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Preliminary results of a 7kW single-effect small capacity pre-industrial LiBr-H2O air-cooled absorption machine

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Abstract

The aim of the paper is to describe the thermal design of an own-developed pr-industrial absorption machine of small capacity, single effect configuration using LiBr-H_2O as working fluid and air-cooled, and then evaluating the preliminary operational results, i.e. the performance and the cooling capacity. This chiller is conceived for low temperature heat sources, e. g. , solar cooling or waste heat. A mathematical modelisation under transient condtions of the single-effect LiBr-H_2O - air-cooled absorption chiller has been developed. Thisis very valuable to predict the thermal behaviour of the absorption machine and their interaction with the other components and, therefore for defining its control strategy. Numerical results are obtained and the validation is performed against experimental data, due to there are dependences with some empiric parameters. The numerical model to simulate the thermal and fluid dynamical behaviour of the absorption machine has been implemented using the NEST platform (object-oriented numerical tool).

Keywords: LiBr-H_2O, air-cooled, absorption, single-effect, object-oriented, numerical simulation, transient

1. Introduction

In the last years there is a renewed interest of sorption systems due to the increasing price of the primary energy, which leads to the more efficient distributed model of energy production. In this distributed model, sorption systems could play an important role. Small capacity systems (less than 15 kW) could be an interesting option in the present situation. There have been many industrial developments in the last decade mainly in Europe, USA and China. However, up to now its implementation has been limited due to the initial high investment. One the main reasons could be the lack of standardisation, both in the components and systems. Based on these issues a 7 kW single-effect air-cooled absorption chiller has been thermally designed, mounted, and ready to be tested in Terrassa (Barcelona), Spain. By using the numerical developed platform tool several enhancements regarding the regulation and control of the device can be addressed.

2. Thermal design of the air-cooled absorption machine

2.1 Mathematical modeling

For performing the simulations, a modular object-oriented simulation platform is used (NEST), which allows the linking between the different components (solar collectors, pump, valves, heat exchangers, etc.) of each system. In this numerical platform each component is an object, which can be either an empirical-based model (e. g. heat exchangers, solar collector) or a more detailed CFD calculation if necessary. With this numerical platform, parallel computing is allowed.

A lumped parametric dynamic model based on mass, momentum and energy balances and applied to internal components of the absorption machine (absorber, generator, condenser, evaporator and solution heat

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exchanger) has been implemented to investigate on transient behaviour of the air-cooled absorption machine (Evola et al, 2013), (Kohlenbach P. and Ziegler F, 2008) (Farnós et al, 2014).

Moreover, thermal and mass storage in each one of the components are considered in the transient evaluation. The pressure losses in the solution heat exchanger are evaluated by means of a resistance coefficient, and the analysis of part-load or activation/deactivation transient operation procedures can be carried out. Thus, the transient simulation of the absorption chiller is useful to obtain realistic information on fast transient procedures, i.e. switch-on, switch off, by-pass or failure operations.

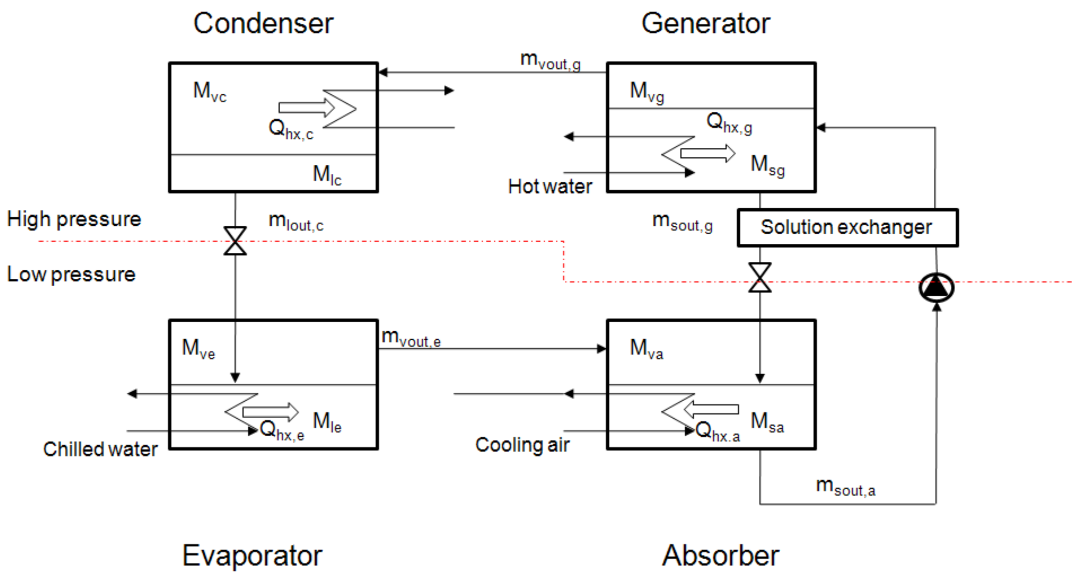

Fig. 1 LiBr-H_2O single effect absorption cycle

Some assumptions are considered for the formulation of the process: i) temperatures, pressures and concentrations are homogeneous inside each component, ii) pressure inside the generator is equal to the one in the condenser, iii) fluid transport delay between two components is neglected, iv) each heat exchanger has a constant overall heat transfer coefficient (UA), v) throttling valves are adiabatic, vi) vapour produced in the evaporator is saturated, vii) volumetric flow conveyed by the solution pump is assumed constant and, viii) mass transfer coefficients ($k\cdot A$) are recalculated according to the conditions of the different sources.

The assumption of not considering vapor mass accumulation on vessels is in accordance with numerical results obtained from other research (Köhlenbach and Ziegler, 2008) (Farnós et al, 2014). In the present work, the equation system is closed using $\dot{m} = k \cdot A \cdot C_{lm}$ in order to obtain absorbed and desorbed mass; where k (mass transfer coefficient) is difficult to obtain accurately.

Preliminary numerical results have been obtained and they have been validated with the numerical results of a case proposed by Evola et al. (2013) and with experimental data obtained by Castro et. al. (2008) (see section 4.1). Temperatures, pressures, concentrations, mass fluxes and power of the most significant points of the whole absorption machine can be obtained.

2.2 Numerical implementation

Regarding numerical simulation, an adaptable in-time Runge-Kutta method has been implemented in order to optimize the time step without reducing accuracy in an explicit method. The explicit method is capable to adapt its time step depending on the convergence of the components (a numerical stability analysis has been done), and on the pseudo-stationary performance of the machine. Therefore, it is important to simulate accurately regulation and control periods when transient phenomena are very relevant. In those cases, the time step can be reduced up to $1\cdot10^{-5}$ seconds, while in a stabilized mode time steps can be increased until 0.1 seconds without losing accuracy.

The concentrated solution rate is obtained by adapting the expression described by Evola et al. (2013) or Köhlenbach and Ziegler (2008) in order to overcome the possibility of emptying the desorber or the condenser. This expression is highly dependent on the ζ which value is a function of piping and solution heat exchanger. Therefore, it is important to obtain manufacturing characteristics. Moreover, ζ is recalculated at each time step depending on the total amount of mass substance at the vessels, which is permanently controlled in order to avoid emptied vessels and to analyze the thermal inertia of accumulated terms. On the other hand, volumetric flow is set constant imposed according to solution pump characteristics. Finally, according to Kohlenbach and Ziegler (2008a), Kohlenbach and Ziegler (2008b), and Yeung et al. (1992) the fouling factor is also considered in the UA_{int} and UA_{ext} expression.

3. Technical Characteristics of the air-cooled absorption machine

There are two types of heat exchangers: i) serpentine tube bundle: generator and evaporator; ii) fin and tube: absorber and condenser. Absorption and desorption are numerically modeled taking account heat and mass transfer in absorption phenomena (Castro et al, 2008). Evaporation and condensation phenomena are calculated using the correlations provided in the literature by Schnabel and Wiegand (2007). In the case of small capacity absorption chillers, flooded evaporators configuration is a very interesting option to remove the refrigerant pumpand therefore decrease the electrical expense. However, their design implies low heat transfer coefficients with respect falling film evaporators because normally they do not work under boiling conditions, but under natural convection regime, due to the small temperature differences between the primary and secondary streams (Castro et al, 2008) .

Table 1. Equipment specifications and main results of the cycle simulation for the nominal operation point

	Units		Specifications	
Nominal capacity	kW		7.0	
COP	-		0.7	
Dimensions	WDH, mm		800-800-2100	
Volume	m^3		1.5	
Electricity consumption	kW		0.37 (fan)+0.25 (pump)	
Hot water stream	ºC	kg/h	88	1730
Cooling air stream	ºC	m^3/h	35	16500
Chilled water stream	ºC *	kg/h	9	1200

Key Issues of the chiller: *Flooded evaporators, Arrangement of heat exchangers, New heat exchangers, Valves under vacuum conditions, reduction of NPSHr, internal corrosion analysis in deep, heat and mass transfer coefficients, and % of air permitted assuring a good heat and mass transfer.*

4. Experimental validation of the LiBr-H_2O small capacity air-cooled absorption machine

Experimental data have been obtained from Castro et al., 2008. In order to validate the numerical simulation for this device, authors have considered this 2 kW air-cooled machine where it is possible to understand and design strategies of regulation and control. This chiller has been simulated in a transient mode, and was tested at the laboratory of the Heat and Mass Transfer Technological Center at the Polytechnical University of Catalonia (UPC) in Terrassa (Barcelona), Spain. The research has been mainly focused on avoiding crystallization phenomena, avoiding freezing and geometrical aspects related in the simulation.

4.1 Experimental validation of a 2 kWc air-cooled absorption machine

Regarding the original configuration of the 2kW air-cooled absorption cycle, a liquid-vapour separator tank between condenser and evaporator, which has been considered adiabatic on the numerical modeling, is integrated in the cycle. This is due to the falling film configuration of the evaporator. Therefore a constant inlet mass flow is fixed.

The stabilization of the performance of the chiller has been observed at around 1200 seconds after switching-on the refrigerating system. Moreover, It can be assumed a stationary operational point of the chiller when $\dot{m}_{abs}$ has a very closed value of $\dot{m}_{des}$. The outlet temperature of the air at the absorber is the inlet at the condenser.

Table 2. Main input parameters for the simulation of a 2kW LiBr-Water air-cooled absorption machine

	UA_{int}[W/K]	UA_{ext}[W/K]
Evaporator	6595	1373
Absorber	1553	1494
Desorber	1566	5200
Condenser	4495	967
Solution Heat Exchanger	575	

Next it is shown a comparison between numerical and experimental data in order to validate the mathematical model. As it can be observed, numerical results overestimate the cooling capacity of the chiller. It can be explained by the $(UA)_e$ where its dependence of semi-empirical information on the heat and mass transfer coefficients brings to a range of working values. As the numerical modelisation needs a fixed value, (UA) has been implemented on the code as fixed input data. At the same time, there are uncertainties regarding the wettability of the evaporator. Therefore, $(UA)_e$ should be numerically reduced in order to assure a better approximation from experimental data. It is needed an strong effort to obtain semi-empirical values which are constant along the numerical simulation. They need to be as accurate as possible.

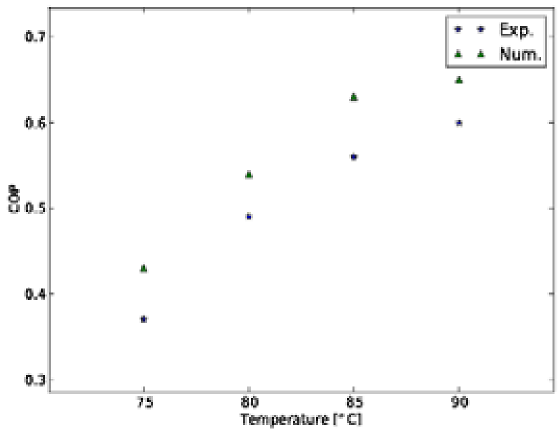

Fig. 2i. Experimental obtained COP Vs numerical calculated COP calculated

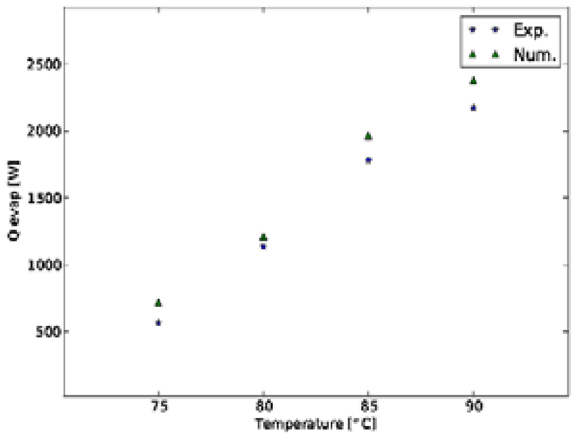

Fig. 2ii. Experimental cooling capacity Vs numerical cooling capacity

4.2 Numerical results of a 7kWc air-cooled absorption machine

In the case of the 7 kW chiller, instead of fixing the inlet values of temperature and mass flows, a virtual solar cooling facility has been implemented in a numerical platform. Therefore, the evaporator and generator inlet temperatures are determined by two tanks which are used as a buffer in order to avoid oscillations on the driving temperature. Temperatures inside the tanks are controlled so that the high temperature circuit (solar circuit) is activated only when is required (below 80ºC). Hysteresis of the tanks is considered when the high temperature source is activated or deactivated. Moreover, there is an auxiliary circuit if solar circuit is not available to overcome heating demand. The low temperature circuit is controlled in order to avoid freezing at the chiller (T_e<1.5ºC).

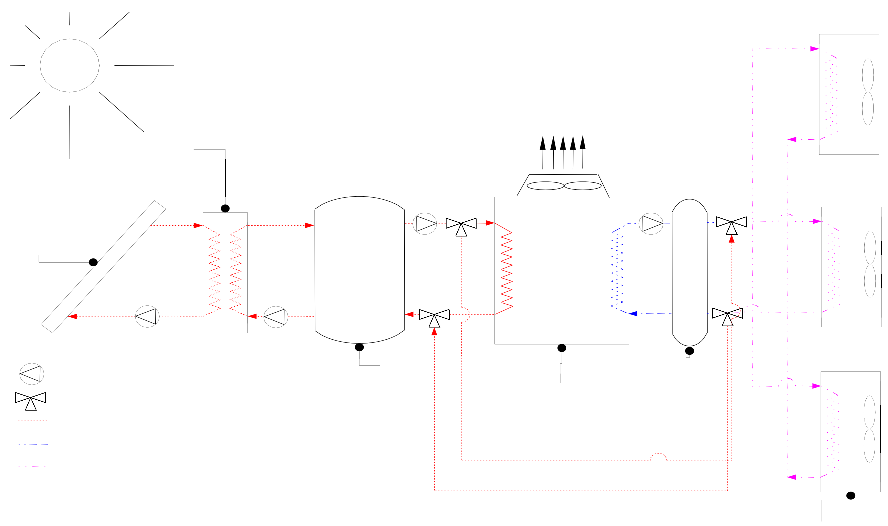

Fig. 3. Solar cooling facility

Once the mathematical modeling has been validated with experimental data from an air-cooled absorption chiller, an example of the necessity of a good control and regulation strategy of an absorption machine is demonstrated below. Based on the numerical simulation, the performance of an air-cooled LiBr-H_2Or absorption machine of 7 kW of cooling capacity is clearly affected by demand.

Table 3. Thermal design of a 7kW LiBr-Water direct air-cooled absorption machine

	UA_{int} [W/K]	UA_{ext}[W/K]
Evaporator	2675	14310
Absorber	4262	3539
Desorber	3125	12866
Condenser	3189	1885
Solution Heat Exchanger	1600	

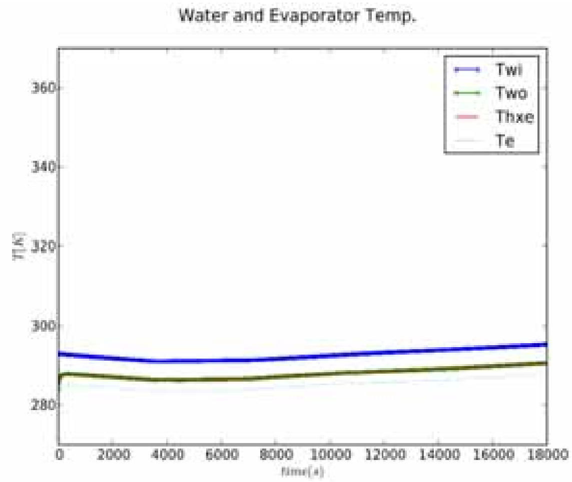

Fig. 4i)Evaporator performancewith high cooling demand

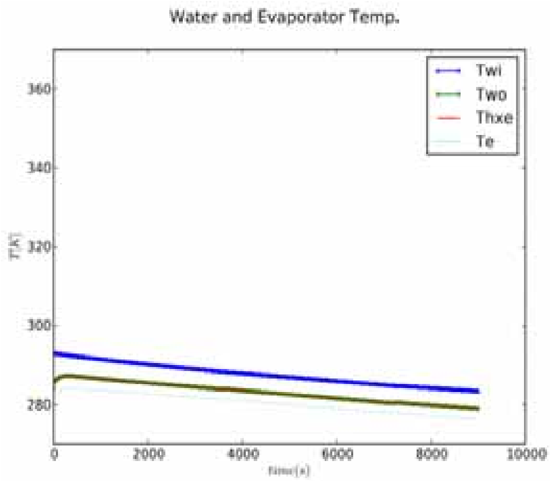

Fig. 4ii)Evaporator performance with low cooling demand

Fig. 4 Evaporator performance in a Mediterranean climate
(Te=Te,inside vessel, Twi=Tw,i secondary flow, Two=Tw,o secondary flow, Thxe =Thx,e)

As it is described on the figures 4i and 4ii, on the same characteristic day from July in Barcelona (Spain, Mediterranean climate), from 9 am to 11:30 am, it is shown that there is not enough cooling demand to overcome the delivering cooling power of the machine. In that case, a cold tank, which acts as a buffer, modulates this issue, but freezing control must be activated. If not, as it is shown in Fig. 4 ii) the temperature at the vessel of the evaporador decreases until freezing appears, In that case, strategies as flow modulation of secondaries or by-pass activation are reliable to overcome operation failures. If control is activated, i.e. by pass from desorber to absorber, computational costs are increased due to convergence of parameters bring the numerical simulation to use time steps of 10^{-5} seconds.

The data obtained from numerical simulation is used to manage the regulation and based in a transient analysis (Mirzaei et al, 2011), taking into account several heat and mass transfer phenomena like crystallization, inertia, etc., aiming to assure a good control strategy on the start-up, shut-down, nominal conditions, etc. With this valuable information different control strategies are considered to be used using industrial devices such as PLC, EPC, etc.

The experimental validation of the 7 kW machine is currently being carried and first operational results may validate numerical data obtained by means of thermodynamic modeling. The experimental facility allows to simulate several service conditions by varying the parameters of internal and external circuits, as the driving temperature, pressure and flow rate, all abled to be recorded in real time and obtaining a first experience in real-life. Finally, the experimental facility monitors the performance focusing on heat rejection temperature dependency and crystallization phenomena.

4.3 Control strategies

Internal and external controls must be focused on the operation of air-cooled absorption chillers. In this work only internal chiller processes are addressed by reviewing several control strategies present at the scientific literature and also from the industry. External control deals with external parameters of the chiller, such as the ones concerning low, medium and hot circuitry. Controlling hot and cooling water is one of the most promising approaches but in air-cooled machines is highly complex, at least at the medium focus. Even though, an optimal control of the chiller concerns on defining a good strategy for the whole system,

including heating and cooling components (hot and cold storage, solar circuit, etc.). The principal controlled parameters of a solar cooling system may be hot water inlet temperature and mass flow, cooling air or water inlet temperature and mass flow and chilled water outlet temperature and mass flow. Chilled water must be kept in certain limits to allow satisfactory operation (Kohlenbach and Ziegler, 2008b). At the same time differential on/off operation of the external circuits should be avoided (Yeung et al, 1992; Bong et al, 1987; Kohlenbach, 2008; Labus, 2012).
The air-cooled absorption chillers are attractive due to no use of cooling tower and associated installation is needed. However, the key technical barrier to operate an air-cooled absorption chiller using LiBr-Water as working fluid is crystallization because of high temperatures achieved inside the absorber which derives in a concentrated solution. It is the main difference towards water cooled absorption chillers. As air is used as coolant, low heat transfer characteristics must be taken into consideration because operational concentrations of the solution are very close to the crystallization limit. Therefore, not always the evaporator temperature can be maintained as low as it would be necessary to overcome the absorption pressure increase. LiBr begins to crystallize when the temperature solution is reduced under the crystallization limit or when concentration ratio is increased. A numerical control strategy has been implemented considering a wide variety of causes which may interrupt the operation.

On the other hand, an interesting control to regulate the cooling capacity of the chiller is by varying mass flows at the medium temperature source. It brings to deal with a high variation of the mass transfer coefficient, which may be recalculated for each performance condition. This control is high complicate to simulate in terms of mass transfer coefficients available data, Nevertheless, the electrical COP would be increased and, therefore, energy savings would be greater. Several regulation strategies can be planned with the aim of electricity consumption reduction.An other approach to regulate the cooling capacity may be considering the variation of the mass flow of the high temperature source. Cooling capacity may be affected.

When high condensation temperature is required due to high ambient temperature (Florides, 2003) (Izquierdo et al, 2004), high temperature at the desorber is needed, with a possible overloading power to the desorber. Therefore, a high concentrated solution has to return to the absorber through the solution heat exchanger. On the other hand, for condensation temperatures over 40ºC, single-stage air-cooled absorption chillers can not be operated by single glaze flat plate collectors(McNeely, 1979) (Izquierdo et al, 2004).

Other causes which allow the creation of slush are low ambient temperature and full load (Florides et al., 2003), electrical failure (shutdown dilution process) and when the chilled temperature is set too low (Liao and Radermacher, 2007). The presence of non-absorbable gases (air and hydrogen) which may vary according to the power input at the desorber (Liao and Radermacher, 2007) and their effect on the performance of the absorption process phenomena are studied at a basic research level (Garcia-Rivera et al., 2012) and will be discussed in future works on the performance of the air-cooled absorption prototype.

Moreover, several approaches can be found on the scientific literature as self-decrystallisation technique (DeVuono et al, 1992) by driving high temperature secondary fluid through the solution heat exchanger. It is only possible if high temperature drives the desorption process. Other techniques as controlling the absorbed mass by monitoring the fluid level at the evaporator or boosting the absorber pressure have also been studied by different authors (Zogg et al, 2005) (Xie et al, 2012). These approaches and an overview of control strategies can be found at Labus et al. (2012), where a new Artificial Neural Network (ANN) modeling and optimization approach is described. Another focus is to define a control strategy based on minimizing specific costs or the price for generation of cold (Albers, 2013).

Finally a J-tube technology or by-passing the solution from the desorber to the absorber in order to avoid crystallization is widely applied on the industry (Johnson Control, 1997) and has been numerical simulated by the authors of the present work to prevent crystallization on the operational mode of the system. When crystallization phenomena appears, a connection between the desorber and the absorber, which by-passes the solution heat exchanger, is opened. Therefore, immediately the concentrated solution is diluted and its temperature decreases, warming the low concentrated solution. At the same time, the mass flow over the solution heat exchanger will allow to avoid crystallization at the concentrated solution which is at low temperature.

Summarizing, regulation and control of the cycle is quiet challenging due to crystallization phenomena and environmental temperature dependence. In fact, mass flow driven from the generator to the absorber depends on the pressure gradient, which at the same time depends on the geometrical configuration. Therefore, it is important the mechanical design in order to define correctly all the input parameters at the numerical simulation. The transient numerical modelling developed will help to define the optimum control strategy.

5. Conclusions

A mathematical model has been used to obtain preliminary results of a small capacity pre-industrial LiBr-H_2O air-cooled absorption machine, with the aim of defining a control strategy to avoid undesirable working conditions. Two different air-cooled absorption machines have been integrated in a virtual laboratory and all experimental data show that the machines are far from crystallization. However, there is a strong dependence on the mass transfer coefficient, which may induct the desorber into a crystallization process and to an overload at the evaporator. This can be assumed constant for a pseudo-stationary case, but in a transient mode where control strategies are taken into consideration, must be calculated according to the specific conditions where the chiller is working each time step. At the same time, ζ must be defined accurately in order to control the circulated solution and refrigerant and to avoid emptying vessels, which could interrupt the operation of the machine. In order to avoid this phenomenon, different strategies are considered, as the control of mass substance inside vessels (J-tube), freezing especially when the machine is operated with low cooling demand, and the integration of a decrystallisation line for example (by-pass), etc.

A good agreement between numerical results and experimental data of a 2kW single-effect LiBr-Water air-cooled machine is achieved. Discrepancies are small and can be explained as consequence of the mathematical model dependence on semi-empirical information. Therefore, the numerical simulation of a 7kW single-effect LiBr-Water air-cooled machine has been carried out with the implemented and validated model. Next steps are to obtain experimental data of a 7 kW air-cooled machine to compare and validate numerical results of the model. Some uncertainties, as mass transfer coefficient or an accurate value of ζ, which determines mass flow of the thermochemical compressor, may be defined accurately.

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NOMENCLATURE

Variables

$\dot{Q}$	thermal flow,W	T	temperature, K
$\dot{m}$	mass flow rate, kg/s	M	mass, kg
COP	Thermal Coefficient of Performance	s	second
NPSH	Net Power Suction Heat		

Greek letters

ζ	pressure loss coefficient (-)

Subscripts and superscripts

abs	absorbed	des	desorbed
int	internal	a	absorber
g	generator	ext	external
e	evaporator	v	vapour / gas
l	liquid	c	condenser
s	solution	i	inlet
o	outlet	w	water
num	numerical	exp	experimental
hx	heat exchanger	d	losses
lm	logarithmic mean	env	environmental
r	required		

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