LiBr - Absorption Chiller
for
Building Air Conditioning
with efficient and flexible operation

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CONTRACT JOE3 CT 97 0059

Publishable Final Report
30.06.2000

Research funded in part by
THE EUROPEAN COMMISSION
in the framework of the
Non Nuclear Energy Programme
Joule III

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1 Synopsis Joule III project

Project Title:

Lithiumbromide Absorption Chiller for Building Air Conditioning with efficient and flexible operation

Objective:

The main objective of the project is the development of a

direct fired,

high performance

energy efficient absorption chiller which is adapted to the needs of the european air conditioning market.

Since the cooling towers and the related water consumption is one of the factors with the most important economical and environmental impact, a new Lithiumbromide absorption chiller process for air conditioning in buildings had to be developed with lowest consumption of water for the cooling tower and low energy consumption.

The new components shall be tested in laboratory on a semi-industrial size (300kW) which can be scaled up with limited risk to realize larger sized industrial units.

Technical Approach:

The development of a new direct fired absorption chiller was mainly based on the newly designed direct fired regenerator. The accompanying improvement of the refrigeration cycle led to an unmatched overall primary energy efficiency which can be up to 20% above commercially available units.

Expected achievements and exploitation

The newly designed double-effect cycle with the regenerator will improve the application of gas for building air conditioning due to lower operational cost than conventional equipment.

Even though water prices are still unexpectedly low in the hotter and arid regions of Europe, investors for future plants are already expecting higher water prices for the times when public subsidies will no more be legal in a privatized water market. This will in which in turn favour cooling technologies with low or negligible water consumption.

The new technology absorption chiller with its highly efficient direct fired generator and the improved refrigeration process can not only be applied to direct fired units but also in cogeneration plants, where a recent marketing survey (Entropie) has shown a significant improvement of the overall energy balance of a tri-generation plant.

Since cooling loads are increasing more and more, while electric consumption in buildings stagnates, a typical tri-generation plant has to produce more and more cooling capacity with a decreasing quantitiy of waste heat from the gas or diesel engines.

The newly developed absorption process and its components can increase the cooling capacity for a given gas engine by more than 20% due to more efficient and thermodynamically optimized use of the waste heat.

2 Abstract

In this final report the results of the research work for the development of an advanced direct fired and partially dry cooled absorption chiller is presented.
The application of absorption chillers in the building sector is currently dominated by installations in cogeneration plants, where the motive energy is available at low marginal cost and the waste heat is transformed into cooling with single-effect absorption chillers.
Unfortunately the implemetation of direct fired chillers is limited by several economical and technical obstacles. Even though the ranking of the economical or environmental obstacles to the installation of direct fired absorption chillers may vary in the different countries throughout Europe, they are more or less the same:

- too high water consumption of the cooling tower
- too high fuel consumption due to too low coefficient of performance
- too low primary energy efficiency

Tasks

In this research project the different group's tasks were therefore split up in order to solve the relevant problems as efficient as possible and partly in parallel:

- identification of climatic and economic conditions for various sites in Europe
- identification of a new, water saving cooling tower concept
- development of a refrigeration cycle with high COP and high cooling water temperature
- design and construction of a regenerator with high primary energy efficiency
- test of the equipment on an industrial size test unit

Cooling Tower and Water Saving

The climatic study was the basis for the work related to the improvement of cooling tower concepts.
As a result of the cooler tower study it can be stated clearly, that considerable water savings can be achieved by using proper combinations of cooling towers readily available on the market without using expensive hybid cooling towers. The relative size of wet and dry part of cooling tower as well as the mode of operation can be determined with a model for various sites and buildings.

Refrigeration Cycle improvement

An important development is related also to the refrigeration process itself, where the integration of supplementary internal heat exchanger can improve the efficiency of the cycle by more than 10% in addition to an improvement of the efficiency of energy recovery from the flue gas which in total allow to design direct fired chillers with a COP 1,25-1,30 instead 1,0 to 1,1 of units already on the market.

New Direct Fired Generator

The new regenerator and the newly introduced economizer has been tested thoroughly on a semi-industrial scale (350 kW). The generator has precisely met the predicted performance and has shown very good and stable operational results which allow industrial implementation with only limited risk for scale-up in larger installations.

3 Objectives of the research project

The main objective of the project is the improvement and/or development of the key components required for a direct-fired or flue gas fired absorption chiller which can be operated with a dry or hybrid cooling tower.
The chiller shall use water and lithium bromide as working medium and hence be be free of (H)CFC.
The refrigeration market is dominated by electric driven compression chillers using (H)CFC or ammonia (mainly industrial). Absorption chillers represent only a niche market mainly concentrated on large cogeneration plants where the motive heat is available at low marginal cost.
Direct fired absorption chillers are available on the market but their market segment is restricted and only a small number of chillers are installed throughout Europe. Japanese and US manufacturers undertook a great effort in the past to push direct fired absorption chillers - sometimes in close collaboration with gas and electricity companies - since direct fired gas absorption can reduce the electric peak consumption at noon due the compressor air conditioners and increase the summer untilization of natural gas.
No european manufacturer was acitve in this field at the date of beginning of this project.
The partners therefore decided to launch the development of the technology required for a market introduction of

- direct fired absorption chillers

or flue gas fired absorption chillers

- which consumes less water in its cooling tower
- which can be integrated into european building with less problems

A critical review of the state-of-the art has revealed the need of new technology in the following fields:

	state-of-the-art	objective
Thermal efficiency	1,0 - 1,1 full load	> 1,25 at full load
	<0,8 at 50% load	> 1,00 at 50% load
Water consumption	3,2 m³/ MWh cooling	< 1,6 m³/ MWh
CO2 emission	proprtional to increase in thermal efficiency	
	reduction by 25% - 50% in part load	
Integrability		
cogeneration	not directly	adaptable
cooling water	no variation allowed	adaptable
european standards		full conformity

Necessity of particular efforts has been identified in

- analysis and optimization refrigeration process
- new flexible boiler/ generator design
- cooling water system design

4 Scientific and technical description of the project

In this research project the different group's tasks were split up in order to solve the relevant problems. The results of each group's work are presented in the following chapters :

Chapter 4.1 Climatic survey
- identification of climatic and economic conditions for various sites in Europe (mainly ZAE Bayern and Gas Natural)

Chapter 4.2 Design of Cooling tower system
- identification of a new, water saving cooling tower concept (mainly Gas Natural and ZAE Bayern)

Chapter 4.3 Optimization of heat recovery and performance of the chiller
- development of a refrigeration cycle with high COP and high cooling water temperature (mainly ENTROPIE)
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Chapter 4.4 LIBRAC absorption Unit Design (1 MW plant)
- complete thermal and hydraulic desing of a 1 MW chiller using the newly developed direct fired regenerator (mainly ENTROPIE)

Chapter 4.5 Design of Flue Gas heated LiBr generator
- general layout and design of a regenerator with high primary energy efficiency with a new burner (mainly British Gas)
- detail design and construction of regenerator (mainly British Gas)

Chapter 4.6 Test Rig for 350 kW demonstrator
- construction of a test rig (mainly ZAE Bayern)
- test of the equipment on the test rig (mainly ZAE Bayern)

Chapter 4.7-4.10 Test campaign with Generator Test Rig
(mainly ZAE Bayern British)
and Evaluation (all partners)

Generally all partners collaborated and contributed to each of the different tasks. The main work has been performed by the partner cited at each chapter.

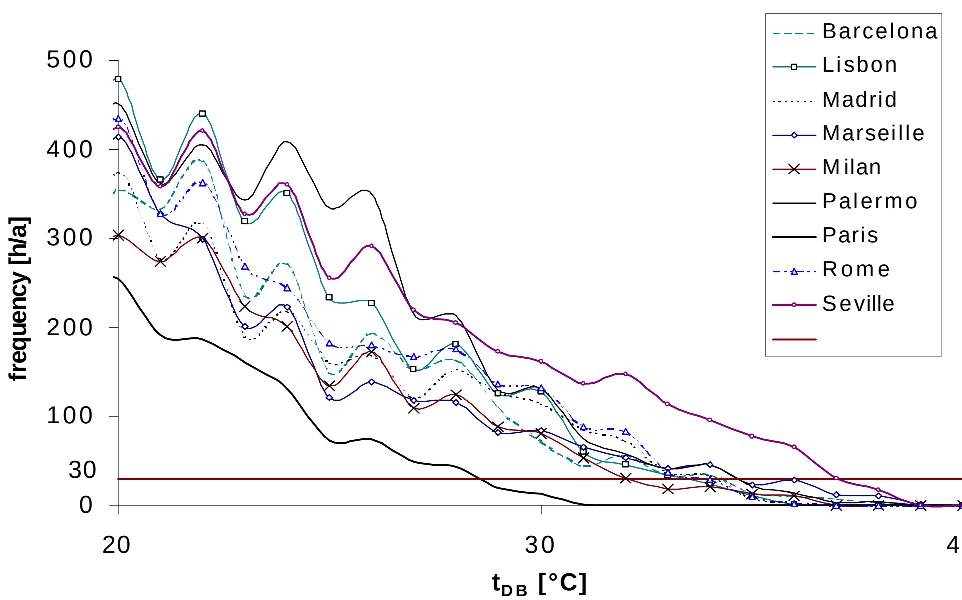
4.1 Climatic data survey

As data basis for the climate of different European regions test reference year data (TRY) have been used. The data sources available in public include mean hourly values for the dry bulb temperature, the radiation (global, diffuse, direct normal), sunshine duration, relative humidity and wind speed.

Key data for the thermal study of the whole system, including the cooling demand and the performance of the cooling tower, are dry bulb temperature t_{Dry} and wet bulb temperature t_{Wet} or relative humidity, respectively. These values describe the state of the intake air of the building air conditioning system and of the cooling tower.

A screening of these climatic data for fifteen representative locations spread over Europe has been carried out: Barcelona, Berlin, Bordeaux, Frankfurt, Hamburg, Lisbon, London, Madrid, Marseille, Milan, Munich, Palermo, Paris, Rome and Sevilla. Some cities, especially those on the coast, show a very narrow and pointed temperature distribution, i.e. the climate is very homogeneous without long cold or hot periods. Such cities are for example Barcelona, Paris, Lisbon or London. However, most of the remaining cities show a broader and flatter temperature distribution with a main field, in which the temperatures vary in a small frequency range. This can be seen very clearly for Berlin, Hamburg, Rome and Madrid.

In Figure below the warm sections of the temperature distributions for the warmest cities, with Paris for comparison, are plotted and compared. Sevilla is clearly warmer than any of the other cities in the range between 28 °C and 38 °C, whereas Palermo is slight warmer in the sector till 28 °C. It is also interesting to see that the extreme warm temperatures are very similar for all cities with values around 36 °C – 39 °C.



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the warm peaks as a function of their cumulated annual frequency, again for the example of the city of Paris. The abscissa shows the sum of the yearly hours but only daytime values -between 7 and 19 o'clock are taken into consideration though. On the ordinate axis the values for dry and wet bulb temperature are plotted. The upper curve represents the dry bulb temperature, whereas the curves below show the percentiles of maximum wet bulb temperature correlated to the respective dry bulb temperature (upper curve).

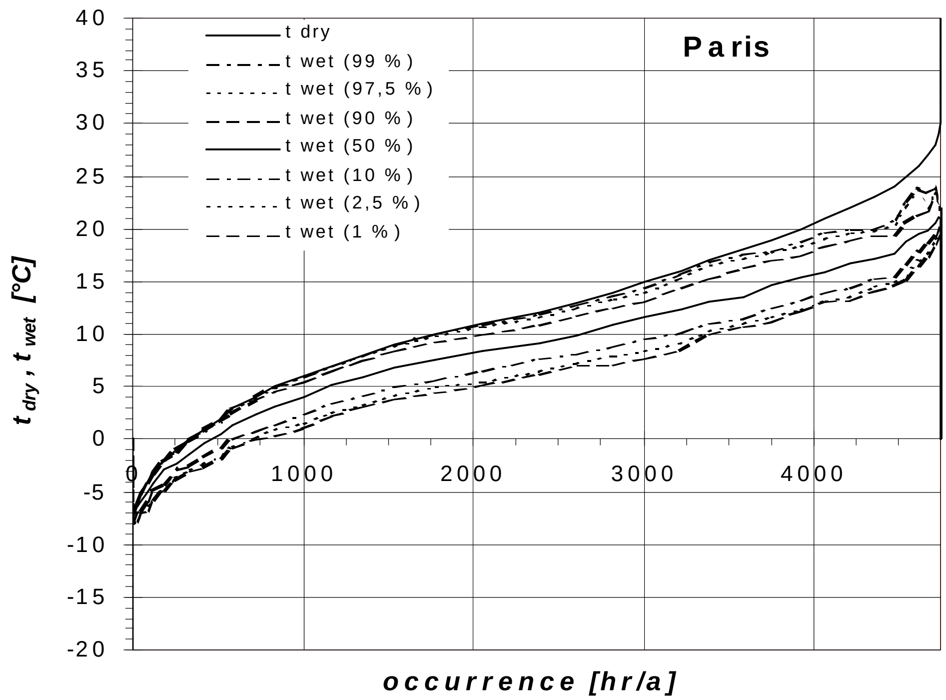


Fig. 2 Yearly correllated probabilities of dry and wet bulb temperatures for Paris. The dashed lines represent the percentiles (in %) of max. wet bulb temperatures related to dry bulb temperature.

The maximum dry and wet bulb Temperatures of the cities investigated are summarised in table 1. To verify the chosen data basis (Meteonorm) the design temperatures obtained from the climatic data analysis have been compared to the temperature limits given by ASHRAE. As both data sources are of statistical nature, they can not perfectly reproduce the naturally occurring weather conditions; in paritcular no effect of warming by global effect, nor localized effects in city centres are taken into consideration. This is the reason why generally 2-3 K are added to these data when design temperatures have to be fixed.

On the other hand they turn out to be very useful for a comparative modelization buildings or cooling machines design among different climatic regions.

It must be noted, that the data extracted from Meteonorm data, generally indicate higher temperatures, in particular for the wet bulb temperatures which are of much higher importance for air conditioning, not only because of higher cooling water temperatures but also higher cooling demand due to dehumification must be expected.

Table 1 Design temperature for absorption chillers and the cooling towers for different locations.

	ASHRAE		METEONORM	
location	t DB max	t WB mn	t DB max	t WB
	[°C]	[°C]	[°C]	[°C
Barcelona	[illegible]	22.4	24.1	[illegible]

4.2 Design of a Cooling Tower System (Paris)

A complete cooling tower system has been designed for a specific location within the task 2. The following concepts have been studied in order to illustrate the capabilities and the limitaitons of the systems:
A building energy load model has been used to run simulations on the cooling load vs. ambient conditions (temperature and humidity).
In parallel a number of different cooling tower configurations have been studied:

- **Open Wet Cooling Tower**
  The classical and cheapest cooling tower design.
  Cooling effect by evaporation of water only.
  - + lowest first cost
  - + low fan power (lowest air flow)
  - + cooling water tempature 3-5 K above wet bulb
  - + standard equipment
  - - high water consumption
  - - suspect to sanitary problems (legionellae)
  - - plume (in particular on colder days)
- **Hybrid Cooling Tower**
  This type of cooling towers is commercially available for larger unit sizes.
  Capacity of dry section is generally limited to approx. 20% ot total capacity.
  - + almost no plume
  - + reduced water consumption
  - - quite expensive
  - - higher fan power
  - - noise problems due to higher air flow
  - - suspect to sanitary problems (legionellae)
- **Dry Cooling Tower**
  Frequently used equipment, available in all relevant sizes.
  Predominant in smaller sizes air conditioners with compressors
  - + no plume
  - + no water consumption
  - + no sanitary problems
  - - high ccoling water temperature 5 K above dry bulb
  - - very expensive
  - - high electrical consumption
  - - noise problems due to high air flow

- **Adiabatic cooling tower**
  This is a relatively recent development to overcome the problems of high cooling water temperatures of dry cooling towers. The cooling tower is itself operated as dry cooling tower. During the hottest days water is sprayed into the air upfront the cooling tower an hence a lower air temperature at the inlet in obtained.
  - + no plume
  - + no sanitary problems
  - + low cooling water temperature 5K above wet bulb

+ low water consumption (partial dry cooling)
- expensive
- water treatment for sprayed water required
- no standard equipment

➢ **Combination of an open wet cooling tower and a dry air cooler**

+ standard equipment
+ considerably reduced water consumption
+ free cooling in winter
+ very flexible design
(fraction of dry part can be adjusted deliberately)
+ practically no plume
(no wet operation on cold days)
- more space required

Physical constraint of chiller which cannot be overcome
Due to the physical properties of the lithium-bromide which is commonly used in absorption chillers, the temperature of cooling is restricted essentially by two conditions:

- The cooling water temperature at the chiller inlet (absorber) must not exceed 35°C
- the outlet temperature also has an influence on the required temperature of the generator heat and hence a prctical limit is about 50°C for double-effect chillers

Two alternatives for the cooling water circuit have studied:

- low flow , an hence high outlet temperature for example 30 / 45°C instead of the commonly used 30/35 °C systems with high flow
- split system with independent circuits for absorber and condenser

Due the high complexity of split systems this alternative has been abandoned even though theoretically interesting.

In order to evaluate **manufacturing constraints** of such hybrid systems, design parameters for LiBRAC chiller are set at:

- Chilling capacity: 1000 kW
- Cooling tower load 1675 kW at full load
- Cooling water temperature at inlet: 30ºC (full load)...36ºC (load below 30%)
- Cooling water temperature at outlet: 40ºC (full load)...47ºC (30% load)
- Maximum cooling water flow: 144 m^3/h
- Minimum cooling water flow: 43.2 m^3/h
- Maximum COP (PCI): 1.37 at full load
- Minimum COP (PCI): 0.79 at 10% load
- Cooling water needs: HDR = (COP + 1) / COP
- Climatic conditions in Paris are:
  Design dry bulb temperature: 32ºC
  Design wet bulb temperature: 22ºC
- Hybrid cooling system design parameters for wet section are:
  Cooling capacity: 100% of full load demand
  Approach temperature: 8ºC at full load
  Cooling range: 10ºC at full load
  Cooling tower configuration: open cycle, axial fans
- Hybrid cooling system design parameters for dry section are:

Cooling capacity: 30% of full load demand
Approach temperature: minimum 5ºC at full load
Cooling range: 10ºC at full load
Start air temperature: 28ºC

4.2.1 Manufacturer Specification for Wet Part (Paris)

Design parameters for the wet part of the hybrid system are exactly the same as if it were a standard design of cooling tower. Taking into account range of products of one of the most important cooling tower manufacturers in Spain (TEVA-DECSA), **main technical specifications** of selected equipment are:

- Cooling capacity: 1675 kW
- Cooling water temperatures: 30ºC inlet, 40ºC outlet (at full load)
- Air temperature: 22ºC (wet bulb)
- Selected model: TVA-111
- Air flow: 20.3 m^3/s
- Number of fans: 1
- Electric consumption: 1 x 7.5 kW
- Weight: 1600 kg transport; 4000 kg in service
- Size: 2470 x 2470 x 3940 mm
- Noise level: 67 dBA at 15 m
- Cost (standard supply): 12000 EUR

4.2.2 Manufacturer Specification for Dry Part (Paris)

Similarly, design parameters for the dry part of the hybrid system are considered the same as if it were a standard design of dry heat exchanger. Taking into account range of products of one of the most important air-water heat exchanger manufacturers in Spain (GEA Ibérica – group GEA), **main technical specifications** of selected equipment are:

- Cooling capacity: 510 kW
- Cooling water temperatures: 33ºC inlet, 43ºC outlet (at full load)
- Air temperature: 28ºC (dry bulb);
  for higher temperatures, only wet part of the system will be used
- Selected model: TGR-10-06-42-C
- Air flow: 61.1 m^3/s
- Number of fans: 10
- Electric consumption: 10 x 3.4 kW
- Weight: 1600 kg in service
- Heat exchanger area: 1375 m^2
- Size: 6600 x 2185 x 1350 mm
- Noise level: 68 dBA at 10 m
- Cost (standard supply): 16500 euro

4.2.3 Comparison of simulation model with manufacturer's data

The following table shows a comparison between real and simulated results (using LiBrac-03 model for Paris):

WET SECTION	REAL	SIMULATION
Electricity consumption (kW)	7.5	8.0
Cost (euro)	12000	12119
DRY SECTION	REAL	SIMULATION
Electricity consumption (kW)	34.0	34.6
Cost (euro)	16500	14026

Table 2: Comparison of simulation model with manufacturer's data

Model shows similar electricity consumption to real data. It is important to notice that electricity consumption in dry systems is much higher than in wet systems (four times more consumption but only one third of capacity).
It will be then important to consider that reduction in water consumption will be linked to an increase in electricity demand. Relative costs for water and electricity will finally define economic feasibility of using hybrid systems.

4.2.4 Hybrid Cooling Towers already in the Market

Apart from hybrid cooling systems (standard wet cooling tower plus standard dry section), market also offers hybrid cooling towers with the same aim of reducing water consumption. In this report, two manufacturer designs are being evaluated: GEA-Soramat and Baltimore-HXI as examples.
Main differences between these designs and expected values are linked to the fact that hybrid cooling towers are always close-loop equipment whist hybrid cooling systems can also be open-loop equipment (in wet part). This difference in concept is also reflected in the price of equipment (close-loop are always more expensive due to the fact that energy transfer efficiency is lower). On the other side, close-loop design reduces fouling risk and other maintenance problems in chiller.
Main technical specifications for GEA-Soramat design:

- Cooling capacity: 1675 kW
- Cooling water temperatures: 30ºC inlet, 40ºC outlet (at full load)
- Air temperature: 32ºC (dry bulb)
- Air temperature: 22ºC (wet bulb)
- Selected model: GVM-75S32 (see attached brochure)
- Air flow: 49.7 m^3/s
- Number of fans: 2
- Electric consumption: 2 x 30 kW
- Weight: 6650 kg transport; 10450 kg in service
- Size: 8100 x 2400 x 3850 mm
- Cost (standard supply): 52300 euro

Load factor	100%	50%	30%	20%	10%
Load (kW)	1675	900	550	400	170
Water T (inlet)	40ºC	45ºC	46ºC	44ºC	40ºC
Water T (outlet)	30ºC	35ºC	36	36ºC	36ºC
Air T (needed)		22.4ºC	29.9ºC	31.9ºC	34.5ºC
Air T (statistics)		24.5ºC	21.5ºC	20.0ºC	18.5ºC
Running (real)	Wet	Wet-dry	Dry	Dry	Dry
Running (expected)	Wet	Dry	Dry	Dry	Dry

Table 3: Part-load operation of GEA -SORAMAT hybrid cooling tower

This kind of system is based on an standard closed-loop wet cooling tower and includes an additional bundle of finned tubes in order to enlarge heat transfer area. This heat exchanger is located just under water spraying system. Air is forced through a first conventional heat exchanger and, afterwards through the finned tubes.

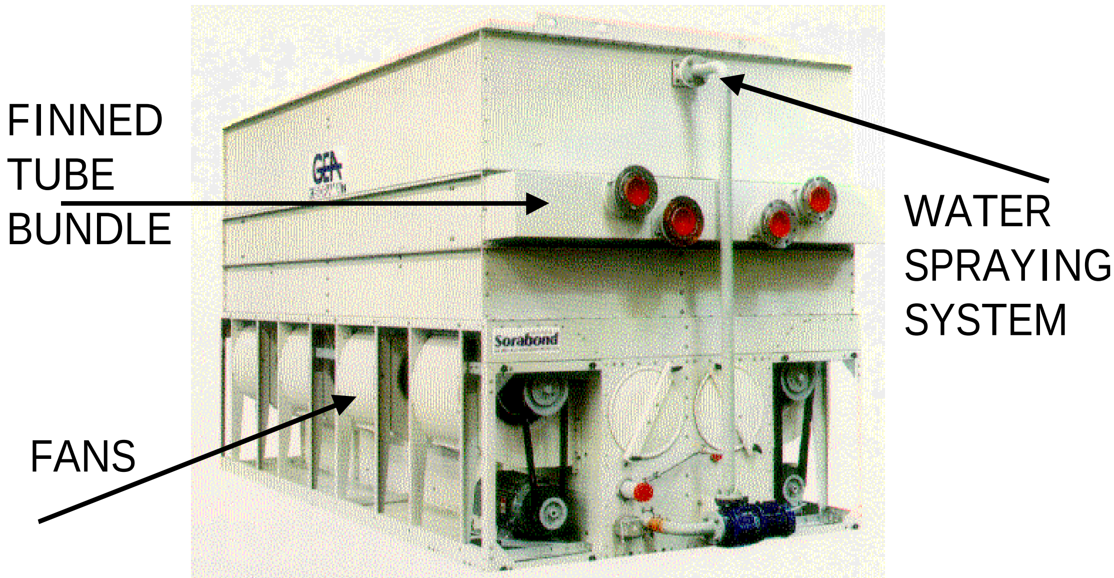


Fig. 3 Hybrid Cooling Tower (GEA SORAMAT)

It can be seen that this equipment does not match expected design. In particular it is pointed out that dry part of this tower will be able to cover full dissipation needs only below 40% of the maximum load of the chiller (instead of 70% in specification). This is due to the fact that dry section capacity of this tower is less than 30% of full load capacity as specified.
It is also possible to see that price of this hybrid cooling tower is twice as much as pre-defined hybrid cooling system.

- **Main technical specifications for Baltimore-HXI design**:
- Cooling capacity: 1675 kW
- Cooling water temperatures: 30ºC inlet, 40ºC outlet (at full load)
- Air temperature: 32ºC (dry bulb)
- Air temperature: 22ºC (wet bulb)
- Selected model: 2 x HXI-440 (see attached brochure)
- Air flow: N.A.
- Number of fans: 2 x 2
- Electric consumption: 2 x (15 + 2.2) kW
- Weight: 2 x (4520 kg transport;7010 kg in service)
- Size (unit): 3690 x 2385 x 4855 mm
- Cost (standard supply): 69100 euro

Load factor	100%	50%	30%	20%	10%
Load (kW)	1675	900	550	400	170

Water T (inlet)	40ºC	45ºC	46ºC	44ºC	40ºC
Water T (outlet)	30ºC	35ºC	36	36ºC	36ºC
Capacity (avail. dry)		548 kW	567 kW	580 kW	557 kW
Running (real)	Wet	Wet-dry	Dry	Dry	Dry
Running (expected)	Wet	Dry	Dry	Dry	Dry

Table 4: Part-load operating data of Baltimotre Hybrid Cooling Tower

This kind of system is based on an standard closed-loop wet cooling tower and includes an additional bundle of finned tubes in order to enlarge heat transfer area. This heat exchanger is located just over water spraying system. Fans are located between conventional heat exchanger and finned tubes heat exchanger.

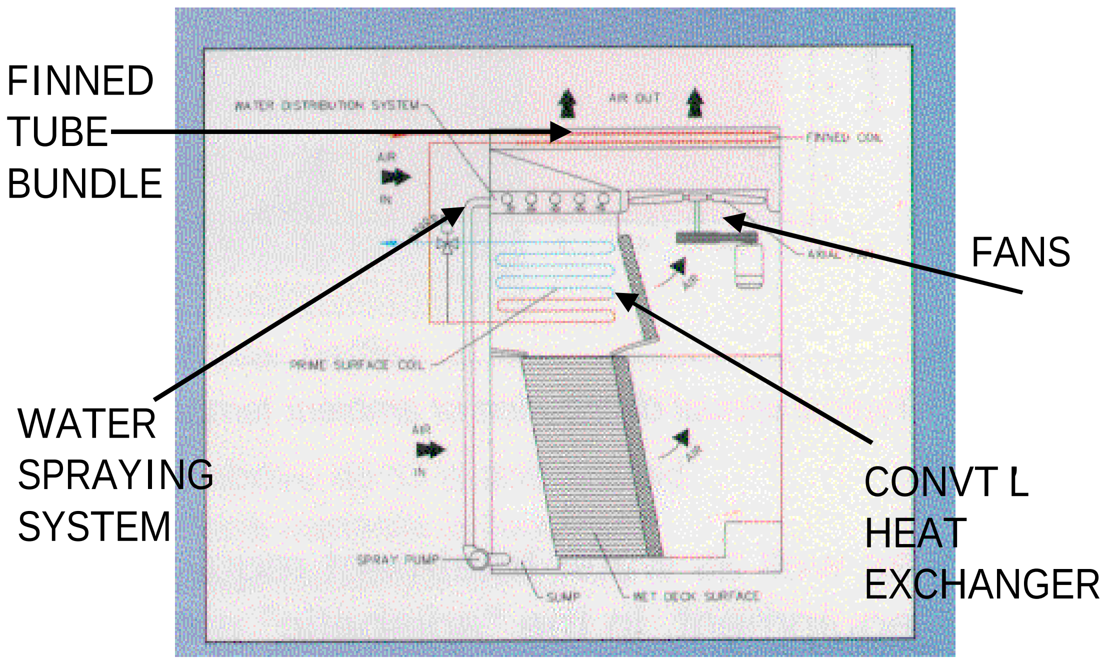


Fig. 4: Cross section of Baltimore Hybrid Cooling Tower

It can also be seen that this equipment does not match expected design. In particular it must be pointed out that dry part of this tower will be able to cover full dissipation needs only below 50% of the maximum load of the chiller (instead of 70% in specification). This is due to the fact that dry section capacity of this tower is less than 30% of full load capacity as specified.
The price of this hybrid cooling tower is two and a half time the price as pre-defined hybrid cooling system.

4.2.5 Overall Comparison of Hybrid Systems

Following table summarises previously presented results:

	HYBRID SYSTEM	HYBRID TOWER	HYBRID TOWER
MANUFACT. – MODEL	TEVA TVA-111 / GEA TGR-10-06-42-C	GEA GVM-75S32	BALTIMORE 2 x HXI-440
PRICE (euro)	28.500	52.300	69.100
OPERATING WEIGHT (kg)	5.600	10.450	14.020
HEIGHT (mm)	3940 (wet), 1350 (dry)	3850	4855
WIDHT (mm)	2185 (wet), 2470 (dry)	2400	2385
LENGTH (mm)	6600 (wet), 2470 (dry)	8100	2 x 3690
AIR FLOW (m^3/h)	20.3 – 61.1	49.7	N.A.
E-CAPACITY (kW)	7.5 – 34	60	34.3

Table 5: Comparison of cooling tower data

The only possibility to match specifications of LiBrac project is a combination of both a wet cooling tower plus an air-to-water heat exchanger (dry cooling tower). Moreover, this combination is much more economical and even less complex of implementation (size and weight).
Baltimore hybrid cooling tower is a more similar design compared to Gea hybrid cooling tower but it is also much more expensive and under no circumstances (price of water, price of electricity) feasible.

4.2.6 Specifications for the new Design

Previous conclusion brings to a situation in which hybrid cooling towers should not be considered as a real possibility against hybrid systems (wet+dry cooling tower) or standard systems (only wet cooling tower with open-loop).
Quite often, standard systems showed better economic performance than hybrid systems (where water to electricity price ratio is low enough or where total running hours of the building being air-conditioned is limited (offices, for instance) or where climate in summer conditions are severe (high temperature and/or high degree of humidity).
The only approach of hybrid systems/cooling towers to those markets must be considered in two stages:

- First, increase the presence of standard wet cooling towers but, in closed-loop instead of open-loop. The reason for this change is to reduce interferences (and problems) between tower and chiller through cooling water. Closed-loop towers work on independent water circuits for evaporation and cooling of chiller.
- Second, to promote hybrid systems in comparison with standard closed-loop cooling towers. Differences in terms of investment will then be sharply reduced and economics will improve.
-

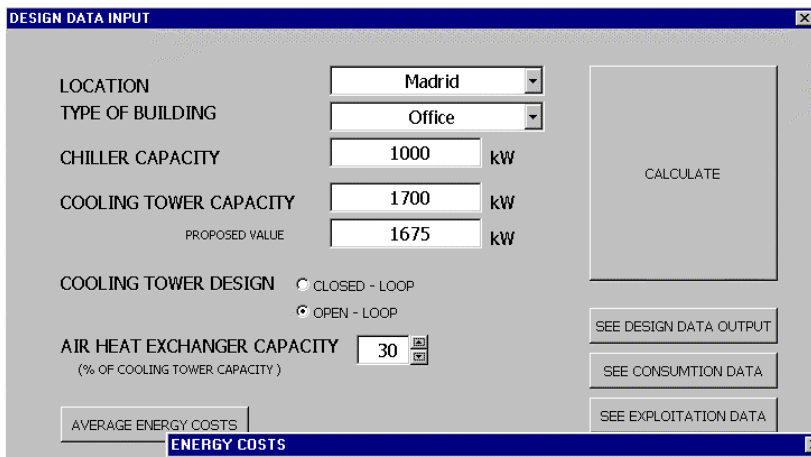
Just as a matter of comparison, standard wet open-loop cooling tower (model TVA-111) which costs 12000 euro, would then be changed to a model which costs between 41500 euro (Gea TCF-DBF-55-4) and 38000 euro (Baltimore FHV643-O). Hybrid towers can therefore compite against standard designs.

4.2.7 Software Tools (LiBrac-03)

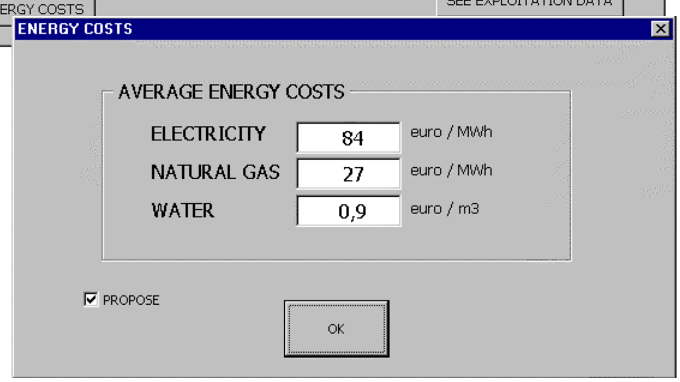
As a result of previously developed activities, a simple software tool has been designed to help in analyzing different possible configurations of cooling systems for a given building in a given climate and economic framework. It integrates modelization of chiller, cooling towers, climates and buildings.
The programme consists in three parts:

1. Inputs

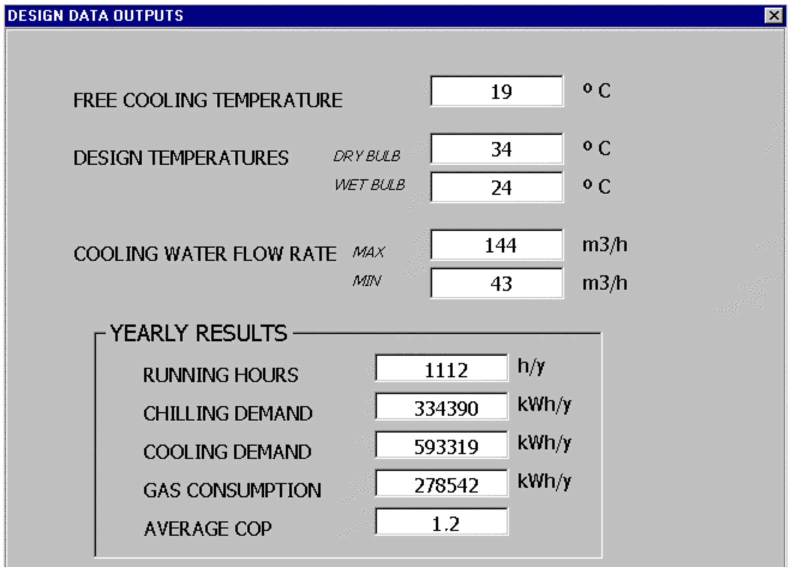
- Design data: Selection of climate (database); selection of building (database); definition of chilling and cooling capacities and definition of cooling towers to be compared (one standard design against one hybrid design). See figure.



- Average cost scenario: introduction or selection from a database of average electricity, natural gas and water costs. See figure.



2. Outputs



- Design parameters: design temperatures (database); free-cooling temperature (calculation) and yearly calculated results (running hours, average load factor, chilling demand, cooling demand, gas consumption, average COP). See figure.

- Comparative balance: electricity and water consumption for selected standard and hybrid towers. See figure.

- Exploitation and investment costs: comparison between standard and hybrid design. See figure.

3. Print/Save Reports: File management facilities.

4.2.8 Final Remarks and Conclusions

Standard design for dry cooling systems shows unbalancement against wet cooling systems when keeping specifications for the project:

	DRY SYSTEM	WET SYSTEM
Maximum air flow (m^3/s)	61,1	20,3
Electricity consumption (kW)	10 x 3,4	1 x 7,5
Footprint (L x W x H)	6600 x 2185 x 1350	2470 x 2470 x 3940

Optimum combination of both systems is a series schema with a three-way valve to by-pass dry section when air temperature is over a certain defined set point (28ºC in case of Paris). Integration of both systems into the same package will not cover specifications. Critical parameters are minimum capacity of dry section (30% of full load capacity) and cut-off temperature of wet section (28ºC in Paris).
Existing hybrid cooling tower manufacturers such as GEA-Soramat or Baltimore HXI offer products that show good performances in dry section when air temperature is reduced below 12 to 14ºC. At those temperatures, cooling demand in typical air-conditioning systems are negligible.
Hybrid cooling towers are, at the present moment, not an optimum choice for air-conditioning. Only when considering large number of running hours (industry, for

instance), in moderate-cold climates and in areas with high cost for water and low cost for electricity can be, to some extent, competitors to standard design.
Only when standard design is changed to closed-loop wet cooling towers, hybrid cooling towers will be able to effectively compete in market.
The following table, compares in a summarized way all evaluated designs for cooling systems (even those that don't exactly match specifications):

COOLING SYSTEM	SPECIFIC WATER CONSUMPTION (l/kWh chilling)	SPECIFIC ELECTRICITY CONSUMPTION (kWhe/MWh chilling)	YEARLY COSTS (water plus electricity plus anuity (10 years))
Standard (open-loop) – reference	3.53 l/kWh	25 kWh/MWh	5194 EURO
Standard (closed-loop)	3.53 l/kWh	41 kWh/MWh	8478 EURO
Hybrid cooling system (separated)	0.68 l/kWh	65 kWh/MWh	4643 EURO
Integrated hybrid cooling tower (GEA)	1.49 l/kWh	51 kWh/MWh	7711 EURO
Integrated hybrid cooling tower (BALTIMORE)	0.91 l/kWh	60 kWh/MWh	8920 EURO

Table 6: comparison of operating cost of three specific design option of cooling towers

It remains to notice that the hybrid cooling system (separated) is already competitive against standard systems (Paris conditions) whilst hybrid cooling towers (integrated, GEA design) are only competitive against standard closed-loop systems.
Summarised results for an **optimised hybrid cooling system** in Paris covering cooling needs of a 1000 kW LiBrac chiller installed in a hotel are as follows:

	STANDARD DESIGN	**HYBRID DESIGN**
Water cooling demand (MWh/y)	831.6	
Natural gas consumption (MWh/y)	410.2	410.2
Electricity consumption (MWh/y)	11.2	21.3
Water consumption (m^3/y)	1585	305
Total running* costs (€/y)	9737	7537
Proportion of water costs	**37%**	**9%**

Table 7: energy and water consumption of cooling towers

(* water, electricity and natural gas)

4.2.9 Total Cost of Chiller & Cooling Tower System

Using the design parameters of an new and more efficient double-effect lithium bromide absorption chiller which will be described later in this report a comparative simulation of total cost of operation has been performed.
The data relevant to the cooling water system are summarized in this chapter.

Based on the extensive data collection has been performed in order to identify a small but typical selection of climatic cases which could be applied to a large number of locations in Europe.The optimisation of the cooling tower concept for a specific application has to include the a number of additional parameters:

cost of electricity
cost of water
cost of water treatment for make-up
cost for blow-down water
avaialable space
various (e.g. noise etc.)

A typical evaluation was carried out in order to identify the influence of the cooling tower design on the total operational costs of the chiller. The right hand bar is for a conventional open cycle wet cooling tower. Dry and hybrid cooling towers, although more expensive, can prove economic because of the savings in water particularly in Berlin where water is expensive.

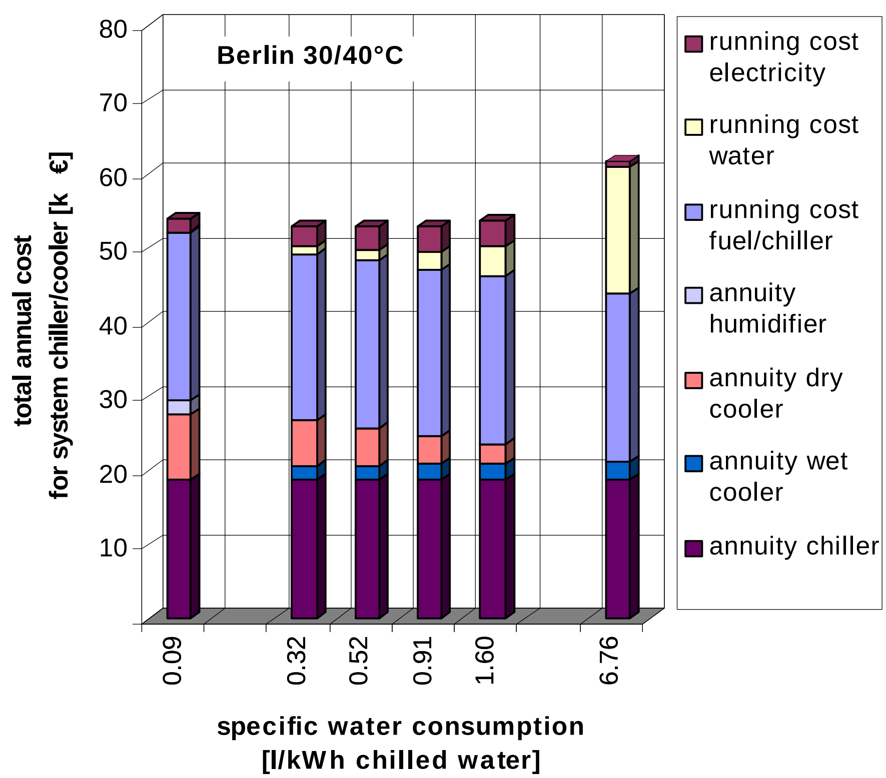


Fig. 5: Contribution of running and investment costs (kEUR) to the total annual cost of a chiller and cooling tower in Berlin

4.3 Optimization of Refrigeration Cycle

A double effect (DE) direct gas fired chiller offers various possibilities of heat recovery enabling an increase of its COP (defined on lower calorifc value LCV). The basic method used to identify the potential of improvement was based on a pinch point analysis, taking into account the constraints imposed by the working fluid pair and the heating medium. This method not also ensures, that the heat is used to the highest possible degree, but also that the heat is used with the best thermodynamical efficiency. Bearing in mind that the temperature of the lithium bromide solution shall not exceed 165°C in the bulk of the liquid due to uncontrolled corrosion, the optimization is limited to the following options:

- Increase Number of Stages (or effects)
  A maximum of two generator stages cannot be exceeded due to corrosion at high temperature even though a third stage would increase the performance much more than any other of the methods cited below.

- Use of low temperature heat of flue gas in the refrigeration cycle
  - in the low temperature generator with an extra coil for flue gas
  - in an additional solution heat exchanger

- Reduction of internal irreversibilites by optimization of the internal heat recovery
  - use heat of condensates from C2
  - reduction of solution flow
  - parallel or serial flow of solution

- Counter-flow pattern in the generator to reduce flue gas temperature at generator outlet.

- Use of other heat sources
  Absorption chillers in cogeneration plants shall not only use the flue gas, but also the low temperature heat sources like intercoolers, oil coolers etc.

- Other low temperature heat sinks in the process
  - air preheater for the burner (relevant only for direct fired units)

Two types of chiller thermal circuit have also been studied :

- a serial solution circuit (see diagram on next page) or with a
- parallel solution circuit (see diagram on next page).

The BGT model of the direct gas fired generator G2 has been integrated into an ENTROPIE calculation program. The used generator G2 design values are similar to those used for the generator design.

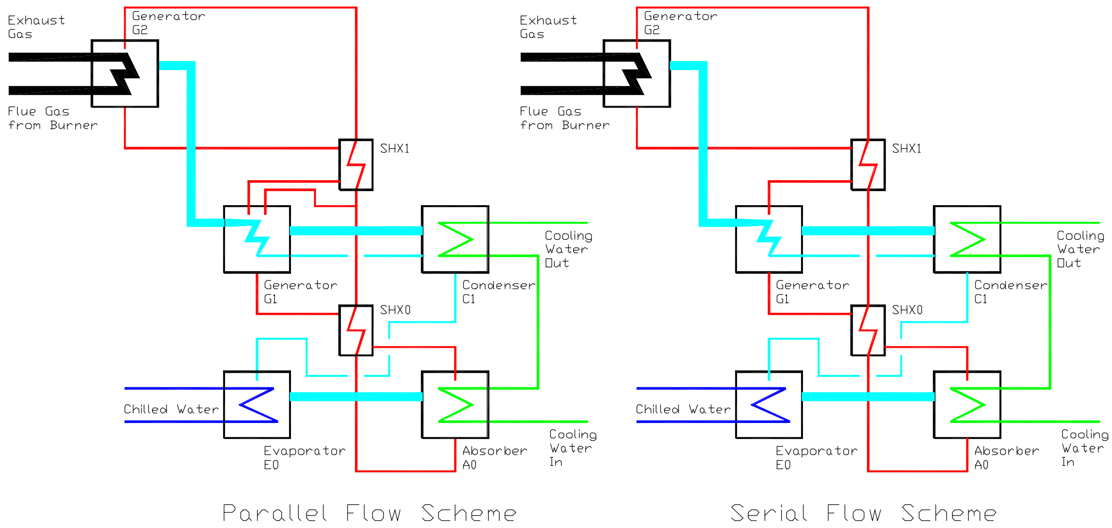


Fig. 6 : Double effect absorption chiller with its main components

Optimization has been limited by relaistic closest approach temperatures of:

15 K for gas-gas or gas-liquid

3 K for liquid-liquid or liquid-phase change exchangers.

The gas burner exhaust temperature has also be kept above 100°C in this first approach, even though the technical solution found would allow flue gas temperatures down to less than 60°C. The latter option is of particular importance for flue gas heated units due the relatively high flue gas flow.

4.3.1 Improvement of Heat Recovery from Flue Gas

The cardinal question to solve was the use of the heat in flue gas after having passed through the high temperature generator. Normally the flue gas temperature is approx. 180°C (assuming sLiBr solution at 165°C a closest approach temperature of 15 K).
The following heat recovery exchangers on gas burner exhaust have been studied (see diagrams above) :

- Air preheater (GE) : Preheater of the gas burner inlet air
- SSHX1-G2 : Exchanger connected to the generator G2 solution inlet
- SSHX0-SHX1: Exchanger between the two solution heat exchangers SHX0 and SHX1
- SSHX1-G1 : Exchanger connected to the generator G1 solution inlet
- SA0-SHX0 : Exchanger at the absorber A0 solution outlet
- SA0-G2: Exchanger connected to an additional solution circuit bypassing the main solution circuit
- SSHX0-SHX1 + SSHX1-G2 : The serial connection of SSHX1-G2 and SSHX0-SHX1 on the flue gases stream was also studied

The effect of each of the exchangers cited above on the chiller COP is summarised on the following bar graph (see fig.13).
Other cases , like recovery from engine exhaust have not been studied quantitatively in such detail within the frame of this project, since a large number of cases would have to be considered, in particular exhaust temperatures vary a lot from engine to engine.

Among the different exchangers which have been studied, the most efficient of them have been selected as leading to an increase of chiller COP higher than 7% :

- for parallel flow scheme :
  SSHX1-G2 + SSH0-SHX1 : + 8.8% of COP increase
  SA0-G2 : +7.3% of COP increase
- for serial flow scheme :
  SSHX1-G1 : +9.6% of COP increase
  SSHX0-SHX1 + SSHX1-G2: +7.4% of COP increase

Fig. 7 H
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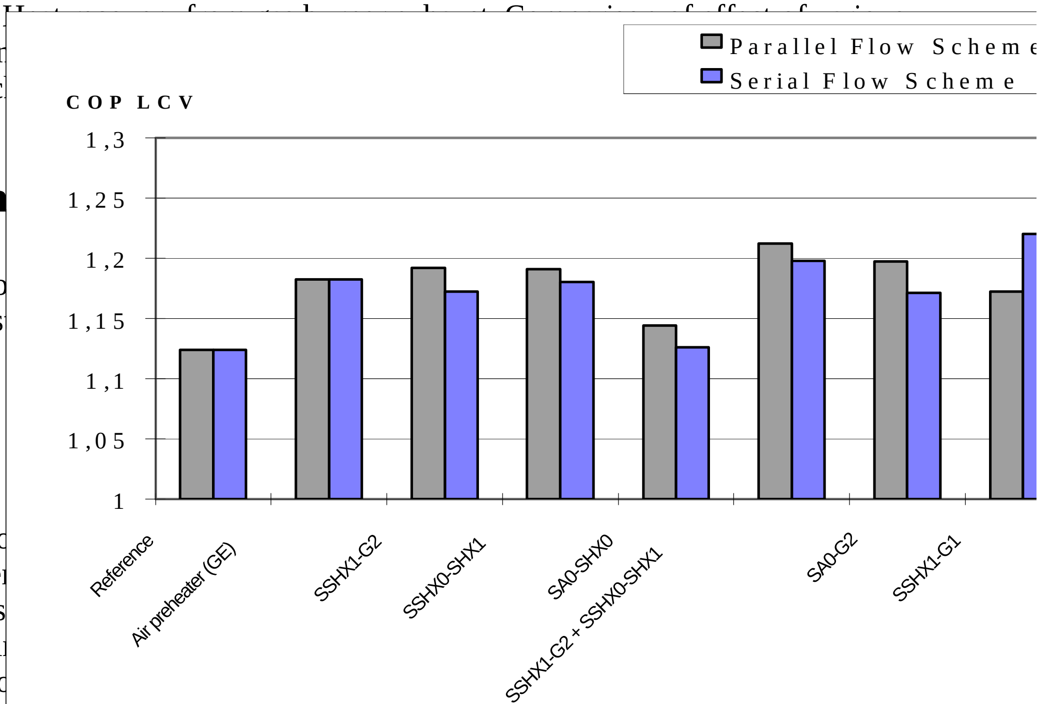

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The SG1-C1 exchanger influences differently the parallel and the serial chillers : COP increase is of 4.6 % for the parallel schema and is negligible for the serial schema.

4.3.3 Other improvements

A number of other potential improvement have not been considered since they are very difficult to implement physically, for example

- Add a flue gas heated section to low temperature generaor (G1)
  This option has been abandoned for direct fired units, since the total amount of heat to be recovered from flue gas after the G2 is about 10%. In case of a unit using flue gas from engines, this is not the case and a direct flue gas heated section may be inevitable.,although an intermediate hot water section may be an interesting alternative as well.
- solution preheater by absorption,
  This option, although feasible is relatively complicated to integrate into a chiller
- condensate - refrigerant vapour exchange in the evaporator
  There is no heat exchanger available on the market for that particular application - extremely high volume flow rates and low presure drop on gas side, very low volume flow rate on liquid side.

- Counter-flow scheme in the flue gas heated generator. This is of interest only for units operating on flue gas engines, where the temperature of the gas is already comparably low.

Even though the the two latter could improve the COP by approx. 5% in theory these options have been abandoned since no reasonable technical solution could be found.

4.3.4 Optimum Heat Recovery System on Direct Gas Fired Chiller

The most efficient out of the previously studied exchangers have been combined and optimized for the direct gas fired chiller thermal scheme.
The following selected systems have been compared :

- <u>SSHX1-G1 + air preheater (G1C)</u> for parallel and serial schemas
- <u>SSHX1-G1 + SG1-C1</u> for parallel and serial schemas
- <u>SSHX0-SHX1 + SSHX1-G2 + SG1-C1</u> for parallel and serial schemas
- <u>SA0-G2 + SG1-C1</u> for parallel schema The effect of

The different groups of exchangers is shown on the following bargraph. The 4 most efficient groups of heat exchangers lead to an average COP increase of 13%. Considering a global heat exchange coefficient of 10 W/m²K for liquid / gas exchangers and of 1000 W/m²K for liquid / liquid exchangers, the different selected groups are hereunder compared with the heat exchange surface required for the heat recovery exchangers.

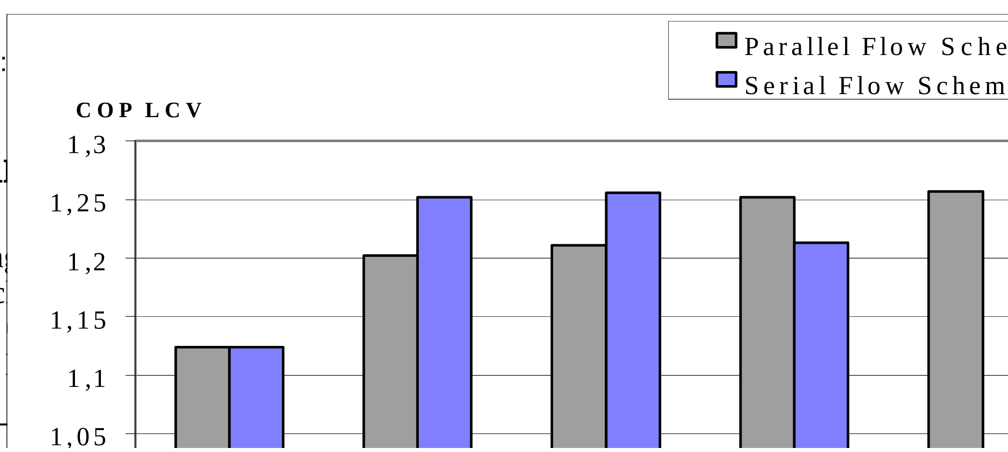
- For serial flow scheme ***:***

SSHX1-G1 + air preheater (G1C) : +12.8 % of COP increase,
total surface : 160 m²
SSHX1-G1 + SG1-C1 : +13.2 % of COP increase,
total surface : 109 m²
(<u>SG1-C1</u> surface : 22 m²)

- For parallel flow scheme :

SSHX0-SHX1 +SSHX1-G2 + SG1-C1 :
+12.8 % of COP increase,
total surface : 139 m²
(<u>SG1-C1</u> surface : 15 m²)
SA0-G2 + SG1-C1 : +13.3 % of COP increase,
total surface : 125 m²
(<u>SG1-C1</u> surface : 14 m²)

Fig. 14 :



4.4 Opti

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Step 1:
Aim:

We studied the influence of LiBr solution flowrate on the chiller heat exchange surfaces. The optimum nominal solution flowrate has then been fixed to 8.7 times the vapour flowrate produced in the evaporator E0. This solution flowrate enables to achieve a COP of 1 at 30% part load operation without any solution flowrate variation.

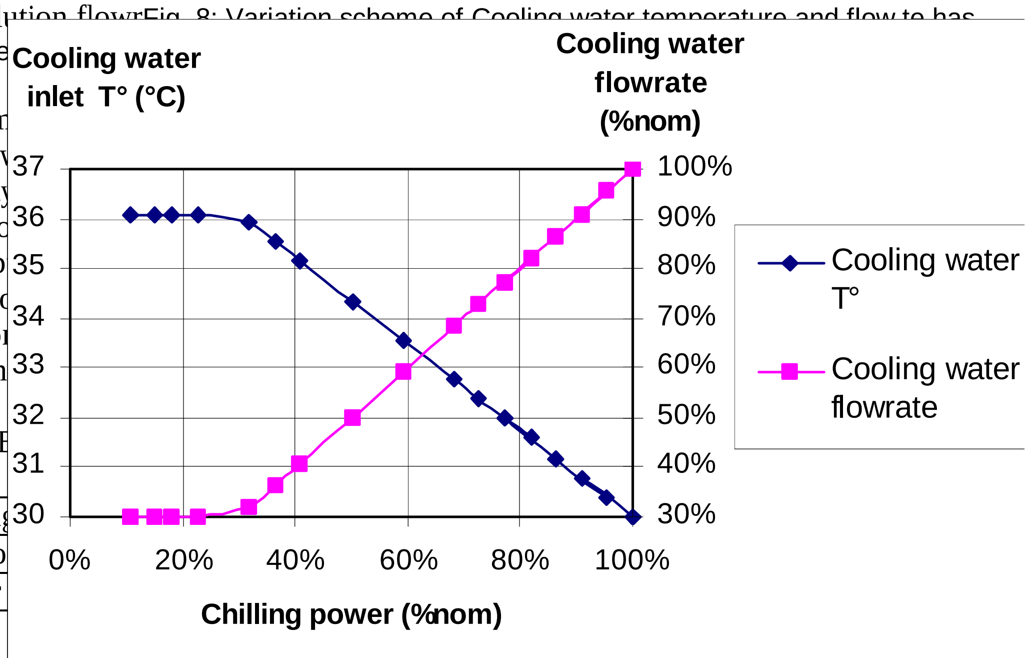
Step 2: Optimisation for part load operation

Aim: increase of cooling water temperature to increase capacity of dry cooling tower

The operation of the DE parallel schema chiller has been simulated for degressive loads. Variations of cooling water flowrate and inlet temperature have been optimised in order to minimise the water and electrical consumptions for part load operation. The cooling water inlet temperature has then been raised and the cooling water flowrate has been in the same time diminished down to reasonable level.

These variations have been optimised in order to maintain the generator G2 pressure below its full load operation level.

The solution flowrFig. 8: Variation scheme of Cooling water temperature and flow te has also bee



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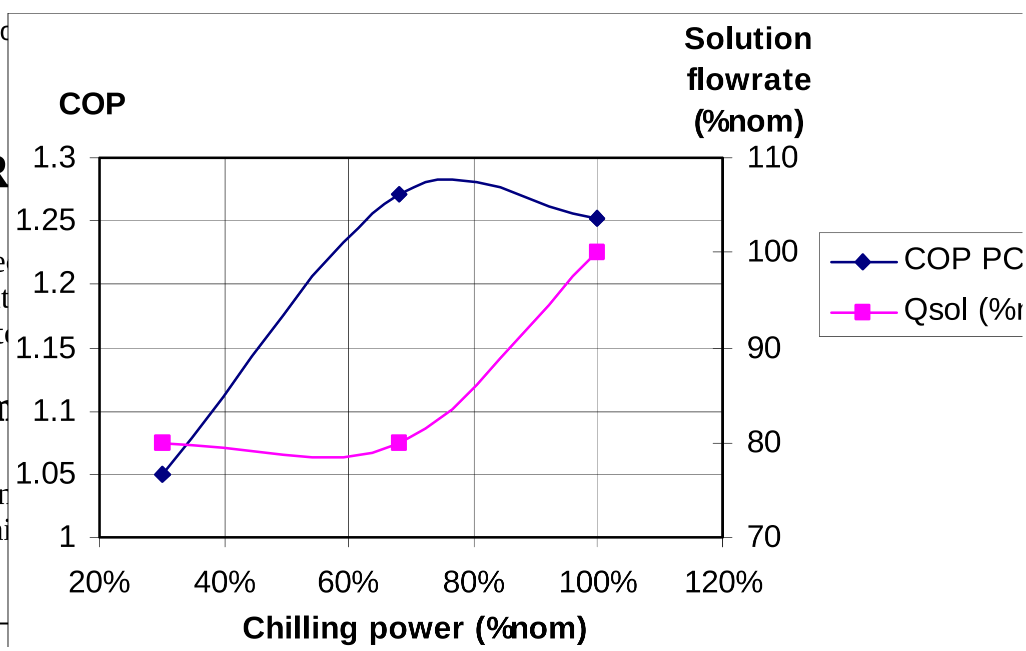
The optimised part load operation of this chiller is shown in the diagrams hereunder.

Fig. 9 : Lo



4.5 LIBR

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been the reduction of this volume. The reduction of the solution volume leads also to the reduction of the chiller investment cost as one litre of solution costs 11.4 EUR .

The thermal inertia of a chiller mainly consits of two terms:

- one related to the mass of the equipment and the corresponding temperature variations these masses have to undergo during a load change
- the second related to the heat of evaporation of the mass of water that has be evaporated to change the concentration of the solution.

Since the temperature changes mainly occur in the generator and rarely exceed 20 K when increasing output from 50% to 100% the related energy to heat up the generator is not very important. On the other hand, the mean concentration of the whole inventory of solution has to change by more than 10% of salt content leading to an equivalent of more than hundred kilograms of water to be evaporated per ton of solution inventory.

Therefore the reduction of liquid volume is one of the key problems to be solved if a rapid reaction of the chiller shall be achieved.

The G2 boiler has indeed been designed with a solution volume reduced by more than 75% compared to standard multi-tube designs. The solution heat exchangers SHX0 and SHX1 have also been selected for their solution volume more than three times reduced compared to the shell-and-tube designs used in the current offered chillers.

The generator G1 has also been designed as a sprayed tube bundle design has been preferred to the current flooded tube bundle one.

4.5.2 LIBRAC Chiller Operation

The whole LIBRAC chiller regulation is based on the chilling power control, usually carried out with the comparison of the chilled water outlet temperature with a set point. The different following parameters will be directly controlled by the chilled water regulator :

- gas burner or waste heat exhaust gases power rate
- cooling water flowrate
- set point inlet temperature of the absorption unit cooling water
- the cooling tower air flowrate (or fan speed) will be adjusted for an inlet temperature of the absorption unit cooling water reaching the set point value

The absorption unit internal process control is roughly the following :

- solution flowrate regulated at inlet and outlet of the G2 boiler
- decristallisation : injection of evaporator E0 condensate into the solution circuit at the absorber A0 solution outlet.
- regulation for part load operation of the main solution flowrate at the absorber Ao outlet

Chiller safeties have been studied and defined in order to be conformal to the EU 97/23 directive for prrssure equipment.

European pressure equipment directive classifies the pressure equipment by the vapour pressure of the liquid at the operating temperature and the corresponding volume. In order to ease the integration into a building and the cost related to workshop testing and eventual third party approvals in workshop testing and site approval it is convenient to be in lowest possible class.
In general the pressure in bodies of the exchangers are below atmospheric and hence the bodies are not even subject to CE marking. The only exception may apply to the gas fired generator, where high temperatures may occur. Due to the properties of the LiBr solution the pressure in the direct fired genreator and hence also in tube sid e of low temperature generator are below 0,5 bar relative if the exchangers are properly designed. In this case the whole absorption chiller falls into the lowest class of EN97/23. The contropl and safeties of burner have to be extended only by an approved pressure switch and eventually a rupture disc on the generator.

4.6 Design of Flue Gas heated LiBr regenerator

4.6.1 Generator Configuration

The generator is required to concentrate the lithium bromide solution by boiling and separating water from the solution as low pressure steam. Boiling point temperatures are likely to reach 160°C when the salt concentration reaches 64% and the generator pressure is approaching 0.5bar g. The design pressure was later increased to 1bar g to allow a larger range of operating conditions for the tests.

The generator was designed to be flexible, scaleable, efficient, low cost, compact and light-weight with improved part load efficiency and controllability. After considering the different heat exchanger options available it was decided that a suitable unit could be designed around a cross flow heat exchanger technology, with the lithium bromide solution on the inside of the tubes. Some in-house expertise on this type of design already existed albeit with a lower duty and using all finned tubes. The generator incorporates a lower solution volume and by careful thermal design can operate with lower solution side surface temperatures which helps to avoid both corrosion and breakdown of additives such as corrosion inhibitors in the solution.

Weak lithium bromide solution is pumped into the generator. It then circulates through the heat exchanger tubes by natural thermo-syphon action. The solution in the tubes rises due to boiling action caused by heat transfer from burner combustion products or from another heat source. A two phase mixture leaves the heat exchanger at velocities around 10m/s into a separation area (top header) containing impingement plates and demister pads.

Weak solution is pumped into the top header at the front side of the generator (burner side) and mixes with the hottest concentrated solution in the generator. Pumping weak solution into the bottom header was initially avoided as it could flash and preferentially dry-out some of the heat exchange tubes or interfere with the thermo-syphon operation. A calming zone is maintained at the rear (exhaust side) of the header by using an underflow - overflow weir system which helps to restrict any inlet solution by-passing the generator tubes and provides a more stable region for effective control of solution level in the generator. The weir system enables removal of the most concentrated solution and prevents weak solution by-passing the heat exchanger. Out flow of concentrated solution is from the calmed region, which also helps to avoid entrainment of vapor bubbles down the outlet pipe.

The thermo-syphon tubes are positioned on the side of the heat exchanger to ensure the front tubes are always fully supplied (wetted) with liquid solution. Steam leaves the generator through the top of the top header after passing a baffle system and a demister pad.

4.6.2 Heat Exchanger Design (1MW and 350kW)

The heat exchanger was designed to be easily scaled for operation over the full range from 250kW to 2.5MW using either direct firing with a fuel gas or turbine exhaust gases or industrial waste gases. Initial detailed designs have concentrated on the 1MW and 350kW direct fired units but calculations have also been carried out to show its suitability to lower temperature waste heat or exhaust gases. The 350kW heat exchanger is arranged in the same manner as the 1MW unit but one third of the width of the 1MW.

The heat exchanger is a cross flow plain and finned tube bundle with lithium bromide solution passing through the inside of vertically mounted plain and finned heat exchanger tubes all connected in parallel.

Hot combustion products from the burner flow initially through a plain tube section then through a finned tube section in a single pass. They then leave through a flue in the opposite side of the heat exchanger. Additional heat is recovered in the flue gas recuperator, a compact finned heat exchanger.
Corrosion from high temperature lithium bromide placed a design constraint on the tube spacing and material choice. The initial material choices were narrowed to more expensive cupronickel, with high temperature corrosion resistance, or to lower cost carbon steel where the corrosion rates were predictable. Carbon steel was chosen as it is more readily available, cheaper and easier to machine and fabricate. Copper was chosen as the fin material due to its high thermal conductivity.
A heat exchange design calculation was carried out using computer models based on published heat transfer correlations. The thermal duty was assumed to be 1 MW of gas input with 30% excess air; this provided a safety margin of about 10% to allow for variations in the heat transfer correlations and reduce the risk of not achieving the necessary high efficiency. The target thermal efficiency was 80% based on the gross calorific value of natural gas.

4.6.3 Predicted Thermal Performance

Computer Model Development
A computer model was developed at BG Technology to simulate operation of a tubular cross-flow generator operating from either direct natural gas firing or from the lower grade exhaust products from natural gas firing (e.g. from an engine or turbine). A variant of this model was used to size the flue gas recuperator.
This model was used to size the direct gas-fired 350kW test unit and predict the performance under a range of conditions. The model can be used to predict overall heat transfer, combustion side and flue gas temperatures, tube wall temperatures, and gas side

pressure drops and fan power requirement. Input data includes the tube configuration (length, diameter, number and spacing/pitch), fin details, gas firing rate and excess air, and solution temperature.
The model undertakes a row by row calculation of the gas side heat transfer starting from the burner (hot) side of the tube bank. A simpler spreadsheet model was also written based on the results of this model; this is used for quick parametric study or can be incorporated into an overall absorption cycle analysis.

Predicted results
The inside nucleate boiling heat transfer coefficient was calculated to be around 5,000 to 10,000 $W/m^2 K$ after allowing for the wall resistance and some fouling. The overall heat transfer coefficient, based on outside surface area, was calculated to be about 65 $W/m^2 K$ in the plain tube section and 22 $W/m^2 K$ (based on the overall finned area) in the finned tube sectionRecuperator/Economiser Designs
The recuperator unit was designed to recover a further 4% of the gross thermal input to the system by further cooling the flue gases by about 100°C to 109°C. The novel absorption cycle design derived by Entropie, makes this more feasible by passing a small flow rate of cooler weak solution through the recuperator tubes before entering the generator. This solution does not pass through the solution heat exchanger and allows this to operate more favourably by balancing the hot and cold streams more effectively.

4.6.4 Generator Burner Design

During the design stages all major burner types were considered and assessed for their suitability at the 1MW+ size and down to 300kW. Off-the-shelf burners, including packaged burners were investigated for the new generator design. These were found to be either too expensive or incompatible with the generator configuration. A list of burners considered is shown in Table 3.

Burner Type	Examples	Cost £ 350kW 1MW	Comments
Packaged	Weishaupt GP Burners	3,500 5,500	Low cost particularly for larger sizes. Long combustion chamber required.
FireStream Grater	Stordy	5,070 N/A	High excess air requirements/large fan
Flatflame	Stordy	N/A	Not practical -too big for heat exchanger
Metallic Fibre	Acotech	5,267 High	High cost of Fecralloy, controls could be expensive for larger burners.
Ceramic Plaques	BG Technology	5,000 Med	Low cost burner, controls could be expensive for larger burners.
Immersion Tube	Process Burners ltd	N/A	Not suitable. Too large a LiBr solution inventory ouside tubes.

Table 8: Burner package examples

To aid size, weight, turn-down and to minimise emissions, a compact, intense flame, premix burner was chosen. Acotech(Belgium) were contacted to supply a prototype metal fibre matrix burner (Table 4). A vertical burner ($0.14m^2$) was selected with firing intensities of 250 to 2500 kW/m^2 (35 to 350kW), to operate mainly in blue 'lifted flame mode. Two or three of these units would be used in the 1MW design.

The burner operates in part radiation mode at lower firing rates but the highest heat fluxes will always be observed during the high firing rates. BG Technology have tested an earlier prototype of these burners and produced an operating map. Premix burner designs offer improved emissions but can suffer from resonant noise if incorrectly designed.
In order to reduce the cost of the burner, further development has been undertaken using a novel design of radiant ceramic plaque. These plaques are cheaper to fabricate than the fecralloy and offer similar performance in

Burner Specification (350 kW)			
Type	ACOTECH Metal Fibre Premix		
Model	ACO 98 ...350kW		
Combustion Surface Area	m^2	0.1400	
Opening for Electrodes		3/4" Dia.	
Burner Housing		2mm thick stainless steel	
		High Fire	Low Fire
Rating	kW	350	35
Gas Flow rate (Nat.Gas)	m^3/h	32.67	3.27
Firing Intensity	kW/m^2	2,500	250
Excess air	%	30	30
Air Flow rate	m^3/h	412.02	41.2
	L/s	114.45	11.44
Mixture Flow rate (air equivalent)	m^3/h	466.47	46.65
Burner Mixture Pressure	mbar	1	0.01
	Pa	100	1
Air Inlet Temperature	°C	20	20
Mixture Inlet Dia	mm	Square gas/air inlet adapted for the Ziehl-ebm fan G1G170-AB31-04.	
Gas Pressure	mbar	30	

Table 9: Burner specification for low and high fire

terms of thermal loading, flame stability. The unique design allows high thermal input per area without
risk of resonance. A picture of a prototype burner under test is shown below . This burner has undergone open firing tests in radiant and lifted flame mode but the budget has not permitted testing on the generator.

Fig. 10: Impro

4.6.5 Gas Burn

The gas cont ... and standards to minimise any ... mark. Components are off-the-shelf ... tion and detailed design of the burner ... ufacturer, Stordy

Combustion Engineering so that appropriate certification (EN676 and TRD412) and guarantees could be obtained.

4.6.6 Operational Experience

The experimentally measured performance of the generator under test in Germany, was found to be in good overall agreement with that predicted by the model. There were no serious technical problems encountered during its prolonged operation. The test unit performed well, giving confidence that a robust, efficient and compact target design had been achieved. In summary, these predictions were as follows:

- Overall boiler efficiency (based on gross CV) of 80.5% (target 80%)
- Maximum inner tube wall temperature of 182°C (target max 185°C)
- Overall gas side pressure drop through boiler of 52 Pa
- Flue temperature of 244°C
- Heat transferred to lithium bromide solution of 282 kW
- Combustion air fan power requirement of 27 W

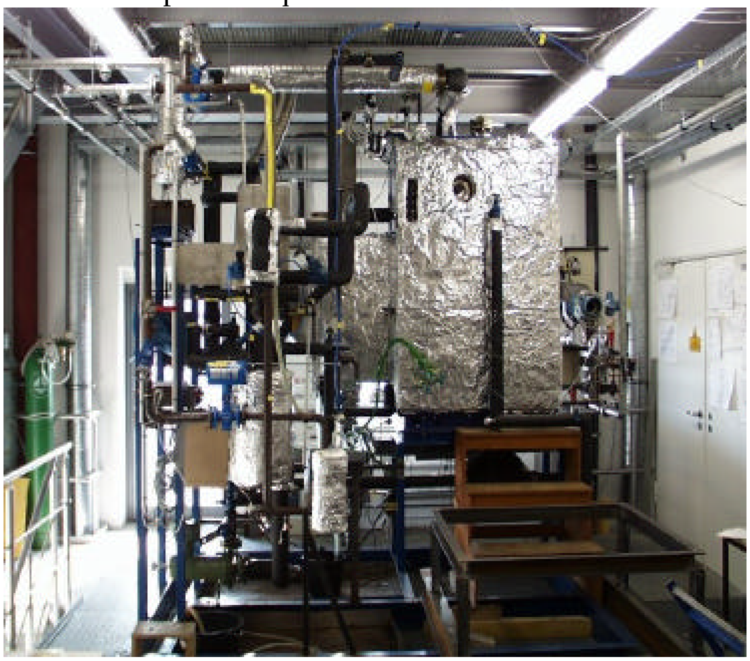


Fig. 11: A side view of the generator unit installed on the test rig in Germany

Fig. 12: End view of the generator under test at BG Technology

4.6.7 Waste Heat Firing Of Generator

During the generator design process, careful consideration was given to the potential use of waste heat from cogeneration sets (engine exhaust products) as a way of providing the necessary heat to solution. The use of engine waste heat to fire an absorption chiller could offer an attractive proposition in terms of tri-generation development. Engine exhaust products would typically be at a temperature of 560ºC (i.e. significantly lower than from a direct gas flame), thus resulting in far less heat transfer driving force within the generator. This limitation can be overcome by increasing the heat transfer area. Alternatively the mass flow rate of combustion products can be increased to improve the convective heat transfer into the solution. In this case, the pressure drop through the generator has to be balanced against the available exhaust side pressure from the engine. The model was used to investigate this scenario, with summary results presented in the table 13.

Mass flow (kg/s)	Plain tube section DP (Pa)	High fin section DP (Pa)	Total gas side DP (Pa)	Fan Power (W)	Heat transferred into solution (kW)	Flue Temp (C)
0.2	27	29	56	22	85	198

0.5	150	171	321	318	193	234
1.0	481	653	1134	2300	344	271

Table 10: Predicted (350kW) generator performance on engine exhaust products at 560^{o}C with unchanged geometry .

4.6.8 Scale-Up Of Generator

The computer model developed and applied to the design of the 350kW test rig is capable of being applied to thermally design a range of generators. It is anticipated that scale up from 350kW to 1MW will be through the simple scaling up of the width of the existing 350kW test rig generator. In practice, the overall width of a larger unit becomes proportionately less when scaled up because the insulation thickness at the side walls does not increase by the same factor.

4.7 Design of Test Rig with 350 kW gas fired regenerator

4.7.1 Flow scheme of the regenerator test rig

The lithium bromide solution regenerator prototype being tested is one major component of an absorption chiller where the working solution is reconcentrated by evaporating some of the water which acts as refrigerant. Instead of setting up a complete absorption cycle, however, here only a special test station was built where the working characteristics of the other chiller components can be simulated. Besides this concept was more time and cost efficient, it also enables a flexible adjustment of various operation modes of the prototype avoiding limitations or interference of other chiller components.

Absorber, evaporator and solution heat exchanger of a real absorption chiller are replaced with a simple mixing. Hence only relative small heat exchangers are necessary to manage the inflow conditions of the regenerator and economizer.

The steam is condensed in one plate heat exchanger (S0) and mixed with the cooled concentrated solution from the regenerator outlet. This mixed solution is fed into the regenerator inlet. In addition a part of the mixed solution is cooled in a heat exchanger (S2) and passes through the economizer. At the outlet of the solution storage vessel a pump is installed, which feeds both circuits through the regenerator and the economizer. The outlet solution flow from the regenerator is controlled by a proportional membrane valve in order to maintain a certain solution-level in the regenerator. The heat exchangers S1 and S2 have bypass control valves to adjust the transferred cooling power. To adjust a certain concentration of the solution the condensate storage vessel is needed. Here a certain amount of the evaporated water can be stored. In the condensate line “CD1” a condensate deflector is installed to prevent steam from flowing into the solution storage vessel. The line “IN1” is closed during normal operation as it is just used for evacuating the system.

The cooling water passes serially through the heat exchangers S2 and S1 and parallel to it through the heat exchanger S0, which works as condenser. The cooling water flow is variable in each line.

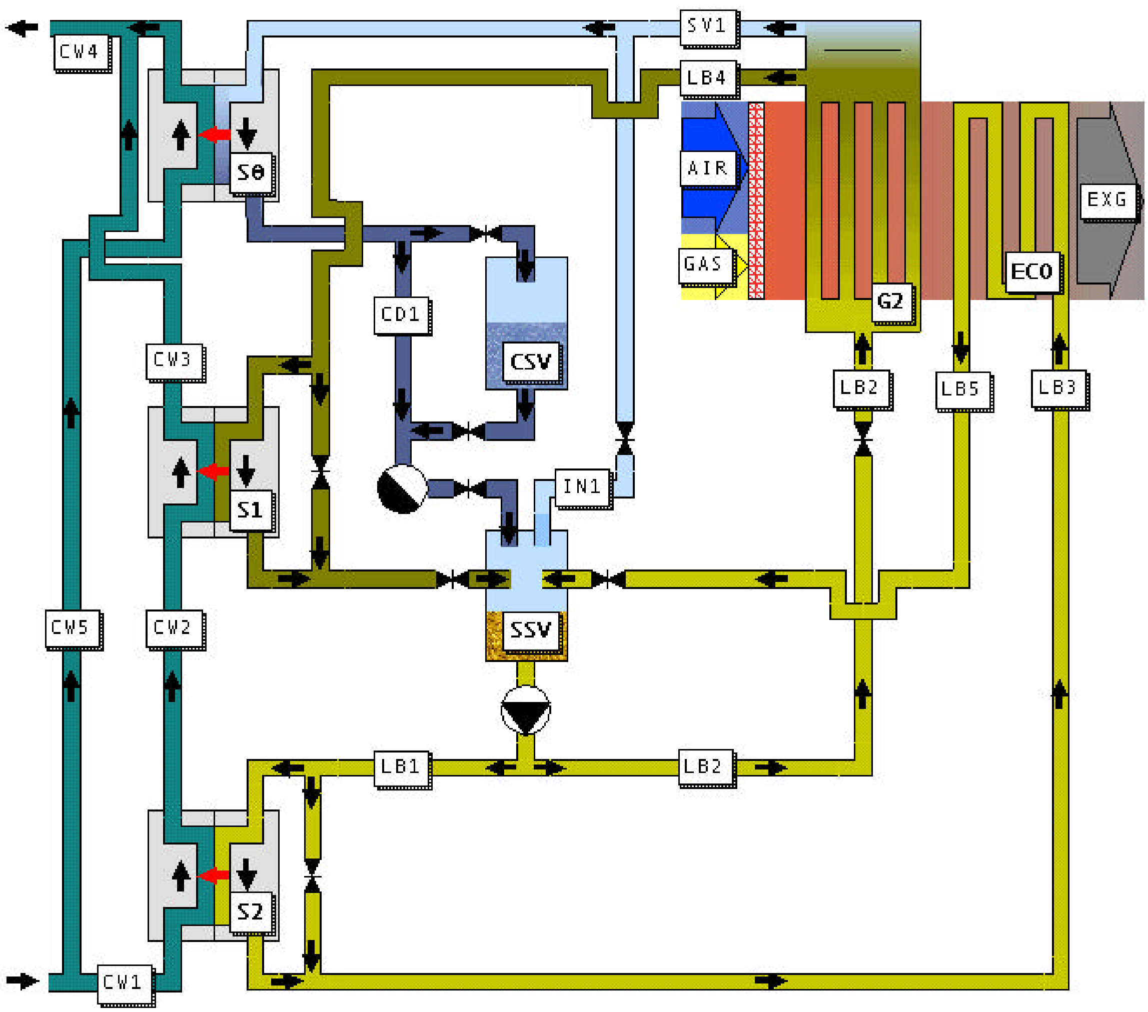


Fig. 13: **Flow scheme of the regenerator test station with all components.**

G2 lithium bromide solution regenerator
ECO economizer, i.e. economizing exhaust gas heat recuperator
S0... 2 simulation heat exchangers 0... 2
CSV condensate storage vessel
SSV solution storage vessel

connection lines:

CW1... 5 cooling water lines 1... 5
LB1... 5 lithium bromide solution lines 1... 5
SV1 saturated vapor
CD1 condensate
IN1 incondensable gases
GAS natural gas input
AIR combustion air input
EXG exhaust gas output

4.7.2 Construction and Operation of Test Rig

The operation as a component of a absorption chiller with lithium bromide solution sets high requirements concerning vacuum tightness as well as corrosion resistance of the test station. Therefore all solution pipes are welded and the unavoidable flanges are especially sealed. For the whole plant a vacuum tightness better than 10^{-3} mbar×l/s was proven.
To ensure proper and safe operation of the gas fired regenerator plenty of safety equipment had to be installed. Emergency shutdown has to be ensured in case of overpressure, over-temperature or improper solution level at the boiler to prevent damage of equipment or harm to men. All safety devices have been cautiously adjusted and checked after installation to ensure proper functioning and to avoid interference of normal operation states. Also some additional installations for operation and control of the system had to be commissioned. Finally, to prevent thermal losses and hence to ensure accurate measuring of enthalpy fluxes, all tubings have been insulated. All this tasks have been completed between June and September 1999 before starting the measuring program. A photograph of the test stand at this state is shown below.
For a further reduction of the thermal losses of regenerator and economizer, a secondary outer insulation has been applied on the casing of the prototype. A photograph of the completely insulated prototype is shown in the next photgraph.

Fig. 14: **Photograph of the regenerator test station after complete insulation of the prototype.**

4.7.3 Measurement concept and instrumentation

Plenty of measured data are necessary to control the operation of the test station and to gain the necessary information about performance, efficiency and the operational

behavior. All external cooling water and gas powers are to be determined as well as the internal cycle conditions.
The following table (see the next page) shows a detailed list of all measurement points. During operation all these measured data are scanned, processed, visualized and stored automatically and continuously. At each component the inlet and outlet temperatures of the fluids are measured, supplementary some additional interesting temperatures. For the enthalpy balance also the fluxes through each component are needed.

Channel no.	P & I D-Tag	Measuring Point Description	type of Instrument
		Temperatures refrigeration circuit	
T01	LB1 T1	LiBr Solution G2 in	PT100
T02	G2 T2	LiBr Solution G2 bottom	PT100
T03	G2 T1	LiBr Solution G2 top	PT100
T04	LB4 T2	LiBr Solution G2 out	PT100
T05	LB4 T1	LiBr Solution Storage in	PT100
T06	LB3 T1	LiBr Solution Storage out	PT100
T07	LB5 T1	LiBr Solution Economizer in	PT100
T08	LB1 T2	LiBr Solution Economizer out	PT100
T09	SV1 T1	Saturated Vapor G2 out	PT100
T10	CD1 T1	Condensate Storage in	PT100
		Temperatures cooling water	
T21	CW1 T1	Cooling Water S2 in	PT100
T22	CW2 T1	Cooling Water S2 out / S1 in	PT100
T23	CW3 T1	Cooling Water S1 out	PT100
T24	CW5 T1	Cooling Water S0 in	PT100
T25	CW4 T2	Cooling Water S0 out	PT100
T26	CW4 T1	Cooling Water out (S1 + S0)	PT100
T27	GAS T1	Gas Burner in	PT100
T28	AIR T1	Air Burner in	PT100
T29	FG T1	Flue Gas G2 in - Thermocouple	Thermocouple
T30	FG T2	Flue Gas G2 out - Thermocouple	Thermocouple
T31	FG T3	Flue Gas G2 out	PT100
T32	FG T4	Flue Gas	PT100
		Temperatures G2 Tubes	
T41 - 51	G2T 1T1-11	G2 Tube 1 - 11	Thermocouple
Pressure			
P61	G2 P1	G2	Absolute Pressure
P66	GAS P1	Gas	Absolute Pressure
P67	AIR P1	Delta P Air	Differential Pressure
Flow intern			
F71	LB1 F1	LiBr Solution G2 in	Coriolis-Massflow
F73	G2 F1/2	LiBr Solution G2 Circulation Tube front/back	Ultrasonic-Volumeflow
Flow extern			

F81	CW1 F1	Cooling Water S2 / S1	Magn.Induktive-Volumeflow
F82	CW4 F1	Cooling Water S0	Magn.Induktive-Volumeflow
F83	GAS F1	Gas	Gasvolume-Counter
Density			
D91	LB1 D1	LiBr Solution G2 in	Coriolis-Density
Level			
L111	G2 LC1	LiBr Solution Level in G2	capacity level sensor
		Flue Gas Temperatures	
T121			Thermocouple
T122			Thermocouple
T123			Thermocouple
T124			Thermocouple
T125			Thermocouple
T126			Thermocouple
T127			Thermocouple
T128			Thermocouple
T129			Thermocouple
T130			Thermocouple
T131			Thermocouple

Table 11: Complete list of measuring points including the flue gas temperature sensors which were not installed during the initial runs.

4.8 Operational experience with Gas Fired Regenerator

The generator has been operated for mor than 5 months (naturally with interruptions) and it turned out that the major problems of operation were not really related to generator but mainly to the peripherals of the test rig.
The generator has been equipped with a fully automatic burner control system for safety reasons which also included a number of essential securities for the boiler.
In contrast to the generator the peripherals were purely manual, except regulation.

In the beginning the adjustment of the securities turned out to be quite delicate, since the instruments used were not all designed for operation with LiBr-solution.

Even though, after discussion with the TÜV in Munich, the generator has been classified in the lowest group of pressure equipment which simplified the approval prcedure considerably, had to equipped with insturments to detect various conditions of potential risk:

- temperature switches in the downgoing pipes for solution in order to detect crystallisation which would lead to insufficient natural convection and hence overtemperature in the boiler tubes. A hardware solution has been chosen (thermostat switches set to a fixed temperature)

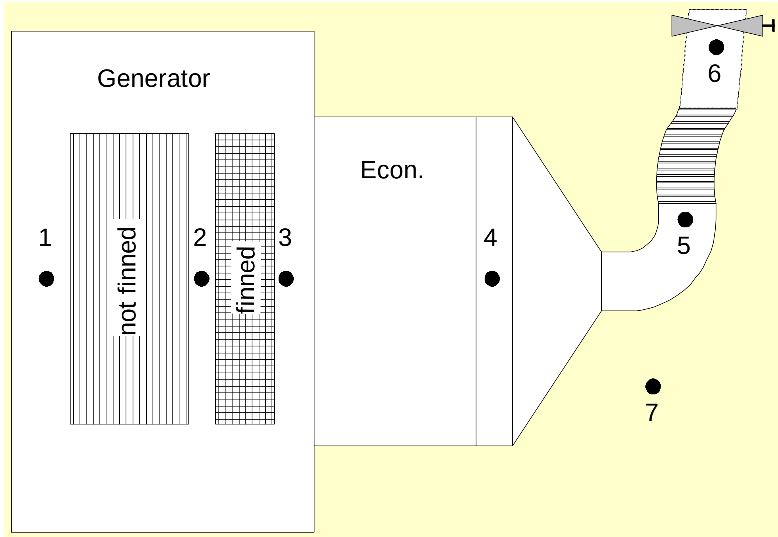
- conductive level sensors to detect low level which could lead to dry out of tubes
- conductive level sensors to detect high level
- max. temperature switch
- max. pressure switch (a hardware switch of approved type must be installed)

4.8.1 Thermal Performance, Flue gas flow and uniformity

The thermal performance measurements that have been carried out, turned out to be quite demanding in temrs of instrumentation and time.
The original intention was to measure
pressure drop
and flue flue gas temperature at different locations
i.e. in front of first row
after plain tube section
after finned tube section
after economizer
While the pressure measurement showed no particular surprises , apart from the absolute value of pressure drop which turned out to be lower than expected, the flue gas temperatures were too inaccurate and inconsistent for a calculation of heat transfer coefficient of the individual tube rows.
A new method which is described below had to be developed and turned out to give satisfactory results.
In addition a precise measurement of the flue gas temperature in front of the first row has been performed using a suction thermometer. This measurement was used to estimate the influence of radiative heat transfer which was an area of concern for the first two rows initially.

4.8.2 Method and points of measurement for pressure drop

In order to determine as well the corresponding pressure drops over orifice plate, fan and burner matrix, refined pressure measurements have been carried out. In addition to the earlier investigated measuring points 1 to 7 within the flue gas train (see figure below), measurement was performed at two extra locations. A differential pressure transmitter with an accuracy better than 10^{-2} mbar was employed for all measurements. For safety reasons, during all measuring the positive port of the pressure transmitter stayed connected to point *B* between fan and burner matrix (gas/air mixture under pressure!) while the negative port was switched between the other locations. Therefore all pressure data including ambient pressure have been measured relative to *B*. The flue gas suction fan was running during the whole series of measuring.



pressure measuring points

A	*between orifice plate and fan (BGT-P1)*
B	*between fan and burner matrix (BGT-P2)*
1	*combustion chamber (BGT-P3)*
3	*between regenerator and economizer (BGT-P4)*
4	*immediately after economizer*
5	*after cone and bend*
6	*before chimney inlet valve*
7	*ambient pressure*

Fig. 15: - Location of the individual points of pressure measuring. Additional points A (in correspondence to BGT point P1) and B (BGT point P2) were investigated for this series of measuring, earlier measuring point 2 was omitted.

4.8.3 Results

For four operating conditions from 120 kW to 246 kW gas power (net calorific value) the pressure profiles along the air and flue gas train from the air inlet to the economizer outlet according to the measuring points shown in figure 1 were recorded.

System Pressure [mbar] relative to Ambient				
Gas Power (net.)	*120 kW*	*160 kW*	*215 kW*	*246 kW*
7 - Ambient Pressure	0,0	0,0	0,0	0,0
A - After Orifice Plate (BGT-P1)	-1,4	-2,3	-4,2	-5,8
B - Before Burner Matrix (BGT-P2)	0,7	2,7	5,7	7,6
1 - Combustion Chamber (BGT-P3)	-1,0	0,0	1,4	2,1
3 - Between G2 and ECO (BGT-P4)	-1,1	-0,1	1,2	1,9
4 - Immediately after economizer	-1,8	-1,0	0,1	0,6
5 - after cone and bend	-2,0	-1,3	-0,4	0,2

Table 12: Calculated system pressure relative to ambient at four operating statuses with different gas power (net calorific value).

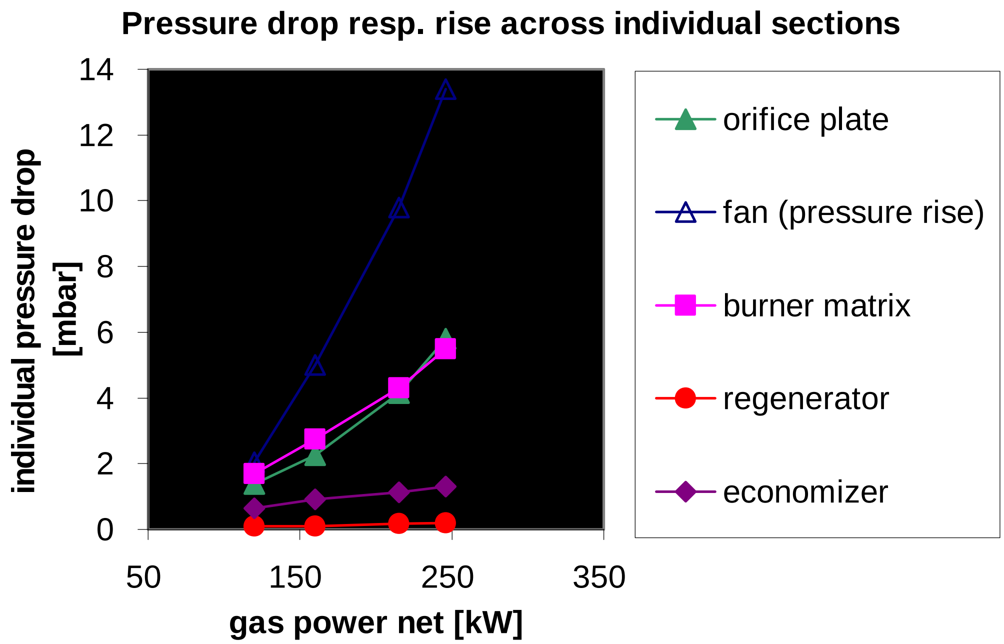


Fig. 16: Individual pressure drop resp. pressure rise across different sections of the air/flue train from air inlet to economizer outlet at four operating statuses with different gas power (net calorific value).It shows that the total pressure drop of the system burner/regenerator is heavily dominated by the pressure drops across the inlet air orifice plate and the burner matrix.

4.8.4 Generator Thermal Efficiency

A high thermal efficiency of the new generator was one of the key objectives of the project. Thermal efficiency has been calculated by two different methods:

an energy balance on the gas side

an energy balance on the solution / cooling water side

The energy balance can be calculated relatively straightforward

Q in = LCV * m gas

Q out = m flue * cp flue * (Tflame - Texhaust) / (Tflame -Tambient)

Boiler efficiency can be expressed by simply measuring exhaust temperature and the calculation of the adiabatic flame temperature:

efficiency = (T_flame - T_exhaust) / (T_flame -T_ambient)

T_flame is the theoretical adiabatic flame temperature, which can be calculated from air ratio. All these variables can be measured with high accuracy with standard equipment available for boilers.

The other method of calculating the efficiency is much more complex. It requires a large number of mesaurements of relatively high precision and it therefore expected to be prone to larger errors and uncertainties. The heat input can be calculated from the metered gas flow and the LCV of the gas, whereas the heat output has to be calculated from enrgy balances in the coolong water and an assumption of heat. The values obtained on both ways showed good consistency, however a systematical difference remains unexplained. The flue gas efficiency which – in contrast to the cycle efficiency – does not reflect the thermal losses of regenerator and economizer results directly from the measured flue gas outlet temperature and the corresponding excess air ratio.

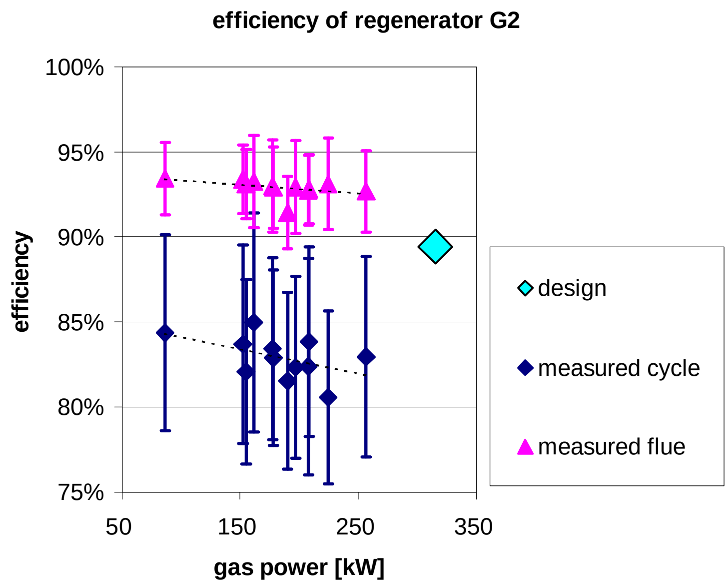


Fig. 17 Thermal efficiency of the regenerator and economizer with respect to the lower calorific value LCV of natural gas. Data marked "measured cycle" have been calculated from heat and mass balances of the solution cycle. Data marked "measured flue" are calculated from flue gas temperature and cumbustion air-to gas ratio.

4.8.5 Plain tubes total heat transfer coefficient

As reported earlier, the convective heat transfer coefficient for the plain regenerator boiling tubes has always been calculated assuming that the actual flue temperature before the first tube row was slightly colder than indicated by the installed thermocouples. However, this assumption might not hold true and it is not out of the question that the actual temperature might be even higher then indicated by the installed measuring equipment (see section Radiation) .
Therefore it is not possible so far, to provide precise values for the convective heat transfer coefficient for each row. However, it is possible to give the total heat transfer coefficient of the plain regenerator boiling tubes (see next figure). These figures are as well a valid upper limit for the convective heat transfer coefficient under question.

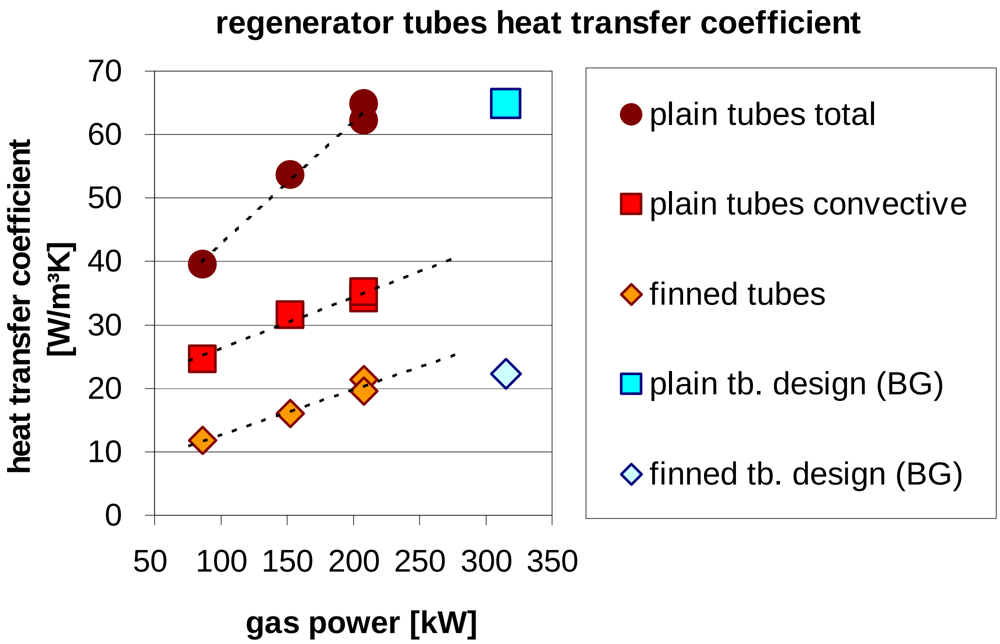


Fig. 18: Plain tubes total heat transfer coefficient in comparison to the earlier reported convective heat transfer coefficient at four operating conditions with different gas power (net calorific value).

Flue gas temperature measurements utilizing a suction thermocouple provided by BGT revealed clearly a non-negligible effect of the fraction of radiative heat transfer and the true convective heat transfer coefficient (see section Radiation). The data marked "plain tubes convective" have been calculated from raw thermoclouple readings and must be considered as non-relevant. The convective heat transfer coefficient is larger than 85% of the the total.

4.8.6 Correction of thermocouple readings

The calculation of heat transfer from the bare readings of temperature measured with thermocouples turned out to give largely false results.
Therefore a set of test measurements with a special suction thermocouple provided by BGT have been made in order to verify the gas temperature upfront the first row.
These measurement clearly confirmed a true gas temperature of approximately 1500 to 1550°C upfront the first tube row instead of 1100-1200 K with the bare thermocouple.
This is more than 300 K hotter than the reading of a bare thermocouple.
With a theoretical gas temperature of 1700 °C a radiative effect of less than 10% can be calculated from this.
This measurement confirmed precisely the predicted values of radiative effect on the first tubes.
The finned at not affected by radiation anymore.
Thus, excellent agreement between theoretical and empirical data could be achieved.

4.8.7 Thermal losses

4.8.7.1 Insulation losses calculation

A measurement of the regenerator casing temperatures as well as the insulation heat loss obtained from these data have already been reported earlier [*ZAE periodic report, January 2000*]. The calculation of the heat loss was based on approximate formulas [*RECKNAGEL 97/98, Oldenbourg Verlag, 1997*].
The total heat loss was estimated as sum of its convective (*"K"*) and radiation (*"S"*) part:

$$\dot{q} = (a_K + a_S) * Dt$$

where Dt [°C] is the temperature difference between surface and ambient
The coefficient for the convective heat loss was estimated by the approximate formula:

$$a_K = 1{,}6 * Dt^{0,3}$$ (approximation after Glück)

The radiation heat transfer coefficient was calculated by:

$$a_S = C * b$$

where b = $\frac{T_1^4 - T_2^4}{t_1 - t_2}$ [K³] is a temperature factor
and C [W/m²K^4] is the radiation coefficient
For a small surface compared to the surrounding surface an approximation can be given:

$$C = 5{,}67 * 10^{-8} * e_1$$

where ε is the emissivity of the surface.
For surfaces painted in blue an emissivity of 0,93 was assumed, for polished steel 0,7 was used. the temperature factor *b* was between $1{,}2*10^8$ and $1{,}5*10^8$ K³.
From the measured surface temperatures for operation at 160 kW gas power for the fire box region heat fluxes of about 2 KW/m² and for the remaining surface fluxes of about 0,5 kW/m² were obtained.
Total insulation losses of 5,6 kW (convenction + radiation) were calculated.

4.8.7.2 Operation as solution heater without boiling

Due to restrictions in the piping of the test rig, the mass flow meter in the condenser cooling water line had to be mounted at an unfavorable position. To ensure completely that the enthalpy balance is not affected by measuring errors concerning the condenser coolant flow, the regenerator was operated as solution heater without any boiling at 84 kW gas power. In this operating condition the condenser was out of operation and the corresponding coolant flow meter had no influence on the determination of the enthalpy balance.
For the operation of the regenerator without boiling at 84 kW gas power from the enthalpy balance thermal losses of 7,6 kW resp. 9,0% were obtained. These values are almost identical to the corresponding numbers of 7,8 kW resp. 9,1% thermal losses at the regenerator, which were obtained for normal operation at 86 kW gas power.

4.8.7.3 Operation without overpressure in the furnace

For the flue gas train of the regenerator no tightness can be specified and operating experience shows that there is always some flue leakage under normal operation conditions. However, it is not possible to determine the quantity of the flue mass loss and the corresponding enthalpy loss of the regenerator.
To check whether a significant fraction of the total energy input is lost by flue leakage a flue gas suction fan was installed and the test rig was run without any overpressure in the

combustion chamber. It was possible to achieve a gas power of 161 kW at a very small negative pressure of about 0,05 mbar in the furnace. For this operational state no flue loss nor any significant air intake have to be considered.
However, the evaluation of the enthalpy balance still revealed thermal losses at the regenerator of 15,5 kW resp. 9,7% of the energy input. These values are in the middle of those of many other similar operational states without suction fan (see fig. 7). Hence no influence of the combustion chamber overpressure on the thermal losses of the regenerator can be stated. Therefore it is not likely that there is a significant enthalpy loss due to flue leakage.

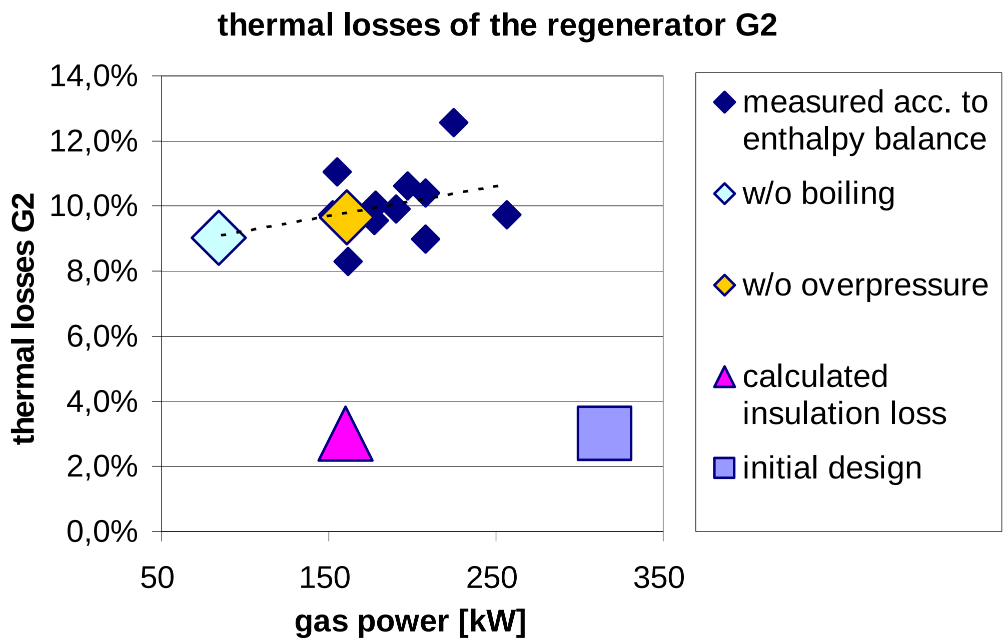


Fig. 19: Thermal losses of the regenerator according to the measured enthalpy balance in comparison to predicted values. Measurements at operating statuses without solution boiling and without overpressure in the combustion chamber are especially marked .

5 Comparison of initially planned activities and work actually accomplished

Initially planned activites	Actually performed activies
Desing of a Absorption chiller Cycle with higher COP State of the art : COP = 1,0 to 1,1 based on LCV	A modification of refrigeration cycle has been found which allow to increase the COP of a direct fired absorption chiller from 1,1 to almost 1,3
Cooling tower system which consumes 50% less water	A cooling tower study on the basis of a climatic study has shown that a properly designed cooling tower in conjunction with an absorption chiller operating a high cooling water temperature can save much more than 50% of the cooling water. Three factors are important to achieve this: 1. High efficiency chiller (=less waste heat) 2. increased cooling water temperature in part load 3. a well-designed combination of wet and dry part of the cooling tower
Cooling tower evalution	The new concept of cooling tower has been simulated for a number of different sites in Europe. The examples showed, that the differences a quite important not only in the different countries but also from building to building , or application respectiveley . Therefore a software tools has been developed in order to simulate various configurations for an application.
development of a new direct fired generator with high performance	A new generator with a completely new design has been tested successfully. The generator turned out to meet precisely the predicted performance during an extensive test campaign. Regulation and securities have been tested and improved and will be fuilly conformal to EN pressure equipment directive
Design of absorption chiller with 1 MW	The complete calculations for the design of an absorption chiller with 1 MW cooling capacity have been made before starting the construction of the generator and the test rig, in order to be sure that the chosen size an geometry is fully

	scaleable
Applicability of the design to cogeneration plants	The generator design can be adapted to operate on flue gas from gas engines instead of direct firing. Heat transfer surface must be increased, pressure drop requirements can be met by properr choice of lateral spacing and number of tubes per row.

6 Conclusions

The research work performed within the frame of this research contract was successful and the objective could be reached by a joint effort of all partners.
After critical review of the development work, it can be stated clearly, that the development of a new environmentally friendly absorption chiller technology has led to an almost immediately marketable product.
With the results of this research contract, the partners will be able to build a an absorption chiller with a number of new features:

1. very high COP based on lower calorific value of the gas
2. low water consumption due to high COP (less heat to be dissipated) and the new cooling tower configuration which , in combination with a new regulation scheme can operate as dry system for relatively long period of time throughout the year.
3. Due to low inventory of solution the chiller can respond quicker to load variations, this may also eliminate cold storage tanks etc.
4. The design of the generator is very flexible and can be adapted to a large number of installations with various heat sources, e.g. direct firing, use of flue gas form gas engines and gas turbines.
5. The design can be used in an industrial demonstrator plant immediately if the size is limited to approx. 1-2 MW in order to limit technological riscs when scaling up.

Some further development is needed however before a fully marketable product will be available:

1. Scale-up beyond 2 MW needs a demonstration unit since the natural convection of solution has to be proven for large arrays of tubes.
2. The newly developed generator has not yet been tested on other fuels e.g. fuel oil, biomass etc.
3. Large cogeneration plant usually operate on mixed fuel (like gas-diesel engines or gas turbines operating on natural gas or fuel oil in case of emergencies) . The generator has not yet been tested with the types of flue gases .

Market situation

The market situation of absorption cooling has completely changed this year due to relatively high prices of gas relative to electric energy. On the other hand cogeneration plant owners are more and more obliged to improve the economics of their plant by installation of absorption chillers, since these are in many cases the only available summer users of the heat. The market potential of high efficiency chillers connected to cogeneration plants seems to represent approx. 25% of all absorption chillers. In Germany this segment represents approx. 25 MW of installed capacity per year. For the instance the primary obstacle to market introduction is the lack of large scale demonstration plant (size at least 2 MW , preferrably larger) since the potential market is mainly concentrated on large installations.
A number of potential locations (mainly hospitals, airports) have been identified and discussions have been started with the clients.

7 List of publications

C. Schweigler, T. Dantele, C. Kren, F. Ziegler, D. Clark, M.-H. Fulachier, J. Sahun: Development of a direct-fired partly air-cooled double-effect absorption chiller. Proc. of the 20th International Congress of Refrigeration, Sydney, 19.-24. September 1999.

C. Kren, T. Dantele, C. Schweigler, F. Ziegler: Entwicklung einer zweistufigen gasgefeuerten Absorptionskälteanlage. Tagungsbericht der Deutschen Klima-Kälte-Tagung 17.-19. November 1999, Berlin. Deutscher Kälte- und -Klimatechnischer Verein, Stuttgart.

C. Schweigler, T. Dantele, C.Kren, F. Ziegler, D. Clark, M.-H. Fulachier, J. Sahun: Improvements in absorption Li/Br chiller for air conditioning. Proc. of the 21th World Gas Conference, Nice, 6.-9. June 2000.

to be published:
C.Kren, T.Dantele, C.Schweigler, F.Ziegler, J.Sadler, R.J.Tucker, C. Larger, J.Sahun: An efficient LiBr absorption chiller for the European air conditioning market. Proc. of the 2001 International Gas Research Conference, Amsterdam, 5.-8. November 2001.

DIPLOMA Thesis

Tobias Dantele: Entwicklung und Versuchsbetrieb eines gasgefeuerten Austreibers für eine zweistufige Wasser-Lithiumbromid-Absorptionskälteanlage. Physik Department der Technischen Universität München, Dezember 1999.

Other documents

ENTROPIE:
New Direct fired and waste heat fired high performance absorption chillers commercial circular on new developments to be distrubuted novmeber 2000