2018

# Charge Equation For Small Charge Hydrocarbon Based Commercial Refrigeration Appliances

Marcel van Beek
*Re/genT BV, Netherlands, The*, marcel.van.beek@re-gent.nl

Thijs van Gorp
*Re/genT, Netherlands, The*, thijs.van.gorp@re-gent.nl

# Charge Equation For Low Charge Hydrocarbon Based Commercial Refrigeration Appliances

Marcel van Beek[1], Thijs van Gorp[1]

[1]Re/genT,
Helmond, the Netherlands
Tel +31(0)492 476365, e-mail: marcel.van.beek@re-gent.nl

## ABSTRACT

To assist manufacturers in their early stage of appliance design (i.e. component selection) an easy to implement charge equation, for hydrocarbon refrigerants R-290 and R-600a, has been developed. The equation estimates the appliance total refrigerant charge based on the type of refrigerant, the operating conditions and the internal volume of the components of the cooling system.

This paper presents the charge equation developed and discusses the derivation, assumptions and underlying calculations in detail. It is shown that the influence of refrigerant mass flux cannot be neglected. However, a simple approximation of the effect of mass flux is shown to provide good results. The equation is validated based on the total refrigerant charge of 10 charge optimized glass door bottle coolers, varying in refrigerated volume from 50 to 1200 $dm^3$, having a cooling capacity between 50 and 1500 W and using hydrocarbon as the refrigerant. Validation showed an agreement within 15% between the calculated and the actual refrigerant charge of the appliances.

## 1. INTRODUCTION

The use of hydrocarbon refrigerants has become common practice for small commercial refrigeration appliances. As the charge of hydrocarbon is limited by safety regulations, refrigerant charge has become a critical design parameter for these appliances. Charge equations have been developed before; for commercial refrigeration units Anymark and Rollsgord (as cited in Dmitiyev and Pisarenko, 1984) suggested a charge equation, only depending on the volume of the evaporator. Dmitiyev and Pisarenko (1984) mention that this equation overestimates the refrigerant charge for domestic appliances and they proposed a charge equation for domestic refrigerators depending on both the volume of the condenser and the evaporator. The main shortcoming of both these equations is that they do not account for the effect of important design aspects of refrigeration appliances, i.e. refrigerant mass flux within the heat exchanger and its effect on the void fraction, internal volume of the liquid line, compressor shell volume and the mass and solubility of refrigerant in the oil. More advanced appliance simulation models, including charge estimation, are available in literature, for example: Li *et al.* (2011), McKinley and Alleyne (2008), Jin and Hrnjak (2016). Although providing good charge estimations, implementation of these detailed simulation models is time consuming and requires computational expertise.

This paper presents an engineering equation to estimate the required refrigerant charge for low charge hydrocarbon based glass door bottle coolers using a capillary tube as the expansion device. The purpose of the equation is to provide a tool assisting engineers in the process of designing the cooling system of a bottle cooler (i.e. selection of the components) such that the total appliance refrigerant charge is within a specified charge limit. The equation is based on dividing a cooling system, of known design, into several control volumes and appliance total refrigerant mass results from summation of the refrigerant mass calculated for each control volume, equation (1).

$$M = \sum_{i=1}^{n} a_i V_i + a_{\text{oil}} M_{\text{oil}} \tag{1}$$

The difficulty in such equation is to obtain a proper estimation of the average density within each control volume for a complete range of appliances. For volumes containing single-phase refrigerant, density can directly be derived from the system operating conditions and refrigerant property data. For volumes containing two-phase flow, the mean void fraction has to be known.

The proposed equation is derived by fitting equation (1) to calculation results of a more detailed numerical charge model. This numerical model, including 22 different void fractions correlations, was used to calculate the refrigerant charge of 10 fully characterized hydrocarbon based glass door bottle coolers ranging in refrigerated volume from 50 to 1200 $dm^3$ with known refrigerant charge.

In the following sections, first the characteristics of the 10 bottle cooler appliances used in the analyses are presented. Hereafter the numerical model is discussed and the results of the charge estimations are given. This is followed by the simplifications suggested resulting in the engineering equation proposed. Finally, the conclusions are given.

## 2. CHARACTERISTICS OF APPLIANCES

Table 1: Appliance characteristics

| | | |
|---|---|---|
| Cabinet | Type | Glass door bottle cooler |
| | Refrigerated volumes | 50 to 1200 $dm^3$ |
| | Refrigerants | R-290 (7x), R-600a (3x) |
| | Design ambient temperature | 32.2 °C |
| | Design cabinet temperature | 3 °C |
| | Heat load during steady state (i.e. including peripherals) | 35 to 380 W |
| | Refrigerant charge | 13 to 95.6 g |
| | Condensing temperature at steady state and 32.2 °C ambient | 38 to 60 °C |
| | Evaporating temperature at steady state and 32.2 °C ambient | -0.5 to -13 °C |
| Compressor | Type | Reciprocating |
| | Displacement | 3 to 12.5 $cm^3$ |
| | Oil type | Polyol ester (R-290), Mineral (R-600a) |
| | Oil charge | 0.067 to 0.24 kg |
| | Cooling capacity at design condition | 55 to 1116 W |
| | Refrigerant flow rate | 0.23 to 3.61 $gs^{-1}$ |
| Condenser | Type | Forced air: Folded Tube and Wire (9x)<br>Forced air: Microchannel (1x) |
| | Airflow | 99 to 662 $m^3h^{-1}$ |
| | Tube length | 0.7 – 20.9 m |
| | Internal diameter | Tube and Wire: 3.2 to 3.6 mm<br>Microchannel: 0.6 mm (31 tubes in parallel) |
| | Overall heat transfer (UA) value at design condition | 7 to 60 $WK^{-1}$ |
| | Mass flux | 31 to 455 $kgs^{-1}m^{-2}$ |
| Evaporator | Type | Forced air: Fin and tube |
| | Airflow | 113 to 483 $m^3h^{-1}$ |
| | Tube length | 0.92 to 24.4 m |
| | Internal diameter | 3.6 to 6.0 mm |
| | Internal volumes | 9 to 688 $cm^3$ |
| | Overall heat transfer (UA) value at design condition | 11 to 198 $WK^{-1}$ |
| | Mass flux | 15 to 129 $kgs^{-1}m^{-2}$ |
| Expansion device | Type | Capillary |
| | Length | 0.8 to 4.6 m |
| | Internal diameter | 0.8 to 1.5 mm |
| | N2 flow @10 bar ΔP | 10.7 to 38.7 $dm^3min^{-1}$ |

All appliances, i.e. glass door bottle coolers, included in the analyses are designed towards low refrigerant charge, i.e. top to bottom refrigerant flow of the heat exchangers, using a capillary tube as the expansion device, short liquid line and no refrigerant accumulators and the appliances were charge optimized. The charge optimization was based on energy utilization measurement (Coca Cola 1, 2014) and half reload recovery testing (Coca Cola 2, 2014) applying various refrigerant charges. The refrigerant charge resulting in the lowest energy consumption, while meeting the half reload recovery performance specification was selected. Nine of these appliances are regarded as conventional bottle coolers using state of the art components. The 10th appliance, however, is fitted with a variable speed compressor, relatively large evaporator and a microchannel condenser, and is specifically designed towards minimum temperature lift and hence low energy consumption. An overview of the main characteristics of these appliances is presented in Table 1.

## 3. NUMERICAL MODEL

A numerical model has been set up in Matlab using the refrigerant property data Refprop 9.1 (Lemmon *et al*, 2013). Using equation (1) the model estimates the total refrigerant charge of the cooling system for stationary operation at a specific operating condition. In the model the cooling system is split up in the following sections (i.e. components): discharge line, condenser, liquid line, filter / drier, evaporator, suction tube, compressor shell and the lubricant oil. Component dimensions, heat transfer values and appliance operating conditions (i.e. system pressures and temperatures) are input parameters and are expected to be known from appliance design. Except for the condenser, evaporator, filter /drier, and the lubricant the refrigerant mass is derived using equation (2), where refrigerant density is derived from the pressure and temperature using Refprop.

$$m_i = \rho_i V_i \tag{2}$$

The model is based on the assumption that two-phase flow exists only within the condenser, evaporator and filter / drier. The condenser and the evaporator are split into two-phase and single-phase regions. For the two-phases regions calculations are performed applying various void fraction correlations, (22 in total). These correlations, including both slip ratio correlations and drift flux correlations were selected from available literature, see section 3.2 were a brief summary of this research is presented.

For the condenser and the evaporator, the length of the two-phase flow region is calculated by subtracting the length of the subcooled and superheated region from the total length of the heat exchanger. The length of these single-phase regions is calculated using 50 calculation elements of equal temperature step (i.e. pre-scribed refrigerant temperature change and hence refrigerant heat transfer). For each element ($k$) the corresponding length ($L_k$) is calculated from the heat absorption / rejection, the overall heat transfer value of the element ($UA_k$) and the temperature difference between the refrigerant and the air following equation (3). $UA_k$ of the element (subcooled or superheated) is estimated assuming a constant thermal resistance between the air and the outer surface of the heat exchanger and between the tube wall and the outer surface, and applying 1-D heat transfer theory. In such case the only difference in thermal resistance between the various sections of the heat exchanger (i.e. subcooled, two-phase, superheated) results from the differences in the refrigerant side heat transfer coefficient ($h_r$), and $UA_k$ can be derived from the known overall heat transfer value based on two-phase flow ($UA_{HEX}$) following equation (5) to equation (7). The heat transfer coefficients ($h_r$) are estimated using the correlations of Gungor and Winterton (1987), for evaporating sections, the correlations of Mathur (1998), for condensing sections and using Janna (2000), for single-phase flow. For each element the refrigerant mass is calculated from the density and the volume and total mass is derived from summation over all elements.

$$L_k = \frac{1}{Q_k} \frac{\Delta T_{a-r,k}}{h_{r,k} P_k + \frac{\text{Const}}{L_{\text{HEX}}}} \tag{3}$$

With,

$$Q_k = UA_k \Delta T_{a-r,k} = \dot{m} \Delta h_{r,k} \tag{4}$$

$$UA_k = \frac{1}{\frac{1}{h_{r,k}P_k L_k} + \frac{\text{Const}}{L_{\text{HEX}}} L_k} \tag{5}$$

$$\text{Const} = \frac{1 - UA_{\text{HEX}} R_{(e,c)}}{UA_{\text{HEX}}} \tag{6}$$

$$R_{(e,c)} = \frac{1}{h_{(e,c)} P_{HEX} L_{HEX}} \tag{7}$$

The two-phase flow regions of the heat exchangers are divided into 100 calculation elements of equal length. The calculations are based on conservation of mass and assuming uniform heat flux, constant pressure, and equilibrium between the phases. Based on this, the mass of condensing or evaporating refrigerant is equal for each element. For each element, the local vapor quality is determined and the slip ratio is calculated using one of the 22 correlations, hereafter the void fraction is calculated using equation (8). Note: Drift flux correlations were converted into a slip ratio correlation format.

$$\alpha_k = \frac{1}{1 + \frac{1 - x_k}{x_k} \frac{\rho_v}{\rho_l} S_k} \tag{8}$$

Finally, the total refrigerant mass within the two-phase section is calculated using equation (9).

$$M_{(e,c)} = \sum_{k=1}^{n} [(\alpha_k \rho_v + (1 - \alpha_k)\rho_l)V_k] \tag{9}$$

For the filter / drier the refrigerant mass is calculated using equation (10), where void fraction $\alpha_f$ needs to be between 0 (completely liquid) and 1 (completely vapor).

$$m_f = [\alpha_f \rho_v + (1 - \alpha_f)\rho_l]V_f \tag{10}$$

The mass of refrigerant dissolved in the lubricant oil is derived using equation (11) applying solubility data of R-600a and R-290 (Polyolester SEZ 68 with R-290 and mineral oil ISO VG5 with R-600a, (Bock and Puhl, 2010)), assuming that all lubricant oil is located inside the compressor shell being at suction pressure and shell temperature.

$$m_{oil} = a_{oil} M_{oil} \tag{11}$$

### 3.1 Background in the selection of the void fraction correlations

Kuijpers *et al.* (1987) showed, by experimental validation of small heat exchangers (i.e. domestic appliances), that both the Premoli and the Hughmark correlation show acceptable agreement when calculating the refrigerant charge in evaporators. They concluded that for calculation of the mean void fraction in both condensing and evaporating flow the Premoli correlation can be considered to be superior. F. Poggi *et al.* (2008) concluded that some of the most used correlations, depending on mass flux, are Hughmark, Premoli and Tandon. De Rossi *et al.* (2011) studied the influence of the refrigerant charge on the steady state working conditions of a vertical domestic freezer. Their total refrigerant mass calculation (complete appliance), showed best agreement with the actual charge for applying the void fraction correlation developed by Rouhani and Axelsson. Woldesemayat (2006) presented a detailed comparison of void fraction correlations for two-phase flow in horizontal and upward inclined flows in his Master thesis report. The work, based on more than 80 void fraction correlations, showed that best agreement results for the Toshiba, Rouhani-Axelsson, Dix, Premoli, Hughmark and Filimonov correlations. Jin and Hrnjak (2016) developed and validated a semi-empirical model to predict the refrigerant and lubricant quantity in both a microchannel condenser and a plate-and-fin evaporator for an air conditioning system. They evaluated six void fraction correlations for the condenser; Zivi, homogeneous, Premoli, Niňo, Hughmark and a Zivi correlation modified for the effect of the oil. For the evaporator they evaluated four void fraction correlations, namely Zivi, Mandrusiak and Carey, Jassim and the homogeneous model. Their validation showed best agreement with the actual refrigerant charge for the Hughmark correlation in the evaporator and for the Jassim correlation in the condenser. All correlations mentioned above are included in the analyses. Next to this several other correlations, taken from Woldesemayat (2006) are included.

### 3.2 Calculation results

Using the numerical model, applying all 22 void fraction correlations, the refrigerant charge was calculated for the 10 fully characterized appliances and compared with the actual total refrigerant charge. In Figure 1 the calculation results for one of the appliances (appliance 3, bottle cooler with a storage volume of 500 $dm^3$) is presented as an example. Based on all appliances best agreement in total refrigerant charge resulted for using the void fraction correlations of Premoli, Hughmark, and Dix in both the condenser and evaporator. Therefore, only these results are presented in this paper, see Table 2.

The calculations showed best agreement in appliance total refrigerant charge at a void fraction of $\alpha = 0$ for the filter / drier (i.e. assuming that the filter is completely filled with liquid refrigerant). However, flow visualization, by Martínez-Ballester *et al.* (2017) and Lee *et al.* (2016), however, have shown two-phase flow in the filter drier (i.e. liquid level at capillary inlet), therefore this needs further evaluation.

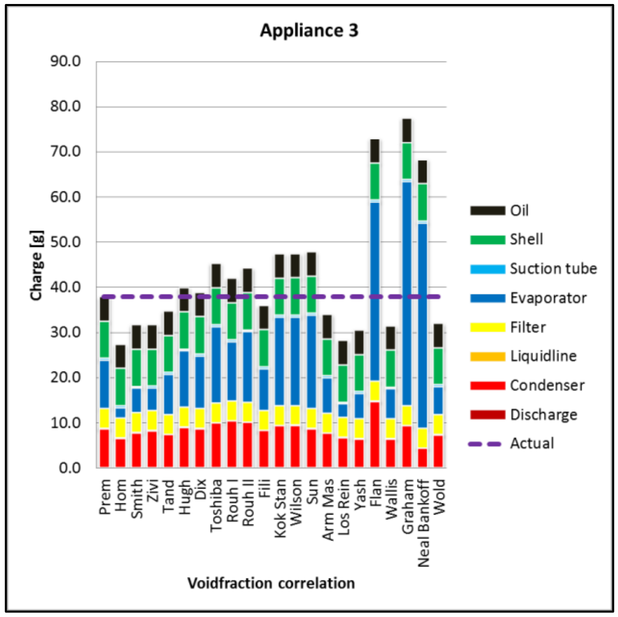


Figure 1: Results of charge estimations for appliance (bottle cooler with storage volume of 500 $dm^3$).

Table 2: Results charge estimation

| Slip ratio correlation | Appl. 1 (50 $dm^3$, 13 g R-600a) | Appl. 2 (300 $dm^3$, 25 g R-600a) | Appl. 3 (500 $dm^3$, 38 g R-290) | Appl. 4 (1200 $dm^3$, 64 g R-290) | Appl. 5 (300 $dm^3$, 46 g R-290) | Appl. 6 (500 $dm^3$, 61 g R-290) | Appl. 7 (1200$dm^3$, 96 g R-290) | Appl. 8 (300 $dm^3$, 53 g R-290) | Appl. 9 (500 $dm^3$, 58 g R290) | Appl. 10 (500 $dm^3$, 40 g R-600a) | Average | Standard deviation |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Premoli | -15% | 4% | 0% | 2% | -2% | 6% | -7% | -13% | 5% | 2% | **-2%** | **7%** |
| Hughmark | -15% | 4% | 5% | 8% | 1% | 10% | -2% | -9% | 9% | 1% | **1%** | **8%** |
| Dix | -13% | 2% | 2% | 0% | 2% | 8% | -9% | -9% | 7% | 20% | **1%** | **10%** |

## 4 CHARGE EQUATIONS

An engineering equation (equation (12)) is developed for using both R-290 and R-600a as the refrigerant. The equation is based on equation (1), and the coefficients and constants are derived from the calculation results of the numerical model using the slip ratio correlation of Premoli *et al.* (1970). It is assumed that the internal volumes of the components, and the system pressures and temperatures are known by design and that the corresponding density of the refrigerant can be derived from a refrigerant property program or looked up from refrigerant property tables. The basis of the underlying numerical model is a bottle cooler appliance specifically designed for low refrigerant charge, having top to bottom refrigerant flow of the heat exchangers, low subcooling (2 K), small superheating (4 K), having a shell temperature of approximately 60 °C, and fitted with a capillary suction gas heat exchanger.

$$M = a_1V_D + (\alpha_c a_2 + (1-\alpha_c)a_3)V_c + a_3V_l + (1-\alpha_f)a_3V_f + \alpha_f a_2 V_f + (\alpha_e a_4 + (1-\alpha_e)a_5)V_e + a_6V_s + a_7V_{sh} + 0.033M_{oil} \quad (12)$$

With refrigerant dependent void fraction coefficients,

| | | | |
|---|---|---|---|
| R-600a | $\alpha_e = 0.685G_e^{0.05} + 0.08$<br>$\alpha_c = 0.62G_c^{0.05} - 0.05$ | R-290 | $\alpha_e = 0.65G_e^{0.05} + 0.09$<br>$\alpha_c = 0.58G_c^{0.05} - 0.03$ |

Where,

| | | | |
|---|---|---|---|
| $a_1 = \rho_{dis}(P_c, 0.5(T_{dis}+T_{c,inlet}))$ | $a_3 = \rho_{l,c}(P_c, x=0)$ | $a_5 = \rho_{l,e}(P_e, x=0)$ | $a_7 = \rho_{shell}(P_e, T_{shell})$ |
| $a_2 = \rho_{v,c}(P_c, x=1)$ | $a_4 = \rho_{v,e}(P_e, x=1)$ | $a_6 = \rho_s(P_e, 0.5(T_{e,out}+T_{suc}))$ | $\alpha_f = 0$ (i.e. liquid refrigerant) |

Equation (12) shows an agreement within 15% between the estimated and the actual charge for the 10 fully characterized appliances, see Figure 2.

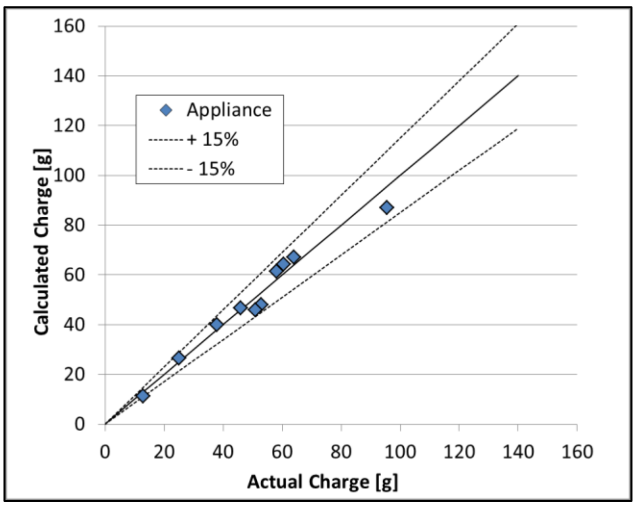


Figure 2: Results of charge estimations using engineering equation

### 4.1 Derivation of constants for void fraction

The coefficients in the proposed engineering equation, representing the average density of the refrigerant within a specific section, require that the void fraction within the heat exchangers is known. The average void fraction ($\bar{\alpha}$) of the heat exchangers is derived from the results of the numerical model following equation (13). For the 10 appliances the results of this calculation are presented in Table 3. Note: The average void fraction includes the refrigerant mass in the subcooled and the superheated sections of the heat exchangers.

$$\frac{M_{HEX}}{V_{HEX}} = (1-\bar{\alpha})\rho_l + \bar{\alpha}\rho_v \Rightarrow \bar{\alpha} = \frac{\frac{M_{HEX}}{V_{HEX}} - \rho_l}{\rho_v - \rho_l} \quad (13)$$

Table 3: Calculated average void fraction of heat exchanges using the selected slip ratio correlations

| | Condenser | | | Evaporator | | |
|---|---|---|---|---|---|---|
| Appliance | $\bar{\alpha}$ (Prem) | $\bar{\alpha}$ (Hugh) | $\bar{\alpha}$ (Dix) | $\bar{\alpha}$ (Prem) | $\bar{\alpha}$ (Hugh) | $\bar{\alpha}$ (Dix) |
| 1 (50 $dm^3$, R-600a) | 0.72 | 0.73 | 0.63 | 0.90 | 0.90 | 0.91 |
| 2 (300 $dm^3$, R-600a) | 0.74 | 0.75 | 0.75 | 0.91 | 0.90 | 0.91 |
| 3 (500 $dm^3$, R-290) | 0.72 | 0.71 | 0.72 | 0.90 | 0.89 | 0.89 |
| 4 (1200 $dm^3$, R-290) | 0.74 | 0.71 | 0.75 | 0.93 | 0.92 | 0.93 |
| 5 (300 $dm^3$, R-290) | 0.73 | 0.72 | 0.73 | 0.89 | 0.87 | 0.87 |
| 6 (500 $dm^3$, R-290) | 0.73 | 0.73 | 0.74 | 0.89 | 0.88 | 0.88 |
| 7 (1200 $dm^3$, R-290) | 0.75 | 0.74 | 0.77 | 0.91 | 0.90 | 0.92 |
| 8 (300 $dm^3$, R-290) | 0.73 | 0.72 | 0.73 | 0.89 | 0.87 | 0.87 |
| 9 (500 $dm^3$, R-290) | 0.73 | 0.73 | 0.75 | 0.89 | 0.88 | 0.88 |
| 10 (500 $dm^3$, R-600a) | 0.65 | 0.66 | 0.65 | 0.85 | 0.85 | 0.80 |
| Average | 0.722 | | | 0.889 | | |

Appliance total refrigerant charge estimations were made based on the average void fractions of 0.722 for the condenser and 0.889 for the evaporator. This showed good agreement, within 17% for the 9 conventional appliances. For the 10$^{th}$ appliance, however, this calculation showed a 23% lower charge than actually applied. The difference with the numerical model, which showed good agreement for appliance 10 when using the void fraction correlations of Premoli or Hughmark (see Table 2), showed to be mainly resulting from the charge calculation of the evaporator. Appliance 10 is an appliance fitted with a microchannel condenser and a variable speed compressor in combination with a standard fin and tube evaporator and is characterized by its low evaporator mass flux ($G_e$ = 15.7 $kgs^{-1}m^{-2}$).

To evaluate the effect of mass flux on void fraction, the slip ratio correlation of Premoli *et al.* (1970) (equation 14) was used to calculate the average void fraction for both a condenser and an evaporator. Calculations were performed for a mass flux ranging from 5 to 505 $kgs^{-1}$, tube diameters varying between 3 to 8 mm, evaporating temperatures between -15 and 0 °C, and condensing temperatures between 35 and 65 °C for using both R-600a and R-290 as the refrigerant. The calculations were performed using Matlab 2016, applying 19 calculation elements of increasing vapor quality (from x = 0.05 to x = 0.95). For each element the slip ratio is calculated, the void fraction is calculated following equation (8), the average refrigerant density is calculated and finally the average void fraction along the tube is derived from the average refrigerant density along the tube and the liquid and vapor density at the saturated conditions. See figure 3 were the results of calculating the average void fraction of the evaporator using R-290 as the refrigerant are presented as an example.

$$S = 1 + K\sqrt{\frac{Y}{1+CY} - CY} \tag{14}$$

Where,

$$Y = \frac{x}{1-x}\left(\frac{\rho_l}{\rho_v}\right) \qquad K = 1.578 Re_l^{-0.19}\left(\frac{\rho_l}{\rho_v}\right)^{0.22} \qquad C = 0.0273 We_l Re_l^{-0.51}\left(\frac{\rho_l}{\rho_v}\right)^{-0.08}$$

and,

$$Re_l = \frac{GD_i}{\mu_l} \qquad We_l = \frac{G^2 D_i}{\sigma \rho_l}$$

The analyses showed that for the evaluated domain, the mass flux is having the largest impact on the void fraction, followed by the type of refrigerant used. Correction parameters for both the mass flux and the refrigerant applied were derived using the results of the average void faction calculations at a tube diameter of 5 mm, an evaporation temperature of -10 °C and a condensing temperature of 45 °C. Fitting of this data, showed best agreement for applying a power function, see figure 4. As a final step, a constant was included and the equation was fitted to improve agreement with the void fraction estimations presented in table 3, resulting in equation (15) with the void fraction coefficients presented in equation (12).

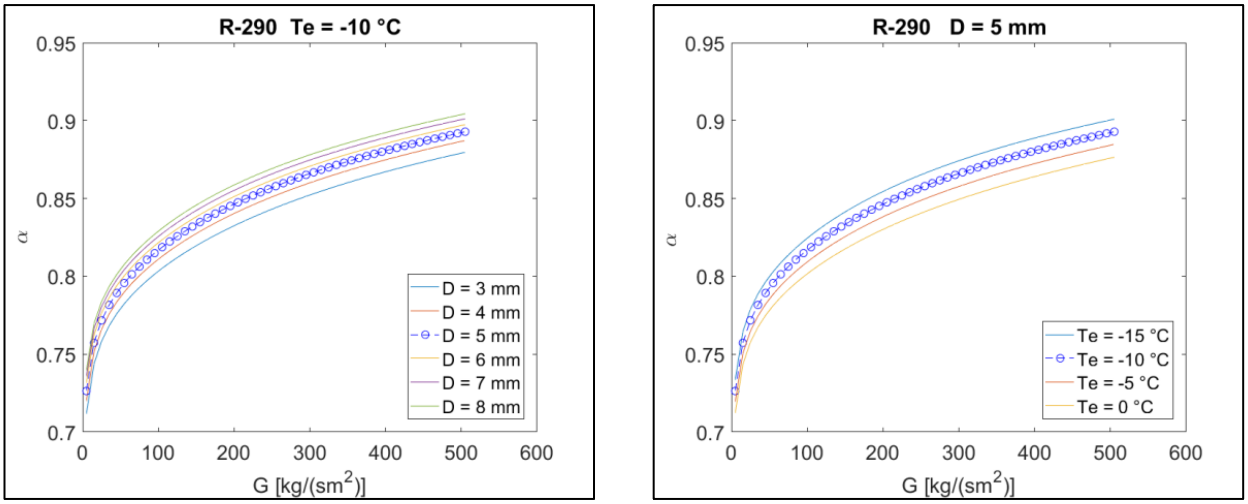


Figure 3: Average void fraction versus mass flux for an evaporator using R-290 as the refrigerant. Left: influence of the evaporating temperature. Right: influence of the tube diameter.

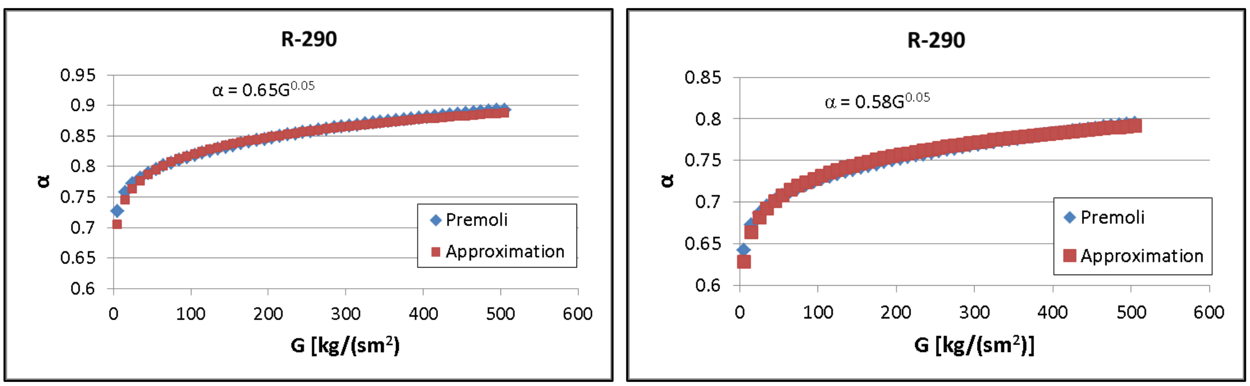


Figure 4: Calculation and approximation using power function of the average void fraction using the slip ratio correlation of Premoli *et al*, 1970, for a tube with inner diameter of 5 mm. Left evaporator. Right condenser.

$$\bar{\alpha} = AG^{B} + C \tag{15}$$

### 4.2 Derivation of constants for solubility

The solubility of the refrigerant within the lubricant oil depends on the temperature, the pressure and the type of oil and refrigerant used. In the numerical model the amount of refrigerant dissolved was calculated for all ten appliances using refrigerant solubility data. Using equation (16) an appliance specific solubility coefficient can be derived from the results of the numerical model, see Table 4.

$$a_{\text{oil}} = \frac{M_{\text{dis}}}{M_{\text{oil}}} \tag{16}$$

Table 4 shows that except for appliance 10, the deviation in $a_{oil}$ is relatively small (within 10%). Appliance 10, is designed for low energy consumption (i.e. small temperature lift), therefore it differs from the other appliances due to its operation at much larger suction pressure (evaporating temperature of -0.5 °C for appliance 10, other appliances between -6 °C and -13 °C). In the engineering equation the average of all appliances is used as estimation for $a_{oil}$( $a_{oil}$ = 0.033).

Table 4: Mass of refrigerant dissolved calculated with the empirical model, total mass of oil and the estimated solubility coefficient.

| Appliance | Oil type | Refrigerant | $M_{dis}$ [g] | $M_{oil}$ [g] | $a_{oil}$ |
|---|---|---|---|---|---|
| 1 | Mineral | R-600a | 2.2 | 67 | 0.0329 |
| 2 | Mineral | R-600a | 4.4 | 124 | 0.0359 |
| 3 | Polyolester | R-290 | 5.4 | 184 | 0.0292 |
| 4 | Polyolester | R-290 | 6.6 | 240 | 0.0275 |
| 5 | Polyolester | R-290 | 5.5 | 184 | 0.0298 |
| 6 | Polyolester | R-290 | 5.9 | 184 | 0.0323 |
| 7 | Polyolester | R-290 | 8.1 | 240 | 0.0337 |
| 8 | Polyolester | R-290 | 5.5 | 184 | 0.0301 |
| 9 | Polyolester | R-290 | 6.0 | 184 | 0.0326 |
| 10 | Mineral | R-600a | 4.5 | 92 | 0.0486 |
| Average | | | | | 0.033 |

## 5. CONCLUSIONS AND DISCUSSIONS

An easy to use charge equation, showing agreement within 15% in appliance total refrigerant charge for 10 charge optimized, low refrigerant charge, cooling appliances has been proposed. The equation has been developed for hydrocarbon based glass door bottle coolers designed towards minimum refrigerant charge (i.e. no liquid accumulation and top to bottom flow of the heat exchangers). The equation can be used to obtain an indication of required refrigerant charge during early stage cooling system design (i.e. selection and design of the components during development of bottle coolers with low refrigerant charge).

The equation calculates the refrigerant charge for the various sections of the cooling circuit and the total charge is derived by summation over these sections. The validation is performed on total system charge, only. Therefore, possible deviations between the actual and the calculated charge in various sections of the cooling system could cancel out. Therefore, care needs to be taken when using the equation for charge estimation at component level or when evaluating appliances having characteristics other than presented in section 2 of this paper.

## NOMENCLATURE

| | | |
|---|---|---|
| $A$ | cross area tube | ($m^2$) |
| $A$ | constant | ($kg\ s^{-1} m^{-2})^{-B}$ |
| $a$ | average density | ($kg\ m^{-3}$) |
| $a_{oil}$ | solubility of refrigerant in lubricant oil | ($kg\ kg^{-1}$) |
| $B$ | constant | (-) |
| $C$ | constant | (-) |
| $D_i$ | inner diameter | (m) |
| $G$ | mass flux | ($kg\ s^{-1}\ m^{-2}$) |
| $h$ | heat transfer coefficient | ($W\ m^{-2}\ K^{-1}$) |
| $L$ | length | (m) |
| $M$ | total refrigerant mass | (kg) |
| $M_{oil}$ | mass of lubricant oil | (kg) |
| $m$ | refrigerant mass | (kg) |
| $\dot{m}$ | mass flow | ($kg\ s^{-1}$) |
| $P$ | tube perimeter | (m) |
| $P$ | pressure | (Pa) |
| $Q$ | heat flow | (W) |
| $Re$ | Reynolds number | (-) |
| $R$ | thermal resistance | ($K\ W^{-1}$) |
| $S$ | slip ratio | (-) |
| $T$ | temperature | (°C) |
| $UA$ | overall heat transfer rate | ($W\ K^{-1}$) |
| $u$ | velocity | ($m\ s^{-1}$) |
| $V$ | volume | ($m^3$) |
| $We$ | Weber number | (-) |
| $x$ | vapor quality | ($kg\ kg^{-1}$) |
| $\alpha$ | void fraction | (-) |
| $\bar{\alpha}$ | mean void fraction | (-) |
| $\Delta h$ | specific enthalpy change | ($J\ kg^{-1}$) |
| $\Delta T$ | temperature difference | (K) |
| $\mu$ | dynamic viscosity | (Pa s) |
| $\rho$ | refrigerant density | ($kg\ m^{-3}$) |
| $\sigma$ | surface tension | ($N\ m^{-1}$) |

Subscript

| | | | |
|---|---|---|---|
| $a$ | air side | $k$ | calculation element index |
| $c$ | condensing / condenser | $l$ | liquid |
| $D$ | discharge line | $oil$ | lubricant oil |
| $dis$ | dissolved refrigerant | $r$ | refrigerant side |
| $e$ | evaporating / evaporator | $out$ | outlet |
| $f$ | filter / drier | $s$ | suction line |
| $HEX$ | heat exchanger | $sh$ | shell |
| $i$ | control volume index | $suc$ | suction tube compressor |
| $in$ | inlet | $v$ | vapor |

## REFERENCES


Bock, W., Puhl, C. (2010), *Kältemaschinenöle*, Berlin: VDE Verlag GMBH.

Coca Cola 1 (2014), *Energy Utilization: SM-PR-5040*, The Coca-Cola Company

Coca Cola 2 (2014), *Half-Reload Recovey- Cooler: SM-PR-5070*, The Coca-Cola Company

De Rossi,F., Mauro A.W., Musto M., Vanoli G.P. (2011), Long-period food storage household vertical freezer: Refrigerant charge influence on working conditions during steady operation, *Int. J. Refrig. 34*(5), 1305-1314.

Dmitriyev, V. I., Pisarenko V. E. (1984). Determination of optimum refrigerant charge for domestic refrigerator units. *Int. J. Refrig., 7* (3), 178-180.

Gungor, K.E., Winterton, R.H.S. (1987). Simplified general correlation for saturated flow boiling and comparison with data. *Chem Eng Res Des, 65*, 148–156

Janna, S. W. (2000). *Engineering heat transfer, second edition* (page 324). Boca Raton, FL: CRC Press.

Jin, S. and Hrnjak, P. (2016), Refrigerant and lubricant charge in air condition heat exchangers: Experimentally validated model, *International Journal of Refrigeration,*67, 395-407

Kuijpers, L., Janssen, M., de Wit, J., (1987), Experimental verification of liquid hold-up in small refrigeration heat exchangers, in *XVIIth International Congress of refrigeration*, *Vienna*, 307–315

Lee, WJ., Seo, JY., Ko, J., Jeong, J.H., (2016), Non-equilibrium two-phase refrigerant flow at subcooled temperatures in an R600a refrigeration system, *Int. J. Refrig., 70,* 148-156.

Lemmon, E.W., Huber, M.L., McLinden, M.O. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, National Institute of Standards and Technology, Standard Reference Data Program. Gaithersburg, 2013.

Li, B., Peuker, S., Hrnjak, P. S., & Alleyne, A. G. (2011). Refrigerant mass migration modeling and simulation for air conditioning systems. *Applied Thermal Engineering, 31*(10), 1770-1779.

Mathur, G.D. (1998), Heat transfer coefficients for propane (R-290), isobutene (R-600a), and 50/50 mixture of propane and isobutane, In ASHRAE Transactions 104,2, *ASHRAE Annual meeting, Atlanta, United States of America* (1159-1172)

Martínez-Ballester, S., Bardoulet, L., Pisano, A., Corberán J.M., (2017), Visualization of refrigerant flow at the capillary tube inlet of a high-efficiency household refrigerator, *Int. J. Refrig., 73,* 200-208.

McKinley, T. L., & Alleyne, A. G. (2008). An advanced nonlinear switched heat exchanger model for vapor compression cycles using the moving-boundary method. *International Journal of refrigeration, 31*(7), 1253-1264.

Poggi, F., Macchi-Tejeda,H., Leducq D., Bontemps A. (2008), Refrigerant charge in refrigerating systems and strategies of charge reduction, *International Journal of Refrigeration, 31*(3), 353-370.

Premoli, A., Di Francesco, D., Prina, A., (1970),Una correlazione adimensionale per la determinazione della densita di miscele bifasiche, in *XXV Congressor Nazionale ATI*, Trieste, Italy, 120-129

Woldesemayat, M. A. (2006). *Comparison of void fraction correlations for two-phase flow in horizontal and upward inclined flows* (Oklahoma State University).


## ACKNOWLEDGEMENT

We would like to express our sincere thanks to The Coca-Cola Company, for their financial and technical support in carrying out this research work.