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Research Article

# A new approach to simulate the critical and the onset nucleate boiling heat fluxes for a thermal generator-bubble pump

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## ABSTRACT

The main factor for the proper functioning of the thermal generator-bubble pump of the diffusion-absorption cycles is the amount of heat required to maintain the necessary vaporization for pumping fluids. An excess of heat or the opposite reduces its efficiency.
The present study aims to simulate the critical heat flux (CHF) and the onset nucleate boiling heat flux (ONBHF) versus mass flow in the generator-bubble pump of absorption-diffusion machines. The bubble pump studied is a vertical heated tube of diameter range is between 4 and 12mm, in which flowing an ammonia-water mixing. To achieve our goal a new approach based on the curves of the variation of the pressure drop as a function of the mass flow rate has been adopted. The pressure drops have been simulated using the two-fluid model. The critical heat fluxes simulated as a function of mass flow rate are compared with those obtained from four correlations. A good agreement has been obtained with the correlation of Shah and Zhang et al.
Simulation results allow defining the optimum range of mass flow that should be used in the same application. We found that the mass flow rate should be higher than 40kg/m²s for the diameters of the studied tubes.



## 1. Introduction

Diffusion-absorption refrigeration (DAR) has the advantage of being thermally powered by renewable heat [1]. The heat flux imposed on its driving element called the generator-bubble pump is one of the essential parameters of the absorption-diffusion refrigeration operation. The generator-bubble pump is generally, one or more vertical tubes heated directly or indirectly[2–5].

The working fluid undergoes a phase change by receiving a heat flux and, pumping action in the entire system will be caused by steam generation.The influence of the heat supplied to the generator-bubble pump has been the subject of several works[5–9].

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**Nomenclature**

| | | | |
|---|---|---|---|
| $a$ | A parameter in Eq. (4) | $X$ | Vapor quality [-] |
| $b$ | A parameter in Eq. (4) | $z$ | Flow direction (m) |
| $C_P$ | Thermal capacity (J /(kg.K)) | $\Delta h$ | Enthalpy variation (J /kg) |
| $D$ | Hydraulic diameter (m) | *Greek symbols* | |
| $F_{GI}$ | Interfacial force for the vapor due to the mass exchange (N/ m$^3$) | $\alpha$ | void fraction |
| $F_{LG}$ | The interfacial force between the two phases (N/m$^3$) | $\rho$ | Density (kg/ m$^3$) |
| $F_{LI}$ | Interfacial force for the liquid due to the mass exchange (N/ m$^3$) | $\Gamma$ | Vapor or liquid generation rate per unit mixture volume (kg/(m$^3$. s)) |
| $F_{WG}$ | The force between the wall surface and the vapor (N/ m$^3$) | $\mu$ | Dynamic viscosity (kg/(m.s)) |
| $g$ | Gravity acceleration (m/s$^2$) | $\sigma$ | Surface tension (N m$^{-1}$) |
| $G$ | Mass flow in the tube (kg/(m$^2$. s)) | *Subscripts* | |
| $k$ | Thermal conductivity (W/(m.K)) | *CHF* | Critical heat flux |
| $h_{fg}$ | Latent heat of evaporation (J/ kg) | $f$ | Fluid |
| $h$ | Fluid enthalpy (J/ kg) | $g$ | Gas |
| $L$ | Heated length of the test section (m) | *in* | inlet |
| $P$ | pressure (Pa) | $L$ | Liquid |
| $u$ | Velocity (m/ s) | | |

Among them: is the influence of the different modes of heating. In this regard, Zohar et al., [10] compared the efficiency of the bubble pump in the case of local heating at its inlet and over its entire length. The coefficient of performance of DAR versus heat received has been widely discussed. A maximum coefficient of performance of 0.33 has been defined by Jacob &Eicker, 2002[11] for a prototype 2.5 kW water-ammonia diffusion absorption chiller which provided 1.5 kW of cooling load with a driving temperature of 150 to 170 ℃. Also, the temperatures vary in the different compounds in industrial absorption-diffusion refrigerators (DAR) were studied for different heat flux imposed by the generator-bubble pump [6]. The operating fluid in the generator-bubble pump undergoes a phase change under the heating process. The two-phase flow in the bubble pump has been the subject of the study conducted by Benhmidene et al. [12]. In this work, the authors proved the use of a two-fluid model to describe the behavior of two-phase flow in the bubble pump. Another work has been focused on the influence of heat flux on flow regime distribution along with the bubblepump [13]. The authors showed that a low heat flux results in the dominance of the bubble regime, where the refrigerant release is incomplete, as well as the pumping action, is weak. On the other hand, an annular regime along the bubble pump is the result of a thorough heating process. In this case, we see a blockage of the pumping action too [14]. Benhmidene et al., [15] correlated the optimal heat flux in the function of mass flow rate and tube diameter too.

Accordingly, optimal functioning of the generator-bubble pump and consequently the machine (DAR), request a selection of heat flux according to the operating parameters. Indeed, the known minimum heat flux, i.e. the onset nucleates boiling heat flux and the critical heat flux is a primordial step before the dimensioning of such a machine. The reduction of the subcooled zone in the generator-bubble pump tube allows the required heat quantity to be optimized. Excess heat also impairs the operation of the generator-bubble pump. The appearance of an annular flow regime along the bubble pump indicates a critical heat flux value achieved. Critical heat indicates a situation in the heating system in which the heated surface dries out and the wall temperature suddenly rises. For heating systems such as the generator-bubble pump, the deterioration of heat transfer causes a considerable increase in its temperature. This can cause flow fluctuations to occur, and can even lead to its destruction [16, 17]. Thus, critical heat flux is one of the most important parameters in two-phase thermodynamics, the accurate prediction of which is essential to ensure the safety and efficiency of the bubble pump. Regarding critical flux, in the generator-bubble pump, there is a complete absence of works addressing this problem. However, some works are interested in the healing processesin a similar geometry to the bubble pump. The bubble pump is a vertical conventional channel with a diameter of more than 2.5 mm [18] which flows a two-phase mixture of refrigerants. Many applications involving boiling flows in the conventional channel (D>=2.5mm) have been found in industrial sectors. Indeed, the critical heat flux has been studied for steam tubes in boilers, compact evaporators [19], and compact heat exchangers [20], tubes in refrigerating industries [21].

Since heat transfer of boiling fluid is quite complex, the prediction of critical heat flux is essentially based on empirical correlations derived from experimental databases. There are many correlations in the literature reflecting this parameter. In the case of cylindrical tubes such as the bubble pump, the majority of studies have been carried out on mini-tubes [22, 23] (diameter less than 2.5mm) and micro-tubes (D<25 µm) [24, 25]. However, some correlations have dealt with the critical flux behavior in the conventional channels. Katto[26] proposed a generalized correlation of the critical heat flux of forced convective boiling in uniformly heated tubes.Katto and Othno[27] postulate that there is a hydrodynamic condition responsible for the critical heat flux. An analysis of the experimental results extracted from the

litteraturefor seven different fluids allows us to distinguish four flow regimes and to give a general correlation of the critical heat flux. Similarly, Kottowski[28] proposed a correlation for the same tube geometry based on 170 data from 11 sources. His correlation depends on thermal and hydrodynamic flow parameters such as the undercooling state, inlet steam quality, mass velocity, and latent heat of vaporization.

Another correlation of critical heat flux was defined by Zhang et al. [29]. This correlation was formulated for water under saturated and subcooled flow conditions in vertical tubes with diameters between 0.33 and 6.23mm. A good concordance is shown by comparing the results of this correlation with those of Hall-Mudawar[30] and Shah [31].Shah [31] has also invented a new correlation were translates the critical heat flux behavior for upward flow in a vertical tube. Shah correlation has been defined for saturated and subcooled flows. The author had referred to data from 62 independent sources including 23 fluids for tube diameters ranging between 0.315 and 37.5mm.

These correlations show a high dependence of CHF on mass flow, heated length, and inner diameter. Therefore, the applicability of existing CHF correlations to small diameter channels must be carefully examined in detail. A reliable CHF correlation applicable to a wide range of parameters for small-diameter channels has not yet been developed.

In the present study, the simulation of the critical flux, and the onset nucleates boiling heat flux along the bubble pump were investigated. The operating fluid was a water-ammonia mixture commonly used in previous studies. In this simulation, a new approach based on the pressure drop versus mass flow curves was used. The results obtained are compared with those calculated from some literature correlations.

## 2. New approach to simulate CHF

In a review study of the two-phase flow instability, Ruspini et al. [32] show in Fig .1 the different kinds of fluid states as a function of mass flow. The pressure drop N-shape curve is limited by the total pressure drop of theoretical cases of all liquid and all vapor phases (dot line). In addition, they defined the values of vapor quality in the maximum and minimum N-shape curves (x = 0 and x = 1). The indicated values present the onset of generation vapor (ONB) and the end of the liquid phase.
In the other works, different authors [33-36] were based on the N-shape curve of pressure drop vs. mass flow to define the nature of the fluid.

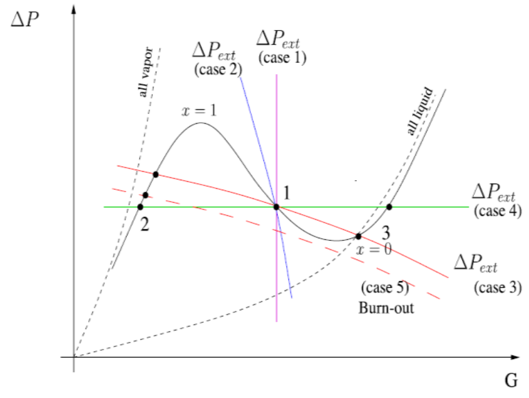


**Fig. 1.** Flow rate characteristic curve for a boiling system. In addition, five different external characteristics curves (cases) [32]

In a previous paper, Benhmidene and Chaouachi [37] simulated pressure drops of ammonia water mixing in a bubble pump tube. The diameters of the test tube are ranging between 4 and 12mm, its length is 1000 mm. The authors used a two-fluid model to conduct their simulation. Balance equations of the two-fluid model used are given in the appendix. They obtained N-shaped curves of pressure drop versus mass flow for different heat flux and tube diameters. An example of these curves is given in Fig. 2, for a heat flow value of 30W/m ²and 8mm of diameter.

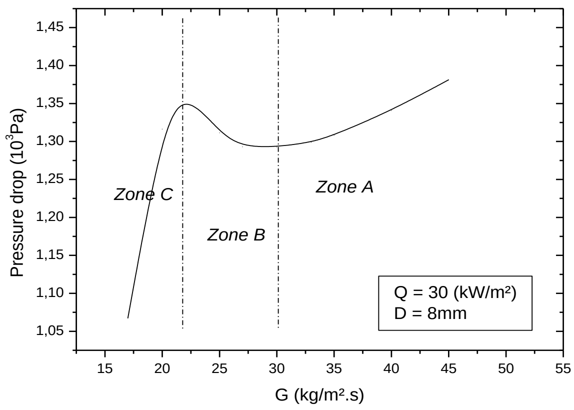


**Fig. 2.** Example of pressure drop vs. mass flow curves

As shown in this curve, three areas corresponding to three fluid states as a function of mass flow can be distinguished:

- Zone A that of pure liquid for the higher mass flow
- Zone B is that of a two-phase liquid-vapor mixture
- Zone C is that of pure steam for the poor mass flow

On the right of the curve, when the mass flow is large enough, flow is a single-phase liquid. Flow begins its boiling by reducing mass flow, and a two-phase flow appears. From this curve the onset nucleates boiling point (ONB) at which the steam quality leaves its zero value can be defined. In this region, pressure drop increases by reducing mass flow.

The limit of the two-phase region corresponds to the extremity of the N curve. At this point, the steam quality is very close to unity. From this point, the mass velocity corresponding to the critical heat flux can be defined. In addition from the ONB point, the mass flow of onset net boiling can be defined too.

Our new approach for the simulation of the onset nucleates boiling and critical heat fluxes is based on the determination of the mass velocities corresponding to the beginning and the end of the two-phase respectively. Knowing the mass flow of the critical flux and its onset nucleate boiling allows us to study the variation of these heat fluxes as a function of the mass flow for different ammonia-water conditions at the inlet. The results obtained will be compared with other correlations.

## 3. CHF correlations

As the majority of studies are interested in critical heat flux, the results obtained are compared with some correlations defined by other authors. In our application, CHF wasn't in the interest of researchers working on the subject of the bubble pump. However, our simulation results are compared to other correlations whose fields of application intersect with ours, such as the heating side (all along the tube, or local), the direction of flow (ascending or descending), the operating fluid (water, coolant, metal solution, etc….), the flow direction (ascending or descending), the geometry of the pipe (cylindrical, annular space, square), the dimension of the pipe (micro pipe, mini pipe, macro pipe), the state of the fluid at the entrance of the pipe (sub-cooled or saturated) and finally the operating conditions, i.e. the ranges of mass velocity and pressure.

Through the comparison and analysis of dozens of round tube CHF correlations, this study has selected the Katto[26], Kottowski[28], Zhang, et al. [29], and Shah [31] correlations which all cover the majority of parameters studied. The parameters range of each correlation and our simulation is shown in Table 1.

### *3.1. Katto correlation*

Critical heat flux was correlated by Katto[26] for low flow rate as follows:

$$q_{CHF} = 0.25Gh_{fg}\left(\frac{\sigma\rho_f}{G^2L}\right)\left(\frac{D}{L}\right)\left(1+1.16\frac{\Delta h_{in}}{h_{fg}}\right) \quad (1)$$

In Eq. (1) there are two terms:

- The first is associated with heat latent heat transport
- The second term is associated with the subcooling effect.

The applicability of the correlation in Eq. (1) is limited to low massflow defined by:

$$\left(\frac{G^2L}{\sigma\rho_f}\right)^{0.29} < \frac{0.4\left(\rho_f/\rho_g\right)^{0.133}}{\frac{D}{L}+0.0031} \quad (2)$$

Beyond this limit, a forced convection CHF criterion should be used. This is given by:

$$q_{CHF} = 0.1Gh_{fg}\left(\frac{\rho_g}{\rho_f}\right)^{0.133}\left(\frac{\sigma\rho_f}{G^2L}\right)^{1/3}\frac{1}{1+0.0031L/D} \quad (3)$$

### *3.2 Kottowski correlation*

Kottowki[27] had defined his correlation from its with Gorlov et al. [38]. He was importing some modifications to be applied to the metal solution. The final expression of Kottowski correlation is as follows:

$$q_{CHF} = a\left(\frac{G^b}{(L/D)^{0.8}}\right)(1-2X_{in})h_{fg} \quad (4)$$

For tube geometries, a = 0.216 and b = 0.807.

### *3.3 Shah Correlation*

Based on 2562 critical heat flux data points in a uniformly heated round tube, Shah [31] presented his new correlation. Some local parameters are the average compound of this correlation such as "true" mass quality (X), mass flow (G), and tube diameter (D).

The expression of the correlation of Shah is as follows:

$$\frac{q_{CHF}}{Gh_{fg}} = 0.124\left(\frac{L}{D}\right)^{0.89}\left(\frac{10^4}{Y}\right)(1-X) \quad (5)$$

$$Y = G^{1.8}D^{0.6}\left(\frac{C_p}{k_L\rho_L^{0.8}g^{0.4}}\right)\left(\frac{u_L}{u_g}\right) \quad (6)$$

**Table 1.**
Parameter ranges of CHF simulated and correlations [26-31, 39, 40]

| Parameters | Present simulation | Zhang et al. Correlation | Shah Correlation | Katto Correlation | Kottowski Correlation |
|---|---|---|---|---|---|
| Canal | Vertical tube | Vertical tube | Vertical tube | Vertical tube | Vertical tube |
| Flow direction | Vertical Ascendant | Vertical Ascendant | Vertical ascendant | Vertical ascendant | Vertical ascendant |
| State of the inlet flow | Saturated | Saturated | Saturated & subcooled | Saturated | Subcooled |
| Working fluid | Ammonia-water | Water | Water, ammonia, freon, potassium | Water | Metal solution |
| Diameter (mm) | 4-12 | 0.33-6.22 | 0.6-38 | 1-3.8 | 2-10 |
| Mass flow (kg/m ²s) | 10-140 | 5.33-1.34x105 | 6-24300 | 10.5-8800 | 50-800 |
| Pression (MPa) | 12-21 | 0.10-19 | 0.023-19.6 | 0.5-20 | 0.004-0.12 |

### *3.4 Zhung et al. correlation*

For a saturated flow, Zhang et al. [29] correlated the critical heat flux during flow boiling in mini and macro-channels. The indicated correlation is expressed by:

$$\frac{q_{CHF}}{Gh_{fg}} = 0.0352\,(W_{ed}\,0.0119\,(\frac{\rho_g}{\rho_L})^{0.361} \qquad (7)$$

$$(\frac{L}{D})^{2.31})^{-0.295}(\frac{L}{D})^{-0.311}[2.05(\frac{\rho_g}{\rho_L})^{0.170} - X_{ir}$$

$W_{ed} = \frac{G^2 L}{\rho\sigma}$ : Weber number of dispersed phase

## 4. Simulation results

Our simulation of critical heat flux and onset nucleate boiling heat flux has been deduced from the results of pressure,drop versus mass flow curves. Table 2 resumes the operating conditions adopted nto simulate pressure drop in the bubble pump [37].

### *4.1 Critical heat flux (CHF)*

As is explained in the last section, by knowing the mass flow corresponding to the critical heat flux for different tube diameters, the critical heat flux curves vs. mass flow are shown in Fig. 3 for different pipe diameters. The curves of Fig. 3 have an affine shape for all pipe diameters studied. This behavior is similar to the usual curves obtained by other authors who studied the critical heat flux as a function of mass flow [24, 26-31, 39-43]; the result shows the success of our approach in predicting the critical heat flux. However, we notice a small difference between the critical

**Table 2**
Operating conditions considered for simulation

| **Parameter** | **Values** |
|---|---|
| Heat flux ($kW.m^{-2}$) | 2 - 150 |
| Tube diameter (mm) | 4 ; 6 ; 8 ; 10 ; 12 |
| Mass flow rate ($kg.m^{-2}.s^{-1}$) | 5 -200 |
| Tube length (mm) | 1000 |
| Ammonia concentration at the inlet | 0,4 |
| Inlet pressure pump (bar) | 18 |

heat flux values for the low mass flow. It increases with the increase of mass flow. Indeed, for low velocities, the fluid resistance time in the tube is sufficient to reduce the effect of gravitation due to the increase in diameter.

For the same mass flow, an increase in pipe diameter increases the CHF values. This is because the hydraulic load increases with the pipe diameter and consequently the volume weight this required additional heat input to reach the CHF. Therefore, the critical heat increases with the volume weight for a constant mass flow. However, this effect is weaker with the reduction in mass flow. Same results were obtained by others authors [44, 45] where CHF increase with the increasing of tube diameter. The simulated critical heat flux was compared to the results calculated from the correlation of Katto, Kottowski, Shah,and Zhang et al., for an inlet pressure of 18 bar,and an inlet mass fraction of 0.4. (Figs. 4 & 5) show the comparison results for the diameters of 6 and 8mm respectively.A global view shows that the simulated critical heat flux takes the same shape as those calculated from correlations for the diameters studied.Moreover, our simulation results arecloser to those of Shah and Zhung than of Katto and Kottowski. The results of the Katto correlation are the farthest from our results.

This can be explained by the difference between these parameters and ours. Table3 shows the relative errors of the simulated critical flux and those calculated from correlations. The relative error is greater than 50% and reaches 63% for the Katto correlation.

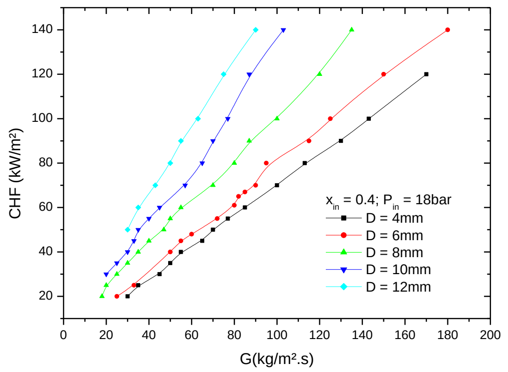


**Fig. 3.** The influence of tube diameter on the critical heat flux behavior

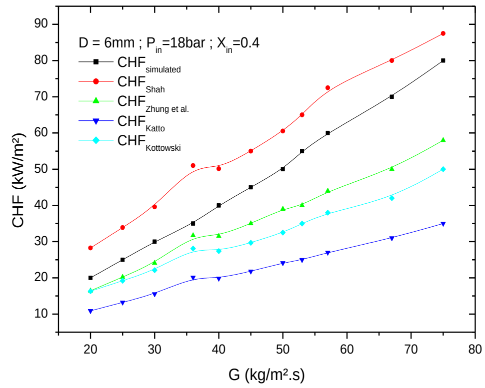


**Fig. 4.** Comparison of CHF simulated & calculation from correlations for D = 6mm

Indeed, this correlation is a compound of two terms, the first is associated with the latent heat transport and the second is the subcooling effect. In our simulation, the second term is null since the inlet fluid is assumed to be saturated. In addition, the Katto correlation takes into account the viscosity and density of the operating fluid which are different from the water-ammonia mixture, another factor that favored the deviation from our simulation results.

The Kottowski correlation is also a function of the subcooling effect and the heat transfer effect. However, the deviation from our simulation results reduces in comparison to those of the Katto correlation. Error is between 30.8 and 47.5%. As it's indicated by Eq. (4), the heat transfer effect term depends only on the mass flow, tube diameter, tube length, and the latent heat of vaporization. The same parameters have been taken into account in our simulation. This has the effect of reducing the deviation from that obtained by applying the Katto correlation.

**Table 3**
Relative errors of simulation and correlation results.

| Diameter (mm) | Relative error (%) | | | |
|---|---|---|---|---|
| | Shah | Zhung et al. | Katto | Kottowski |
| 4 | 15.787 | 20.630 | 63.215 | 47.084 |
| 6 | 24.61 | 22.73 | 51.55 | 31.829 |
| 8 | 13.617 | 23.97 | 60.534 | 47.558 |

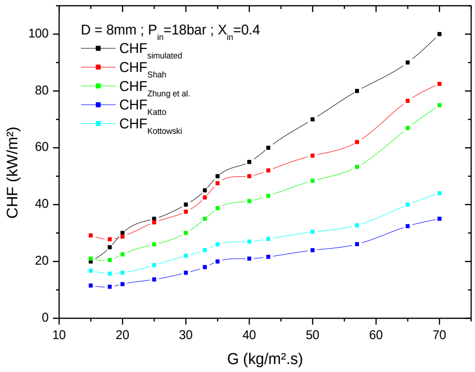


**Fig. 5.** Comparison of CHF simulated & calculation from correlations for D = 8mm

In contrast to the Katto and Kottowski correlations, the comparison of our simulation results with those calculated by Shah and Zhung et al., correlations show a much smaller difference. Indeed, the relative error compared to the results calculated by Shah's correlation does not exceed 26%. A comparable error is obtained with Zhung's correlation. These correlations sufficiently describe our simulation results. The deviations are acceptable because of the difference in the physicochemical properties of the fluids used.In the Zhung et al. correlation where Weber's number is given, which expresses the ratio of inertial forces to gravitational forces, in addition to the effect of heat transfer by evaporation and the effect of pressure (density ratio), we obtain an error in around of 22%.

However, it is difficult to distinguish the impact of each of these parameters on the critical heat flux, the same for the presence of the heat transfer evaporation effect.

In addition, the effect of tube diameter shown in Table 2 for 4, 6, and 8mm doesn't allow giving a hard recommendation for this parameter by referring to relative error for the different diameters.

### *4.2 Onset Nucleate Boiling Heat Flux*

The pressure drop curve against mass flow allows also studying the onset nucleate boiling heat flux versus mass flow too. Since the minimum pressure drop versus mass flow curve corresponds to the zero vapor quality, i.e. the vapor quality corresponding to the beginning of boiling, we can also use this curve to follow the evolution of the heat flux of ONB as a function of the mass flow. If the mass velocity corresponding to the start of boiling is known, the heat flux of net generation vapor that corresponds to the mass flow rate for which the vapor quality is slightly greater than zero.

Fig. 6 shows the variation of onset of nucleate boiling heat flux versus mass flow for a tube diameter of 4, 6, 8, 10, and 12mm. The curves have a linear shape for different diameters. The deviation between them becomes important with the increase of mass flow and for decreasing diameters. In the same way, the slope of the curves also decreases with decreasing pipe diameter.

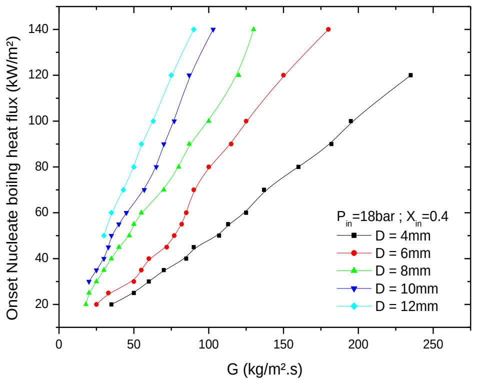


**Fig. 6.** Onset nucleate heat flux vs. mass flow

a constant heating flux can cause a subcooling zone to appear, while the opposite can cause the formation of a drying phenomenon before the fluid leaves the tube to the separator.

For known geometric and operating conditions, the heat flux-mass flow duality is linked. Setting one of the parameters will influence the choice of the other. Indeed, a choice of cooling capacity for a refrigerator operating according to the absorption-diffusion cycle requires the fixing of the refrigerant flow in the evaporator and therefore a choice of mass velocity in the bubble pump. As a result, the heat flux in the bubble pump should be chosen so that the occurrence of subcooling and/or drying out phenomena is eliminated. We aim to locate the gap between the two heat

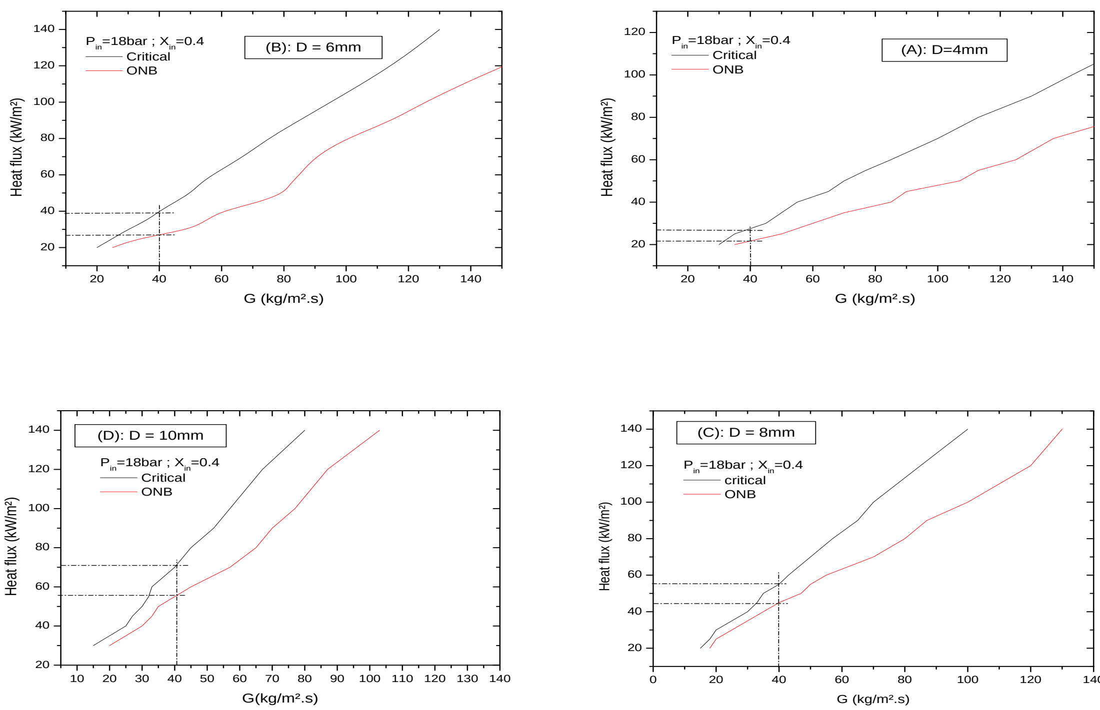


**Fig. 7.** CHF qand ONB heat flux curves: 4mm (A), 6mm (B), 8mm (C), 10mm (D)

### *4.3 Mass flow optimization*

Heat flux is of primary importance for the dimensioning of heat exchangers before the bubble pump. A reduction in the mass flow for fluxes as a function of the mass flow for a chosen bubble pump diameter.

To reach this goal, we plot on the same figures the variations of the critical flux and the onset nucleate boiling heat flux as a function of the mass flow. These variations are shown in Fig. 7 (a), (b), (c), and (d) for the respective diameters of 4, 6, 8, and 10mm. As a global observation, it can be seen that the slopes of these curves increase with increasing tube diameters. However, the minimum difference between the fluxes is about 5kW/m² while the maximum is about 30kW/m². For the 8 and 10mm diameters, the deviation remains close to the minimum value for mass flow lower than 40kg/m²s. In general, this deviation is not influenced enough by the variation of the tube diameter.

The present result has the importance to allow defining the mass flow interval, which must be not used. Indeed, by adopting mass flows lower than 40 kg/m²s we risk falling into the drying or subcooling zone, thus a fluctuation phenomenon occurs in the bubble pump due to the small difference between the critical flow and the net boiling flux[46]. A bubble pump powered by a solar flux, whose intensity is variable over time, requires a heat flux interval allowing the use of this energy source without fluctuation. The adaptation of a low mass flow is not recommended for this application whatever the diameter of the tube.

## 5. Conclusion

The role of the bubble pump in the absorption-diffusion machine is to circulate the working fluid by using the thermal effect. Heat supplied to the bubble pump is limited by the onset nucleate heat flux as the minimum value and the critical heat flux as the maximum. In our study, these parameters have been simulated by using the results of the simulation of pressure drop versus mass flow rate for tube diameters between 4 and 12 mm. Our results have the same shape for the variation of critical and onset nucleate heat flux against the mass flow obtained in the literature. A comparison of our simulated critical heat flux with those calculated from Shah Correlation shows a relative error of less than 16% for the tube diameter of 4 and 8mm. a comparable error has been obtained against Zhang et al. correlation where it's about 23%. However, an error of more than 31% has been found in comparison with the Kottowski correlation. It increases to achieve 61% for the Katto correlation. Our results of simulation of critical and net generation vapor heat fluxes allow defining a value of 40kg/m²s as the limit

In the two-phase region, the general conservation equations of mass, momentum, and energy were formulated by Ishii and Mishima [47]. For the steady-state flow with negligible kinetic and potential energy, the conservation equations of the two-fluid model are written as follows:

## Appendix

- Phase mass equations

$$\frac{d}{dz}[\alpha\rho_g u_g] = \tau_g \qquad \text{(A1)}$$

$$\frac{d}{dz}[(1-\alpha)\rho_L u_L] = \tau_L \qquad \text{(A2)}$$

- Phase momentum equations

$$\frac{d}{dz}[\alpha\rho_g u_g^2] + \alpha\frac{dP}{dz} + \alpha\rho_g g = -F_{WL} - F_{gL} - F_{gI}$$

(A3)

$$\frac{d}{dz}[(1-\alpha)\rho_L u_L^2] + (1-\alpha)\frac{dP}{dz} + (1-\alpha)\rho_L g = -F_{WL} - F_{Lg} - F_{LI}$$

(A4)

- Mixture energy equation

$$\frac{d}{dz}[(1-\alpha)\rho_L u_L h_L + \alpha\rho_g u_g h_g] = \frac{4q}{D} \qquad \text{(A5)}$$

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