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DESIGN AND CONSTRUCTION OF A COMPACT AIR-COOLED ABSORPTION MACHINE FOR SOLAR ENERGY APPLICATIONS

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ABSTRACT
This paper presents a methodical procedure for the design and sizing of a compact, water fired, air- cooled absorption chiller. The proposed compact machine uses Lithium-Bromide/Water as absorbent/refrigerant fluid pair. The machine was designed using a detailed heat transfer analysis for each individual component (i.e. desorber, condenser, absorber and evaporator). The condenser uses conventional fin-tube heat transfer surfaces while the evaporator uses an inner corrugated surface to increase the heat transfer area. The generator has a concentric tube arrangement in which the dilute solution is in the inner section and the heating water flows in the outside section. This arrangement results in a regeneration effect at temperatures close to 75ºC which can be easily provided with solar collectors. The absorber and evaporator work together as a single unit. The vapor exiting the evaporator comes into thermal contact with the concentrated LiBr solution that enters the absorber from the top and falls inside the vertical tubes, creating the absorption effect. Moreover, air is cooling the outside surface of the tubes removing the heat released during the absorption process. The evaporator was designed to be a falling film evaporator such that when applying the cooling load, condensed water falling on the evaporator tubes evaporates and rises through the vertical tubes of the absorber. A set of highly efficient fans are used to bring outside air to remove the necessary heat in both the absorber and condenser, respectively.

Furthermore, all system components have been constructed and assembled into a working prototype of variable cooling capacity between 10.5 to 17.5 kW having final dimensions equivalent to a volume of 5 m^3. The preliminary characterization of the thermal performance of this prototype is presented in the paper with the objective of validating the design methodology.

Keywords: absorption, air cooled, thermal performance.

INTRODUCTION
In the Caribbean one of major areas of energy consumption is room comfort for commercial, industrial and residential applications. Due to the hot and humid conditions that are characteristics of the region, room comfort has become a need instead of a commodity. In a study conducted in Florida it showed that for hot and humid climates about 40% of household energy consumption is used in room space conditioning [1]. Statistics from planning board in Puerto Rico shows about 27,000 new units households were constructed in the year 2000 alone [2]. The demand of energy for residential purposes is high and increasing and therefore there exists a very high priority to attend this need.

The use of solar driven absorption machines has been suggested during the last few years to offset the energy consumption in the Caribbean for air conditioning applications [3-4]. The experimental results for a 35 kW closed absorption system installed in the city of Cabo Rojo, Puerto Rico, showed that such systems are technically feasible [3] for medium cooling loads. A market study to define the market potential in Puerto Rico was conducted to define market of small businesses that potentially can use this technology, taking into account space requirements of the equipment. The study determined there was potential in the small business sector, but suggested that residential sector is also very attractive market [4]. The price targeted in this study was $1200 to 1400 per KW installed ($4K to $5K per ton) for an air conditioning functioning eight hours per day resulting in a pay back of about 5 years.

To address both the needs of energy savings and residential applications it is proposed in this work compact air cooled absorption chiller. The proposed machine will target small cooling loads within the range of 10.5 to 17.5 KW and will be air cooled without significantly affecting the heat exchangers sizes but eliminating the use of cooling tower. The absorption machine will be single-effect having regeneration temperatures in the order of 75°C which make it feasible to be powered by conventional flat plate collectors.

During the past years air-cooled absorption chillers have been investigated from a mathematical and theoretical point of view [5-7] as possible alternatives to scale down existing absorption machines. In these air-cooled closed absorption conditioning conceptual models, fans replace the cooling tower. A prototyping of such compact systems have not been reported yet. In this paper we report results from a design exercise for an air-cooled absorption machine. Specific knowledge of functionality and validity of this model can be found in previous works [5,8]. Our purpose is to consider matters of components or system design, along with the fabrication and assembling of machine prototype.

THE ABSORPTION CYCLE

Fig. 1 below shows the conceptual schematic of an air-cooled absorption machine using lithium-bromide/water as solution pair. As can be seen in the schematic, the main components of the absorption machine are: desorber (generator), condenser, evaporator, a solution heat exchanger and the absorber.

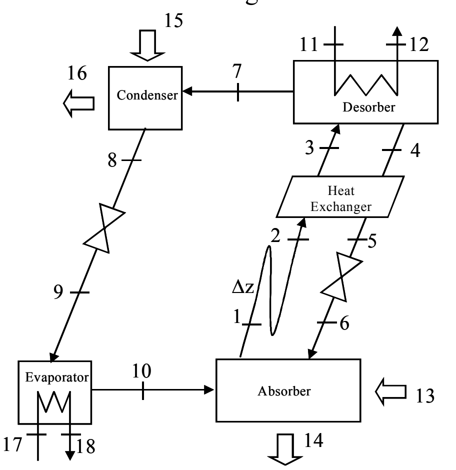

Figure 1- Conceptual air-cooled absorption machine

The process starts when the hot water entering the desorber (coming from solar collectors hot water circuit), induces a violent boiling process within the desorber. The boiling takes part of the lithium bromide solution out of the desorber. Furthermore, part of the water escapes as vapor into the condenser and the remaining concentrated solution heads to the absorber. In the condenser, the vapor is condensed to saturated liquid and further expands decreasing the temperature as it goes through the expansion device. This sub-cooled liquid is finally taken to the evaporator where the cooling effect is obtained and the liquid goes to a superheated water vapor state. The absorption of the water vapor by the aqueous salt finally takes place as the superheated water vapor and the concentrated solution come together to force the absorption of water vapor into the solution. Simultaneously the heat released to absorb the vapor is removed by air. The resultant diluted solution then goes back into the desorber.

NOMENCLATURE

C_1, C_2 – constants given in plate correlation
h_m – average heat transfer coefficient, W/m^2K
Ra_s – average rayleigh number
Nu - average nusselt number
S – spacing between parallel plates, m
T – temperature, K
Greek Symbols:
α – thermal diffusivity, m^2/s
β - volumetric expansion coefficient, K^{-1}
ν – kinematic viscocity, m^2/s
ρ – density, kg/m^3
Sub-indices
g – gas
l – liquid
m – mean
s – surface of plate
∞ - surroundings

DESIGN STRATEGY AND CONSTRUCTION

At the early stages of the project the technical and economical feasibilities of the proposed air cooled absorption machine had been completed. The entire thermodynamic cycle was modeled in EES software [9], as shown in Fig. 2.

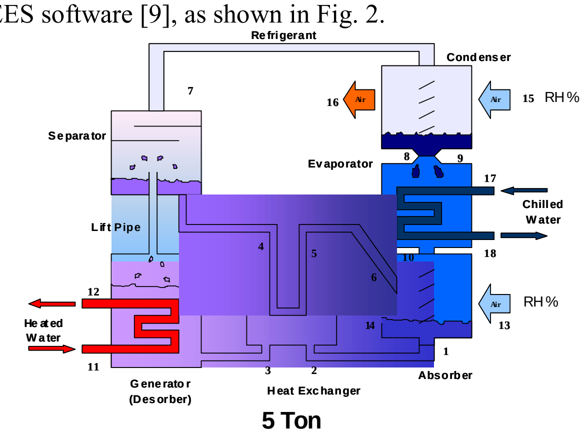

Figure 2- EES simulation of absorption cycle

The EES software outputs the properties of every state shown in Fig. 2, and this data was the basis of our design calculations (i.e. heat transfer). Table 1 shows the temperatures, pressures and concentrations of the LiBr solution at the different states.

Table 1- Table of thermo properties in absorption machine

States	m_i [kg/sec]	P [Pa]	T[C]	Xi [%LiBr]
1	0.0816	1.147	38.3	55.1
2	0.0816	6.645	40	55.1
3	0.0816	6.645	50.6	55.1
4	0.0742	6.645	83	60.6
5	0.0742	6.645	70.4	60.6
6	0.0742	1.147	50.8	60.6
7	0.0074	6.645	71.3	0
8	0.0074	6.645	38.1	0
9	0.0074	1.147	9	0
10	0.0074	1.147	9	0
11	1.2	--	85	--
12	1.2	--	80	--
13	2	--	29	--
14	2	--	40.8	--
15	2	--	29	--
16	2	--	37.9	--
17	0.8	--	15.2	--
18	0.8	--	10	--

In order to complete the design, component fabrication, and subsequent testing a general strategy had to be developed. Fig. 3 shows a flow chart implemented in order to complete full prototype development

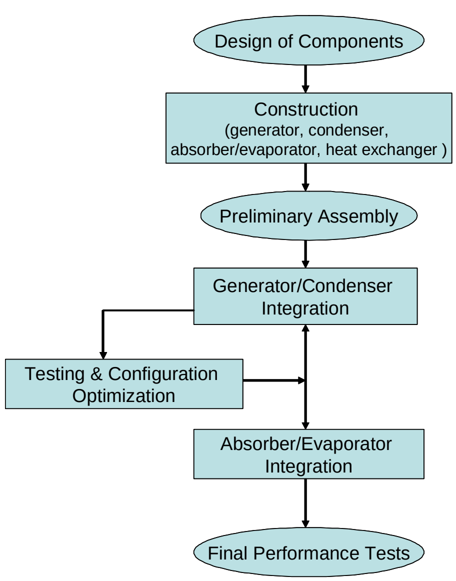

Figure 3- Flow chart of Prototype Development

GENERATOR DESIGN

From a simulation exercise, the quantity of heat transfer in the generator which would provide the amount of water vapor flux required was obtained. In the 17.5 kW absorption machine, for example, the heat transfer necessary was computed to be 25.2 kW. The initial proposal suggested the use of a heliflow coil for the heat transfer process. This idea was soon rejected, as calculation of the size of coil and price would be outrageous. A closer study demonstrated that in the desorber shown below the boiling phenomenon was both a function of heat transfer, velocity and ability of vapor bubbles to drag the solution out of desorber.

To design the generator a parallel plate channel model of flow between plates model was used [10]. The boundary conditions applied were the isothermal parallel plate conditions, for which the corresponding Nusselt and Raleigh number equation are:

$$Nu = \left[\frac{C_1}{\left(Ra_s \cdot S/L\right)^2} + \frac{C_2}{\left(Ra_s \cdot S/L\right)^{1/2}} \right]^{1/2} \qquad (1)$$

$$Ra_s = \frac{g \cdot \beta (T_s - T_\infty) S^3}{\alpha \upsilon} \qquad (2)$$

Fig. 4 shows the final isothermal schematic of the generator based on the model shown above (for isothermal conditions C_1=756 and C_2=2.87). The idea underlining the selection of this model is based on the concept of confining the lithium bromide solution among the walls of concentric cylinders. The cylinder wall is used as the heat transfer surface, and for compactness, water runs through the inner and outer walls of concentric cylinders that contain lithium bromide solution. This geometry provides many advantages, providing wall surface for which solution can climb, due to the shear force acting upward during the boiling process. The diameter is based on the heat transfer necessary in the generator and the velocity of the water vapor at the generator exit.

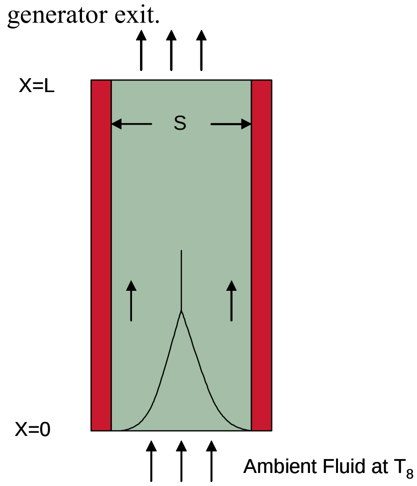

Figure 4- parallel plate model use to model generator

Fig. 5 shows the velocities corresponding to given heat inputs to the generator while Fig. 6 shows the internal diameter as function of this velocity. These graphs take into account a spacing of 1.27 cm in between the concentric cylinders that contain the lithium bromide solution. The generator is the component with the greatest unknown set of variables in the absorption machine. The estimates for diameters of concentric cylinders were based on a design velocity of 22 m/s. This parameter was chosen to meet the minimum velocity of refrigerant in vertical tubes of 4.9 m/s [10], which provided a diameter that was reasonable to construct.

The generator has already been constructed using the described design criterions. Experimentation will allow to clear doubts and validate the design process. There has been extensive testing of the generator coupled to the condenser at this stage and results will be presented in the next sections.

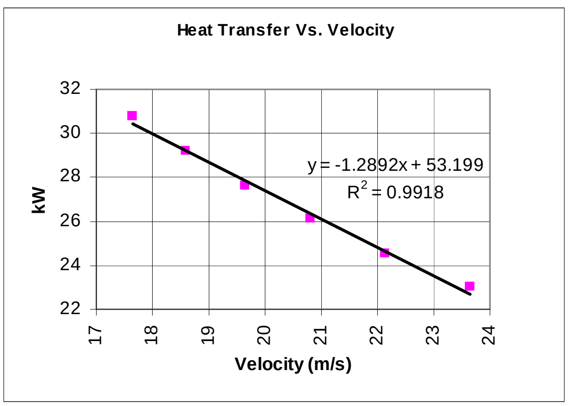

Figure 5- The Velocity estimated for given heat input into the generator

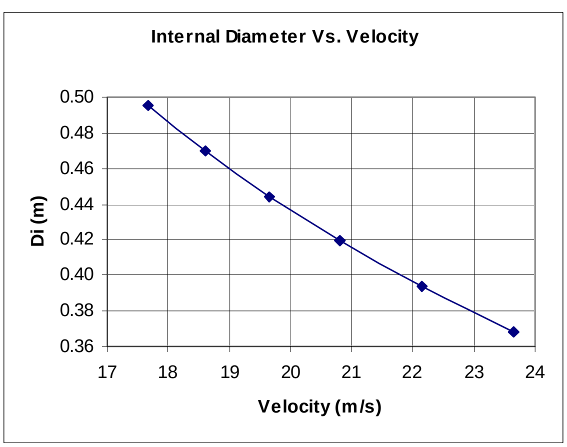

Figure 6- The corresponding inner generator sizing for a given velocity

DESIGN OF CONDENSER

In order to design the condenser, inlet and outlet vapor conditions were specified. Using a computational simulation program for thermodynamic properties [8] of binary mixtures, the thermodynamic states were defined. Table 1 mentioned previously, shows all states and property values for a simulation corresponding to a 17.5 kW absorption machine.

The condenser inlet conditions in this simulation correspond to those of the state 7 and outlet at state 8. To simulate and design the condenser a software for heat exchanger design referred as HYPROTECH™ was used. The condenser chosen for the modeling was a finned tube fin since it is more efficient for air-liquid exchange. Resulting dimensions for length x width x height of condenser were: 1.65 m x 1.16 m x .22 m. In the design 7.08 m^3/s of air were required to remove 18 KW of heat.

The condenser was sent into actual manufacturing to a company that specializes in heat exchangers. Some redesigning was required for the final version of the condenser design resulting in final dimensions of: 1.75 m x .99 m x .19 m , which did not differ much from the simulations. The total air provided was 4.72 m^3/s for the removal of 18 KW. The condenser is a two pass aluminum fin condenser.

Fig. 7 and 8 are illustrations of shop drawings of the condenser that was designed. The dimensions of the condenser were previously specified.

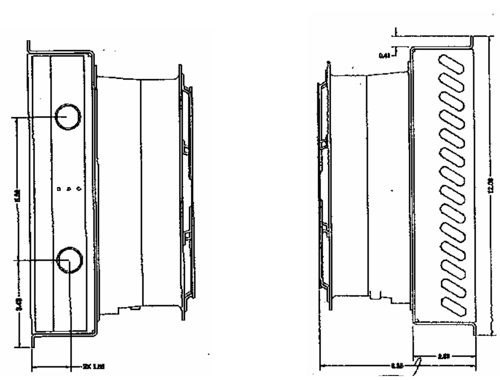

Figure 7- Side views of condenser with fans mounted

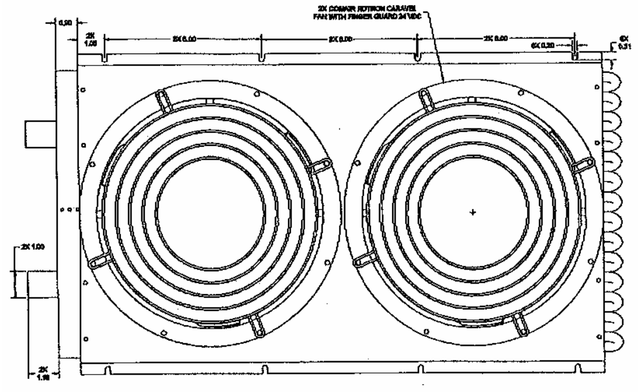

Figure 8- Front view of condenser

ABSORBER-EVAPORATOR UNIT

In the absorption machine, the absorber and evaporator work together as a unit. The vapor exiting the evaporator needs to come into thermal contact with the concentrated LiBr solution that enters the absorber from the top and falls inside the vertical tubes, creating the absorption effect. Air is cooling the outside surface of the tubes removing the heat from the concentrated solution and enabling the absorption process.

The evaporator was designed to be a falling film evaporator to meet the required needs. To model the evaporator it was used a film evaporation model on a horizontal tube [11]. The following expression was used to determine the Nusselt number which will provide the average heat transfer coefficient.

$$\frac{h_m d}{k_l} = 0.728\left[\frac{\rho_l(\rho_l-\rho_g)gh_{fg}d^3}{\mu_l(T_{sat}-T_w)k_l}\right]^{1/4} \qquad (3)$$

Using this estimated average heat transfer coefficient the heat load of the evaporator can be estimated by completing the design process for the evaporator.

For the previous estimate it is assumed that the sub-cooled liquid enters the evaporator after a constant enthalpy process and applying the cooling load goes to a completely saturated vapor state. The vapor formed rises through the vertical tubes of the absorber due to differences in concentration meeting the strong solution coming from the generator and completing the absorption cycle. The tubes used in the evaporator are enhanced surface tubes for greater efficiency.

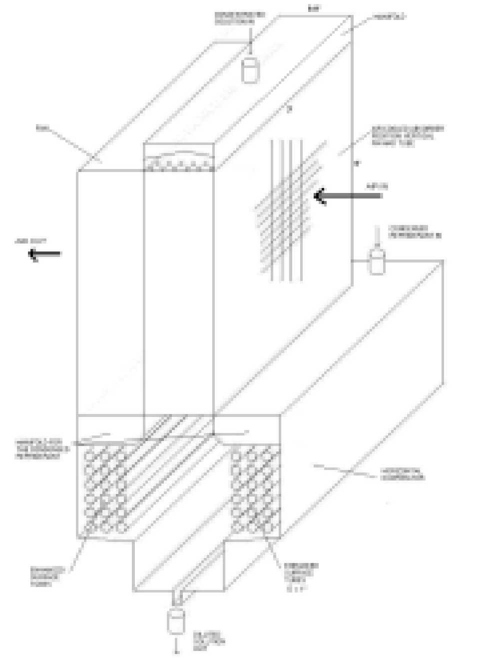

Figure 9 – Absorber Evaporator Schematic

ASSEMBLING OF THE MACHINE

Designing and constructing the components was the initial step in constructing the entire machine. A complete flow analysis had to be done in order to have an assembling diagram of all components. In Fig. 10 and 11 there are two different views of the absorption machine connections in detail. The heights of one component relative to the rest must be defined for flow to occur using the minimum amount of energy.

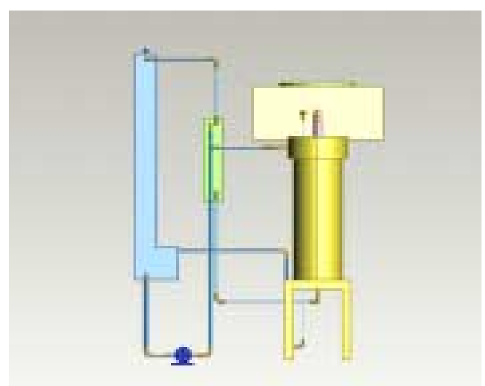

Figure 10- Side view of components relative heights

As seen from Fig. 10 the system contains a small pump after the absorber that takes the diluted solution back to the desorber. This pump is needed in the air cooled machine because the diluted solution exiting the absorber unit is too low to flow back by gravity to the desorber.

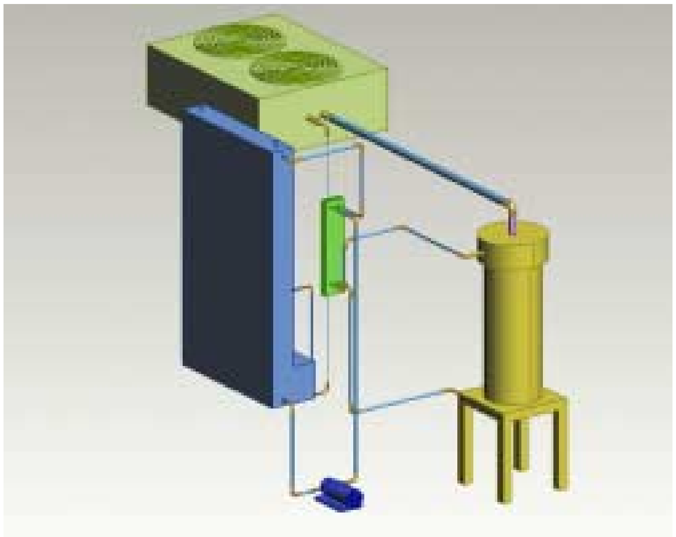

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Figure 11- View of fully assembled prototype

ABSORPTION MACHINE TESTING

Testing the absorption machine is necessary to validate the design process. The testing process is iterative starting from the initial assembling configuration and making changes along to optimize the final configuration. The testing can be divided into two stages preliminary testing and machine performance testing. During the first stage of the testing major changes in the machine configurations occurred to accommodate for adequate flow, improved the throttling process and other changes that made machine to function as a unified unit.

The machine performance stage starts after the prototype is fully functional to test the effect of varying the external parameters such as: mass flow rates, temperatures, vacuum conditions, etc. and measure how there affect the machine performance parameters (i.e. COP).

To start the construction of the prototype the first element tested was the generator. The generator is the heart of the absorption machine where heat (instead of work as in vapor-compression) is injected producing a flow of vapor to the condenser and strong solution to the absorber. The focus of the initial testing was assembling the generator to measure the production of vapor by coupling it with condenser and measuring the volume of water produced. Table 2 is a summary of the initial tests of the generator-condenser assembly. As seen in Table 2 the first tests were qualitative to observe the phenomenon occurring, subsequently condensate volume was measured. After the second test a deposit was connected to the strong solution leaving at the top of the generator to observe if there was flow out of the generator (eventually to absorber). The idea was not to introduce lithium bromide (corrosive) into the generator until an optimal generator-condenser configuration was obtained. Also the functionality of the lithium bromide is to release water vapor in the generator and recover it in absorber. Since the absorber was not yet assembled lithium bromide was not needed.

The generator-condenser configuration results breed optimism since worries of adequate functionality of design were overcome. According to the simulations, 27 liters per hour of condensate were required of which 19 were obtained by the seventh test. Some of the challenges of constructing the machine arose during these trials such as: reaching adequate vacuum and temperature conditions.

Table 2- First Generator-Condenser Tests Results

Parameters/Tests	First	Second	Third	Fourth	Fifth	Sixth	Seventh
Solution Used	Water	Water	Water	Water	Water	Water	Water
Initial Vacuum (inches of Hg)	27	25.5	25	25	28.5	28	27.5
Volume Charged (liters)	12.98	13.25	13.25	13.25	15.14	30.28	26.50
Flow Rate (Lpm)	3.79	7.575	94.63	94.63	75.71	75.71	45.42
Duration (minutes)	107	65.1	70.1	65.3	12	60	60
Amount of Condesate (L)	Condesate was Observed	Condensate was Observed	2.93	3.50	3.79	20.82	18.60

Subsequently, other configurations have been tested by connecting, generator, condenser, evaporator and absorber. Initially, it was believed that the expansion would occur in a U-tube. After installing the prototype, no expansion was observed in the configuration. In a set of separate tests it was demonstrated that expansion occurred by volume in expansion more efficiently that using other expansion methods such as orifice, short tube, or capillary tube.

Applying this concept the prototype was reconfigured and tested. Fig. 12 shows the optimal configuration that includes the expansion by volume. Results of preliminary tests with this configuration are presented in Fig. 13. It can be observed from this figure that temperatures lower than 20°C were observed while the evaporator pressure remained close to the design values.

The prototype has been fully constructed with modifications to the expansion section. The generator, condenser, absorber-evaporator and heat exchanger have been joined, and the machine is full testing.

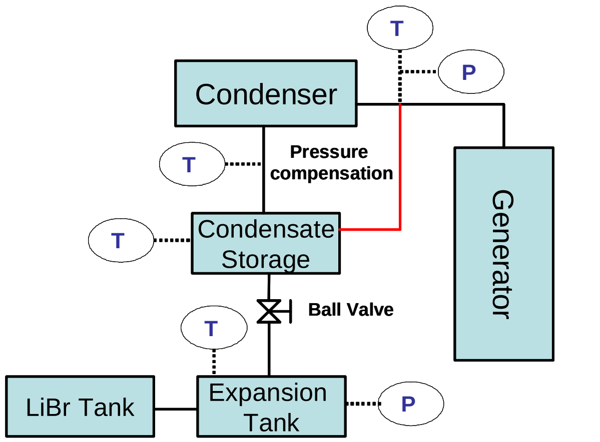

Figure 12 – Schematic of Absorption Machine with Expansion Device (P=Pressure sensor; T=Temperature Sensor)

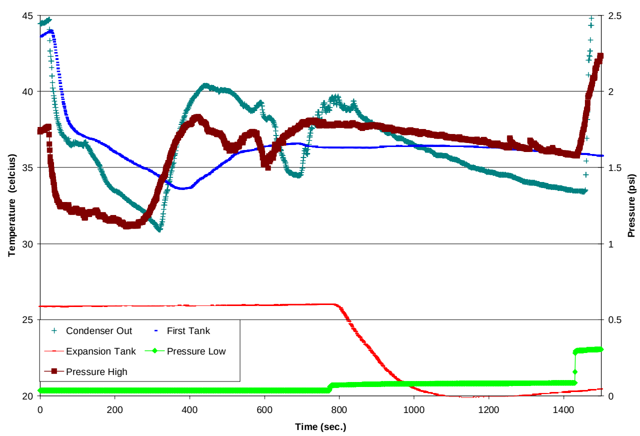

Figure 13- Results of Optimal Expansion Configuration

CONCLUSIONS

Absorption machine theory has existed for many years, however just recently has this technology reached a stage were it is also a commercially viable option.

In this paper the design process of an air cooled single effect absorption machine prototype was completed. The prototype has not yet fully been tested for its performance parameters (i.e. COP) but partial tests measuring functionality of the individual components suggest there a working prototype is on the way. The generator, condenser and throttling process of the prototype have all been tested extensively. The evaporator, absorber and solution heat exchanger have been coupled and tested qualitatively.

There is a lot of hope in absorption machine technology having a breakthrough in market, but this will depend on the adaptability of this technology for residential use. Although some more time will better answer the uncertainties of the air cooled absorption machine, there are high expectations in the potential downsizing of absorption machines for small cooling loads by air cooling these.

ACKNOWLEDGMENTS

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