1988

The First Industrial Application of Non Azeotropic Mixture

J. C. Blaise
Electricite de France

T. Dutto
Electricite de France

J. L. Ambrosino
Institut Francais du Petrole

THE FIRST INDUSTRIAL APPLICATION

OF NON AZEOTROPIC MIXTURE

J. C. Blaise
T. Dutto

Electricite de France
Direction des Etudes et Recherches
B. P. N°1 - 77250 Moret S/Loing - FRANCE

J. L. Ambrosino

Institut Francais du Pétrole
B. P. 311 - 92506 Rueil Malmaison Cedex - FRANCE

Abstract

Many controls, simulations and tests have been performed in laboratories or test-rigs on non azeotropic mixtures. Consequently, Electricite de France (EDF), the Institut Francais du Petrole (IFP), and Company QUIRI have decided to build together such a unit on an industrial site, using all their own top technologies to optimize the operation and the control. Such a test in an industrial environment is absolutely necessary to measure the reliability of this new technology.

Experiments were carried out on a heat pump which recovers energy from a refrigerating installation and warms water from 58 to 68° C. Characteristics of the installation being:

- 2 open-type reciprocating compressors powered by electric-motors of 75 kW; the total swept volume is 496 m^3/h at 1450 r.p.m.
- A perfect counter-flow condenser.
- Electronic expansion valves to feed the dry-ex evaporator.

As the aim of our experimentation is to compare the COP and the thermal power of the heat pump, we replace pure fluid R12 by a non azeotropic mixture. So first we determine performances of the heat pump with R12 to have a reference, then we replace R12 by a ternary mixture. As the installation includes 31 sensors (temperature, pressure, flow meter, electric power) connected to a computer, we can evaluate the influence of a non azeotropic mixture on the volumetric and isentropic compression efficiencies, and on the overall coefficient of heat transfer of the condenser. Tests with the non azeotropic mixture, give the following results (compared with R12).

* \+ 20% for the thermal power at the condenser,
* \+ 1, 5% for the coefficient of performance,
* \- 2% for the volumetric efficiency,
* \- 10% for the overall coefficient of heat transfer for the same heat flux.

At the end of the tests we successively created leakages respectively at the input of the dry ex evaporator (low pressure, aliquid and gas phases) and at the top of the receiver where gas is in equilibrium with liquid. We took a sample of one kilogramme of liquid from the receiver before and after each leakage to know the concentration of each component of the mixture. The following table shows mass concentration of each component.

mass concentration %	At the beginning of test	After leakage at the input of the evaporator	After leakage at the top of the receiver
component A	1,54	1,54	1,25
component B	71,52	71,53	69,85
component C	26,94	26,93	28,90

Initial load of the heat pump is 300 kg.

Experimental results show no variation of mass concentration when a leakage of 30 kg occurs at the input of the evaporator. The leakage at the top of the receiver lasts until gas crosses the expansion valve. Between 20 and 30% of the total load was lost. We can therefore conclude that leakages do not modify the composition of this mixture enough to change working conditions of the heat pump.

LA PREMIERE APPLICATION INDUSTRIELLE DE MELANGE NON-AZEOTROPIQUE.

RESUME : De nombreux essais, contrôles et vérifications ont été menés en laboratoire sur des boucles d'essais à propos des mélanges de fluides. E.D.F., IFP et la Société QUIRI ont pris la décision de construire ensemble en site industriel, en utilisant leurs technologies les plus évoluées, une unité mettant en oeuvre ces mélanges. Un tel essai est absolument indispensable pour mesurer la faisabilité de cette nouvelle technologie.

Les essais ont été menés sur une P.A.C. qui récupère l'énergie d'une unité de réfrigération et qui chauffe de l'eau de 58°C à 68°C. Les caractéristiques de l'installation sont :

- 2 compresseurs à pistons de type ouvert entrainés par moteur électrique de 75 kW chacun, le volume global balayé est de 496 m^3/h à 1450 tr/mm;
- un condenseur à contre-courant parfait ;
- des détendeurs électroniques pour alimenter l'évaporateur à détente sèche.

Comme le but de notre expérience est de comparer le C.O.P. et la puissance thermique de la P.A.C., nous remplaçons le fluide pur R12 par un mélange non azéotropique. Nous déterminons tout d'abord les performances de la P.A.C. avec du R12 pour obtenir le point de référence, puis nous remplaçons le R12 par un mélange ternaire. Comme l'installation comporte 31 capteurs (température, pression, débit, puissance) connectés à un ordinateur, nous pouvons évaluer l'influence du mélange sur les rendements volumétriques et isentropiques globaux et sur les coefficients d'échange globaux au condenseur. Les tests menés avec le mélange donnent les résultats suivants (comparés au R12) :

* + 20 % pour la puissance thermique au condenseur
* + 1,5 % pour le C.O.P.
* - 2 % pour le rendement volumétrique
* - 10 % pour le coefficient d'échange global du condenseur à même flux.

A la fin de ces tests, nous avons créé successivement des fuites respectivement à l'entrée de l'évaporateur à détente sèche (basse pression, phase liquide et gaz) et en partie haute de la bouteille où le gaz est en équilibre avec le liquide. Nous avons prélevé un échantillon d'un kilogramme de liquide de la bouteille avant et après chaque fuite pour connaître la concentration de chaque composant du mélange. Le tableau suivant montre la concentration massique de chaque composant.

Concentration massique %	Au début du test	Après la fuite à l'entrée de l'évaporateur	Après la fuite en haut de la bouteille
Composant A	1,54	1,54	1,25
Composant B	71,52	71,53	69,85
Composant C	26,94	26,93	28,90

La charge initiale de la P.A.C. est de 300 kg.

Les résultats expérimentaux montrent qu'il n'y a pas de variation de la concentration massique quand une fuite de 30 kg se produit à l'entrée de l'évaporateur. La fuite en haut de la bouteille a duré jusqu'à ce que du gaz traverse le détendeur. Entre 20 et 30 % de la charge totale ont été perdus. Nous pouvons donc conclure que les fuites ne modifient pas la composition de ce mélange suffisamment pour changer les conditions de fonctionnement de la P.A.C..

THE FIRST INDUSTRIAL APPLICATION OF NON AZEOTROPIC MIXTURE

J.C. BLAISE - T. DUTTO

ELECTRICITE DE FRANCE, DIRECTION DES ETUDES ET RECHERCHES
B.P. N° 1 - 77250 MORET S/LOING - FRANCE

J.L. AMBROSINO

INSTITUT FRANCAIS DU PETROLE
B.P. 311 - 92506 RUEIL-MALMAISON Cedex - FRANCE

1 - INTRODUCTION

Many controls, simulations and tests have been performed in laboratories or test-rigs on non azeotropic mixtures. Consequently, Electricité de France (EDF), the Institut Français du Pétrole (IFP) and the company QUIRI have decided to build together a unit on an industrial site, using all their own top technologies to optimize the operation and the control. Such a test in an industrial environment is absolutely necessary to measure the reliability of this new technology.

2 - TEST INSTALLATION

Experiments were carried out on a heat pump which recovers energy from a refrigerating installation and warms water from 58°C to 68°C. As we can see on figure 1, characteristics of the installation are :

- 2 open type reciprocating compressors powered by electric motors of 75 kW ; the total swept volume is 496 m^3/h at 1450 r.p.m.,

- a perfect counter-flow condenser,

- Electronic expansion valves to feed the dry-ex evaporator.

The heat pump includes 31 sensors on heat source (ammonia), on cold source (water) and on refrigerant (R12 or non azeotropic mixture). The flowmeters and the temperature and pressure gauges are connected to a computer. The installation has already been described with more details in [1].

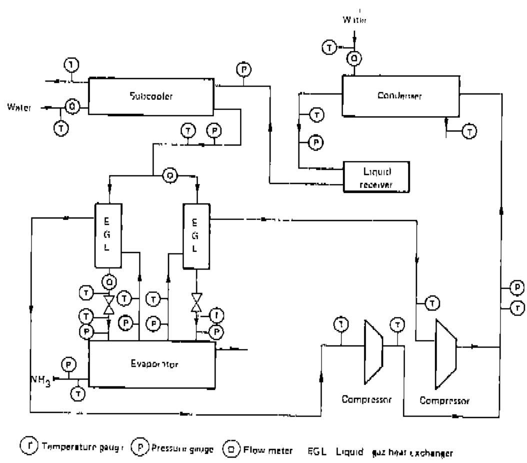

Figure 1

3 - INVESTIGATIONS METHOD

When we examined the measurements obtained we were faced with all the problems of getting precise measurements in an industrial environment (meat salting factory) and particularly the problem of stability. The instability of the heat pump in operation is mostly due to the variations of the ammonia condensing temperature in the refrigerant evaporator. As a matter of fact, the activities of the salting vary during the day and this brings about important variations of the needs of refrigeration and upsets the stability of the heat pump.

So, among all the obtained measurements, we looked for operating zones in which the variations of the ammonia pressure are small. Then we could study the performances of the heat pump versus the temperature of the heat source.

On a stable operating source, we determined an average operating point in order to calculate all the performances.

To evaluate the influence of a non azeotropic mixture on the performances of a heat pump compared with R12 we determined the following efficiencies for each fluid.

- Volumetric efficiency : ratio of the flow-rate of refrigerant at the suction point to the swept volume of the compressor.

- Isentropic efficiency : ratio of the power absorbed by an isentropic compression of the refrigerant to the power determined from the characteristics of the refrigerant at the suction and the discharge point.

- Coefficient of performance (COP) : ratio of the power Q obtained at the cold source to the electric power used by motors.

- Overall coefficient of heat transfer of the condenser h defined as :

$$h = \frac{Q}{DTL\ S}\ (kW/°C\ m^2)$$

S : external surface of the condenser (m^2)
Q : thermal power rejected by the condenser (kW)
DTL : Logarithmic mean temperature difference (°C). It is obtained from the logarithmic mean temperature differences of the desuperheating and condensing area [2].

4 - TESTS RESULTS

The curves for the average COP and thermal power of heat pump versus ammonia pressure for the same input and output temperatures of water are plotted on figure 2 and 3.

These results correspond to the expected values. As a matter of fact, simulation showed that the use of a ternary mixture would increase the thermal power by 21 % with the same COP under a ammonia pressure equal to 12,5 bars. The COP does not change because there is no temperature drop on heat source, therefore no decrease of irreversibilities on the evaporator.

TABLE 1 - INCREASE OF THERMAL POWER AND COP COMPARED TO R12

PNH_3 (Bar)	10,5	11	11,5	12
Thermal power (%)	+ 19,1	+ 21,3	+ 20	+ 23
COP (%)	+ 1,5	+ 1,6	+ 1,4	+ 1,5

We can also observe the small influence of variations of the warm source on the advantages obtained with non azeotropic mixture. So a mixture is a fluid made to measure, but it keeps its advantages even when the working conditions differ from the initial ones.

- Volumetric efficiency

Figure 4 shows the variations of volumetric efficiency versus pressure ratio. First we can notice that the values obtained are close to the values usually given for this type of compressor according to the gap of pressure between suction and discharge. For the same pressure ratio, volumetric efficiency decreases about 2 % when we use non azeotropic mixture instead of pure refrigerant R12. This small decrease of volumetric efficiency may be due to a variation of the ratio Cp/Cv which is higher for the non azeotropic mixture.

- Isentropic efficiency

The variations of isentropic efficiency for different values of the pressure ratios can not be noticed from figure 5. We can just conclude that the mean isentropic efficiencies are 70 % for pure refrigerant and 72 % for ternary non azeotropic mixture.

So we can conclude that the use of a non azeotropic mixture does not reduce isentropic and volumetric efficiencies of an open type reciprocating compressor.

- Overall coefficient of heat transfer

Experimental results (figure 7) show a small decrease of the overall coefficient of heat transfer of the condenser when we use a non azeotropic mixture. On the heat pump equipped with a coiled condenser with bare pipes, the decrease can be equal to 10 % for the same density of energy.

But we can also compare the overall coefficient of heat transfer with the same temperature of the water at the input and the output of the condenser. To obtain the same temperature drop we make the water flow-rate of the cold source vary in proportion to the thermal power rejected by refrigerant. In this case, the overall coefficient of heat transfer is about the same with R12 or non azeotropic mixture. For the example shown on figure 6 the overall coefficient of heat transfer is 916 $W/m^2\ °C$ for R12 and 933 $W/m^2\ °C$ for non azeotropic mixture.

5 - INFLUENCE OF LEAKAGES

After having evaluated the performances of a heat pump using a non azeotropic mixture we studied the influence of refrigerant leakage on the mixture composition.

For this purpose, we created two leakages : one at the input of the dry ex evaporator (low pressure, liquid and gas phasis) and the other at the safety valve of the receiver where gas is in equilibrium with liquid (high pressure). In order to determine the variations of concentration of all the components, we took a sample of one kilogram of liquid from the receiver before and after each leakage.

A leakage of 30 kg occurred at the input of the evaporator, the initial load was 300 kg and the leakage lasted 4 hours.

The leakage at the top of the receiver lasted 4 hours until gas crossed the expansion valve because refrigerant load of the heat pump became insufficient for a good working. The refrigerant loss is comprised between 55 and 90 kg, i.e 20 to 30 % of the initial load 300 kg.

The following table (table 2) shows the mass concentration of each component of the ternary mixture at the end of each leakage.

TABLE 2 - MASS CONCENTRATION AFTER EACH LEAKAGE

Mass concentration %	At the beginning of test	After leakage at the input of the evaporator	After leakage at the top of the receiver
component A	1,54	1,54	1,25
component B	71,52	71,53	69,85
component C	26,94	26,93	28,90

We can notice that the leakage at the input of the dry ex evaporator does not modify the concentration of the different components. These results obtained in an industrial installation agree with the results obtained in laboratory [3]. So we can conclude that the leakages created on a pipe, where a diphasis mixture flows, does not modify the composition of a non azeotropic mixture.

The leakage at the top of the receiver causes a bigger decrease of the most volatile components A and B.

We know the concentration of each component before and after the leakages, but we do not know the exact quantity of lost refrigerant. Although it is difficult to know the exact concentration of each component in the lost refrigerant, we can suppose that it is very close to the concentration of the mixture itself. Then if we add the lost quantity of refrigerant, with initial concentration, to the remaining quantity, the concentration is about the same as the initial load.

We can therefore conclude that leakages do not modify the composition of the mixture enough to change the working conditions of the heat pump.

6 - CONCLUSION

In this publication we have presented the series of measurements obtained on the first industrial heat pump using a non azeotropic mixture of refrigerants.

The tests carried out on the heat pump with a ternary non azeotropic mixture and then with a pure fluid, the R12, allowed us to verify the following :

- The agreement of the increase of the thermal power and of COP with the expected values.
- The absence of maintenance problems specific to the use of non azeotropic mixtures.
- The very small influence of the refrigerant leakages on the mixture composition.

These tests have showed the reliability of a heat pump using a non azeotropic mixture in an industrial environment.

This type of mixture can be a substitute for R12, the use of which is controlled because of its bad influence on the zone layer.

REFERENCES

[1] BLAISE J.C. - DUTTO T. (EDF) - CHERON AMBROSINO (IFP) - TORREILLES (QUIRI) An industrial application of non azeotropic mixture - 1987 - IIR Commission E2 VIENNE.

[2] Donald KERN - Process heat transfer - Mac Graw Hill Book Company Inc 1950.

[3] BLAISE J.C - DUTTO T. - Some practical results obtained with non azeotropic mixture of refrigerants in high temperature heat pump - 1986 - IIR Commission E2 PURDUE.

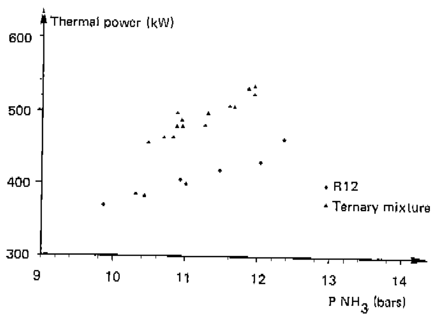

Figure 2 — Thermal power versus ammonia pressure

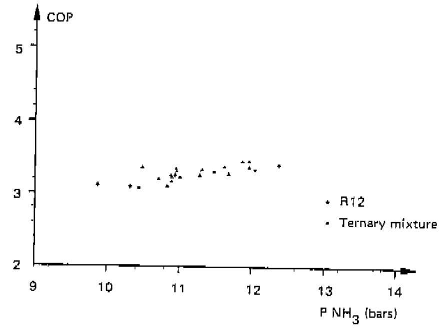

Figure 3 — COP versus ammonia pressure

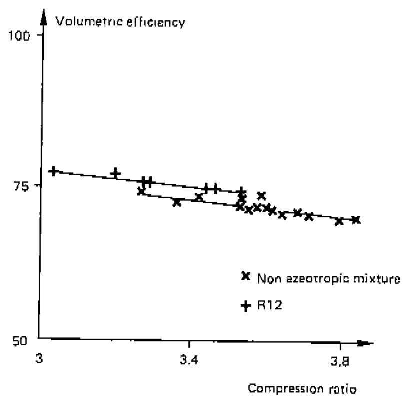

Figure 4 — Volumetric efficiency.

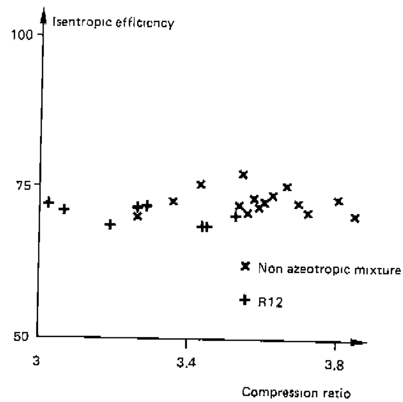

Figure 5 — Isentropic efficiency.

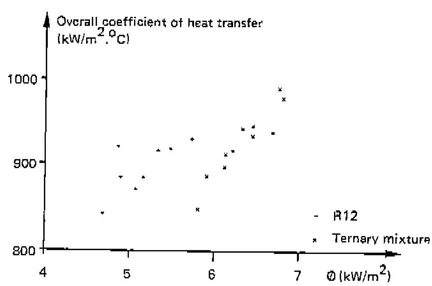

Figure 7 — Overall coefficient of heat transfer of the condenser.

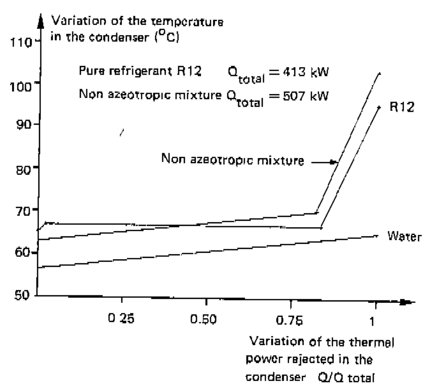

Figure 6 - Variation of the temperature of R12 and non azeotropic mixture with the same temperatures of water at the input and the output of the condenser.