1992

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COMPRESSION / ABSORPTION CYCLE USING THE WATER-AMMONIA PAIR DESIGNED FOR AIR CONDITIONNING

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ABSTRACT

A study is performed on hybrid cycles of compression-absorption. The fluid used is a mixture water-ammonia.

This system presents two key advantages, compared with compression: the large difference of temperature between the hot source and the cold, while pressure limits are kept acceptable; a glide of temperature in absorber and desorber. Therefore using it as a thermo-frigo pump (joint production of heat and cold) is possible.

The applications fit domains of HVAC (air conditionning together with hot water production, latent energy storage) and with food process.

An investigation of various types of hybrid cycles is presented. Energetic performances of those systems are compared, refered to the pure compression system using ammonia.

Various parameters such as temperatures and concentrations are studied and specially the effects on the coefficient of performance of the system has produced results enabling its optimization.

NOMENCLATURE

- Pabs, Pdes (bar)	absorber and desorber pressures	
- Tcmax, Tcmin	temperatures of absorption start and end	(K)
- Tfmax, Tfmin	temperatures of desorption start and end	(K)
- dTmax	maximum difference of temperature in the system	(K)
- Tevap, Tcond	temperatures of evaporation and condensation for compression system	(K)
- Xr, Xp (kg/kg)	liquid mass fraction of NH_3	
- ΔX (kg/kg)	difference mass fraction Xr-Xp	
- Y (kg/kg)	vapour mass fraction of NH_3	
- COP PAC	coefficient of performance when used as a heat pump	
- COP Frigo	coefficient of performance when used as a frigorific generator	

INTRODUCTION

The implication of the CFCs in the damaging of the stratospheric ozone and the consecutive international restrictions for their use have necessited the developpment of researches, partly to identify replacement refrigerants and to define replacement systems.

Hybrid systems of compression / absorption constitute an answer to the problem, for the fluid can be a mixture of components considered as harmless to environnment. Another advantage is issued of their structure, which gives them some advantages of both systems

They can provide wide differences of temperatures and a glide in heat and mass exchangers can reduce irreversibilities in heat transfers. Moreover the working pressure levels are less important. The use of mixture gives an additionnal degree of freedom to the system, which allows some flexibility. A rating gets determined by a combined set of pressure, temperatures of sources and concentrations of the solution.

Description of the system

Hybrid cycle means association of a compression process and an absorption cycle. The working fluid consists in a mixture of a solvant and a solute. The more volatile component is the refrigerant. For chosen fluid, water-ammonia, refrigerant is NH_3 and solvant H_2O. Figure A presents, through

scheme and Oldham diagram (Y-axis : Ln P, X-axis : $\frac{1}{T}$), the simplest hybrid system , considering the number of components : absorber, desorber, compressor, pump and internal heat exchanger.
More complex schemes can be designed by using several times these elements and adding two complementary elements : a condenser and an evaporator. Some authors have suggested various solutions [1,2,3,4] .

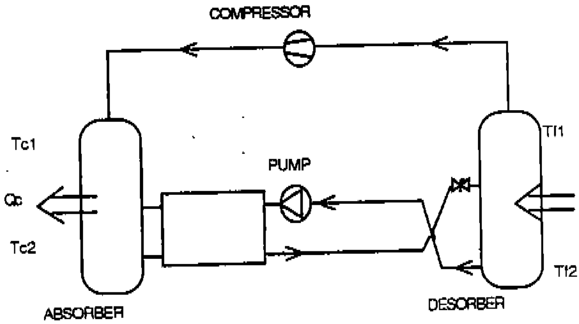
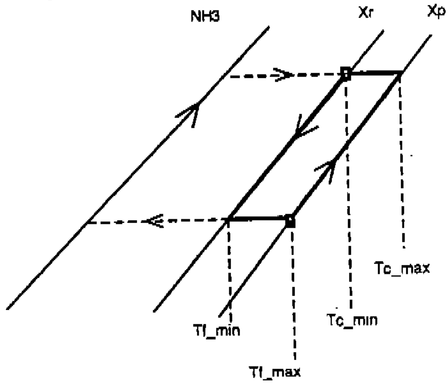


Figure A : Basic hybrid cycle (scheme and Oldham diagram)



First studied cycle is composed of a low pressure desorber producing form a rich mixture : high concentrated NH_3 vapour and a mixture poor in ammonia. The separation phenomenon require a contribution of heat at a defined temperature, which is here lower than ambient, this is producing the frigorific effect. Next, that vapour gets compressed to a high pressure to be mixed with the poor mixture in the absorber. Absorption is exothermic and can provide disposable energy. Poor mixture pumped out of desorber absorbs a part of the heat from the rich mixture out of the absorber. Required temperature levels get determined by pressure and composition conditions.

AVANTAGES AND CHARACTERISTICS FROM CYCLE

They are those of the use of refrigerant mixture :
- Disposable heat is exchanged with "gliding temperatures", in extents, there is temperature evolution during mixing and separation phenomenons in heat and mass exchangers. Depending on differences in concentration, gliding can reach important values (over 20K). That feature allows the lowering of irreversibilities during heat transfers.
- The gap of average temperatures between the cold and heat source can get very important (over 90K) whith reasonnable pressure levels in the range 1 to 25 bar (0,1 to 2,5 MPa).
- The introduction of a mixture offers an additionnal degree of freedom as regard to the compression cycle. This allows, at constant pressure, a noticeable change of the temperature level of heat transfers, then the variation is determines by a change in composition of mixtures. The process provides adaptativity to variable function rates.
- The working fluid, NH_3- H_2O, is not prohibited by rules about CFC, concerning environnement

safe. Moreover ammonia is a valuable refrigerant, of common use and properties to the pair are fairly known.

APPLICATIONS

The hybrid cycle is most convenient for joint production of heat and cold, such as for a thermo frigo pump design. Fluid covers a temperature range from -40°C to +200°C. The hybrid cycle is especially fit for applications joining large temperature difference between sources and important required variations on exchangers on either cold and hot side.

Applications are planed in the domain of climatization for tertiary industries as a thermo frigo pump together with hot sanitary water production. Our prospective extends to an interesting possibility of cold storage via ice production.

INTRODUCTION TO SIMULATION MODEL

We have based calculations of fluid state at each point of the cycle and heat exchanges on energy, mass and species balances, which are established on frontiers of constitutive elements of the system.

Main hypothesis are:
- absorption and desorption are non ideal phenomenons, therefore, we use coefficients of efficiency to describe the processes;
- the highest temperature in the absorber is the one to the poor mixture, at absorption pressure, which is assumed as a constant;
- the lowest temperature in desorber is the one to the rich mixture, in equilibrium at desorption pressure, following the expansion, which is supposed isenthalpic;
- the regenerative solution heat exchanger is modelled with an efficiency;
- the compression is calculated, including an effective coefficient, depending on compression ratio.

The properties of the solution H_2O-NH_3 are modelled according to the equations proposed by Ziegler et Trepp [1]. The properties program offers four entries : (P,T), (P,X), (P,Y) et (X,Y); other variables : enthalpy, entropy, specific volume and the two left state characterics among (P,T,X,Y) are calculated.

The simulation program offers two kind of inputs :
- whether equilibrium temperatures in absorber and desorber together with frigorific power;
- or pressures in absorber and desorber together with frigorific power;

Iterations are performed on the mass fractions Xr and Xp with a variation on extreme temperatures in exchangers : Tcmax et Tfmin.

From these inputs, properties are calculated for rich (SR) and poor (SP) solutions out from absorber and desorber, respectively. Temperatures at the entry of the regenerative exchanger are determined, then output temperatures are calculated, using an exchange efficiency of 0,8. After expansion, the two phase state of SR at equilibrium gets defined from its enthalpy before expansion, low pressure and species balance. The simulation program does not include the secondary exchange of application in extents, no external exchanger is modelled. A vapour rectifier condensing water excess is assumed in order to provide quasi-pure ammonia (over 99,5%) to the compressor, (it is not displayed on figure A). Actually, it was proved this element is not necessary for the systems we have investigated, for the reason that the pressures and temperatures are sufficient to satisfy this stress. The level of superheating is controlled during compression, which is necessited by technological limits of the compressor. Another cooling brings vapours to the temperature of absorption start; the cooler, contributes to calorific production of the machine; not displayed on figure A, it is situated just before absorber.

Study of the basic hybrid cycle and comparison with the compression cycle

First, we study performances of the system with the minimum temperature difference between absorber and desorber, Tcmin - Tfmax. We define the maximum difference that can be realized while keeping a compression ratio lower than 10. This limit is the one of commonly used compressors and is not corresponding to limits on fluid, which can fit for a large range of temperature. Therefore, as displayed figure B, an additionnal compressor is implemented, in order to increase the capacities of the structure. That very system is composed of two successive compressors with a cooling loop for vapours rejected by the low pressure compressor. The circuit is of "total injection" type (i.e. evaporation in high pressure circuit balances condensation of low pressure circuit) . A part of forced vapour at high pressure gets condensed and expanded in a tank at average pressure where hot vapours from low pressure compressor bubble. The condensation causes a significative loss of energy for the system. The performances of this second cycle are calculated, too, in order to examine the transition

between both structures.
We add a comparison with the compression cycle using pure ammonia. The hypothesis for calculations for compression part are the same as exposed before, for hybrid cycle, moreover, we neglect pressure drops in both evaporator and condenser.

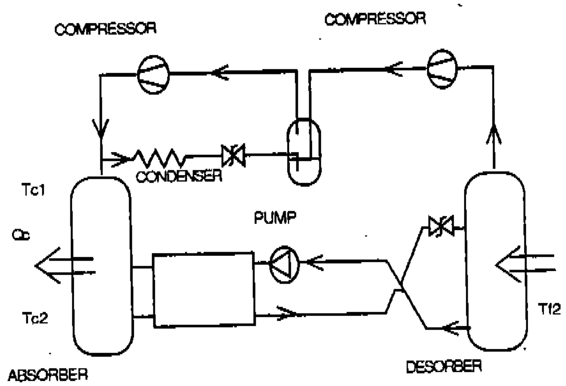


Figure B: Basic hybrid cycle with two compressors

Comment about the comparison :

The choice of temperature level is of great importance for the comparisons of various cycles [5].

The three fonctionning types of the system can be represented by Carnot cycles, where temperatures keep constant and Lorentz, where temperatures are variing during heat exchanges.

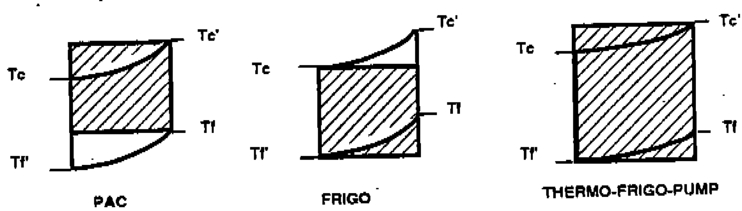


Figure 1 : Functionning types (Carnot cycle is signaled by stripes)

For a disposable heat exchange, two temperatures are to be considered, concerning the fluid to be heated or cooled : starting and end of exchange. The first temperature is not necesseraly known (by instance ambient air temperature), the second is the one required.
In a transfer not considered as part of the production, one single temperature is taken in account : the one to the source exchanging with the system. These are stresses to be assumed by the system.

In the case of a chiller, these stresses are defined for the two following systems :

* hybrid system :
  - the desorber provides the frigorific production which maximum (Tf) and minimum (Tf') temperatures are those of beginning and end of exchange.
    The temperature glide in the desorber is corresponding to the one of the fluid to be cooled (minus temperature drop necessary for exchange);
  - heat is rejected at a temperature level over (at least equal) disposable source; it will be the minimum temperature in the absorber (Tc), the maximum (Tc') is not valorized.

* compression system :
  - the frigorific production is realized at the temperature at the end of the exchange (Tf') which is the constant temperature of evaporation;
  - the condenser rejects heat at the temperature of disposable source (Tc)

We re-define those stresses in the case of a thermo frigo pump :

* hybrid system :
  - extreme temperatures in the desorber are corresponding to those of the fluid to be chilled (minus temperature drop necessary for exchange);
  - extreme temperatures in the absorber (Tc' and Tc) are corresponding to those of the fluid to be heated (increased of temperature drop necessary for exchange).
* compression system :
  - the evaporation temperature (Tf') is the minimum temperature at the end of exchange of the fluid to be chilled;
  - the condensation temperature (Tc') is the maximum temperature at the end of exchange of the fluid to be heated.

Correspondances for temperatures as exposed above is valid only in the case of opposite current exchangers.

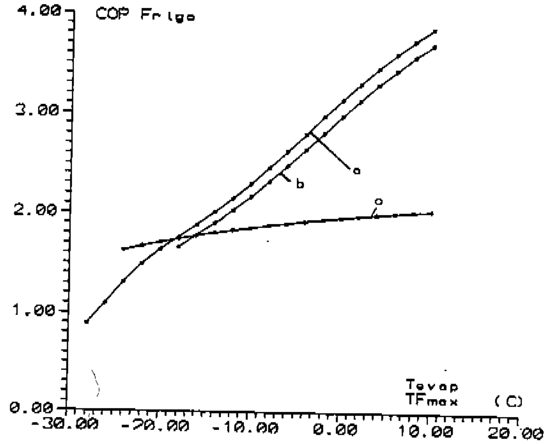


Figure 2: Frigorific COP as a function of ΔT_max for systems
(a-compression, b-hybrid 1 comp., c-hybrid 2 comp.)

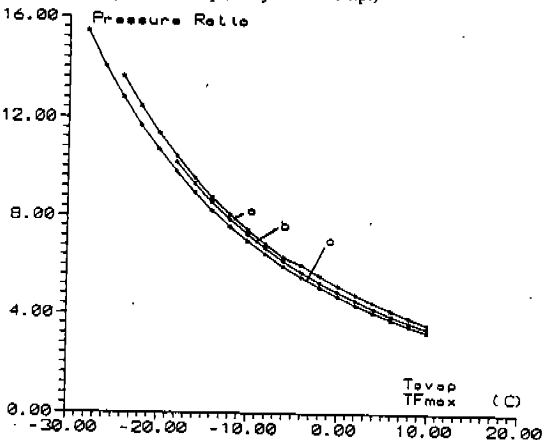


Figure 3: Pressure ratio as a function of Tf_max and Tevap for systems
(a-compression, b-hybrid 1 comp., c-hybrid 2 comp.)

Comments about graphics based on temperature difference between sources :

We have performed the simulating calculations for a chiller using Tcond= Tc_min = 50°C and Tevap = Tf_min = [10,-20°C]. Hybrid cycles are optimized for each temperature level according to the concentration of the mixtures. Figure 2 displays the variations of frigorific COP with Tevap = Tf_min, figure 3 shows the variations of total compression ratio (Pabs/Pdes and Pc/Pe).

We can remark that :
- the compression cycle has a performance better than the one of the cycle presented figure A (some%) the difference proceeds from the compression ratios;
- the twin compressor hybrid cycle is noticeably less performant for the following reasons : the energy loss created by the total injection circulation, first, and a lower isentropic efficiency of the compression due to the low pressure ratios of both compressors. When the compression ratio reaches 10, the single compressor system approaches its technological limits and its COP becomes lower than COP of two compressors system.

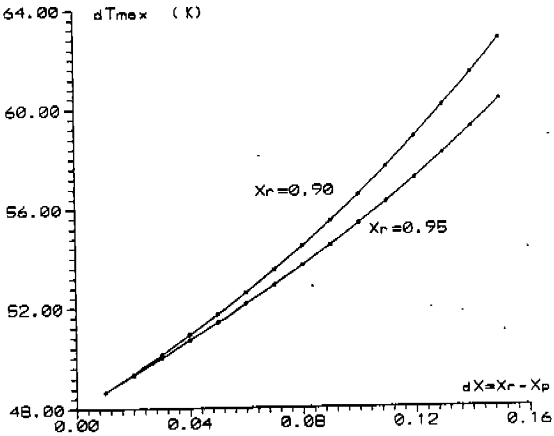


Figure 4: ΔT_max as a function of ΔX for various X_R

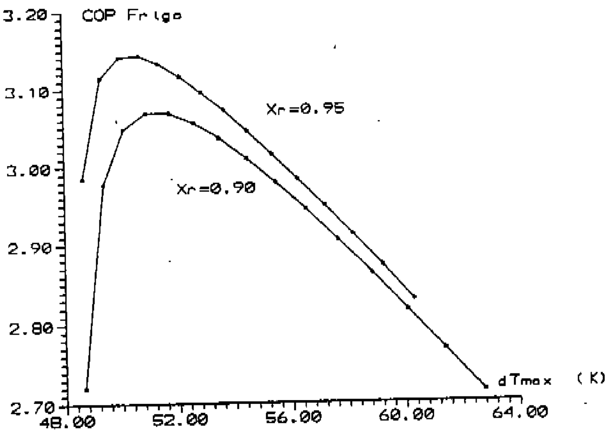


Figure 5: Frigorific COP as a function of ΔT_max for various X_R

The limit of the single stage hybrid cycle is determined by a compression ratio of 10 at a desorber temperature Tf_max = -10°C. That is to say when the difference of temperature between sources is about 60K. That is the reason why the twin stage compression structures provides a better performance.

Comments about graphics based on temperature glide in exchangers :

Considering variations upon temperatures in the desorber, we fix up the following conditions :

Tc = 50°C, Tf = 2°C, X_R = 0,95 et X_R = 0,9 with variing X_P. Figure 4 shows a variation ΔT_max = Tc_max - Tf_min, of the difference between extreme temperatures in the system with the concentration gap ΔX, monotoneus and increasing with ΔX. The temperature glide in both exchangers is extending with ΔX. The two curves show the major sensitivity of the glide to concentration levels, when low in ammonia.

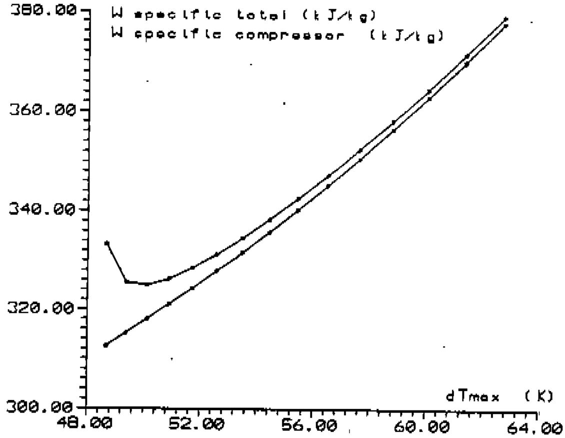


Figure 6: Total specific work (compressor+pump) as a function of ΔT_max

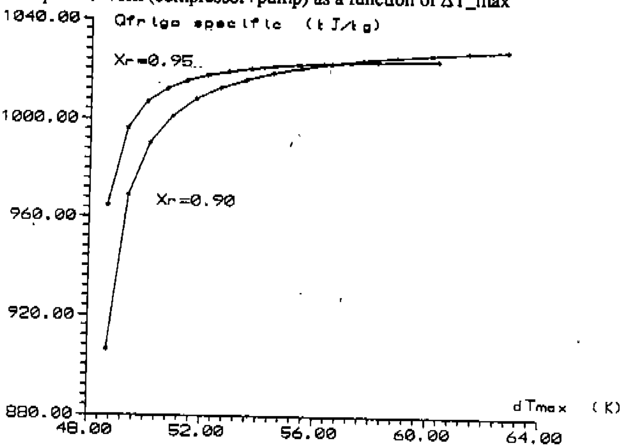


Figure 7: Frigorific specific production as a function of ΔT_max for various X_R

The frigorific COP is displayed on figure 5, it presents a maximum. Figures 6 and 7 show that mixing energy for low ΔX decreases noticeably whereas specific work increases; that phenomenon explains the maximum of COP. The difference of COPs on figure 5 has two origins:
- the compression ratio is higher for low concentrations, this is valid for any value of the maximum difference between temperatures in absorber and desorber (ΔT_max), as shown figure 3;

- the second reason to COPs differences is that for a low concentration gap (ΔX close to 0,02), the heat needed for desorption is strongly dependent to the level from mass fraction of the rich solution (X_R), which explains that maximum COP difference is reached for low ΔT_max.

Figure 6 displays total mechanical work used in the system by both pump and compressor. For high values of the maximum temperature difference ΔT_max, the part of energy consumption required by pumping is low and can even be omitted in energy balance. On the contrary, when ΔT_max decreases, the relative need of pump gets important, which explains the inflexion of the curve (ΔT_max=50K) and the increase of total work for lower ΔT_max. This steep increase illustrates the strong dependance of pump requirement to concentration gap ΔX, acknowledged that ΔX determines ΔT_max.

Best performances are provided when concentration levels of ammonia in solutions are important. The difference of concentration between the two solutions allows the optimization of the cycle it terms of performance. This result had been exposed by Ahlby and al.[6] for a heat pump.

PRESENTATION OF DIFFERENT TWO STAGE HYBRID STRUCTURES

We study here different cycles which appeared interesting. They are designed for providing simultaneously heat and cold. The different systems are simulated by using the same modellisation program, calculating for the single loop of solution described by figure A. We add a coupling module designed for connecting the two elementary loops. The optimization, on COP criterion, of the complete system is realized through iteration performed on every free variables.

Two stage hybrid cycle with solution mixing (figure C) :

This system is very close to the compression system. Two loops at different pressures link an absorber at high pressure, a desorber at low pressure and a mixing tank at mid-pressure. That last element is behaving as the desorber of the rich mixture. Desorption energy in provided by the cooling of vapours out from the low pressure compressor. The mid-concentration (X_m) solution is formed in the mixing tank through the mixing of S_R, rich solution (X_R) and S_P, poor solution (X_P) and with desorption. The choice of the value of Xm allows to overlay concentrations X_R and X_P and such, the temperature gradients in desorber and absorber.

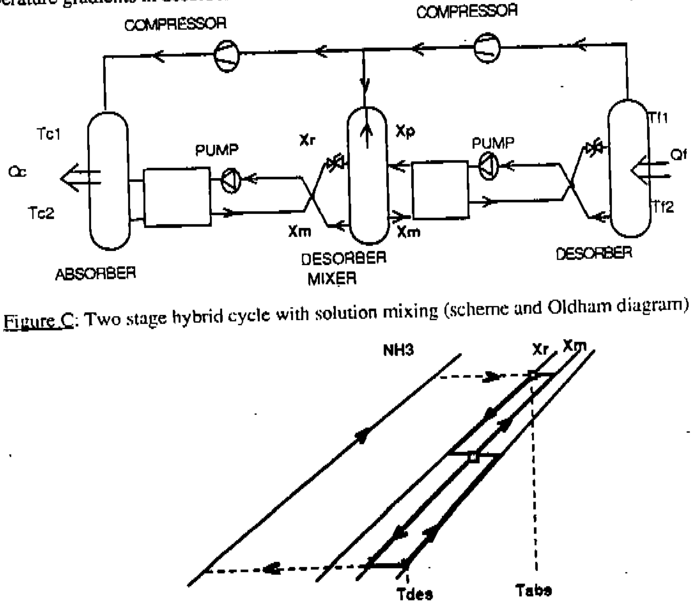


Figure C: Two stage hybrid cycle with solution mixing (scheme and Oldham diagram)

Two stage hybrid cycle with heat coupling (figure D) :

Two single stage hybrid systems get coupled through a thermal exchange between the absorber of first stage and desorber of second. That exchange can be realized by simple contact on walls or through a common annexe circulation. The stage of high pressure produces the disposable heat for absorber, and the stage of low pressure provides the frigorific effect at desorber. This design allows no communication between solutions of both loops, so the concentrations are free variables. And this results in the fact that each loop can be optimized by variations of compositions; consequently, the whole system can be optimized.

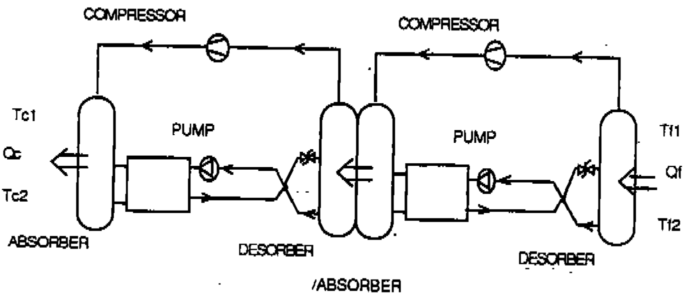
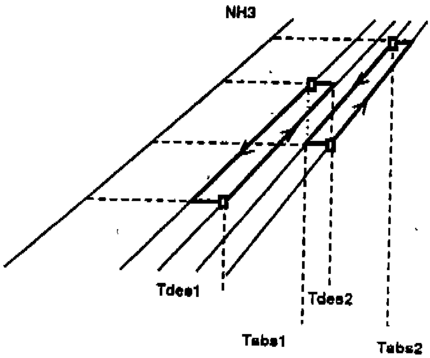


Figure D: Two stage hybrid cycle with heat coupling (scheme and Oldham diagram)



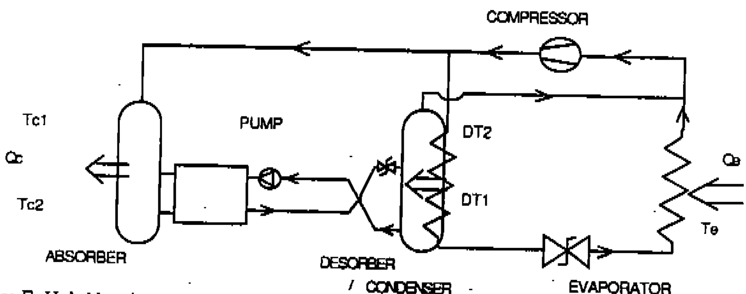


Figure E: Hybrid and compression cycle coupled with a common compressor

Hybrid and compression cycle coupled with a common compressor (figure E) :

Two systems, one of hybrid type using water-ammonia mixture, the other of compression type using pure ammonia, are coupled through thermal exchange. Moreover, the condenser pressure is the same as this of in desorber, the pressure of evaporator, the same of this of in desorber. As both temperatures of absorber, for disposable heat and from evaporator for cold energy, are fixed, there remains one single free variable, for instance XR or XP. Actually, the evaporation pressure fixes desorption pressure and such, the coupling temperature (if XP is fixed) that determines pressure in condenser and absorber. The condenser is cooled by the desorber.

The advantage of this system is to provide low compression ratios for high differences of hot and cold temperature. It combines the effect of lowering the saturating vapour pressure, due to mixture in absorber in comparison with low evaporation pressure of pure ammonia (compared to the mixture NH_3 - H_2O). Yet, on the other hand, one single exchanger is working with a temperature glide, which, once fixed by the remaining free variable with Tc_min and Tevap, determines the other parameters of the system. This can lead to concentrations XR and XP that do not optimize the hybrid cycle and so, decreases the performance of the whole system.

Thermal coupling of hybrid and compression cycle (figure F) :

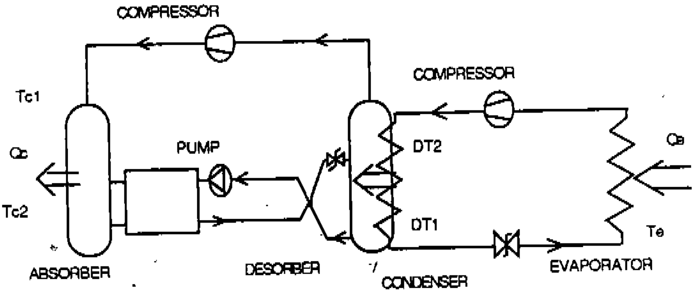


Figure F: Thermal coupling of hybrid and compression cycle

A heat exchange couples a hybrid loop with a compression one. The scheme of figure D is used again with differenced pressure circuits of both loops. Compression ratios are, in this case too, lower than for other systems.

Results of comparisons :

We have compared these cycles between them and with the reference of the two stage compression system (G). As formarly explained, two types of calculations were lead : COP evolutions according with temperature gradients in exchangers, and with difference of temperatures between hot and cold sources. The fonctionning is the one of a thermo frigo pump, i.e. we are considering that temperatures of system are equal to those of external fluid (+necessary temperature gap for exchange).

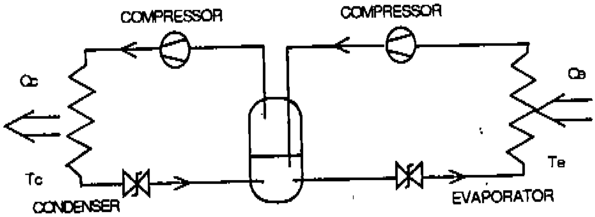


Figure G : Two stage compression system

Evolution of ΔT_min :

Results are given here for Tc_min = 80°C and Tf_max = 2°C. Figure 8 shows COPs of these systems as function of ΔT_min. The Coefficient of Performance is defined as the fraction of disposable heat (calorific and frigorific production) to total work. The ΔT_min is the difference between the minimum temperature on hot side and the maximum temperature on cold side.

We remark :

- system D and F have best performances until ΔT_min reaches 90K and system E gets better over. This last system is moreover requiring a lower compression ratio;
- system C gets second as regards to performances for $\Delta T\_min<80K$, which, once taken account of its simplicity places it as interesting solution for that range of temperature;
- system B has lowest performance, because of the losses in the cooling circuit for exhaust vapours from low pressure compressor;
- compression system G keeps lower in terms of performances to system C until a ΔT_min of 90K and lower to systems C,D, E under all conditions.

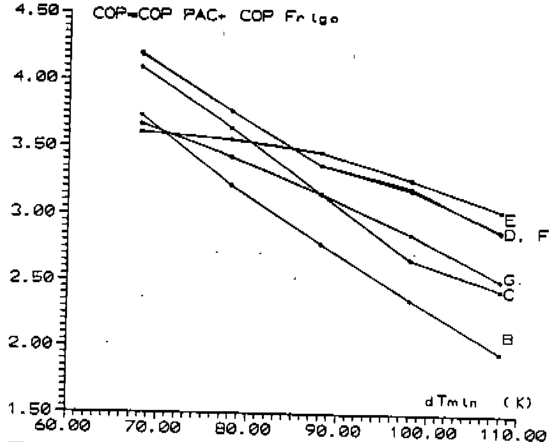


Figure 8: Thermo frigo pump COP as a function of ΔT_min for the 6 systems

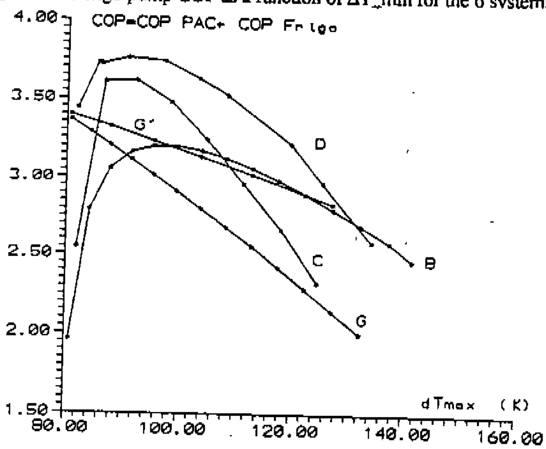


Figure 9: Thermo frigo pump COP as a function of ΔT_max for the 6 systems

The choice of the system depends together of required differences of temperatures and of economical stresses for the cost of the machine is linked to the complexity of its design.

Sensitivity to the glide in exchangers :

In this study results are given for the same value of glide in absorber and desorber for each system.These two gradients are variing with the mass fraction gap ΔX.

A detailled approch of glide for system B has been presented by authors [7] .

Figure 9 presents the variation of the COP of the thermo frigo pump using systems B, C and D compared to the compression cycle in two cases :
- G-case : evaporation temperature equal to Tf_min in desorber and condensation temperature corresponding to Tc_max in absorber;
- G'-case : condensation and evaporation temperatures defined as the average temperatures respectively in absorber and desorber.

We suppose, in this comparision, that the output temperature of the fluid to be heated reaches the maximum temperature in absorber (in fact, slightly lower due to the drop necessited for exchange). The respective hypothesis is made for hot external fluid with lowest temperature in desorber. Performances of compression system G are noticeably lower than those of other systems with exchangers with gliding temperatures. This functionning is the one advantaging the more the hybrid cycle, which was not the case in the former comparison. Then we can note that the way of considering reference temperatures for comparison of hybrid and compression cycles is essential. Besides, the hybrid cycle B, C and D provides better performances for applications where the suitable glide in exchangers can be valorized.

CONCLUSION

We have performed a detailled study of various systems using hybrid cycles. A comparison has been made between them refered and to the results of the compression of pure ammonia. The hybrid system is using a mixture and gets its advantages such as gliding temperatures.

Performances are different according to the structure. The performances are similar in one stage structures, for hybrid and compression systems and the hybrid may may be more efficient if temperatures in exchangers can be valorized, that is to say if input and output temperatures of treated fluids are close to temperatures in system exchangers.

Concerning two stage cycles it can be defined hybrid cycles with better performances than compression systems, but the complexification of their structure leads to balance solution between energy save and conception cost.

REFERENCES

1- B.Ziegler and Ch. Trepp
Equation of state for ammonia-water mixtures.
Revue Internationale du Froid, Vol.7, N°2, Butterworth & Co ltd and IIR 1984.

2 - L.Ahlby, D.Hodgett and T.Berntsson
Optimization study of the compression / absorption cycle.
International Journal of Refrigeration, Vol 14, January 1991.

3 - E. Morawetz
Sorption compression heat pumps.
International Journal of Energy Research, Vol.13,83-102 , 1989.

4 - S.Arh and B.Gaspersic
Faculty of mechanical Engineering, Murnikova 2,61000 Ljubljana, Slovenia, Yugoslavia.
Developpment and comparaison of different advanced absorption cycles.
Revue Internationale du Froid, Vol. 13 janvier 1990.

5 - M.O.McLinden and R.Radermacher
Methods for comparing the performance of pure and mixed refrigerants in the vapour compression cycle.
Revue Internationale du Froid, Vol.10 Novembre 1989.

6 - L.Ahlby
Compression / absorption systems. Simulation of two cycles for different applications.
Department of heat and power technology, Chalmers university of technology, Gothenburg, Sweden.

7 - D.Alo, J.M.Baleynaud, D.Clodic
Systeme à compression / absorption pour la climatisation, utilisant le couple eau-ammoniac
Colloque SFT, Sophia Antipolis (F) mai 1992.