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OPTIMIZED DESIGN OF HEAT EXCHANGERS FOR "REVERSIBLE" HEAT PUMP USING R-407C

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ABSTRACT

Temperature glide of R-407C can present either an advantage or a disadvantage depending on the heat exchanger design. There is a need for counter-current heat exchangers whatever the type either air/refrigerant or liquid/refrigerant.

For air/water heat pump, perfect counter-current design is possible for liquid/refrigerant heat exchangers whereas crossed current with counter-current tendency design is appropriate for air/refrigerant heat exchangers. The definition of external and internal flow rates permitting the limitation of the mean logarithmic temperature difference is an additional constraint.

Tests performed on single row and multiple rows refrigerant/air heat exchangers are presented. Comparisons with R-22 performance indicate that higher energy efficiency can be obtained with R-407C and appropriate heat exchanger design.

1.INTRODUCTION

R-22 is one of the most widely used hydrochlorofluorocarbons (HCFCs) for application such as residential room air conditioners, heat pumps, and refrigeration systems. The update of the European regulation EC3093/94 implies a quick phase out of HCFCs in the next 4 years (2001 to 2004). One of the potential substitutes, R-407C, is a mixture of R-32/R-125/R-134a (23/25/52 wt%). Therefore new heat exchangers design aiming at maximum energy efficiency when using this non-azeotropic mixture deserve to be investigated.

Tests performed on single-row and multiple-row refrigerant/air heat exchangers are presented. Comparisons between R-22 and R-407C condenser and evaporator performances are also presented

2. R-407C PROPERTIES

2.1. Heat-Exchange Coefficient In Evaporating And Condensing

Because of different volatility of components, the phase change of refrigerant blends is associated with a mass transfer and a composition shift. Consequently, an associated mass resistance will exist. Many works conclude that the heat exchange coefficient decreases with refrigerant blends.

For appropriate analyses of tests and calculations, thermophysical and thermodynamic properties need to be taken into account. Thermophysical properties of the liquid phase of R-407C are more suited, as shown in table 1.

The ratio between the thermal conductivity and the liquid thermal viscosity indicates the ability to the thermal heat exchange taking into account the viscosity limitation. Because R-407C contains R-32, the blend presents both a better thermal conductivity and a much lower viscosity than R-22. Based on this comparison, R-407C could be expected to present higher heat transfer coefficient.

Table 1 – Heat exchange properties of 4 refrigerants in liquid phase
From H.M.Pham and values according to Refprop 6

Refrigerant	R-22	R-410A	R-407C	R-134a
λ Liquid conductivity (W/m.K) at 0°C	0.095	0.114	0.1	0.092
μ Liquid viscosity (mPa.s) at 0°C	0.218	0.166	0.209	0.271
Ratio λ/μ	0.435	0.687	0.478	0.339
Ratio λ/μ referred to R-22	1	1.57	1.1	0.78

Tests were performed on a test bench developed by the Centre for Energy Studies of Ecole des Mines de Paris. Figures 1 and 2 present the results for a single-row heat-exchanger, with only one circuit. The tube external diameter is 7mm, the pitch of slit fins is 1.6mm. Testing parameters are as follows:

- various mass flow rates
- constant superheating at the evaporator and
- super-cooling ranging between 3 and 5 K.

Comparisons of tested heat exchangers are based on the global heat exchange coefficient U for each measurement point. This global heat exchange coefficient is calculated from the mean logarithmic temperature difference ΔTLM.

$$U = \frac{Power}{A_{ext} \cdot \Delta TLM} \quad \text{where} \quad \Delta TLM = \frac{(Te_{air} - Ts_{air})}{ln\left(\frac{Te_{air} - \tilde{T}}{Ts_{air} - \tilde{T}}\right)} \quad \text{and} \quad \tilde{T} = \frac{h_e - h_s}{s_e - s_s}$$

Note: it is essential to use the average entropic temperature $\tilde{T}$ to compare pure refrigerants and zeotropic blends. $\tilde{T}$ permits to take into account both the temperature variation associated with the refrigerant blend phase change and the pressure losses associated with the various mass flow rates, for pure refrigerants as well as for blends.

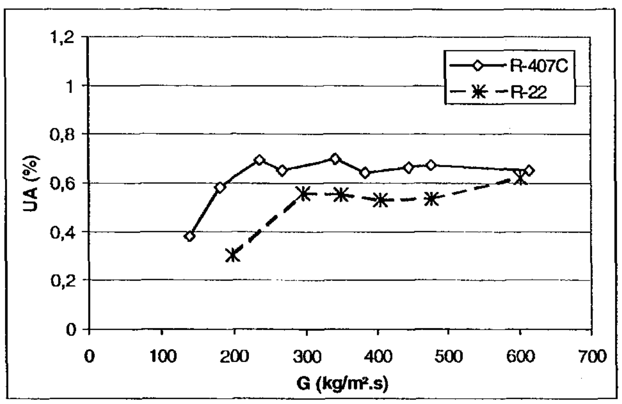


Figure 1 : UA evaporator

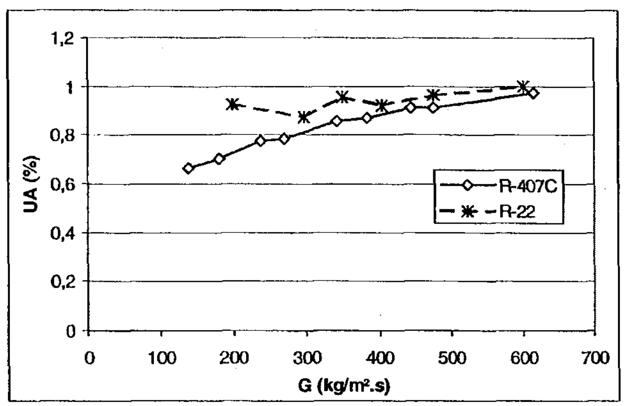


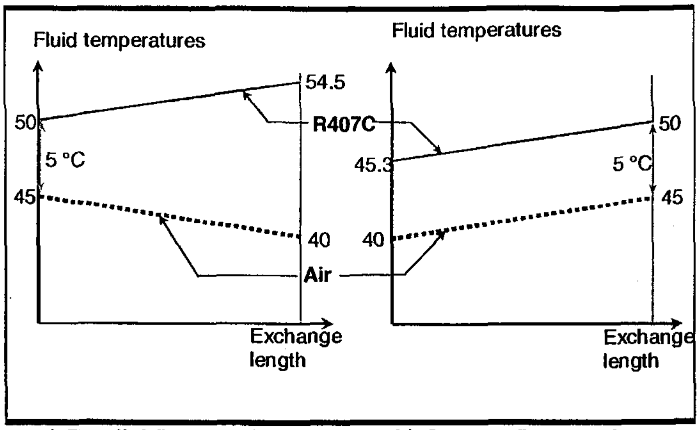
Figure 2 : UA condenser

The comparison of the two refrigerants indicate that

- at the evaporator the U coefficient is about 15 to 20% higher for R-407C than for R-22;
- at the condenser, the tendency would be reversed, the R-22 U coefficient is slightly higher than for R-407C (approximately 5 to 10 %).

2.2 Heat Exchangers Configuration: Fluid Circulations

In order to minimize irreversibilities associated with temperature differences, heat exchangers using zeotropic blends shall present a structure allowing counter-flow circulation of the heat exchange fluids. Figures 3a and 3b present the temperature evolutions of a zeotropic blend and the external flow, respectively in parallel and counter flow condensers.



a) Parallel-flow condenser b) Counter-flow condenser

Figure 3 – Temperature evolutions

Figure 3 indicates that the surface ranging between the two profiles is smaller with counter flow than with parallel flow. The generation of entropy being proportional to this surface, the counter flow circuit is better in terms of energy efficiency.

The same study can be performed with the evaporator and would lead to the same conclusion.

2.3 Optimized Design of a Multi-Row Heat Exchanger Using R-407C

For a given external surface area, several choices are possible as for the number of rows, the number of parallel circuits and the circuit configuration.

For a counter-flow circuit, the reduction of the temperature difference between the external flow and R-407C is possible. Several design parameters shall be considered:

- **Increase in the number of rows**
  This increase constitutes the possibility of counter-flow operation, but the depth of the heat exchanger shall be limited by choosing preferably fins of small width.
- **Increase in the number of parallel circuits**
  It permits the limitation of counter-flow circulation between air and refrigerant.
- **Appropriate mass flow rate**
  The refrigerant mass flow rate shall be higher than 200kg/m^2.s to prevent the degradation of internal heat exchange. If the number of parallel circuits is increased, the flow rate in each tube can become too slow. Consequently, tubes with smallest diameters shall be preferably selected.
- **Intermediate circuit length for pressure losses limitation**
  Temperature glide at the evaporator is reduced when pressure losses are significant. Because pressure losses occur mainly in association with high vapor qualities, the evolution of the state change temperature is no longer linear and presents an inflection point. For counter-flow circuit design, pressure losses shall be limited.

From a technological viewpoint, fin dimensions vary significantly as a function of the tube diameter. For example, the total depth of a two-row heat-exchanger with 9,52-mm diameter tubes is 43.2 mm (2 x 21.65mm) whereas the total depth of a three-row battery with 7-mm diameter tubes is only 38.1 mm (3 x 12.7mm). 7-mm diameter tubes with small-width fin are thus adapted for R-407C.

3. EXPERIMENTAL RESULTS OF A THREE-ROW EVAPORATOR

According to the defined design parameters and to available technical solutions, a three-row heat exchanger was designed with 7-mm diameter tubes. The three rows of this heat exchanger are separated physically permitting the measurement of air intermediate temperatures. They are alike three single-row heat exchangers in series in the air flow direction. The heat exchanger includes three parallel circuits. Figure 4 presents a partial layout of this heat exchanger.

Rows are not connected as usual because the elbow length is not minimized. For this heat exchanger where the evolution of the quality is linear, the temperature variation of each row is identical for all rows.

Tests have been performed at the evaporator for three different mass flow rates (12, 15 et 20 g/s) and 5 air velocities (0,7 – 1 – 1,5 – 2 – 3 m/s).

Figures 5 and 6 present the relative variations of UA and the mean temperature difference variation for the three refrigerant flows and various air velocities.

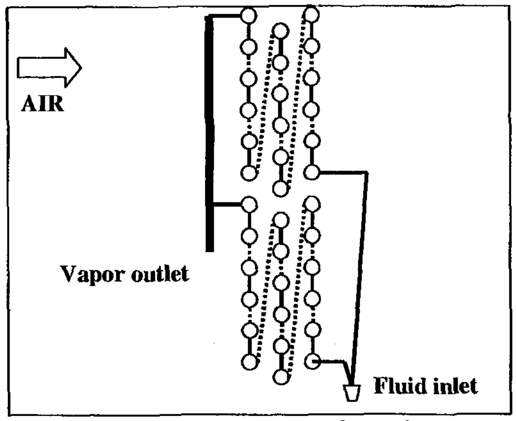


Figure 4: Circuit structure of the three-row evaporator

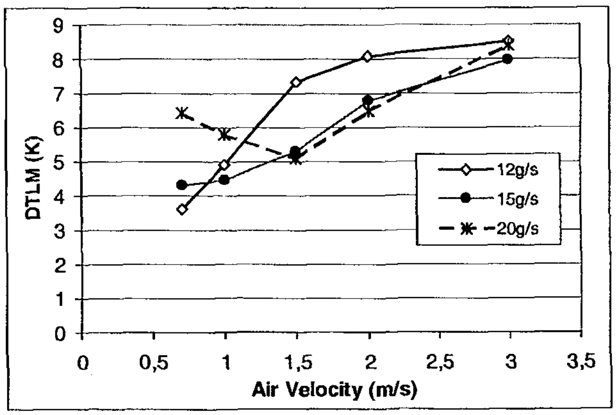


Figure 5 : ΔTLM (K) – Three-row evaporator

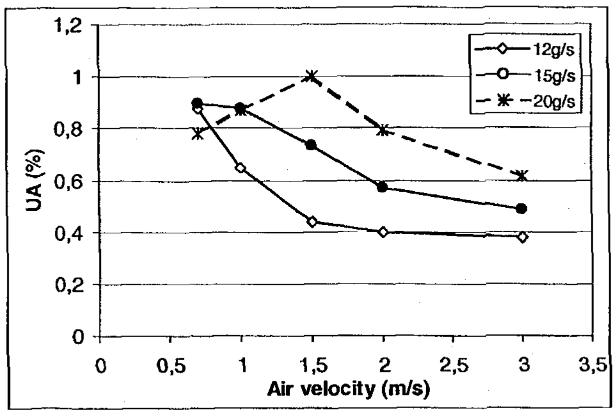


Figure 6: UA (%) – Three-row evaporator

Figures 5 and 6 indicate that for each refrigerant flow rate an optimal operation point exists for which the ΔTLM is minimized, and where the heat-exchange coefficient is thus maximum. Measurements of the intermediate air temperatures and those of the refrigerant in the heat exchanger permit to analyze the basis of this optimum.

Figures 7 to 11 present the air and refrigerant temperature differences when tests are performed for a 20 g/s mass flow rate.

Air : A : Heat-exchanger inlets
B : 1st row outlet
C : 2nd row outlet
D : Heat-exchanger outlet

R-407C : A : Heat-exchanger inlet temperature
B : Evaporator average temperature
C : saturated temperature at evaporator outlet
D : Heat-exchanger outlet temperature

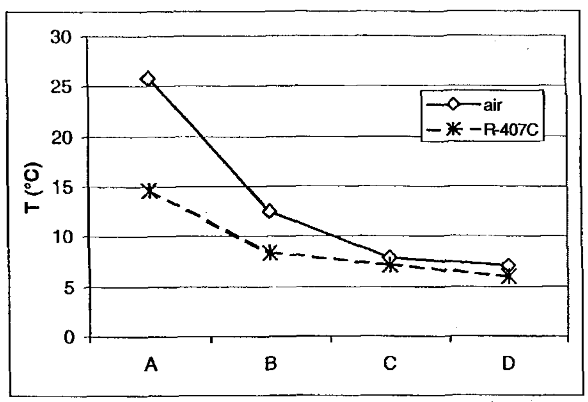


Figure 7 : Vair =0.7 m/s

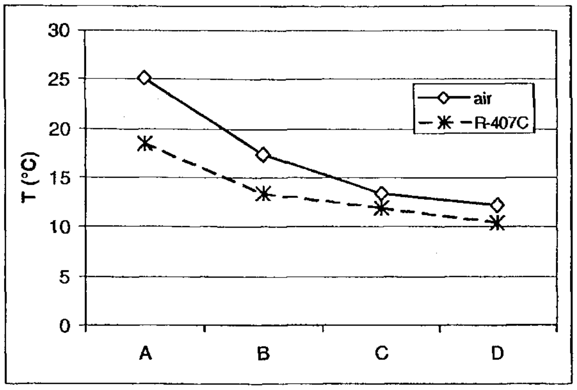


Figure 8 : Vair = 1 m/s

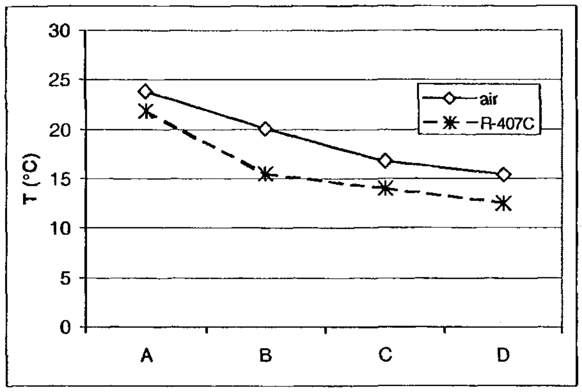


Figure 9 : Vair=1.5m/s

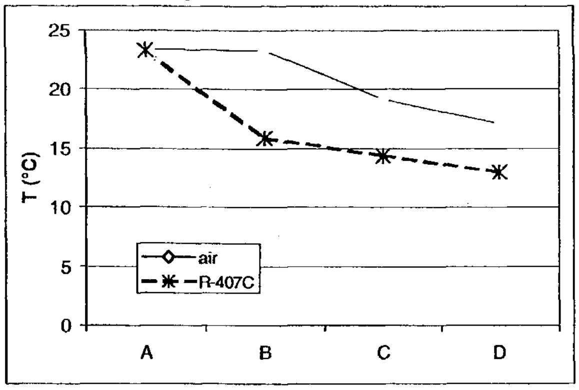


Figure 10 : Vair = 2m/s

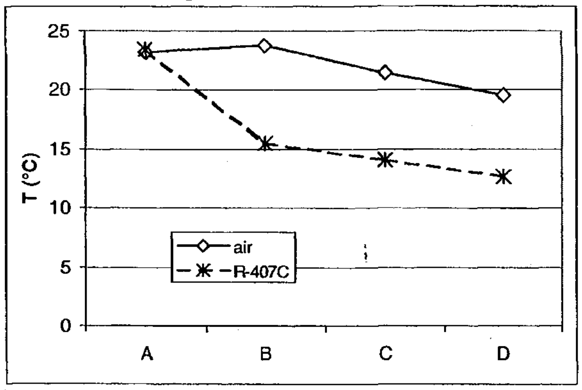


Figure 11 : Vair = 3m/s

The ΔTLM is minimum when the air and refrigerant temperature gradients are similar (Vair=1.5m/s).

If the air-flow rate is too high, the temperature differences (between air and refrigerant) at the evaporator outlet will be the limiting factor (see Fig. 11).

If the air-flow rate is too slow, the temperature differences (between air and the refrigerant after expansion) will be the limiting factor (see Fig. 7).

4. EXPERIMENTAL RESULTS OF A THREE-ROW CONDENSER

The previous heat exchanger is then tested as a counter-flow condenser (see Fig. 4). Tests have been performed for several refrigerant mass-flow rates and different air velocities.

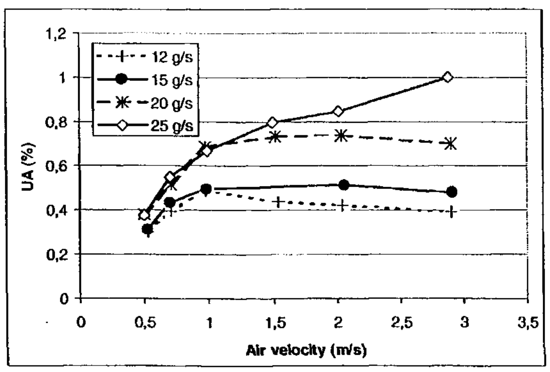


Figure 12: UA (%) – Three-row condenser

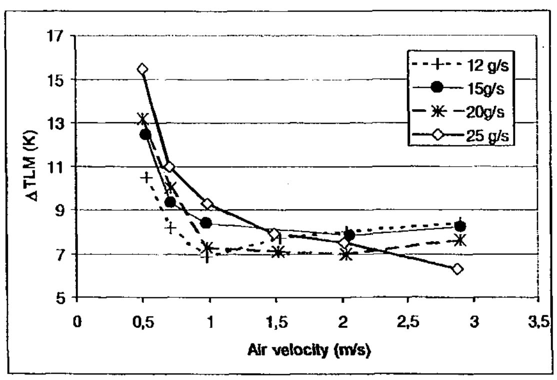


Figure 13: ΔTLM (K) – Three-row condenser

Figures 12 and 13 indicate that, similarly to the evaporator, an optimal air velocity exists for which the temperature difference is minimized.

5. IMPACT OF PARALLEL OR COUNTER-FLOW FOR A THREE-ROW CONDENSER

The use of a multi-row heat exchanger favorable to counter-current tendency of R-407C and air constitutes a potential advantage for air-conditioning systems and heat pumps. But for reversible systems, when the refrigerant flow is reversed, the advantage can turn into a disadvantage, and therefore the evaluation of this disadvantage is much appropriate.

Tests have been performed on the three-row condenser assembled in opposite direction (parallel flows). Three fluid mass flow rates (15 - 20 - 25 g/s) associated with five air velocities have been implemented.

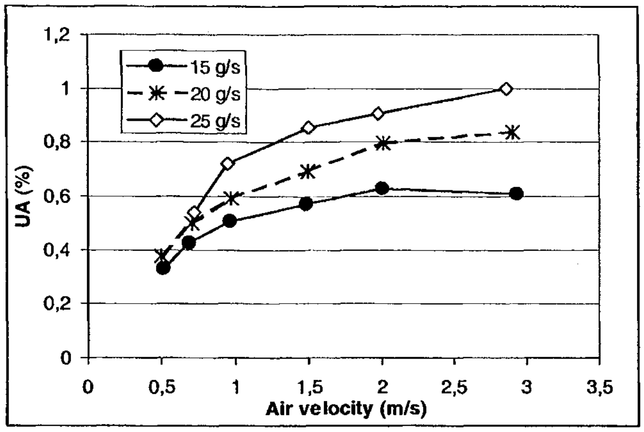


Figure 14: three-row condenser with parallel flows

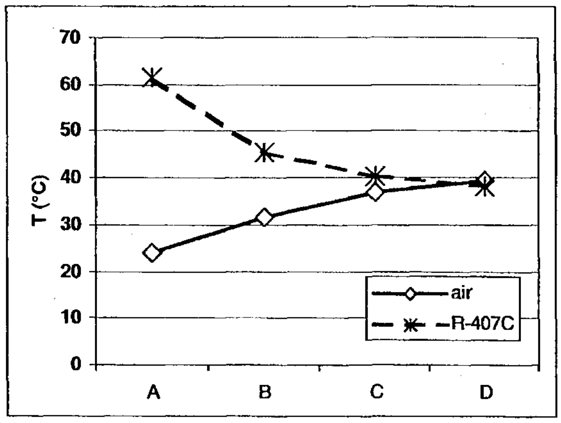


Figure 15: Vair=1m/s at 20 g/s

Air and refrigerant temperature evolutions are presented on figure 15 for a 20 g/s refrigerant mass-flow rate and air velocity air of 1 m/s. The temperature difference between air and refrigerant is not minimized, the air outlet temperature is identical to the temperature of the sub-cooled liquid; the third row is thus ineffective.

Figure 17 allows the comparison between heat-exchange coefficients of parallel flow and counter-flow three-row condensers for a 25 g/s mass flow rate. Parallel flow entails more than 50% of heat exchange coefficient loss.

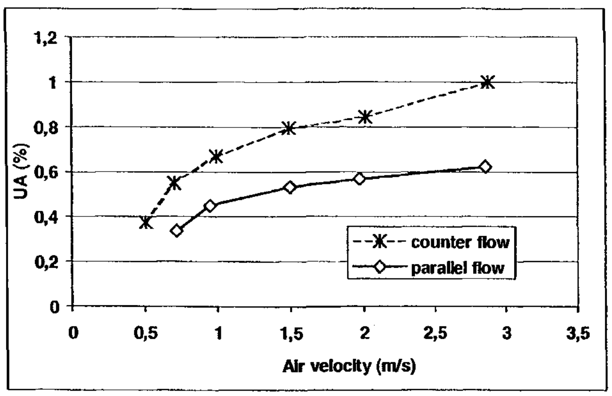


Figure 17: parallel-flow / counter-flow comparison

	Counter-flow	Parallel-flow
Air inlet Temp (°C)	25.2	23.9
Air outlet Temp (°C)	40.6	39.3
Pcond (W)	4075	4060
Entropic Temp (°C)	39.5	44.6

Table 2

Table 2 indicates that the heat-exchange coefficient decrease implies a 5 to 6 K increase in the condensing temperatures when air velocity is 1 m/s. This also entails a significant COP decrease.

6. TRADITIONAL AND OPTIMIZED HEAT-EXCHANGER COMPARISON

6.1. Evaporator

Characteristics of the evaporator are as follows:

- air / water system using R-22,
- two-row evaporator including a usual circuit
- tube diameter 9.52mm (3/8 "), assembled in two circuits
- pitch of slit fins: 1.8mm.

Figure 18 presents the circuit layout.

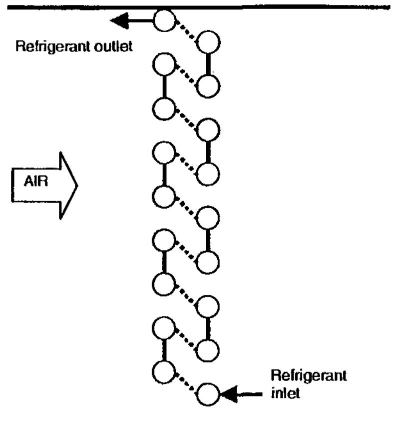


Figure 18: usual heat exchanger

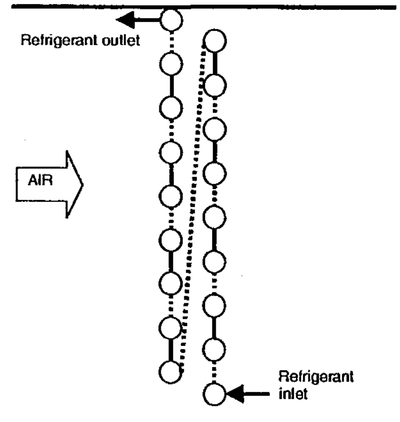


Figure 19: optimized R-407C heat exchanger

To take advantage of the temperature glide, the design of a R-407C system requires appropriate heat exchanger circuit arrangement. A heat exchanger with identical heat-exchange surface to the previous one has been modified aiming at the realization of counter flow circulation. Figure 19 present the circuit layout.

The test bench is used in the water-cooled condenser configuration. The outlet water temperature is maintained at 40°C by mass flow rate control. Air temperature at the condenser inlet is +7°C. Tests are performed for several air velocities. Three configurations are compared:

- R-22 air/water system (evaporator with usual circuit)
- R-407C air / water system (evaporator with usual circuit)
- R-407C air/water system (parallel-flow evaporator).

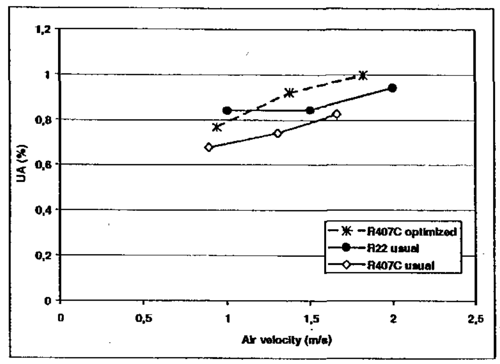


Figure 20: two-row evaporator

The replacement of R-22 by R-407C in a non optimized heat exchanger entails 10 to 15% energy efficiency loss. Appropriate arrangement of the evaporator permits to take advantage of the temperature glide and to increase the heat-exchange coefficient by about 20% compared to a usual configuration.

6.2 Condenser

The system is used in cooling mode. The heat exchanger is used as a parallel flow condenser. Condensing tests are performed for the two possible configurations, to quantify the impact of a two-row heat-exchanger. Three R-407C system configurations are compared:

- usual condenser circuit
- counter-flow condenser
- parallel-flow condenser.

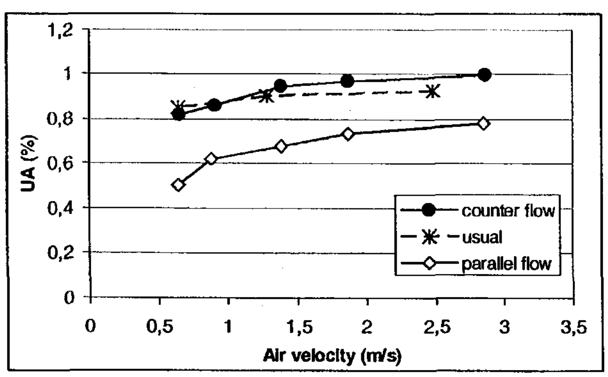


Figure 21 : two-row condenser using R407C

The global heat-exchange coefficient of the two-row condenser decreases by 25 %. The energy efficiency loss is lower than for the three-row condenser (see figure 17) because with two rows, the flow parallelism is more difficult to realize, and thus the damage is less significant.

7. CONCLUSIONS – DESIGN CRITERIA

Tests performed have permitted to define the main design criteria for design of heat exchangers using a zeotropic fluid such as R-407C. Primarily, single-row and multiple-row heat exchangers present significant efficiency differences.

For single-row heat exchangers, test results indicate that R-407C presents internal heat-exchange coefficients higher than R-22. At condenser the difference is reversed but remains low. Generally, for this heat exchanger category, the circuit design shall supply internal mass-flow rate higher than > 200 $kg/m^2.s$ in order to prevent degradation of the global heat exchange coefficient.

When dimension constraints exist and require the use of multiple-row heat exchangers the design is more complex. The use of R-407C in heat exchangers designed for R-22 entails energy efficiency loss. Although the internal heat-exchange coefficient at evaporator is higher, the global heat-exchange coefficient is lower because heat exchangers designed for R-22 do not take into account the temperature glide.

Tests have highlighted that appropriate design of air-refrigerant counter-flow heat exchangers using R-407C permits a 20% increase in the global heat exchange coefficient at evaporator and 10% at the condenser compared to R-407C usual system. Appropriate counter-flow circuit shall include a large number of rows. This implies short depth of fins. Increase in the number of parallel circuits permits to reduce counter flows. Appropriate mass flow rate presents an advantage for small diameter tubes. The minimization of the temperature difference between R-407C and air requires similar temperature glides for both of them. For a given exchanged capacity, an optimal air flow rate exists.

8. ACKNOWLEDGEMENT

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9. REFERENCES

[1] D.B. Bivens, D.M. Patron ; A. Yokozeki, Performance R-32/R-125/R-134a mixtures in systems with accumulators or flooded evaporators, 1997, Ashrae Transactions

[2] J. Swinney, W.E. Jones, J.A. Wilson, The impact of mixed non-azeotropic working fluids on refrigeration system performance, 1998, International journal of refrigeration, Vol.21, No.8, p. 607-616.

[3] C.C. Wang, J.Y. Jang, C.C. Lai, Y.J. Chang, Effect of circuit arrangement on the performance of air-cooled condensers, 1999, International journal of refrigeration, Vol.22, p. 275-282.

[4] M. Marques, P.A. Domanski, Potential coefficient of performance improvements due to glide matching with R-407C, 1998 International Refrigeration Conference, Purdue University, West Lafayette, Ind. USA. P82-91.

[5] C. Gabrielii, L. Vamling, Changes in optimal design of a dry-expansion evaporator when replacing R-22 with R-407C, 1998, International journal of refrigeration, Vol.21, No.7, p. 518-534.

[6] R. Yajima, N. Domyo, S. Taira, I. Tarutani, Selections and applications of new refrigerants for air conditioners, 1998 International Refrigeration Conference, Purdue University, West Lafayette, Ind. USA. P82-91.