Refrigeration system with integrated liquid line and low environmental impact

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REFRIGERATION SYSTEM WITH INTEGRATED LIQUID LINE AND LOW ENVIRONMENTAL IMPACT

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ABSTRACT

A design of a low charge refrigerating system based on microchannels technology for the condenser, a compact evaporator and a reduced diameter liquid line is presented. The design of the liquid line of a refrigerating system is specially detailed. Usually designed to avoid pre-expansion before entering in the expansion device, it is shown in this study that a drastic reduction of the pipe diameter is possible by cooling progressively the liquid. An adapted design using this principle, called "integrated liquid line" (ILL) has been developed and its impact on refrigerant charge and energy performance has been evaluated experimentally for a cold room refrigerating unit. A numerical model of the ILL, validated on this experimental device for various running conditions, has been developed and can be used as a design tool for other systems. Experimental measurements were performed and compared to a reference system. Coupled with micro-channel exchangers, a significant reduction of the internal volume and consequently of the refrigerant charge can been reached.

Key words: integrated liquid line, volume reduction, heat transfer coefficient, refrigerant charge, TEWI, coefficient of performance.

RÉSUMÉ

Une installation à faible charge en frigorigène a été conçue à partir de condenseurs à mini-canaux, d'un évaporateur compact et d'une ligne liquide à diamètre réduit. Les règles de conception de la ligne liquide sont plus spécialement présentées. En effet, bien que ce composant contienne une quantité non négligeable de fluide, cette quantité n'est pas prise en considération lors de la conception. Nous présentons ici le concept de « ligne liquide intégrée » (LLI) consistant à réduire le diamètre de la ligne liquide et à sous-refroidir le liquide afin d'éviter une pré-détente. L'impact de cette LLI sur la charge et la performance a été caractérisé expérimentalement. Le modèle numérique permettant la détermination des diamètres de la LLI en fonction du fluide utilisé et de la puissance est également présenté. La réalisation expérimentale a montré que cette technologie, couplée à celle des échangeurs compacts, permet une réduction significative de la quantité de frigorigène dans une installation.

1. INTRODUCTION

Whatever the fluid used in a refrigerating plant, a reduction of the refrigerant amount has positive aspects. In the case of HFC, it means less environmental impact; in the case of natural refrigerants as hydrocarbons, ammoniac, it means more safety; in all cases, it means a cost reduction. Thus, a general tendency for the design of refrigerating systems is a relatively "new" concern about the refrigerant amount required. Until now, most design principles were based on a robust functioning of the systems without taking into account the refrigerant charge. A good example is the calculation rules for liquid lines: the criterion used is mainly the pressure drop along the tubes, nearly ignoring the impact of the resulting refrigerant charge.

The refrigerant charge depends on the internal volume of each component: pipes, heat exchangers, receivers, compressors and the various accessories. The components where the refrigerant is liquid or in two-phase state contain much more refrigerant than the components with vapour only. Figure 1 shows an evaluation of the refrigerant charge in each component for a 5 kW (-28/+40°C) refrigeration system with a roof top condenser unit. It can be seen that the refrigerant is mainly located in the components where it is in liquid or two-phase state: liquid receiver, condenser, evaporator and liquid pipes. Thus, the design of a low charge system has to focus on those critical components.

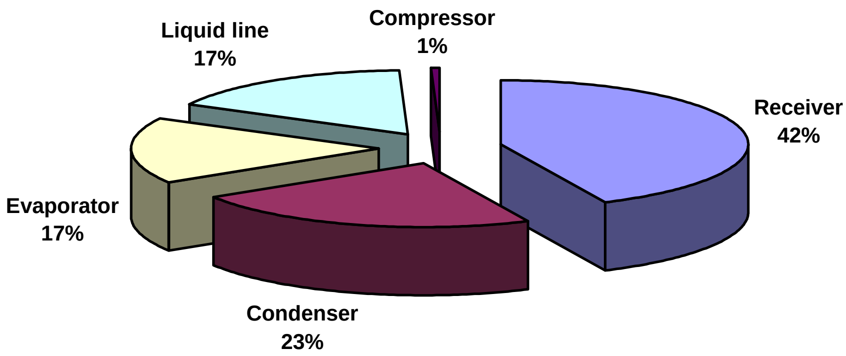

Figure 1 : Refrigerant charge distribution of an air / air refrigeration plant with separated condensation unit (5 kW refrigerating capacity and 15 kg refrigerant at -28/+40°C).

Using compact heat exchangers is an efficient way to reduce the charge of the systems [1] and many research works focused in the recent years on these components. Microchannel heat exchangers for example allow a drastic reduction of the internal volume of condensers with a high air performance. Although the distribution of the two phase flow makes the design of microchannel evaporators difficult, evaporators with a small fin spacing and reduced diameters remains a technological solution available.

The design of the liquid line is also important and the diameter of this pipe may have a large impact on the total charge. A consequence of a diameter reduction of this pipe is a higher pressure drop and a possible pre-expansion. Since most of the expansion devices used in a refrigerating system are not designed to be used with a two-phase flow, its consequence would be a reduced refrigerating capacity and a low performance of the system. An alternative is to use a reduced diameter with a progressive cooling of the liquid refrigerant to avoid the pre-expansion. This cooling may be provided by the system itself, and particularly by the vapour leaving the evaporator.

In this paper, the design of a low charge refrigerating system is presented, with an emphasis on the liquid line. An experimental comparison with a classical system using usual diameters for heat exchangers and pipes is given.

2. DESIGN OF THE LOW CHARGE REFRIGERATING SYSTEM

2.1 Heat exchangers

The design of a low charge system means a reduction of the internal volume of each component, including pipes and requires a good knowledge of the consequences of the sizing reduction on pressure drops in small diameters pipes, particularly for a two-phase flow. Usual diameters used for heat exchangers and pipes are from 10 to 14 mm. The small-channel technology uses 0.2 to 3 mm hydraulic diameters. The result is a drastic reduction of internal volumes while keeping a similar thermal efficiency [1].

This technology, already widely used in car air conditioning, was selected in this study for the condenser. The condenser unit is composed of two parallel small-channel condensers. Internal volume of each condenser is 0,38 liter to be compared to the 6 liters of the 8 mm diameter condenser of the reference system.

The small-channel technology for the evaporator at negative temperatures was first considered. However the compactness of such exchangers, which is an advantage considering the refrigerant charge, is a severe drawback considering the behaviour during frost formation, due to the fin spacing. Moreover many authors report a non homogeneous refrigerant distribution in microchannel evaporators [4, 5, 6] and have even reported a 20% performance degradation due to maldistribution [7]. For both reasons, a round tube evaporator with a diameter as small as possible, has been selected.

2.2 Liquid line

Another important point to investigate was the refrigerant amount in the liquid line. In order to reduce its diameter, this pipe had been incorporated in a liquid-vapour heat exchanger. With this configuration, pressure drop caused by the low diameter is balanced by an extra-cooling of the liquid in order to avoid a pre-expansion (ie evaporation before entering the expansion device).

A consequence of a small diameter liquid line is a pressure drop as presented between the points 4 and 4' in the Figure 2. To avoid a phase change, the refrigerant may be cooled by the vapour leaving the evaporator. This heat exchange results in a vapour overheat at the inlet of the compressor, as shown in figure 3 (points 1-1'). The design has also to take into account another constraint: an excessive discharge temperature may result in an oil deterioration.

In order to determine the best diameters, a model of the heat transfer and pressure loss has been developed and is presented later.

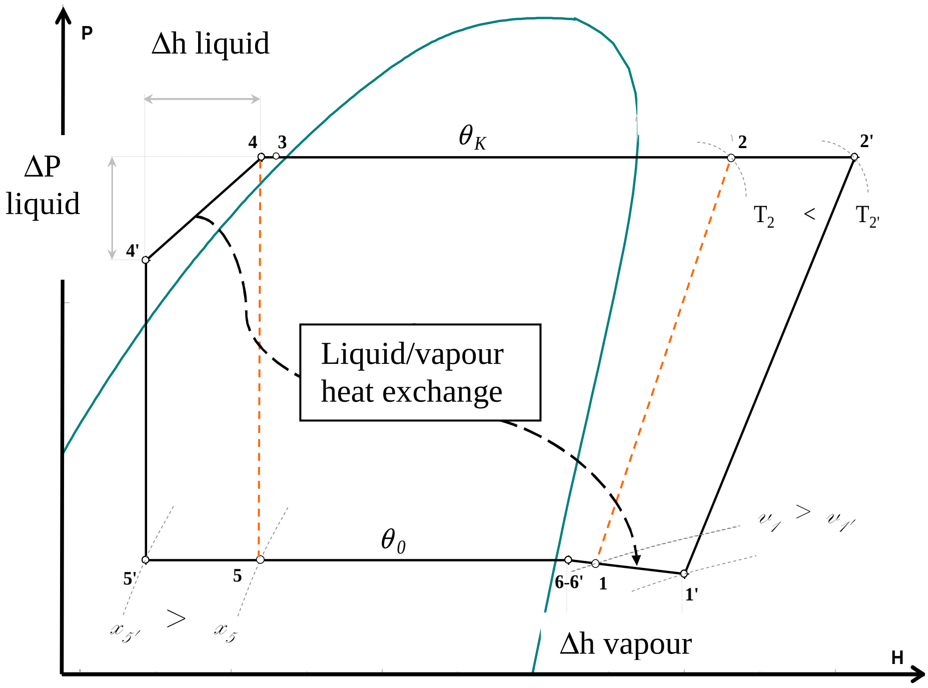

Figure 2 : P-h Diagram for a ILL system

The regular pressure drop along the liquid line can be evaluated by :

$$dP = 4f\frac{L}{D_h}\frac{\rho u^2}{2}$$

where the parameter f can be computed by:
for a laminar flow ($\mathrm{Re} \leq 2300$) by the Poiseuille equation:

$$4f = \frac{64}{\mathrm{Re}}$$

for a turbulent flow (if $2300 < \mathrm{Re} < 10^5$) by the Blasius equation

$$4f = 0.316\ \mathrm{Re}^{-0.25}$$

and, if $\mathrm{Re} > 10^5$, by the Herman equation

$$4f = 0.0054 + 0.3964\,\mathrm{Re}^{-0.3}$$

The heat transfer between the liquid and the vapour refrigerant can be computed by:

$$dQ = KdS(T_l - T_v)$$

$$dQ = \dot{m}Cp_v dT_v = \dot{m}Cp_l dT_l$$

where the heat transfer coefficient K is given by :

$$K = \frac{1}{\alpha_l} + \frac{1}{\alpha_v} + \frac{e}{\lambda}$$

The local heat transfer coefficients h_l and h_v are evaluated by the Mac Adams correlation [2] for conventional diameters

$$Nu = 0.023\,\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{1/3} \text{ with } 0.5 < \mathrm{Pr} < 100$$

and for reduced diameters, by the Gnielinski correlation [3] :

$$\alpha_{mono} = Nu_{Gniel} \times \left(1 + 7{,}6.10^{-5} \times \mathrm{Re} \times \left[1 - \left(\frac{D_h}{1{,}164}\right)^2\right]\right) \times \frac{\lambda}{D_h}$$

$$with\ \ Nu_{Gniel} = \frac{(f/8) \times (\mathrm{Re} - 1000) \times \mathrm{Pr}}{1 + 12{,}7\sqrt{f/8} \times (\mathrm{Pr}^{2/3} - 1)} \times \left[1 + \left(\frac{D_h}{L}\right)^{2/3}\right]$$

This model has been used to build the graph of the Figure 3, for a -20°/+40°C cycle and R404A refrigerant. This figure shows the possible diameters for suction and liquid pipes depending on the refrigerating capacity. The graph presented here is valid only for a -20/+40 cycle with R404A refrigerant for a 0 to 30 m liquid line.

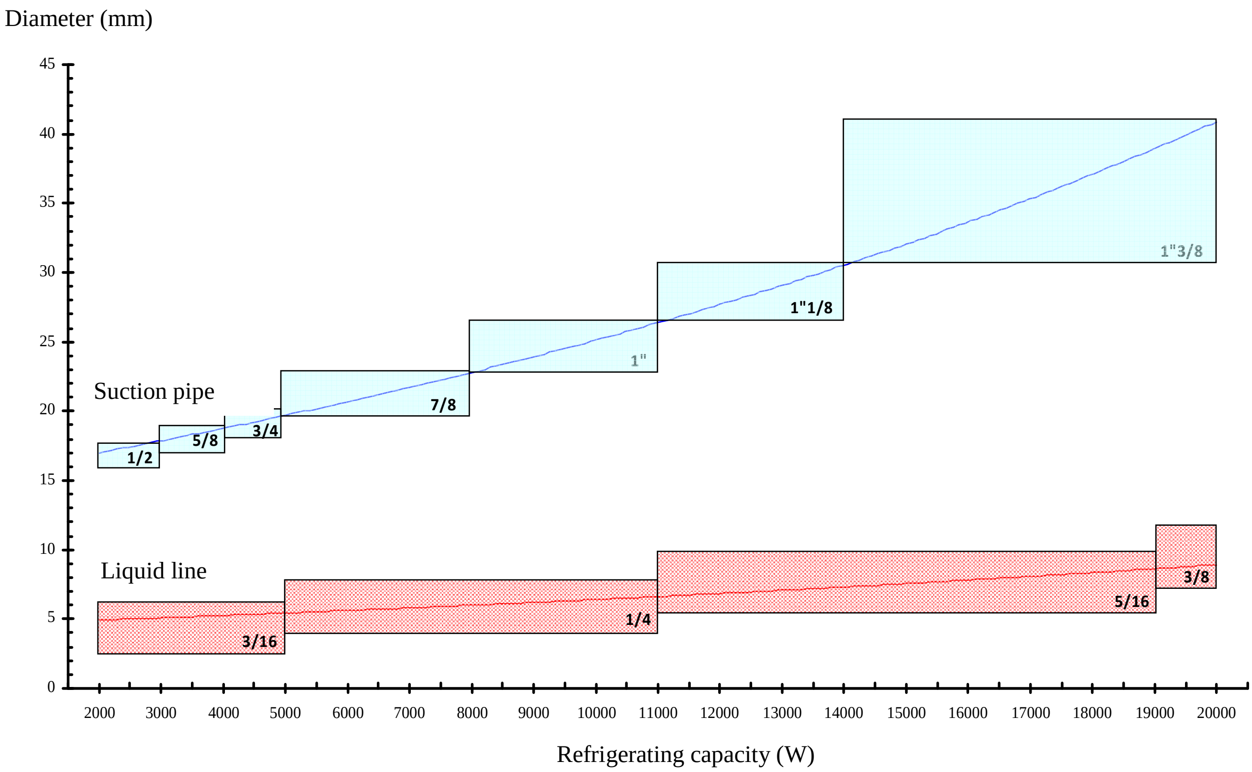

Figure 3 : Graph for the selection of inner and outer diameters of the liquid line (-20/+40 °C R404A)

1. EXPERIMENTAL RESULTS

3.1. Internal volumes

The condenser unit is composed of two parallel small-channel condensers, with a 86% reduction of the internal volume compared to the reference system.

The internal diameter of the 26 meters liquid line of the prototype was reduced from 9.5 mm to 6.4 mm (1/4"). A consequence of this reduction is a higher pressure drop, which varies from 1 to 3 bar, depending on the operating conditions. In order to avoid a pre-expansion of the refrigerant in the line, a long coaxial liquid / vapour heat exchanger (Figure 4) has been developped to maintain the refrigerant in liquid state despite the pressure drop. Such a coaxial exchanger enables a progressive subcooling of the refrigerant keeping the refrigerant in a liquid state. The global result is a 55% reduction of the internal volume of the liquid line.

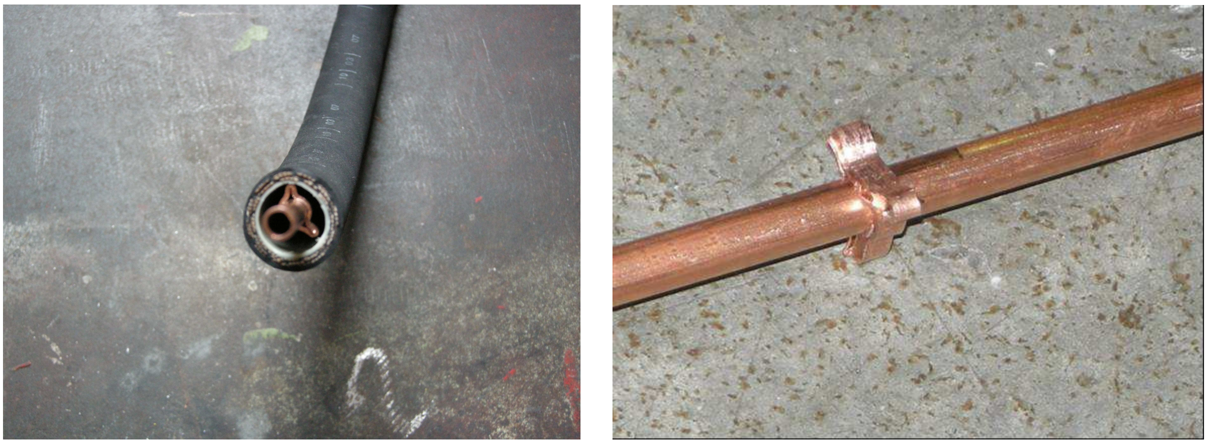

Figure 4 : Coaxial liquid/vapour pipe (ILL)

Details of the volume distribution can be found in Table 1 for the reference system (without ILL) and the system with the reduced diameter liquid line.

Table 1 : Details of volume distribution (reference and low charge systems)

	Reference system	*ILL system*	*ILL/Ref.*
	Evaporator		
Manufacturer	Frigerst PC245-980EB	Friga-Bohn LUC840E	
Internal diameter	10,7 mm (1/2’’)	9,53 mm (3/8)	
Vint	**14.8 l**	**6.0**	*-59%*
Length (total)	117 m (4x29,2)	84 m	
S air side	47,4 m²	32,3 m²	
S refrigerant side	3,9 m²	1,26 m²	
Fin pitch	6,35 mm	4,23mm	
Δp evaporator	1,2 bar (for 45 g/s)	0,11 bar (à 38 g/s)	
Δp suction pipe	0,2 bar (for 45 g/s)		
Heat exchange coeff.	**9.9 W/m².K**	**22 W/m².K**	
Air flow rate	11 800 m3/h	1135 tr/min, 2 fans (≅ 5 370 m^3/h)	
	Condenser		
Manufacturer	Searle MDA 22-6	microchannel Valeo	
Internal diameter	7,93 mm (3/8’’)	1,49 mm	
Internal volume	**6 l**	**0,77 l**	*-86%*
Total length	122 m (4x30,3)	250 m (2x32x7x0,56)	
S air	33 m²	32 m²	
S refrigerant	3,0 m²	1,5 m²	
Fin pitch	2,12 mm	1,28 mm	

Δp condenser	0,5 bar (à 45 g/s)	0,36 bar (à 45 g/s)	
Heat exchange coeff.	**20 W/m².K**	**32,8 W/m².K**	
Air flow rate	5 200 m^3/h	4 600 m^3/h + VEV	
Liquid line			
Internal diameter	***1,0 m 3/8'' + 25 m 3/8''***	***1,0 m 3/8'' + 25,0 m 1/4''***	
Total length	26 m	26 m	
Internal volume	*1,34 l*	*0,60 l*	*-55%*
LVE	*/*	***Coaxial 3/4 - 1/4'' (v_{liq} < 2,5 m/s)***	
Compressor			
	1 cp piston - Bock HAX34P/215-4	1 cp piston - Bock HAX22P/160-4	
V balayé	18,8 m^3/h	13,7 m^3/h (50 Hz)	
Regulation	On/Off control	VEV 20/70 Hz - v controlled with $\theta_{cellule}$	
VOLUME TOTAL	*22,1 l*	*6,97 l*	*-68%*

3.2. Comparison of the refrigerant charge and TEWI

An experimental characterisation of the energy performance of the prototype and of its environmental impact has been carried out and compared with the reference system, and particularly the refrigerant charge, the electrical consumption, the refrigerating capacity and the TEWI criteria.

As it could be expected from the volume reduction, the global refrigerant charge has been reduced (Table 2). The final specific charge ratio (refrigerating capacity/ refrigerant charge) is 0.4 kg/kW for the prototype, which has to be compared to the initial 1.3 kg/kW ratio.

Table 2 : Comparison of the estimated refrigerant charge both in the traditional system and in the reduced charge system

	Reference system	**ILL system**	
θ_0 / θ_K (°C) **Φ_0 (W)**	*-17 / +38* *5 200*	*-17 / +38* *5 200*	
Refrigerant charge (kg)	**6.9**	***2.3***	***-67%***
-liquid pipe	1.1	*0.5*	*-55%*
-evaporator	2.7	*1.1*	
-condenser	1,8	*0.3*	
-vessel	1,3	*0.4*	
Charge ratio (kg/kW)	**1.3**	***0.4***	

The influence of a liquid/suction heat exchanger on a refrigerating system has already been analysed by a few authors. Domanski and Didion [9] presented an analysis of the performance of systems using liquid-suction heat exchangers for 29 different refrigerants. They showed that the specific heat capacity among the relevant thermodynamic properties have the most significant impact on the COP change. Klein and al. [8] analysed the impact of liquid-suction heat exchangers on refrigeration system performance for new refrigerants. In this study, they conclude that liquid/vapour heat exchangers are useful for systems using R404A, R507A, R134a, R407C, and R410A, and can be detrimental to system performance in systems using R22, R717 and R32.

During the tests, the average subcooling of the liquid refrigerant was about 27°C, as presented in Figure 5, and the average heat flux between liquid and vapour was 1 300 W. For a same refrigerating capacity, the mass flow was consequently reduced by 25%. This reduction has a direct impact on the electrical consumption of the compressor (Table 3) and the COP was increased 18%. Several factors affected the performance : a better efficiency of the heat exchangers, a frequency inverter on the compressor, and finally the influence of the liquid line-suction line heat exchange in the ILL.

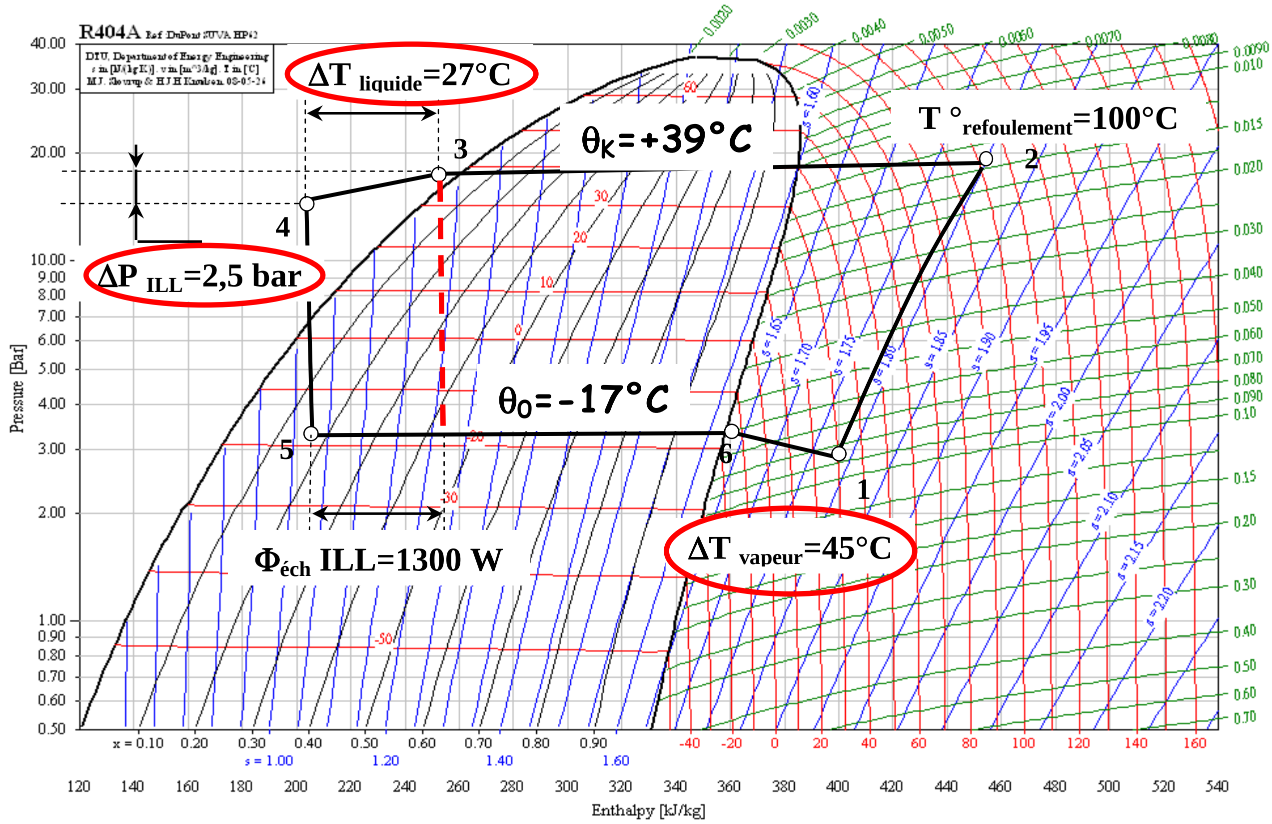

Figure 5 : Mollier diagram for the system with a small diameter liquid line (experiment)

Table 3 shows a 43% TEWI reduction between both systems, mainly due to the direct effect reduction (figure 5) : the indirect TEWI, calculated with a 430 g of CO2 rejection by kWh (average rejection in Europe, AIE 1999), is reduced by 18% and the refrigerant charge reduction yields to a 67% reduction of the direct TEWI.

Table 3 : Performance and TEWI comparison

	Reference system	**ILL system**	
θ_0 / θ_K (°C)	*-17 / +38*	*-17 / +38*	
Φ_0 (W)	*5 200*	*5 200*	
Q_0 (kW.h)	89	*89*	
$W_{C.p}$ (kW.h)	48,2	*39,1*	*-19%*
Q_{f_cond} (kW.h)	2,23	*1,03*	
Q_{f_evap} (kW.h)	8,92	*8,58*	
$Q_{resistance.}$ (kW.h)	5,4	*4,2*	
Q_{elec} total (kW.h)	64,7	*52,9*	*-18%*
Q_{LVE} (kW.h)	/	*23,11*	
TEWI direct (Ton CO_2)	34,1	*11,2*	*-67%*
TEWI indirect (Ton CO_2- Europe)	152,4	*124,6*	*-18%*
TEWI total (Teq.CO_2 - Europe)	186,5	*135,8*	*-27%*

The ratio between direct and indirect TEWI effect depends on the energy policy of the country. For France, one kilowatt-hour causes the rejection of 90 g carbon dioxide. This figure leads to a 43% reduction of the total TEWI, which is considerable.

3. CONCLUSION

The main objective of this project was to demonstrate that it is possible to reduce drastically the refrigerant charge in a refrigerating system using compact exchangers and low charge design rules, particularly for the liquid pipe, while preserving the energy performance (or even improving it).

Using small-channel technology for the condenser, a low diameter evaporator and a coaxial liquid-vapour heat exchanger composed of a reduced diameter liquid line and the vapour suction line, the internal volume of a refrigerating system for a negative temperature cold room could be reduced by 68%. Experimental measurements have been carried out to evaluate the impact of this low charge design on the environment, by measuring energy performance, refrigerant charge and TEWI. It has been shown that the reduction of the global TEWI criteria was 27% (calculated for Europe), and that the refrigerant mass could be reduced by 67%.

The example selected in this paper is highly representative of a large number of refrigerating systems; for example in France, a generalization of this design to all the potential applications could lead to a reduction of more than 10000 tons of refrigerant.

NOMENCLATURE

Cp heat capacity, *J/K*	u speed, *m/s*
D_h hydraulic diameter, *m*	z distance along the distributor, *m*
e thickness, *m*	
f friction factor	*Greek symbols*
K heat transfer coefficient, *W/m²K*	α heat transfer coefficient, *W/m²K*
L length, *m*	γ aspect ratio
$\dot{m}$ mass flow rate, $kg.s^{-1}$	λ conductivity, *W/mK*
Nu Nusselt number	ϕ heat flux, *W*
P pressure, *Pa*	ρ mass density, $kg.m^{-3}$
Pr Prandtl number	*Indices and exponents*
Q heat, *J*	f fans
Re Reynolds number	l liquid
T Temperature, *K*	v vapour

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