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Low charge transport refrigerator (II). Theoretical and experimental investigation

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1. Introduction

The first part of this work [1] presented a review of refrigerant charge studies and analysed general strategies of charge reduction. Also from the analysis of available experimental data it was found that for low charge system it is advantageous to reduce subcooling in condenser – the charge should just ensure complete condensation. Such system could be the system with high pressure liquid receiver, in which liquid subcooling is not allowed by design.

Further the applicability of general charge reduction strategies for refrigerating plant with eutectic system is analyzed together with some aspects specific to low temperature systems. Experimental results obtained for low charge eutectic systems are also presented.

2. System architecture consideration

General strategies 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 and proper receivers sizing, (iii) capillary tube used as expansion device, the diameters and lengths of pipes, (iv) use compressor with small internal volume and small oil charge, (v) use nonmiscible oils, (vi) use of compact, low-volume heat exchangers, preferably mini-channel heat exchangers, (vii) use evaporator's direct expansion supplying.

The recommendation to use indirect system can be explained by significant pipe lengths in split systems and high refrigerant amounts in pipes. Macchi et al. [2] reported that in the studied cases the ratio between the charge in the liquid pipes and the total charge was 60% for the direct expansion system and 40% for the flooded system. Small plants with split system have separate condensation groups and also often contain a significant part of their charge in the liquid pipes. Therefore minimization of liquid pipe diameter and length is one of necessary steps for charge reduction. With indirect systems the distance between system components may be minimal, which reduces refrigerant charge. Palm [3] gives another example, when the charge of indirect supermarket system is expected to be 11% of the charge of conventional centralized system. Here analyzed transport refrigerators are already made compact and implementation of indirect system will not offer further decrease of liquid line length.

Jensen and Skogestad [4] investigated optimal operation of simple refrigerating systems and compared different designs. In their analysis the "active" charge is defined as the total mass accumulated in the process equipment in the cycle (condenser, evaporator, compressor…) but excluding any adjustable mass in liquid receivers. It is assumed that refrigerant holdup change by filling or leaking in receiver with a constant active charge does not affect the operation of the cycle. Thus receiver makes operation independent of the total charge in the system. The two main cases where the receiver is placed are (i) on the high pressure side after the condenser and (ii) on the low pressure side after the evaporator. The superheat in evaporator is not optimal, but some subcooling in condenser is optimal contrary to popular believe. This claim is supported by theoretical and experimental investigation by Corberán et al. [5] as well as experimental data from Tassou and Grace [6], Choi and Kim [7, 8], Cho et al. [9] and Primal et al. [10] (see discussion in [1]).

Jensen and Skogestad [11] also investigated selection of controlled variables for simple refrigeration cycles. It was found that for ammonia refrigeration cycle a good policy is to have no subcooling. Further savings at about 2% are obtained with some optimal sub-cooling (larger savings are expected for cases with smaller heat exchanger areas). It was found that the best control strategy is to fix the temperature approach at the condenser exit (the difference between temperature of sub-cooled liquid before sub-cooling control valve and ambient temperature).

Palm [3] also recommends using a low pressure receiver in the suction line rather than the common high pressure receiver and capillary tube as expansion device in order to decrease refrigerant charge. Low pressure receiver with the correct charge should ensure operation of the system without superheat at the evaporator outlet and highest Coefficient of performance (COP). Capillary tube ensures that no charge is trapped in a liquid line. Such expansion device is cheap and requires no initial setting once correctly sized. Evans et al. [12] used the system with low pressure receiver and four capillary tubes as expansion devices in their multitemperature commercial refrigerator cabinet.

The low pressure receivers are often called accumulators. Parameters of available low pressure accumulators and their influence on system performance are investigated by Wang et al. [13, 14]. They experimentally and theoretically studied low pressure accumulator with an oil bleeding hole at the bottom of j-tube.

Such system with low pressure receiver may be very promising for eutectic systems. Still, its successful implementation requires development of the receiver with proper parameters. We tested such system with currently commercially available low pressure receivers and found the results not good enough – the measured system capacity and COP was lower comparing to results of baseline system with thermostatic expansion valve (TEV). The available low pressure receivers (accumulators) do not provide the outlet vapour quality close to saturated outlet, which would be required to ensure high efficiency. Since the hardware components for implementation of the system with low pressure receiver are not currently available, the traditional system with a high pressure receiver and TEV or electronic expansion valve (EEV) was chosen for low charge eutectic system development.

3. Features of eutectic systems and the baseline system

Most of the works, discussed previously investigate refrigerating system working at static conditions. As distinct from these systems, the eutectic systems always operate at nonstatic conditions. In this sense the eutectic system is similar to household refrigerator, but the evaporation temperature in case of eutectic system is varying in much wider range. Another difference in operation mode is that eutectic systems are built for maximal cooling capacity and sometimes operate nonstop through the night, while in household refrigerators the duration of compressors on-off cycle can be used as additional control variable.

The major tests for eutectic systems performance estimation is temperature pull-down test. During this test the warm refrigerator with the eutectic plate temperature equal to ambient temperature is turned on, and runs until the thermostat cut-out temperature is reached. In the analyzed refrigerators the eutectic mixture with -33°C crystallization temperature was used and the cut-out temperature (air) for summer conditions was -36°C. The pull-down time and energy consumption may be measured and performance estimated for different inside air temperature intervals. Still, when refrigerator is back to depot and turned ON during everyday operation, the eutectic solution in the plates is almost completely thawed and temperature of air in refrigerated space is close to -20°C. The period when average air temperature in refrigerated space goes from -20 to -33°C is considered as best indicating the everyday performance of the system

Usually it is required that the eutectic system was able to perform pull-down at 35°C ambient temperature for ‘north’ and at 38°C for ‘south’ applications. During the initial stage of pull-down test the temperature of eutectic plates is equal to ambient temperature and the evaporation temperature is lower than eutectic plate temperature by the certain temperature difference. At higher evaporation temperatures the heat rejection, condensing pressure and compressors current would increase a few times comparing to nominal operating conditions. In order to keep these parameters inside of operation envelope, the suction pressure in such system must be limited.

Most of eutectic systems are equipped with direct expansion (DX) evaporator with mechanical TXV and high pressure (HP) liquid receiver. Usually these systems also have suction - liquid heat exchangers (SLHX). Simplified diagram of such system is given on the Fig. 1.

The eutectic refrigerators chosen for the investigation and described in this paper are developed and produced as refrigerated bodies with an integrated refrigerating plant for delivery vans (permissible total weight 3.500 kg), to be used for the distribution of refrigerated goods (vegetables, meat, ice cream etc.) to the end consumers. The baseline systems (A) and (B) are traditional design, use R507A refrigerant and are equipped with scroll compressors with liquid injection through the discharge temperature control (DTC) valve. The mechanical outlet pressure downstream regulator (OPR) is used for maintaining a predetermined maximum suction pressure. This regulator is mounted on a suction line just before the compressor. The maximum operation pressure (MOP) is chosen experimentally during the pull-down test at maximal rated ambient temperature (35 or 38°C) and must keep the condensing pressure and compressors current within allowable limits. The specifications of the major components of both baseline systems are given in Table.

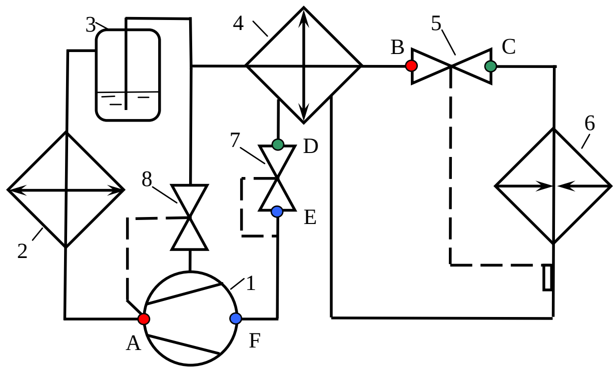

Fig. 1 Traditional eutectic refrigerating system; *1* – compressor, *2* – condenser, *3* – HP receiver, *4* – SLHX, *5* – TXV, *6* – evaporator (eutectic plates), *7* – OPR, *8* – DTC valve

Table

Specifications of the main components of systems

Components	Specifications	System			
		(A)	(B)	(C)	(D)
Refrigerant	Charge, kg.	4.1	6.0	2.4	2.9
Condenser	Inner volume, (l)	2.25	3.98	2.25	2.25
	Inner area, (m^2)	1.19	1.58	1.19	1.19
	Outer area, (m^2)	26.4	25.3	26.4	26.4
Evaporator	Tube, (mm)	Ø15×1	Ø15×1	Ø18×1	Ø15×1
	Total length, (m)	45	60	45	60
	Inner volume, (l)	5.97	7.96	9.05	7.96
SLHX	Liquid line, (mm)	Ø10×1	Ø10×1	Ø6×1	Ø6×1
	Length, mm	5.5	7.5	5.5	7.5
Receiver	Inner volume (l)	4.6	6.9	1.0	1.5

The experimental systems (C) and (D) were designed and realized in collaboration between Kaunas University of Technology and JSC Carlsen Baltic. The project was supported by Lithuanian State Studies Foundation. The objective of the project was to investigate the possibility of charge reduction for System (A) below 2.5 kg and for System (B) below 3 kg.

The initial specific charge of system (A) was about 2650 g/kW, and for system (B) was about 2817 g/kW. The specific charge of baseline systems is on the same level as specific charge of competing eutectic systems, but is significantly higher comparing to other refrigerating systems. Hrnjak and Litch [15] give compilation of specific charges, and according to their data for commercially available air-cooled ammonia chillers the specific charge is in a range of 125 – 160 g/kW. However, these systems can not be directly compared – the used refrigerants have different properties, the systems operate at different temperatures and have different designs of condensers and especially evaporators.

Since the control system was not defined in the specifications of the project, various control systems were considered. The most cost-effective solution would be a system with capillary tube as expansion device. As mentioned previously, prototype of such system was tested, but did not offer acceptable performance comparing to baseline system. According to our estimation the main obstacle here was the lack of suitable low pressure accumulator (with saturated outlet). Also, the capillary tube capacity ensuring effective operation of eutectic system at all the range of operation conditions could not be selected. This means that the additional valve is needed.

If we consider a system with a small liquid line diameter and additional control valve, we get a system similar to the one proposed by Barsanti [16]. The patent describes refrigerating system, which is claimed to offer the refrigerant charge reduction of 80%. The system has no thermostatic valve and the expansion is performed through the piping. The diameter of these pipes is much smaller comparing to liquid lines usually used. In addition to that, evaporator liquid feeding is limited by pulsing solenoid. This would be similar to pulse-width-operated EEV, except that the valve has no metering orifice. Actually it is claimed that the system should work with a plain on-off solenoid valve. Since the valve has no metering orifice, the system is claimed to be not sensitive to gaseous phase refrigerant before the valve. The system also has no receiver on high or low pressure sides.

Let's analyze of the proposed system. The system without receiver is possible – most of the works discussed in [1] analyze systems without receiver. The charge minimization is achieved by charging the refrigerating system the minimal refrigerant quantity necessary to system operation. Often the refrigerating systems include significantly refrigerant receivers that permit to store a refrigerant quantity definitely higher than strictly necessary working quantity. Such system may face minor leaks without adding refrigerant. However, from the ecological point of view such practice is unacceptable since it will increase total leakage significantly. In the system designed for minimal refrigerant charge the destination of receiver should be only to compensate fluctuations of active charge, which may occur at different operating conditions. If system is working at close to constant operating conditions the receiver is not needed, which is common case for heat pump systems. The eutectic system analyzed here must comprise of properly seized receiver since its operating conditions vary in a wide range.

The clause considering small diameter liquid line is also clear – the reduction of liquid pipe diameter is one of the success keys in a charge minimization strategy. The diameter must be optimized taking into account the associated pressure losses - the decreased diameter increases a risk of partial liquid vaporization prior to expansion device and a bad evaporator's supply. Possible strategies to design a refrigerating plant with reduced diameter of liquid pipes depend on system architecture. For refrigerating systems without liquid recirculation the smaller liquid line diameter may be compensated by (i) better liquid subcooling in suction –SLHX or economizer or by (ii) an additional pump at the condenser outlet.

We performed simulation of refrigerating system with SLHX using the mathematical model presented in [16, 17] with refrigerant properties from [18] and compressors parameters from [19]. According to the simulation, smaller liquid line diameter also has additional advantage in the systems comprising of counter-flow welded tube SLHX. While the diameter of liquid line decreases, the increase of heat transfer coefficient from the liquid side is faster than the decrease of heat transfer area, and kA value of SLHX increases. The pressure losses in the liquid line will also increase rapidly, but that is not a problem as far as required refrigerant capacity is ensured. Therefore the combination of smaller diameter liquid line and EEV without metering orifice makes sense for charge reduction as well as higher thermal efficiency of SLHX.

One may doubt the suggestion in [20] to use refrigerating system without SLHX. This could be considered when the system with absolutely minimal charge is on target, since generally without the SLHX the vapour quality at the inlet of evaporator will be higher, bringing in some charge decrease. However, this is the case when measure reducing refrigerant emissions also decreases system energy efficiency. In-deep investigation of SLHX influence on system performance is given for example in Klein et al. [21]. We also performed experiments in order to estimate the influence of SLHX to low temperature refrigerating system. Two refrigerating systems were tested in parallel - the reference system (A) and otherwise identical system with SLHX from the system (B). The system with higher efficiency SLHX at 'OFF' instance reached both lower average eutectic plate temperature and lower average air temperature. From ON to OFF the pull-down energy consumption decreased by 1.74 kWh (15% decrease). The measured performance improvement is higher than obtained through simulation of SLHX. In order to explain the observed improvement, the deeper analysis including influence of SLHX on heat transfer and hydraulic losses in evaporator is required. Still, the experiment demonstrates the importance of the SLHX for efficiency of eutectic system.

When analyzing the baseline eutectic system in its nominal operation mode we noticed that the system may operate with significantly lower refrigerant charge without negative impact on performance. However, such system with 'filled on demand' charge demonstrated inadequate performance during pull-down test at higher ambient temperatures. We come to conclusion that this performance degradation is related to one more design feature, which as far as we know is entirely overlooked in literature, but still strongly influences the total refrigerant charge in eutectic systems. This feature is a mean to ensure the MOP.

4. MOP and cycle parameters

The eutectic system is low evaporation temperature system with the maximum operation pressure limitations imposed by compressors maximal electrical current as well as maximal pressure in condenser. The eutectic systems mostly use the mechanical outlet OPR for maintaining a predetermined maximum suction pressure, which is chosen experimentally during the pull-down test at maximal rated ambient temperature.

Let's analyse performance of such system during the initial stage of pull-down at high ambient temperatures. The ambient temperature t_{amb} = 38°C and the same is initial eutectic plate temperature. The TXV is limiting the refrigerant supply to evaporator according to superheat at outlet, i.e. expansion device capacity (EDC) or MOP are not a limiting factors. For the analyzed cycle conditions the difference between condensing and evaporation pressures is about 11 bars, which is enough for operation of TXV. In such case the heat transfer surface dedicated to evaporation is almost constant and the difference between the plate and evaporation temperatures correlates well with cold capacity of the compressor - as cold capacity decreases, the temperature difference also decreases. For the baseline system (A) the OPR pressure is set to 1.7 bar and the condensing temperature should not exceed 55°C. The effectiveness of SLHX in system (A) is approximately equal to $\eta_t = 0.5$. The compressor's cold capacity according to manufacturer's data is approximately 1.5 kW (a bit lower than nominal) and the temperature difference in evaporator can also be assumed a bit lower than in nominal mode of operation (7 K); the evaporation temperature then is equal to 31°C. Let's assume the vapour superheat at evaporator outlet equal to 6 K and the temperature before OPR will be about 46°C. With such assumptions the simplified diagram of the cycle is given on the Fig. 2.

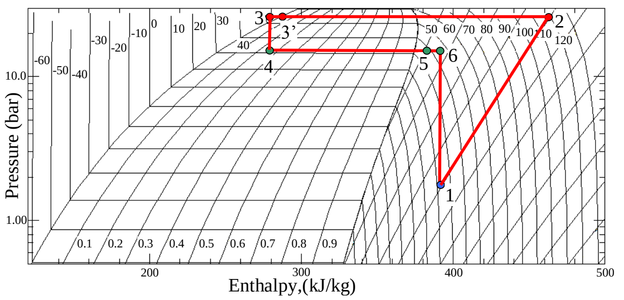

Fig. 2 The p-h diagram for OPR controlled system, initial stage of pull-down at $t_{amb} = 38$°C; *1 -2* – compression, *2 -3'* – desuperheating and condensing, *3'-3* – subcooling in SLHX, *3-4* – expansion in TXV, *4-5* – evaporation and superheating in evaporator, *5 -6* – superheating in SLHX, *6 -1* – expansion in OPR

If the refrigerating system is controlled by EEV, the OPR valve is not used, since the controller itself may limit the refrigerant supply to evaporator according to suction pressure – MOP function is overriding the superheat control. Simplified diagram for such system is given on Fig. 3. Let's analyse such system at the same conditions as in previous example. Since OPR valve is not present in the system, the EEV is limiting refrigerant supply to the evaporator so that the pressure in the evaporator does not exceed the MOP settings, and evaporation temperature is approximately equal to -35°C. Difference between the plate and evaporation temperatures is then greater than 70 K – about 10 times bigger, than assumed in the previously analyzed case. Since the compressors cold capacity in both cases is approximately the same, the heat transfer area dedicated for evaporation proportionally decreases, increasing the heat transfer area dedicated for refrigerant superheating. At the initial stage the temperature of superheated vapour at evaporator outlet is close to plate temperature. During further operation the plate temperature decreases, the evaporator is gradually filled with liquid and the superheat in evaporator decreases to ~6°C. The corresponding p-h diagram is the Fig. 4 (cycle *a*).

During the baseline system operation its p-h diagram also undergoes transformations – the evaporation temperature decreases, vapour superheat in SLHX increases and pressure drop in OPR decreases. Eventually the initial cycle from Fig. 2 also transforms into the cycle *a* from Fig. 4. From this moment neither OPR nor MOP function of electronic controller does influence the system performance.

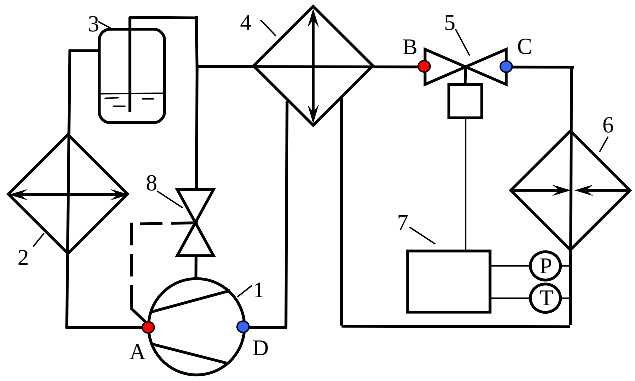

Fig. 3 Diagram of eutectic refrigerating system with EEV; *1* – compressor, *2* – condenser, *3* – HP receiver, *4* – SLHX, *5* –EEV, *6* – evaporator (eutectic plates), *7* – electronic controller, *8* – DTC valve

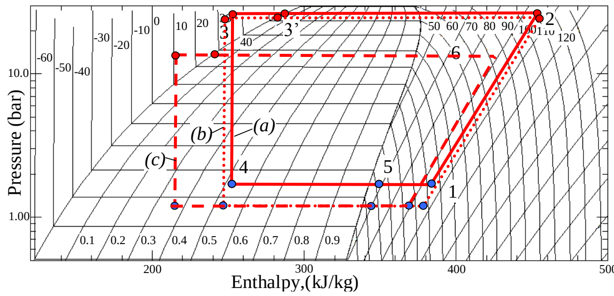

Fig. 4 The p-h diagrams for EEV controlled system; *(a)* – all evaporator working at $t_{amb} = 38$°C, $t_o =$ = -35.5°C, *(b)* – crystallization at $t_{amb} = 38$°C, $t_o =$ = -43°C, *(c)* – at nominal conditions $t_{amb} = 20$°C, $t_o = -43$°C; *1 -2* – compression, *2-3'* – desuperheating and condensing, *3'-3* – subcooling in SLHX, *3 -4* – expansion in EEV, *4 -5* – evaporation and superheating in evaporator, *5 -6* – superheating in SLHX

In real refrigerating system the discharge temperature is controlled by DTC valve, but on the given diagrams (Figs. 2 and 4) the compression is assumed isentropic. This assumption makes no difference for analysis of the evaporator. However, for analysis of the condenser the vapour temperature after compression directly influences the refrigerants mass flow in condenser (when liquid injection is used, the refrigerant mass flow in condenser is bigger than refrigerant mass flow in evaporator).

Another simplification is constant condensing temperature. For the conditions on the Figs. 3 and 5 (cycle *a*) the condenser's heat rejection is not strictly the same, and the condensing temperatures should be different. However, the heat rejection differs by less than 6%, and the average temperature difference would change about 0.5°C. For the simplicity reason this change was neglected.

5. Simulation results

In order to estimate the refrigerant charge distribution in two phase regions, the Premoli et al. [22] correlation was chosen. According to Rice [23], Premoli correlation is one of the most effective for refrigerating systems. According to Kuijpers et al. [24], only the Premoli correlation gives satisfactory results for the charge calculation in a condenser. According to Farzad and O'Neal [25], models depending on the mass flux, such as Premoli correlation, predict a refrigerant charge that is more significant and is more effective.

The refrigerant properties required for the simulation were taken from REFPROP database [18]. The influence of oil in the condenser and evaporator for fluid properties was not taken into account. The refrigerant mass flow rate in the evaporator was estimated according to the data, provided by compressors manufacturer [19].

The evaporators of both baseline systems comprised of two parallel branches, made of Ø15×1 tube. Each branch was controlled by separate TXV according to superheat in the branch. However, such evaporator setup is not suitable for system with EEV. The controllers currently available on the market can only control one branch per controller. In the best case the setup for two branches system can comprise of cold room controller + additional slave controller, both sharing the signal from the single pressure probe. While such setup can be used, the cost increase is too high for competitive transport refrigerator.

For the systems with EEV therefore are two options – to use parallel design with distributor or serial design. The parallel design makes it possible to decrease the evaporator's tubes diameter which would result in lower evaporator's volume, lower cost, weight, refrigerant charge and even potentially better efficiency. We made the preliminary tests of such system, and obtained controversial results. The tests confirmed the higher efficiency of the parallel system, but also revealed some problems caused by uneven refrigerant distribution (mainly caused by technological restrictions). Therefore the serial evaporator design was chosen for eutectic system with EEV. In order to keep hydraulic losses within acceptable limits, the diameter of evaporator's tube was increased to Ø18×1.

Calculating the refrigerant charge we considered the Ø15×1 two branch evaporator design for baseline system and Ø18×1 serial design for system with EEV.

For the baseline system (A) at the conditions corresponding to Fig. 2 the calculated refrigerant charge in evaporator is approximately 1.4 kg, while for conditions corresponding to Fig 4 (cycle *a*) the charge decreases to only 0.44 kg. During this cycle transformation at initial stage of pull-down the refrigerant charge in the evaporator decreases by almost 1 kg, which makes approximately 25% of total charge in baseline system (A). For the system (C) with EEV at the conditions corresponding to Fig 4 (cycle *a*) the refrigerant charge in the evaporator is equal to 0.54 kg, which is more than for baseline system (A); the charge increases due to bigger volume of the evaporator. However, at the initial stage of pull-down for the system with EEV the charge in the evaporator is smaller due to higher vapour quality and smaller tube length dedicated to evaporation. During further stage of pull-down – crystallization of eutectic plates (Fig. 4, cycle *b*) – the heat rejection decreases, condensing temperature slightly decreases, evaporation temperature decreases to -43°C. As a result the vapour quality after expansion changes insignificantly and the charge in evaporator remains almost the same.

The required refrigerant charge at high pressure side remains almost constant for all analyzed stages of the baseline system (A) operation at 38°C ambient temperature. According to the simulation, about 0.84 kg of refrigerant can be found in condenser, 1.2 kg in receiver and about 0.27 kg in liquid line (tube Ø10×1, L = 5.5 m). Calculating the charge in the receiver we made the same assumption as Rajapaksha and Suen [26] that 1/6 of the receiver is filled with liquid refrigerant. Then in the system (A) at initial stage of pull-down the charge is about 3.71 kg, at -35.5°C evaporation temperature it decreases to 2.75 kg and during crystallization remains about 2.73 kg. Some additional charge remains dissolved in the compressor oil and in filter – dryer. The calculation of charge dissolved in oil is quite complicated, since oil is circulating in the system and may be at different temperatures and pressures. Thorough evaluation of all the piping was also not performed. Still, the maximal calculated value of refrigerant charge will be close to the charge, which is actually filled into the baseline system (A).

In Corberán et al. [4], Primal et al. [27] the charge distribution between the condenser and evaporator is different, but these sources analyze the systems with liquid sub-cooling in the condenser. As noted Hrnjak and Litch [15], the liquid sub-cooling is a large contributor to total charge since the sub-cooling region holds most of the total charge inventory. In systems with the high pressure receivers the sub-cooling in the condenser is not possible, and refrigerant charge in the evaporator is relatively bigger. For the eutectic system the charge in the evaporator is also increased due to relatively big volume of the evaporator.

Neglecting the charge increase in the evaporator during the initial stage of pull-down, the active charge could be considered almost constant. Even for different ambient temperatures the variation of active charge is very small. For example during the crystallization at nominal conditions (Fig. 5, cycle *c*) the active charge in baseline system is 2.53 kg. Some charge increase in the evaporator is compensated by the charge decrease on the high pressure side. The charge increase during the initial stage of pull-down is caused by interaction of TXV and OPR, while in the system with EEV during this stage of operation the charge in the evaporator even decreases.

Unfortunately, the system with EEV may not always be competitive due to higher cost. For the low-cost low-charge systems the alternative control system was proposed. Such system could comprise of the mechanical TXV with MOP function. The lowest MOP pressure for the available expansion valves corresponds to -25°C evaporation (2.6 bar for R507A), which could be considered acceptable. The experimental system with such TXV was built out of baseline system (B) – for this system the OPR settings correspond to -28°C, which is very close to nominal MOP of TXV. Unfortunately, during the tests the MOP function in mechanical valves was not accurate enough, and suction pressure regulator OPR was still required. When OPR settings were set to nominal, the evaporation pressure increased, and resulted evaporation pressure at initial stage of pull-down was about 6 bar. The systems (B) and (D) are also equipped with higher efficiency SLHX with $\eta_t = 0.5$.

Operation of such system has similarities with both previously analyzed systems. Initially the refrigerant supply to the evaporator is limited by MOP function of TXV and only fraction of evaporator surface is used for heat transfer. While the plate temperature decreases, the evaporator is filled with liquid until all the evaporator is working and TXV is controlled according to superheat in evaporator. This point corresponds to the biggest refrigerant charge in the evaporator due to low vapour quality and high pressure. Note that evaporation pressure at this point was found to be much higher, than rated MOP pressure of TXV. Resulting p-h diagram for such system at previously described conditions is given on Fig. 5 (cycle *a*). Estimated refrigerant charge is equal to 1.1 kg, which is significantly less than maximal charge estimated for baseline system (A). Comparing the charges one should also have in mind that evaporator's volume in baseline system (B) is ~33% bigger. From this point the evaporation pressure decreases until it equalizes with OPR pressure (Fig. 5, cycle *b*). The charge in evaporator at this point decreases to 0.68 kg.

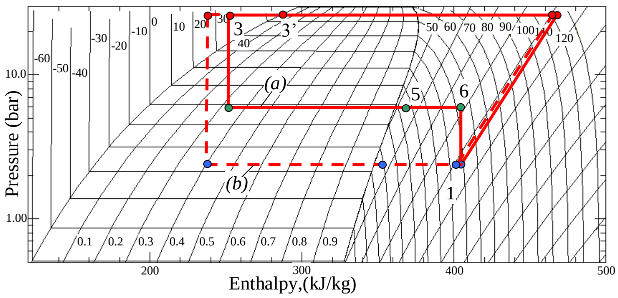

Fig. 5 The p-h diagram for pull-down of TXV (with MOP) + OPR system, all evaporator working, t_{amb} = 38°C; *(a)* – t_o = -1.2°C, *(b)* – t_o = -28°C;*1-2* –compression, *2-3'* – desuperheating and condensing, *3'-3* – subcooling in SLHX, *3-4* – expansion in TXV, *4-5* – evaporation and superheating in evaporator, *5-6* – superheating in SLHX, *6-1* – expansion in OPR

The successful charge minimization strategy also include careful selection of the receiver. Previously, when the charge minimization was not considered a priority, the fundamental principle in selecting the size of liquid vessels was to choose them large enough that during operation they never become completely full of liquid or completely empty. However, the complete filling of the receiver with liquid should not cause a problem if no valve is present between the receiver and the condenser. During the normal operation the receiver should only compensate the variation of active charge in evaporators. The service technicians commonly wish having a possibility to move all the charge from the low pressure side to the high pressure side (e.g. for replacing compressor). In such case the liquid refrigerant can be stored not only in receiver, but also in the condenser, i.e. the common volume of the receiver and condenser should be able to hold the entire refrigerant from the system (with some reserve volume).

Also, it is considered that liquid should not be permitted to completely drain from the HP receiver, because this would adversely affect the performance of control valves. If the SLHX is implemented, significant amount of vapour may be condensed in its liquid line without affecting the control valves. From our experience, the refrigerating system performance is not affected significantly as far as clear liquid with some bubbles of the vapour can be seen in a sight glass (after receiver). Still we support the recommendation to have at least 1/6 of the receiver volume filled with liquid in order to avoid incomplete condensation. The baseline system (A) comprised of 4.3 litres receiver. Following the discussed principles the volume of the receiver for corresponding low charge system (C) was decreased to 1 litre. Then at 55°C condensing temperature the refrigerant charge in the receiver (with 1/6 volume full of liquid) decreased from 1.23 kg to 0.29 kg.

Another parameter – the diameter of liquid line – was already discussed previously. The tube for liquid line was changed from Ø10×1 to Ø6×1, which decreased refrigerant charge in liquid line from 0.265 kg to 0.065 kg. The liquid line diameter may be decreased even further (system with Ø5×1 liquid line was successfully tested), but further charge reduction is insignificant.

For the low charge system (D) with mechanical control the 1.5 litre receiver was chosen. Also the condenser from the baseline system (B) is not suitable for low charge system – at high ambient temperatures such condenser alone holds almost 1.5 kg of refrigerant, which makes the system with 2.9 kg charge beyond the reach. This condenser was replaced with special low-volume (2.25 instead of 3.98 liters) condenser with the same capacity. According to the simulation, such system should be able to work with 2.9 kg of refrigerant. The decreased tube diameter causes some increase of pressure drop, but estimated effect on condensing temperature is insignificant.

6. Test results of low charge systems

The low charge experimental systems (C) (with 2.4 kg charge) and (D) (with 2.9 kg charge) were developed and built using previously discussed principles and recommendations. Their test results were compared to baseline systems (A) and (B). The tests were successful – the low-charge systems demonstrated similar performance as baseline systems with regular charge in a whole range of ambient temperature. No evidences of insufficient charge were observed.

Also it was found that 2.4 kg system with EEV (system (C)) has some reserves for charge decreasing – no performance degradation was observed when the charge was further decreased by 400 g. The thorough experiments were not done since the objectives of the project were already reached. The system with mechanical TXV and OPR (system D) does not have reserves for further charge decrease with current components.

Another question is considering significance of MOP control for eutectic systems operating at high ambient temperatures. One could think, that previously discussed 'initial period' lasts for a few minutes, and therefore could be neglected when selecting the charge. The Fig. 6 gives the temperatures, measured on the outside surface of the tubes, entering the eutectic plates. As can be seen, it takes more than 5 hours from the beginning of the test to the moment, at which all the evaporator is working and the MOP function is no longer interfering with control. During this period the system filled 'on demand' may operate with incomplete condensation, causing significant degradation of performance.

On the Fig. 6 also two MOP corrections can be seen, when the system was turned off by the high pressure protection, after which the MOP settings were reduced.

All the previously discussed systems were equipped with copper tube/aluminium fin coil condensers. Parameters of (B) condenser would be typical to the condensers, used in refrigerating industry, while the (A) condenser is special low-volume design. The further charge decrease is possible with the implementation of new low-volume condenser. The aluminium microchannel condenser, similar to the one described in [15], was also tested and demonstrated superior performance. The inner volume of the condenser was only 1.345 litres, which is only 60% of current low-volume condenser, used in systems (A), (C) and (D). Comparing to condenser of system (B), the volume of new condenser is only 34%. Implementation of such condenser alone allows decreasing the refrigerant charge by 0.3 kg for systems (A), (C) and (D), while for system (B) the charge decrease is 1 kg. For the system based on system (B) and equipped with lower diameter liquid line and such condenser we may decrease the volume of the high pressure receiver from 6.9 to 4.6 litres, while the refrigerant charge for such system decreases from 6.0 to 4.5 kg even without implementing the new control system. Such the system was also tested and no performance degradation was observed.

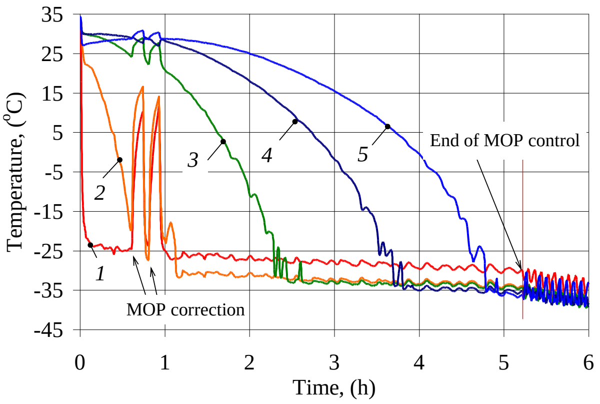

Fig. 6 Temperatures of the evaporator tubes at the entrance of eutectic plates during the pull-down test at t_{amb} = 37°C for the system with EEV; *1-5* numbers of the plates according to serial connection

When microchannel condenser was implemented together with new control system, even better results were achieved. The new system was based on the baseline system (A), but equipped with the new condenser, higher capacity SLHX from system (D) (liquid line Ø6×1 mm), 2.3 litre high pressure receiver and new mechanical control system (TXV with MOP + OPR). The system was tested thoroughly and proved to be perfectly functional with 2.5 kg refrigerant charge. The efficiency of the system even increased due to higher capacity of SLHX – the measured COP was higher by ~9% when compared to baseline system (A). Such system is also very cost-efficient, since the refrigerant charge and energy consumption is decreased without increasing system cost.

7. Conclusions

The objective of this project was to develop two low-charge eutectic refrigerating systems. During the research it was found, that big variation of active refrigerant charge in baseline systems is related to significant increase of refrigerant charge in evaporator at initial stages of pull-down. This charge increase depends on the controls used in the system in order to maintain maximum operation pressure. The biggest charge increase is for traditional combination of TXV and OPR – in this case the charge increase was ~1 kg, i.e. 25% of total charge in baseline system (A). For low-charge systems it was proposed to use EEV or combination of TXV with MOP function and OPR. As far as we know the influence of MOP control to refrigerant charge was not previously discussed in literature.

Next, it is important to reduce the internal volume of condenser and evaporator. With implementation of microchanel condenser the charge in the condenser may be decreased by 40-70%, comparing to traditional copper tube/aluminium fin coil condensers. This makes 7 to 17% of the total refrigerant charge in the baseline systems. With the parallel plate evaporator setup, the further charge reduction is possible.

The minimization of liquid line diameter may also offer some charge reduction without any negative effect on performance. The potential of this measure depends on the length of the liquid line. In analysed case the charge decrease was ~5% of total charge in baseline system.

The last important measure for charge reduction is to decrease the volume of receiver. If low-charge system is on target, the function of receiver should be limited to compensation of active charge variation. In the analysed case the refrigerant charge in the receiver was decreased by ~1 kg - from 1.23 kg in system (A) to 0.29 kg in system (C). This charge decrease makes ~25% of total charge in baseline system (A).

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MAŽOS DOZĖS TRANSPORTINIS ŠALDYTUVAS (II). TEORINIS IR EKSPERIMENTINIS TYRIMAS

R e z i u m ė

Aplinkosaugos iššūkių kontekste vienas iš svarbiausių šaldymo, vėdinimo ir oro kondicionavimo technikos uždavinių yra sumažinti šaldymo agento dozę. Šiame darbe pateikiama transportinio šaldytuvo su eutektinėmis plokštėmis dozės minimizavimo studija.

Tyrimai parodė, kad įprastinėje sistemoje pradiniame atšaldymo režimo etape dozės garintuve gerokai padidėja dėl termostatinio droseliavimo ventilio ir siurbimo slėgio reguliavimo ventilio tarpusavio sąveikos. Bazinei sistemai (A) šis dozės padidėjimas buvo ~1 kg, t.y. 25% visos dozės. Šį dozės padidėjimą galima sumažinti naudojant skirtingas valdymo sistemas. Kiti svarbūs veiksniai yra minimalus resiverio, šilumokaičių bei skysčio linijos tūris.

Buvo sukurti ir išbandyti du mažos dozės šaldytuvai su skirtingomis valdymo sistemomis. Palyginti su bazinėmis sistemomis, šaldymo agento dozė sumažėjo 40 – 50%.

L. Vaitkus

LOW CHARGE TRANSPORT REFRIGERATOR (II). THEORETICAL AND EXPERIMENTAL INVESTIGATION

S u m m a r y

Considering the environmental challenges the refrigerant charge minimization is one of the most important targets for refrigerating and air conditioning applications. This paper presents a charge minimization study of transport refrigerator with eutectic plates.

It was found that in traditional system significant charge increase in evaporator during the initial stage of pull-down was caused by interaction of thermostatic expansion valve and outlet pressure downstream regulator. For the baseline system (A) the charge increase was ~1 kg, i.e. 25% of total charge. This charge increase may be minimized through implementation of different control systems. Other important factors are the volumes of receiver, heat exchangers and liquid line.

Two low-charge refrigerators with different control systems were developed and tested. The charge decrease was 40-50% comparing to baseline systems.

Keywords: transport refrigerator.

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