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A study on the fluid refrigerant charge in a Two-Phase Mechanically Pumped Loop system using R134a and R1234yf

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Abstract

Heat management by means of two-phase mechanically pumped loop (TMPL) possesses promising advantages such as long-distance heat transport, high heat density and good temperature control. Knowledge of the fluid charge has great importance for TMPL system efficiency. The present work is a theoretical/experimental study on the refrigerant charge in TMPL system using R134a and R1234yf refrigerants. The mass of refrigerant in the TMPL system was measured for several cooling capacities. The effects of vapor quality in the evaporator (varying from 0.1 to 1), mass flux ($300\text{kg.s}^{-1}\text{m}^{-2}$ and $400\text{kg.s}^{-1}\text{m}^{-2}$), and saturation temperature (25°C and 30°C) on refrigerant charge in the TMPL were experimentally investigated. About 70 experimental points were analyzed and compared with a theoretical model employing eight different prediction tools for the determination of the void fraction. The best results were obtained by Zivi and Hughmark correlations with mean absolute errors of 12% and 13%, respectively.

Key words: TMPL, R1234yf, R134a, void fraction, refrigerant charge.

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evaporator remains constant [5] –within the limits of the temperature drop that may be induced by the pressure drop-. This makes TMPL systems a valuable option for applications with high heat fluxes in small surface areas or systems that require very stable conditions, that is, minimum oscillations and small temperature changes.

Terpstra [6] studied a TMPL system working with R134a. This refrigerant has a high global warming potential (GWP=1340 kg eq. CO₂); it does not satisfy the Directive 517/2014 to reduce the use of high-GWP greenhouse gases in the European area to limit global climate change in accordance with the objectives marked by the EU Research and Innovation program Horizon 2020 [7]. R1234yf has emerged as a replacement for R134a [8]. R1234yf has GWP less than 1 kg eq. CO₂ and has similar thermodynamic properties as R134a [9]. R1234yf exhibits a low GWP because this fluid contains one C=C double bond in molecular structure, which makes the compound reactive towards OH radicals in the troposphere, decreasing the lifetime and global warming potential [10]. The drop-in performance of R134a and R1234yf was evaluated in various systems: mobile air-conditioner [11-13], domestic refrigerator [14], residential heat pump water [15], residential heating split air [16] and ejector refrigeration cycle [17]. These studies have shown that R1234yf can be substituted to R134a with comparable performance and no substantial modifications to the original system. Up to the Authors' knowledge, no study focused on the replacement of R134a by R1234yf in TPML, which is the focus of the present work.

To replace R134a, it is very important to know the R1234yf thermohydraulic behavior of two-phase flow because it has an important role in design and optimization of cooling and refrigeration systems (evaporator, condenser and intermediate heat exchanger). Therefore, two-phase flow characteristics of R1234yf in smooth tubes were studied. Del Col et al. [18], [19] measured condensation and boiling heat transfer coefficients of R1234yf and R134a at mass fluxes ranging from 200 kg.s⁻¹.m⁻² to 1000 kg.s⁻¹.m⁻² and saturation temperatures ranging from 30 °C to 40 °C. They found lower values of condensation heat transfer coefficients compared to R134a for the same experimental conditions and no significant differences between the flow boiling performance of R1234yf and R134a. Lu et al. [20] also obtained the same conclusion for convective boiling heat transfer performance for a 3.9 mm smooth diameter tube and saturation temperature of 10 °C. Saitoh et al. [21] and Li et al. [22] investigated the flow boiling of R1234yf, R134a and R32 inside a smooth horizontal tube with diameter of 2 mm. The heat

transfer coefficient and pressure drop were measured at heat fluxes up to 24 kW.m^{-2} , mass fluxes between $100 \text{ kg.s}^{-1}\text{m}^{-2}$ and $400 \text{ kg.s}^{-1}\text{m}^{-2}$, and at a saturation temperature of 15°C . The results showed that the heat transfer coefficient of R1234yf was almost the same as that for R134a and lower than that for R32. Wang et al. [23] studied the heat transfer during condensation of R1234yf, R124a and R32 in a horizontal tube with diameter of 4 mm at a mass flux ranging from $100 \text{ kg.s}^{-1}\text{m}^{-2}$ to $400 \text{ kg.s}^{-1}\text{m}^{-2}$, and different saturation temperatures (40°C , 45°C and 50°C). In addition, the pressure drop was measured and related to the observed flow patterns. Padilla et al. [24] visualized two-phase flow regimes for R1234yf and R134a in a straight tube whose inner diameter was 6.7 mm. These flow regimes were satisfactorily predicted by Wojtan et al. [25] flow pattern map. The two-phase pressure drop was also measured for a tube diameter varying from 7.9 mm to 10.9 mm and a saturation temperature ranging from 5°C to 21°C . Conventional prediction tools of pressure drop used for R134a are suitable for R1234yf. Finally, Sempértegui-Tapia and Ribatski [26], [27] developed correlations to predict the heat transfer and pressure drop during evaporation of R134a, R1234ze(E), R1234yf and R600a in tube of 1.1 mm diameter. This brief summary underlines that the behavior of R1234yf was mostly characterized from the viewpoints of two-phase heat transfer, pressure drop and flow patterns, but not from that of void fraction in systems using R1234yf, while this parameter is directly related to the fluid charge.

The refrigerant charge is the total mass of fluid contained in a refrigeration system. The distribution of the refrigerant mass into the components of the system represents the refrigerant charge that is placed before starting the system. The equipment has a different performance when charged with more or less fluid [28]. This means that there is an optimum fluid amount that maximizes the heat transport capacity and improves the stability of system [29]. The determination of void fraction is one of the main issues that must be addressed for developing an accurate model on the overall refrigerant charge.

As a conclusion, the designers of TMPL systems should shift to the use of the next generation of refrigerants with low GWP as R1234yf, like such a shift was done for other applications (essentially refrigeration and air-conditioning) with success; The calculation of void fraction and mass charge of R1234yf was not studied in the literature. For this reason, the present work is an experimental study on the refrigerant charge in TMPL system using the R134a and R1234yf fluids. Additionally, this study presents a model to calculate the refrigerant mass charge in TMPL

system and a comparison with the obtained experimental results. Besides, the model is useful to assess the performance the several prediction tools for the void fraction in the evaporator and condenser of the TMPL system using R1234yf and R134a.

2. Experimental setup

2.1. Experimental apparatus

Fig. 1 shows a schematic of the experimental setup. It mainly consists in a closed refrigerant loop that includes a magnetically coupled gear pump (series GJ of MICROPUMP INC). The refrigerant flow rate is controlled by adjusting the frequency of the magnetic gear pump (drive model Eagle of MICROPUMP INC). The fluid flows into the mass flow meter type micro-turbine (KOBOLD, model DPM-1130) and is pumped to the evaporator, made of a copper tube heated by an electrical resistance. The electrical power is adjusted to obtain desired values of vapor quality. The outlet of the evaporator is connected to the inlet of the condenser using a 1.5 m adiabatic pipe that keeps the vapor quality constant as flashing is negligible in such conditions (this can for instance be inferred from the discussion by Revellin et al. [30]). Finally, the fluid is condensed and subcooled and returns at the pump inlet. The condenser is a coaxial tubes heat exchanger, where the refrigerant flows in the inner tube in counter flow with cold water in the annular section. The cold water loop is supplied with water by a chiller with a control of the water temperature at the inlet of the heat exchanger. The system pressure can primarily be set owing to the condenser temperature and can be fine-tuned with the help of a small refrigerant tank that is connected to the refrigerant loop and placed in a temperature- controlled bath with a second water loop. A scale is placed under the refrigerant tank to measure the amount of refrigerant mass and to determine the remaining refrigerant mass in the rest of the system. The whole setup has internal diameter tube of 4.77 mm and is insulated to minimize thermal loss (<5% heat losses were observed). The refrigerant temperatures (with T-type thermocouples) and pressures (NOVUS, manometer model NP460) are measured in several locations throughout the circuit, in order to determine the fluid thermodynamic state. For the data reduction the properties of the refrigerant were calculated from Engineering Equation Solver (EES, developed by F-Chart Software).

Fig. 1. Schematic of the experimental setup.

2.2. Data reduction

The mass flux is calculated as:

$$G = \frac{4\dot{m}_r}{\pi D} \quad (1)$$

where $\dot{m}_r$ the refrigerant mass flow rate and D is the internal cross-sectional area of the tube. The vapor quality at the outlet of the evaporator can be determined through a simple energy balance.

$$x = \frac{1}{i_{lv}} \left(\frac{Q}{\dot{m}_r} - (i_l - i_{in}) \right) \quad (2)$$

where i_{in} is the enthalpy of the liquid at the inlet of the evaporator, i_l is the enthalpy of the saturated liquid and i_{lv} is the latent heat of vaporization considering the saturation temperature at the outlet of the evaporator. Q is the heat rate in the evaporator:

$$Q = EI \quad (3)$$

where E and I are respectively the voltage and current electrically applied in the preheater.

2.3. Uncertainty analysis

The uncertainty analysis was carried out to evaluate the quality and the repeatability of the experimental data following the procedure exposed by Moffat [31]. The known accuracy of instruments (refrigerant and water mass flow meters, current sensor, voltage sensor, pressure sensors, and scale) coupled with calibrated accuracy of thermocouples was used to evaluate the uncertainty in the calculated parameters as vapor quality and mass flux. The uncertainties are summarized in Table 1.

Table 1. Uncertainties for parameters

Parameter	Uncertainty
Internal diameter	±0.4%
Tube Length	±0.1%
Refrigerant mass flow rate	±2.5%
Water mass flow rate	±1.7 %
Electric current	±1.5 %
Electric voltage	±1.8 %
Pressure	±0.2 bar
Temperature	±0.5 °C
Refrigerant charge	±0.5 g
Refrigerant mass flux	±3.4%
Vapor quality	±4.8 %

2.4. Experimental conditions

The experimental tests were performed for R134a and R1234yf (99.8% purity). The mass flux were set equal to 300 or 400 $\text{kg.s}^{-1}\text{m}^{-2}$, the saturation temperature were set equal to 25 or 30°C at the outlet at evaporator for both fluids, and the vapor quality ranged from 0.1 to 1.

3. Refrigerant charge model

3.1. Basics Concepts

A mathematical model for the prediction of the mass of fluid in the experimental apparatus was developed. It is based on the method of Porto et al. [28] and Otaki [32] for the prediction of the fluid inventory of refrigerating systems. The inputs from the refrigerant loop are the mass flow, the vapor quality at the test section, the subcooled and saturation temperatures. It is also necessary to know the mass flow rate and the water temperature at the inlet and outlet of the condenser. Other important parameters of the model are the geometric characteristics (diameter and length) of the tubes in each part of the experimental setup. In the model the pressure drop in the system is neglected, the error of this consideration is small and is compensated with the selection of the heat transfer coefficient of fluid and water.

The first step is to calculate the mass of subcooled liquid ($M_{l,sc}$) in the section that connects the outlet of the condenser to the inlet of the evaporator.

$$M_{l,sc} = \rho_{l,sc} V_{l,sc} \quad (4)$$

where $V_{l,sc}$ and $\rho_{l,sc}$ are respectively the volume of the section mentioned above and the refrigerant density at the subcooled temperature.

The second step is to calculate the mass of refrigerant within the evaporator. Figure 2 shows the evaporator is divided into two regions. The liquid zone is considered as a unique control volume. In the two-phase region, the density of refrigerant varies significantly in a non-linear way. Thus, to obtain accurate results, the evaporation region should be divided into N control volumes (mesh independence was found for N greater than 150), considering that the vapor quality varies linearly with length. The lengths of the liquid ($L_{l,h}$) and two-phase ($L_{b,h}$) regions in the evaporator referred in Fig. 2 are found by considering the heat flux to be constant in the evaporator and applying the first law of thermodynamics:

$$L_{l,h} = \left(\frac{i_l - i_{l,sc}}{i_x - i_{l,sc}} \right) L_h \quad (5)$$

Fig. 2. Schematic of the evaporator.

Then, the liquid mass is calculated by multiplying the refrigerant density (using the average temperature $((T_{l,sc} + T_{sat})/2)$) by the liquid region volume.

$$M_{l,h} = \frac{\pi}{4} \rho_{l,h} L_{l,h} D^2 \quad (6)$$

The mass in the two-phase zone ($M_{b,h}$) is the summation of mass in each control volume, which is calculated by the product of the two-phase local density (ρ_i) to the fix volume of the control volume (ΔV_i).

$$M_{b,h} = \sum_{i=1}^N (\rho_i \Delta V_i) \quad (7)$$

$$\rho_i = \rho_v \alpha_i + \rho_l (1 - \alpha_i) \quad (8)$$

$$\Delta V_i = \frac{\pi}{4N} L_{b,h} D^2 \quad (9)$$

where ρ_l and ρ_v are respectively the liquid and vapor densities at saturation state, and α_i is the void fraction of the control volume, which is function of the vapor quality. $L_{b,h}$ is the evaporation region length, which is obtained by subtracting the liquid region length from the total evaporator length.

The third step is to calculate the mass of two-phase flow contained in the adiabatic section between the outlet of the evaporator and the inlet of the condenser. The vapor quality is constant and the mass is calculated as follows:

$$M_b = \frac{\pi}{4} L_b D^2 (\rho_v \alpha_x + \rho_l (1 - \alpha_x)) \quad (10)$$

where $L_b = 1.5 \text{ m}$ is the length of the section and α_x is the void fraction at the outlet of the pre-heater and considered as constant. Revellin et al. [30] discussed the channel length from which the flashing effect should be taken into account, and calculated to R134a for a mass flux of $500 \text{ kg.s}^{-1} \text{ m}^{-2}$, diameter of 6 mm, and vapor quality of 0.7. The length was between 8 m and 21 m in saturation temperatures of 10°C and 30°C , respectively.

The fourth step are dedicated to the calculation of the liquid and two-phase mass in the condenser. In this heat exchanger; refrigerant flow is considered as counter current as shown in Fig. 3. The energy balance between the refrigerant side and the water side in the subcooling section results in Eqs. (11)-(13). The conduction resistance is neglected.

Fig. 3. Temperature distributions in of the condenser.

$$\dot{m}_r(i_l - i_{l,sc}) = \dot{m}_w C_{p,w}(T_w - T_{w,in}) = \pi L_{l,c} U_{l,c} \Delta T_{l,c} D \quad (11)$$

$$U_{l,c} = \left(\frac{1}{h_l} + \frac{1}{h_w} \right) \quad (12)$$

$$\Delta T_{l,c} = \frac{(T_{sat} - T_w) - (T_{l,sc} - T_{w,in})}{\ln \left(\frac{T_{sat} - T_w}{T_{l,sc} - T_{w,in}} \right)} \quad (13)$$

$\dot{m}_r$ is the refrigerant mass flow rate, and $\dot{m}_w$ and $C_{p,w}$ are the mass flow rate and the specific heat of the water, respectively. Additionally, $L_{l,c}$ and $\Delta T_{l,c}$ are the subcooling length and the logarithmic mean temperature difference, respectively. $U_{l,c}$ is the overall heat transfer coefficient, which is a function of h_l and h_w , the heat transfer coefficients of the liquid refrigerant and water, respectively. $C_{p,w}$ varies slightly and can be calculated at the mean water temperature $((T_{w,in} + T_{w,out})/2)$. The subcooling length ($L_{l,c}$) can be calculated by Eq. 11. Then, the mass of the subcooled liquid contained into the condenser is calculated by using the liquid refrigerant density (at the mean temperature $((T_{l,sc} + T_{sat})/2)$).

$$M_{l,c} = \frac{\pi}{4} \rho_{l,c} L_{l,c} D^2 \quad (14)$$

Finally, the condensation region length ($L_{b,c}$) is obtained by subtracting the liquid region length from the total condenser length. The procedure for calculating the refrigerant mass in a condensing region consists in dividing the volume into N control volumes as shown in Fig. 4 for fixed difference enthalpy, Δi .

Fig. 4. Control volume in condensation region.

Equations (15)-(18) express the energy balance that allow to calculate the length of the control volume (L_j). This energy balance depends on the internal heat transfer coefficient that varies non-linearly.

$$\dot{m}_r(\Delta i) = \dot{m}_w C_{p,w} (T_{w,j} - T_{w,j+1}) = \pi L_j U_j \Delta T_j D \quad (15)$$

$$U_j = \left(\frac{1}{h_{b,j}} + \frac{1}{h_w} \right) \quad (16)$$

$$\Delta T_{b,c} = \frac{(T_{sat} - T_{w,j}) - (T_{sat} - T_{w,j+1})}{\ln \frac{(T_{sat} - T_{w,j})}{(T_{sat} - T_{w,j+1})}} \quad (17)$$

$$\Delta i = \frac{i_x - i_l}{N} \quad (18)$$

The heat transfer coefficient for water in Eq.(16) depends of the tube wall temperature, which is unknown. The tube wall temperature distribution is calculated by iteration process of Eq. (15)-(17) until Eq. (19) is satisfied

$$L_{b,c} \approx \sum_{j=1}^N L_j \quad (19)$$

Thus, the mass in the condensation section is calculated as follows:

$$M_{b,c} = \frac{\pi}{4} D^2 \sum_{j=1}^N (\rho_v \alpha_j + \rho_l (1 - \alpha_j)) L_j \quad (20)$$

The total refrigerant charge in the system is:

$$M = M_{lsc} + M_{l,h} + M_{b,h} + M_b + M_{b,c} + M_{l,c} \quad (21)$$

The liquid heat transfer coefficient was computed using the Gnielinski [33] correlation for both fluids.

$$h_l = \frac{(f/8)(Re_l - 1000) Pr_l}{1 + 12.7(f/8)^{0.5} (Pr_l^{2/3} - 1)} \left(\frac{\mu_l}{\mu_{wall}} \right)^{0.11} \frac{k_l}{D} \quad (22)$$

where Pr_l , k_l and μ_l are the Prandtl number, thermal conductivity and liquid viscosity of fluid, respectively. μ_{wall} is liquid viscosity evaluated in tube wall temperature. f is the friction factor. For smooth pipes, it is calculated as a function of Reynolds number:

$$f = [1.82 \ln(Re_l) - 1.64]^{-2} \quad 2500 \leq Re_l \leq 10^6 \quad (23)$$

where

$$Re_l = \frac{GD}{\mu_l} \quad (24)$$

The heat transfer coefficient for refrigerant during condensation was calculated using Shah [34] correlation.

$$h_{cond} = h_I \quad \text{if} \quad J_g \geq 0.98(Z + 0.263)^{-0.62} \quad (25)$$

$$h_{cond} = h_{III} \quad \text{if} \quad J_g \leq 0.95(1.254 + 2.27Z^{1.249})^{-1} \quad (26)$$

$$h_{cond} = h_I + h_{III} \quad \text{Otherwise} \quad (27)$$

where

$$h_I = 0.023 Re_l^{0.8} Pr_l^{0.4} \frac{k_l}{D} (1-x)^{0.8} \left(1 + \frac{3.8}{Z^{0.95}}\right) \left(\frac{\mu_l}{14\mu_v}\right)^{(0.0058 + 0.557 p_{red})} \quad (28)$$

$$h_{III} = 1.32 \left[\frac{\rho_l(\rho_l - \rho_v) g k_l^3}{(1-x)\mu_l^2 Re_l} \right]^{\frac{1}{3}} \quad (29)$$

$$J_g = \frac{xG}{(gD\rho_v(\rho_l - \rho_v))^{0.5}} \quad (30)$$

$$Z = \left(\frac{1}{x} - 1\right)^{0.8} p_{red}^{0.4} \quad (31)$$

where p_{red} is the reduced pressure of fluid and g is the gravity.

3.2. Void fraction prediction tools

The estimation of refrigerant mass in two-phase flow inside condenser and evaporator depends on void fraction. Thus, this parameter is the most important parameters of the whole system in order to obtain accurate results. There are several models (or families of models) that can be suitable for evaluating the void fraction in two-phase flow systems; they can be classified as follows:

- The homogeneous model is derived by treating the gas or vapor phase and the liquid phase as a homogeneous mixture flowing with the same velocity. This major assumption implies that:

$$\alpha_{hom} = \left[1 + \left(\frac{1-x}{x} \right) \left(\frac{\rho_v}{\rho_l} \right) \right]^{-1} \quad (32)$$

- The slip ratio (S) defined as the ratio of the gas velocity to the phase velocity, can be introduced. A variety of models or correlations for S are available in the literature. With the slip ratio, the void fraction can be expressed as:

$$\alpha = \left[1 + \left(\frac{1-x}{x} \right) \left(\frac{\rho_v}{\rho_l} \right) S \right]^{-1} \quad (33)$$

- Another possible way to correct the homogeneous model is to multiply the homogeneous void fraction by a correction factor K (i.e. $\alpha = K\alpha_{hom}$) that is also determined by correlations of models.
- The drift flux model expresses the void fraction as a function of the so-called “distribution parameter” C_0 , the drift velocity u_{gm} , the superficial gas velocity u_{sg} , and two-phase mixture velocity u_m . The parameter C_0 is predicted by a specific model or correlation.

$$\alpha = \frac{u_{sg}}{C_0 u_m + u_{gm}} \quad (34)$$

- Finally, several miscellaneous other prediction tools are available in the literature: they often include the Lockhart-Martinelli parameter (X), and various dimensionless numbers like the Reynold number (Re), the Froude number (Fr), etc.

Table 2 shows a set of classical available tools that were used to calculate the void fraction in the present work. These tools were used for evaporation and condensation in the same way, to simplify the calculation routine.

Table 2. Void fraction prediction tools.

Authors	Type
Homogeneous model	-
Zivi [35]	Slip ratio
Hughmark [36]	$K\alpha_{hom}$
Premoli et al. [37]	Slip ratio
Domanski and Didion [38]	Miscellaneous
Steiner [39]	Drift flux
Xu and Fang [40]	Miscellaneous
Kanizawa and Ribatski [41]	Miscellaneous

4. Experimental results

This section reports the experimental results of refrigerant mass charge as a function of the operating test conditions. The influence of different parameters on the refrigerant charge was investigated. The mass flux, the vapor quality at the outlet of the evaporator and the saturation temperatures were varied. A total of 30 and 40 experimental tests were performed for R134a and R1234yf respectively.

4.1. Effect of mass flux

Figure 5 reports the effect of the mass flux on the refrigerant charge as a function of the vapor quality at the outlet of the evaporator, at two different saturation temperatures. Fig. 5 shows that for vapor quality between 0.1 and 0.5, the refrigerant charge is slightly affected when the mass flux increases. It is observed for both fluids and any saturation temperature. This trend can be explained considering the slip ratio that controls the void fraction and, consequently, the refrigerant charge. At low vapor qualities, the mass flux does not influence the slip ratio due to vapor bubble distribution. Additionally, when the vapor quality exceeds 0.5, the stratified flow or annular flow can proceed. Wojtan et al. [25] showed that for the same quality, a transition of stratified flow to annular flow can occur when the mass flux increases in the range studied in this work. These flow patterns result in slip ratios quite different, so that, for the same vapor quality, the void fraction is greater by increasing the mass flux. This can explain the difference between curve of refrigerant charge for $x > 0.5$ in Fig. 5.

Fig. 5. Effect of mass flux (G in $\text{kg.s}^{-1}\text{m}^{-2}$) on the refrigerant charge at two different saturation temperatures.

4.2. Effect of saturation temperature

The refrigerant charge is plotted against vapor quality at the outlet of the evaporator for two saturation temperatures and for two mass fluxes on Fig. 6. The refrigerant charge was slightly higher at a saturation temperature of 30°C. The influence of saturation temperature on thermophysical properties for the two refrigerants is shown in Table 3. When the saturation temperature decreases, the reduction of the density ratio (ρ_l/ρ_v) induces a decrease of the vapor velocity, the slip ratio and void fraction [42]. As a result, the refrigerant charge increases.

Table 3. Thermodynamic properties of R134a and R1234yf at similar operating conditions.

Fluid	T_{sat} °C	P_{sat} bar	σ mN.m^{-1}	ρ_l kg.m^{-3}	ρ_v	ρ_l/ρ_v -	μ_l mPa.s	μ_v	μ_l/μ_v -
R134a	25	6.62	8.05	1207.3	32.20	37.49	0.1953	0.0117	16.69
	30	7.67	7.40	1188.0	37.37	31.79	0.1835	0.0119	15.42
R1234yf	25	6.80	6.19	1092.5	37.76	28.93	0.1454	0.0125	11.63
	30	7.80	5.58	1073.9	43.54	24.66	0.1366	0.0128	10.67

Fig. 6. Effect of saturation temperature on the refrigerant charge at two different mass fluxes.

4.3. Effect of refrigerant nature

Figures 5 and 6 are useful to capture the effect of fluid nature. To explain the trend is necessary to consider the difference between the thermodynamic properties of both fluids shown in Table 3. It can be seen from these figures that R134a has a lower refrigerant charge than R1234yf at high vapor qualities. This can be explained by the higher saturation pressure and vapor density of R1234yf with respect to R134a, which causes the vapor phase to flow at a lower velocity and the void fraction to be lowered. At low vapor quality when the liquid phase is dominant, the trend of refrigerant charge curve is inverse; R1234yf has a lower refrigerant charge than R134a because R1234yf has a lower liquid density than R134a, an important parameter as can be caught from Eq. (35).

4.4. Cooling capacity of TMPL system

The electric power applied in the evaporator of system is the cooling capacity of TMPL system. Fig. 7 presents the cooling capacity in function of vapor quality for all test conditions and both refrigerants. When the mass flux increases by around 33%, the cooling capacity increases by 40% for R134a and 38% for R1234yf. When the vapor quality increases, the cooling capacity of system using R134a is higher than that for R1234yf; because R134a has higher vaporization enthalpy than R1234yf. Finally, the saturation temperature does not influence the cooling capacity.

Fig. 7. Cooling capacity of TMLP system vs vapor quality.

Figure 8 shows the cooling capacity as a function of the refrigerant charge for two different mass fluxes. One observes that when increasing the mass flux, the operating range of fluid charge in the TPML is getting larger (e.g. [1000 g – 1200 g] for R1234yf, $T = 30^{\circ}\text{C}$ and $G = 300 \text{ kg.m}^2.\text{s}^{-1}$ against [500 g – 1300 g] for the same fluid and temperature but at $G = 400 \text{ kg.m}^2.\text{s}^{-1}$). This range of refrigerant charge should be considered in the design of the accumulator (refrigerant tank in Fig. 1), that is used to control the mass of refrigerant into the loop in order to keep to the desired saturation temperature when the heat load changes.

Fig. 8. Cooling capacity of TMLP system vs refrigerant charge.

5. Comparison with available models

The experimental data were compared with the results of the model of refrigerant charge presented in section 3, with the different prediction tools for void fraction available in the literature described in section 3.2. The comparison is based on the Mean Absolute Relative Deviation that is defined as:

$$MARD = \frac{1}{n} \sum \left| \frac{M_{pred} - M_{exp}}{M_{exp}} \right| * 100 \quad (35)$$

A summary of the comparisons according to the prediction tools used is given in Table 4. Bold numbers correspond to the four best predictions for each case. For R134a, the best prediction tools are proposed by Zivi [35] Hughmark [36], Xu and Fang [40], and Kanizawa and Ribatski [41] with MARD less than 13%. For R1234yf, the void fraction prediction tools developed by Premoli et al. [37] and Steiner [39] replace the last two tools in the previous top four with all MARD less than 14%. Finally, considering all experimental data, the best predictions tools are those of Zivi [35] Hughmark [36], Premoli et al. [37], and Kanizawa and Ribatski [41], with MARDs between 12.0% and 13.6%. The difference between the tools of Zivi [35] and Hughmark [36] is 0.6%, which is significant with respect to the uncertainty that was given previously in Table 1 (± 0.5 g for the refrigerant charge, i.e. less than 0.2% of relative uncertainty).

Table 4. MARD values for experimental refrigerant charge data.

Model	R134a data	R1234yf data	All data
Homogeneous model	24.20	23.64	23.92
Zivi [35]	12.17	11.86	12.02
Hughmark [36]	13.35	12.30	12.83
Premoli et al. [37]	13.57	13.20	13.39
Domanski and Didion [38]	16.99	18.57	17.78
Steiner [39]	13.66	13.64	13.65
Xu and Fang [40]	12.95	15.04	14.00
Kanizawa and Ribatski [41]	12.60	14.47	13.54

Figure 9 presents graphically the comparison between experimental mass charge and those calculated by the model including the four best prediction tools mentioned above, along with $\pm 15\%$ error bands. The comparison of the experimental data shows a small over-prediction for low mass charge (high vapor qualities), whereas for high mass charge, the results rather show an

under-prediction. In the range of refrigerant charge of [800 g – 1200 g], a great discrepancy between experimental and predicted data is observed. It is due to the fact that the void fraction grows quickly in the low vapor qualities, which makes it difficult to predict with the correlations.

Fig. 9. Comparison of experimental data with correlations from the literature.

Figure 10 shows the cumulative distribution of experimental data in the maximum error for each correlation of top four. For example, all correlations present approximately 100% of experimental data for both refrigerants within a $\pm 30\%$ error band. Applying the Zivi [35] correlation in the refrigerant charge model allows to predicting 70% of the experimental data within a $\pm 15\%$ error band. The use of Kanizawa and Ribatski [41] correlation results in 60% of data within a $\pm 15\%$ error band.

Fig. 10. Percentage of experimental data as function of maximum error.

6. Conclusions

An experimental analysis of the refrigerant charge of a TMPL system for R1234yf and R134a under several operating conditions was reported in this article. The experimental data were also compared to a model using different void fraction prediction tools from the literature. The main findings are summarized below:

- The effects of vapor quality, saturation temperature, and mass flux on refrigerant charge for R1234yf and R134a were analyzed. R134a results show a lower refrigerant charge than R1234yf in conditions used in evaporator and condenser (high vapor qualities). This difference can be between 2% e 16%, depending on experimental conditions. Thus, the size of TMPL system does not need to be changed considerably when replacing R134a by R1234yf, if the operating conditions are selected correctly.
- R1234yf refrigerant stands as a good substitute to R134a for TMPLs, like it was found to be a good substitute for mobile air-conditioners, and recently, for other systems like domestic refrigerators and heat pumps. The capacity cooling of R1234yf is 20%-30%

lower than R134a, but its environmental advantages and new policies in the world concerning global warming make it be a suitable alternative. Moreover, the difference in cooling capacity can be compensated by slightly increasing in the mass flux.

- The experimental results of refrigerant charge were compared against a model including void fraction that was calculated by various classical models or correlations available in the open literature. The best prediction tools are, in order, those developed by Zivi [35] Hughmark [36], Premoli et al. [37], and Kanizawa and Ribatski [41], with prediction of the experimental data points with MARDs between 12.0% and 13.6%.

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Appendix

The experimental data are presented in Table A.1 and A.2.

Table A.1. Experimental data of R134a.

$T_{sat} (^{\circ}C)$	$G (kg.s^{-1}.m^{-2})$	$x (%)$	$Q (W)$	$M (g)$
25	300	100.0	1001.3	782.0
25	300	93.0	917.9	830.0
25	300	78.0	804.9	936.0
25	300	69.0	720.0	1040.0
25	300	53.0	568.9	1162.0
25	300	35.0	391.5	1254.0
25	300	22.0	277.4	1272.0
25	400	100.0	1392.0	400.0
25	400	73.0	1042.3	452.0
25	400	63.0	907.1	666.0
25	400	54.0	789.4	1024.0
25	400	40.0	612.2	1220.0
25	400	35.0	521.3	1272.0
25	400	25.0	406.9	1322.0
25	400	15.0	285.4	1376.0
30	300	100.0	985.6	1054.0
30	300	89.0	884.6	1104.0
30	300	76.0	755.9	1150.0
30	300	67.0	681.4	1198.0
30	300	58.0	596.1	1250.0
30	300	46.0	495.2	1310.0
30	300	35.0	388.1	1364.0
30	300	23.0	297.3	1416.0
30	400	100.0	1354.0	424.0

		0	4	
30	400	85.0	1170.8	474.0
30	400	78.0	1076.4	574.0
30	400	67.0	948.7	846.0
30	400	52.0	760.6	1156.0
30	400	45.0	665.8	1262.0
30	400	36.0	540.5	1326.0
30	400	26.0	421.2	1368.0
30	400	16.0	277.4	1416.0

Table A.2. Experimental data of R1234yf.

$T_{sat} (^{\circ}C)$	$G (kg.s^{-1}.m^{-2})$	$x (%)$	$Q (W)$	$M (g)$
25	300	100.0	832.1	924.6
25	300	93.0	796.6	964.7
25	300	82.0	701.7	995.9
25	300	75.0	645.2	1035.8
25	300	67.0	584.7	1075.7
25	300	60.0	538.2	1115.6
25	300	50.0	452.9	1155.8
25	300	40.0	374.4	1185.9
25	300	34.0	327.7	1200.9
25	300	27.0	277.4	1215.5
25	400	100.0	1129.2	440.0
25	400	88.0	1002.0	469.9

			3	
25	400	78.0	903.7	600.6
25	400	71.0	837.1	761.4
25	400	63.0	749.8	911.9
25	400	58.0	701.6	1012.4
25	400	47.0	577.1	1099.3
25	400	36.0	477.3	1154.6
25	400	26.0	370.9	1189.7
25	400	18.0	277.4	1225.6
30	300	100.0	820.8	1000.0
30	300	88.0	735.2	1050.0
30	300	77.0	665.0	1096.0
30	300	67.0	577.4	1142.0
30	300	54.0	479.1	1172.0

30	300	46.0	416.9	1186.0
30	300	37.0	351.4	1204.0
30	300	32.0	316.5	1212.0
30	300	28.0	285.4	1216.0
30	400	100.0	1105.9	500.0
30	400	95.0	1069.9	545.0
30	400	89.0	1001.3	603.0
30	400	82.0	923.1	679.0
30	400	76.0	865.6	765.0
30	400	65.0	747.4	981.0
30	400	50.0	599.4	1141.0
30	400	36.0	452.9	1241.0
30	400	23.0	324.0	1291.0

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Highlights

- An experimental study of mass charge for TMPL system using R134a and R1234yf
- The effect of major flow parameters on mass charge for TMPL systems was discussed
- The effect of void fraction correlations on the calculation of mass charge for TMPL system was evaluated