SACMO: SOLAR ASSISTED COOLING MACHINE WITH OPTIMIZED UTILIZATION OF SOLAR ENERGY

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1 Partnership

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2 Objectives of the project

The result of this project is an absorption cooling machine optimised for the use of solar energy and with an high efficiency for the use of natural gas. We believe that solar assisted cooling will enhance the installation of solar collector systems especially in Southern Europe due to its higher potential compared to solar heating.

In fact the present active solar thermal market is mainly in hot water production. Space heating is not done. One reason is that the heating season is relatively short and that there is little use for solar collectors for the rest of the year (a space heating system requires more collector area than for hot water production) with the result that the economics of space heating is not there. However with solar cooling becoming possible, the economics for the use of solar systems becomes more interesting, with the system now providing all the needs for heating (including water) and for cooling. This means that the potential impact on the development of a solar market for this project is a multiplication by a factor of 5 to 10, that being the ratio of collector area for all thermal needs to the collector area needed just for hot water production.

With the results gained we hope to stimulate the engagement of different European companies not only into the field of solar cooling but also into absorption chillers in general which already have a large market in the USA and Japan.

Last but not least this project can help to gather experience for the development of a small solar assisted absorption cooling machine for residential housings as well as for the development of a pure solar cooling system to be used in developing countries where cooling for food storage and hospitals is a major problem.

A pilot plant of an optimised solar assisted absorption cooling machine in the capacity range of 25 *kW*, which can be driven simultaneously with heat supplied by a thermal solar collector field and a gas furnace was developed.

The concept of the absorption machine developed here is a double-effect absorption chiller which is gas fired at the high-pressure generator but with a modified middle-pressure generator allowing to drive it not only by the internal heat of the high-pressure condenser but also directly by hot water from thermal collectors. Essentially this machine acts as a high efficient double-effect absorption chiller with an high COP of about 1,1 when driven by gas and as a single-effect absorption chiller with a COP of about 0,6 when using solar energy. The main task was the development of this new middle-pressure generator and the coupling to both the high pressure condenser and the solar collector field. Especially the varying shares of solar to gas put strong emphasis to the control strategy which was of course influenced by the technical design of the machine.

The major tasks of the project are summarised below. For each task also the partners involved are indicated.

1. Definition of the principle layout of the machine and the control strategy (solar <u>or</u> gas versus solar <u>and</u> gas with variable share) (ZAE, ATECNIC, INETI, INTA). Especially in solar systems with different components of collector, heating or cooling equipment, the combination of the parts and the control system are extremely important to get a good performance of the entire system. This is even more valid for the absorption machine where solar

energy and natural gas will be used simultaneously. Therefore the control of the machine was one major part of the work programme. Most of the parts of the machine (evaporator, absorber, middle-pressure condenser, high-pressure generator) are near to state of the art components. However, the required compactness posed additional questions for these components, too. The research and development has been focused on the construction of the middle-pressure generator which is able to use alternatively or simultaneously condensation heat from the high-pressure condenser as well as from the solar collector field.

2. Development of software tools (ZAE, Uni Valencia) to incorporate performance curves of different absorption machines in a standard simulation program like TRNSYS has given the possibility to judge the solar yield of a certain system including the special absorption machine developed for different applications and climates.
3. Hardware design and dimensioning of the components for the prototype; development of the final control and regulation strategy (ZAE, ATECNIC). In this low capacity region the required small dimensions of the machine put further constraints to the used technique and the actual design of the components.
4. Set-up and first tests (ZAE, ATECNIC) of the prototype. After the construction of the parts in the technical workshops, the set-up of the machine and preliminary checks (like leakage tests) the principal performance should have been tested and necessary changes in parts of the machine performed to obtain the performance looked for.
5. Extended tests (ZAE, ATECNIC, INTA, INETI, Univ. Valencia). The prototype had to be measured under realistic conditions with special regard to its COP, refrigeration capacity and dynamic behaviour. The measured detailed performance curve should have been used in a simulation including different collector types to judge the annual solar yield etc.
6. Review of the design (ZAE, ATECTNIC). With the results and experience of the tests the layout of the machine and especially the constructive details should have been reviewed and necessary changes and improvements towards a commercial product discussed.

3 Scientific and technical description of the project

3.1 Machine layout and general control strategy

According to the concept it was necessary to find a design which allows for two different heat coupling points in the machine.

In particular a middle pressure generator, which works at about 100 °C was developed. It is normally heated by an internal heat exchange but is able to use external hot water (solar) producing cold with single effect performance.

A further objective was to develop a control strategy which takes two aspects into consideration:

The machine has to be able to utilise the whole solar heat gained by the collector field. This means that the cycle should not only be able to switch between double and single effect operation mode but also to be driven in intermediate modes. Simultaneously the solar input, although with lower temperature, should have priority to the gas input.

The water consumption for the chiller waste heat dissipation should be minimised, since the market for such solar assisted units is concentrated in countries with unsatisfactory water supply. One possible strategy is to dissipate the heat to the ambient through an air stream as long as the needed driving temperature difference is available and to turn on a supplementary water sprinkling device (in order to cool down the ambient air) if the ambient air temperature rises.

An effort of the project partners has been, like already mentioned, to conceive the machine components with respect to the later standard production. As a result of this it has been decided to build the main heat exchangers, i.e. evaporator, absorber, middle pressure condenser, middle pressure generator and high pressure generator, as shell and tube heat exchangers in a rectangular casing. Merely the high pressure condenser should be built as a plate heat exchanger. This decision was based on the fact that the square form allows a much more compact design than the conventional cylindrical form which, however, allows to use a thinner sheet metal.

A special attention was paid to the partition of the tube bundles of the main heat exchangers. This means the way of positioning the tubes on top of each other. For each main heat exchanger we chose different geometries of partition depending on following criteria:

- Compactness of the heat exchanger bundle
- Good wetting behaviour
- Keeping of the minimal drip height between rows for absorption processes
- Keeping of the minimal distance between tubes meeting the requirements for the tube rolling

Another investigated item concerned the kind of manufacturing of the heat exchanger tube bundle.

Due to the calculated thermal expansions it was decided to build the high pressure generator with one floating header and the other heat exchangers with fixed headers.

The evaporator and condenser are more or less standard components, so the further layout efforts focused on the three chiller components absorber, middle pressure generator and high pressure generator, which are described in detail below.

3.1.1 Absorber

A further step concerning the design of the machine was to investigate different heat dissipation options. Like mentioned above it should be tried to build the chiller air cooled. Waste heat is produced at the middle-pressure-condenser and at the absorber (low-pressure-level). The latter is the more critical component due to the low heat transfer coefficient and the crystallisation risk.

Three options for the absorber design have been taken into consideration:

a) Direct air cooling in order to avoid multiple heat transfer and thus to keep the internal absorber temperature as low as possible. Here the most important item is the design, and consequently the efficiency, of the heat exchanger, which must assure an acceptable heat transfer coefficient between absorbing solution and ambient air.
b) A so called adiabatic absorber, which permits a separation of heat and mass transfer. The real mass transfer (absorption) happens in an adiabatic vessel while the heat transfer from the LiBr-solution to the cooling air (absorption heat) happens in an air cooler.
c) A very common water cooled absorber with an additional air cooler connected through a water loop.

Again the necessary heat exchanger area has been calculated for each case.

Case b) yields the largest area due to a higher temperature difference required compared to the other cases. The other two options are very similar, in spite of the additional water loop of case c). The reason is the large heat transfer resistance on the air side, which leads to the fact that the heat transfer on the liquid side is of reduced importance. Finally case c) has been favoured because with a water loop the absorber geometry is completely uncoupled from the air cooler geometry. In this way an unit can be built, that could also operate with a wet cooling tower in applications which require a guaranteed cold supply at a fixed temperature (e.g. hospitals etc.).

A further item concerning the absorber development was the investigation of a solution distribution which allows for very cheap manufacturing and which is able to work with very small admission pressures. This last point means that the apparatus should be able to distribute the solution stream regularly on a large area, without using a high admission pressure with the associated high pumping work. As this is of great interest a test facility has been built and tested at ZAE Bayern although this was not included in the work programme. This distribution concept represents an alternative to the conventional spraying system with nozzles which is very frequently used in the industry. The latter has for sure the advantage of a very fine distribution due to the spraying effect but it needs admission pressures in the range of 300-1000 mbar which must be generated by a corresponding high pump work.

3.1.2 Middle pressure generator

Concerning the solar heat coupling into the middle-pressure-generator different alternatives have been discussed. Four cases have been regarded as useful:

a) The vapour from the high-pressure-generator flows in the high-pressure-condenser (same vessel like middle-pressure-generator: internal heat exchange!) like in a standard DE chiller. After the heat exchanger the condensate pipe is branched off. The two branches lead to the evaporator and to a “solar heat exchanger”, which could be of the plate heat exchanger type. If solar heat is supplied the condensate is pumped to the solar heat exchanger where it evaporates (solar heat is taken up) and afterwards it is fed back into the vapour pipe between high-pressure-generator and high-pressure-condenser. In gas operation mode the condensate is led after the middle-pressure-generator into the evaporator as usual.
b) The vapour from the high-pressure-generator flows in a plate heat exchanger (high-pressure-condenser) where it condenses. The condensation heat is taken by a heat transfer medium which flows to the middle-pressure-generator, heats up the solution (regeneration) and returns to the high-pressure-condenser (loop). If solar heat is supplied this loop is opened, the medium is heated up in a solar heat buffer, flows through the inactivated high-pressure-condenser, gives off its heat in the middle-pressure-generator and is finally pumped back into the solar buffer.
c) The vessel for the middle-pressure-generator contains two heat exchange bundles: a first one which is supplied with vapour from the high-pressure-generator and a second one which is supplied with the heat transfer medium from the solar buffer. On both of them, which could be arranged on top of each other, the solution will then trickle down and be regenerated.
d) Option c) can be changed in order not to design the two heat exchanger bundles on top of each other but into one another (coaxial pipes). In this way only one pipe-package would be needed, which the solution trickles on. Compared to case c) this would result in a compact design and avoid problems with the solution distribution.

These four options have been compared. A double effect chiller cycle has been simulated with calculation programs developed at ZAE. The corresponding temperatures, mass flows and heat transfer coefficients have been used for calculating the total heat exchanger area needed for the middle pressure generator. Total area means the area of both heat exchangers needed for the double effect and for the solar operation mode. Case a) results in a large area because of the two serial heat transfers between solar heat source, intermediate loop and LiBr-solution combined with a small total driving temperature difference. Case b) also has the drawback of two serial heat transfers, but in the DE mode, i.e. with a big driving temperature difference at its disposal. Advantage of this circuit type is on the other hand the permanent use of the generator heat exchanger area in both operation modes, that leads to a reduction of the total installed area. Case c) and d) have in this respect the drawback of a limited use of the heat exchanger area, i.e. each apparatus only works either in the DE or in the SE (solar) modus, which leads to a higher installed area. Finally option b) has been chosen due to its easier way of manufacturing, which is in fact a combination of standard “shell & tube” and commercial plate heat exchanger construction.

3.1.3 High pressure generator

It was decided to build the high pressure generator as a “shell and tubes” pool generator and to heat the solution indirectly , i.e. by a heat transfer medium between water/LiBr-solution and gas burner. This was decided since a direct firing of the generator, even if it is for sure the most economical solution, needs extensive tests in order to prevent corrosion processes caused by so called hot spots.

Two alternatives for indirect heating have been recognised: a fluid loop with thermal oil and a heat pipe system.

Two heat pipe types have been investigated:

a) Conventional heat pipes with capillary feedback (internal structure). The manufacturers interviewed do not have any experience with direct heating and are afraid of an evaporator destruction in case of a too little heat exchange in the condenser (not enough feedback in the evaporator). Highest operation temperatures are 150 °C.
b) Two-phase thermosiphon heat pipes. They have to be installed vertically or at least with 5° inclination. The heat capacity is limited by different mechanisms:

- Evaporator dries out, e.g. due to insufficient contents or too high heat flux density.
- Burn out limitation due to film boiling: with too strong evaporator heating the evaporation changes from pool boiling to film boiling. This way the working fluid can be decomposed and the wall material destroyed.
- Mutual obstruction of vapour and condensate flow, which can even stop the feedback to the evaporator.

From all this we regarded the risk of a heat pipe damage or of its unsatisfactory operation as quite high. In fact the design of heat pipes for such a high temperature range of operation requests a well assessed know-how as well as the optimisation for the part load behaviour.

After intensive investigation of this option it was decided to focus on the option of a heat carrier loop. As possible heat transfer media steam and thermal oil have been taken into account. After receipt of several manufacturers offers the steam option was rejected due to the high investment costs caused by the gas fired steam generator system for little capacities like 20 kW. Thus it was decided to use a thermal oil heater.

3.2 Development of software tools

The goal of this task was to develop appropriate software tools in order to allow an economic and energetic comparison among the chosen operation.

In order to get more exact information about solar contribution and gas consumption an operation simulation on the basis of the TRNSYS program has been carried out similar to the POSAC study. For four different cities on the Iberian peninsula, both on the coast and in the interior, several parameters like total energy demand for heating, cooling and hot water, real solar contribution, gas consumption and economics, have been simulated for two different systems: a small house with 120 m² and a small hotel with 600 m² living space. Different types of absorption chillers were calculated in order to find out if coupling the solar heat into the double effect generator gives an essential improvement. Three kinds of absorption chiller and solar collector have been compared:

a) A standard single effect solar absorption chiller with a COP of about 0.7 and a driving temperature at the generator of about 80 °C;
b) A single/double effect solar/gas absorption chiller with COP respectively of 0.7 at 80 °C and 1.1 with gas direct firing,
c) A double effect solar absorption chiller with a COP of 1.1 and a solar driving temperature of 150 °C.

Of course also the pure single and double effect units use gas for heating the generator (water fired type), if no solar radiation is available. The three investigated collectors are of type CPC (Compound Parabolic Concentrating), which in previous simulations proved to have a better price/performance relation than other types (POSAC-study):

a) A commercial one of the Setsol company with an optical efficiency of 0.75 and a global heat loss coefficient of 3.7 W/m²K;
b) The same one but with a better transparent insulation and therefore with an optical efficiency of 0.71 and a global heat loss coefficient of 2.5 W/m²K,
c) A new one with higher concentration (optical efficiency of 0.74 and a global heat loss coefficient of 2.5 W/m²K).

Figures 1 and 2 show some simulation results by using collector type b), which can be regarded as new state of the art. The diagrams show the yearly total energy consumption and the respective costs for the three chiller variants. In both cases, house as well as hotel, the system using the single/double effect chiller saves the largest amount of energy with only a small increase in cost as compared to the single-effect system. The combination with the pure double effect turns out to be less effective due to the very high driving temperature required that of course handicaps the solar collectors.

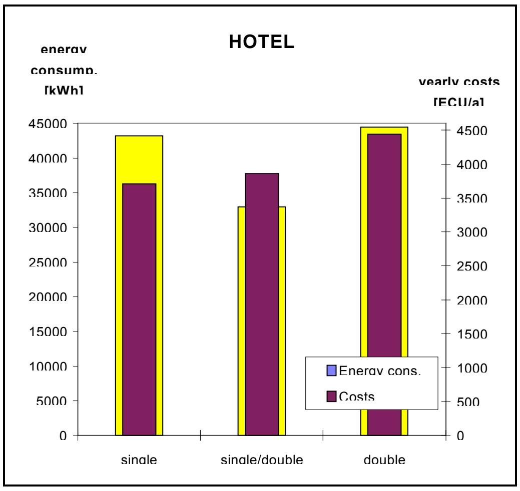

Fig. 1. Yearly costs and total energy consumption for a hotel air conditioning. Iberian average.

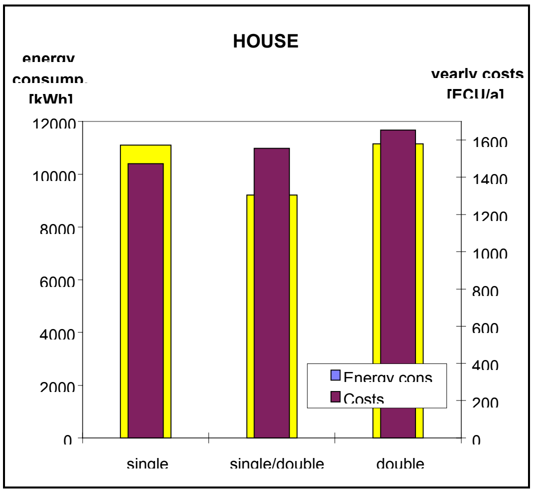

Fig. 2. Yearly costs and total energy consumption for a house air conditioning. Iberian average.

The simulation results again justified the decision to develop a new working cycle: namely the single effect/double effect absorption chiller.

3.3 Machine hardware design

A further fundamental decision concerned the absorption chiller design, i.e. the dimensioning and the arrangement of the components described above.

For that purpose it was necessary first to calculate all internal operation parameters like temperatures, pressures, solution concentrations, mass flow rates and last but not least the overall heat transfer coefficients in the main heat exchangers. By means of these calculations it was possible to determine the design data for an “area minimised” (and therefore for a “cost minimised”) construction of the main components.

Although incorporating requirements of a following market product the design had to allow laboratory operation, i.e. it was designed such that it was possible to vary fluid levels, flow rates and pressure levels in the main components. This made necessary the use of controlled pumps and throttle valves. Moreover the behaviour of the whole unit and the observation of the internal processes like absorption can be recorded by the installation and adaptation of appropriate measurement devices.

Like explained above, in order to achieve a high compactness it has been decided to build the main heat exchanger containments in a rectangular form and not in the usual cylindrical form. The compactness is supposed to play an essential role for the economics of units in the small and medium cooling capacity range, aimed for in this project.

With a calculation program specially written for this purpose the cooling machine process could be completely checked. With the physical data of the used water/LiBr solution and with the chosen absorption cycle (double effect) the complete process parameters like temperatures, pressures, concentrations, mass flows as well as thermal capacities and the coefficient of performance (COP) could be calculated. Using known heat transfer coefficients the resulting overall heat transfer values and the corresponding heat exchanger areas could be determined. Parametric studies have been carried out in order to determine the optimal operation point for the chiller. Specially for the solar operation it was important to find out the optimum between heat exchanger area and COP, since the latter strongly influences the economics of the solar collector field

Finally, after calculation and optimisation of the water-side pressure losses inside the heat exchanger bundles and after calculation of the solution distribution ratio on the same heat exchanger bundles, it was possible to determine the most compact bundle geometry for the main heat exchangers. Out of the results for the thermohydraulic design, corresponding drawings which serve as basis for the chiller manufacturer have been done.

Starting from absorber and middle pressure generator layout, it has been decided to build also evaporator and middle pressure condenser in the “shell & tubes” way.

On the contrary the high pressure condenser has been chosen of the plate heat exchanger type because at the high pressure level (about 1 bar) little pressure losses do not affect the machine performance and therefore allow a cheaper heat exchanger manufacturing.

Also the solution heat exchangers have been designed as plate components. The dimensioning was done using a standard software refined by introducing the physical data of the water/LiBr solution.

Like mentioned above with an own developed program the tube bundle geometry of all “shell & tube” components has been optimised in order to achieve both good overall transfer values and good tube wetting coefficient. Regarding to this also the necessity of a water and solution recirculation in the components absorber, evaporator and middle pressure generator has been considered. The advantage is that the recirculation increases the working fluid mass flow on the heat exchanger bundle and improves their wetting. The drawbacks consist on one hand in the pumping work needed in order to maintain the recirculation mass flow at the desired level and on the other hand in the investment costs for the pumps. Recirculation is not necessary for the condensers, of course, and for the high pressure generator since it has been designed as a pool heat exchanger where the bundle is fully immersed in the solution.

Concerning the absorber first it was considered to renounce a recirculation, but this means that beside the usual design restrictions given by the problematic bundle wetting, also for the solution distribution very hard conditions would have been given due to the very small admission pressure of the solution from the generator into the absorber like already discussed in the section “machine layout - absorber”. Therefore for the first set-up the chiller was designed to be equipped with nozzles and a recirculation circuit in the absorber. Nevertheless it was foreseen to investigate also the cheaper variation discussed above.

Regarding the middle pressure generator normally it is not strictly necessary to provide the machine with a recirculation circuit. Nevertheless this solution allows for a very flexible operation e.g. varying between pool and falling film mode. In particular because this component is the main object of this project high priority was given to the flexibility.

Due to delays in the manufacturing phase has been decided to install spraying nozzles -a well-known technique which does not need any additional development work- in the absorber, middle pressure generator and evaporator heat exchangers instead of the “low admission pressure” system tested at ZAE.

Nozzles with rectangular full cone spraying characteristics have been preferred since in this way it was possible, with only one nozzle row, to wet completely the upper fore-part of the heat exchanger bundles. Moreover nozzles with a cone angle of 120 ° have been chosen compared with the usual 90 ° in order to decrease the distance between the nozzle head and the first bundle row and therefore to achieve a more compact vessel design.

3.4 Control strategy

Goal of the control concept for the developed cooling machine was to meet two different needs which had to be optimised jointly:

The control of the cooling capacity in dependency of the actual load on the customer side and

The minimisation of the energy consumption in dependency of the real solar energy at disposal.

The first aspect is common for all chillers. The cooling demand shows in any case, except for special applications, time variations that have to be quickly and safely met by the cooling machine. This is commonly guaranteed by observing and controlling of the outlet chilled water temperature.

This temperature depends on the cold production which is again influenced by the driving heat coupled into the high pressure generator.

So this part-control concept aims at the control of the heat supply of the high pressure generator when the cold production rises above the actual cold demand and therefore the outlet chilled water temperature drops under the given set value.

In our case the driving heat is produced by an adjustable gas burner which is able to vary its power between 100 % and 30 % of the nominal value in dependency on the outlet chilled water temperature. Once the value of 30 % is crossed the burner is switched off completely till cold production is needed again. It is obvious that the intermittent operation should be possibly avoided, since this causes in the burner so-called intermittent-operation losses as well as mass flow oscillations in the solution circuit of the chiller which influence negatively its efficiency.

The second aspect of the needed control strategy occurred because of the special features of the developed chiller. It is based on fact that two different energy sources, gas and sun, should be used separately or complementarily and that the use of the one or the other lets change the working mode of the chiller from double to single stage. Consequently, always in dependency on the actual cold demand, following different control tasks arise:

As long as sufficient solar energy is available in the water buffer to keep the necessary driving temperature in the middle pressure generator, the unit is driven as a single effect. This means that the high pressure level of the chiller is separated from the rest of the unit and the gas burner is switched off.

If the cold demand can not be completely covered by the solar energy, the high pressure components are connected back to the cycle and the gas burner is switched on as additional driving heat source. Thus the heat supply for the middle pressure generator is taken over in series by solar buffer and high pressure condenser.

In case that the solar buffer does not reach or keep the necessary temperature to drive the middle pressure generator the cooling machine is finally switched over to the double effect modus and the gas burner takes over the full heat production like in a commercial unit, till the solar buffer reaches its set value again.

3.5 Prototype set-up

The prototype built at ATECNIC is shown in figure 3 in a total view of the unit. One can recognise very well the two main heat exchanger pairs evaporator-absorber on the left and middle pressure condenser-generator on the right. Further on the right hand are situated the solar buffer and the heating set-up. Beneath the heat exchangers on the top of the frame there is the place for the solution pumps and water pumps, the plate heat exchangers, the high pressure generator (not yet built in at the time of the picture) and all the measuring sensors.

In order to allow easy access to the components the machine was not built in the most compact way possible.

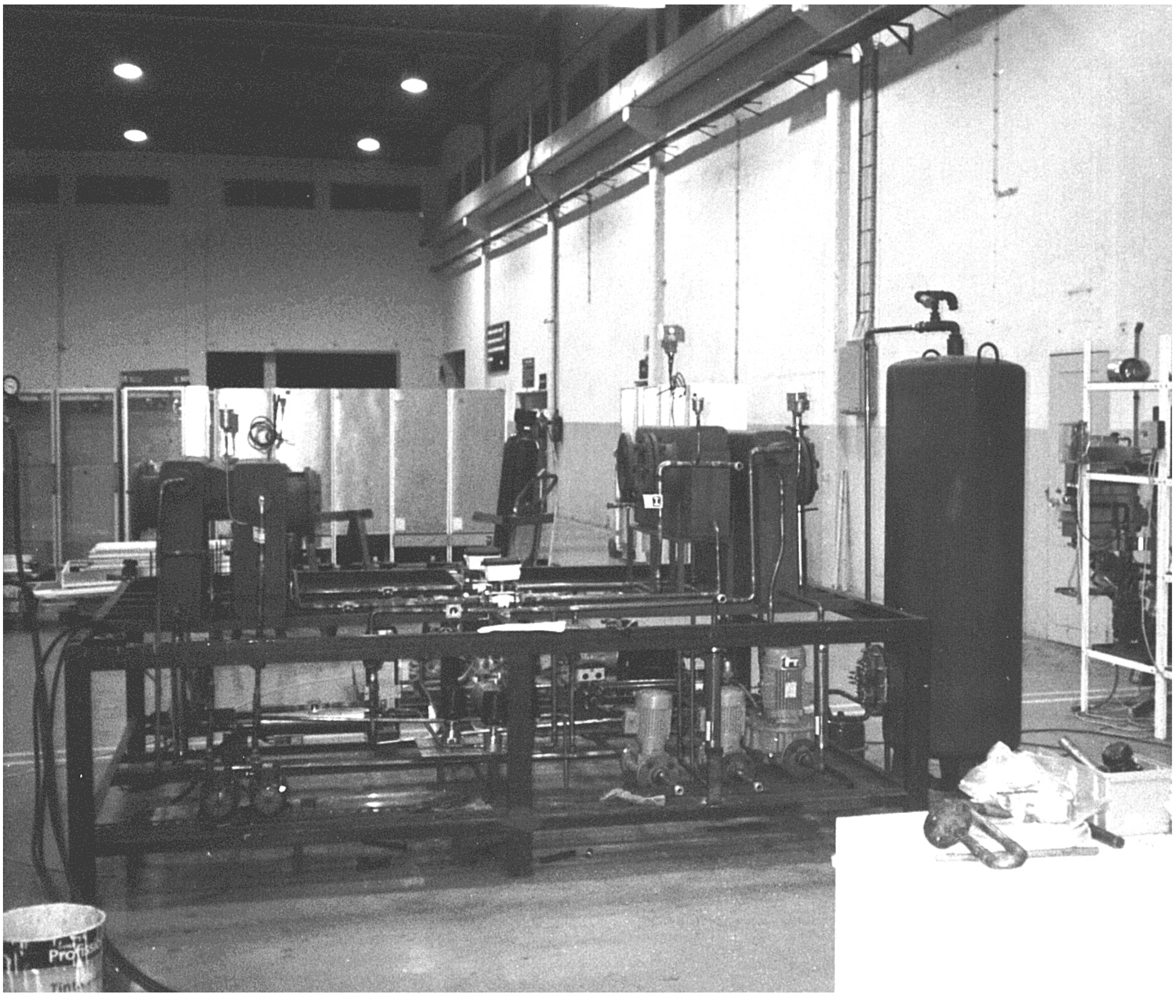

Figure 3. View of the absorption cooling machine set up in this project.

3.5.1 Main components

After the successful completion of the design phase the corresponding drawings have been made and the main heat exchangers manufactured.

Moreover, all peripheral components needed have been designed and purchased or manufactured. The most important are listed and explained below.

<u>Heating set-up</u>

Gas fired systems are rather expensive and not common in use for little capacities. Instead of purchasing a commercial device it was decided to set up a laboratory prototype of a gas fired heat transfer system of small capacity with thermal oil. Since the expenses for commercial heat transfer systems has been one important argument for an own set up, a budget-priced gas fired flow heater for water has been chosen and converted. This flow heater has a heating capacity of roughly 20 kW and is able to modulate the heat capacity from 100% to 30%. Hence it is suited quite well to follow the changes of the heat demand in the high pressure generator. With the modifications made it is able to permit thermal oil temperatures up to some 200°C instead of some 90°C with water. Its main advantage, compared to an usual burner with vessel, is a much lower thermal inertia allowing for a faster response to load changes. Moreover its modular atmospheric burner improves the part load efficiency, again compared to an usual vessel.

Solution throttle valve

The solution throttle valve is situated between high and middle pressure generator and takes care of two tasks: to expand the solution from the high pressure level to the middle one and to hinder the overfill of the high pressure generator. The first task is done by the right dimensioning of the throttle accordingly to the mass flow and the pressure levels. The second one is done by controlling the level in the high pressure generator and the corresponding adjustment of the valve. In most cases motorised valves are used with good results. However, their drawback is the high cost due to the control device needed and the motor. Therefore a hydraulic driven floating valve has been developed and built in. This valve can adjust itself automatically by opening the floating valve inside the throttle when the level in the generator drops. Thus it assures a constant solution level in the generator without any external signal or aid.

Purge device

In a double effect absorption cooling machine working with water as refrigerant there are working pressures of some millibar up to about 1 bar according to the chosen working temperatures. This requires a very good vacuum tightness of the unit and therefore sets strict conditions for the manufacturing. As often experienced, a very small amount of non-condensable gases is enough to disturb the stable operation of an absorption unit. Two main precautions should consequently be taken in order to achieve a regular and successful operation standard:

- Thorough vacuum tests of the single components during the manufacturing phase as well as of the whole unit before starting the operation, and
- Installation of purge devices at the places where the non-condensable gases are expected with high probability.

The first point is necessary since it is practical out of question to achieve a faultless manufacturing of the whole unit, and all the more in a laboratory set-up which is equipped, due to the aimed scientific purposes, with a multitude of additional accessories which very often can not be welded in the unit but must be screwed. More about the vacuum test procedures adopted is explained in chapter 3.5.3.

Concerning the purge devices as already explained the non-condensable gases are accumulated in condensers and absorbers. In the cooling machine developed there is one condenser and one absorber of the type ”shell and tube”, i.e. with the heat exchanger tube bundle inside of a vessel, and one condenser of the type ”plate heat exchanger”.

For the ”shell and tube” components it has been decided to purge the complete vessel inventory between the measuring sessions by means of a flange attached at the vessel wall.

On the other hand a special purge device has been built for the high pressure condenser (plate heat exchanger) which allows to catch the non-condensable gases flowing out of the heat exchanger (they are pushed out by the vapour entering) and to prevent their further transport to the middle pressure condenser.

Nozzles

In accordance with the calculated data of mass flow and recirculation ratios it was decided to install spraying nozzles in following heat exchangers:

Evaporator

Absorber

Middle pressure generator

In the remaining heat exchangers, i.e. middle and high pressure condenser and high pressure generator, there was no need for such distribution since the former two do not need distribution devices at all and the latter one was built as a pool generator in which the solution entering is not distributed on the heat exchanger bundle (immersed in the solution) but mixed in the existing pool.

The nozzles are able to build a pyramidal full cone, i.e. the liquid sprayed covers a quadratic area under the nozzle itself. Thus a complete wetting of the upper heat exchanger rows is ensured.

Pumps

In contrast to the theoretical needs of a double effect chiller with only one solution pump, in the prototype three pumps have been foreseen in order to ensure a faultless operation. This was necessary due to the use of nozzles for the distribution of water or working solution in evaporator, absorber and middle pressure generator, since nozzles show a too high pressure drop which can not be compensated by a convenient static arrangement of the different vessels. This has already been discussed in chapter 3.1.1 and an alternative solution has been presented.

For the use in the chiller a suitable pump has to meet two main characteristics: vacuum tightness and possibility of flow control. Accordingly to these requirements magnetic driven pumps show an optimal set-up.

A last consideration related to the control concept required a further peculiarity for the absorber solution pump, namely the capability to stand cavitation. With such a pump it would be possible to keep the corresponding vessel continuously empty and thus to save the otherwise necessary control device. This is already known from chiller manufacturers

Additives recovery device

As mentioned above additives are used to enhance the heat and mass transfer in particular in the absorber. To assure that this additives are continuously available in the absorber special measures have to be taken. Thus in the middle pressure generator a device for the sampling of some liquid from the top of the LiBr-solution pool has been foreseen and built in. The aim is to recover additives and to lead them into the main flow coming from the bottom of the generator and going to the pump and further to the absorber distribution.

3.5.2 Measurement set-up

For the data acquisition program the QuickBasic v 4.5 language has been used, due to the features of the PC for the data logging.

In its actual version the program registers all the data continuously with the time span fixed by the user, and among the 50 different signals in the machine, the program allows the simultaneous graphical display of 16 signals in two groups of 8 signals.

In order to see the data while recording two modes have been devised: graphical and numerical.

In the meantime all the measuring sensors have been connected to the data acquisition program which is so ready for operation.

3.5.3 Vacuum tests

As already explained in chapter 3.5.1 ”purge devices”, the vacuum tests were of essential importance in order to guarantee the continuous operation of the unit, which is necessary to obtain useful measuring periods. Therefore a methodology was devised to guarantee a good vacuum tightness of the prototype. This can be seen in table 1.

(1).- Test of each individual component.	(2).- Test of each pair of vessels after welding one component to each other.	(3).- Final test of the whole machine.
Absorber (A)	A-E	Test of the piping and Connection of the Measurement instruments
Evaporator (E)		
Condenser 1 (C1)	C1-G1	
Generator 1 (G1)		
Condenser 2 (C2)	C2-G2	
Generator 2 (G2)		

Table 1. Methodology for the vacuum tests.

The general procedure to ensure the vacuum tightness at each stage was to get first a sustainable degree of vacuum and afterwards to use a He-leak detector equipment. In all cases it was very difficult to reach a satisfactory vacuum pressure, given poor manufacturing practices, as for instance, a lot of dirt inside the vessels. Different procedures have been followed, like injection of inert gas to drag the dirt by pumping. Furthermore, the vessels have been heated to facilitate dirt removal.

A problem of tightness arose by coupling the solution pumps and the pipes. These pumps were made of polypropylene and Teflon as already explained in chapter 3.5.1 ”pumps”, and so no direct soldering or welding to the piping was possible. In order to find a suitable sealant for the connections between plastic solution pumps and piping several leading manufacturers have been contacted. Specially for the Teflon pump it was difficult to find an adequate product since Teflon shows an extreme inadhesivity which makes problematic the connection which almost all commercial glues and sealants.

3.6 Extended tests

This task could not be carried out due to a delay in the manufacturing. There is a chance that the measurements will be done after the end of the project. The results will then be published elsewhere.

4 Conclusions

In the actual project a solar-assisted gas-driven absorption cooling machine has been designed and built up.

First an extended simulation phase allowed to compare in a dynamic procedure the real cooling demand of an ordinary house and hotel for several Iberian cities. With these data as input the use of three solar-assisted cooling systems, namely a single-effect, a double-effect and a mixed-mode single/double-effect machine have been simulated in connection with three solar collectors of the type Compound Parabolic Concentrating: namely a commercial model, the same one with better insulation and as last a model with higher concentration. The resulting energetic and economic features could be compared. As shown in chapter 3.2, under the given technical conditions for the cooling machine and the solar collectors it turned out that the mixed-mode single/double-effect machine could achieve the maximal energy savings with only a small increase of the yearly costs as compared to standard single effect machines.

On the base of these results a prototype has been designed and manufactured. Beside the thermohydraulic optimisation of the unit for the solar operation, the attention has been turned to a ”commercial-friendly” design of the unit. In this sense the main heat exchangers have been built with a rectangular shape which allows to achieve very compact total dimensions of the unit.

Two further aspects of great importance have been investigated. These are the improvement of a solution distribution device operating with very low admission pressures and the rebuilding of a commercial water heater as high temperature driving set-up for the prototype. Both experiments delivered positive results as documented in the chapters 3.1.1 and 3.5.1.

An appropriate control strategy has been worked out with the target to maximise the share of solar heat coupled in the cooling machine. The followed philosophy was to be enabled to meet the nominal cooling demand in solar driven mode by full solar availability and to glide continuously towards the gas-driven (high efficient) mode if necessary.

Last but not least the completion of the data acquisition system and the implementation of vacuum tests for the main components of the unit have been done.

A detailed measurement of the performance of the prototype chiller could not be achieved due to the time delay during the manufacturing of the machine.