Vol.1 pp. 34-39 International Journal of Scientific Research & Engineering Technology (IJSET)

EFFECT OF TUBE DIAMETER ON THE PERFORMANCE OF THE BUBBLE PUMP

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Abstract – **The mathematical model will be able to predict the operated condition (required tube diameters, heat input and submergence ratio….). That will result in a successful bubble pump design and hence a refrigeration unit. In the present work a one-dimensional two-fluid model of boiling mixing ammonia-water under constant heat flux is used to predict the outlet liquid and vapor velocities and pumping ratio for different heat flux input to pump. The influence of tube diameter on the functioning of the bubble pump is presented and discussed.**

It was found that, the liquid velocity and pumping ratio increase with increasing heat flux, and then it decreases. Optimal heat flux depends namely on tube diameter variations. Vapor velocity increases linearly with increasing heat flux under designed conditions.

Keys words – **Bubble pump, two-fluid model, simulation, heat flux, tube diameter.**

I. INTRODUCTION

A diffusion-absorption refrigeration cycle or a pumpless vapor absorption refrigeration cycle holds a great significance in noiseless refrigeration applications. The diffusion-absorption cycle relies on a bubble pump to pump the solution from the absorber to the boiler [1,2]. A bubble pump is a fluid pump that operates on thermal energy to pump liquid from lower level to the higher level.
Heat applied to the pump causes formation of bubbles and the density of strong solution in the vertical pump tube is reduced so that the solution is forced to the top by the static head of solution in the absorber vessel. The vapor which is released by boiling the solution will eventually become the condensed refrigerant, and its mass rate will dictate the refrigeration capacity of the refrigerator. According to established theory of absorption refrigeration, this mass rate of refrigerant is supported by the circulation rates of the strong and week solutions and their concentration difference [3].
The experiments and theoretical considerations showed that, for a specific heat input, the diameter of the lift tube has no effect on the pumping rate if the pump is running in the slug or churn flow regimes [4, 5]. When the maximum lift-tube diameter is exceeded, the flow pattern changes from slug flow to an intermittent churn-type flow [5, 6]. After a certain pumping height is exceeded, the pumping action stopped.
Pfaff *et al.* [7] studied the bubble pump with a lithium bromide–water vapor absorption cycle. They developed a mathematical model using the manometer principle to evaluate the bubble pump performance. They found that the pumping ratio is independent of the heat input. However, the frequency of the pumping action increases as the heat inputs to the bubble pump increases, or if the tube diameter decreases. The model was then used to analyze an ammonia-water system and it was found that the diameter that maximizes the efficiency of the bubble pump is between 4 mm and 26 mm for a liquid pumping rate between 0.0025 kg/s and 0.02 kg/s. However, the efficiency rapidly decreases when diameters below the optimum values are used; therefore it is recommended that the diameter should be slightly larger than the optimum value.
To increase its refrigeration capacity, a multiple lift-tube bubble pump can be used, in order to increase the volume flow rates of the fluids, which are directly related to the amount of refrigerant produced. Vicatos and Binnet [3] testing on a diffusion-absorption plant using a multiple lift tube bubble pump, and the effects of additional tubes on the system's performance have been recorded. Although a full range of heat inputs could not be implemented, because of the limitations of the components of the unit itself, it was observed that the refrigeration cooling capacity was increased without a significant drop in Coefficient of Performance (COP). It was concluded that the multiple lift tube bubble pump has no limitation to the fluid flow rate and depends solely on the amount of heat input. This gives the freedom to design the lift tube pump according to the refrigeration demand of the unit, and not the other way round which is the current approach by the manufacturers world wide.
The bubble pump model presented by Delano [8] assumed that all flow takes place in the slug flow regime. At first, it may seem that a pump tube with a larger diameter would always be advantageous. However, increasing the diameter when the liquid flow is constant will eventually cause the assumed slug flow to change to bubbly flow. Delano concluded that increasing the heat input to the bubble pump for a fixed submergence ratio increases the flow rate of the liquid through the bubble pump to a maximum and any further increase in the heat input decreases the liquid flow rate. White [9] showed a rapid decrease in efficiency when the diameter drops below a particular value. A numerical

ISSN: 2356-5608

and experimental study of convective boiling of a binary organic solution in a vertical tube was conducted by Levy *et al.* [10]. They obtained a detailed description of the flow characteristics. The influence of the heat source and operating pressure on the flow characteristics was examined both numerically and experimentally. Jakob *et al.* [11] reported that the indirectly heated generator with its bubble pump is the main new feature of a solar heat driven ammonia-water diffusion-absorption cooling machine and that all the prototypes constructed performed well. The performance of three diffusion absorption refrigeration (DAR) systems, which differ in their generator and bubble pump configuration, was studied numerically by Zohar *et al.* [12]. They showed that the configuration that integrated both the generator and the bubble pump is of great interest. An experimental investigation of an air-cooled diffusion-absorption machine operating with a binary light hydrocarbon mixture (C_4H_{10}/C_9H_{20}) as working fluids and helium as pressure equalizing inert gas is presented by Ben Ezzine *et al*. [13]. The experimental results show that the bubble pump exiting temperature as well as those of the major components of the machine but the absorber is very sensitive to the heat power inputs to the bubble pump. For bubble pump heat inputs from 170 to 350W, the driving temperature varies in the range of 120–150 °C.

Except of Levy *et al*. [11], most of the models reported in the literature for two-phase flow in a bubble pump, use the Beattie and Whalley's method [14] and the drift flux method (Zuber and Findlay [15]) to calculate the two-phase friction factor and the gas void fraction, respectively. The difference between these models is the value of the coefficients used in the drift flux model.

In the recent decades, significant developments in the two-phase flow formulation have been accomplished by introducing and improving the two-fluid model. The two-fluid model can be considered the most detailed and accurate macroscopic formulation of the thermo-fluid dynamics of two-phase flow [11, 16-18]. This model treats the general case of modeling each phase or component as a separate fluid with its own set of governing balance equations.

In the present study, the bubble pump was a vertical uniformly heated tube with an ammonia-water mixture. A numerical study based on two-fluid model was carried out to investigate the influence of heat flux input in the bubble pump, for different operating conditions such as tube diameter, ammonia fraction in solution and inlet pressure on the optimum heat flux input in the bubble pump.

2. MATHEMATICAL MODELING

The diffusion-absorption refrigeration cycle consists of a generator bubble pump, an absorber, an evaporator and a condenser, and usually operates with ammonia/water/hydrogen or helium as working fluid. Fig. 1 shows the main components of an absorption-diffusion refrigeration cycle and flow configurations in the bubble pump [19].

In the diffusion-absorption cycle, the bubble pump is a heated tube that lifts fluid from a lower reservoir to a higher one (fig. 1). The generator configuration is of great importance. Heat is usually supplied at the bottom of the tube [3, 7, 9]. In the present work, heat is applied along all the tube length. This configuration of the bubble pump has two advantageous. First, it increases the coefficient of performance of the cycle (COP) using minimum heat as possible and desorbing as much refrigerant as possible [12]. Second, it can be heated using solar thermal energy by integrating the bubble pump tubes and the solar collector [11, 20, 21].

In the present work, the two-fluid model was used for the two-phase flow region considering the hydrodynamic non-equilibrium between the liquid and vapor phases. The flow configuration was not limited to the slug regime [8-9], starting as bubbly and ending as annular (Fig.1). The mathematical model used is described below.

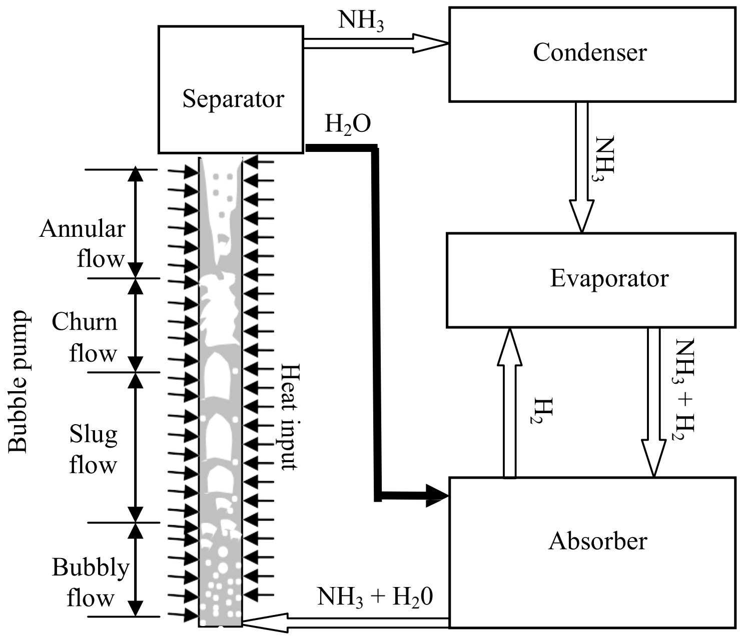

Fig. 1 Main components of an absorption-diffusion refrigeration cycle and flow configurations in the bubble pump

A. The Two-fluid Model

The liquid and vapor superficial velocities and the void fraction throughout the tube are predicted using a one-dimensional two-fluid model**.** In the two-phase region, the general conservation equations of mass, momentum and energy were formulated by Ishii and Mishima [22]. For the steady state with negligible kinetic and potential energy, the conservation equations are reduced to the following five equations [23, 24]:

ISSN: 2356-5608

- Phase mass equations

$$\frac{d}{dz}\left(\alpha\rho_G u_G\right) = \Gamma_G \qquad (1)$$

$$\frac{d}{dz}\left[\left(1-\alpha\right)\rho_L u_L\right] = \Gamma_L \qquad (2)$$

- Phase momentum equations

$$\frac{d}{dz}\left(\alpha\rho_G u_G^2\right)+\alpha\frac{dP}{dz}+\alpha\rho_G g = -F_{WL} - F_{GL} - F_{GI} \qquad (3)$$

$$\frac{d}{dz}\left(\left(1-\alpha\right)\rho_L u_L^2\right)+\left(1-\alpha\right)\frac{dP}{dz}+\left(1-\alpha\right)\rho_L g = -F_{WL} + F_{LG} - F_{LI} \qquad (4)$$

- Mixture energy equation

$$\frac{d}{dz}\left[\left(1-\alpha\right)\rho_L u_L H_L + \alpha\rho u_G H_G\right] = \frac{q_w P_h}{A} \qquad (5)$$

B. Numerical Resolution [23, 24]

Equations 1 to 5, which govern the evaporation of refrigerant flowing in the vertical tube, have five unknown parameters, namelyα, u_L, u_G, P, and h_L. To solve this set of equations numerically, the tube was divided into infinitesimal sections. The heat flux, q_W, and the following inlet operating conditions for the refrigerant were considered:

– Inlet saturation pressure or inlet mixture temperature;
– Mixture mass flow rate;.
– Inlet flow quality.

The equations were solved using the fourth order Runge-Kutta method and for operated conditions illustrates in the tab. 1.

Table I
OPERATING CONDITIONS CONSIDERED FOR SIMULATION

	Parameters	**Values**
Heat flux, *q*	$kW.m^{-2}$	1 – 30
Tube diameters, *D*	mm	4, 6, 8, 10
Mass flow rate, *G*	$Kgm^{-2}s^{-1}$	50
Tube length, *L*	m	1.000
ammonia fraction in inlet solution, X_{in}	-	0.4
Inlet pressure, P_{in}	bar	15

III. SIMULATED RESULTS

A. Liquid velocity

Fig. 3 presents the outlet liquid velocity evolution *vs.* heat flux for five different tube diameters at a constant operating pressure of 15 bars and ammonia fraction in inlet solution of 0.4. For tube diameters, namely, 4 and 6 mm, the liquid velocity increases, reaches a maximum and then decreases. But it increases for a diameter exceeds 8 mm. The value of heat flux that corresponds to the maximum liquid velocity depends on tube diameter. The heat flux required to produce the maximum liquid velocity is about $15 kW/m^2$ for 4 mm and 25 kW/m^2 for 6 mm tube diameter. While for 8, 10 and 12 mm tube diameter, maximum outlet velocity was not achieved for the range of operating heat input. As the tube diameter increases, the frictional pressure drop decreases and the occurrence of the maximum liquid velocity is shifted to the right side, *i.e.*, to the higher heat input side.

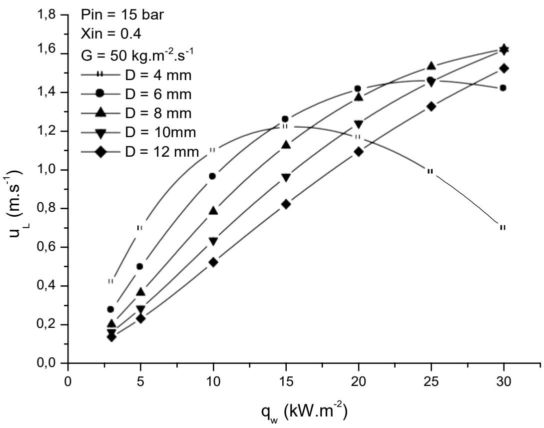

Fig. 3 Influence of heat flux on the outlet liquid velocity for different tube diameter

B. Vapor velocity

The effect of heat flux on vapor velocity is presented in fig. 4. They show a linear increasing of outlet vapor velocity with the heat flux. The outlet vapor velocity decreases if tube diameter increases. This is related to the increase in the hydraulic load in tube.

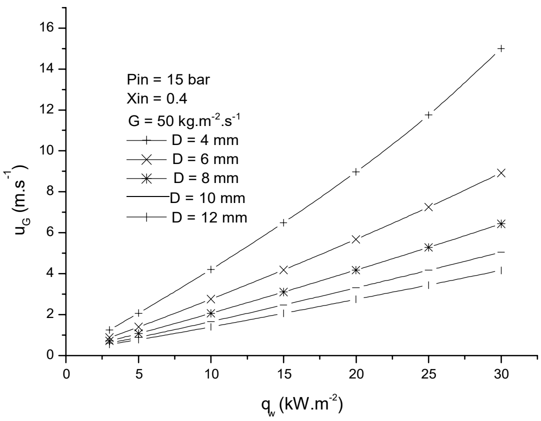

Fig. 4 Influence of heat flux on vapor velocity for different tube diameter

ISSN: 2356-5608

C. Pumping ratio

The pumping ratio R_p is the ratio of liquid velocity (u_L) to vapor velocity (u_G). The variation of the pumping ratio with the heat flux is plotted for different tube diameters in fig. 5.

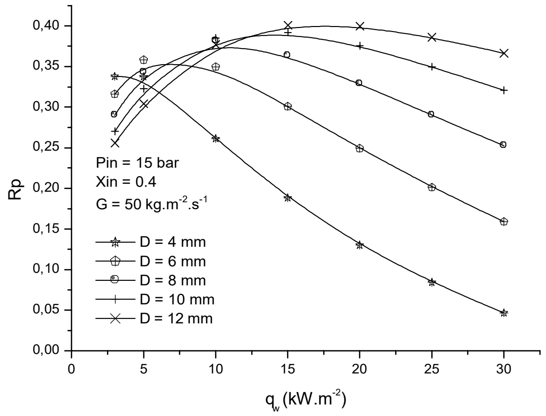

Fig. 5 Effect of heat flux on pumping ratio for different tube diameter

From theses curves, the general behavior of the pumping ratio with respect to the heat flux seems similar. The evolution of pumping ratio versus heat input at different operating conditions presents a maximum. This which gives the optimal value of heat flux input. Indeed, the optimal value of heat flux is the value that corresponds to a maximum pumping ratio.

The variation of the tube diameter has a significantly affects on the pumping ratio. Figure shows that for the small diameters (D = 4 and 6 mm), the maximum pumping rate is reached at a lower heat flux, ranging between 3 and 5 kW/m^2. It increases when the tube diameter increases (10, 15 and 17.5 kW/m^2 for D = 8, 10 and 12 mm respectively). However Pfaff *et al.* [7] showed that the pumping ratio is independent of heat input for a rung tube diameter between 10 and 18 mm, which is due to the fact, there's no transition flow regime for the tubes diameters studied. They showed an increase with driving head, a result indicates a transition regime.

As can be see, a certain minimum heat input denoted (Q_{min}) is required for the pump to operate. Before Q_{min} was reached, the driving force was not sufficient for pumping action. The mixing ammonia-water boiled off but the solution only oscillated in the pump without being conveyed to the boiler [7]. It is important to notice that for most significant diameters, the range of heat flux corresponding to maximum pumping ratio is broader. In fact, pumping ratio is slightly influenced by moderate variation of heat flux.

IV CONCLUSION

In the present work, the two-fluid model was used for the two-phase flow region of ammonia-water mixing, considering the hydrodynamic non-equilibrium between the liquid and vapor phases. On the basis of the numerical simulations, a detailed description of the flow characteristics was obtained. The influence of the heat input and tube diameter on the flow characteristics was examined. The following conclusions are deduced:

- Vapor velocity varies linearly with heat flux, whereas the liquid velocity and the pumping rate variations present a maximum values that depend on operating conditions.
- The bubble pump optimum functioning are defined when the pumping ratio is maximal. The results shows, the value of heat flux corresponding to the optimal functioning are more influenced by tub diameter (3kW/m^{-2} for a 4mm of tube diameter and 5, 10, 13 and 17 kW/m^2 respectively for 6, 8, 10 and 12 mm of tube diameter). It increases with tube diameter.
- It is important to notice that for most significant diameters, the range of heat flux corresponding to maximum pumping ratio is broader. In fact, pumping ratio is slightly influenced by moderate variation of heat flux.

NOMENCALTURES

A	- tube cross-section area, [m^2]
D	- tube diameter, [mm]
F_{GI}	- interfacial force for the vapor due to the mass exchange, [Nm^{-3}]
F_{LG}	- interfacial force between the two phases, [Nm^{-3}]
F_{LI}	- interfacial force for the liquid due to the mass exchange, [Nm^{-3}]
F_{WG}	- force between the wall surface and the vapor, [Nm^{-3}]
F_{WL}	- force between the wall surface and the liquid, [Nm^{-3}]
g	- gravity acceleration, [ms^{-2}]
G	- mass flux flowing in the tube, [kgm^{-2}s^{-1}]
H	- enthalpy, [Jkg^{-1}]
L	- length, [m]
P	- pressure, [Pa]
P_h	- heating perimeter of the channel, [m]
q	- total wall heat flux, [Wm^{-2}]
T	- temperature, [K]
u	- velocity, [ms^{-1}]
v	- vapor quality, [-]
z	- axial location along the flow direction, [m]

Greek symbols

α	- void fraction, [-]
ρ	- density,[kg.m^{-3}]
Γ	- vapor or liquid generation rate per unit mixture volume, [kgm^{-3}s^{-1}]

Indice

ISSN: 2356-5608

L - liquid
Go - vapor only
G - vapor
w - wall

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ISSN: 2356-5608