_c = 0.003 Re_f^{0.74} Pr_f^{0.5}\frac{\lambda_f}{\delta} f_i, \quad (22)$$

where f_i is an interfacial roughness correction factor, given by $f_i = 1 + (u_v/u_f)^{1/2}((\rho_f-\rho_g)g\delta^2/\sigma)^{1/4}$ which is corrected by (G/G_{strat}) when the regime is fully stratified.

The film condensation contribution, α_f is defined by

$$\alpha_f = 0.728\left[\frac{\rho_f(\rho_f-\rho_g)g i_{fg}\lambda_f^3}{\eta_f D q}\right]. \quad (23)$$

The value of the void fraction used in the previous formulae is obtained by the logarithmic mean of the homogeneous void fraction, ϵ_h, and the Rouhani–Axelsson void fraction, ϵ_{ra},

$$\epsilon = \frac{\epsilon_h - \epsilon_{ra}}{\ln\left(\frac{\epsilon_h}{\epsilon_{ra}}\right)}, \tag{24}$$

where

$$\epsilon_h = \left[1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_f}\right)\right]^{-1}, \tag{25}$$

$$\epsilon_{ra} = \frac{x}{\rho_g}\left\{\left[1 + 0.12(1-x)\right]\left(\frac{x}{\rho_g} - \frac{1-x}{\rho_f}\right) + \frac{1.18}{G}\left[\frac{g\sigma(\rho_f - \rho_g)}{\rho_f^2}\right]^{1/4}(1-x)\right\}^{-1}. \tag{26}$$

El Hajal et al. defined the necessary flow pattern map for condensation inside horizontal plain tubes in [13]. The expressions for the mass flow rates used to determine the regime encountered each time along the tube can be found in this reference.

In order to illustrate the models presented above, the values of the heat transfer coefficient provided by the correlations described previously are displayed in Figs. 2 and 3 for different values of the mass flow rate (50, 125 and 200 kg/m^{2}s) and the heat flux (2000, 8000 and 14,000 W/m^2). All of them correspond to a pressure $p = 20$ bar and a horizontal tube diameter $D = 11.9 \times 10^{-3}$ m. Refrigerant in all cases is R-290. Special attention has been paid to five models, namely:

- Cavallini et al. correlation [8],
- Dobson and Chato correlation [7],
- Shah correlation [6],
- Thome et al. correlation [9],
- Cavallini–Zecchin correlation [5],

all of which, except for the Shah and Cavallini–Zecchin models, cover either totally or partially the pattern map.

A larger dependence on the mass flow rate is observed for the Dobson–Chato, Shah and Cavallini–Zecchin correlations than for the others.

3.2. Friction factor correlations

As in the study of evaporators, the prediction of pressure drops in condensers is an important issue for the accurate design and optimisation of refrigeration, air-conditioning and heat pump systems. In the case of the two-phase friction factor, this is also estimated by using the Friedel correlation [14] as in the evaporation case [1].

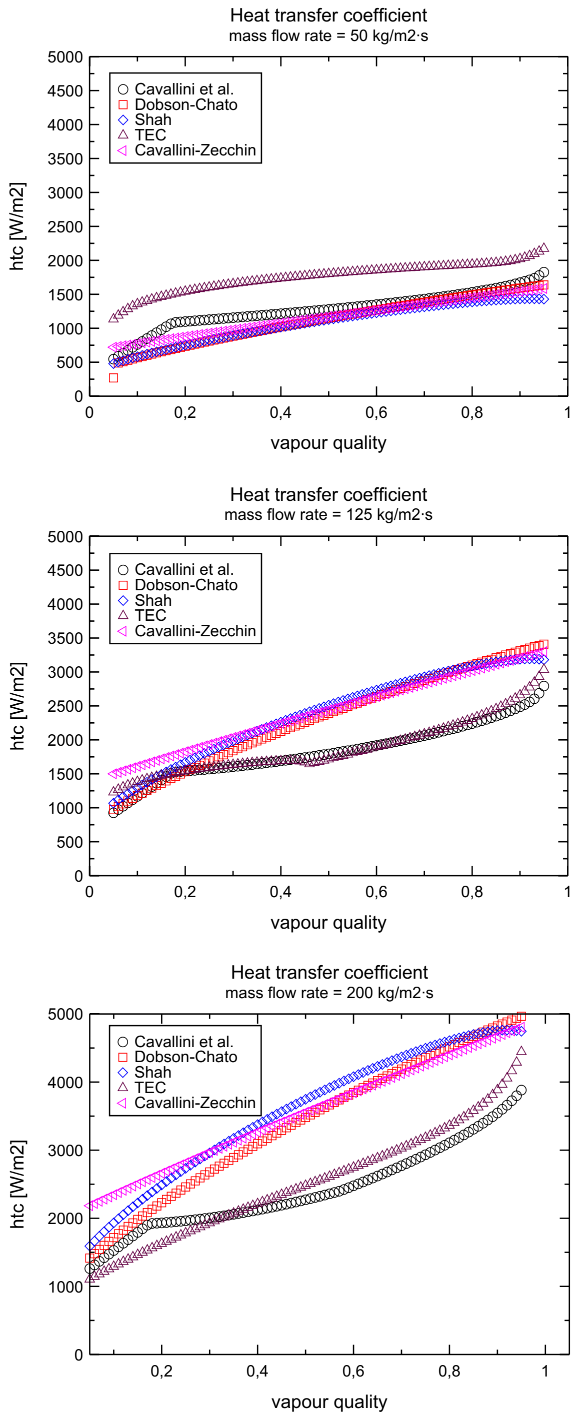

Fig. 2. Heat transfer coefficient at different mass flow rates (50, 125 and 200 kg/m^{2}s) and constant heat flux $q = 5000$ W/m^2.

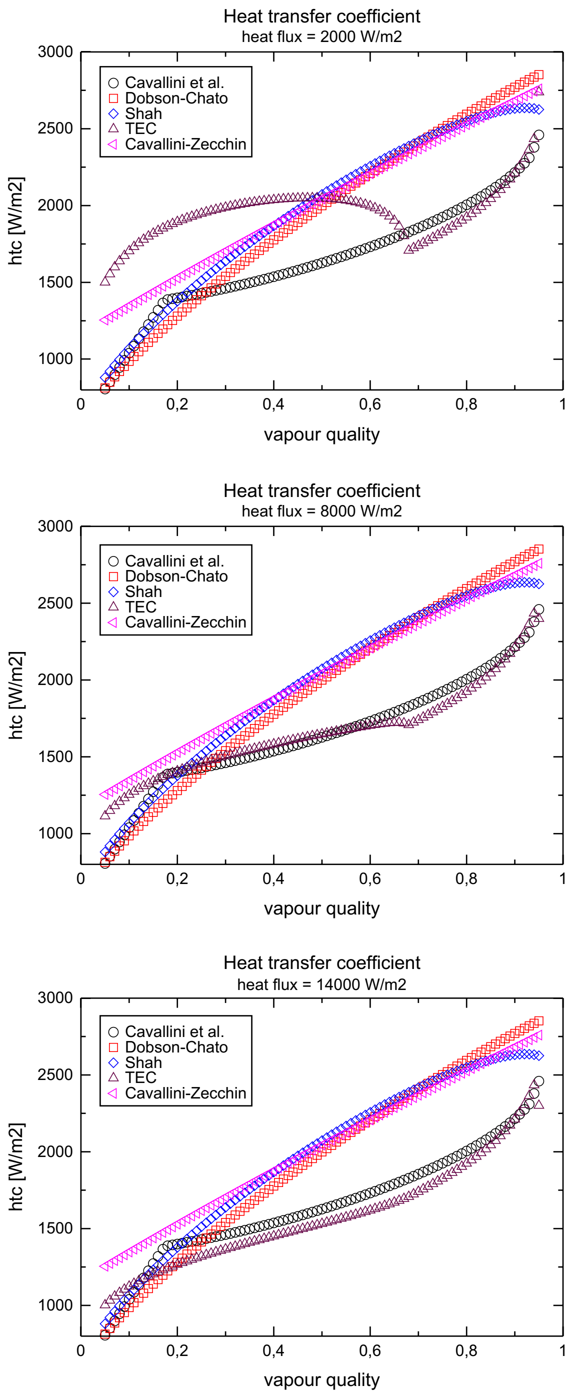

Fig. 3. Heat transfer coefficient at different heat fluxes (2000, 8000 and 14,000 W/m^2) and constant mass flow rate $m = 100$ kg/m^{2}s.

3.3. Air side correlations

In the case of condensers, the correlations for describing heat transfer exchange and friction on the air side are those proposed by Wang and Chi [15], the same as in the case of dry air for evaporators.

4. Experimental results

The experimental equipment used during the measurement stage is a heat pump that is a modified version of a R-22 catalogue model of CIATESA (IWA-95). This is specifically adapted, from the point of view of safety as well as performance, to be used with propane as the refrigerant. All materials and measurement devices utilised were intrinsically safe. Gas detectors were installed in the heat pump in combination with a switch which isolated the electrical circuits in accordance to the Spanish standard UNE 20-318 (for electrical installation in potential flammable atmospheres) and the UNE-EN 60079-10 (for electrical equipments in explosive environment).

4.1. Experimental equipment

The heat pump unit (Fig. 4) was equipped with: a reciprocating compressor; a thermostatic expansion valve; different brazed plate heat exchanger with 38 or 46 plates acting as evaporators/condensers and, of course, the different fin and tube heat exchangers studied in this paper that acted as condensers/evaporators depending on the case. In this study the fin and tube heat exchangers were tested as condensers of the heat pump, in such a way that the direction of the refrigerant was as displayed in Fig. 4. Two different heat exchangers, RHPU and AHPU described in [1] were tested as in the evaporation case. Additional data on the facility can also be found in this reference.

4.2. Measurements

As commented above, the main objective of this study is to evaluate the goodness of the model and check the behaviour of the correlations used in the characterisation of the condensation process in heat exchangers. Thus, some measurements were carried out with the objective of correctly obtaining the refrigerant conditions at the inlet and outlet of the tested heat exchangers. It was necessary to characterise the behaviour of the compressor with the refrigerants tested in this study. The compressor was analysed under the same range of working conditions during the measurement campaign. Fig. 5 shows the volumetric and isentropic efficiencies obtained experimentally and used in the characterisation of the compressor. These points correspond to different pressure ratios and have been depicted for the refrigerants used in the experiments.

For each measurement, superheat and subcooling temperatures of the refrigerant had to be stabilised. Furthermore, the mass flows of all loops of the experimental system were required to be stabilised. The level and possible appearance of bubbles in the flow of refrigerant were controlled through

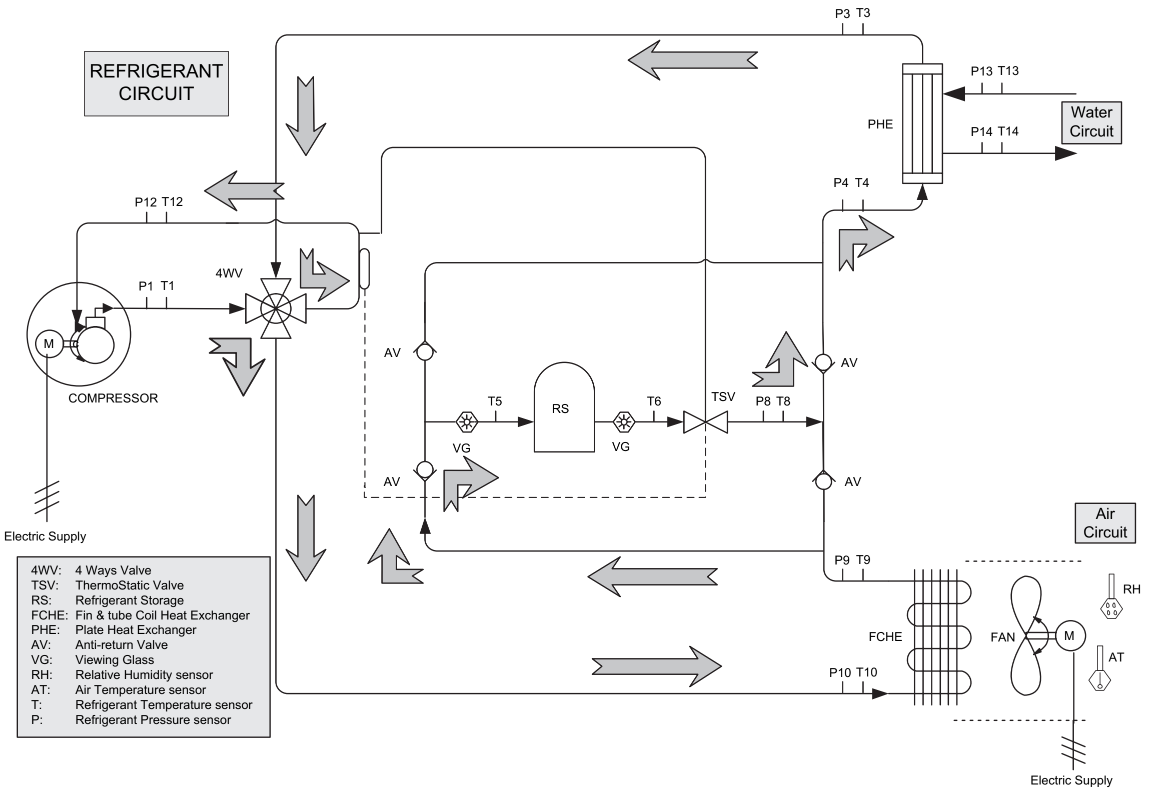

Fig. 4. Sketch of the heat pump tested and direction of the refrigerant flow.

the viewing glasses. The temperatures, pressures and mass flow rates in all loops of the experimental test rig were controlled and recorded automatically by a computer.

5. Modelling procedure

The correlations described above have been included in the global model of the ART© [1] code and are studied below using extensive collections of experimental results [2]. As mentioned before, the model incorporates several submodels for the integral components of the heat pump, compressors, brazed plate heat exchangers, finned tube heat exchangers, expansion devices, and connecting tubes. It is able to characterise the behaviour of each heat pump component, thus allowing for their separate examination and analysis. It enables the user to undertake a comparison between the experimental measurements and results calculated by each submodel separately. In consequence, this allows the user to check the influence of the condensation models on the model behaviour. The model does not consider wet wall conditions in the de-superheating region. Its effect on the condensation temperature calculation is almost negligible. A variation of 50% in the heat transfer coefficient yields a difference of a few hundredths of a degree.

For the sake of comparison, it has been necessary to run the model with the same conditions as in the experiment. The following considerations regarding the modelling procedure have to be taken into account.

- The refrigerant inlet conditions in the condenser are determined by the outlet conditions of the refrigerant discharged by the compressor. Therefore, it is necessary to know the exact working conditions of the compressor. Firstly, the correct suction conditions must be provided, i.e. the experiment evaporation temperature, the superheating, and a suitable model for the compressor. This is carried out taking into account both volumetric and isentropic efficiencies obtained from the experimental results displayed in Fig. 5. Secondly, the experiment subcooling is imposed to guarantee convergence.
- The same outlet temperature at the evaporator is obtained as in the experiment if the evaporator is modelled as an infinite-surface heat exchanger. The outlet temperature of the refrigerant is conditioned by the secondary fluid (water) outlet temperature which has been fixed at a value equal to the evaporation temperature measured in the experiments.

[1] Advanced Refrigeration Technologies.

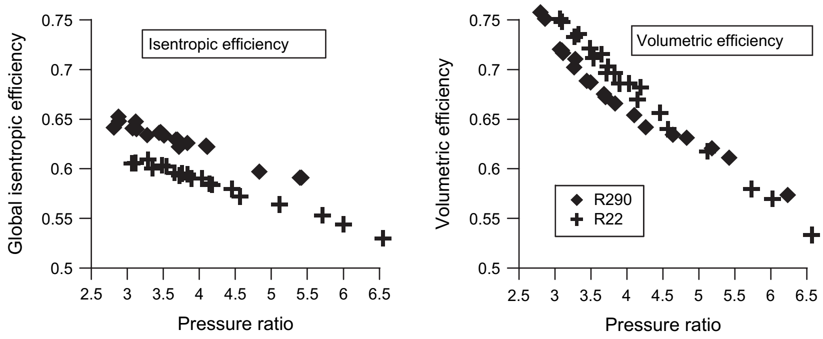

Fig. 5. Volumetric and isentropic efficiencies of the tested compressor.

- Refrigerant mass flow rate is set to the same value as in the experiment.
- Inlet conditions of the secondary fluid (i.e. air from the climate chamber) are also needed. Air inlet conditions (pressure, temperature, humidity, and mass flow rate of air) at the condenser are therefore imposed equal to the data measured under controlled weather conditions during the tests.

6. Experimental results versus calculated results

Several measurements were carried out, some with R-22 and others with R-290. Only the experimental data corresponding to the condensation temperature have been considered throughout this analysis. The same test cases performed experimentally have been studied following the numerical procedure described above, where the condensation correlations have been implemented. The experimental results have not been compared with the results provided by the Cavallini–Zecchin correlation. This has not been implemented in the code as it is only for the annular regime and was developed for a smaller data bank than that used for the Cavallini et al. correlation which covers several regimes and far more refrigerants. The experimental results and the calculated values for the different correlations have been displayed in Figs. 6–8. Lines corresponding to ±5% differences have been depicted only for the sake of comparison and clarity.

In all cases, certain agreement is encountered between the experiments and the computational results. Differences of only two or three degrees have been found. In general, similar tendencies and behaviours are shown for both refrigerants. In the AHPU heat exchanger, working with R-290, the mass velocity ranges from 55 to 67 kg/m^2s and the heat flux from 1500 to 6000 W/m^2. The values of the condensation pressures range from 13.5 to 19.35 bar. With the RHPU heat exchanger, the mass velocity of R-290 is higher than in previous cases, ranging from 85 to 95 kg/m^2, and pressure varies from 14.27 to 20 bar. The heat flux has values from 2000 to 8000 W/m^2. Regarding the R-22 cases the values of pressure are slightly higher, going from 15 to 23 bar. The mass velocity has greater values than in the propane case, varying from 150 to 175 kg/m^2s, and the heat fluxes range from 3000 to 9000 W/m^2.

With respect to the regimes encountered during the calculations, the Cavallini et al. correlation identifies the flow as stratified-wavy. Only in some cases, when almost all the

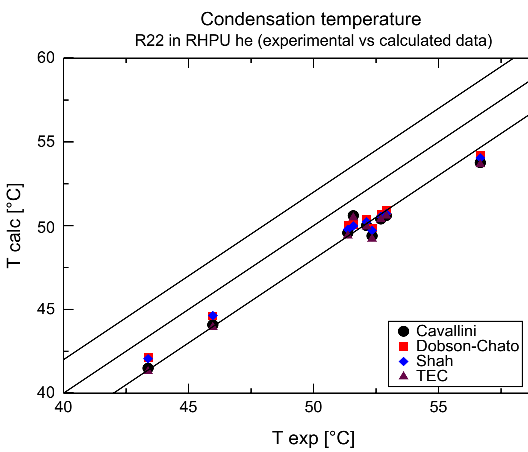

Fig. 6. Experimental results versus numerical results.

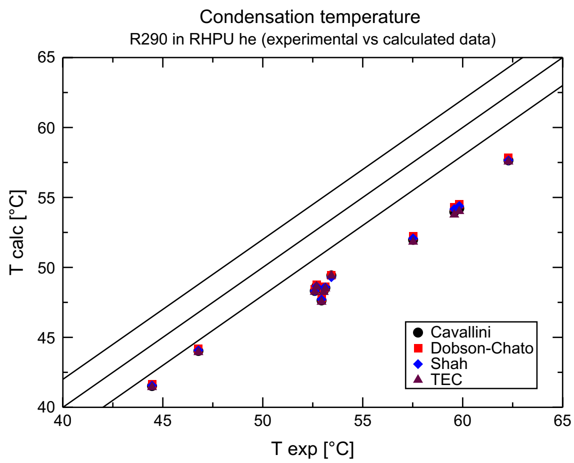

Fig. 7. Experimental results versus numerical results.

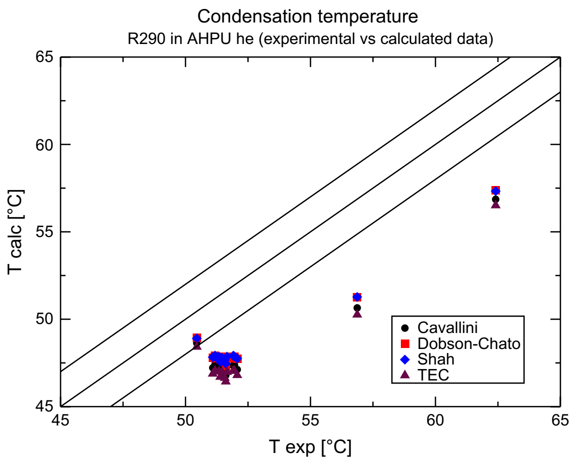

Fig. 8. Experimental results versus numerical results.

vapour has condensed the regime changes to stratified-slug. From Dobson and Chato's point of view, the flow is annular and the expression used for the heat transfer evaluation is that proposed for this regime. This happens for both refrigerants. With the Thome et al. correlation, the regime encountered is predominantly stratified-wavy, but mist flow can also been found at mass flow rates greater than 95 kg/m^2s for R-290 and greater than 178 kg/m^2s for R-22, where high vapour qualities are present.

7. Conclusions

In this paper, the assessment of a refrigeration cycle model has been carried out by using the condensation temperature as a design parameter. This is justified by the greater sensitivity of this parameter to changes in the heat transfer coefficient than other cycle parameters such as the heating capacity or the COP. A change in the heat transfer coefficient of 50% leads to variations in the condensation temperature greater than 6% when the heating capacity and the COP vary less than 2% or 5%, respectively, in most cases.

- Under these considerations, both the methodology followed in the numerical study and the experimental facility used for the model validation have been presented.
- Various correlations for the estimation of the condensation heat transfer coefficient have been introduced and contrasted to show their main differences. They have been used to model the refrigeration cycle and compare the experimental data with those provided by the numerical analysis of the experiments.
- The methodology used in the modelling has been described and the local variation of the friction factor and the heat transfer coefficients taken into account.
- The results shown in the paper correspond to two geometries and two different refrigerants.

Thus, it can be concluded that this model characterises the condensation process quite well when analysing refrigeration cycles with fin and tube coils and when proper condensation correlations are taken into account. These correlations provide satisfactory results when they are used in real problems and in the analysis of real heat exchangers with different conditions from those for which the correlations were developed. This work confirms that the use of the condensation temperature as a design parameter is an interesting proposition.

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