OPTIMAL DESIGN AND ECONOMICAL STUDY FOR SOLAR AIR-CONDITIONING BY ABSORPTION CHILLERS

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ABSTRACT:

The aim of our paper is to study a Lithium-Bromide absorption machine thoroughly. We started by a simple introduction showing the amount of fuel used in the last years for air-conditioning systems and how an absorption machine can save us a large amount of that fuel. The second part of our paper was concerned by studying each main part of the machine by that we mean the condenser, the evaporator, the generator, the absorber, the heat exchanger and the collector. Studying these parts means studying all the necessary parameters: the optimal temperatures used the flow rates, the required pressures, and concentrations of the fluids at every step. Using all these given parameters and the proper conduction-convection heat transfer equations, we can find the length of the tubes required to insure the transfer of heat (either by a software or by mathematical calculations). Then, we moved to the secondary parts such as: the pumps, the valves used, the divergent and all the other things that help in the good performance of the machine. The paper is concluded by studying if this machine is economical or not. Therefore, a simple application is done on a supermarket that needs an absorption machine of a 600 ton capacity. The study shows that the machine needs 6 years and eight month to retain its costs with an annual payback of 120000$.

I - INTRODUCTION:

According to an investigation done by the Lebanese society on solar energy, we were able to establish the price balance of the production and the sartorial consumption of electricity in Lebanon as follows:

Yearly invoice price of necessary fuel-oil for the production of electric energy is about 350 millions dollars, 40% of the electric energy production are consumed in the residential and commercial domains such houses, apartments, stores and commercial enterprises…, 33% from this 40% are consumed in the residential domain which is equivalent to 115 millions dollars (per year).

The main consumption factors in the residential field are: heating and air-conditioning, sanitary hot water, lighting and electric devices. Air-conditioning and heating consumes the largest part of the invoice then comes the sanitary hot water and then the lighting. This consumption can reach 75% of the total electric energy production, If one supposes that the consumption of the electric resistance in the water heater is of about 45% of 75%, then one can estimate that the invoice of the residential electric resistance is about 52 millions dollars per year. The increase in the yearly demand of electric energy is about 6 to 8% that implies that in the next 15 years the consumption will be 2 times the present consumption, which is a dangerous.

That is why we searched for an inexhaustible energy source which is the "The sun".

The main reason for choosing this energy source is certainly because it is inexhaustible; the second reason is the risk of the nuclear energy waste and its storage. The third reason is the notion that is recognized as "lasting development."

To Lebanon, the renewable energies are as in most countries, little developed. For example hydraulic energy that is not exploited to its maximal potential, it becomes useful and profitable if it used in a region isolated of high mountains and not joined to the national network as well as in the case of an emergency station as that found in the plain of Bekaa.

II - SOLAR AIR-CONDITIONING:

The most common technique consists in using solar collectors to provide the heat that is directed toward an absorption machine. This machine dissociates, by boiling point, a solution of water and bromide of lithium. After cooling, the recombination of the two components produces the cold air which is distributed then into the zones like classic air-conditioning.

The water and bromide of lithium is a clean, efficient and silent solution. It reduces the CO_2 emissions, the use of liquids refrigerants and the urban noise, but its technique is still in development phase [Ale91].

The use of the vacuum solar collectors is recommended to assure temperatures superior to 80°C that is necessary to the efficient working of the refrigerated absorption machine. A vacuum solar collector is composed of transparent glass tubes of 5 to 15 cm of diameter. Every tube contains an absorber to capture the solar radiance and an exchanger to permit the transfer of the thermal energy. The tubes are vacuumized to avoid the thermal convective losses of the absorber that receives a selective treatment to prevent the radiance [Ale91].

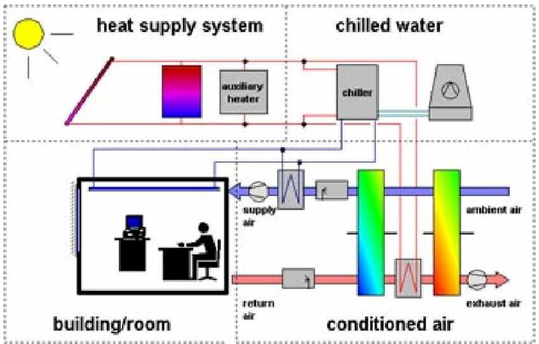


Figure 1: Solar air-conditioning

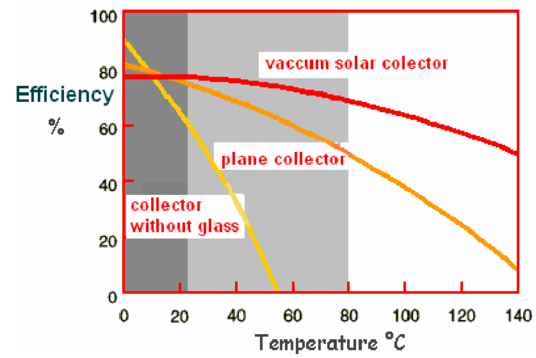


Figure 2: solar collectors efficient

To be efficient the vacuum must be lower in 10^{-3} Pa. A tube becomes useless if it is not completely hermetic and it is necessary to change it to preserve the performance of the whole collector. For the same equal absorber surface, the heat output is generally better than that of a plane collector, especially for elevated temperatures superior to 60°C.

III – ABSORPTION SYSTEM

The cycle "begins" when high-pressure liquid refrigerant from the condenser passes through a metering device (1) into the lower-pressure evaporator (2) and is collected in the evaporator sump. The heat transfer from the chilled water to the cool refrigerant causes the later to evaporate (2), and the resulting refrigerant vapor migrates to the lower-pressure absorber (3).

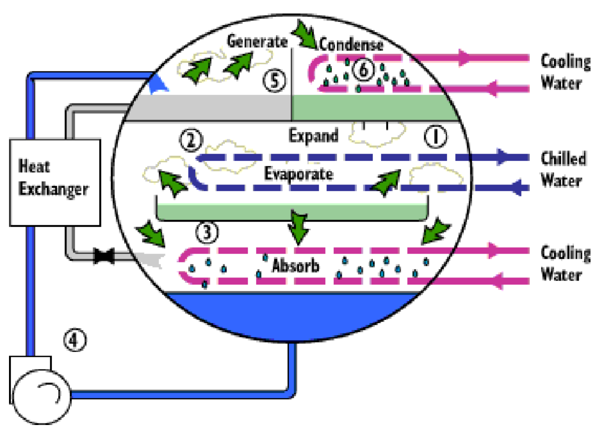


Figure 3: Absorption system

There, it is "soaked up" by an absorbent lithium-bromide solution. This process not only creates a low-pressure area that draws a continuous flow of refrigerant vapor from the evaporator to the absorber, but also causes the vapor to condense (3) as it releases the heat of vaporization picked up in the evaporator. This heat along with the heat of dilution produced as the refrigerant condensate mixes with the absorbent is transferred to the cooling water and released in the cooling tower [Miao98].

a. Solution Pump – A dilute lithium bromide solution is collected in the bottom of the absorber shell. From here, a hermetic solution pump moves the solution through a shell and tube heat exchanger for preheating.

b. Generator – After exiting the heat exchanger, the dilute solution moves into the upper shell. The solution surrounds a bundle of tubes which carries either steam or hot water. The steam or hot water transfers heat into the pool of dilute lithium bromide solution. The solution boils, sending refrigerant vapor upward into the condenser and leaving behind concentrated lithium bromide. The concentrated lithium bromide solution moves down to the heat exchanger, where it is cooled by the weak solution being pumped up to the generator.

c. Condenser – The refrigerant vapor migrates through mist eliminators to the condenser tube bundle. The refrigerant vapor condenses on the tubes. The heat is removed by the cooling water which moves through the inside of the tubes. As the refrigerant condenses, it collects in a trough at the bottom of the condenser.

d. Evaporator – The refrigerant liquid moves from the condenser in the upper shell down to the evaporator in the lower shell and is sprayed over the evaporator tube bundle. Due to the extreme vacuum of the lower shell [6 mm Hg (0.8 kPa) absolute pressure, the refrigerant liquid boils at approximately 39°F (3.9°C), creating the refrigerant effect. (This vacuum is created by hygroscopic action - the strong affinity lithium bromide has for water - in the Absorber directly below.) The single effect absorption cycle uses water as the refrigerant and lithium bromide as the absorbent. It is the strong affinity that these two substances have for one another that makes the cycle work. The entire process occurs in almost a complete vacuum.

e. Absorber – As the refrigerant vapor migrates to the absorber from the evaporator, the strong lithium bromide solution from the generator is sprayed over the top of the absorber tube bundle. The strong lithium bromide solution actually pulls the refrigerant vapor into solution, creating the extreme vacuum in the evaporator. The absorption of the refrigerant vapor into the lithium bromide solution also generates heat which is removed by the cooling water. The now dilute lithium bromide solution collects in the bottom of the lower shell, where it flows down to the solution pump. The chilling cycle is now completed and the process begins once again.

IV- THERMODYNAMIC STUDY OF AN ABSORPTION MACHINE:

1. Evaporator work conditions: the volume flow of water must cool from $\theta_{e,FP}$ = 12 °C to $\theta_{s,FP}$ = 7 °C. The net refrigerated power to be produced is "P" KW. The temperature of evaporation of water is θ_F = 3 °C. This temperature corresponds to a pressure of saturation of water p_F = 0.0076 bar. The regulator of supply of the evaporator ΔT allows to maintain the exit of the evaporator at temperature of 3 °C.

2. Volume flow of liquid refrigerant: The liquid refrigerant is water, its density is ρ_{FP} and its thermal capacity mass is presented under certain pressure , its volume flow is:

$$\dot{V}_{FP} = \frac{\phi_F}{\rho_{FP} \times c_{FP} \times (\theta_{e,FP} - \theta_{s,FP})} \qquad [1]$$

3. Condenser work conditions: The temperature of water available is $\theta_{e,CD}$ = 35 °C. Let us admit heating of water in condenser about $\Delta\theta_{éch,CD}$ = 5 K. The exiting temperature of water is $\theta_{s,CD}$ = 40 °C. The condensation temperature of water is θ_C = 45 °C. This temperature corresponds to water pressure of saturation
p_C = 0.096 bar.

4. Absorber work conditions: To improve absorption, we must attempt to cool the absorber as much as possible and reduce heating of water to $\tilde{\theta}_{éch,AB}$ = 4 K. The rich solution temperature in the exit absorber is θ_a = 41 °C. The steam temperature entering the absorber is θ = 47 °C. To simplify, one will suppose that the pressure losses entering evaporator and absorber is unimportant, the last one is under pressure p_F = 0.0076 bar.

5. Boiler work conditions: The thermal power of the boiler comes from the heated glycol with a solar collector until the temperature of 100 °C. The poor solution temperature of boiler exit is then θ = 100 °C. Otherwise, in the outlet boiler, the steam temperature can be considered like being in the saturation conditions. If we suppose that the pressure losses between generator and condenser is negligible, then the pressure that reigns in the generator is p_C = 0.096 bar.

6. Thermo dynamical values: Thermo dynamical values of the absorption cycle are listed in the next table [Chen92]:

Table 1: Thermo dynamics value

	T [^{O}C] Temperature	P [bar] Pressure	h [kj/kg] Enthalpy	X_{H2O} Composition in H_2O	x_{LiBr} Composition in LiBr	% in water vapor
1 Outlet absorber	45	0.0076	133.3	0.3882	0.6118	0.0000
2 Entering of the heat exchanger	45	0.096	133.4	0.3882	0.6118	0.0000
3 Entrance of boiler	86	0.096	210.8	0.3882	0.6118	0.0000
4 Outlet of boiler	100	0.096	250.2	0.3633	0.6367	0.0000
5 Outlet heat exchanger	56	0.096	169.6	0.3633	0.6367	0.0000
6 Entrance absorber	50.2	0.0076	169.6	0.3633	0.6367	0.0042
7 Entrance condenser	94.2	0.096	2674.5	1.0000	0.0000	1.0000
8 Outlet Condenser	45	0.096	188.4	1.0000	0.0000	0.0000
9 Entrance evaporator	3	0.0076	188.4	1.0000	0.0000	0.0705
10 Outlet evaporator	3	0.0076	2506.0	1.0000	0.0000	1.0000

7. Fluid descriptions :

Circulating Water mass flow $\dot{m}_F$: $\Phi_F = \dot{m}_F (h_6 - h_5) = \dot{m}_F\ h_6 - h_4$ [2]

Thermal power exchange in the condenser: $\Phi_{CD} = \dot{m}_F (h_2 - h_3)$ [3]

Volume flow of vapor, entering in 1, at absorber: $\dot{V}_{FV,1} = \dot{m}_F V_{FV,1}$ [4]

Volume flow of vapor, outlet in 2 of generator: $\dot{V}_{FV,2} = \dot{m}_F V_{FV,2}$ [5]

Volume flow of vapor, entering in 4, at regulator: $\dot{V}_{FL,} = \dot{m}_F V_{FL,4}$ [6]

Rate of circulation and flow of solutions: $\tau_{c,r} = \dfrac{x_{FR} - x_P}{x_r - x_p}$ [7]

Mass $\dot{m}_{SR}$ *flow of rich solution:* $\dot{m}_{SR} = \tau_{c,r}\ \dot{m}_F$ [8]

Volume flow $\dot{V}_{SR,a}$*, of rich solution outlet of the absorber:* $V_{SR,a} = \dot{m}_{SR}\ V_{SR,a}$ [9]

The rate of circulation of the poor solution: $\tau_{c,p} = \dfrac{x_{FR} - x_r}{x_r - x_p}$ [10]

Mass flow $\dot{m}_{SP}$ *of poor solution:* $\dot{m}_{SP} = \tau_{c,p}\ \dot{m}_F$ [11]

Volume flow of poor solution at outlet of the boiler: $V_{SP,d} = \dot{m}_{SP}\ V_{SP,d}$ [12]

8. Power of the PS pump for rich solution: This pump moves the volume flow of rich solution, $\dot{V}_{SR,a}$, of the absorber, where reigns pressure p_F = 0.0076 bar, towards the generator where pressure is p_C = 0.096 bar. If the functioning was ideal, the mechanical power that it would consume would be (by neglecting the pressure losses): $P_{PS,i} = \dot{V}_{SR,a}\ (p_C - p_F)$. To take into account the imperfection of the group motor-pump, an mechanical efficiency η_{MP} = 0,7 is admitted. The real power of this pump is had then: $P_{PS,r} = P_{PS,i}/\eta_{MP}$.

9. Equation of the enthalpy variation in the limits of the pump:

$$h_b - h_a = w = P_{PS,r} / \dot{m}_{SR} \qquad [13]$$

10. Thermal power exchanged in the heat exchanger EI:

$$\Phi_{EI} = \dot{m}_{SR} h_c - h_b \qquad [14]$$

11. Energy balance of the absorber: $\Phi_{AB} = \dot{m}_F h_1 + \dot{m}_{SP} h_f - \dot{m}_{SR} h_a$ [15]

12. Energy balance of the generator: $\Phi_{GE} = \dot{m}_F h_2 + \dot{m}_{SP} h_d - \dot{m}_{SR} h_c$ [16]

13. Performance coefficient for absorption cycle :

$$COP = \frac{T_F}{(T_C - T_F)} \frac{(T_M - T_C)}{T_M} \qquad [17]$$

14. Cooling water flow for the condenser and of the absorber:

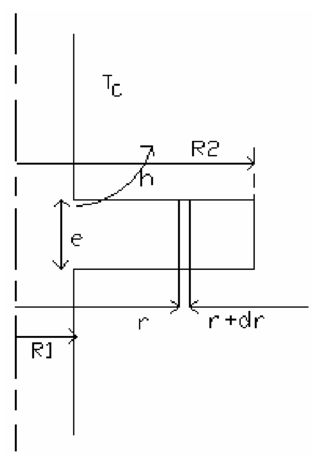
$\Phi = \Phi_{CD} + \Phi_A$ è $\Phi = \dot{V}_{eau,CD}\, \rho_{eau}\, C_{eau} \theta_s - \theta_e$ è $\dot{V}_{eau,\,CD} = \dfrac{\phi}{\rho_{eau} C_{eau} (\theta_s - \theta_e)}$ [18]

15. Flow of the heating fluid from the boiler: $\dot{V}_{glycol\ de\ chauffage} = \dfrac{\phi}{\rho_g C_g (\theta_s - \theta_e)}$ [19]

16. Dimensions of heat exchangers: Every thermal heat exchanger (evaporator or boiler), is supposed a circular pipe having internal radius R_1 and external radius R_3, and constituted of a solid material with thermal specific conductivity λ. The pipe is provided with the circular fins of radius outside R_2 and thickness e_l, the distance between fins is dx, L is the pipe length.

a. Calculation method for the evaporator and the boiler:

The thermal balance for the fins gives:



$$\frac{d^2T}{dr^2} + \frac{1}{r}\frac{dT}{dr} - m^2 T = 0 \qquad \text{with} \quad m^2 = 2h/(\lambda e_l)$$

Figure 4 : circular pipe configuration

For this circular fin: $S(r) = 2\pi r e$

Conditions at limits: $T(R_3) = T_0$ et $(dT/dr)_{R_2} = 0$

Temperature outside is T_c, The temperature of entering cool water is T_e, and that of the exiting is T_s, T_0 is temperature of the wall of the pipe.

hi: convection coefficient inside the tube

he: convection coefficient outside the tube

K: specific conductivity coefficient of the copper

m: Flow of fluid inside the tube

$T_{E:}$ Temperature outside (evaporator, boiler, as the case)

The equation solution of temperature variation in the fin is of shape is:

$$T(r) = (T_0 - T_E)\frac{besselK(1,mR_2)\times besselI(0,mr)+besselK(0,mr)\times besselI(1,mR_2)}{besselI(0.mR_3)\times besselK(1,mR_2)+besselK(0,mR_3)\times besselI(1,mR_2)} + T_E \quad [20]$$

The stream of heat through fins is: $\varphi = -2.\pi.\tilde{\lambda}\ \mathbf{d_1.e\ (dT/dr)}_{\mathbf{r=\tilde{R}_3}}$

$$d\varphi = m\ c\ dT$$

è and

$$\begin{cases} mc(T_2-T_1) = \varphi(T_0) \\ (T_1+T_2)/2 - T_0 = R_{th}\ \varphi(T_0) \end{cases}$$

with: ***c:*** the specific heat of the water, ***m***: Flow of cooling water, ***R_{th}:*** Thermal resistance of the pipe, ***$(T_1+T_2)/2$*** : average temperature of the water.

It is a system of two equations of two unknowns. The solution is of the shape:

$$T_2 = \frac{2\times m\times c\times T_1 - k_{fi}\times T_1 + 2\times R_{th}\times k_{fi}\times m\times c\times T_1 + k_{fi}\times T_E}{2\times m\times c + k_{fi}\times el + 2\times R_{th}\times m\times c\times k_{fi}} \quad [21]$$

With: $k_{fi} = 2\times\pi\times R_3\times e_l\times\lambda\times m\times Ft$

$$Ft = \frac{-besselK(1,mR_2)\times besselI(1,mR_1)+besselK(1,mR_1)\times besselI(1,mR_2)}{besselI(0.mR_1)\times besselK(1,mR_2)+besselK(0,mR_1)\times besselI(1,mR_2)} \quad [22]$$

The thermal balance of the temperature between fins (every two fins):

$$m.c.dT = K\ (T-T_E)\ ds \quad [23]$$

With $$K\ ds = ka = 2*\pi*dx/(1/\lambda * \log(R_3/R_1)+1/(hi*R_{13})+1/(he*R_3)) \quad [24]$$

$$m.c.\ \frac{\boldsymbol{T_1 - T_0}}{\mathbf{2}} = ka*(\ (T_1+T_2)/2 - T_E) \quad [25]$$

Then $$T1 = (T_E*ka+T_0*(m*c-ka/2))(m*c+ka/2) \quad [26]$$

By MatLab Software, we can solve the system.

b. Calculation method for the evaporator and the boiler (EI):

Thermal power Φ of the heat exchanger is bound in ΔT_{LM} by the relation:

$$\Phi = K\ S\Delta T_{LM} \qquad \Delta T_{LM} = \frac{\Delta T_2 - \Delta T_1}{\mathrm{Log}\frac{\Delta T_2}{\Delta T_1}} \quad [27]$$

The knowledge of the numbers of PRANDTL and REYNOLDS allows then to calculate the number of NUSSELT $N_u = \frac{h_c D_h}{\lambda}$ (D_h: is the hydraulic diameter). COLBURN's formula are used :

$$\mathbf{N_u = 0.023\, P_r^{1/3}\, R_e^{0.8}} \qquad [28]$$

The global exchange coefficient K is given by:

$$K = \frac{1}{\frac{1}{h_c} + \frac{1}{h_f} + \frac{e}{\lambda}} \qquad [29]$$

c. Calculation method of the heat exchanger for the condenser and the absorber:

A method **shell_tube_condensers** is used with the MatLab software [War02]. This method gives the number of tube heat exchangers and the length of every tube in each application.

17. Calculation Program: We built a MatLab program based on the previous equations for whatever climatic load. As example, for P = 3 KW, we find the following results:

$\dot{V}_{FP} = 1.43\times10^{-4}$ m³/s ;	$\dot{m}_F = 1.2944\times10^{-3}$ kg/s	$\Phi_{CD} = 3.218$ kW
$\dot{V}_{FV,1} = 0.2176$ m³/s	$\dot{V}_{FV,2} = 0.0228$ m³/s	$\dot{V}_{FL,} = 1.2944*10^{-6}$ m³/s
$\tau_{c,r} = 25.57$	$\dot{m}_{SR} = 33.097$ g/s	$\dot{V}_{SR,a} = 13.2\times10^{-6}$ m³ /s
$\tau_{c,p} = 24.57$	$\dot{m}_{SP} = 31.8$ g/s	$\dot{V}_{SP,d} = 12.37\times10^{-6}$ m³ /s
$P_{PS,r} = 0.14586$ W	$h_b - h_a = 0.0044$ kJ/kg	$\Phi_{EJ} = 2.565$ kW
$\Phi_{AB} = 4.218$ kW	$\Phi_{GE} = 4.44$ kW	*COP* = 0.675
$\dot{V}_{eau,CD} = 1.6\times10^{-4}$ m³/s	$\dot{V}_{FP} = 1.43\times10^{-4}$ m³/s	$\dot{V}_{glycol\ de\ chauffage} = 1.14\times10^{-4}$ m³/s

Length the evaporator = 5.417 meters for a tube of diameter ½ inch and a fin of beam equal in 4 times that of the tube.

Length the boiler = 4.161 meters for a tube of diameter ½ inch and a fin of beam equal in 4 times that of the tube.

Length of the tube for the outside heat exchanger: L = 4.38 m.

V - ECONOMY OF ABSORPTION MACHINE

The use of solar collectors is suitable to the absorption-refrigerated machines to a range of power, available on the market, from 50 to 5000 kW. The applications of this machine are many in our daily life (residential, hospitals, restaurants…). The following calculation is going to show an important yearly payback relative to an electric machine with same climatic power.

<u>1 - Calculation method:</u>

a- price and consumption of an absorption machine:

A PGE power in kilowatts requires a collector surface equal to S = PGE/(η*1.2), where η are the generator efficiency. Every two meters square of vacuum collector costs 500$ which implies that the vacuum collector price is equal: PSC=S*500/2

If it works 22 hours, 8 hour in the morning and 6 hours in the following day, the solar collector can assure the necessary power during the day for a 8 hours middle time. The rest (14 hours) will be assured by a fuel-oil furnace that could be used in winter for heating.

The seasonal consumption of fuel-oil in liter (150000 BTU/GAL) is then:

$$PAP=14*PGE*1000*30*5*3600*3.875/(150000*1055*0.8) \qquad [30]$$

NB: 0.8 is the efficiency of the furnace, 1055 is the factor conversion of BTU to JOULE and we can consider that this machine can be used 5 month each year.

If we consider that the price of 20 liters fuel is equal to 10000L.L, then the price of fuel-oil consumption by the furnace is:

$$PEM=PAP*10000/(20*1500). \qquad [31]$$

The electric consumption of the water circulation pump for the generator is: PUMP= $\hat{m}$ *g*H/η with $\hat{m}$ is the water mass debit necessary to the generator heating, H is the pump pressure in meter of water (15 meter), η is the pump efficiency. The water mass debit $\hat{m}$ is equal to: $\hat{m}$ =PGE*1000/(C*ΔT) with ΔT: difference of water temperature enters the entry and exit of the generator.

The electric consumption in kWh of the pump generator during the cooling season (5 months and 22 hs/day) is equal PUMPG = PUMP*22*5*30/1000.

The electric consumption of the water circulation pump to cooling the condenser and the absorber is also: PUMP=m*g*H/η . The flow of water mass necessary to cool the condenser and the absorber is equal to: m= (PAB+PCND)*1000/(C*ΔT).

We will be able to find thus that the electric power absorbed by the pump of the condenser and the absorber is: PUMP=m*g*H/η

The electric consumption of the pump in kWh during (5 months and 22 hs/day) is equal: PUMPCA = PUMP*22*5*30/1000.

Averagely we can consider that the price of an electric air chiller is 200$/tonne and the price of the solar absorption chiller is 2 to 3 times more.

The figure below present generally the installation price of the equipments for the air-conditioning machines (electric, solar absorption, tower of cooling…) for large power (>200 tons). The figure shows the installation price for absorption machine, centrifugal machine and the auxiliary equipments (cooling tower, pump…). We notice that the absorption machine and its equipment cost more than the centrifugal machine and its equipment.

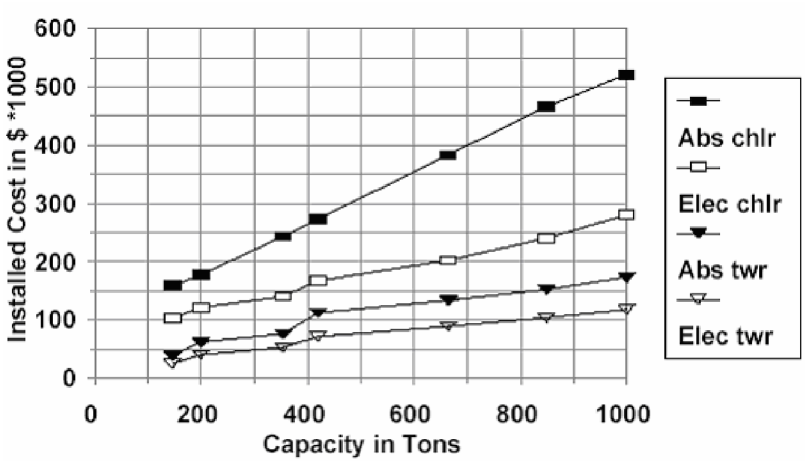


Figure 5 : Chiller and auxiliary equipments cost – electric and absorption

These values are based on an evaluation done in 1996. So, it is necessary to take into account the decreasing price, the modification and the development that air-conditioning systems undergo each year since 1996.

According to all what precedes, we can estimate the initial cost of a solar absorption machine by the following relation:

PRICE= (2.5*a*1000+PSC+200*5*2*1.5+S*0.5*5*2*1.8+S*5+35000)*1.05 [32]

- a: is the average price of one ton of electric centrifugal machine (200$/tonne).
- 2.5: factor we multiply the one-ton of an electric chiller by in order to get the price of an absorption machine that is 2.5 times more.
- 200: 200 meter of steel galvanized tubes are needed for piping construction where the meter is considered equal to 5.5 $. The coefficients (1.8, 2, 1.5...) are done to include the accessories of the pipes (valves….).

Then this price includes: The price of the absorption machine, the price of the vacuum collectors, the installation price of vacuum collectors with all the necessary accessories and their installation price (necessary quantity in meter of galvanized steel and their appropriate accessory….), the price of circulation pumps and an overestimate of 5% of this entire sum.

b - Price and consumption of an electric machine:

The price of an electric machine centrifuges air-conditioning having the same power that the one to absorption is equal to: PME=a*1000

The electric consumption in kWh of the compressor of such a machine during (5 months and 22 hs/day) is then: PMEE=1000*3.575*22*30*5*1.15/(COP*η)

3.575: conversion from ton to kilowatt, 22,: hours working number of the compressor per day, the work month number is 5, COP: performance coefficient of the electric centrifugal machine. This consumption is surmounted by 15% that include the consumptions of the pumps, of the cooling tour and fans.

c- Back time estimation:

According to everything that proceeds we will be able to find the yearly payback of an absorption machine (solar) in relation to the one electric.

The yearly payback is then:

DPA=PMEE*150/1500-(PEM+ (POMPEG+POMPECA)*2*150/1500) [33]

The difference between the price of an absorption machine and an electric centrifugal machine is equal to: DFA = (PRICE - SME) [34]

The back time is then: PAYBACK=DFA/DPA [35]

2 - Applications in Lebanon:

The below tables summarizes the back time and the yearly payback for a machine 15 tons and 1000 tons but first we considered that the machine worked for 5 month in the second we considered that it worked for 6 month.

Table 2: the back time and the yearly payback

power (1000 tons)	5 working months	**6 working months**
Initially price (PRICE)	1474000 $	**1474000 $**
yearly economy (DPA)	167240 $	**191800 $**
back time	**7.6 ans**	6.7 ans
power (15 tons)	5 working months	**6 working months**
Initially price (PRIX)	31435 $	**31435 $**
yearly economy (DPA)	2815 $	**3200 $**
back time	10.5 ans	9.5 ans

3 – Economical modeling for an absorption machine:

The following curves present the variation of the yearly payback and the back time for the machine price according to the power. (The application is calculated for an application in Lebanon: 22 hours per day and of 5 months).

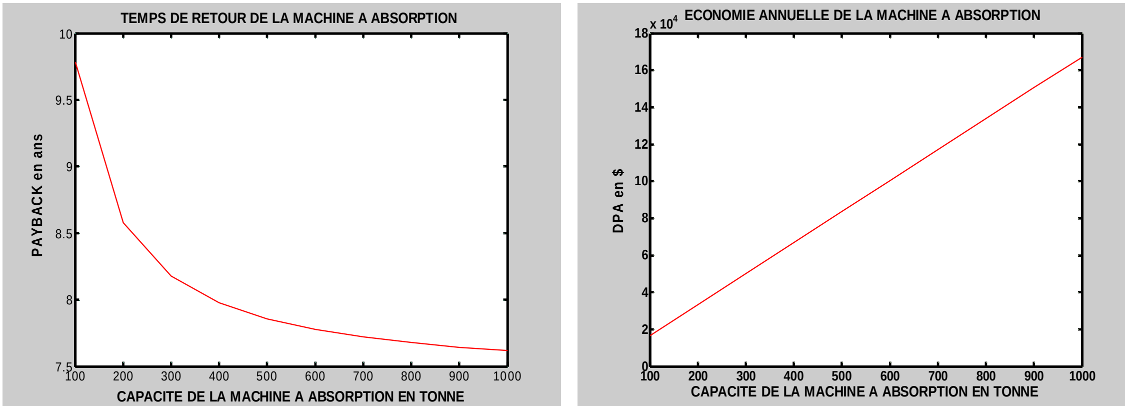


Figure 6: the variation of the yearly payback and the back time

These figures show that with an absorption (power between 100 and 1000 tons), we can achieve a yearly payback going from 20000 $ (100 tons) to 170000 $ (1000 tons) and a back time from 9.5 years (10 tons) to 7.7 years (1000 tons). These values are acceptable enough.

The followings curves present the same variation above for small power between 10 and 50 tons. (The application is calculated for in Lebanon for working conditions: 22 hours per day and of 5 months).

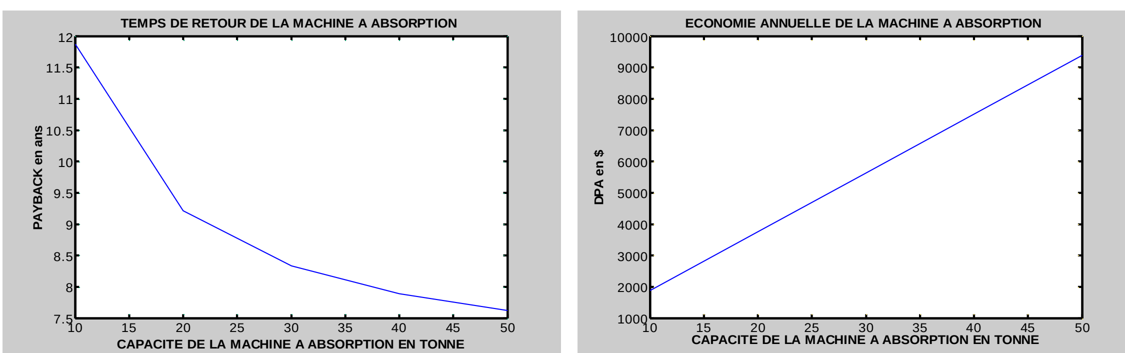


Figure 7: the variation of the yearly payback and the back time

We conclude that an absorption machine begins to be economic from 20 tons with a yearly payback going from 3000$ (20 tons) to 9200$ (50 tons) and for a back time respective from 9.5 years to 7.6 years. These values appear to be even acceptable.

VI - CONCLUSION:

The previous figures show that, the time back curve is decreasing with the increase of the absorption machine power but the curve of yearly payback variation is increasing with the increase of the power.

By comparison between the two above studies, concerning the yearly payback and the time back, it is concluded that the back time is nearly the same. But one sees that the yearly economy is more important for the larger powers, in relation to their back time (especially powers between 100 and 1000 tons).

But in spite of the elevated collector price, the elevated price of the fuel oil, and also the price raised of the absorption machine in relation to the electric centrifugal machine, the solar absorption machine will be able to be very economic for an application in Lebanon.

It is as obvious as such systems must not only be used for the air-conditioning in summer but also for the heating during the winter. During the intermediate season, the energy could be used, for example, for the sanitary hot water production. Also, such systems permit to avoid problems of environment that would have can be caused by the fossil energy use that makes it an interesting and promising system. Such a system can also be used in regions where the solar radiance is weaker, as for example in Germany, in France, in Belgium, with important surfaces collectors. However, the surface of the collector in those countries can be decreased the need for air-conditioning system is less there.

On the other hand, the air-conditioning by solar collectors is very interesting because the needs in cold weather and the solar radiance are nearly in phase. But one wishes that in a near future, the price of the solar collector decreases, which is foreseeable from the moment they are produced in big quantities, and that the price of the absorption machine decreases to equalize that of the electric centrifugal machine.

VII - REFERENCES:

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