DOMESTIC SOLAR POWERED HEAT PUMP SIMULATION: SUMMERTIME CONDITIONS

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ABSTRACT

This paper presents the simulation of the performance of a small scale water-ammonia absorption heat-pump using solar energy as heat source. The heat-pump is currently being tested at IST and the results show that the system can be used for ambient cooling providing comfort conditions although additional equipment is necessary.

RÉSUMÉ

Cet article présente la simulation du fonctionnement d'une petite pompe de chaleur à absorption fonctionnant à l'eau-ammoniaque et qui utilise l'energie solaire comme source de chaleur. Actuellement cette machine est testée à l' IST, et les résultats prouvent que le systéme peut être utilisé pour le refroidissement d'une habitation, bien qu'il soit nécessaire un équipement d'appui complémentaire.

KEYWORDS

Heat Pump, Solar Cooling, Buildings, Comfort, Absorption

1. INTRODUCTION

1.1 Air conditioning use of absorption Heat Pumps

Indoor conditions in Portugal have steadily improved in the last decades. This improvement is due to better building envelopes, as enforced by the Portuguese regulations as well as an increasing use of air conditioning systems. The present increase in the number of units sold is around 8% per year. With few exceptions all systems use electricity, and although Portugal has an important percentage of hydro-electricity, most of the electricity used is produced in conventional power stations. The result has been an increase in electricity demand and a resulting increase in CO_2 and NOx emissions.

Solar energy in Portugal is an important endogenous energy source. However it is seldom used and its use for cooling systems is almost non-existent. The use of adsorption systems has the benefit of not using halocarbons and heat can be provided by conventional fuels, biomass or solar energy.

1.2 Simulation using the experimental rig

At IST a prototype is being developed. The cooling power is 5kW and the heating power is 9kW. The heat source to be used is the solar energy. Complementing the experimental work, a procedure to simulate the working conditions of the system was developed. This tool provides the information on the system performance under outdoor and indoor conditions different than those used when making the rig experiments.

The model describing the system and the developed program use climatic parameters as well as the system characteristics and mixture properties. The program uses the EES (Engineering Equation Solver) utility for determining several property data. The basic function provided by EES is the numerical solution of a set of algebraic equations. The results of simulation program are presented in this paper. They show that the system being developed can cope with the heat and cooling loads in households.

2. SIMULATION MODEL PARAMETERS

2.1 Solar collectors

A prototype of a compound parabolic concentrators (CPC) solar collector was developed by AoSol - Energias Renováveis Lda, specially for this purpose. The solar radiation considered is based on the equations in /1/ for a CPC collector, with a tilt 20º, surface facing south in July.

The collectors were dimensioned to provide enough heat to the generator during the peak of radiation. As a result 13 collectors with an area of 26 m^2 and the following characteristics: optical efficiency ($F'\eta_0$) of 0.74 and a heat loss factor ($F'U_L$) of 2.5 $W.m^{-2}.K^{-1}$. Table 1 shows the hourly radiation on the CPC collectors, the air temperature, the inlet (T14) and outlet

(T13) water temperatures, the water flow, the collectors power output and their efficiency for a typical day in July.

Table 1 – Solar collectors output for a day in July

Solar hour	Rad [W/m²]	T_{air} [ºC]	T_{13} [ºC]	T_{14} [ºC]	m_{13} [kg/s]	Q_{util} [kW]
9.00	503.5	23.9	79.3	72.4	0.21	6.1
10.00	695.2	25.5	100.3	90.7	0.21	8.5
11.00	837.0	27.4	116.9	105.3	0.21	10.3
12.00	889.3	29.2	124.6	112.3	0.21	10.9
13.00	837.0	30.6	121.6	110.1	0.21	10.2
14.00	695.2	31.5	109.3	99.9	0.21	8.3
15.00	503.5	31.8	91.4	84.8	0.21	5.8

2.2 Heat loads

The conditioned space was considered has a single room having an area $100m^2$ and a height of 2.6m. The building has a light thermal inertia and the global heat transfer coefficient of the roof, walls and windows is 0.6, 0,5 and 4,0 $W.m^{-2}.K^{-1}$, respectively. The external walls face the main directions (N, E, S,W) and are similar, with an opaque area of $20m^2$ and a window area of $6m^2$. The roof has an area of $100m^2$.
The external heat loads were determined using the sol-air temperature difference data in /2/ and /3/ and an indoor temperature of 25ºC. The internal loads considered an occupation of 4 persons with a moderate activity, lighting of $10W.m^{-2}$ and an equipment heat load of 200 W. The fresh air rate is considered equal as the one imposed by the Portuguese regulation, e.g., $30m^3.h^{-1}$.person. The internal heat loads were determined according to ASHRAE /4/.
The calculated heat loads in July can be viewed in figure 1.

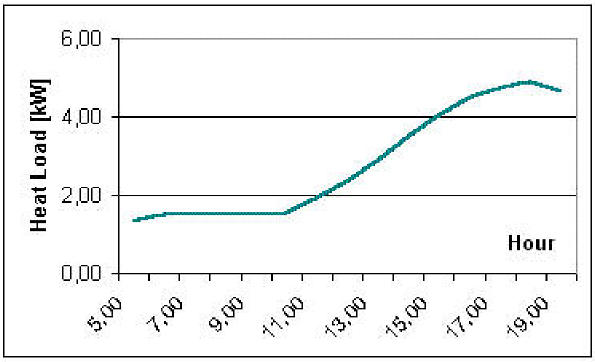

Figure 1 – Thermal heat load in July

2.3 Absorption machine

The absorption machine is a single effect heat pump using water-ammonia with a cooling capacity of 5 kW. This equipment is under test as described in /5/ and is represented in figure 2. Refrigerated water produced by the system is carried out to fan-coils in the volume under study. The heated water in the solar collectors is used to provide the necessary heat to the generator. Heat is rejected by means of a water circuit that goes through the condenser, the absorber and an air-water heat-exchanger in succession as shown in figure 2.

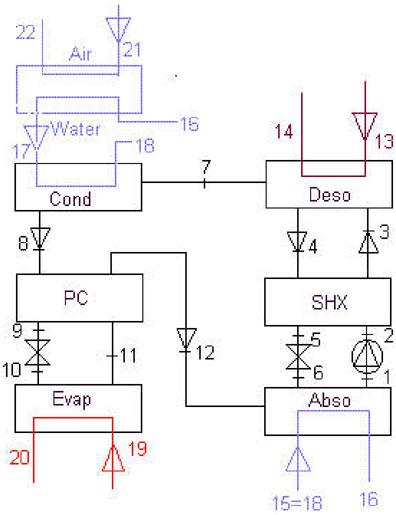

Figure 2 – Layout of the absorption machine

A model that simulates the working conditions of the machine was developed to work in the EES. The input data are the inlet water temperature in the desorber (T13), the outside air temperature (T21) and the inlet water temperature in the evaporator (T19). The heat-exchange in the different equipments of the system is determined considering the generic equation 1 for heat transfer in heat-exchangers and the equation 2 for the evaporator.

$$Q = U \times A \times \Delta T \ln \qquad (1)$$

$$Q = U \times A \times \Delta TM \qquad (2)$$

The UA values were obtained from experimental results. and related through empirical expressions for the condenser and the evaporator, because the mass flow weren't constants. These empirical expressions is given in table 2.

Table 2 –UA values of the equipment

	UA [W/K]
Desorber	0.80
Condenser	-0.012+0.0807Qcond
Absorber	0.73
HX air-water	1.50
Evaporator	-0.245+0.6288Qevap

The system does not use a rectifying column, as it is under study the economical effects (costs versus efficiency) of this conception. As a result the vapour concentration leaving the generator is 98.0% and the temperature in the evaporator (T20-T10) is 3ºC. The solution is still evaporated in the heat-exchanger (PC), sub-cooling the liquid leaving the condenser. It is considered that the flow through the pump is constant and equal to 0.035kg.s^{-1} The system COP under nominal conditions is 0.58 Tables 3 and 4 present the results obtained with the model for nominal running conditions (T13 =110ºC and T21=31ºC).

Table 3 – Nominal running conditions

Pt.	Fluid	T [ºC]	X	P [bar]	m [kg/s]	h [kJ/kg]
1	NH_3-H_2O	49.7	0.463	5.64	0.035	-16.3
2	NH_3-H_2O	49.8	0.463	15.26	0.035	-15.2
3	NH_3-H_2O	85.3	0.463	15.26	0.035	146.8
4	NH_3-H_2O	102.8	0.388	15.26	0.031	235.5
5	NH_3-H_2O	62.0	0.388	15.26	0.031	50.0
6	NH_3-H_2O	62.0	0.388	5.64	0.031	50.0
7	NH_3-H_2O	87.8	0.980	15.26	0.004	1458.0
8	NH_3-H_2O	40.1	0.980	15.26	0.004	169.2
9	NH_3-H_2O	18.2	0.980	15.26	0.004	70.6
10	NH_3-H_2O	8.1	0.980	5.64	0.004	70.6
11	NH_3-H_2O	16.6	0.980	5.64	0.004	1188.0
12	NH_3-H_2O	34.3	0.980	5.64	0.004	1287.0
13	Water	110.0			0.21	
14	Water	100.4			0.21	
15	Water	42.2			0.38	
16	Water	47.2			0.38	
17	Water	38.6			0.38	
18	Water	42.2			0.38	
19	Water	17.0			0.20	
20	Water	11.1			0.20	
21	Air	31.0			2.40	
22	Air	36.6			2.40	

Table 4 – Equipment power output for nominal conditions

Equipment	Q [KW]
Evaporator	5.0
Generator	8.5
Condenserr	5.7
Absorber	7.8
HX air-water	13.5

3 THE MODEL

The model developed, uses the solar collector temperature determination, the heat load calculation and the operating temperature calculation of the system to determine the overall working conditions, for predefined outside conditions and space use.

The water heated in the solar panel enters the generator at T13 and leaves at T14, entering again in the solar collector. The water flow in the collector and the generator is thus the same.

It is considered that the absorption machine can only operate if the waster temperature leaving the collector is above 78ºC. As a result the machine can only operate from 09 to 15 hours (solar) as can be seen from Table 1.

The cooling power of the absorption machine is considered equal to the heat load. The model does calculate a minimum possible indoor temperature. Coupling the heat load calculations with the machine power output, an operating condition curve can be determined.

From the values in Figure 1 and Table 1, a correlation between the outside air temperature and solar radiation and the thermal loads can be established. However the cooling loads can take place for periods when no solar radiation exists. Thus the absorption machine can only remove the thermal loads throughout the summer days if another heat source other then solar radiation is used. This alternative heat source should be used as back-up.

The results of the model can be viewed in table 5 for July. The inlet water in the generator varies from 79.3 to 124.6ºC. The heat provided by the solar panels, that is the generator heat input varies from 5.8 to 10.9 kW. The inlet temperature in the evaporator is 17ºC, while the outlet temperature varies from 9.6 to 12.9ºC. The COP increases with decreasing cooling power output, a natural result in absorption machines.

The minimum indoor temperature changes from 7.6 at 11:00 a.m. to 29.4ºC at 15:00 p.m. Only at 15:00 p.m. the indoor temperature is superior at comfort conditions, this is because the heat load is superior at cooling power.

Table 5 – Indoor temperature for a day in July

Solar hour	COP	T_{19} [ºC]	T_{20} [ºC]	Qevap [kW]	Heat Load [kW]	Tindoor [ºC]
9.00	0.67	17	12.1	4.1	1.5	**14.0**
10.00	0.63	17	10.6	5.4	1.6	**8.9**
11.00	0.59	17	9.8	6.1	1.9	**7.6**
12.00	0.56	17	9.6	6.2	2.4	**8.9**
13.00	0.56	17	10.1	5.7	3.5	**15.5**
14.00	0.58	17	11.2	4.8	4.1	**21.8**
15.00	0.59	17	12.9	3.5	4.5	**29.4**

4 CONCLUSION

The above results show that the use of absorption refrigeration systems for air conditioning must be viewed with care, particularly if one is using the usual methodology for vapour compression systems. The first reason is that the cooling capacity of a vapour compression system with air cooled condenser is determined for the outside air conditions close to the maximum air temperature. As a result during cooler temperatures of the outside air, the system can provide a higher cooling capacity. In the absorption system the maximum cooling capacity takes place for the lowest outside air temperature. As a result minimum cooling output is to be expected in the period where cooling is more necessary, e.g., from 14 to 17 hours, usually.

The absorption system to operate efficiently should be coupled with an cold storage vessel. If this solution is provided, the excess cooling capacity of the system during the morning hours will provide the additional cooling necessary for the period from 16 hours till the end of the day. If this solution is not used, the increase of temperature needed for the generator could only be obtained using large areas of solar panels or with an back-up heating (from a conventional boiler), and both these solutions are less interesting than the use of a storage vessel.

If cold storage is used, an efficiency higher than 0.58 is to be expected for the absorption system, while the necessary heat source is freely provided by the solar radiation under clear sky conditions.

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NOMENCLATURE

$F'\eta_0$– Collector optical efficiency factor

$F'U_L$ – Collector heat loss factor

U - global heat transfer coefficient

A – heat exchanger area

Δ Tln - mean logarithmic temperature difference

Δ TM - mean temperature difference