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Refrigerant Charge Reduction in a Small Commercial Refrigeration Systems

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Abstract: In this paper different aspects of heat transfer and pressure drop in the heat exchanger as designed for the charge minimization are discussed. Focusing on the two-phase model prediction related to the charge in a small channel evaporator as a design choice for a significant charge reduction. Comparisons of the determined results between four evaporators with different channel diameters are presented in figures. Choice of void fraction correlations appears with greater effect on the refrigerant charge and three mass- flux void fraction correlations are recommended and compared. Concluding that the heat transfer mechanisms of these small diameters are different from those of larger diameters.

Key words: Charge reduction, heat exchanger, refrigeration system, hydraulic diameter, void fraction, channel evaporator, two-phase flow, pressure drop, mass flux

INTRODUCTION

New requirements on the service of refrigeration systems have set on the technological developments and the use of efficient and environmentally refrigerants, the ozone depletion layer, greenhouse warming potentials and the increased energy prices as well as increasing the reliability, whole of these demands determines the necessity for developing and designing a new systems with low refrigerant charge and keeping leakage at lowest possible level.

The channel diameter reduction in the heat exchangers for lowering the refrigerant charge as a design choice is well demonstrated by the most correlations that express the relation ship between the channel diameter lowering and a heat transfer increase, so for lowering the refrigerant charge, the heat exchangers (with small diameters) with small internal volume must be used, then in turn the refrigerant mass is lowered to serve for lowering the emissions.

Refrigerant charge minimizing is an attractive choice for several advantages and reasons such as:

- Reducing the direct environmental effects (ozone depletion and global warming issues).
- Low charge may gives a possibility for using a very good refrigerants with high cycling efficiencies and better heat transfer, this situation give the possibility for the application of small channeled heat exchangers with lower volume.

For two-phase flow in heat exchanger with small hydraulic diameter in the condenser and evaporator channels, the design correlations for pressure drop, heat transfer and flow regimes (Flow pattern) as used for larger tube diameters cannot be used for predicting the case of small diameter tubes. The scope of this paper is to evaluate and establish the suggested correlations in order to perform the analytical prediction of the charge in a small channel evaporator in order to give a good aid for the wanted system design.

For two-phase flow condensation as well as boiling several different correlations are being used. Comparison and prediction of the refrigerant charge is not a simple task, also the analysis is complicated by the fact that the pressure drop will results in a change of the saturation temperature, adding that the two-phase flow is one of the most complex fields of fluid dynamics, difficulty in charge inventory analysis for a proper prediction in the two-phase zones of both condenser and evaporator, is caused by the vapor-to-liquid slip at each cross section and the variation of refrigerant quality a long the tube that causes changes in the flow pattern, the flow pattern is determined by some important physical parameters such as :surface tension and gravity forces.

Generally the correlations predict an increase in the heat transfer coefficient with decreasing diameter. The void fraction correlations choice is found to give a greater aid for the charge prediction, in which the output of the refrigerant charge models are based on different void fraction correlations that can vary by a factor of ten[1,2].

In this paper the focus is on evaluating and comparing between three void fraction correlations, in which different channel diameters are used, assuming constant heat flux to evaluate the effect on the refrigerant mass predictions over the two-phase zones in the evaporators operating at the given conditions.

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Few investigations have been made on the charge amount, distribution and issue[3] from the University of Illinois Air Conditioning and Refrigeration Center (UI ACRC) with the research group they found and adjust the void fraction correlation to suit the calculations for evaporation and condensation in all tubes of 4 to 9 mm diameters for rectangular tubes. Rice[1] classified void fraction prediction approaches and applied them to refrigerant charge prediction.

Little information regarding the small diameter tubes is available in the literature of specialty; one information is reported by Yang and Webb[4] in which they investigate the influence of pipe diameter on the vapor quality, mass flux, pressure drop and heat transfer, they uses the refrigerant R-12 condensed in two different flat extruded aluminum tubes with the external dimensions of (16x3 mm) and a wall thickness of 0.5 mm, at the inside the tubes were divided into four channels by thin longitudinal walls. The difference between the two tubes was that one had smooth internal surfaces while the other having longitudinal micro-fins, 0.2 mm in height. It was found that the heat transfer coefficient increased with increasing vapor quality, increasing mass flux and increasing heat flux. The results were compared with the predicted and reporting that a good agreement for the plain tube at low mass flux was obtained, but (10-20%) lower than predicted at higher mass flux.

Current study is based on the subject of charge minimization in a small commercial air-cooling application that working on (R134a) by using a fined heat exchanger evaporator (6 mm spaced fins), where the saturation temperature for a typical mode of evaporator ($T_{sat.}$= -8°C), having a heat capacity of (Q= 5 kW) and a total mass flux (G= 30 g sec^{-1}), assuming that the refrigerant input quality to the evaporator is 25% (x= 0, 25) while the output quality is as saturated vapor, the internal tubes of evaporator are smoothly rounded.

Investigation and comparison are to be done between four evaporators with different inner diameters as following: D_1=13 mm (the largest diameter), D_2= 9 mm, D_3= 3 mm and D_4= 1 mm (the smallest diameter).

MATERIALS AND METHODS

For the refrigerant flow in narrow tubes having diameters of ≤ 3 mm, the physical processes seems to be different from those with larger diameter tubes, because the surface tension forces start to have a larger influence on the process, while the gravity forces are still negligible, the effect of these forces can be expressed by the dimensionless number:

$$Co = \sqrt{\frac{\sigma}{g(\rho_L - \rho_V).D^2}} \quad \text{(Confinement number)} \quad (1)$$

Confinement number is introduced by Kew and Cornwell[5] stating that the heat transfer mechanisms are different above and below Co= 0,5 concluding that for the application of refrigeration and heat pumps with minimum charge, the diameter of interest is in the range from 0.5 mm to 3 mm and the surface tension forces have a larger influence on the process in narrow tubes, which means that the flow stratification does not appear and the bubble diameter is restricted by the diameter itself[6,7].

The flow pattern and heat transfer mechanisms can be described in summary as:

Isolated bubble flow: In this regime the bubbles are still smaller than the tube diameter, the heat transfer coefficient is highly dependent on the flux and marginally on the mass flow.

Confined bubble boiling: In this regime the bubble diameter is restricted by the tube diameter, the heat transfer coefficient is only dependent on the heat flux[5].

Convective boiling: When the vapor fraction increases the liquid plugs in between the bubbles will decrease in length and the bubbles grow up rapidly in length.

Partial dry out: In this regime, the thin liquid film will tend to dry out and the heat transfer can be modeled as in the above regimes. So it is important to mention that in the two-phase flow, the change from the isolated bubble flow into a partial dry out flow may take place in a very short section of the tube, that is different from larger diameter tubes, because the ratio of volume and heat transfer area is smaller than in the large diameter tubes, the heat flow will be higher and then it lead to a faster liquid evaporation, that will push the liquid slugs in the channel.

Generally it is founded that the heat transfer coefficient is highly dependent on the heat flux and weekly on the mass flux in small diameter tubes[8]. Many void fraction and pressure drop correlations have been developed for rounded tubes of different diameters. The goal of this paper is to determine which of the existing models gives the close prediction for round tubes of different diameters in order to be used in the rectangular heat exchanger with the given diameters for the case study.

Design consideration: For commercial refrigeration systems it has been demonstrated that the use of a secondary fluid on both hot and cold sides of the heat pumping systems may reduce the charge by a factor of ten[1]. For further charge reduction the heat exchanger should be done with care, in which channels with small diameters are to be used, the volume contained in the rounded tube is proportional to the squared diameter and the surface area direct proportional to the diameter.

A reduction in the diameter to its half is compensated with doubling the tube length for keeping the surface area thus will result in the internal volume reduction to the half of its original.

In this paper, the focus is on examining the consequence of how to manage the same flow of energy in different tube diameters (D_1, D_2, D_3 and D_4), in which the same heat exchanger type and same refrigerant are used.

The expected problem of a charge analysis for small diameter evaporator is the choice of the heat exchanger geometry and. So a uniformly heated rectangular heat exchanger as a simplest choice will be assumed as a general geometry to allow for a range of possible tube diameters.

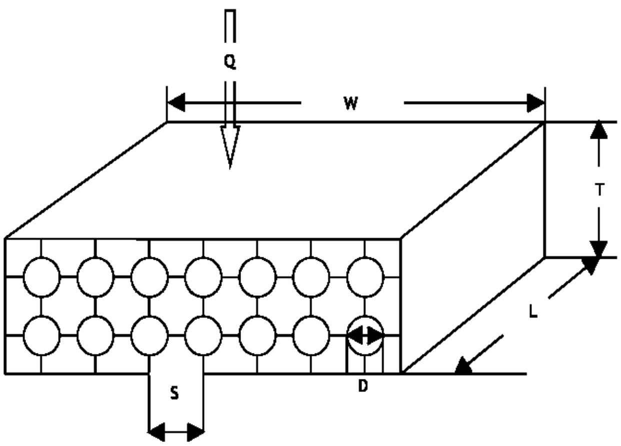

Fig. 1: Heat exchanger shape

As shown in (Fig. 1) the rectangular heat exchanger with a number of rounded tubes (y), each tube having a number of rectangular fins with a side length dependent on the tube diameter and the necessary data input are; the flux of heat Q (W), flow rate of the refrigerant G (kg sec^{-1}) the tube diameter D (mm).

By taking two different tube sizes for the same mass flux and operating conditions of saturation temperature (T_{sat}) and heat flux (Q), the flow area will be the same for each diameter that can be expressed as:

$$\frac{y_1}{y_2} = \frac{D_2^2}{D_1^2} \tag{2}$$

Then, with D_1= 10 D_2, the tube number increases by a factor of 100.

Refrigerant amount prediction: For determining the amount of a refrigerant in the two-phase zones of evaporator, internal tubes in the evaporator are divided into 70 sections for each one, the change of quality in each section along the tube is 0,01 assuming constant thermal flux the heat flux per each section is given by:

$$Q_{sec} = \frac{Q_{channel}}{Y} = G_{channel}.H_v.\Delta x \tag{3}$$

Where Y is the number of sections.? x is the change of quality in every section and $G_{channel}$ is the mass flux in every channel. A coefficient of 100 (W/m^2 K) taken to represent the air-side while the refrigerant side heat transfer coefficient for the larger diameters in our study case are the largest ones (D_1=13 mm) and (D_2=9 mm), both diameters can be computed by using the correlation of Dobson *et al.*[9]. For the small diameters (D_3= 3 mm) and (D_4= 1 mm), the heat transfer mechanisms are different from those with larger diameters, so the design correlations of standard larger diameters may no longer be valid, as stated by several correlations, that the heat transfer coefficient increase with the diameter decrease.

For the small channel diameter of D_1= 1 mm[10], correlation as developed on experimental tests with refrigerants in small diameter tubes, it predicts a slightly small error of (4%). For computing the section length, the heat transfer coefficient is used:

$$Q_{section} = K.\Delta T_m \tag{4}$$

Where ΔT_m is the refrigerant and air temperature difference in the evaporator and K is the overall heat transfer coefficient (W/K) of the wanted section z along the tube, the symbol (z) is chosen to indicate the section location, which having a range from (0 to Y-1) so the global coefficient is used to compute the section length as in the following expression:

$$\frac{1}{K} = \frac{1}{\alpha_{refrigerant}.A_{refrigerant}} + \frac{1}{\alpha_{air}.A_{air}} \tag{5}$$

Where:

$A_{ref} = \pi . D . L_Z$ and

$$A_{air} = \pi.D.L_z + \frac{A_{fin}}{meter}.L_z$$

The two- phase density ρ_Z, in every section that depends on the void fraction is expressed as:

$$\rho_Z = (1 - \alpha_Z) . \rho_l + \alpha_Z . \rho_v \tag{6}$$

By definition, void fraction is the time- averaged fraction of the cross-sectional area or volume occupied by the gas phase in two- phase media, Precise prediction of the void fraction is important for calculating the amount of refrigerant in evaporator. So the used model for fraction prediction must be carefully chosen. Depending on the used model, the void fraction accuracy can vary from 2 to 15%, giving a variation in the calculated amount of refrigerant of up to 75%.

One of the factors that influence the accuracy of prediction is the vapor quality that changes along the tube. As the section length is determined by using the equation (5) and (6).

Then the mass of the refrigerant in every section ca be computed as:

$$m_Z = \rho_Z \cdot A_{cr} \cdot L_Z \tag{7}$$

Where, A_{cr} is the cross-section area, then the total amount of refrigerant can be determined with the summation of the mass in every section along the total tube length.

Void fraction: The void fraction is an important parameter for evaluating the amount of refrigerant, several correlations are used to evaluate the void fraction in the two-phase flow and finally the amount of refrigerant in the evaporator is computed by using these correlations, the void fraction as a quality dependent is highly influenced by the vapor quality that changes along the tube, so in our work we have to chose the most appropriate correlation and model for reaching the goal of this paper.

The correlations that are used for determining the refrigerant mass in the system are the followings, the homogenous flow, slip-ratio correlation[11], Premolli correlation and Hughmark models.

The homogeneous flow: The main assumption is that both phases of two-phase flow are mixed and have the same velocity so the flow behaves as one-phase.

The general equation for the void fraction is deduced from the mass balance and continuity equation that is expressed as:

$$\alpha = \frac{1}{1 + \left(\frac{u_v}{u_l} \frac{1-x}{x} \frac{\rho_v}{\rho_l} \right)} \tag{8}$$

Where the term u_v/u_l is the slip ratio (S) and the velocities for both phases in the homogeneous flow are the same, S=1.

Hugh mark correlation: Is one of the most complex ways of calculating the void fraction due to its iteratively determined coefficient. This correlation is a function of vapor quality and viscosity of each phase, mass flux and the channel geometry.

Many authors for predicting the amount of refrigerant such as used this correlation:[1,12,9]. The void fraction is given by a correction factor K_H that is introduced to the homogeneous equation:

$$\alpha = K_H \cdot \alpha_{homog} \frac{K_H}{1 + \left(\frac{1-x}{x} \frac{\rho_v}{\rho_l} \right)} \tag{9}$$

Where

$$K_H = 0,7266 - 0,0003482.Z - \frac{0,8454}{Z} + 0,06011.Z^{1/3} \tag{10}$$

and

$$Z = f(Re, Fr, \alpha_{hom\,og}) \tag{11}$$

K_H is dependent on the correlating parameter Z (tabulated), Z is a function of the void fraction α-weighted Re number, Fr number and the liquid volume fraction. This is the main reason of iterations for evaluating and determining the value of K_H. Hughmark correlation was developed originally for two-phase flow of air-liquid mixture closely to the atmospheric pressure it shows reasonable results for vertical as well as for horizontal flow, more details are detailed in Rice[1].

Slip-ratio correlation: It uses the same equation of the void fraction as expressed previously in which the slip-ratio is determined as:

$$S = \left[1 - x \left(1 - \frac{\rho_l}{\rho_v} \right) \right]^{1/2} \tag{12}$$

This correlation has been included as the one the best correlations that predicts the closest void fractions to the experimental values, slip-ratio correlations were investigated by many investigators and several correlations for the slip-ratio exist. Premoli correlation, was also identified by Rice[1] as one of the most suitable correlations for the refrigerant application.

This correlation is of interest because of its tendency for minimizing the prediction errors of the liquid density. Because the fluid density is directly related to the refrigerant mass prediction, this correlation could be the best choice for our case study, in which the slip-ratio is represented as

$$S = 1 + E_1 \cdot \left(\frac{J}{1 + J.E_2} - J.E_2 \right) 1/2 \tag{13}$$

Where

$$j=\frac{x}{1-x}\frac{\rho_l}{\rho_v},\ E_l=1.578.Re_{10}^{-0.19}.\left(\frac{\rho_l}{\rho_v}\right)^{0.22}$$

$$\text{and } E_2=0.0273.We_{10}.Re_{10}^{-0.51}.\left(\frac{\rho_l}{\rho_v}\right)^{-0.08}$$

Lockhart and martinelli model: Is based on their research on pressure drop in the two-phase flow in pipes, they introduce a new parameter as called the Lockhart-Martinelli parameter (X_{tt}), that is the ratio of a viscous dissipation to the vapor internal energy and X_{tt} squared is defined as the ratio of the liquid pressure drop to the vapor pressure drop. This new parameter is computed as:

$$X_{tt}=\left(\frac{1-x}{x}\right)^{0.9}.\left(\frac{\rho_v}{\rho_l}\right)^{0.5}.\left(\frac{u_l}{u_v}\right)^{0.1} \quad (14)$$

A mass-flux-dependent correlation was developed by (Tandon), for annular flow that is based on the X_{tt} parameter and Re_{10} and expressed in the following form:

$$\alpha = f(Re_{10}, x_{tt}) \quad (15)$$

It is found also that the void fraction can be represented as a function of X_{tt}[1] that is formulated as in the following way:

For $X_{tt} \le 10$: then $\alpha= (1 + x_{tt}^{0.8})^{0.378}$ and
For $X_{tt} \le 10$: then $\alpha= 0.823 - 0.157 \ln X_{tt}$ (16)

Pressure drop prediction: For two-phase flow, the total pressure drop in tubes is the sum of three components, 1- is the gravitational (static) pressure drop, 2- is the momentum pressure drop and 3- is the frictional. Pressure drop that is the most component in which the total pressure is influenced, the total pressure drop:

$$\Delta p = \Delta p_{gr} + \Delta p_{mom} + \Delta p_{fr} \quad (17)$$

Both gravitational and momentum pressure drops are relatively ignored. The pressure drops were compared to the Martinelli two-phase multiplier model. Frictional pressure drop in two-phase flow can also be predicted in two main ways: by using two-phase multiplier for one-phase pressure drop, or by using friction factors developed for two-phase flow. Frequently used model that is based on two-phase multiplier is the Friedel correlation for the frictional two-phase pressure gradients.

A special friction factor to be used in two-phase frictional pressure drop calculations was suggested by Pierre[13] in order to calculate the frictional pressure drop in horizontal two-phase flow.

Friedel correlation: Is written in terms of the two-phase multiplier, is defined as:

$$\phi_l^2=\frac{(-dp/dl)_{lg}}{(-dp/dl)_l} \quad (18)$$

Where $(-dp/dl)_{lg}$ is the frictional pressure gradient in two-phase flow and $(-dp/dl)$ is the frictional pressure gradient in single-phase liquid flow as in case it would flow at the same mass flow rate as the two-phase flow.

The two-phase flow pressure drop as computed by using Friedl two phase multiplier is:[14]

$$\phi_l^2=E+\frac{3.24FH}{Fr^{0.045}.We^{0.035}} \quad (19)$$

Where

$$E=(1-x)^2+x^2\frac{\rho_l.C_{fg}}{\rho_g.C_{fl}},\quad F=x^{0.78}(1-x)^{0.224}.$$

$$H=\left(\frac{\rho_l}{\rho_g}\right)^{0.91}.\left(\frac{\mu_g}{\mu_l}\right)^{0.19}\left(1-\frac{\mu_g}{\mu_l}\right)^{0.7}$$

$$F=\frac{G^2}{gDh.\rho_m^2},\quad We=\frac{G^2.D_h}{\sigma.\rho_m},\quad \rho_m=\left(\frac{x}{\rho_g}+\frac{1-x}{\rho_l}\right)^{-1}$$

Granryd[14] has investigated frictional pressure drop for the rectangular heat exchanger type and results in terms of pressure drop were correlated by using a modified Pierre model resulting in the following equation:

$$\Delta p = f_m . G^2 . V'' . L/D_h^3 . \quad \text{Where } f_m = 0.083) \quad (20)$$

RESULTS AND DISCUSSION

Void fraction in evaporator: The choice of the model for the void fraction computation has a big influence on the prediction of the amount of refrigerant in evaporator. The void fraction was predicted by using several available correlations, out of which the Hughmark, Tandon and Premoli models seems to give the best approximation. Fig. 2 shows the effect of the channel diameter and the selected correlation on the void fraction results, as a function of refrigerant quality.

As can be seen from the graphs, we can observe that for all selected correlations, the void fraction decreases with the channel diameter decrease. For the qualities superior than 25% (0,25), the void fraction is observed in

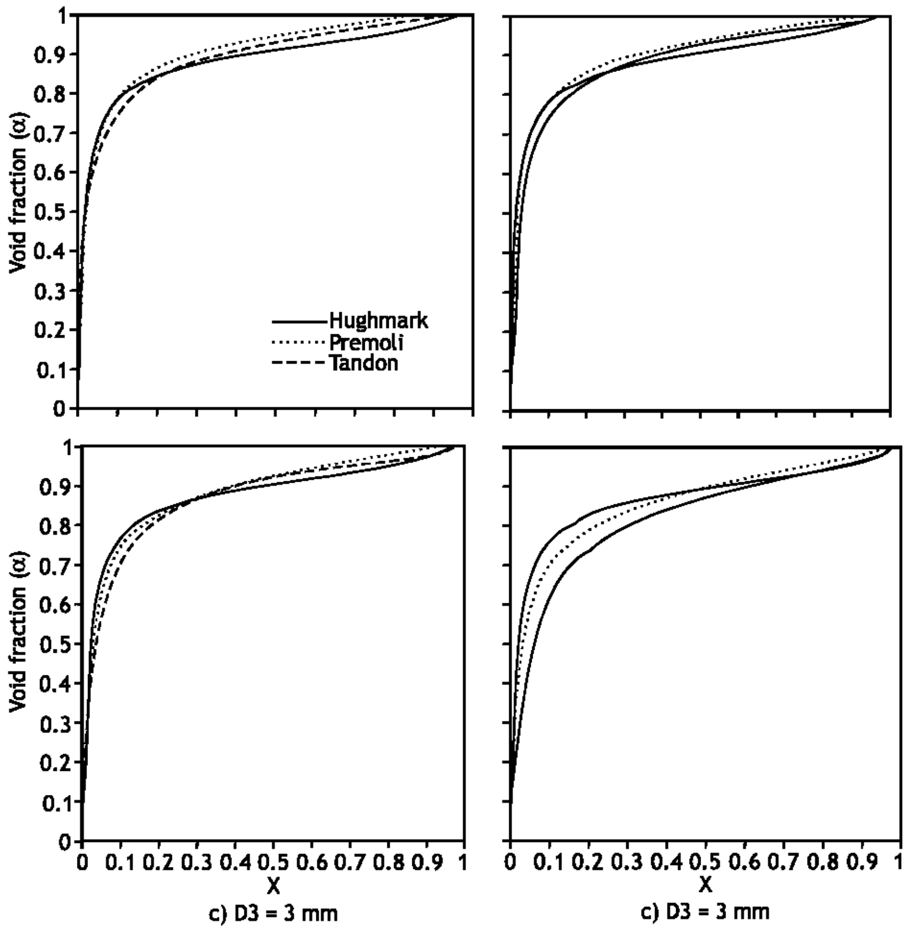

Fig. 2: Comparison of the void fraction (α) vis-à-vis the quality (x) for different evaporator diameters, in which the selected models are shown for the same working conditions

the curved shape for the range of values as given by the selected correlations of Hughmark, Tandon and Premoli.

For the smallest diameter size of 1mm, the Hughmark correlation results gives a higher values of the void fraction and of the amount of refrigerant in the evaporator tubes than the values

obtained by using Premoli correlation (for vapor qualities inferior than 0,55). This behavior may be related to the absence of Weber number in Hughmark correlation.

Higher precision of Hughmark correlation (up to 5% maximum deviation) is achieved by the iterative calculation of the correlating parameter Z, which is dependent on void fraction (the void fraction is defined by correction factor of the homogeneous flow) that is in turn a function of the correlating parameter Z. For conventional channel sizes, the surface tension force effect was found as insignificant.

As the channel diameter size decreases, the surface tension force increases relatively to gravitational forces. So the dimensionless groups of Reynolds and Frouds numbers should be included in the slip-ratio correlation as suggested by Zietiow and Pedersen[12] they suggest that these numbers should be included in the correlation for the conventional pipes size.

Refrigerant mass comparison: The refrigerant charge in the evaporator was computed by using the previous equations for the heat flux for each section (eq 3), the section length by using the heat transfer coefficient (eq 5), the section position along the tube (eq 7) and the mass in each section as in (eq 8).

Refrigerant charge results are obtained from the selected correlations of a void fraction where the heat flux was assumed to be constant for all the diameter sizes (four evaporators with $D_{1,}$ $D_{2,}$ D_3 and D_4). Fig. 3 shows the results obtained by using both correlations of Premoli and Tandon, in which they predicts a reduction in the refrigerant mass of about 85% as the diameter start decreasing from the largest diameter (13 mm) to the smallest one (1 mm), a mass reduction of about 75% is also obtained as the diameter decreases from (9 to 1 mm).

In comparison with the values obtained by using the Hughmark correlation, a significant important reduction is observed to be of about 90% when the diameter is reduced from 13 mm to 1mm and 85% when the diameter is reduced from 9 to 1 mm. Finally we conclude that an

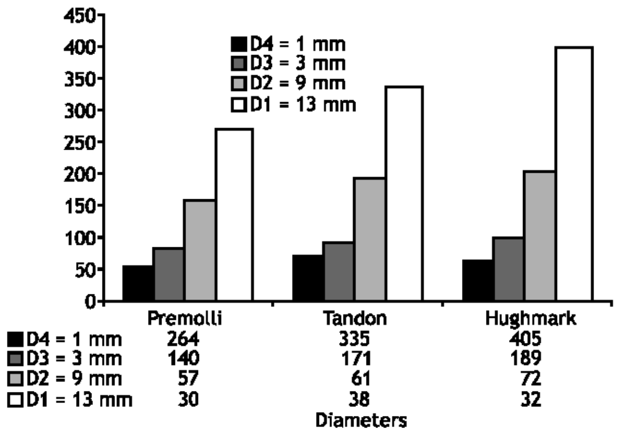

Fig. 3: The comparison R-134a charge reduction in different evaporator diameters as shown by the selected models

important refrigerant charge reduction can be detected accurately by using the above three correlations.

The pressure drop: The pressure drop was calculated by using the majority of the interrelated correlations as applied for the small size diameters, the main observation is that the pressure drop penalty is higher than expected, in which the deviation is expected to be in the range of 30% for two-phase flow.

Walley[15] describes Friedle correlation as the best available and generally applicable method; noticing that even the best empirical correlations developed for two-phase flow pressure drops calculation, give errors of 40%. Granryd model seems to give the best approximation, by giving a deviation of 20%.

The closest prediction for the two-phase void fraction (amount of refrigerant) in the evaporator channels was achieved when using Hughmark, Tandon and Premoli models.

The Hughmark model superiority for determining the refrigerant mass reduction with the same operating conditions and heat flux is observed clearly, Hughmark model as compared with the other two models demonstrates its superiority when the diameter size decreases from (13 to 1 mm) where the mass reduction is of about 90% and from (9 to 1 mm) the mass reduction is

Nomenclature

Latin symbols		Greek symbols	
D_h Hydraulic diameter	[m]	α Void fraction	[%]
D Tube internal diameter	[m]	ρ Density	[Kgm^{-3}]
A Heat transfer area	[m^2]	σ Surface tension	[$N m^{-1}$]
L Channel section length	[m]	μ Viscosity	[Nsm^{-2}]
G Mass flux	[$Kg\ s^{-1}$]	ϕ Two-phase multiplier	[-]
H_v Enthalpy of vaporization	[$KJ\ Kg^{-1}$]	Subscripts	
m Refrigerant mass	[Kg]	g Gas phase	
p Pressure drop	[Pa]	l Liquid phase	
Q Heat flux	[W]	l-g Liquid-gas (two-phase)	
q Surface heat flux	[$W\ m^{-2}$]	m Mean	
x Vapor quality	[%]	cr Cross section	
y Number of channels	[-]	v Vapor phase	
T Temperature	[°C]	tt Turbulent two-phase flow	
S Slip-ratio	[-]	Dimensionless numbers	
v Specific volume	[$m^3\ kg^{-1}$]	Fr Froude nr. ($v^2/D.g$)	[-]
f_m Pierre friction factor	[-]	Re Reynolds nr. ($\rho.v.D/\mu$)	[-]
Δp Pressure drop	[Pa]	We Weber nr. ($\rho.v^2.D/\sigma$)	[-]
z Section location	[-]	Nu Nusselt nr. ($\alpha.D/\lambda$)	[-]

of 85%. Also a significant lower fraction is observed in the smallest rounded tube with a diameter of 1mm than in the classical tubes.

Comparison between correlations and the corresponding experimental data indicates a good agreement for the total refrigerant charge in evaporator.

Tubes with internal diameters (D_h) less than 3mm, the surface tension forces have much greater influence than for the larger diameter tubes. From the pressure drop models that have been investigated, the Pierre model adjusted by Grnryd[14] indicates the best approximation in which the deviation is about 20% where part of deviation is probably due to the measuring difficulties, mentioning that the advantage of using the small diameter can be reached as the pressure drop is minimized, that in turn increase the system efficiency.

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