Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 82 (2015) 577 – 583

Energy Procedia

ATI 2015 - 70th Conference of the ATI Engineering Association

Modeling, Design and Construction of a Micro-scale Absorption Chiller

Giuseppe Franchini[a]*, Ettore Notarbartolo[a], Luca E. Padovan[a], Antonio Perdichizzi[a]

[a]*Department of Engineering and Applied Sciences, University of Bergamo viale Marconi 5, Dalmine (BG) 24044, Italy*

Abstract

In last decades, much effort has been made to drive cooling cycles exploiting renewable energy sources. The use of solar energy is one of the most attractive solutions especially for air conditioning, as availability of solar radiation and cooling loads are approximately in phase.
Solar cooling based on water-lithium bromide absorption chillers is the most promising technology for low-medium temperature heat sources (80-100°C). Small (15-40 kW cooling capacity), medium (50-250 kW) and large scale (up to tens of MW) units are currently at commercial stage. In the present work, the development of a novel micro-scale LiBr absorption chiller (around 5 kW) is presented. The objective is to demonstrate the technical feasibility and to investigate the performance under different operating conditions.
A computer code has been developed to simulate a LiBr absorption chiller. The model computes mass flow rates, temperatures, pressures and mass concentration of LiBr-water solution in all the chiller components, both in design and off-design conditions. Giving as inputs inlet temperature and mass flow rate of the external circuits (hot water source, cooling water and chilled water), the computer code is able to evaluate the efficiency (COP) and the actual cooling capacity.
The simulation code has been used to size the heat exchangers and to design a prototype of a micro-scale chiller. The chiller prototype with a 5 kW nominal cooling capacity has been manufactured and fully instrumented in order to monitor all physical quantities in the internal and external circuits. The prototype is installed on a test rig at the Energy System and Turbomachinery Laboratory of Bergamo University. Measurement devices, data acquisition system and in-house monitoring software are described in the paper. Preliminary results of the experimental investigation are presented.

Peer-review under responsibility of the Scientific Committee of ATI 2015

Keywords: Absorption chiller; solar cooling.

* Corresponding author. Tel.: +39-035-2052-078; fax:+39-035-2052-077.
E-mail address: giuseppe.franchini@unibg.it.

Peer-review under responsibility of the Scientific Committee of ATI 2015
doi:10.1016/j.egypro.2015.11.874

1. Introduction

The proliferation of energy consumption and carbon dioxide emissions in the built environment has made energy efficiency and savings strategies a priority objective for energy policies in most countries. In recent years particularly relevant has been the growth of energy consumption in HVAC systems, especially for cooling. Hence, the growing interest for the exploitation of renewable energy sources to meet the building thermal loads. Solar cooling is particularly promising to reduce electricity consumption in summer due to air conditioning, allowing to alleviate the problem of electrical brownout or power blackout due to peak loads in hot summer days. Closed-cycle thermally driven absorption chillers are a proven technology among the commercially available options [1,2]. In the last decade, many researchers investigated the mutual interaction of the main components in a solar cooling system (solar collector field, storage tank, absorption chiller): the analysis and optimization of single components and of the whole system are typically carried out with a simulation software able to predict the performance under variable operating conditions [3,4].

Nomenclature

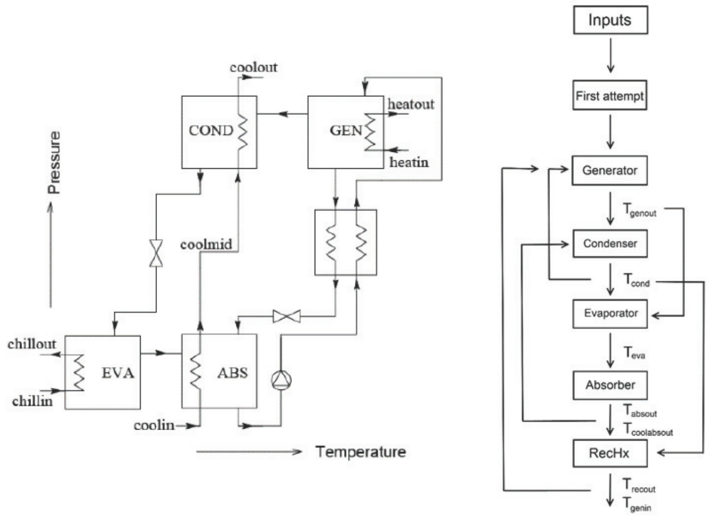
A, ABS	absorber; exchange area	out	outlet
C, COND	condenser	P	pressure (Pa)
C_p	specific heat (kJ/kg/K)	Q	heat transfer rate (kW)
chill	chilled water	R, rig, rec	recuperative heat exchanger
cool	cooling water	r	refrigerant
E, EVA	evaporator	s	strong solution
G, GEN	generator	T	temperature (°C)
in	inlet	U	heat transfer coefficient ($W/m^2/K$)
h	enthalpy (kJ/kg)	X	concentration
heat	hot water	w	weak/diluted solution
m	mass flow rate (kg/s)		

The most critical component is by far the absorption chiller. Therefore, a detailed and accurate modeling of the chiller, including the water-LiBr absorption cycle, is necessary to predict cooling energy production both in design and off-design conditions. A large number of papers in the open literature present computer codes for simulation of absorption refrigeration systems using LiBr-water as a working pair: most of them are steady state models [5,6]. Kohlenbach and Ziegler developed a dynamic model for a single-effect LiBr absorption chiller based on internal energy and mass balances. Dynamic behavior is implemented via thermal and mass storage terms as well as by delay times [7,8].

Absorption chillers are available on the market with different sizes, starting from 15-20 kW cooling capacity up to tens of MW. The size range from 3 to 10 kW is substantially unexplored, with very few exceptions. Only a few researchers addressed the development of novel prototypes of micro-scale absorption chillers [9]. The goal of the present work is the design and the construction of a LiBr absorption chiller for solar cooling applications with about 5 kW nominal cooling capacity.

2. Simulation code

First step was the development of a computer model able to accurately simulate the operation of a LiBr absorption chiller. The aim was to have a tool to correctly size all the heat exchangers, to define all the physical quantities (pressure and temperature levels, mass flow rates, solution concentration, heat transfer rates) across every component. Fig. 1a shows the considered scheme of the chiller: it is a single-stage hot water driven absorption unit, with a recuperative heat exchanger between generator and absorber and a single cooling circuit devoted to collect the heat rejected by absorber (firstly) and condenser (secondly).



$$
\begin{cases}
\dot{Q}_E = \dot{m}_{chill}\, C_{pH2O}\,(T_{chillin} - T_{chillout}) \\
\dot{Q}_E = \dot{m}_r\,(h_{evaout} - h_{evain}) \\
\dot{Q}_E = UA_E\,\Delta T_{LN,E} \\
\dot{Q}_A = \dot{m}_{cool}\, C_{pH2O}\,(T_{coolabsout} - T_{coolin}) \\
\dot{Q}_A = \dot{m}_r\, h_{evaout} + \dot{m}_s\, h_{rigout} - \dot{m}_w\, h_{absout} \\
\dot{Q}_A = UA_A\,\Delta T_{LN,A} \\
\dot{Q}_G = \dot{m}_{heat}\, C_{pH2O}\,(T_{heatin} - T_{heatout}) \\
\dot{Q}_G = \dot{m}_r\, h_{vapgenout} + \dot{m}_s\, h_{genout} - \dot{m}_w\, h_{genin} \\
\dot{Q}_G = UA_G\,\Delta T_{LN,G} \\
\dot{Q}_C = \dot{m}_{cool}\, C_{pH2O}\,(T_{coolout} - T_{coolabsout}) \\
\dot{Q}_C = \dot{m}_r\,(h_{vapgenout} - h_{evain}) \\
\dot{Q}_C = UA_C\,\Delta T_{LN,C} \\
\dot{Q}_R = \dot{m}_w\,(h_{genin} - h_{absout}) \\
\dot{Q}_R = \dot{m}_s\,(h_{genout} - h_{recout}) \\
\dot{Q}_R = UA_R\,\Delta T_{LN,R} \\
\dot{m}_w = \dot{m}_s + \dot{m}_r \\
X_s\,\dot{m}_s = X_w\,\dot{m}_w \\
\dot{Q}_A + \dot{Q}_C = \dot{Q}_E + \dot{Q}_G \\
X_s = f(P_{cond}, T_{genout}) \\
P_{cond} = f(T_{cond})
\end{cases}
$$

Fig. 1. (a) Schematic of a single stage LiBr chiller; (b) algorithm block diagram;(c)linear and nonlinear equations

The algorithm solves the mass and energy balance for all the heat exchangers and for the whole unit. Fig.1b and 1c show the algorithm block diagram and the system of equations that is solved.

The equilibrium of the LiBr-water solution imposes the relationship between pressure, solution temperature and concentration given by the Dühring chart. The algorithm reads as input the mass flow rate and the inlet temperature of the external circuits (hot water, cooling water and chilled water). The solution concentration (X_w) depending on the initial charging process and the weak solution mass flow rate depending on the solution pump are two additional inputs. An iterative procedure solves the equation system with the logic shown in the block diagram of Figure 1(c).

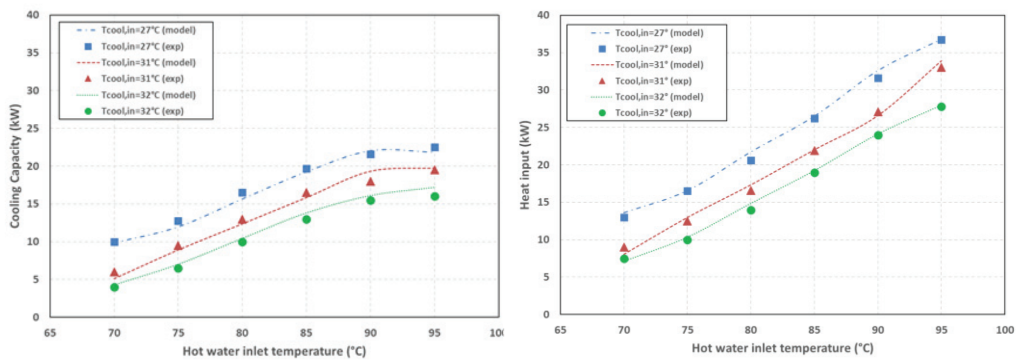


Fig. 2. Comparison model vs. manufacturer data: (a) cooling capacity; (b) heat input

Experimental data related to the hot water driven absorption chiller Yazaki WFC-SC5 have been used to validate the computer model. Figure 2 shows a comparison between data provided by the manufacturer and the model.The model exhibits a good performance prediction: the average deviation is 5.4% for the cooling capacity and 3.2% for the heat input.

3. Micro-scale absorption chiller prototype

The validated computer codewas used to design a prototype of micro-scale absorption machine. The chiller has been designed with the typical two-shell layout, where the upper cylinder includes generator, condenser and recuperative heat exchanger, whilst absorber, evaporator and pump chamber are located in the lower tank. All heat exchangers have a spiral configuration (generator and recuperative heat exchanger have two concentric spiral channels) and are copper made. The external envelop as well as the cylinder's caps are made of polypropylene. Heat exchange surfaces of every component have been determined based on the model prediction, in order to obtain a nominal cooling capacity of 5 kW at design conditions. Table 1 reports the main geometric characteristics of the 5 heat exchangers.

Table 1. Heat exchanger's geometric characteristics

	Absorber	Condenser	Evaporator	Generator	Recup. HX
Finned tube diameter (mm)			25.4 (outer) - 16.4 (inner)		
Heat exchange area (m^2)	5.8	5.8	4.54	3.17	1.79
Nr. of spirals	23	23	21	15 (x2)	8 (x2)
Spiral pitch (mm)	27.5	27.5	27.5	27.5	27.5
Spiral outer diameter (mm)	335.5	335.5	196	184 - 96	193.5-137.5

Figure 3 shows the predicted absorption cycle related to the micro-scale chiller on the PTX chart.

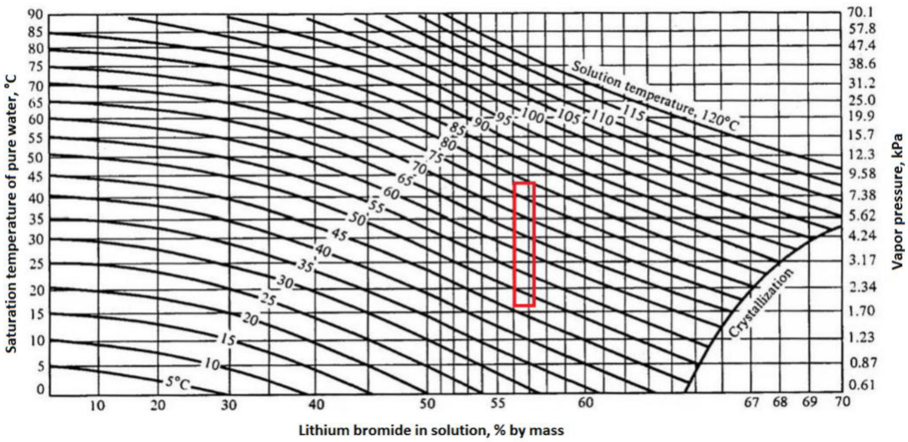


Fig. 3. Absorption cycle on the PTX chart (model prediction).

The prototype has been assembled and installed on a testing facility at Bergamo University Labs. Figure4a shows a 3D rendering of the system, where generator (red spirals) and condenser (green) are visible on the left, and evaporator (cyan) and absorber (yellow) on the right. A picture of the prototype on the test rigis shown in Figure 4b.

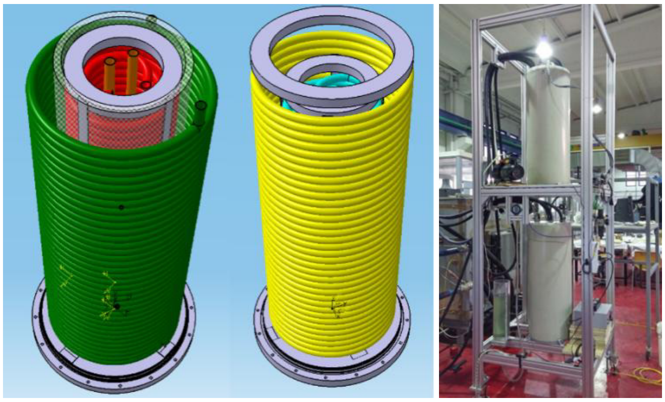


Fig. 4. (a) Virtual image and (b) picture of the micro-scale absorption chiller prototype

The prototype is fully instrumented in order to monitor all physical quantities in the external and internal circuits. The instrumentation devices are listed in Tab. 2 with their full scale and accuracy.

Table 2. Instrumentation devices

Measurement Instruments	Sensors	Range	Accuracy
Pressure transducer	Piezoresistive membrane	0-350mbar	± 0.2% F.S.
RTD	Platinum sensor	-50/+250°C	± 0.12% F.S.
Flowmeter	Ultrasonic sensor	0-0.06 l/min	<0.01% F.S.
Flowmeter	Turbine with optical sensor	0-10 m/s	± 0.50% F.S.

A multichannel data acquisition system acquires the input analog signals from the instrumentation devices and a LabView virtual instrument (see Figure 5) displays and records the test data.

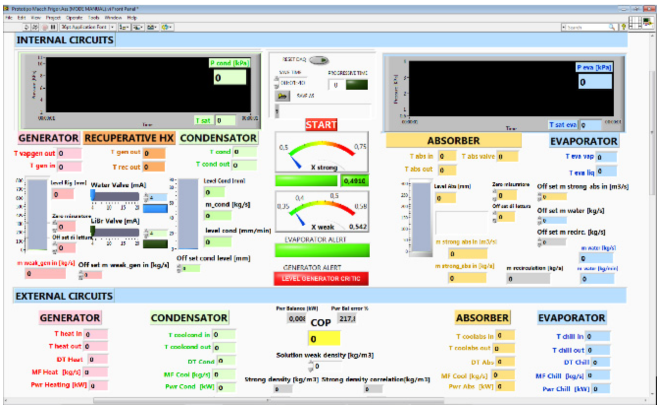


Fig. 5. Front panel of the LabView virtual instrument

The prototype is installed on a testing facility equipped to carry out experimental investigations under different operating conditions. Two temperature controllers with 70 kW total thermal power allow to heat waterup to 95°C and to regulate the water temperature entering the chiller's generator. A dry air cooler (60 kW cooling capacity) dissipates the heat rejected by condenser and absorber.

4. Preliminary results

First tests were devoted to check the leak tightness: absorber and evaporator operate at a nominal pressure of 1kPa, whilst the design pressure level in the generator and condenser is 9kPa.

After checking an adequate leak tightness, the water-LiBr mixture was charged and the measurement campaign for the performance evaluation has started. Current tests are focused on the chiller start-up and on the settings to stabilize the prototype operation. Table 3 summarizes the average values of the main monitored physical quantities in a preliminary test. In the prototype, the pressure gradient has been established between evaporator (1.8 kPa) and condenser (9.1 kPa) and an effective cooling capacity of 3.25 kW (corresponding to a COP of 0.358) was documented when the inlet hot water temperature was set to about 88°C. It has to be pointed out that the reported results are only related to the first measurement campaign, finalized to verify the start-up and the operational continuity. Results of the experimental investigation will be presented in next papers.

Table 3. Measured physical quantities (preliminary test)

Qg	9.08 kW	m,chill	0.2443 kg/s	m,cool	0.2988 kg/s
Qc	4.37 kW	Tchill,in	19.8°C	Tcond,in	21.9°C
Qe	3.25 kW	T,chill,out	16.6°C	Tcond,out	25.4°C
Qa	2.76 kW			Tabs,in	19.8°C
		m,heat	0.2780 kg/s	Tabs,out	22.0°C
COP	0.358	Theat,in	88.2°C		
		Theat,out	80.5°C		
m,s	0.03153 kg/s	Tcond	34.9°C	Tevap	8.5°C
m,w	0.07124 kg/s	Pcond	9.1 kPa	Pevap	1.8 kPa

5. Conclusions

A computer code has been developed to accurately simulate the behaviour of hot-water driven LiBr absorption chillers under variable operating conditions. The model predictions have been compared with experimental data provided by a manufacturer and a good agreement has been observed.

Then the validated code has been used to design a micro-scale absorption chiller. A fully instrumented prototype was assembled at Bergamo University Labs and currently is under experimental investigation. Preliminary results document the activation of the absorption cycle within the prototype. Further improvements to the operation stability and the overall efficiency are expected in the next future.

Acknowledgements

The authors acknowledge financial support from Regione Lombardia, which funded the present work as part of "RELAB-Renewable heating and cooling Labs" project.

References

[1] Grossman G. Solar-powered systems for cooling, dehumidification andair-conditioning. *Solar Energy* 2002;**72**:53–62.

[2] Fedrizzi R, Franchini G, Mugnier D, Melograno PN, Theofilidi M, Thuer A, Nienborg B, Koch L, Fernandez R, Troi A, Sparber W. Evaluation of Standard Solar Combi Plus Systems for Small Scale Applications. CISBAT 2009 International scientific Conference, Lausanne; September 2009.

[3] Assilzadeh F, Kalogirou SA, Ali Y, Sopian K. Simulation and optimization of a LiBrsolar absorption cooling system withevacuated tube collectors. *Renewable Energy* 2005,**30**:1143–1159.

[4] Giovanni N, Franchini G, Perdichizzi A. Combined solar heating and cooling systems: simulation and design optimization. Proceedings of ES2008, ASME International Solar Energy Division 2008, Jacksonville, FL, USA;August 10-14, 2008.

[5] Joudi KA, Lafta AH. Simulation of a simple absorption refrigeration system. *Energy Conversion and Management*2001;**42**:1575–1605.

[6] Somers C, Mortazavi A, Hwang Y, Radermacher R, Rodgers P, Al-Hashimi S. Modeling water/lithium bromide absorption chillers in ASPEN Plus. *Applied Energy*2011;**88**:4197–4205.

[7] Kohlenbach P, Ziegler F. A dynamic simulation model for transient absorptionchiller performance. Part I: The model. *Int. J. of Refrigeration*2008;**31**:217–225.

[8] Kohlenbach P, Ziegler F. A dynamic simulation model for transient absorptionchiller performance. Part II: Numerical results and experimental verification. *Int. J. of Refrigeration*2008;**31**:226–233.

[9] Ugwu HU. Design, Fabrication and Performance Evaluation of aMicro-Absorption Refrigerator. *Int. J. of Engineering and Technology*2012;Vol. 2, No. 11.

Biography

Prof. Giuseppe Franchini is currently Associate Professor at the Department of Engineering and Applied Sciences of Bergamo University, where he has been researcher for 9 years. He got his PhD at the Politecnico of Milan in 2005.He is author of about 60 papers of international relevance on the modeling of energy conversion systems.