EXPERIMENTAL ANALYSIS OF BUBBLE PUMP LiBr SOLUTION WORKING FLUIDS FOR ABSORPTION SYSTEM

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Abstract

The bubble pump provides the drive for moving the solution in an absorption refrigeration system where the. low cost energy for moving solution is its advantage. An experimental research test rig to study the performance of the bubble pump for absorption refrigeration units was made. The characteristics of bubble pump determine the efficiency of the absorption refrigeration system. Different diameter tube of bubble pumps was selected. Two internal and external heaters gave the energy to the solution, where the first is in the container of bubble pump and the other around the tube. The LiBr was selected as the tested liquid. The effects of changing the quantities such as submergence ratio (H/L), diameter of tubes and the power of internal heater were studied. The experimental results show that the performance of the bubble pump depends mainly on the power of internal heater and diameter of tube. It was also seen that for the same input power for larger tube diameter the vapor flow rate is increased where the liquid flow rate is decreased. The results show that as the power of internal heater increases, the liquid flow rate increase too. It can be seen that the vapor flow rate increases to a maximum value as the power of the internal heater increases, where finally it reaches a steady value (slug flow).

Keywords: absorption refrigeration system, bubble pump, LiBr, Submergence ratio

Presenting Author's biography

Mohammad Naghashzadegan. I am an assistant professor in mechanical engineering in Guilan University in Iran. My major interested area is air-conditioning, refrigeration and heat-transfer where I involve with several research projects including the design a pump-less absorption system, build a computer simulation program, modeling and simulating cooling and heating load calculation for Iranian climate.

ISBN 978-3-901608-32-2

1. Introduction:

The solution pump in the conventional vapor absorption Refrigeration system (VAR S) has two tasks: conveying the solution from the absorber to the generator and bringing the solution up to the condenser pressure to create the required pressure difference between condenser and evaporator. The pump forms a critical part of the absorption refrigeration system since it requires a high grade energy source like mechanical /electrical energy. Hence, significant advantage lies in the fully heat-operated pump less VARS, especially with the reference to small capacity units.

Saravanan and Maiya [1] have modeled the bubble pump for pump-less continuous vapor absorption refrigerator using the manometer principle based on intermittent slug flow of absorbent solution and refrigerant vapor. Pfaff [2] evaluated the performance of the bubble pump with an experimental test set-up built in glass and suggested that a pump with diameter 10 mm with a heat input of about 40W is suitable for a refrigerator of about 100W cooling capacity. The results also indicated that the pumping ratio is independent of the bubble pump heat input within the operating ranges studied but showed an increase towards higher driving head, lower pump lift and smaller tube diameter.

The bubble pump operated vapor absorption cooler makes use of bubble pump technique and hydrostatic principle. To restrict the height of the refrigerator to a reasonable height of 1.5 m, there is a constraint on the selection of the working fluids. The refrigerator should operate at low absolute pressure (vacuum) to achieve low-pressure difference between the condenser and evaporator so as to restrict the refrigerator height to practical limits. The pressure difference between the high and low pressure sides (generator and absorber sides) of water–salt and alcohol–salt systems is low enough for the bubble pump to pump the solution from absorber to generator, thus eliminating the need for the solution pump. It is an essential requirement for miniaturization of the absorption refrigerator [3,4].

Water is an excellent refrigerant having low vapor pressure as desired by the cooler, high latent heat and low viscosity. Lithium bromide and similar other salts are very good absorbent having negligible vapor pressure and low specific heat. The affinity between water and lithium bromide is also very high and the mixture is safe, non-toxic and environmental friendly.

The problem of crystallization can be overcome by controls and restricting the operating ranges. Hence water-lithium bromide (H_2O-LiBr) has been selected as working fluid for this cooler [5].

H_2O-LiBr vapor absorption systems are normally water-cooled. As these systems operate under vacuum, pressure drops in the components and the connecting tubes are of main concern. These aspects are taken care in water-cooled large capacity H_2O-LiBr systems by locating the acupressure components in the same shell. For small capacity, condenser and absorber must be air-cooled and pressure drops in the components and connecting tubes could be high. Parallel flow paths are proposed to reduce the pressure drop between evaporator and absorber in such units and a combination of _U_ tube and capillary tube is suggested to stabilize the systems at varying ambient conditions [6].

2. Absorption cycle:

The absorption cycle is a process by which refrigeration effect is produced through the use of two fluids and some quantity of heat input, rather than electrical input as in the more familiar vapor compression cycle. Both vapor compression and absorption refrigeration cycles accomplish the removal of heat through the evaporation of a refrigerant at a low pressure and the rejection of heat through the condensation of the refrigerant at a higher pressure. The method of creating the pressure difference and circulating the refrigerant is the primary difference between the two cycles. The vapor compression cycle employs a mechanical compressor to create the pressure differences necessary to circulate the refrigerant. In the absorption system, a secondary fluid or absorbent is used to circulate the refrigerant. Because the temperature requirements for the cycle fall into the low-to-moderate temperature range, and there is significant potential for electrical energy savings, absorption would seem to be a good prospect for geothermal application. Absorption machines are commercially available today in two basic configurations. For applications above 32^{o}F (primarily air conditioning), the cycle uses lithium bromide as the absorbent and water as the refrigerant. For applications below 32^{o}F, an ammonia-water cycle is employed with ammonia as the refrigerant and water as the absorbent.

3. System analysis

To evaluate the performance of the bubble pump, the experimental data was analyzed by using three control volumes. The first control volume, CV1, contains the generator and the volumetric flow meter of the rich solution as shown in Fig. 1. This control volume was used to evaluate the performance of the generator. At the inlet of CV1, it was assumed that the solution is in equilibrium (i.e. all the solution's thermo physical properties are known since both the pressure and the temperature are known) and the volumetric flow rate of the rich solution was measured. At outlet, two-phase

binary flow was obtained. The flow consists of superheated gas and poor solution.
Since the process is fast the poor solution is not at equilibrium and its properties are unknown.
The refrigerant flow rate at the generator outlet is the maximal refrigerant flow rate that could be achieved by the bubble pump, thus the maximal cooling capacity of the cycle could be calculated.

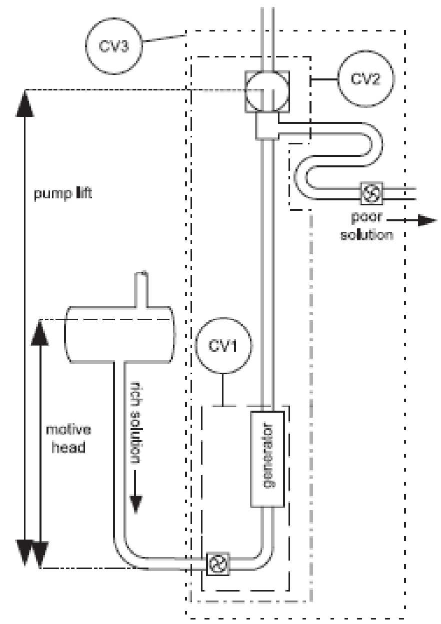


Figure 1. Control volumes description.

The continuity equation for CV1 could be expressed as:

$$\dot{m}_{rich} = \dot{m}_{ref} + \dot{m}_{poor} \tag{1}$$

the continuity equation for the refrigerant only could be written as:

$$\xi_{rich}\dot{m}_{rich} = \dot{m}_{ref} + \xi_{poor}\dot{m}_{poor} \tag{2}$$

The continuity equation includes three unknowns' $\dot{m}_{poor}$, $\dot{m}_{ref}$ and ξ_{poor}. To complete the model another equation is required. The energy balance on the generator could provide the third equation.

$$\dot{Q} = \sum \dot{m}h \tag{3}$$

Substituting Eqs. (1) and (2) into Eq. (3) results in

$$\dot{Q}_{genrator} = \dot{m}_{rich} \times \left(\frac{1-\xi_{rich}}{1-\xi_{poor}} h_{poor} + \frac{\xi_{rich}-\xi_{poor}}{1-\xi_{poor}} h_{ref} + h_{rich} \right) \tag{4}$$

Since the enthalpy of the mixture is expressed as a function of the concentration and the temperature, Jelinek [7], the unknowns in Eq. (4) are the concentration of the poor solution, ξ_{poor} and the heat supplied to the solution, $\dot{Q}_{genrator}$. Assuming the generator heating efficiency (of 100% when the generator is small and well insulated), the poor solution concentration can be found analytically.
The second control volume (CV2) includes the generator, the bubble pump and the volumetric flow meter of the rich solution as shown in Fig. 1. This control volume has one inlet (i.e., rich solution) and two outlets (i.e., superheated gas and poor solution). Assuming that the rich and the poor solutions, at the bubble pump inlet and outlet, respectively, are in equilibrium conditions, Fig. 1. Control volumes description.
The thermophysical properties of the fluids at these locations are defined. The mass flow rate of the poor solution can be obtained by using similar continuity equations (Eqs. (1) and (2)).

$$\dot{m}_{poor} = \dot{m}_{rich} \frac{1-\xi_{rich}}{1-\xi_{poor}} \tag{5}$$

Using this control volume, both the mass flow rate of the poor solution and the gas at the separating vessel inlet can be estimated. Thus the amount of refrigerant that has been absorbed back into the solution while it flows through the bubble pump tube can be estimated. Applying the energy conservation on this control volume will allow estimating the heat lost through the bubble pump walls.
Examining the control volume containing the generator, the bubble pump, the poor solution heat exchanger and both volumetric flow meters of the rich and the poor solutions (CV3 in Fig. 1),
will provide the amount of refrigerant absorbed in the poor solution heat exchanger. Continuity equations have been written for this control volume. Since the poor solution at the exit is subcooled, equilibrium condition cannot be assumed and the concentration of the poor solution cannot be defined from equilibrium conditions (i.e. pressure–temperature–concentration). Hence,
the thermodynamic state of the poor solution is not defined. Nevertheless, since the volumetric flow rate of the poor solution was measured, the concentration of the refrigerant in the poor solution can be found by applying the continuity equations and the relation between the mass and the volume flow rates (quasi-equilibrium state).

$$\dot{m} = \rho\dot{V} \tag{6}$$

Using the last definition, together with the continuity equations results in

$$\dot{m}_{rich} = \dot{V}_{poor}\rho_{poor}\frac{1-\xi_{poor}}{1-\xi_{rich}} \quad (7)$$

Since the density of the solution is a function of the solution concentration and the temperature as presented by Jelinek [7], and the temperature of the poor solution is measured, Eq. (7) is an implicit equation for the concentration of the refrigerant at the poor solution. To find the poor solution concentration, Eq. (7) was solved by an iterative method.

Recall that in a diffusion absorption system, the objectives of the bubble pump unit (i.e., generator and tube) is to circulate the flow and to separate refrigerant from the rich solution.

Since a higher refrigerant mass flow rate will increase the cooling capacity of the diffusion absorption unit, the back absorption that occurs at the bubble pump tube decreases the cooling capacity of the diffusion absorption unit. Thus we define the bubble pump efficiency as follows.

$$\eta = \frac{\dot{m}_{ref\_CV3}}{\dot{m}_{ref\_CV1}} \quad (8)$$

Analyzing the experimental results using Eq. (8) showed that the bubble pump efficiency is 45–55%.[8]

4. Results

Three Pyrex tubes of diameters 15, 20 and 25 millimeter have been selected where the tubes length was 110 centimeters. One heater is in the bottom of container where for the better performance the second heater with low power was twisted around the pipes. The variable parameters are heat input, submergence ratio (H/L) and tube diameter of lift.

The effects of the variation of these parameters on vapor and liquid flow rate were studied where the working fluid was LiBr solution 54% and submergence ratio (H/L) was kept 0.5. The power of external heater was 300 watt for the first where later it changed to 600 watt. The power of internal heater was changed from zero to 1100 watt. The results have been shown at figures 2 and 3.

The figures show as the power of internal heater increases the vapor flow rate increase s too. The vapour flow rate increases for larger tubes for the same power because the external heater has more effect for lager tubes. Although there is external heater but the vapour flow rate is zero when there is no internal heat power that is because the flow regime is changed to the bubbly regime so that the small existence bubbles can not move up..

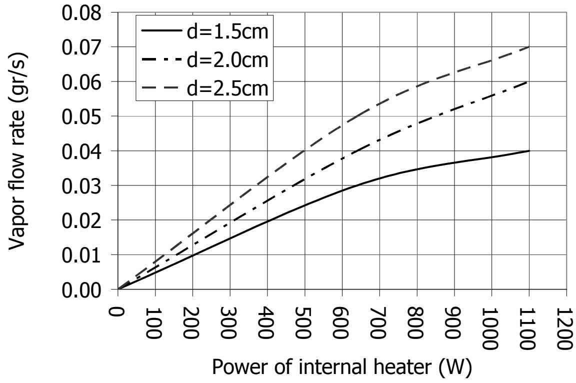


Figure 2. Variation of vapor Flow rate versus internal heat power (external heat power = 300 W)

Due to absorption of LiBr with vapour water the separation process is more difficult than water from vapour water. As a result in similar situation the vapor flow rate at the time of pumping water is more than at time of pumping LiBr.

Figure 3 shows the variation of vapor flow rate versus of the internal heat power.

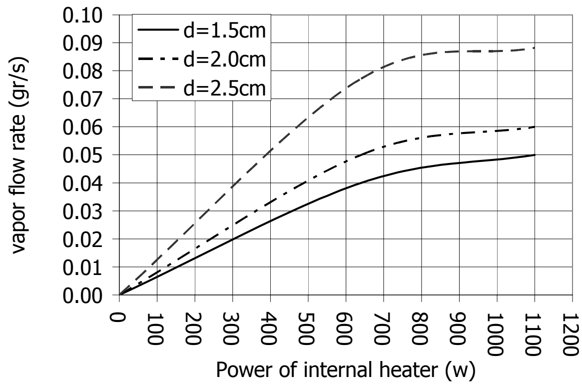


Figure 3. Variation of vapor Flow rate versus internal heat power (external heat power = 600 W)

The vapor flow rate increases to a maximum value as the power of the internal heater increases, where it reaches a steady value. but as the internal power. Here flow regime has changed to churn and annular regime from optimum case (Slug.

Figure 4 shows the variation of liquid flow rate versus of the internal heat power with the same characteristics as before.

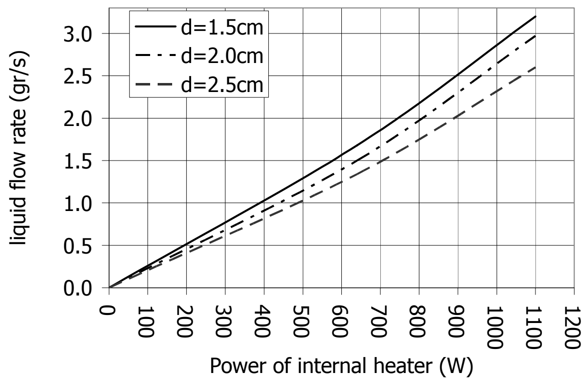


Figure 4. Liquid Flow rate versus internal heat power (external heat power = 300 W)

It is seen that as the power of internal heater increases, the liquid flow rate increase. The results show that the liquid flow rate in the thinner tubes is more and that is because although in thinner tubes bubbles the bubble can fill tubes easier and push the fluid upward and act like a piston. Experiments show that LiBr 54% boils at 140 ^{o}C .Figure 5 shows the variation of liquid flow rate versus of the internal heat power where the external power heater is600 watt.

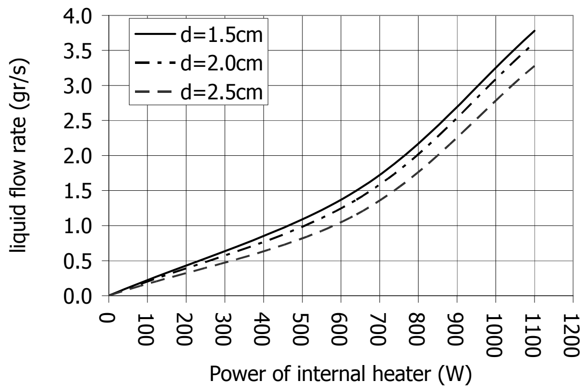
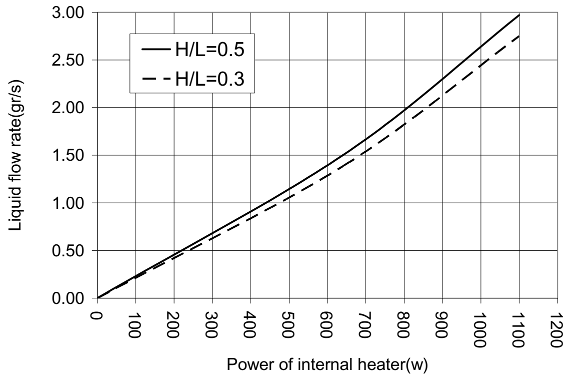


Figure 5. Variation of liquid Flow rate versus internal heater power (external heat power = 600 W)

The figure shows as the power of the internal heat increases the liquid flow rate increases too. Figure 6 compare the effect of submergence ratio (H/L) on liquid flow rate where the external heat was kept 600 watt.
The Figure shows that the larger H/L the larger liquid flow rate.
At H/L less than 0.2 the bubbles reach to the surface of fluid rapidly, because enough fluid doesn't exist between the bubbles.



The study was continued on absorption system with the capacity of 1 KW to find the number of tubes which should be replaced instead of the mechanical pump.
G.A. Florides et al [9]'s simulation has been used here where a 1KW absorption system was analyzed. Figure 7 shows such this schematic and table 1 shows its performance analysis.
The test results show that the tubes with the diameters of 25 millimeter and H/L=0.5 gave similar results of their results, where the LiBr flow rate was 0.00517 kg/s. For this flow rate a bubble pump system with two tubes needed. These tubes can produce 0.176 gr/s vapor. Table 1 shows a 1KW absorption system need 0.431 gr/s vapor, so to complete cycle the system need 0.255 gr/s more vapour and that will be produced in generator therefore less energy will be used in this system comparing with the system with a mechanical pump.

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Table 1. Data for single effect LiBr–water Cooling system

Point	m (kg/s)	X (%LiBr)
1	0.00517	55
2	0.00517	55
3	0.00517	55
4	0.00474	60
5	0.00474	60
6	0.00474	60
7	0.000431	0
8	0.000431	0
9	0.000431	0
10	0.000421	0
11	0.000011	0

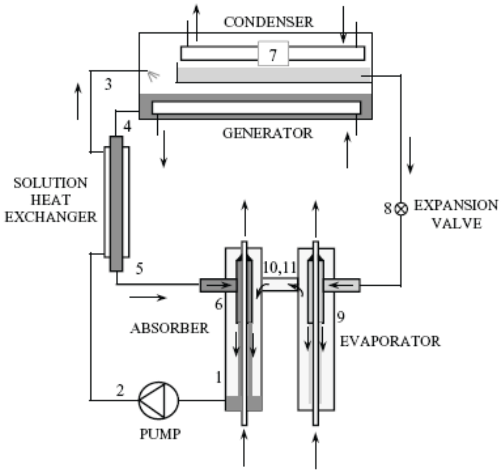

Figure 7. A schematic of a LiBr Absorption system
With a mechanical pump

5. References

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