2000

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Benouali, J.; Chang, Y. S.; and Clodic, D., "Analysis of the Sub-Cooling on Refrigerating Systems Using R-410A or R-404A" (2000). *International Refrigeration and Air Conditioning Conference.* Paper 466.
http://docs.lib.purdue.edu/iracc/466

ANALYSIS OF THE SUB-COOLING ON REFRIGERATING SYSTEMS USING R-410A OR R-404A

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ABSTRACT

Both R-410A and R-404A present low critical temperatures, in the range of 72.5°C. Consequently, the refrigerating capacity and the energy efficiency of the system decrease substantially when condensing temperature raises. Sub-cooling, either by external group or by superfeed, improves drastically the refrigerating capacity and the coefficient of performance.

Comparisons have been established for two evaporating temperatures (-40°C and +2°C) and condensing temperatures from +20°C to 60°C. Simulations have been performed over a one-year period, for two different climatic conditions on systems including floating high pressure strategy.

1. INTRODUCTION

Various strategies to sub-cool the refrigerant flowing out of the condenser were developed during the past . They are particularly useful for refrigerants having a low critical temperature. For example R-410A and R-404A have both a critical temperature near 72.5°C.

When condensing temperature increases, power consumption increases too, but the refrigerating capacity decreases. This effect is penalizing, because it is not consistent with climatic conditions: refrigerating capacity lowers when refrigerating needs increase. To limit these effects, many propositions were made [1, 2, 3].

Two strategies of sub-cooling are presented: the "superfeed" method, and external group. In both cases the COP and refrigerating capacity figures will be presented for various evaporating and condensing temperatures, and on a year basis for two French climatic conditions.

2. SUBCOOLING WITH "SUPERFEED" SYSTEM

2.1 System Layout

For low temperature applications (-40°C) and refrigerating capacity in the range of 500 kW, it is usual to find two-stage or "superfeed" systems. The "superfeed" technology is not adapted to reciprocating compressors but it is very much used for screw compressors. Fig. 1.a presents a typical arrangement of "superfeed" system.

At the condenser outlet, the refrigerant flow is divided in two:

- the main flow passes in a heat exchanger located in the "superfeed" capacity at the intermediate pressure of the two-stage cycle;
- the secondary flow is expanded in this capacity, the evaporated refrigerant is sucked at the intermediate pressure of the compressor.

This arrangement is only possible with screw (or Scroll) compressors where the pressure rises along the gas flow inside the screw. It is thus possible to inject a refrigerant flow at an intermediate pressure. This flow will be partially compressed from this intermediate pressure to the discharge pressure whereas the main flow leaving the evaporator is compressed from the low pressure to the high pressure of the cycle. Fig. 1a and 1b show respectively the superfeed system layout and the h-ln p diagram of the cycle.

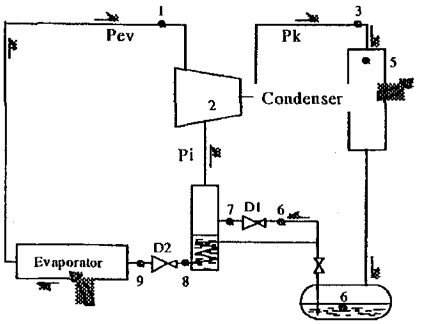

Fig.1.a - Superfeed system layout

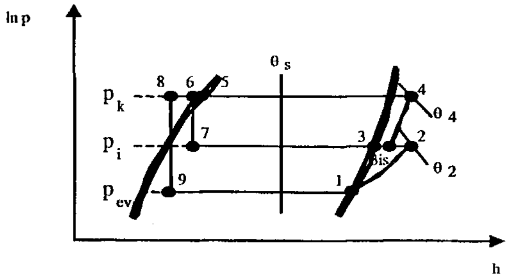

Fig.1.b - h-ln p diagram

2.2 Calculation Method

For modeling, the intermediate pressure is fixed such as the compression ratios Pi/Pev and PK/Pi are equal, which allows the calculation of the intermediate pressure according to: Pi = $\sqrt{P_k \bullet P_{ev}}$

Note: the intermediate pressure Pi of the superfeed system is fixed at a lower level than the one resulting from the previous formula, but the results in terms of refrigerant capacity and COP are quite similar. Parametric study is performed while varying the efficiency of the heat exchanger located in the superfeed capacity for several series of evaporating and condensing temperatures. The heat exchanger efficiency varies from 0 to 90%. From there, the derived refrigerant mass flow rate also varies to provide the sub-cooling capacity which thus depends on the heat-transfer surface.

2.3 Results

For each evaporating temperature, -40°C and +2°C, two figures are presented:

- variation of the refrigerating capacity referred to the single stage system (-40°C: Fig. 2.a and +2°C: Fig. 3.a),
- variation of the relative COP also referred to the COP of the single stage system (-40°C: Fig. 2.b and +2°C: Fig. 3.b).

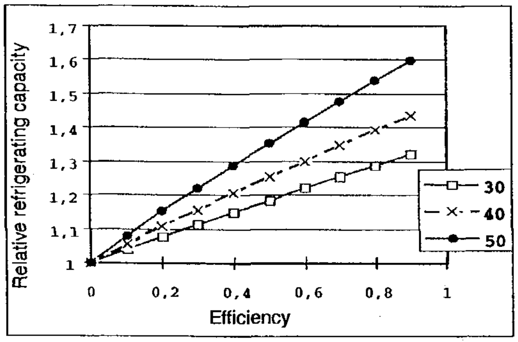

Fig.2.a -

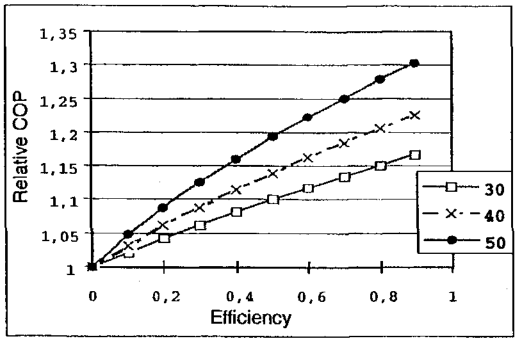

Fig. 2.b

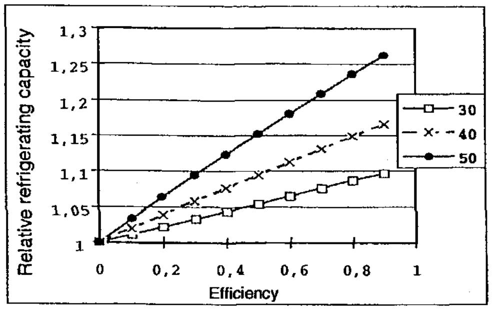

Fig. 3.a

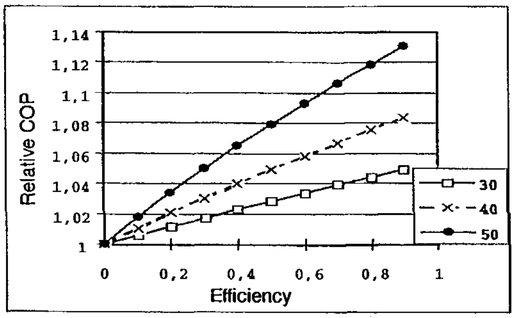

Fig.3.b

The higher the condensing temperature, the higher the cooling capacity. For a sub-cooling heat exchanger the efficiency is 90% and for condensing temperatures of 30°C and 50°C, the cooling capacity gain varies between 30% and 60%. When the condensing temperature rises to 65°C, the cooling capacity gain is about 70% and the COP gain reaches 32%. The COP improvement is significant but relatively less significant than the one in refrigerating capacity. This is due to the dominating role played by the irreversibilities associated with the isenthalpic expansion, particularly when the condensing temperature is high.

For an evaporating temperature of +2°C, the same tendencies are highlighted. The gain in terms of refrigerating capacity or COP is reduced comparatively with the one obtained at -40°C, because the temperature difference between source and sink is reduced (respectively 25% in refrigerating capacity and 12% in COP with condensing temperature of 50°C). However these gains are not negligible when the condensing temperature approaches 50°C.

So, "superfeed" sub-cooling for systems using R-410A implies significant gains in terms of refrigerating capacity and COP for high condensing temperatures. This is more significant as evaporating temperature is low. This solution can be recommended for low temperatures but also for all evaporating temperatures when the condensing temperature is equal or higher than 50°C. These conclusions are also verified for systems operating with R-404A.

3. SUBCOOLING WITH AN EXTERNAL GROUP

3.1 Operation Principle

One of the methods used to reduce the energy consumption (or to improve the coefficient of performance) of refrigerating systems working with low evaporating temperature, is to sub-cool the refrigerant flowing out of the condenser using an auxiliary refrigerating cycle (also called mechanical sub-cooler) [1]. This method is known to yield energy savings, and to increase significantly the main system refrigerating capacity, while necessary modifications are simple and non-expensive compared to new system purchase [2, 3]. Figures 4.a and 4.b presents respectively layout of a system equiped with an auxiliary group and the main loop cycle diagram.

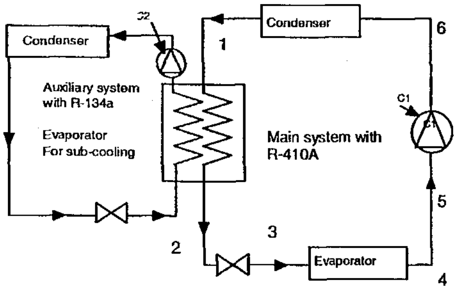

Fig. 4.a – Subcooling with external group

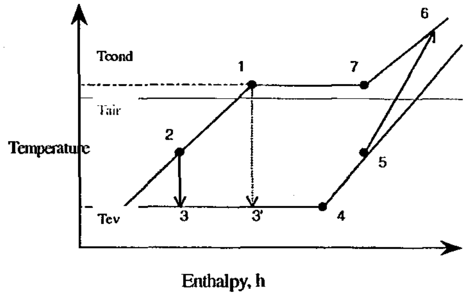

Fig. 4.b : Cycle of the principal loop

P_{MC} : input power for main and additional compressors
Q_{RMC} : refrigerating capacity
T_{cond}, T_{ev}: refrigerant condensing and evaporating temperatures.

Calculation Assumptions

For a system sub-cooled by an external group, the mechanical power supplied to the system is the sum of the power supplied to the compressors of both the main and auxiliary systems.

$$P_{MC} = P_{MC\_Main} + P_{MC\ Auxiliary}$$

The usual refrigerating Q_{RMC} is the one of the main system.

The higher the difference between T_{cond} and T_{ev}, the lower Q_{RMC} and the energy efficiency of the system. Using the refrigerating cycle COP allows to compare performances with different T_{ev} and T_{cond}.

COPs are defined as follows:

$$COP = \frac{Q_{RMC}}{P_{MC}}, \quad COP_{Carnot} = \frac{T_{ev}}{T_{cond} - T_{ev}} \quad \text{and} \quad CEF = COP.\left(\frac{T_{cond} - T_{ev}}{T_{ev}}\right) = \frac{COP}{COP_{Carnot}}, \text{ CEF : cycle efficiency.}$$

Sub-cooling permits to achieve two goals:

- reduction of the isenthalpic expansion irreversibilities (see Fig. 4.a and 4.b),
- increase in the refrigerating capacity.

Notable CEF decrease is observed when T_{ev} is low, for a given temperature T_{cond} , because the difference between source and sink decreases.

The variation between the external group evaporating temperature and the sub-cooled liquid (pinching) is fixed at 5 K.

3.2 Results For A Sub-cooling Auxiliary System Using R-134a

For each evaporating temperature, -40°C and +2°C, two figures are presented:

- variation of relative refrigerating capacity (see Fig. 5.a and 6.a),
- variation of the relative coefficient of performance (see Fig. 5.b and 6.b).

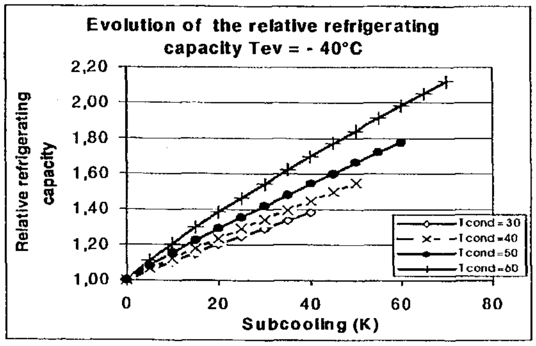

Fig.5.a

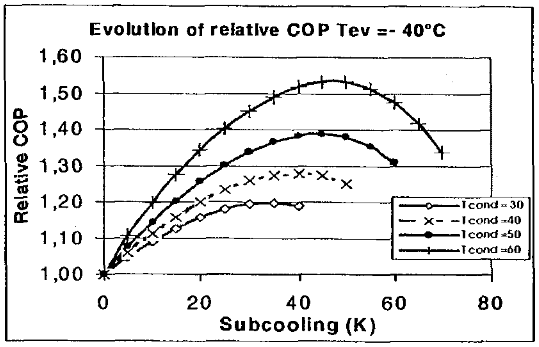

Fig. 5.b

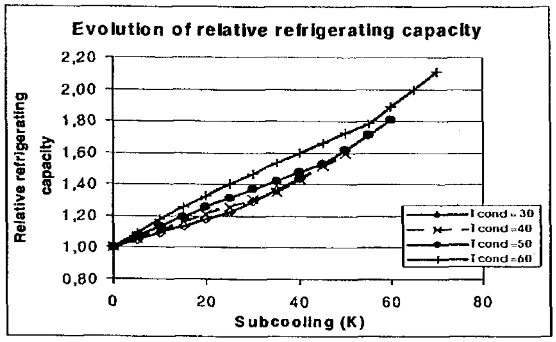

Fig. 6.a

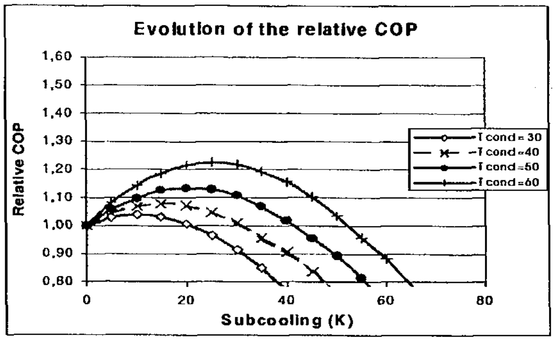

Fig. 6.b

The COP increases up to an optimum according to the sub-cooling level. For each evaporating temperature, the optimum is "drifting" to the left as condensing temperature decreases. Conversely, the higher the condensing temperature, the higher the COP gain. Finally the lower the evaporating temperature, and the higher the source/sink temperature difference, the higher the energy gain associated with sub-cooling. This strategy is more than interesting for hot climates and it is even advantageous for moderate ones. For an evaporating temperature of -40°C, the sub-cooling optimal range is between 40 and 50K, whereas for +2°C, it is between 10 and 25 K.

The higher the condensing temperature, the higher the refrigerating capacity gain. Moreover, the relative increase in the refrigerating capacity is linear according to sub-cooling.

The auxiliary group overcost can be extremely low since sub-cooling limits the over-sizing of the main group that is always designed for high thermal loads.

Influence of the evaporating temperature level

For a single-stage system, over a large variation in temperatures (-40/+40°C) the relative additional consumption of the auxiliary group is low and its COP is high. On the contrary, for the main system with a small variation in temperatures between source and sink (+2/30°C), the main system COP is almost as high as that of the auxiliary group, the global COP improvement is thus very limited.

4. R-410A AND R-404A PERFORMANCES REFERRED TO R-22

COP and refrigerating capacity of a system using R-22 were compared systematically to those of a system using R-410A and R-404A [4].

$\frac{COP_{R-404A}}{COP_{R-22}}$, $\frac{COP_{R-410A}}{COP_{R-22}}$ for COP et $\frac{PF_{R-404A}}{PF_{R-22}}$, $\frac{PF_{R-410A}}{PF_{R-22}}$ for the refrigerating capacities.

Performances of the system using R-22 are evaluated with the same sub-cooling as for R-404A and R-410A cycles.

4.1 COP Evolution

Figures 7.a to 8.b indicate that in all cases the use of R-410A and R-404A involves a performance decrease of the total system since the relationship between COPs is always lower than 1.

In both cases, sub-cooling allows to reduce the performance difference between the system using R-22 and cycles using R-410A and R-404A. Optimum of $\frac{COP_{R-404A}}{COP_{R-22}}$ and $\frac{COP_{R-410A}}{COP_{R-22}}$ exist for a given sub-cooling that depends on evaporating and condensing temperatures.

- –40°C evaporating temperature

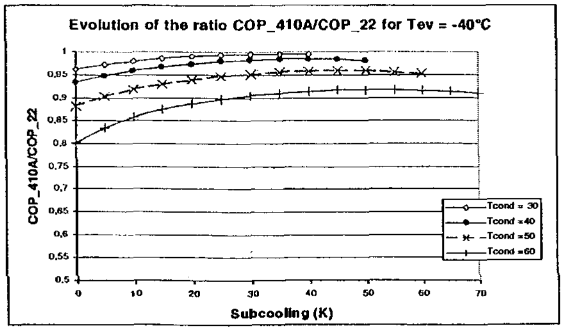

Fig. 7.a

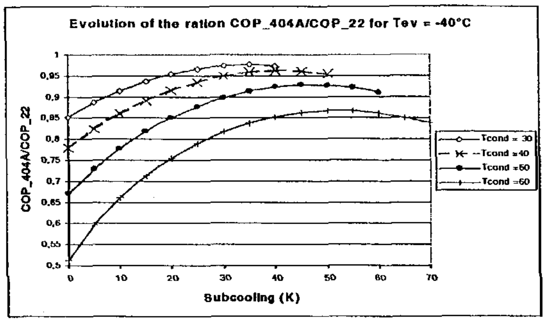

Fig. 7.b

- +2°C evaporating temperature

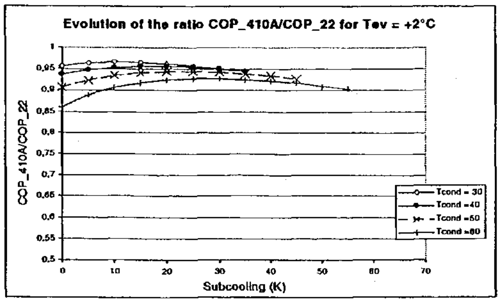

Fig. 8.a

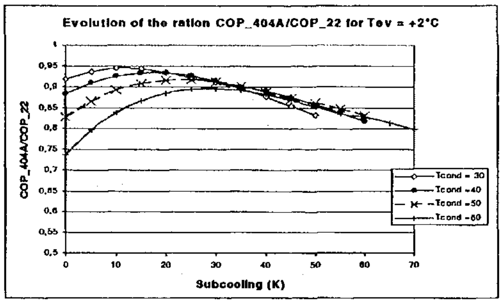

Fig. 8.b

Performances variation according to evaporating and condensing temperatures:

- for a given evaporating temperature, the lower the condensing temperature, the lower the sub-cooling necessary to reach the optimum;
- the lower the evaporating temperature, the higher the COP gain;
- the system using R-410A presents a better energy performance compared to the R-404A cycle; moreover sub-cooling reduces significantly the performance variation between the R-22 and the two blends, particularly for high condensing temperatures.

4.2 Evolution Of The Refrigerating Capacity (see Fig. 9.a to 10.b)

Using sub-cooling with an external group makes possible to increase the refrigerating capacity of systems using R-410A or R-404A compared to R-22 cycle. This evolution is quasi linear according to sub-cooling. With regard to R-404A, it appears that without sub-cooling, the refrigerating capacity supplied by the system is lower than the one available with R-22.

For a given installation, the use of R-410A allows 60% increase in the refrigerating capacity compared to R-22 system.

For R-404A, sub-cooling allows to increase the ratio of cooling capacity compared to R-22.

- 40°C evaporating temperature

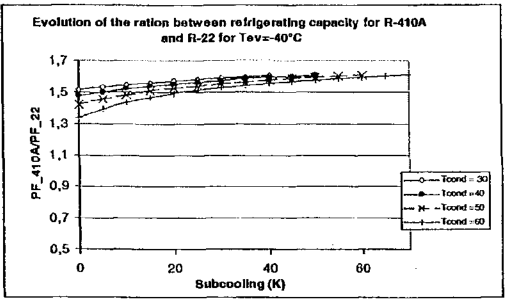

Fig. 9.a

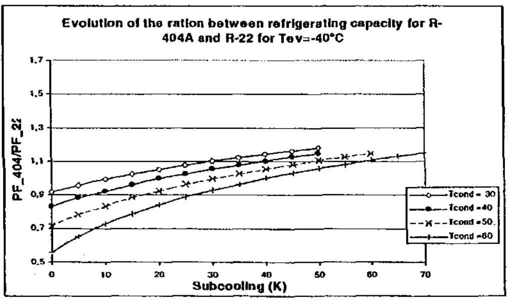

Fig. 9.b

+2°C evaporating temperature

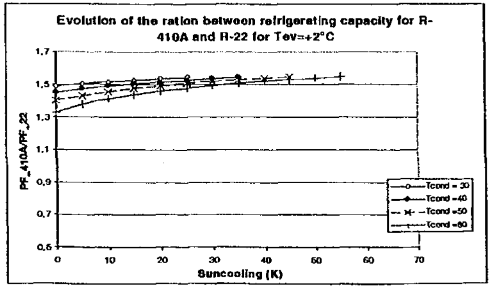

Fig. 10.a

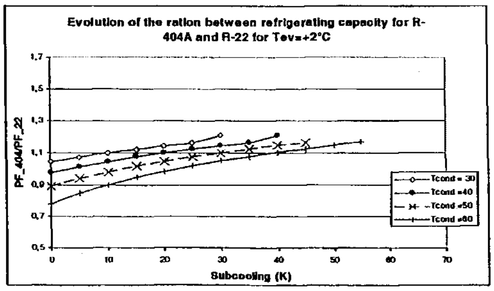

Fig. 10.b

5. COP EVALUATION AND ANNUAL COOLING CAPACITY ACCORDING TO CLIMATIC DATAS

Two French cities were selected to perform evaluations: Trappes and Nice. For each one, the temperature curves according to the number of hours in the year was used (see Fig. 11).

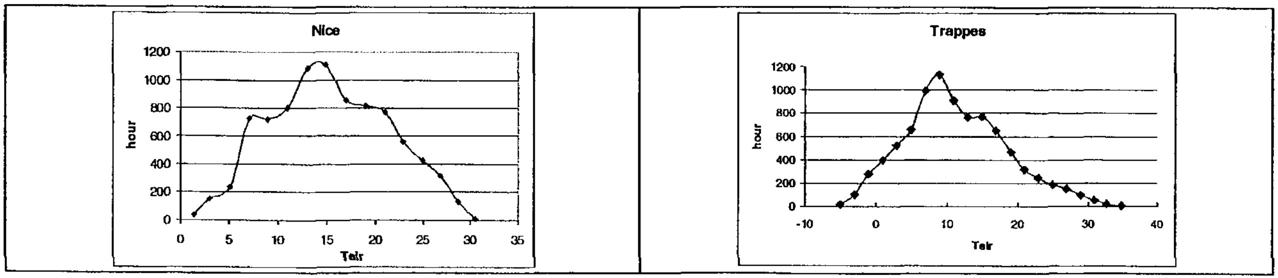

Fig. 11 : Temperatures in Nice and Trappes

Assumption

The condensing temperature is fixed according to the external air temperature and using the formula

$T_{Condensing} = T_{Air} + 8\ K + 2\ K_{subcooling}$

This formula implies that energy gains can be added with those associated with the high floating pressure. This assumption is used to estimate the energy savings obtained with the auxiliary system. For condensing temperature, a 20°C threshold was selected because it corresponds to the limit used with the high floating pressure strategy. The condensing temperature is calculated based on the air temperatures for each site. Then the annual frequency of these temperatures is taken into account for energy consumption calculation. Air temperatures higher than 10°C exist 78% of the year in Nice and 53% in Trappes.

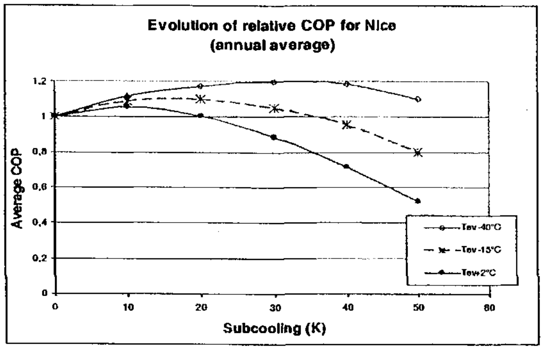

Fig. 12.a

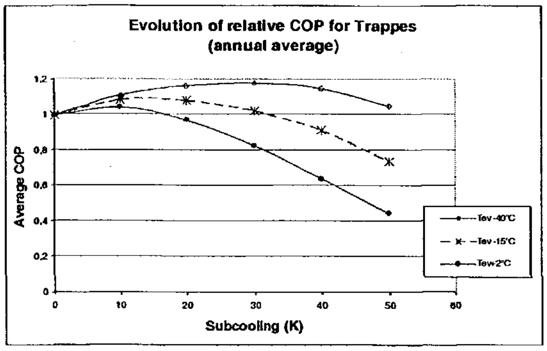

Fig. 12.b

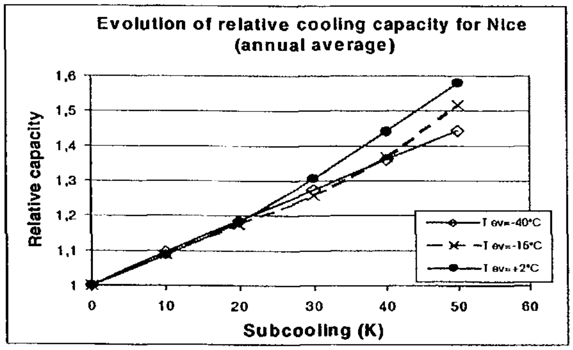

Fig. 13.a

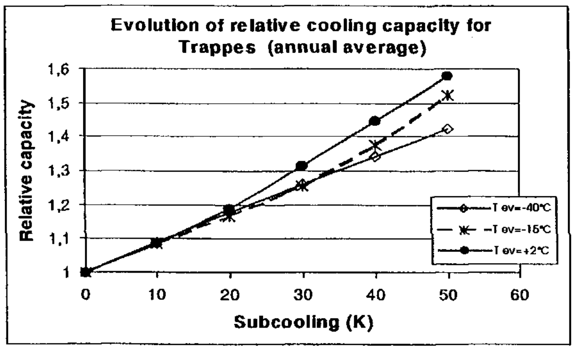

Fig. 13.b

Figures 13.a and 13.b present identical relative cooling capacity gains for Trappes and Nice. These range between 10 and 30% for the optimal sub-cooling associated with the three selected evaporating temperatures.

For an evaporating temperature of -40°C, the annual average energy gains (see Fig. 7.1a and b) are about 20% for the climatic conditions of Nice and Trappes. These gains are obtained for systems already equipped with the high floating pressure and a minimal condensing temperature of 20°C. This COP improvement is obtained for a sub-cooling of 30K. The refrigerating capacity gain is then about 18%. Such a gain allows the modification of the initial system design, which probably makes it possible to directly limit the auxiliary group overcost. Savings in term of energy efficiency and refrigerating capacity for an evaporating temperature of -15°C follow the same tendencies. On the other hand the gain in terms of energy savings is equal to 8% but the refrigerating capacity gain is about 20%. These results are obtained for a sub-cooling of 15K.

For a 2°C evaporating temperature, it is necessary to distinguish systems equiped with either "superfeed" or auxiliary group. For "superfeed" systems, the average COP gain is about 5% and the refrigerating capacity gain is about 10%. On the other hand, if an auxiliary group is used, it is more judicious to make it running only for condensing temperatures higher or equal to 30°C. The major gain is obtained for the refrigerating capacity, which is around 20% for sub-cooling of 20K.

7. CONCLUSIONS

Sub-cooling with an auxiliary group for reciprocating systems or by "superfeed" capacity for screw or Scroll compressors presents the same tendencies. The "superfeed" system permits to obtain energy gains slightly higher than sub-cooling using an auxiliary group, because the compressor output remains high for various evaporating and condensing temperatures. Sub-cooling can be used systematically for -40°C and -15°C evaporating temperatures and for all condensing temperatures. For a +2°C evaporating temperature sub-cooling is interesting when condensing temperature is high (up to 50°C).

8. ACKNOWLEDGEMENT

The results presented in this paper are issued from a study funded by the French Utility (Electricité de France, R&D Department).

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