ISSN 1392 - 1207. MECHANIKA. 2011. 17(6): 665-673

Low charge transport refrigerator (I). Refrigerant charge and strategies of charge reduction

L. Vaitkus
Kaunas University of Technology, K. Donelaicio g. 20, 44239 Kaunas, Lithuania, E-mail: liutauras.vaitkus@ktu.lt

crossref http://dx.doi.org/10.5755/j01.mech.17.6.1012

1. Introduction

For a long time, subjects dealing with the refrigerant charge in equipment were not a priority. For practical purposes a common estimation was to assess that 50% of the evaporator's volume is filled with liquid. Things started to change after the Montreal's protocol (1987) highlighted the destructive influence of some refrigerants on the ozone's layer, and the Kyoto's protocol (1997) underlined that most of refrigerants considerably contribute to the greenhouse effect. Refrigeration units cannot avoid refrigerant leaks, and can thus have a harmful impact on the environment. Therefore recently studies on refrigerant charge reduction have become more numerous.

The charge reduction is a major design objective for refrigerating equipment using hydrocarbons because of their flammability. However, when trying to reduce the atmospheric emissions, the restriction in charge should be also adopted in systems operating with halogenated refrigerants since lower charge per system leads to lower annual emissions. With the low charge systems the double positive effect is gained. Not only annual emissions from systems during their use are reduced, but also greenhouse gas (GHG) stock build up is prevented, which will reduce 'End of Life' emission in the future. Therefore, considering the environmental challenges the refrigerant charge minimization is one of the most important targets for refrigeration and air conditioning (RAC) applications.

However, any measure reducing refrigerant emissions that also decreases system energy efficiency must be cautiously estimated taking into account GHG emissions related to energy use. If energy usage of refrigerating or air conditioning system is included in a comparison, the balance of such a measure can be negative. Therefore, according to Harmelink et al, [1] any system adaptation should only be carried out if it results in lower Total Equivalent Warming Impact (TEWI).

The object of this study is refrigerating systems, used in refrigerated bodies for delivery vans to be used for the distribution of refrigerated goods (vegetables, meat, ice cream etc.) to the end customers. Such refrigerated bodies are usually built with an integrated refrigerating plant with eutectic system. The information and recommendations available from the previous studies are analyzed primarily considering their applicability in such system.

The term 'Eutectic System' refers to a refrigeration system that uses the phase change of a liquid medium to absorb and dissipate large amounts of thermal energy while remaining at a constant pre-arranged temperature. Phase change occurs when we freeze a solution solid by removing its heat or as it thaws into a liquid again while it absorbs heat. The eutectic solution is stored in the steel or aluminium tank (eutectic plate) and acts like a renewable ice block, freezing solid during the refrigeration run cycle and thawing during off periods.

Such a system has some advantages. They operate with big 'hold-over' OFF periods due to the thermal mass of the stored eutectic solution and because of the eutectic's phase change. The cold storage is used for the maintenance of a sufficient low temperature during distribution, where no access to electric supply is available, and the refrigerating plant is not working. Also these plants can generate cold during the night, with the vehicle in garage, by connecting to the electrical supply net. Operation during the cooler periods (the night) ensures lower condensing temperature, which increases system efficiency.

The storage temperature in the goods must not exceed –18°C and the crystallization temperature of the eutectic mixture is usually –33°C. These levels of temperature mean that the refrigerating compressor is working under very high compression ratio and utilization of energy is relatively poor. Condensing temperature is usually about 30°C; evaporation temperature during the crystallization of eutectic mixture is in the range of –43 - –45°C, but at the end of pull down cycle it goes below –50°C and may even reach –57°C. For R507A refrigerant this will give the compression ratio above 24. The stationary refrigerator for such temperatures would use two-stage refrigerating system with intercooler. However in transport refrigerating systems the space and weight are limiting factors and one-stage systems are used instead.

Further a review of studies considering refrigerant charge and charge reduction is presented.

2. Sensitivity of refrigerating system performance to charge levels and optimal charge

One of important aspects is sensitivity of refrigerating system performance to charge levels. Grace et al. [2] investigated the effect of refrigerant charge on steady-state system performance and identified parameters sensitive to charge level for leak detection in vapour compression refrigeration systems. The investigations were performed on a small 4 kW nominal cooling capacity vapour compression water-to-water chiller equipped with plate type heat exchangers. Refrigerant R404A was used as a working fluid. The system was equipped with thermostatic expansion valve (TXV). The liquid receiver was not used. The results indicate that the chiller could operate over a wide range of charge levels, 25% below to 25% above the design value without significant impact on its performance. Outside this range, the performance was found to be strongly dependent upon the charge level - at charge levels below 25% undercharge the cooling capacity falls very rapidly and at 50% undercharge it drops down to 50% of its maximum value. At overcharge levels greater than 25% the cooling capacity also begins to fall slowly, dropping by 7% from its maximum value at 40% overcharge conditions. The impact of charge level was much more pronounced on the condensing temperature and pressure than on the evaporating temperature and pressure. At charge levels below 25% of design value the system operated with incomplete condensation. Moving from 25% undercharge to 25% overcharge provided only a small change in the cycle on the P–h diagram.

Tassou and Grace [3] focused their article on fault diagnosis and refrigerant leak detection in vapour compression refrigeration system. The test facility used for development and validation of the fault diagnosis system was identical to analyzed in [2]. It was found that the coefficient of performance (COP) of this system was relatively constant at its maximum across a broad range of charge levels and that discrimination is required at charge levels differing by more than 33% from the nominal. The sensitivity to refrigerant charge levels is a function of the system considered and liquid-to-liquid systems are less sensitive to refrigerant charge than air-to-air systems. The developed fault diagnosis and leak detection system shows an undercharge fault when the system charge falls below 33% of nominal charge and an overcharge fault when the charge level is above 33% of nominal charge. Superheat out of evaporator was used to diagnose undercharge and subcooling out of condenser was used to diagnose overcharge.

Choi and Kim [4, 5] investigated the influence of refrigerant charge on the performance of heat pumps with electronic expansion valves (EEV) and capillary tube. Water-to-water heat pumps without liquid receiver were analyzed. The results indicate that the variation of the capacity for the EEV system with respect to refrigerant charge was less pronounced than that for the capillary tube system. With the EEV system, the variations of the capacity were almost negligible as refrigerant charge was altered from -10% to +20% of full charge. With the rise of the condensing temperature, the optimum charge amount of the EEV system needs to be increased. For such system the COP was relatively insensitive to refrigerant charge - for R407C system maximum reduction of COP at charge conditions from -20% to +20% of full charge was 3.6%.

The effect of refrigerant charge level on air-conditioning systems was also examined by Goswami et al. [6]. The authors concluded that charge level has a significant effect on the performance of air-conditioning systems at levels below 80% of normal. For a charge level of 90% of normal, the effect on COP and cooling capacity was found to be negligible.

Farzad [7] also studied the effect of improper charging on air conditioner performance for a short tube orifice, a TXV and a capillary tube. He found that the units using TXV were less sensitive to charge amounts. In the -15% to +10% charge range the capacity and efficiency of TXV unit was relatively constant (max. capacity and efficiency degradation was within 3%).

Corberán et al. [8] presented charge optimization study of a reversible water-to-water propane heat pump. The system was equipped with scroll compressor, brazed plate condenser and evaporator and TXV. The liquid receiver was not used. It was found that the COP has a clear maximum with approximately the same value of charge for both cooling and heating conditions, the maximum being more pronounced under cooling conditions. Thus a reversible unit charge optimised at one operating condition will be also at optimum under the reverse condition.

Björk and Palm [9] investigated performance of a domestic refrigerator under influence of varied refrigerant charge, expansion device capacity (EDC) and ambient temperature. Household refrigerator with SLHX and low pressure liquid accumulator (integrated in evaporator) was analyzed. EEV with stepper motor + capillary tube combo was used as expansion device in order to have variable expansion device capacity. It was found, that the energy consumption had a flat and wide minimum for certain combination of EDC and charges. Low sensitivity to charges (flat minimum) was explained by evaporator accumulator, which acts as a buffer that protects the system from either superheat or a cold suction line. It was also suggested that SLHX should insignificantly increase the charge due to lowered evaporator inlet quality and the dry-out point moving closer to the evaporator outlet.

Cho et al. [10] investigated effects of refrigerant charge amount on the performance of a typical transcritical CO_2 heat pump in cooling mode operation. The test system consisted of a variable speed scroll compressor (heating capacity 4.5 kW), finned-tube gascooler and evaporator, and an expansion device (EEV driven by a stepping motor). The system characteristics of the CO_2 system were compared with those for the R22, R407C and R410A systems. Fig. 1 represents the COP ratios of the compared systems with deviation from optimal charge. At undercharged conditions the CO_2 system showed the largest reduction in the COP. The cooling COP of the R410A system showed the largest drop at +5% of optimal charge. For the CO_2 system, the reduction of COP was more significant at undercharged conditions than at overcharged conditions. For -20% of optimal charge, the COPs for the R22, R407C and CO_2 systems decreased by 4, 8 and 25%, respectively, and for -10% of optimal charge, those reduced by 2, 5.5 and 10%, respectively. However, for +5% of optimal charge, the R22, R410A, R407C and CO_2 systems showed reductions of the COP by 2, 5.5, 3, and 2%, respectively. The CO_2 system showed higher performance sensitivity to refrigerant charge than that other compared systems.

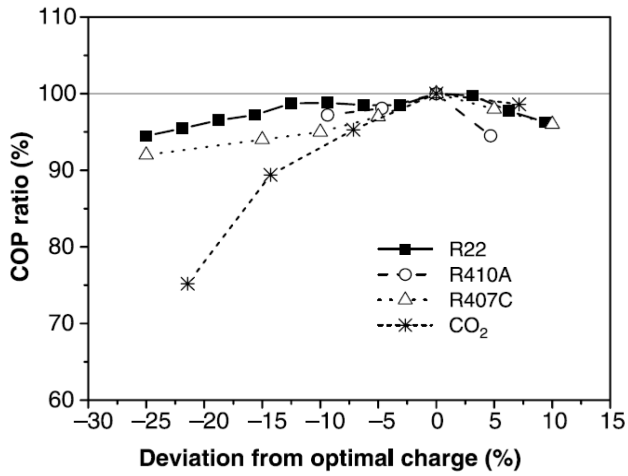

Fig. 1 COP ratios of the R22, R410A, R407C and CO_2 systems with deviation from optimal charge [10]

Rozhentsev [11] investigated a low-temperature Joule–Thomson refrigerating machine, using a non-azeotropic mixture of refrigerants. The machine was equipped with a single-stage hermetic compressor, operated at the temperature level of –75°C and was charged with the working non-azeotropic mixture which comprised of two components with considerably different thermodynamic properties. The behaviour of such refrigerating machine is found to be essentially different from the behaviour of refrigerating machines working by use of pure refrigerants or azeotropic mixtures. The amount of the minimum acceptable mixed refrigerant mass charge for the machine has been found. Under the mass charges below the minimum one, the temperature and power performance of the mixed refrigerant refrigerating machine are considerably higher than the designed ones and those operating modes are taken as inadmissible.

Dmitriyev and Pisarenko [12] suggested a simple correlation to calculate the optimum charge in a domestic refrigerator in which the evaporator and condenser internal volumes are the only parameters:

$$G_r = 0.41V_e + 0.62V_c - 38$$

where G_r is the refrigerator charge in grams, V_e the evaporator internal volume in ml and V_c the condenser internal volume in ml. They found that the COP was more sensitive to over than undercharging. They also found that the optimum charge was independent on the ambient temperature. The correlation was designed for refrigerant R12. However, Björk and Palm [9] tested the correlation for Isobutane household refrigerator multiplying the result with the density ratio of Isobutane to R12 (0.41 liquid/liquid). They have found that suggested optimum charge was an overcharge of only 15% to the nominal charge. Therefore the correlation may be used as a first estimation of optimum charge for household refrigerator.

Vjacheslav et al. [13] proposed rationally based algorithm to evaluate optimal mass charge into refrigerating machines. The model was created for refrigerating system without liquid receiver with capillary tube expansion device. It takes into account the major components of the refrigerating system (condenser, evaporator, expansion device and compressor). The calculated results give an identical trend to those of experimental data for the systems with capillary tube expansion device.

Ratts and Brown [14] presented an experimental analysis of the effect of refrigerant charge level on a cycling-clutch, orifice-tube (CCOT) automotive refrigeration system. The system was equipped with liquid separator – accumulator after the evaporator. Thermodynamic losses were quantified as a function of the refrigerant charge level. The experimental results show that the system is more efficient as the refrigerant charge level decreases. This is accomplished at the expense of increased refrigeration temperature and decreased refrigeration capacity.

3. The charge distribution and refrigerant mass measurement

Another important aspect is refrigerant mass measurement and charge distribution. Most of the refrigerant charge is in the form of liquid inside the unit, and depends on the geometry of the evaporator and condenser, on the volume of the liquid line, on the evaporation and condensation temperatures, on the subcooling, and on the amount of oil in the compressor.

Ding et al. [15] presented quasi on-line measurement method (QOMM) for measuring refrigerant mass distribution inside a refrigeration system. The method is combination of liquid nitrogen method (LNM) and on-line measurement method (OMM). Compared with LNM, QOMM can accelerate the measurement process. The results showed that the maximal prediction deviation of the refrigerant charge in the whole refrigeration system is 1.7%. The article also gives case study using the new measurement method. An R410A inverter air conditioning system was analyzed under steady state conditions. The cooling capacity of the system was 7.1 kW, power consumption of the compressor was 1.7 kW, and the refrigerant charge was 2 kg. The refrigerant in the condenser, the evaporator and the compressor were 50.0±3.4%, 16.0±1.3% and 14.0±2.3% of the whole charge, respectively (the inner volume of the analyzed condenser was 2.4 times larger than that of the evaporator).

Björk [16] presented refrigerant mass measurement technique based on the quick closing valves technique and the equations of state. The system was computer automated and can operate autonomously. The refrigerating system is subdivided into various control volumes by quick-closing valves, in order to trap the refrigerant. Each control volume is then expanded into a tank large enough to achieve a superheated state. When thermodynamic equilibrium is reached, the temperature and pressure in the tank are measured and the charge is calculated. This method is adapted to measure low quantities of refrigerant, but is not so practical for larger quantities.

Björk and Palm [17, 18] used this technique for the investigation of refrigerant mass charge distribution in a domestic refrigerator. These works provide experimental data of the charge distribution in a capillary tube throttled cooling system under varied load conditions. It was found that the condenser and compressor mass charges increased whereas the evaporator charge decreased upon increased thermal load. These trends confirm the simulations from Kuijpers et al. [19]. The largest charge variation is found in the evaporator - from the lowest to the highest thermal load the charge in this component decreases more than 30%. The charge increase in the condenser and in the compressor was explained by the increasing system pressure. The charge decrease in the evaporator cannot solely be explained by an increased inlet quality and increased mass flow and was partially caused by a decreased charge in the accumulator. The results highlight one important feature of the accumulator: it acts as a charge buffer from which refrigerant is displaced to other parts of the system at increased thermal load without starving the evaporator.

Charge distribution in a 5 kW heat pump using propane as a working fluid was presented by Primal et al. [20] (Fig. 2). The experimental system had no liquid receiver, used plate heat exchangers and EEV as expansion device. Optimal charge for the system was 300 g.

Primal et al. [21] also investigated propane heat pump designed for low refrigerant charge. The experimental system was equipped with mini-channel aluminium evaporator and condenser. The total internal volumes of the aluminium heat exchangers were less than 50% of those of the plate heat exchangers in [20]. It was found that the amount of refrigerant charge in the heat pump with the mini channel heat exchangers was about 200 g, comparing to 300 g for system with plate type heat exchangers. The heat pump was tested with different heat source temperatures and with a constant heat sink temperature. The refrigerant quantity in the system was varied for each heat source and heat sink temperature combination and the COP of the system was determined. The superheat was independent of the refrigerant charge and was kept between 4 and 6.5 K. The sub-cooling at the condenser outlet was allowed to increase for higher charges and therefore was strongly coupled to the system charge. Optimum charges for each heat source temperature correspond to a subcooling of 4–5 K.

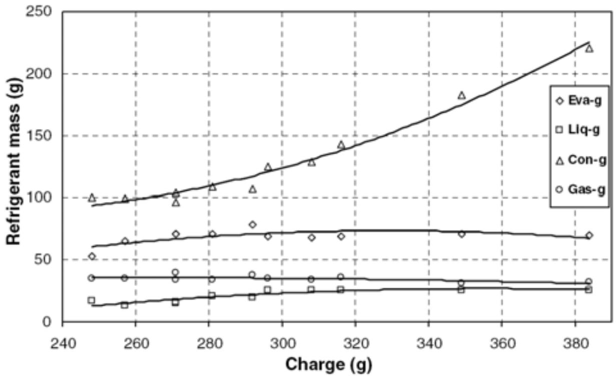

Fig. 2 Charge distribution in heat pump system as a function of total charge (from [20])

It was found that the COP was more or less constant when the charge was above a certain minimum level. Below this level, the COP dropped significantly. As for the COP the capacity is more or less constant as long as the charge is above the optimum charge. The capacity and COP reduction for lower refrigerant charges is caused by the decrease of the evaporation temperature. The reason for this decrease is the 'starvation' of the evaporator.

Charge distribution tests showed that the amount of refrigerant in the evaporator (volume 376 cm^3) was almost same for all four tests (23 – 26 g), even though the evaporation temperature varied from –16.5 to +4.5°C. Also, the amount of refrigerant in the liquid line was the same in all four experiments (23 – 24 g). The amount of refrigerant in the condenser (volume 437 cm^3) varied from 69 to 93 g, showing an increase of 24 g in spite of the fact that the condensing temperature was almost constant. It was concluded that use of mini channel heat exchangers reduces considerably refrigerant charge in heat pumps and refrigerating systems.

Similar results were obtained by Corberán et al. [8] who used IMST-ART software in order to examine the distribution of the charge inside the refrigerating system. It was found that most of the charge of the unit (about 50%) is in the condenser and mainly at its final part and outlet port. The evaporator contains another significant part of the charge (about 14%). Nearly 30% of the total charge is found dissolved in the lubricant oil in the compressor. The amount of refrigerant in the oil is for the most part dependent on the evaporation pressure and on the oil temperature. Therefore, the amount of charge in the compressor is, a function of the evaporation temperature. The authors suggest that since most refrigerant inventory corresponds to liquid refrigerant and that its density is fairly constant, the total mass in all components of the system, with the exception of the condenser, is non-dependent on the variation of the charge. The condenser stores the extra charges in the system varying the subcooling. With the increase of charge in the system the amount of charge in the evaporator also slightly increases, since the increase in subcooling is leading to a decrease in the vapour quality at the inlet of the evaporator.

Since most of the charge of the unit is in the condenser, some charge reduction studies are focused on different low charge heat exchanger designs. Cavalini et al. [22] investigated performance of large capacity propane heat pump with low charge heat exchangers. Shell-and-tube heat exchangers using minichannels were tested along with conventional brazed plate heat exchangers. It is shown that a 100 kW heat pump without a liquid receiver can be run with around 3 kg of propane using a plate condenser and a plate evaporator. Using the minichannel condenser, around 0.8 kg (25% of the total mass) reduction can be obtained with a negligible performance loss.

Hrnjak and Litch [23] presented experimental results from a low charge and compact ammonia chiller. It was equipped with an air-cooled condenser with microchannel aluminium tubes and a plate evaporator. Charge, heat transfer, and pressure drop measurements were taken for a serpentine flat macro tube and parallel flow microchannel tube condensers. Condenser performance was quantified in terms of heat capacity, refrigerant and air side pressure drops, heat transfer coefficient values, and refrigerant inventory. Comparisons show the superiority of the microchannel design. Microchannel ($Dh = 0.7$ mm) charge is an average of 53% less than for the serpentine ($Dh = 4.06$ mm). The ''microchannel'' condenser charge per capacity ratio is around 76% less than for the ''macrochannel'' serpentine condenser.

Another important observation which can be found in [23] considers connection between subcooling and charge. According to the article, the liquid subcooling is a large contributor to total charge. The relative predicted charge contributions from the refrigerant phase zones for the data point with the highest liquid subcooling tested are 0.5% from superheated vapour, 29.2% from the two-phase region, and 70.3% from liquid subcooling. From the data point with the lowest liquid subcooling, the contributions are 0.5% from superheated vapour, 60.1% from the two-phase region, and 39.4% from liquid subcooling. Even though the subcooling region is only a small portion of the total tube length, it holds most of the total inventory. Thus it is advantageous to reduce subcooling not only for increased heat transfer, but to reduce refrigerant charge.

4. Calculation of the refrigerant charge

The calculation of refrigerant charge in a single phase pipe (liquid or vapour), is easy. The charge in a receiver depends on the shape and the filling ratio. The major difficulty is evaluation of the two-phase components (heat exchangers). There are a lot of works, analyzing various aspects of two-phase flow. For example [24] provides an analysis of the heat transfer for condensation process and [25] analyzes pressure changes of condensing annular flow and [27-29] analyzes charge distribution.

The two-phase flow can adopt various flow patterns. To determine the fluid charge, it is necessary to know the proportion of vapour and liquid at any location. This proportion differs from the volume fraction, mass flow rates and densities. In order to evaluate the refrigerant charge, one must know the void fraction or the averaged volumetric fraction of gas in two-phase flow. At the i -th position the void fraction can be defined as

$$\varepsilon_{vi} = dV_{v,i} / dV_i = A_{v,i} / A_i \tag{1}$$

where dV_i is an elementary volume, $dV_{v,i}$ is volume of gas in elementary volume, A_i is cross-section, $A_{v,i}$ is the fraction of the cross section occupied by gas.

The mass contained in the entire volume V (the total charge) is

$$M = \int_0^V \left[\varepsilon_v \rho_v + \left(1 - \varepsilon_v\right) \rho_l \right] dV \quad .$$

where ρ_{vi}, ρ_{li} are vapour and liquid density (kg/m^3).

To calculate the refrigerant charge it is necessary to know the void fraction. It is usually linked to vapour quality x (kg$_v$/kg), which is the fraction of total mass flow which is in the gas form.

The average vapour and liquid velocities u_{vi} and u_{li} are given by

$$u_{vi} = \frac{G_{vi}}{\rho_{vi} \varepsilon_{vi}} \tag{2}$$

$$u_{li} = \frac{G_{li}}{\rho_{li} \left(1 - \varepsilon_{vi}\right)} \tag{3}$$

where G_{vi} , and G_{li} are vapour and liquid mass fluxes (kg/(m^2 s)) at a given position i .

The slip ratio is

$$S_i = u_{vi} / u_{li} \tag{4}$$

Using these equations the void fraction is connected to the quality

$$\varepsilon_{vi} = \left(1 + S_i \frac{(1 - x_i)}{x_i} \frac{\rho_{vi}}{\rho_{li}} \right)^{-1} \tag{6}$$

The thermodynamic terms can be found when temperature and pressure are known. The slip ratio is determined using correlations. The simplest approach is to consider the gas–liquid mixture as a homogeneous flow, where the average liquid velocity is equal to the average gas velocity ($S_i = 1$). Then a void fraction is equal:

$$\varepsilon_{\text{hom}} = \left(1 + \frac{(1 - x)}{x} \frac{\rho_v}{\rho_l} \right)^{-1} \tag{7}$$

This relation serves as a reference for other existing correlations.

Premoli et al. [26] assumed an annular flow in tubes and developed the correlation for the slip ratio:

$$S = 1 + E_1 \sqrt{Y / \left(1 + Y E_2\right) - Y E_2}$$

with

$$E_1 = 1.578 \,\mathrm{Re}_l^{-0.19} \left(\rho_l / \rho_v \right)^{0.22}$$

$$E_2 = 0.0273 \,\mathrm{We}\, \mathrm{Re}_{lo}^{-0.51} \left(\rho_l / \rho_v \right)^{-0.08}$$

where $Y = \varepsilon_{hom} / \left(1 - \varepsilon_{hom}\right)$; $\mathrm{We} = G^2 D_h / \left(\sigma \rho_l\right)$ is Weber number; $Re_l = G D_h \left(1 - x\right) / \mu_l$ Reynolds number for liquid; $Re_{lo} = G D_h / \mu_l$ is Reynolds number for liquid only; D_h is hydraulic diameter, m; σ is surface tension, N/m; G is mass flux, kg/(m^2 s); μ_l is liquid dynamic viscosity, Pa s.

According to Rice [27], Premoli correlation is one of the most effective for refrigerating systems. According to Kuijpers et al. [28], only the Premoli correlation gives satisfactory results for the charge calculation in a condenser, while homogeneous model underestimates the charge in a condenser. For Farzad and O'Neal [29], models depending on the mass flux, such as Premoli correlation, predict a refrigerant charge that is more significant and is more effective.

5. Superheat in evaporator and subcooling in condenser

Superheat in evaporator for TXV and EEV controlled systems does not depend on refrigerant charge, except of the cases of significant undercharge (below 25% of optimal charge [2, 3]). At 50% undercharge [2] the superheat rises to approximately 10 K. The high superheat at low charge levels is a result of insufficient refrigerant in the evaporator which leaded to larger evaporator area devoted to superheating. Zero superheat would be optimal, but TXV and EEV are not able to work at such settings.

Considering the influence of refrigerant subcooling in condenser, different sources give contradictory information. In the literature it is generally taken for granted that for optimal operation there should be no subcooling. According to Stoecker [30], subcooling is not normally desired, since it indicates that some heat transfer surface that should be used for condensation is used for subcooling. This reduces the cooling, and condensing pressure increases. Vjacheslav et al. [13] in their algorithm for optimal mass charge evaluation also assumed that optimal conditions corresponds to the state where the refrigerant is fully condensed, or minimal subcooling just enough to ensure stable operation.

However, in the works [3-5], [8], [10] and [21] we have different results. In [3] it was found that subcooling at optimal charge was more than 5 K. At -33% undercharge the subcooling was still above 2 K. Choi and Kim [4] also found significant subcooling of 4 – 5 K at optimal charge for EEV controlled heat pump with R22 refrigerant (Fig. 3). The subcooling for similar system with R407C [5] was a bit smaller, but still significant 2 – 3.5 K. Comparing [4] and [5] we can see that for different refrigerants subcooling at optimal charge is different – for R22 the subcooling is almost two times bigger, than for R407C. Primal et al. [21] also found that optimum charges for each heat source temperature correspond to a sub-cooling of 4–5 K. From the Fig. 3 we can also see, that even at -20% undercharge we still have full condensation and 1 K subcooling.

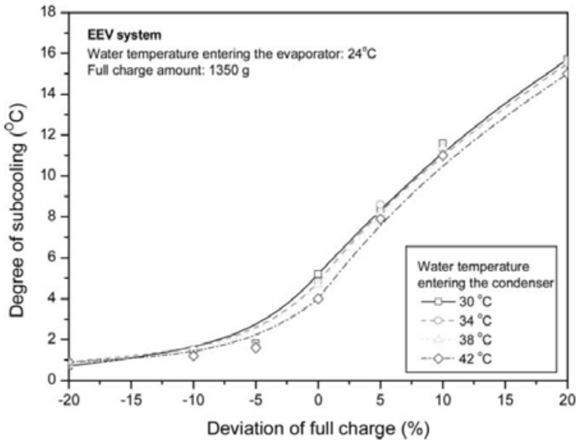

Fig. 3 Variations of subcooling with refrigerant charge in EEV controlled R22 heat pump system [4]

Jensen and Skogestad [31] also investigated optimality of subcooling. For analyzed ammonia refrigerating system contrary to popular belief they found that subcooling by 4.66 K reduces the compression work by 1.74% comparing to the case with saturation out of the condenser. The condensing pressure increases, but it is compensated by reduction in flowrate. According to their estimation the improvement of 2% would be larger if the pressure drop in the piping and equipment was accounted for in the model.

The connection between charge, subcooling and performance for the unit without liquid receiver is analyzed in deep by Corberán et al. [8]. According to their analysis the influence of the charge in such system is attributable to the fact that extra refrigerant must find space inside the refrigerant circuit. The only way for the refrigerant cycle to react when the charge is increased is by increasing the fraction of the condenser which is full of liquid and therefore increasing the subcooling. To increase the subcooling, an increase in the heat transferred from the condenser to the secondary fluid must occur; hence, the condensing temperature and pressure must increase. The increase in subcooling at the highest charges is due to increase of the saturation temperature at higher condensation pressures and not to a decrease in the outlet refrigerant temperature. The increase in the subcooling always has a positive effect on the COP. On the other hand, the necessary increase in condensation pressure to produce the higher subcooling produces a decrease in the COP. Therefore, one effect is positive while the other is negative; this leads to the observed maximum in COP. The rapid decrease of performance, both capacity and COP at low charges, when the charge is further reduced is due to the appearance of bubbles at the inlet of the expansion valve.

The subcooling increases while the charge is increased from the minimum charge (which corresponds to saturation outlet). Low subcooling is obtained with an almost undetectable increase in condensation pressure, this leading to an effective increase in COP. However, when small temperature approach is attained, the increase in subcooling requires a proportional increase in the condensation temperature, quickly degrading the COP. As a result the optimum subcooling is mostly imposed by the temperature approach at the outlet of the condenser for 0 K subcooling. The optimum subcooling of a refrigeration unit will depend mainly on the condenser performance. Higher number of transfer units (NTU) will lead to a lower value of the optimum subcooling since the condensation temperature will become closer to the water temperature. This conclusion is in line with [31] according to which the optimal degree of sub-cooling becomes smaller as the heat transfer (UA-value) is increased. With an infinite heat transfer area the subcooling is not optimal.

6. Strategies of charge reduction

Palm [32] investigated refrigerating system with minimum charge of refrigerant. In addition to mini-channel heat exchangers, his suggestions for charge reduction are (i) use of indirect system, (ii) use of low-pressure receiver in the suction line rather than the common high pressure receiver in the liquid line, (iii) capillary tube used as expansion device, (iv) use compressor with small internal volume and small oil charge, (v) use non-miscible oils. When using secondary refrigeration, the power consumption increase (indirect TEWI) is compensated by the charge reduction (direct TEWI).

The influence of evaporator's supplying mode was investigated by Vrinat et al. [33]. They have found that the liquid filling rate of flooded evaporators would be a little higher than that of direct expansion fed evaporators with a respectively, 25% and a 10% volume filling. A strict concern of charge reduction will result in preferring the evaporator's direct expansion supplying.

Poggi et al. [34] presented a review of the refrigerant charge studies in a refrigerating plant and evaluation of the influence of the refrigerant charge on COP and on the cooling capacity.

7. Conclusions

Because of the influence of HCFC and HFC refrigerants to the direct greenhouse effect, it is important to reduce their atmospheric emission. Therefore a reduction of the charge in the system is significant goal to achieve.

Summarizing the available works considering refrigerant charge we can see that the refrigerant mass charge in the system is linked to the system performance. Independently on the system, too little charge will cause draining of the condenser into the evaporator, two-phase feeding of expansion valve (incomplete condensation) and inadequate filling of evaporator.

As long as the charge is sufficient to ensure the complete condensation, the effect of further charge increase depends on refrigerating system used. Most of the discussed works investigated the systems without a receiver. Björk and Palm [17, 18] and Ratts and Brown [14] investigated refrigerating systems with a low side receiver, but because of different control systems application of their results to transport refrigerating unit with eutectic plates is limited.

If the system allow liquid subcooling in condenser (no high pressure liquid receiver present in the system), the unit performance depends on the charge. For such system it is possible to determine refrigerant charge, which maximizes refrigerating system COP. Such charge is called "optimal charge”. Some COP increase over the case of complete condensation (saturated outlet) is achieved through liquid subcooling in condenser. The expected increase is about 2% plus some positive effect from lower pressure drop in the piping and equipment. The positive effect depends on refrigerant used, type of condenser (air cooled or water cooled), UA-value of condenser, hydraulic losses in evaporator etc.

However, this COP increase is not free. The liquid subcooling is a large contributor to total charge since the subcooling region holds significant fraction of the total charge inventory. Experimental proof for this statement may be found in various sources, for example in [2-7]. In these works different systems were analyzed but similar conclusions were obtained. Significant performance degradation is observed when the charge reduction exceeds 25% from the optimal charge. However, charge reduction up to 20% is possible without significant deterioration of performance, and at 10-15% of charge reduction the effect on performance is negligible for systems with TXV or EEV. It is safe to suggest, that incomplete condensation occurs only when charge reduction is 20 - 25% from optimal and all (rather insignificant) efficiency increase is the effect of liquid subcooling in condenser.

Thus if the low charge system is on target, it is advantageous to reduce subcooling. Actually, when selecting a charge for low charge system one should not try to optimize the charge considering maximum COP, but rather just ensure complete condensation. This will cause some performance degradation, but this may be compensated by other means with lower influence to charge. Such system could be the system with high pressure liquid receiver, in which liquid subcooling is not allowed by design. If liquid level in the receiver is high enough to avoid the appearance of bubbles at the inlet of the expansion valve, the variation of charge should only affects the level of liquid refrigerant in the receiver and, therefore, the unit performance becomes non-dependent on the charge.

The big attention should be paid to condenser, where the biggest part of the charge is located. The microchannel condenser is a good choice for low charge system.

Considering vapour superheat in evaporator, the relationship to refrigerant charge is weak. In the most of discussed systems the superheat was determined by the control system rather than by the charge. The exception is the case of significant undercharge when the inadequate feeding of evaporator causes the increase of superheat. With the increase of superheat, the charge in evaporator decreases, but this can not be considered a charge reduction strategy since the negative effect on performance is more pronounced. Actually it would be preferable to have a system with no superheat. Potentially this could be ensured by the low-charge system with low pressure receiver, recommended by Palm [30].

Considering the expansion device, both the systems with TXV and EEV are considered less sensitive to refrigerant charge, when compared to system with capillary tube. At the given conditions there are no significant differences in the charges of TXV and EEV controlled systems. The one should also have in mind that all here discussed EEV or TXV systems were analyzed at static conditions (constant condensing and evaporation temperature). The dynamic mode of operation may introduce additional factors influencing refrigerant charge.

Also, most of the discussed systems were not equipped with SLHX. While the SLHX will cause some charge increase in evaporator due to lower vapour quality after expansion, it also has some positive effects. First of all, the SLHX may increase the system efficiency (COP). It also helps ensuring proper liquid subcooling before the expansion device. This allows decreasing the diameter of liquid lines, which leads to lower charge. Therefore the SLHX should be carefully estimated when developing the low charge system.

Acknowledgements

Lithuanian State Studies Foundation and JSC Carlsen Baltic provided support for this work.

References

1. **Harmelink, M.; De Jager, D.; Kerssemeeckers, M.; Hendriks, C.; Van Brummelen, M.** 2002. TEWI: Total equivalent warming impact: A practical approach. Non-CO2 greenhouse gases. Millpress, Rotterdam pp. 627-632.

2. **Grace, I.N.; Datta, D.; Tassou S.A.** 2005. Sensitivity of refrigeration system performance to charge levels and parameters for on-line leak detection, Applied Thermal Engineering 25: 557-566.

3. **Tassou S.A.; Grace, I.N.** 2005. Fault diagnosis and refrigerant leak detection in vapour compression refrigeration system, International Journal of Refrigeration 28: 680-688.

4. **Choi, J.; Kim, Y.** 2002. The effects of improper refrigerant charge on the performance of a heat pump with an electronic expansion valve and capillary tube, Energy 27: 391-404.

5. **Choi, J.; Kim, Y.** 2004. Influence of the expansion device on the performance of a heat pump using R407C under a range of charging conditions, International Journal of Refrigeration 27: 378-384.

6. **Goswami, D.Y.; Ek, G.; Leung, M.; Jotshi, C.K.; Sherif, S.A.** 1997. Effect of refrigerant charge on the performance of air conditioning systems, Proceedings of the Intersociety Energy Conversion Engineering Conference 3 (Pt. 4): 1635-1640.

7. **Farzad, M.** 1990. Modelling the effects of refrigerant charging on air conditioner performance characteristics for three expansion devices. Dissertation Information Service, Texas A&M University, UMI, Ann Arbor, MI. 263p.

8. **Corberán, J.M.; Martínez,I.O.; Gonzálvez, J.** 2008. Charge optimisation study of a reversible water-to-water propane heat pump, International Journal of Refrigeration 31: 716-726.

9. **Björk, E.; Palm, B.** 2006. Performance of a domestic refrigerator under influence of varied expansion device capacity, refrigerant charge and ambient temperature,

International Journal of Refrigeration 29: 789-798.

10. **Cho, H.; Ryu, Ch.; Kim, Y.; Kim, H.Y.** 2005. Effects of refrigerant charge amount on the performance of a transcritical CO_2 heat pump, International Journal of Refrigeration 28: 1266-1273.

11. **Rozhentsev, A.** 2008. Refrigerating machine operating characteristics under various mixed refrigerant mass charges, International Journal of Refrigeration 31: 1145-1155.

12. **Dmitriyev, V.I.; Pisarenko, V.E.** 1984. Determination of optimum refrigerant charge for domestic refrigerator units, International Journal of Refrigeration 7: 178-180.

13. **Vjacheslav, N.; Rozhentsev, A.; Wang, C.** 2001. Rationally based model for evaluating the optimal refrigerant mass charge in refrigerating machines, Energy Conversion and Management 42: 2083-2095.

14. **Ratts, E.B.; Brown, J.S.** 2000. An experimental analysis of the effect of refrigerant charge level on an automotive refrigeration system, International Journal of Thermal Science 39: 592-604.

15. **Ding, G.; Ma, X.; Zhang, P.; Han, W.; Kasahara, S.; Yamaguchi, T.** 2009. Practical methods for measuring refrigerant mass distribution inside refrigeration system, International Journal of Refrigeration 32: 327-334.

16. **Björk, E.** 2005. A simple technique for refrigerant mass measurement, Applied Thermal Engineering 25: 1115-1125.

17. **Björk, E.; Palm, B.** 2006. Refrigerant mass charge distribution in a domestic refrigerator. Part I: Transient conditions, Applied Thermal Engineering 26: 829-837.

18. **Björk, E.; Palm, B.** 2006. Refrigerant mass charge distribution in a domestic refrigerator. Part II: Steady state conditions, Applied Thermal Engineering 26: 866-871.

19. **Kuijpers, L.J.M.; Janssen, M.J.P.; Verboven, P.J.M.** 1988. The influence of the refrigerant charge on the functioning of small refrigerating appliances, ASHRAE Transactions 94: 813-828.

20. **Primal, F.; Samoteeva, O.; Palm, B.; Lundqvist, P.** 2001. Charge distribution in a 5 kW heat pump using propane as working fluid. 16 Nordiske Kølemøde og 9, Nordiske Varmepumpedage 29–31, Københvn, p.299.

21. **Primal, F.; Palm, B.; Lundqvist, P.; Granryd, E.** 2004. Propane heat pump with low refrigerant charge: design and laboratory tests, International Journal of Refrigeration 27: 761-773.

22. **Cavallini, A.; Da Riva, E.; Del Col, D.** 2010. Performance of a large capacity propane heat pump with low charge heat exchangers, International Journal of Refrigeration 33: 242-250.

23. **Hrnjak, P.; Litch, A.D.** 2008. Microchannel heat exchangers for charge minimization in air-cooled ammonia condensers and chillers, International Journal of Refrigeration 31: 658-668.

24. **Šinkūnas, S.; Kiela, A.** 2011. Heat transfer for film condensation of vapour, Mechanika 17(2): 144-148.

25. **Saffari, H.; Dlir, N.** 2011. Effect of friction stress of droplets with film on prediction of pressure changes in condensing tubes, Mechanika 17(1): 50-56.

26. **Premoli, A.; Di Francesco, D.; Prina, A.** 1970. Una correlazione adimensionale per la determinazione delle densita` di micele bifasiche. In: XXV Congresso Nazionale ATI, Triete, Italie, septembre. 1970.

27. **Rice, C.K.** 1987. The effect of the void fraction correlation and heat flux assumption on refrigerant charge inventory predictions, In: American Society of Refrigerating and Air Conditioning Engineers Meeting, New York. 341-357.

28. **Kuijpers, L.; Janssen, M.; De Wit, J.** 1987. Experimental verification of liquid hold-up predictions in small refrigeration heat exchanger, In: XVII International Congress of Refrigeration, Wien, Austria. 307-315.

29. **Farzad, M.; O'Neal, D.** 1994. The effect of void fraction model on estimation of air conditioner system performance variables under a range of refrigerant charging conditions, Revue Internationale du Froid 17(2): 85-93.

30. **Stoecker, W.F.** 2004. Industrial Refrigeration Handbook. McGraw-Hill. 688p.

31. **Jensen, J.B.; Skogestad, S.** 2007. Optimal operation of simple refrigeration cycles Part I: Degrees of freedom and optimality of sub-cooling, Computers and Chemical Engineering 31: 712-721.

32. **Palm, B.** 2007. Refrigeration systems with minimum charge of refrigerant, Applied Thermal Engineering 27: 1693-1701.

33. **Vrinat, G.; Macchi, H.; Leducq, D.** 2000. Estimation de la charge en frigorige`ne dans les boucles frigorifiques basse pression (2e`me partie), Revue Ge´ne´rale du Froid. 4p.

34. **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: 353-370.

L. Vaitkus

MAŽOS DOZĖS TRANSPORTINIS ŠALDYTUVAS (I). ŠALDYMO AGENTO DOZĖ IR DOZĖS SUMAŽINIMO STRATEGIJOS

R e z i u m ė

Šiame darbe apžvelgiami šaldymo agento dozės šaldymo sistemoje tyrimai – sistemos jautrumas dozės pokyčiams, ryšys tarp dozės ir perkaitinimo bei peraušinimo, dozės matavimo ir skaičiavimo metodikos, žinomos dozės sumažinimo strategijos ir jų pritaikymo galimybė transportiniams šaldytuvams. Mažos dozės sistemoms tikslinga ne optimizuoti dozę, bet tik užtikrinti pilną kondensaciją. Tai gali būti sistema su aukšto slėgio resiveriu, mikrokanaliu kondensatoriumi ir TRV arba ERV maitinamu garintuvu.

L. Vaitkus

LOW CHARGE TRANSPORT REFRIGERATOR (I). REFRIGERANT CHARGE AND STRATEGIES OF CHARGE REDUCTION

S u m m a r y

This paper presents a review of refrigerant charge studies in a refrigerating system – sensitivity of system performance to charge levels, relationship of charge with superheat and subcooling, methodologies of charge measurement and calculation, existing charge reduction strategies and their applicability to transport refrigerating systems. For the low charge system one should not optimize the charge, but rather just ensure complete condensation. Such system could comprise of high pressure receiver, microchannel condenser and DX evaporator with TXV or EEV expansion device.

Received February 02, 2011
Accepted November 10, 2011