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Egoi Ortego Sampedro, Florent Breque, Maroun Nemer

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# Two-phase nozzles performances CFD modeling for low-grade heat to power generation: mass transfer models assessment and a novel transitional formulation

## Keywords

Low grade heat valorization; motive nozzle efficiency; two-phase flow; interfacial area density; thermal phase change model; ejector ; ORC

## Highlights

- Four phase change models were compared in a two fluids metastable flow description
- A novel transitional area density formulation was explored for motive flash nozzles
- Simultaneous calibration of flow rate and velocity was not possible for some models
- The best models are the ones based on multiple and non-redundant calibration terms

10

## Abstract

The use of two-phase nozzles for low-grade heat valorization by electricity production increases the energy recovery rate using Trilateral Flash or Wet to Dry cycles. A model benchmark for nozzle flow rate and efficiency estimation was conducted on experimental data from the literature. These data come from a series of tests made on geothermal energy production water two-phase nozzles. A new transitional bubble-to-droplet interfacial area density formulation (TA-BD model) is presented by the paper. It is compared to three models from literature. Two nozzles operating with different inlet and outlet conditions were modeled. The calibration flexibility and the robustness of the models are discussed in association with physical analysis. The paper shows how the models using single or redundant adaptation parameters fail to provide good results simultaneously on flow rate and efficiency. It appeared that the TA-BD model is the more flexible and robust. The Homogeneous Relaxation Model (HRM) model gives also good results. Furthermore, TA-BD model gives the lowest average discrepancies. Especially at the best efficiency point of the first test case, TA-BD model shows less than 1% discrepancy where the HRM model has 18% discrepancy in efficiency. The TA-BD model appeared to be easier to calibrate than the HRM model. Finally, regarding the proposed TA-BD model, the sensitivity to the geometry and operating conditions shows that the interfacial area density formulation could be completed to include the effect of nozzle's section profile, the effect of the inlet temperature on bubbles number density, and the effect of outlet pressure on the droplets number density.

24

<b>Nomenclature</b>			
<i>Roman characters</i>		$\alpha$	volume fraction
$A$	interfacial area density [ $m^{-1}$ ]	$\Gamma$	volumetric mass flow rate [ $kg\ m^{-3}s^{-1}$ ]
$a$	empirical coefficient for HRM model	$\eta$	isentropic efficiency
$b$	empirical coefficient for HRM model	$\theta$	relaxation time [s]
$C_D$	drag coefficient	$\lambda$	thermal conductivity [ $W\ m^{-1}$ ]
$C_p$	isobaric heat capacity [ $J\ kg^{-1}\ K^{-1}$ ]	$\lambda_{acc}$	accomodation coefficient
$D_l$	drag force [ $N\ m^{-3}$ ]	$\mu$	dynamic viscosity [Pa s]
$D_T$	thermal diffusivity [ $m^2\ s^{-1}$ ]	$\rho$	density [ $kg\ m^{-3}$ ]
$h^t$	total enthalpy [ $J\ kg^{-1}$ ]	$\tau$	viscous stresses
$h_i$	interfacial heat transfer coefficient [ $W\ m^{-2}K^{-1}$ ]	$\phi$	mass transfer potential
$Ja$	Jakob number	$\psi$	pressure ratio
$K$	relaxation coefficient	<i>Subscripts</i>	
$l_c$	interfacial length scale [m]	1	Liquid phase
$N$	number density [ $m^{-3}$ ]	2	gas phase
$M$	interface forces [ $N\ m^{-3}$ ]	$b$	bubble
$P$	pressure [Pa]	$crit$	critical
$Pe$	Peclet number	$d$	droplet
$Pr$	Prandtl number	$gl$	gas to liquid
$q_1$	interfacial heat transfer [ $W\ m^{-3}$ ]	$i$	interfacial
$R_g$	universal gas constant [ $kg\ kmol^{-1}$ ]	$in$	inlet
$s$	entropy [ $J\ kg^{-1}\ K^{-1}$ ]	$mix$	mixture
$T$	temperature [K]	$ml$	methastable liquid
$U$	velocity [ $m\ s^{-1}$ ]	$out$	outlet
$vof$	volume fraction	$s$	isentropic
$x$	quality	$sat$	saturated
<i>Greek characters</i>		$sg$	saturated gas
		$sl$	saturated liquid
		$slu$	slug
		$T$	Taylor

$p$	first phase
$p'$	second phase
$\Delta P$	Pressure based model
Acronyms	
CFD	Computational Fluid Dynamics
HK	Hertz- Knudsen

HRM	Homogeneous Relaxation Model
SC	Sub Cooling
Sch	Super Heat
TA-BD	Thermal Bubble to Droplet
TFC	Trilateral Flash Cycle
VOF	Volume fraction
WDC	Wet to Dry Cycle

## 25 1. Introduction

26 Electricity production from waste-heat is a promising solution for the reduction of fossil fuels usage. Advanced Trilateral Flash Cycles (TFC) and Wet  
 27 to Dry Cycles (WDC) can increase the profitability of low-grade heat based production plants [1]. Compared to a Rankin cycle, TFC and WDC cycles  
 28 present a higher output power density thanks to the better matching between the hot source and the working fluid temperatures at the boiler [3].  
 29 An example of a trilateral flash cycle in a temperature-entropy diagram is shown in Figure 1. Since the flow rate of the sensible hot stream of  
 30 classical waste-heat sources is finite, the ideal cycle leading to the maximum output power has a triangular shape.

31 In such cycles the key element is the two-phase turbine; it can produce power from an initially sub-cooled or saturated flashing liquid stream [3]. An  
 32 example of a two-phase nozzle and turbine arrangement is presented in Figure 2.

33 The renewed interest in these types of cycles and turbines is attested by recent publications on advanced Organic Rankin Cycles (ORC) like the  
 34 experimental work of Iqbal et al. on two-phase impulse turbines [1] or the numerical exploration of White on wet-to-dry cycles [4].

35

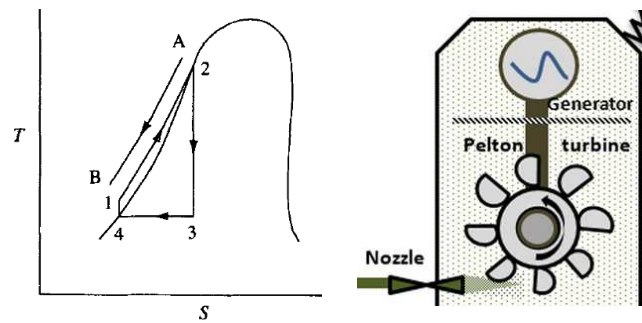


Figure 1 : trilateral flash cycle [2]

Figure 2 : two-phase nozzle and  
impulsion turbine arrangement [1]

36 In such systems, the main part of the expansion occurs in the two-phase nozzle. The nozzle can be static [1] or rotating like in the numerical  
 37 exploration of the very low-temperature turbine of Rane & He [5]. Its behavior modeling is fundamental for such systems design and optimization.

38 The general behavior of the flashing flow in a motive nozzle is presented in Figure 3. The flow's pressure and velocity evolutions in the nozzle are  
 39 represented by the side "A" of the figure; the nozzle inlet is in general in a slightly sub-cooled state then, when flowing through the convergent,  
 40 liquid accelerates so the static pressure decreases and finally, cavitation occurs at the throat. In the divergent, the flow accelerates continuously up  
 41 to the outlet (in adapted flow conditions). The side B of the figure shows, in a P-h diagram, the actual expansion; it follows a path somewhere  
 42 between isenthalpic and isentropic expansions.

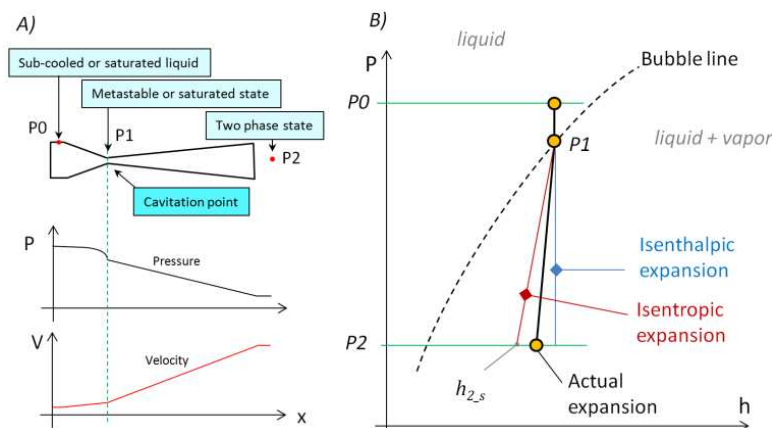


Figure 3 : general motive nozzle behavior

43

44

45 The nozzle efficiency is related to the isentropic velocity (assuming negligible inlet velocity) as follows:

$$46 \quad \eta = \frac{V_{gt}^2}{V_m^2} = \frac{V_{gt}^2}{2(h_{2,s}(P_2, s_0) - h_0(P_0, T_0))}$$

47 The actual outlet velocity corresponds to the liquid and vapor mixture velocity ( $V_{gt}$ ). It depends on several factors such as the nozzle shape, the  
48 operating conditions, the metastability degree, the outlet vapor void fraction, or the friction between phases. So, designing high efficiency nozzles  
49 or even predicting their efficiency can be relatively complex. However, this is fundamental when designing the thermodynamic systems mentioned  
50 before. Nevertheless, the efficiency or even the outlet velocity is never considered in the literature dealing with flash nozzles CFD modeling.

51 Regarding the literature on flash nozzles, most of the early research studies were done on Loss of Coolant Accident (LOCA) nozzles. These studies  
52 were based on experimental and theoretical approaches [6]. One of the main goals was to predict the critical mass flow rate in case of rapid loss of  
53 coolant in nuclear plants. However, after the first two oil crises, increased interest in geothermal power and waste-heat energy recovery arose;  
54 using two-phase turbines looked promising in this context; consequently, various studies were conducted on two-phase turbine nozzles [2]. The  
55 prediction methods were based on 1D two-fluid approaches and on thermal phase change closure models as can be read in the work of Bussac [7].

56 The recent CFD methods mainly benefited the development or optimization of ejectors for transcritical CO<sub>2</sub> cycles. These methods are often derived  
57 from former 1D modeling approaches as can be read in [8]. As for the two-phase turbines, two-phase nozzles are also the motive elements of the  
58 ejectors and nozzle efficiency is a crucial parameter. However, in this sector, the entrainment ratio is studied at the expense of the nozzle efficiency  
59 [9].

60 Furthermore, literature on flash boiling in fuel injectors of IC engines exists. The research in that sector analyses in particular the impact of the early  
61 phase change on the shape of the fuel spray in the combustion chamber. The CFD analysis of Devassi on a fuel injection nozzle [10] is a typical  
62 example of the kind of studies conducted in this sector. Such studies are focused on mass flow rate and liquid jet shape prediction [11].

63 These works and others not cited here raised and tried to answer several questions. Particularly: What is the slip between liquid and gas? What  
64 should be the phase change formulation? At what pressure does the phase change start? How efficiently calibrate the models? An interesting  
65 discussion about these aspects is proposed by [12], a comprehensive literature survey on flash flows.

66 To the best knowledge of the authors, the nozzle efficiency prediction was not discussed in the cited or any other papers. So the present paper  
67 tends to fill this gap. It is done by a phase change models assessment focused on the ability of the models to reproduce the outlet flow conditions of  
68 experimental data from the literature. For this purpose, experimental test cases where nozzle outlet velocities were measured are used as  
69 reference.

70 Furthermore, a number of the papers cited above and many others are based on models that do not exploit the concept of interfacial area density;  
71 this is the case of the well known Homogeneous Relaxation Model (HRM) [13] used either for water or for CO<sub>2</sub> [8] and the case of models based on  
72 standard accommodation coefficients for mass sources [9]. Besides, other families of modeling strategies use the interfacial area density expression  
73 corresponding to the bubbly regime for nozzles where the void fraction could reach up to 0.7, like in the thermal phase-change modeling of [14] or  
74 the Hertz-Knudsen model (HK) based studies of [11]. Knowing that the bubbly regime is limited to void fractions below 0.3 [15], the interfacial area  
75 density formulation using this description is neither valid for the conditions mentioned above nor for two-phase turbines nozzles' since in that last  
76 case void fraction attains easily values up to 0.9 [16] which corresponds to a droplet regime situation.

77 To the best knowledge of the authors, the topology transition from a bubbly to a droplet regime was never before considered in literature dealing  
78 with flash nozzles CFD modeling. To fill this gap, a model never used before for such nozzles is explored in the paper. Indeed, when trying to  
79 calibrate the models on the mass flow and the outlet velocity simultaneously, the poor flexibility and difficulty of calibrating the models available in  
80 literature brought the author to explore a transitional interfacial area formulation that is a variant of the formulation proposed by Wu [15]. This  
81 modified model is also included in the model benchmark; it is named for the article "non-symmetric Thermal-Bubble-Droplet" (TA-BD) model.

82 In summary, this article deals with two aspects of two-phase nozzles modeling never treated before:

- 83 • The ability of literature phase change models to predict the nozzle efficiency in conjunction with the flow rate
- 84 • The exploration of a new variant of a transitional interfacial area formulation that aims to highlight the importance of this parameter in  
85 expanding flash flows especially when trying to predict the efficiency

86 The detail of literature models and their pros and cons are discussed in section 2.1.2.

87 The improved knowledge on CFD flashing flow modeling could improve the shape optimization techniques aiming to design more efficient  
88 components and systems. For this purpose, two nozzles were studied to evaluate the effect of the nozzle shape on the models' behavior.

89 The article is structured as follows. In section 2, the general equations treated by the multiphase CFD solver are presented. From this generality, the  
90 specificities and simplifications done by different models in the literature are introduced. The main phase-change mass transfer models of literature  
91 are discussed before the TA-BD model is presented. Subsequently, test cases based on two nozzle geometries are presented. Then in section 3 the  
92 simulation results and the models comparison are presented. The comparison is focused on the robustness and the ability of models to be  
93 calibrated regarding the mass flow rate and the outlet velocity that defines the nozzle isentropic efficiency.

## 94 2. Methodology

### 95 2.1. Constitutive equations

#### 96 2.1.1. Conservation equations formulation

97 The general formulation of conservation equations for a two phase flow requires a phase per phase description. In the commercial CFD code Ansys  
98 CFX 16.0 these equations, describing the so called Euler-Euler model, are described by equations 1 to 4 for liquid. The same kind of equations is  
99 used for liquid and gas; index 1 is used for the liquid and 2 for the gas.

100 Continuity equation:

$$\frac{\partial}{\partial t}(\alpha_1 \rho_1) + \nabla(\alpha_1 \rho_1 \vec{U}_1) = \Gamma_{12} \quad 1$$

101 where  $\Gamma_{12}$  is the volumetric mass flow rate transferred from liquid to vapor. And  $\Gamma_{12} = -\Gamma_{21}$ .

102 Momentum equation:

$$\frac{\partial}{\partial t}(\alpha_1 \rho_1 \vec{U}_1) + \nabla(\alpha_1 \rho_1 \vec{U}_1 \otimes \vec{U}_1) = -\alpha_1 \nabla p + \nabla \left( \alpha_1 \mu_1 (\nabla \vec{U}_1 + (\nabla \vec{U}_1)^T) \right) + \Gamma_{12} \vec{U}_i + \vec{M}_1 \quad 2$$

103 The term  $\Gamma_{12} \vec{U}_i$  is the momentum source produced by phase change. The term  $\vec{M}_1$  contains the forces transferred from phase 2 to phase 1 i.e. the  
104 interfacial drag forces.

105 Total energy equation:

$$\frac{\partial}{\partial t}(\alpha_1 \rho_1 h_1^t) - \alpha_1 \frac{\partial P}{\partial t} + \nabla(\alpha_1 \rho_1 \vec{U}_1 h_1^t) = \nabla(\alpha_1 \lambda_1 \nabla T_1) + \alpha_1 \nabla(\vec{U}_1 \tau_1) + \Gamma_{12} h_i^t + \dot{q}_1 \quad 3$$

106 where  $\dot{q}_1$  is the interfacial heat transfer rate.

107 The interfacial transfer terms  $\Gamma_{12} \vec{U}_i$  and  $\Gamma_{12} h_i^t$  take into account the flux direction as follows:

$$\begin{cases} \text{if } \Gamma_{12} < 0 \\ \text{if } \Gamma_{12} > 0 \end{cases} \vec{U}_i = \begin{cases} \vec{U}_1 \\ \vec{U}_2 \end{cases} \text{ and } h_i^t = \begin{cases} h_1^t \\ h_2^t \end{cases} \quad 4$$

108 The CFX solver solves also the volume continuity equation, or volume fraction transport equation, expression 5, usually introduced in multiphase  
109 problems. Thus, in the general case 6 (+1) equations are solved.

$$\sum_p \frac{1}{\rho_p} \left( \frac{\partial(\alpha_p \rho_p)}{\partial t} + \nabla(\alpha_p \rho_p \vec{U}_p) \right) = \sum_p \frac{\Gamma_{pp'}}{\rho_p} \quad 5$$

110 The index p refers to the phase (1 or 2) and the index p' to the remaining phase (i.e. 2 or 1).

111 One can read in the literature dealing with flash nozzle flows that the general multiphase description presented above is often reduced to a simpler  
112 formulation, such as:

- 113 • Euler-Euler formulation: this is the general formulation including the 6 generic equations and the volume fraction transport equation (6+1) [14].
- 114 • Mixture formulation: in this case the continuity and energy equations are shared by both phases while the momentum equation can be  
115 completed by a phase slip velocity model. The volume fraction transport equation is also solved for vapor. One gets 3+1 or +2 equations model.  
116 One may refer to [17] for the exact formulation. A simplified description, especially for energy equation, can be read in [18].
- 117 • Homogeneous formulation: here the momentum equation is shared by two phases; velocities are identical. The energy equation can be shared or  
118 not. The original description of this formulation is given by Downar [13].

119 An additional assumption very often found in literature is that vapor phase is saturated.

120 We shall comment on these formulations. Regarding the momentum equation, it seems reasonable to assume that gas and liquid will accelerate to  
121 different degrees given their density disparity and being subject to the same axial pressure gradient along the nozzle. The question is then if the  
122 velocity difference between gas and liquid has any substantial effect on the behavior of the nozzle. According to Yixiang [12] it has an important  
123 effect on the vapor generation rate, possibly because it affects the interfacial heat transfer rate. From an entropic point of view, friction between  
124 phases is a loss of kinetic energy and consequently affects the flow pattern and the exit velocity.

125 Concerning the use of a unique energy equation, since vapor quality attains relatively low values in flash flow nozzles, solving an energy balance  
 126 only for the liquid phase seems reasonable; a number of authors propose this simplification either in a HRM framework [13] or in thermal phase-  
 127 change model situation [14]. This goes often with the assumption that the vapor is saturated.

128 A summary of the type of formulations used in various papers is presented in Table 1; it is not exhaustive but it represents the major research areas  
 129 dealing with flashing flows and the more popular modeling approaches. In addition to the modeling aspects it shows the maximum outlet void  
 130 fraction and the type of modeled geometry.

131 From this table some conclusion can be done:

- 132 • There is a substantial disparity in the interfacial transfer models; this point is treated in the next section
- 133 • Papers using interfacial area density variable ( $A_i$ ) do not take into account the droplet regime, only bubbly regime ( $A_{ib}$ )
- 134 • The majority of references are based on mixture formulation showing the interest of exploring the possibilities offered by Euler-Euler  
 135 formulation
- 136 • There is not a preferred model for turbulence

137 In the next section the most meaningful types of interfacial mass transfer models are presented.

138

139 Table 1 : summary of some flashing flow CFD modeling works

Ref.	General formulation	Evaporative potential	Interfacial area density	Specific models	Turbulence model	Max outlet void fraction	Geometry	Software
[19]	Mixture	$\Delta P$	$A_{ib}$	Slip velocity	k-w SST	0.7	BNL nozzle	Fluent
[18]	Mixture	$\sqrt{\Delta P/P}$ & $\Delta P$		Slip velocity	k-w SST	0.95	CO2 ejector	Fluent
[20]	Mixture	$\sqrt{\Delta P/P}$			k-e	0.7	BNL nozzle	Fluent
[11]	Mixture homogenous	$\Delta P$ $\Delta \dot{x}$	$N_b$ cte or f(T) $d_b$ : cte		k-w SST	0.98	“Moby Dick” nozzle	Fluent
[21]	Mixture homogenous	$\Delta \dot{x}$			Realizable k-e	...	CO2 ejector	Fluent
[22]	Mixture	$\Delta T$	$\dot{N}_b$ : Jones $A_{ib} + A_{istu}$	$N_b$ transport equation	k-e	0.7	BNL nozzle	CFX
[9]	Mixture homogenous	$\Delta T/T$			k-w SST	0.98	CO2 ejector primary nozzle	Fluent
[23]	Mixture	$\sqrt{\Delta P/P}$ & $\Delta T/T$		Slip velocity	Realizable k-e	0.95	R134a ejector primary nozzle	Fluent
[14]	Euler-Euler	$\Delta T$	$A_{ib}$		k-w SST	0.7	BNL nozzle	CFX
[10]	Mixture	$\Delta P$	$A_{ib}; N_b$ f(T)			1	Fuel injector	AVL FIRE
[24]	Mixture	$\Delta \dot{x}$			RNG k-e	1	Fuel injector	CONVERGE

140

## 141 2.1.2. Interfacial transfer terms

142 In this section the mass, energy and momentum interfacial transfer models are presented.

### 143 2.1.2.1. Interfacial mass transfer

144 The mass transfer models are rather diverse. The authors tried here to get a global vision of this diversity and propose a classification in order to  
 145 emphasize their main characteristics.

146 The general formulation of the mass transfer from liquid phase to vapor phase is based on a potential ( $\phi$ ) and on what we will call a relaxation term  
 147 ( $K$ ).

148 The potential represents the distance from a thermodynamic equilibrium. The metastability is an image of this distance to the equilibrium. The  
 149 greater this distance the greater the mass transfer. This is why a metastable situation is intended to disappear. The relaxation coefficient represents  
 150 the relation between the potential and the mass flow; it takes several forms. Here, the models are classified by types of potential. Regarding the  
 151 relaxation terms, only a selection of some formulations used for model comparison is presented here. Also a specific sub-section for the interfacial  
 152 area density is presented at the end of the section.

#### 153 2.1.2.1.1. Pressure difference based models

154 The mass flow potential can be computed as the difference between the saturation pressure and the actual static pressure. The saturation pressure  
 155 is a function of the liquid or mixture temperature. This potential was extensively used in modeling of cavitating flows. The reference model is the  
 156 Singhal model which is based on Rayleigh-Plesset bubble growth analysis [25]. It was also used for depressurized flows [20]. The reader may refer to  
 157 Singhal's work [25] for more details. To the best knowledge of the author, the Singhal formulation is only used for moderate average void fraction  
 158 flows ( $< 0.5$ ). The average void fraction here refers to the same of the Table 1 i.e. the cross sectional device average.

159 For flows dealing with high void fractions, the Hertz-Knudsen (HK) formulation is used. That is the case in the work of Karathanassis [11] and Le [19]  
 160 on water nozzles or Yazdani [18] for CO<sub>2</sub> two phase ejectors.

161 The general expression of the mass flux is based on the kinetic theory of gases and it is named Hertz-Knudsen relation:

$$\Gamma = K_{\Delta P} (P_{cavit}(T) - P) \quad 7$$

162 The relaxation term can take a multitude of forms but the basic relation for  $K_{\Delta P}$  takes the following form:

$$K_{\Delta P} = \lambda_{acc} \alpha_l \rho_l \quad 8$$

163 where  $\lambda_{acc}$  is an accommodation coefficient,  $\alpha_l$  the liquid volume fraction and  $\rho_l$  the liquid density. A multitude of variants of this relaxation terms  
 164 were used by different authors. An interesting benchmark study was done by Karathanassis [11]; he observed that for high temperature water  
 165 flashing flows equation 9 seems to give reasonable results compared to experimental data. It is based on pressure difference potential and an  
 166 accommodation coefficient as described in equation 9 and a constant interfacial area density ( $A_i$ ) defined in equation 10.

$$K_{\Delta P, K1} = \frac{\lambda_{acc} A_i}{\sqrt{2\pi R_g T_{int}}} \quad 9$$

$$A_i = 4 \pi (d_b/2)^2 N_b \quad 10$$

167 The value of  $N_b$  was  $10^{13}$  1/m<sup>3</sup> and the bubble diameter was  $10^{-6}$ m in the cited paper. These values will be used for the computations presented in  
 168 this paper.

169 The interface temperature  $T_{int}$  was assumed to be equal to the local cell temperature.

170 The reference pressure was called the cavitation pressure and was expressed as follows [17]:

$$P_{cavit} = P_{sat}(T) + 0.195 \rho_{mix} k_T \quad 11$$

171 where  $k_T$  is the turbulent kinetic energy and  $\rho_{mix}$  the mixture density. This expression is supposed to capture the effect of turbulence on cavitation  
 172 onset. Note that this expression was originally included in the Singhal model.

173 The conservation equations in Karathanassis were written in a mixture formulation frame work without slip between phases. The details of  
 174 formulations, especially for energy, can be found in Ansys Fluent user's guide [17] or in various articles of which a good example is [26].

#### 175 2.1.2.1.2. Vapor quality difference based model

176 The potential driving the mass transfer can be expressed as the difference between the thermal equilibrium quality and the actual quality. This  
 177 approach was proposed by Downar [13] and reused by Karathanassis [11] for water flash flow nozzles, by Lorenzo [27] for transient water flash  
 178 flows, and by Palacz [21] for a CO<sub>2</sub> ejector.

179 It is called Homogeneous Relaxation Model (HRM) and its general form is:

$$\Gamma = \frac{\rho}{\theta} (\bar{x}(P) - x) \quad 12$$

180 where  $\bar{x}$  is the equilibrium vapor quality and  $\theta$  a relaxation time. It was named "homogeneous" because the two phases were supposed to have the  
 181 same velocity even if they were not sharing the same temperature. Downar [13] expressed the equilibrium quality as follows:

$$\bar{x} = \frac{h_{mix}(P, x) - h_{sl}(P)}{h_{sg}(P) - h_{sl}(P)} \quad 13$$

182 where the mixture enthalpy is:

$$h_{mix} = x h_{sg}(P) + (1 - x)h_{ml} \quad 14$$

183 The authors [13] call  $h_{ml}$  metastable liquid enthalpy; since the mass transfer does not affect the energy equation they used (same for [21]), it seems  
184 to correspond to inlet liquid enthalpy. However, one can read in [28] that liquid enthalpy may change during expansion for HRM model; that  
185 requires including a source term in the liquid energy balance and so was done for the present work.

186 Then the final form of the mass flux is given by Downar et al. [13]:

$$\Gamma = \frac{\rho}{\theta} (1 - x) \frac{h_{ml} - h_{sl}(P)}{h_{sg}(P) - h_{sl}(P)} \quad 15$$

187 The relaxation time  $\theta$  is defined by semi-empirical correlations presented in Downar-Zapolski et al. [13] :

$$\theta = \theta_0 \alpha^a \psi^b \quad 16$$

188 Here  $\psi$  is a non dimensional pressure difference; for pressures lower than 10bar it is defined as:

$$\psi = \left| \frac{P_{sat}(T_{in}) - P}{P_{sat}(T_{in})} \right| \quad 17$$

189 For higher pressures this is defined as follows:

$$\psi = \left| \frac{P_{sat}(T_{in}) - P}{P_{crit} - P_{sat}(T_{in})} \right| \quad 18$$

190 The exponents in equation 16 are fitting coefficients. The recommended values for low and high pressures, in the case of water, are presented in  
191 Table 2.

Table 2 : HRM model empirical constants

$P < 10bar$	$P > 10bar$
$\theta_0 = 6.51 \cdot 10^{-4} s$	$\theta_0 = 3.84 \cdot 10^{-7} s$
$a = -0.257$	$a = -0.54$
$b = -2.24$	$b = -1.76$

193 Note that  $\alpha$  and  $\psi$  are both increasing functions when approaching the outlet of the nozzle. Thus, they should produce a similar effect on relaxation  
194 time. Also the expression of the equilibrium quality (eq. 13) is questionable: the vapor quality is computed from an enthalpy which should be the  
195 equilibrium enthalpy (isentropic or isenthalpic) but it is computed with what seems to be the mixture's meta-stable enthalpy. Nevertheless the  
196 HRM model uses 3 tuning parameters and should therefore have sufficient degrees of freedom for calibration. This ability will be called flexibility in  
197 the next sections.

### 198 2.1.2.1.3. Temperature difference based model

199 The potential associated with the non-equilibrium can also be expressed as the difference between the saturation temperature ( $T_{sat}(P)$ ), function  
200 of the static pressure, and the liquid temperature ( $T_{liq}$ ).

201 The most notable model based on this difference is the so called thermal phase change model. The basic idea is to compute the heat flux associated  
202 with the phase change when mass transfer occurs. The mass flux is:

$$\Gamma = \frac{\dot{q}_i}{\Delta h_{gl}^t} = \frac{h_i A_i}{\Delta h_{gl}^t(P)} (T_{sat}(P) - T_l) \quad 19$$

203 where  $h_i$  is the interfacial heat transfer coefficient,  $A_i$  is the interfacial area density (in  $m^2/m^3$ ) and  $\Delta h_{gl}^t$  is the total enthalpy difference between the  
204 gas and liquid. The interfacial heat flux ( $\dot{q}_i$ ) is applied also to the energy balance. This expression shows the relation between the capacity of the  
205 fluid to internally transfer heat from liquid to vapor and its ability to change phase.

206 This formulation was used by Wu [15] and Yixiang [14] who worked on two phase nozzle flows mostly for nuclear power plant safety issues or in the  
207 work of Bussac [7] who worked of two phase impulse turbines. One of the main goals was to predict the mass flow rate of the nozzles.



208 Regarding the heat transfer coefficient, Yixiang [14] proposes to use the Aleksandrov heat transfer coefficient that was derived from bubble growth  
 209 analysis in superheated liquids [29] to take into account the gas/liquid velocity difference through the Péclet number.

$$h_c = \frac{\lambda_l}{l_c} \left( \frac{12}{\pi^2} Ja^2 + \frac{1}{3\pi} Pe \right)^{1/2} \quad 20$$

210 where  $\lambda_l$  is the liquid's thermal conductivity and  $l$  is the interfacial characteristic length; in the case of bubbly flow, this is the bubble diameter. The  
 211 Jakob number is:

$$Ja = \frac{C_p \rho_l (T_l - T_{sat})}{\rho_g \Delta h_{gl}} \quad 21$$

212 where  $C_p$  and  $\Delta H_{gl}$  are respectively the isobaric heat capacity, and the latent heat of the liquid. The Péclet number is:

$$Pe = \frac{|\vec{U}_g - \vec{U}_l| l_c}{D_{T,l}} \quad 22$$

213 where  $D_{T,l}$  is the thermal diffusivity of liquid. In the cited Yixiang model, the heat transfer coefficient is multiplied by a correction factor  $k_h$  in order  
 214 to add a calibration parameter to the formulation.

215 We could also cite in this section the work of Giacomelli [9] and Geng [23] who use a dimensionless temperature difference for a CO<sub>2</sub> and R134a  
 216 two-phase ejector primary nozzle modeling. They used the  $K_{\Delta P}$  term of equation 7 multiplied by  $(T_{sat} - T)/T_{sat}$ .

217 The interfacial area density  $A_i$ , that can be used in  $\Delta P$  and  $\Delta T$  based models, is an essential parameter of the multiphase flow behavior. That is why  
 218 a specific section is attributed to it in this paper.

#### 219 2.1.2.1.4. Interfacial area density

220 The treatment of the interfacial area density ( $A_i$  in equations 9, 19 and 30) needs to rely on several assumptions if one wants to avoid to use  
 221 detailed interface tracking techniques. A solution proposed by some authors is to model the transport and source terms of variables associated with  
 222 interfacial area. This is the case of the MUSIG (multiple size group [30] ) model that resolves the momentum equation for the dispersed phase using  
 223 an Euler-Euler flow description. It is based on a bubble population balance equation. This model has evolved later to introduce the transition to the  
 224 continuous gas phase (GENTOP model [31]) in an attempt to obtain a general model. The difficulty these methods try to address is the treatment of  
 225 the diversity of characteristic sizes and velocities describing a multiphase flow. It requires additional transport equations (one per sub-size fraction  
 226 for inhomogeneous MUSIG model) and a variety of models for shape transition, coalescence or breakup source terms, etc. Nevertheless the  
 227 generalization remains hard to achieve as can be concluded from the simulation configuration of those models [31].

228 A simpler approach consists in proposing a relation between the void fraction and the interfacial area density. Wu [15] defines it for the first stage  
 229 of phase change ( $\alpha < 0.3$ ) as an increasing function of the void fraction and the number of bubbles per unit volume ( $N_b$ ) Assuming equal sized  
 230 spherical bubbles, Wu writes:

$$A_{i_b} = (6\alpha)^{2/3} (\pi N_b)^{1/3} \quad 23$$

231 The value of the bubbles number was assumed to be constant by Wu. Maksic [22] proposed to use a bubble transport equation including a  
 232 volumetric nucleation rate based on Jones model.

233 When the void fraction increases, the bubbly flow should evolve to a bubbly-slug flow for which Wu gives an approximation based on a Taylor  
 234 bubbles description:

$$A_{i_{slu}} = (4\alpha_T)^{2/3} / (\alpha_{max}^{1/6} D) \quad 24$$

235 where  $D$  is the channel diameter and  $\alpha_T$  is the Taylor bubbles void fraction. This second expression was used, added to  $A_{i_b}$  (equation 23), by Maksic  
 236 to compute the final interfacial are density ( $A_{i_b} + A_{i_{sl}}$ ) [22] with a maximum void fraction ( $\alpha_{max}$ ) of 0.8. No precision about how to get  $\alpha_T$  was given  
 237 by Maksic. Yixiang used  $A_{i_b}$  for void fractions lower than 0.7 in a converging diverging nozzle [14]; in Yixiang's paper, the particle model is  
 238 mentioned. This seems to refer to the CFX particle model [32] in which  $A_{i_b}$  is based on a clipped void fraction ( $\tilde{\alpha}$ ) for high void fractions ( $\alpha > 0.8$ ) in  
 239 order to decrease  $A_{i_b}$  when approaching unit void fraction; this clipping was implemented for the present paper as well.

$$\tilde{\alpha} = \begin{cases} \max(\alpha, \alpha_{min}) & \text{if } (\alpha < \alpha_{max}) \\ \max\left(\frac{1-\alpha}{1-\alpha_{max}}, \alpha_{min}\right) & \text{if } (\alpha \geq \alpha_{max}) \end{cases} \quad 25$$

240 For void fractions higher than 0.7, according to Wu, the flow is considered as being mist flow. The term "mist flow" refers, in gas-liquid flow analysis,  
 241 to a regime where the disperse phase (liquid droplets) is transported by the gas phase. In these conditions, the interfacial area density for constant  
 242 sized droplets is:

$$A_{i_d} = (6(1 - \alpha))^{2/3} (\pi N_d)^{1/3} \quad 26$$

243 The number of droplets per unit volume ( $N_d$ ) was assumed to be equal to  $N_b$  by Wu. Wu proposed a symmetric model assuming that transition  
244 from bubble to droplet regime is done at constant area density from 0.3 to 0.7 void fractions.

245 With an idea similar to the ones of Maksic or Jones, Karathanassis proposed to use a non constant  $N_b$  (equation 27). He referred to a work dealing  
246 with flash boiling sprays in which the authors localized the bubble formation in the initial stage of the flash process and wrote that the nucleation  
247 depends on the degree of superheat of spray corresponding to the injection back pressure at the inlet temperature [33]. It is not clear how this was  
248 implemented in Karathanassis' simulation but it seems that  $N_b$  was computed for every axial location. Otherwise he tested a constant value of  $A_i$ .

$$N_b = 10^{13} \exp(-5.28/\Delta T_{sup}) \quad 27$$

249 Devassi [10] used the same superheat based formulation also for  $N_b$ ; he computed it for every point in the flow. However since the inlet is in a sub-  
250 cooled state, inlet superheat is negative so when going through the expansion, superheating goes through zero and  $N_b$  through infinite; so this  
251 formulation is not adapted for motive nozzles.

252 In the present work a non-symmetric bubble to droplet transition model was tested. To the knowledge of the author, this is the first time this is  
253 implemented and analyzed in any depressurizing two phase flow analysis.

254 The interfacial area density will be modeled by a simple linear interpolation between the maximum bubble regime  $A_{i_b}$  and the maximum droplet  
255 regime  $A_{i_d}$ . This leads to the general formulation expressed by equation 28.

$$A_i = \begin{cases} \text{if } \alpha \leq \alpha_b: (6\alpha)^{2/3} (\pi N_b)^{1/3} \rightarrow A_{i_b} \text{ (bubbly regime)} \\ \text{if } \alpha \geq \alpha_d: (6(1 - \alpha))^{2/3} (\pi N_d)^{1/3} \rightarrow A_{i_d} \text{ (droplet regime)} \\ \text{else : linear interp } A_{i_b} |_{\alpha_b} \text{ to } A_{i_d} |_{\alpha_d} \text{ (transition)} \end{cases} \quad 28$$

256 The values for the transition void fractions were proposed by Wu et al. [15]. These values could be subject to a parametric study, but were  
257 supposed to be  $\alpha_b = 0.3$  and  $\alpha_d = 0.7$ . In contrast to the Wu model, this model does not state for which regime (bubble or droplet) the maximum  
258  $A_i$  is observed and does not assume a constant  $A_i$  between the two regimes.

259 In Ansys-CFX, for Euler-Euler problems composed of two continuous phases, the interfacial area density is expressed as follows:

$$A_i = \frac{\alpha(1 - \alpha)}{l_c} \quad 29$$

260 Please note that in the CFX GUI and theory guide, an Euler-Euler model composed of two continuous phases is named "mixture" model. From an  
261 implementation point of view, the interfacial length scale  $l_c$  has to be given by the user either as a constant either with a CEL expression; so in this  
262 work  $l_c$  was expressed as a function of  $\alpha$  and  $A_i$  ( $l_c = \frac{\alpha(1-\alpha)}{A_i}$  from equation 28).

263 In order to avoid getting non defined values of  $l_c$  at  $\alpha$  values of 0 and 1, the volume fraction is capped to a certain minimum value; a value of  $1e-7$   
264 was taken for this work. Note that liquid volume fraction is  $(1 - \alpha)$  and  $A_{i_b}$  and  $A_{i_d}$  are null at 0 and 1 void fraction respectively. An example of  
265 transitional interfacial area density is given by Figure 4. The part of the curb starting at  $\alpha = 0$  is  $A_{i_b}$  and the part at high values of  $\alpha$  corresponds  
266 to  $A_{i_d}$ .

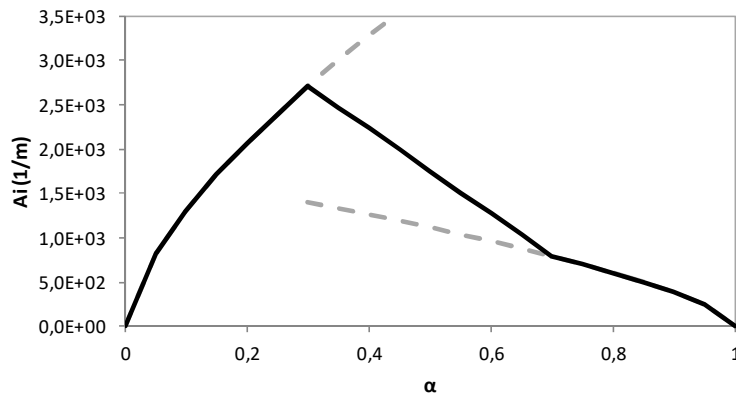


Figure 4 : non symmetric interfacial area density example  $N_b = 2 \cdot 10^9$  and  $N_d = 5 \cdot 10^7$

267

268

### 2.1.2.2. Momentum transfer

270 The interface momentum transfer term ( $\vec{M}_{12}$  in equation 2), includes the drag force between liquid and gas. It has the following form [17]:

$$\vec{D}_I = \frac{C_D}{8} A_i \rho_i (\vec{U}_2 - \vec{U}_1) |\vec{U}_2 - \vec{U}_1|$$

271 No other interfacial forces were modeled here and thus  $\vec{M}_{12} = \vec{D}_I$ .

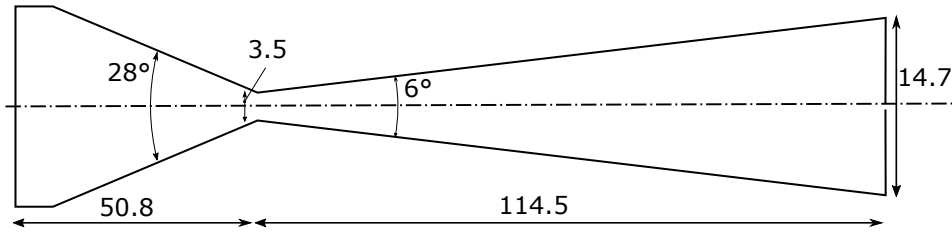
272 The first simulations done for this paper were done using TA-B model [14] where the drag force model is the one of Ishii-Zuber which includes the  
 273 effect of gravity. Some tests were done during the actual research work using Schiller-Neumann [17] and constant drag coefficient models and no  
 274 significant differences were found, notably, the Schiller-Neumann model computed a constant value of 0.44 in the entire diverging section of the  
 275 nozzle. That is why in the models computing liquid and gas velocities separately, equation 30 in conjunction with a constant  $C_D$  value was used.

276 **2.2. Test cases**

277 One of the intentions of this paper is to discuss the advantages of modeling the transition from bubbles to droplets in the interfacial area density  
 278 formulation when trying to predict the nozzle behavior and efficiency in particular. The bubble and droplet numbers may depend on several  
 279 parameters, that is why we selected test cases with different inlet/outlet conditions and geometries. Two nozzle geometries were tested in order to  
 280 explore the effects of nozzle shape of the models.

281 **2.2.1. Ohta B nozzle**

282 In the early 90's, a Japanese team [16] worked on waste heat recovery by impulse turbines using phase change nozzles. Two nozzle geometries  
 283 were tested. The first was called B nozzle which is a fairly simple nozzle. It was widely studied for a wide outlet pressure range. The authors  
 284 measured mass flow rate, efficiency and pressure profiles. The efficiency was obtained thanks to thrust measurements. The dimensions and the  
 285 operating conditions are shown in Figure 5 and Table 3 respectively.



286 Figure 5 : Ohta B nozzle

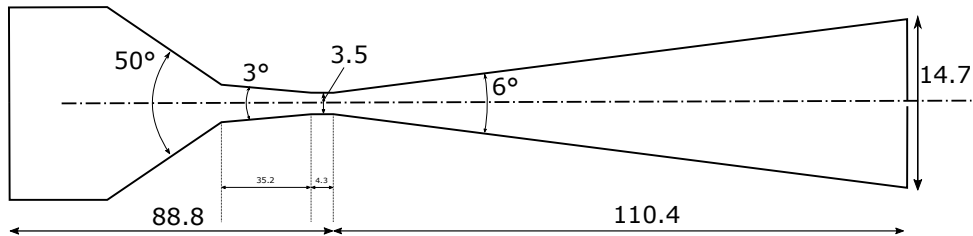
287 Table 3 : Ohta B nozzle operating points

$T_{in}(^{\circ}C)$	$P_{in}(kPa)$	$SC(K)$	$P_{out}(kPa)$	$\dot{m}(g/s)$	Ref.
148	470	1,5	18/45/73/100	122	[16]
137,5	470	12	16/43/70/100	156	[16]

289 The range of operating sub-cooling degrees (SC), inlet temperatures and outlet pressures are particularly useful for the purposes of this article. Note  
 290 that the mass flow in Table 3 does not depend on the outlet pressure; that means that the nozzle operates in critical conditions.

291 **2.2.2. Ohta F nozzle**

292 The Ohta team tested a second nozzle with a long throat called F nozzle; the goal was to increase the cavitation inception pressure and thus the  
 293 efficiency of the nozzle. The authors made mass flow, efficiency and pressure profile measurements as well. The tested operating conditions and  
 294 geometry are presented below.



295 Figure 6 : Ohta F nozzle

296 Table 4 : Ohta F nozzle operating points

$T_{in}(^{\circ}C)$	$P_{in}(kPa)$	$SC(K)$	$P_{out}(kPa)$	$\dot{m}(g/s)$	Ref.
148	470	1,5	20/47/74/100	118	[16]

298 Operating conditions are given in the Table 4. This case allows us to test the validity of the interfacial area formulation for a different section  
 299 variation.

300 The uncertainties of the measured data reported in [16] are within 2% for mass flow rate, 2% for thrust, 1% for pressure and 0.5% for temperature.  
301 This leads to an uncertainty of 2.8% for velocity by uncertainty propagation (knowing that thrust is equal to  $\dot{m} \cdot V$ ) and thus around 5% for efficiency.

## 302 3. Results and discussion

303 In this section the configuration of the CFD solver, a summary of the physical formulation, a mesh sensitivity test and test results for the 3 nozzles  
304 are presented.

### 305 3.1. Configuration and models

306 Four different models were implemented regarding the governing equations and interfacial mass transfer formulation. The first three are based on  
307 models from literature which represent the main families of methods to model the flow and mass transfer phenomena in two phase nozzles. The  
308 last one is a variation of one of them. A summary of the four formulations is given here:

- 309 • HK model based on [11]: mass, momentum and energy conservation equations are shared by both phases. Mass transfer is computed using  
310 equations 7 and 9. Please refer to [11] and [18] for details on governing phase hydrodynamic equations. The main physical characteristics of this  
311 model are: no slip between phases, same energy equation, constant interfacial area density and the use of only one calibration parameter.
- 312 • HRM model based on [13]: mass and momentum conservation equations are shared by both phases. Energy equation is solved only for liquid.  
313 Vapor is assumed to be in saturated condition. Mass transfer is computed using equations 15 to 17. The main physical characteristics of this  
314 model are: no slip between phases, saturated vapor, and the use of three calibration parameters.
- 315 • TA-B model based on [14]: Euler-Euler model assuming saturated vapor. Interfacial drag coefficient equal to 0.45. Mass transfer is computed  
316 using equations 19, 20 and 23. The main physical characteristics of this model are: different phase velocities, saturated vapor, and the use of two  
317 calibration parameters.
- 318 • TA-DB model: Euler-Euler model assuming saturated vapor. Interfacial drag coefficient equal to 0.45. Mass transfer is computed using equations  
319 19, 20 and 28. The main physical characteristics of this model are: different phase velocities, saturated vapor, and the use of two calibration  
320 parameters.

321 The nozzles had as boundary conditions static pressures, temperature and void fraction at inlet and static pressure at outlet. The inlet void fraction  
322 was set equal to 1 for liquid. The wall was configured with default parameters (no slip wall). The nozzle flow was supposed to be axisymmetric. A 3°  
323 revolution form was therefore used to model the control volume with two symmetry planes as its limits.

324 Regarding the state of the fluids, the liquid properties ( $\rho, C_p, \mu$ ) were computed as a function of temperature computed from the enthalpy resulting  
325 from the energy balance which means that the liquid's meta-stable condition was computed by considering it in a temperature based saturation  
326 state and not in a pressure based saturation state. The vapor was supposed to be in pressure based saturated conditions i.e. its properties were  
327 function of the static pressure.

328 The liquid and vapor properties of water were computed from standard IAPWS IF97 tables available in Ansys CFX. The flow is turbulent; the  
329 Reynolds number at the outlet is around  $30 \cdot 10^3$  and around  $130 \cdot 10^3$  close to the throat. The kwSST closure was used as turbulence model since this  
330 is the one preferred for adverse pressure gradient flows that could happen for some operating conditions [17]. One of the advantages of the kwSST  
331 model is its ability to switch from a low-Reynolds boundary layer treatment to a standard boundary layer treatment.

332 The CFX solver is a coupled solver using a pseudo-transient formulation; the coupled option was selected for volume fraction as well. A bounded  
333 second-order upwind scheme was selected for advection. Please refer to Ansys CFX [32] documentation for details on numerical resolution.

334 A steady state simulation was done. The physical time step was set in a range between  $1 \cdot 10^{-4}$  s and  $1 \cdot 10^{-8}$  s depending on the mesh size and the  
335 model. This parameter acts like a relaxation coefficient. The simulation was supposed converged when the mass and energy imbalance was lower  
336 than 0.5%, the inlet mass flow rate was steady and the outlet velocity was steady; all residuals were in this situation lower than  $1 \cdot 10^{-5}$ . The total  
337 energy formulation of the energy conservation equation was selected. The flow field was initialized at 0 vapor volume fraction, at inlet temperature  
338 and pressure and at 0 m/s velocity.

### 339 3.2. Mesh sensitivity

340 In order to maximize the quality within reasonable computational times, a mesh sensitivity analysis was done. The mesh was composed by prisms at  
341 the symmetry axis and hexahedrons everywhere else.

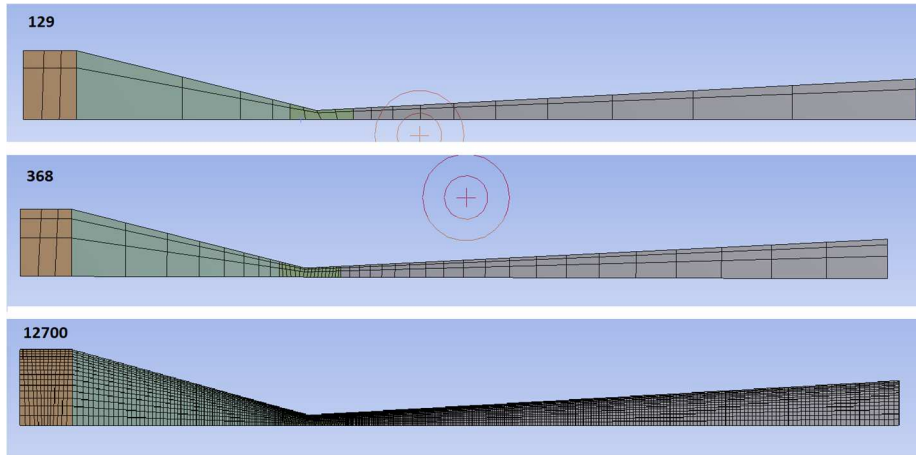


Figure 7 : 3 mesh examples with corresponding number of nodes

342

343

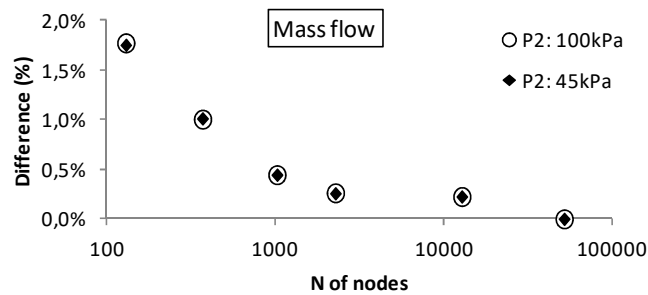
344 During this work it was observed that convergence is improved when several control volumes are used axially dividing the nozzle into several parts;  
 345 this is certainly related to mesh interface control sections added by this operation. The corresponding mesh parts can be seen in Figure 7. In this  
 346 same figure, 3 mesh examples are illustrated; the number in the left top corner is the number of nodes.

347 The tested case is the Ohta B nozzle at inlet condition of 1K sub cooling and two different outlet pressures: 100kPa and 45kPa. The TA-BD model  
 348 was used with the parameters of Table 6.

349 The results were monitored in terms of mass flow rate and outlet velocity. The effect of mesh was quantified comparing the results to those of the  
 350 more dense mesh. The mass flow difference is shown in Figure 8 and the outlet velocity difference in Figure 9. Note that for each calculation the  
 351 case was reinitialized.

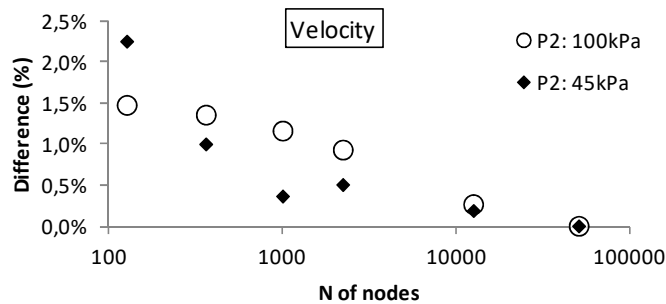
352 The maximum difference to the more dense case was about 2% for both outlet pressures and monitored parameters.

353 For the lighter mesh, the  $Y^+$  values were ranging between 100 and 1000, and for the more dense mesh between 10 and 20. Yixiang [14] obtained  
 354 similar  $Y^+$  values for BNL flash nozzle simulations and the results were relatively well in agreement with experimental observations. That suggests  
 355 that standard wall treatment is quite sufficient to model the dominating physical phenomena of flash boiling in such nozzles.



356

357 Figure 8 : difference with the more dense mesh; mass flow



358

359 Figure 9 : difference with the more dense mesh; velocity

360 The mesh used for simulations was consisted of 12700 nodes, resulting from 15 radial subdivisions and a  $2.10^{-4}$  m of axial mesh size in the central  
 361 part of the control volume.

362 For nozzle F nozzle the same number of cells per unit volume was used with similar boundary layer mesh sizes.

### 363 3.3. Test cases

364 In this section the method by which the models were calibrated and tested are presented followed by the comparisons to the experimental results.  
 365 The CFD results are compared in terms of mean mixture velocity, wall static pressure, inlet mass flow rate or efficiency. The latter is defined as  
 366 follows:

$$367 \eta = \frac{V_{gl}^2}{V_m^2} \quad 31$$

367 This shows the importance of good outlet velocity prediction for nozzle efficiency computation; velocity discrepancies are amplified by the power  
 368 applied to  $V_{gl}$ .

369 The mean velocity is computed as follows:

$$370 V_{gl} = x V_g + (1 - x)V_l \quad 32$$

370 Please note that for models using a unique momentum equation one get's  $V_g = V_l$  (models HK and HRM).

371 The maximum velocity is the isentropic velocity:

$$372 V_m = \sqrt{2 (h_{out,s}(P_{out}, S_{in}) - h_{in}(P_{in}, T_{in}))} \quad 33$$

372 For the models' accuracy comparison, a mean error is given for the available variables ( $\varphi$ ); it is computed as follows:

$$373 er_{\varphi} = \frac{1}{n} \sum \frac{|\varphi_{CFD} - \varphi_{REF}|}{\varphi_{REF}} \quad 34$$

373 where n is the number of measured points. When the error refers to pressure or void fraction profile, the value of  $\varphi_{CFD}$  is obtained by interpolation  
 374 at a given axial position. In the case of a mass flow result comparison, the non absolute relative error is given.

#### 375 3.3.1. Calibration and test process

376 The general procedure to calibrate and test the CFD models was to:

- 377 • get a mass flow rate close to the experimental one
- 378 • Try to get the nozzle outlet velocity for one of the outlet pressures. This is a crucial step when evaluating the ability of models to be calibrated.  
 379 This is done for the highest outlet pressure. The calibration was assumed as achieved when discrepancy was lower than 1% for both mass flow  
 380 rate and velocity at the pressure considered. When velocity calibration appeared as incompatible with flow rate calibration, the parameters  
 381 adapted for the correct flow rate were maintained. At the end of this step the empirical constants of the model are assumed to be calibrated.
- 382 • Test the model on the three other outlet pressures using the calibrated parameters in order to observe the robustness of the models to  
 383 changes in outlet pressure. If models react correctly when changing outlet pressure that means that one outlet pressure point is sufficient for  
 384 calibration.

385 Models' parameters variation for calibration was done by trial and error approach using as departure point the parameter values from reference  
 386 papers.

387 Besides, note that the flow rate is not dependent on the outlet pressure but varies with the degree of sub cooling; thus the first calibration step can  
 388 be repeated when the sub-cooling changes in order to maintain a good precision.

389 We recall the calibration parameters for the 4 tested models: for the HK model, the adaptation parameter  $\lambda$  was tuned, for the TA-B model one can  
 390 tune  $N_b$  and  $k_h$  parameters, for the HRM model the parameters " $\theta_0$ ", " $a$ " and " $b$ " can be modified, and for the TA-BD model  $N_b$  and  $N_d$  are tuned.

#### 391 3.3.2. Ohta B nozzle

392 This nozzle was experimentally characterized in terms of mass flow, outlet velocity and static pressure profiles for two inlet sub-cooling degrees.

##### 393 3.3.2.1. 1.5K sub cooling

###### 394 3.3.2.1.1. Results summary

395 After the first calibration step, the mass flow rate obtained by all the models was of 122g/s with less than 1% error. However, when trying to obtain  
 396 the measured outlet velocity only the TA-BD and HRM models could be effectively calibrated. The values of calibration parameters are given in the  
 397 Table 5.

398 Table 5 : calibrated parameters; Ohta B 1K

$\lambda$ (HK)	$\theta_0/a/b$ (HRM)	$N_b/k_h$ (TA-B)	$N_b/N_d$ (TA-BD)
$2,4 \cdot 10^{-2}$	$1,45 \cdot 10^{-4} / -0,26 / -1,75$	$1 \cdot 10^9 / 8$	$3,5 \cdot 10^{10} / 1,85 \cdot 10^9$

399 The velocities for the 4 outlet pressures are shown in Figure 10. One can check the outlet velocity at 100kPa outlet pressure for different models;  
 400 the points for the HRM model and for TA-BD model are superposed on the measured value. The value for HK model is very close to the measured  
 401 value, but getting a more accurate velocity is not possible when the target mass flow rate is obtained. In the same figure, in the case of TA-BD  
 402 model, the velocities for other pressures show that when pressure is reduced, the model reproduces very well the measurements.

403 The nozzle efficiency chart shown in Figure 11, shows that all models can capture the efficiency dependence on the pressure as measured by Ohta.  
 404 Nevertheless efficiency discrepancies are relatively important for the 3 literature models at optimal efficiency pressure (45kPa); for the TA-BD  
 405 model it is lower than 1% while for the HRM model the discrepancy on maximal efficiency is 18%.

406 These 4 models could be used for optimal outlet pressure research. But the most representative model is the TA-BD model at least for these  
 407 operating conditions. This suggests a better adaptation of this model to high expansion ratios. No obvious reasons for this difference to other  
 408 models was found but the absence of the pressure directly in the source term and the specific formulation of  $A_i$  are probably a part of the  
 409 explanation.

410 Regarding the pressure profiles on Figure 12 close to the nozzle outlet, the TA-B and TA-BD models are most accurate. The HK and HRM models  
 411 produced a shock wave close to the outlet; this seems to be related to an over estimation of the velocity before the shock. These two models use a  
 412 unique momentum equation so it could be related to the absence of velocity slip between phases.

413 At the throat HRM, TA-BD and especially the TA-B model show a pressure plateau; this is related to a vapor pocket located just after the throat and  
 414 absent in the HK model results.

415 Regarding the average errors, the best models are the TA-BD and HRM models. Average errors on outlet velocity, pressure mass flow rate are  
 416 reported in the Table 6.

Table 6 : summary of errors; Ohta B 1K

	TA-BD	HRM	TA-B	HK
$er_v$	0,7%	6,9%	12%	12%
$er_p$	9%	17%	13%	15%
$er_{\dot{m}}$	<1%	<1%	<1%	<1%

418 The TA-BD model is the most robust i.e. it is the one presenting the lowest average error when changing outlet pressure. This is particularly  
 419 interesting in the perspective of two-phase ejector simulations where the mixing pressure i.e. the outlet pressure of the primary flow, is not known  
 420 a priori.

421

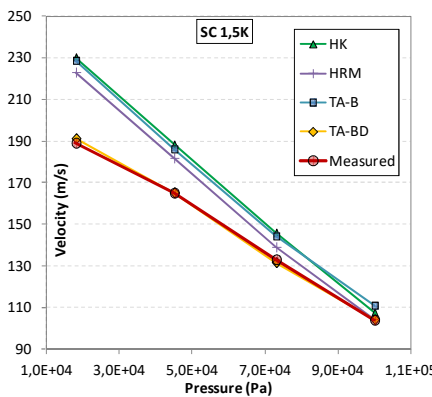


Figure 10 : outlet velocity function of outlet pressure; SC1.5K;nozzle B

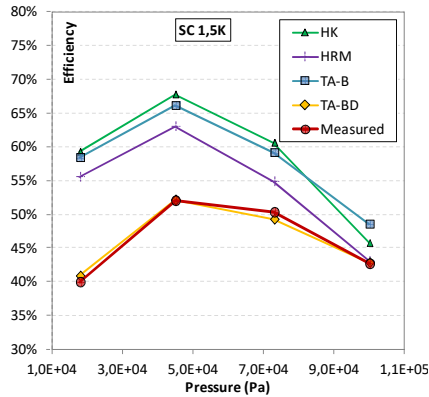


Figure 11: nozzle efficiency function of outlet pressure; SC1.5K;nozzle B

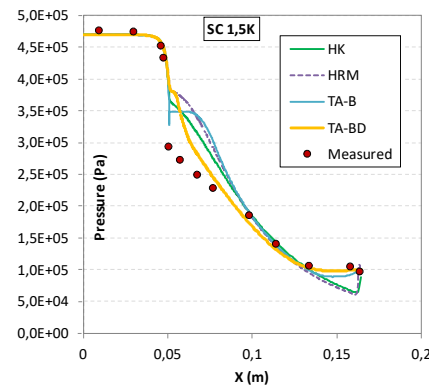


Figure 12 : static pressure profiles; SC1.5K 1bar outlet pressure; nozzle B

### 422 3.3.2.1.2. About calibration and flexibility of models

423 The two calibration objectives were the measured mass flow rate and the outlet velocity for 100kPa outlet pressure. During calibration process a  
 424 certain number of flow rate and velocity couples were obtained; they are shown in Figure 13; the values were divided by the corresponding  
 425 reference flow rate or velocity which results in a non dimensional plot. The resulting non-dimensional velocity was called velocity coefficient and  
 426 the non-dimensional pressure is called pressure coefficient. The closer those coefficients are to unit, the better is the matching with experiments.  
 427 This plot represents a space of solutions and helps during the calibration procedure; also it shows the limited space of possible solutions for some  
 428 models.

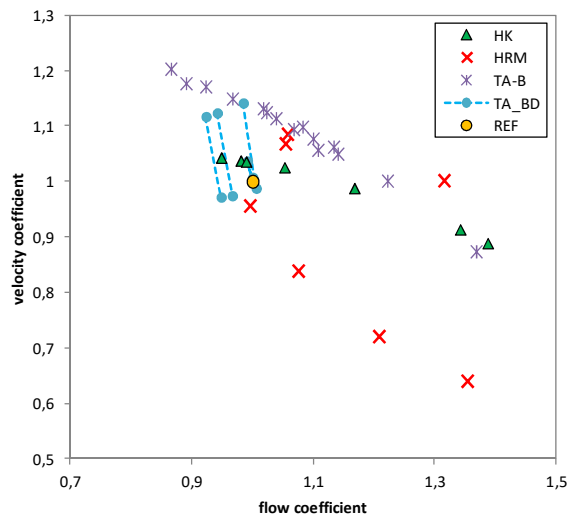
429 For the HK model the adaptation coefficient was modified. An increase in  $\lambda$  produces a flow rate reduction and an outlet velocity increase. A unique  
 430  $(\dot{m}/V)$  fraction is possible; this results on a line of solutions on the solutions space (triangles in Figure 13). The figure shows the case at 1bar outlet  
 431 pressure; for this specific case, the predicted velocity is close to the measured one at the target mass flow rate, but if this procedure has been done  
 432 for another outlet pressure, the target point would have been far from the solutions line (see velocity values in Figure 10).

433 The TA-B model includes  $N_b$  and  $k_h$  as parameters. The former was tested from  $8 \cdot 10^8$  to  $1 \cdot 10^{12}$  and 3 values were tested for the latter: 1, 8 and 16.  
 434 The resulting purple points were obtained in Figure 13. Increasing  $N_b$  or  $k_h$  has a similar effect to increasing  $\lambda$ ; this can be easily understood looking  
 435 to transfer equations. This leads to a narrow band of  $(\dot{m}; V)$  couples of the TA-B model solutions.

436 The HRM model's parameter  $\Theta_0$  was tested between  $1 \cdot 10^{-4}$  and  $20 \cdot 10^{-4}$ . Regarding "a" and "b", a preliminary analysis showed that both have a  
 437 similar effect on the results (see comments on HRM model regarding  $\alpha$  and  $\psi$ ) but changing "a" produces less effects than changing "b". For  
 438 example increasing "a" and "b" of 10% produce an increase of the mass transfer of +4% and +25% respectively.

439 At the end the parameter "a" was maintained at its original value of -0.257 and "b" was varied between -0.9 and -2.24. The resulting red points  
 440 were obtained. The reduction of " $\Theta_0$ " and increase of "b" tend to increase the velocity and reduce the flow rate. Nevertheless the coupling  
 441 between " $\Theta_0$ " and "b" was high and the responses far from linear making calibration laborious. However, the flexibility of this model is good and  
 442 results in a large space of solutions in Figure 13. Thus, both the target mass flow rate and the outlet velocity could be obtained.

443 The calibration process of the HRM model is quite difficult since the parameters are strongly coupled and the associated effects are non linear; that  
 444 explains why specific calibration techniques were developed for HRM using genetic algorithms by some authors [8].



445

Figure 13 : available flow/velocity solutions by model

446

447 Finally the TA-BD model calibration area is marked by blue points in Figure 13. The lines correspond to different  $N_b$  values ( $6 \cdot 10^{10}$  max,  $3.5 \cdot 10^{10}$   
 448 min);  $N_d$  was tested from  $1 \cdot 10^9$  to  $9 \cdot 10^{10}$  values. There was a strong dependence of the flow rate on  $N_b$  and velocity on  $N_d$ . This resulted in a fairly  
 449 easy calibration.

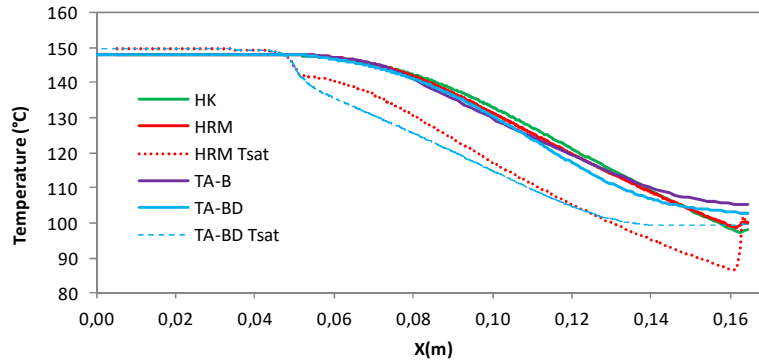
450 Some intermediate conclusion from this analysis:

- 451 • HK and TA-B models allow only a narrow band of solution of  $\dot{m}$  and  $V$  couples. This suggests that the use of 1 or 2 momentum equations does  
 452 not make any difference regarding the flexibility or robustness of the models
- 453 • The number of calibration parameters and their effect on the mass transfer is a key feature of a model for its ability of being calibrated but the  
 454 type of parameter and sensitivity of the model to it can be very different depending on the formulations

### 455 3.3.2.1.3. About physical discrepancies between the models

456 Regarding the energy equation, the heat transfer between phases was included in all the models. The resulting liquid temperature predictions are  
 457 shown by Figure 14. For all cases a temperature drop of 30K to 40K was observed depending on the models. In the case of the TA-B and TA-BD  
 458 models the temperature gradually approaches the saturation temperature when phase change begins (at the throat). Regarding the saturation  
 459 temperatures profiles, the difference between HRM and TA-BD models comes from the pressure profile difference; since the pressure close to the  
 460 nozzle outlet was lower for the HRM model, the reference (saturation) temperature is lower than for the TA-BD model. The same observation was  
 461 done for the HK model close to the nozzle outlet. For that reason, the liquid temperature of HRM and HK models is lower than the one predicted by  
 462 the TA models. Please note that the sudden saturation temperature increase of HRM model close to the outlet is due to the shock wave mentioned  
 463 in the comments of the Figure 12; the HK model predicted very similar saturation profile.



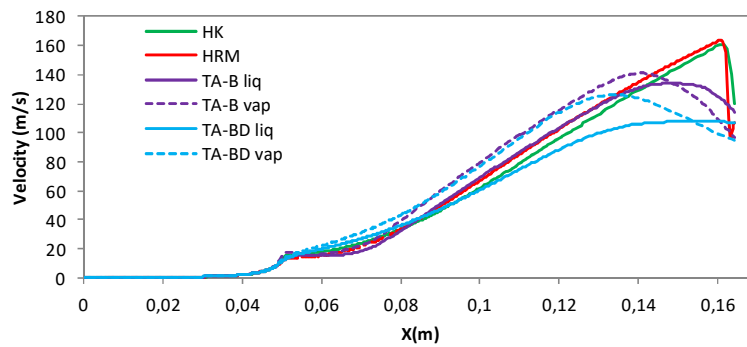


464

465

Figure 14 : temperatures at different axial positions; 100kPa outlet pressure

466 Regarding momentum equation two kinds of models were implemented: the homogeneous (in velocity) and non-homogeneous models. The  
 467 velocities computed by the different models are shown by Figure 15. It appears that only the homogeneous models give a shock wave even if the  
 468 velocity magnitudes in the major part of the nozzle are similar to the ones of the non-homogeneous models. Looking at the pressure profiles close  
 469 to the outlet in Figure 12, it appears that a pressure undershoot of HRM and HK models is related to the velocity overshoot; referring to the  
 470 pressure measurements it is probable that such a velocity variation in this part of the nozzle does not occur.

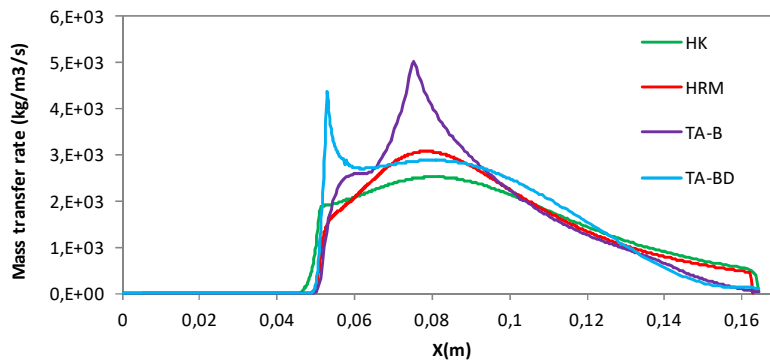


471

472

Figure 15 : velocities at different axial positions; 100kPa outlet pressure

473 Regarding liquid and vapor velocities modeled by the TA-B and TA-BD models one can observe that at the end of the nozzle the vapor is slower than  
 474 the liquid; that means that the liquid acceleration could not be maintained in a longer nozzle.



475

476

Figure 16 : interfacial mass transfer rate; 100kPa outlet pressure

477 The interfacial mass transfer rate along the axis of the nozzle is reported in Figure 16. Globally it increases rapidly after the throat, reaches a  
 478 maximum in the first half of the divergent part and then decreases approaching the outlet. One observes that HK and HRM models compute higher  
 479 mass transfer rates in the last third of the divergent ( $0,14m < x < 0,16m$ ) compared to the other models. This difference is related to the interfacial  
 480 area density formulation of TA-BD and TA-B models that brings it close to zero when void fraction approaches one. The HRM model and HK models  
 481 produced a sudden decrease of the mass transfer rate in the very last part of the nozzle because of the sudden increase of the pressure after the  
 482 shock waves.

483

### 3.3.2.2. Sub cooling of 12K

484

485

In order to observe the sensitivity of the models to a variation in the degree of sub-cooling, the precedent calibrated parameters were maintained  
 for a first simulation series. Then the calibration process was applied as in the precedent section.

486 The pressure data set was incomplete for this degree of sub-cooling in the referred article for this point. The model discrepancies between the  
 487 model and measured data are thus shown in terms of mass flow rate and velocity in Table 7. First two lines give the results for the models using the  
 488 calibration parameters of 1K sub-cooling situation.

489

Table 7 : summary of errors; Ohta B 12K

		TA-BD	HRM	TA-B	HK
1K SC calibration	$er_V$	6,8%	22%	24%	17%
	$er_m$	13%	14%	7,7%	20%
12KSC calibration	$er_V$	2,6%	11%	33%	41%
	$er_m$	<1%	<1%	<1%	<1%

490 Form a general point of view, increasing the inlet sub-cooling leads to higher mass flow rate and lower outlet velocities. Qualitatively, this behavior  
 491 was correctly predicted by all models.

492 The smallest combined error was produced by the TA-BD model. The predicted mass flow rates and velocities were always higher than measured  
 493 ones for all the models. TA-B model computed correct mass flow rate but inaccurate velocities.

494 The last two lines of Table 7 give the flow rate and velocity average discrepancies obtained after calibration. Values of the calibration parameters  
 495 are given in the Table 8.

496

Table 8 : calibrated parameters; Ohta B 12K

$\lambda$ (HK)	$\theta_0/a/b$ (HRM)	$N_b/k_h$ (TA-B)	$N_b/N_d$ (TA-BD)
0,15	$6 \cdot 10^{-4} / -0.26 / -0.05$	$2 \cdot 10^9 / 8$	$1,1 \cdot 10^{12} / 9 \cdot 10^8$

497 After the calibration step, the mass flow rate obtained by all the models was of 156g/s with less than 1% error. The value of the parameter  
 498 responsible of the mass flow rate for HK and TA models ( $\lambda$  and  $N_b$ ) is higher than the value at 1K SC.

499 Please note the increase in velocity error for TA-B and HK after the mass flow rate calibration step. The reason is that the accommodation  
 500 coefficients were increased in order to increase the cavitation inception pressure and thus reduce the mass flow rate; consequently the vapor  
 501 production was increased and so were the velocities. Please refer to Figure 13 to observe the relation between flow rate and velocity for these 2  
 502 models. This illustrates the problem of using a unique calibration parameter or several calibration parameters acting in the same way on all the  
 503 evaporative situations.

504 The velocities obtained after calibrations are shown in Figure 17; corresponding efficiencies in Figure 18.

505 The observation about the flexibility of the models can be applied here as well: only TA-BD and HRM were able to give the right outlet velocity at  
 506 100kPa outlet pressure during calibration. The most robust model is the TA-BD model.

507 The efficiency trend (Figure 18) is well captured by all models. TA-BD and HRM models are the closest to the reality. The efficiency plateau closest to  
 508 the reality is obtained with TA-BD model.

509 The simulated pressures are shown in Figure 19. One sees similar profile characteristics to the 1K sub-cooling case i.e. a pressure plateau for certain  
 510 models after the throat and a shock wave in last quarter of the divergent for the two HRM and for HK models.

511 Regarding the cavitation inception pressure, Ohta observed 2.8bar and 2.1bar pressures for 1K and 12K degrees of sub-cooling respectively. This is  
 512 close to 0.6 times saturation pressure at inlet temperature. Ohta guessed the inception to occur just after the throat. The computed pressure at this  
 513 point by the TA-BD model gives a factor relative to saturation pressure close to 0.8; so inception pressure was overestimated.

514 The assumed  $N_b$  values seem to increase when increasing liquid inlet sub-cooling. This is consistent with the result of [14].

515

516

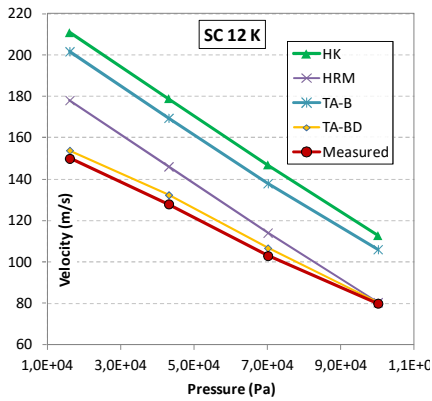


Figure 17 : outlet velocity function of outlet pressure; SC12K;nozzle B; mass flow calibrated models

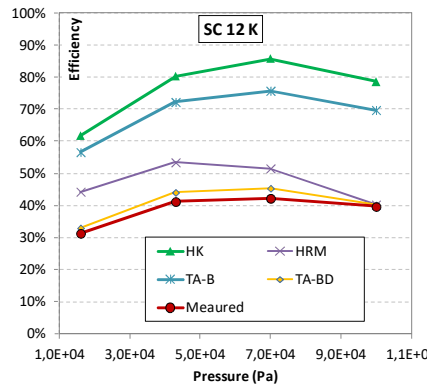


Figure 18 : nozzle efficiency function of outlet pressure; SC12K;nozzle B; mass flow calibrated models

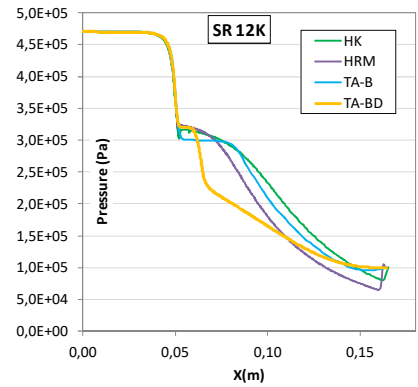


Figure 19 : static pressure profiles; SC12K 1bar outlet pressure; nozzle B; mass flow calibrated models

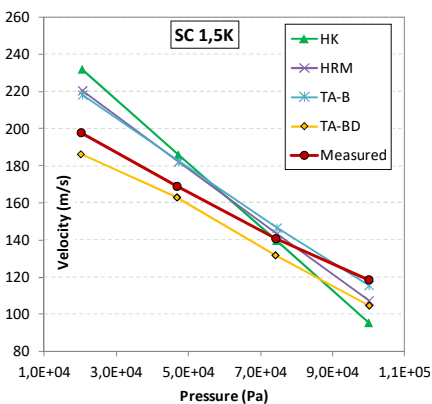


Figure 20 : outlet velocity function of outlet pressure; SC1.5K;nozzle F

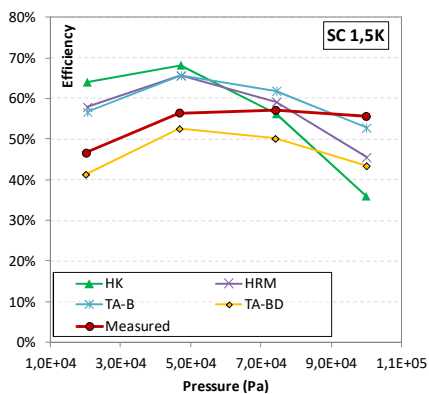


Figure 21 : nozzle efficiency function of outlet pressure; SC1.5K;nozzle F

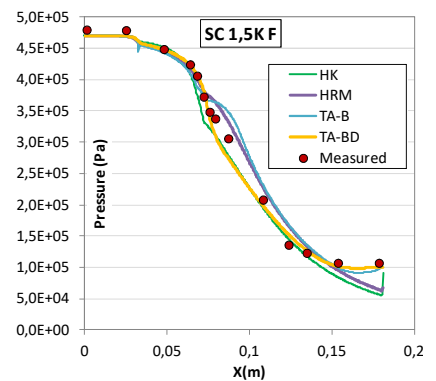


Figure 22 : static pressure profiles; SC1.5K 1bar outlet pressure; nozzle F

### 517 3.3.3.Ohta F Nozzle

#### 518 3.3.3.1. Results summary

519 The inlet conditions for this nozzle were identical to the first case (1K SC degree). The computations were done supposing that only inlet and outlet  
520 conditions affect the phase change regime. The parameters of the models for 1K SC were therefore maintained (the ones of Table 5).

521 The mean errors for mass flow rate, velocity, and pressure are reported in Table 9. For velocity we give the mean of the four outlet pressures cases.  
522 The pressure error is the one of the 100kPa case. The mass flow rate was identical for all the outlet pressures.

523 Table 9 : summary of errors; Ohta F 1.5K

	TA-BD	HRM	TA-B	HK
$er_m$	6,3 %	10%	15%	-8,8%
$er_v$	6,9%	7,6%	6,2%	12%
$er_p$	5,7%	9,6%	7,8%	10%

524 The mass flow rates obtained for this nozzle range from 108 g/s to 136 g/s depending on the model; those values are relatively close to the  
525 measured rate of 118g/s.

526 The obtained outlet velocities and corresponding isentropic efficiencies are reported in Figure 20 and Figure 21. The outlet velocity values and  
527 global trend are quite good captured by all models but the efficiency maximum was not correctly modeled. The measured values show an efficiency  
528 plateau between 45kPa and 100kPa outlet pressures that was not predicted by any model. The efficiency sensitivity to pressure was especially high  
529 for HK model. Also, the TA-BD model shows a moderate efficiency variation as was observed by Ohta experiments.

530 The pressure profiles obtained by the different models for 100kPa outlet pressure are show in Figure 22. A shock wave appears in the last part of  
531 the divergent section for HK and HRM models; this resembles to the simulated pressure profiles of the B nozzle. The pressure plateau of the TA-B  
532 model is shorter than in the precedent case.

533 The most close to measurements are the TA-BD and HRM models for this case. The TA-B model gives quite good values for velocity but over  
534 estimates the mass flow rate; if one tries to calibrate it by increasing  $N_b$ , to get the right flow rate, the velocities would increase a lot.

535 The effect of the geometry on the phase change regime is clearly not negligible; even if some models take it into account relatively well without any  
 536 supplementary calibration, the efficiency in Figure 21 for the TA-BD model is not satisfactory. The sensitivity of the efficiency to the outlet pressure  
 537 is quite low compared to the B nozzle and this is not well captured by the tested and proposed models.

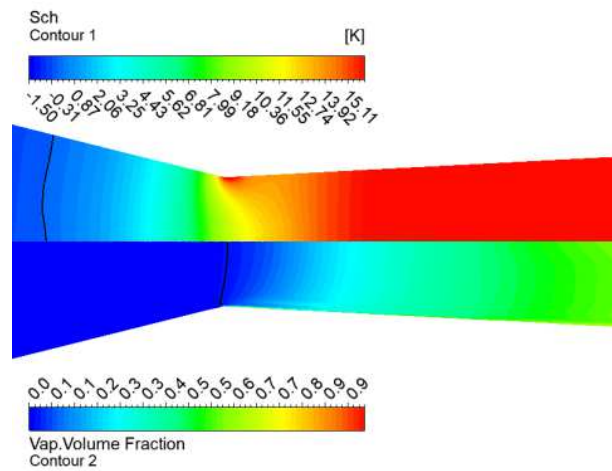
538 Compared to the B nozzle, the higher discrepancies for the F nozzle in terms of efficiency could be explained by a poor formulation of the interfacial  
 539 area density that would be highlighted in the case of the F nozzle.

### 540 3.3.3.2. A flow analysis

541 The difference between B and F nozzles is the geometry of the convergent part. The length of the small section part is higher in the case of F nozzle  
 542 and the angle before the throat is lower. This should affect what happens in the divergent part.

543 In order to analyze the effect of the throat and convergent geometry on the phase change regime and the flow, the contours of the void fraction  
 544 and degree of superheat are shown in Figure 23 for the B nozzle and in Figure 24 for the F nozzle. The black lines on the images are "isolines" for 0K  
 545 superheat (Sch) and 2% void fraction (VOF).

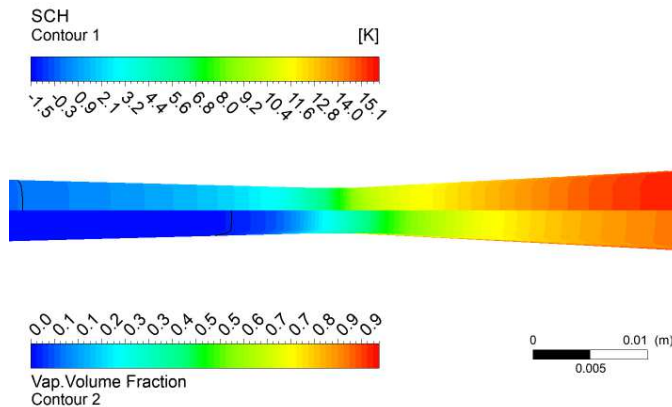
546 In the case of the B nozzle one sees that there is a high Sch close to the throat's outer radius; this is a favorable situation for phase change close to  
 547 the wall. In the case of the F nozzle this local Sch increase does not appear and thus the phase change will be more homogeneous.



548  
 549 Figure 23 : Ohta B nozzle; 1.5K 100kPa; superheat and void fraction

550 Furthermore the phase change begins long before the throat for the F nozzle, thus the void fraction at the throat is higher than for the nozzle B; this  
 551 could explain the lower flow rate of the F nozzle despite its larger throat. It would be useful to confirm this early vapor generation by experimental  
 552 observations.

553 One may point out that the formulation of the interfacial area density of the TA-BD model is only function of the void fraction. So despite its globally  
 554 good results, the formulation is probably not complex enough to model the multitude of topologies constitutive of depressurized phase change  
 555 flows.



556  
 557 Figure 24 : Ohta F nozzle; 1.5K 100kPa; superheat and void fraction

## 558 4. Conclusion and future work

559 This paper deals with CFD modeling of two phase turbine flash nozzles.

560 The paper explored the ability of 4 models to predict nozzles' mass flow rate and efficiency. Experimental data specifically containing nozzle outlet  
561 velocity measurements were used as reference.

562 The four models were implemented in the commercial software Ansys CFX i.e. the HK model based on a pressure difference, the HRM model based  
563 on a vapor quality difference, the TA-B and TA-BD models based both on a temperature difference but different interfacial area density  
564 formulations.

565 The ability of the 4 models to be calibrated was tested on two nozzle geometries. Three models were based on precedent research works. The last  
566 one is a model newly explored in this paper for flash nozzles; this is a variation of the Wu model [15].

567 From the comparisons with experimental data, it can be observed, concerning the calibration and prediction capacity of the models, that:

- 568 > Less than 1% error could be obtained on mass flow rate by all the tested models. The mass flow rate seems to be driven by the vapor  
569 generation rate in the region of the 5K to 10K first degrees of superheat.
- 570 > The calibration of the mass flow rate is incompatible with the one of the outlet velocity (or efficiency) excepted for the HRM and TA-BD  
571 models.
- 572 > The mass flow rate is strongly affected by the degree of sub-cooling at the inlet. This effect is captured by all the models but is overestimated.  
573 Thus, to improve the precision when the sub-cooling degree changes, models need to be recalibrated. For the models using a single or  
574 redundant calibration parameters (HK and TA-B), this produces an increase of the errors on velocity.
- 575 > Regarding the HRM model calibration, for a given sub-cooling, the calibration of two parameters seems to be sufficient. The calibration  
576 process of the HRM model is quite difficult since the parameters are strongly coupled and the associated effects are non linear.
- 577 > Regarding the tested HK-type model, it gave relatively good results but always higher discrepancies in average than other models. This model  
578 is limited by the use of a unique adaptation parameter. It should be noticed that the explored transition interfacial area density formulation  
579 could be used in HK based models.
- 580 > The TA-BD model is the most robust model. Regarding its calibration parameters, the number of bubbles density affects the mass flow rate  
581 and the number of droplets density affects the outlet velocity.
- 582 > Two nozzle geometries were tested. It was observed that the model that best adapts to geometry change is the TA-BD model. However, the  
583 sensibility to the outlet pressure was not well captured by any model for the F nozzle.

584 The last remark underlines a limitation of the TA-BD model: the interfacial area density formulation does not depend on nozzle geometry; however  
585 the interface topology is certainly modified by the nozzle shape. So the interfacial area density formulation should be improved. The shape  
586 optimization CFD techniques could be enhanced by this eventual improvement.

587 Another limit of the TA-BD model is the use of predefined  $N_b$  and  $N_d$  values. Additional work should be done to integrate a nucleation model to  
588 predict  $N_b$ . Regarding  $N_d$  values, the order of magnitude was similar to the one of  $N_b$  values, higher or lower. It is not yet clear how it could be  
589 predicted. It may be useful to introduce a droplet breakup model as a function of liquid/gas velocity slip. Also the effect of the boundary conditions  
590 uncertainty should be explored by uncertainty propagation simulation analysis in order to obtain the confidence interval of the prescribed  
591 calibration parameters.

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## 595 5. References

596

- [1] M. A. Iqbal, S. R. Mahdi, A. A. Date and AliakbarAkbarzadeh, "Experimental study on the prospect of low-temperature heat to power generation using Trilateral Flash Cycle (TFC)," *Applied Thermal Engineering*, vol. 172, 2020.
- [2] D. G. Elliott, *Theory and Tests of Two-Phase Turbines*, Pasadena: Jet Propulsion Laboratory, 1982.
- [3] I. K. Smith, "Development of the trilateral flash cycle system. Part 1 : fundamental considerations," *Proc Instn Mech Engrs*, vol. 207, 1993.
- [4] M. T. White, "Cycle and turbine optimisation for an ORC operating with two-phase expansion," *Applied Thermal Engineering*, vol. 192, 2021.
- [5] S. Rane and L. He, "Numerical analysis of a novel two-phase turbine using thermal non-equilibrium, homogeneous nucleation phase change," *Thermal Science and Engineering Progress*, vol. 22, 2020.

- [6] N. Abuaf, "A study of Nonequilibrium Flashing of Water in a Converging-Diverging Nozzle: Volume1. Experimental. U.S. Nuclear Regulatory Commission," 1981.
- [7] F. Bussac, "Developpement d'une turbine diphasique pour la production d'électricité à partir de la détente de saumures géomothermales," Commission des Communautés européennes, 1985.
- [8] M. Haida, J. Smolka, A. Hafner, M. Palacz, K. Banasiak and A. J. Nowak, "Modified homogeneous relaxation model for the R744 trans-critical flow in a two-phase ejector," *International Journal of Refrigeration*, pp. 314-333, 2017.
- [9] F. Giacomelli, F. Mazzelli and A. Milazzo, "A novel CFD approach for the computation of R744 flashing nozzles in compressible and metastable conditions," *Energy*, vol. 162, pp. 1092-1105, 2018.
- [10] B. M. Devassy, D. Benković, Z. Petranovic, W. Edelbauer and M. Vujanovic, "Numerical Simulation of Internal Flashing in a GDI Injector Nozzle," in *Conference on Liquid Atomization and Spray Systems*, Paris, 2019.
- [11] I. Karathanassis, "Comparative evaluation of phase change mechanisms for prediction of flashing flows," *International Journal of Multiphase Flows*, vol. 97, pp. 257-270, 2017.
- [12] L. Yixiang and L. Dirk, "Computational modelling of flash boiling flows, a literature survey," *International Journal of Heat and Mass Transfer*, pp. 246-265, 2017.
- [13] P. Downar-Zapolski and Z. Bilicki, "The non-equilibrium relaxation model for one-dimensional flashing liquid flow," *International Journal of Multiphase Flow*, vol. 22, pp. 473-483, 1996.
- [14] L. Yixiang and L. Dirk, "3D CFD simulation of flashing flows in a converging-diverging nozzle," *Nuclear Engineering Design*, pp. 149-163, 2015.
- [15] A. N. Wu B.J.C., "A study of Nonequilibrium Flashing of Water in a Converging-Diverging Nozzle: Volume2. Modelling. U.S. Nuclear Regulatory Commission," 1981.
- [16] J. Ohta, T. Fuji, K. Akagawa and N. Takenaka, "Performance and flow characteristics of nozzles for initially subcooled hot water (influence of turbulence and decompression rate)," *International Journal of Multiphase Flow*, vol. 19, no. 1, pp. 125-136, 1993.
- [17] Ansys, *Fluent V20 Users Guide*, 2020.
- [18] M. Yazdani, T. D. Radcliff, A. A. Alahyari and M. Farzad, "Numerical Modeling and Validation so Supersonic Two-Phase Flow of CO<sub>2</sub> in Converging-Diverging Nozzles," *Journal of Fluid Engineering*, vol. 136, 2014.
- [19] Q. D. Le, *FLASHING FLOW MODEL FOR INDUSTRIAL ENERGY APPLICATIONS*, POLITECNICO DI MILANO, 2017.
- [20] G. Palau-Salvador, P. Gonzalez-Altozano and J. Arviza-Valverde, "Numerical modeling of cavitating flows for simple geometries," *Spanish Journal of Agricultural Research*, 2007.
- [21] M. Palacz, M. Haida, J. Smolka, A. J. Nowak, K. Banasiak and A. Hafner, "HRM accuracy comparison for the simulation of CO<sub>2</sub> expansion in two-phase ejectors for supermarket refrigeration systems," *Applied Thermal Engineering*, no. 115, pp. 160-169, 2017.
- [22] S. Maksic and D. Mewes, "CFD-CALCULATION OF THE FLASHING FLOW IN PIPES AND NOZZLES," *Proceedings of ASME FEDSM'02*, 2002.
- [23] L. Geng, H. Liu and X. Wei, "CFD analysis of the flashing flow characteristics of subcritical refrigerant R134a through converging-diverging nozzles," *International Journal of Thermal Sciences*, pp. 438-445, 2019.
- [24] H. Guo, L. Nocivelli, R. Torelli and S. Som, "Towards understanding the development and characteristics of under-expanded flash boiling jets," *International Journal of Multiphase Flow*, 2020.
- [25] A. M. SINGHAL A.K., "Mathematical bases and validation of the full cavitation model," *J Fluids Eng*, vol. 124, pp. 617-624, 2002.

- [26] M. S. Lee, H. Lee, Y. Hwang, R. Radermacher and H.-M. Jeong, "Optimization of two-phase R600a ejector geometries using a non-equilibrium CFD model," *Applied Thermal Engineering*, vol. 109, no. 26, pp. 272-282, 2016.
- [27] M. D. Lorenzo, P. Lafon, M. D. Matteo, M. Pelanti, J.-M. Seynhaeve and Y. Bartosiewicz, "Homogeneous two-phase flow models and accurate steam-water table look-up method for fast transient simulations," *International Journal of Multiphase Flow*, pp. 199-219, 2017.
- [28] W. Angielczyk, Y. B. D. Brtosiewicz and J.-M. Seynhaeve, "1-D Modeling of Supersonic Carbon Dioxide Two-Phase Flow Through Ejector Motive Nozzle," in *International Refrigeration and Air Conditioning Conference*, Purdue, 2010.
- [29] A. A. Avdeev, "Laws of Vapor Bubble Growth in the Superheated Liquid Volume (Thermal Growth Scheme)," *High Temperature*, vol. 52, no. 4, pp. 588-602, 2014.
- [30] E. Krepper, L. Dirk, T. Frank, H.-M. Prasser and J. Zwart Phil, "The inhomogeneous MUSIG model for the simulation of polydispersed flows," *Nuclear Engineering and Design*, no. 238, pp. 1690-1702, 2008.
- [31] S. Hänsch, D. Lucas, E. Krepper and T. Höhne, "A multi-field two-fluid concept for transitions between different scales of interfacial structures," *International Journal of Multiphase Flow*, vol. 47, pp. 171-182, 2012.
- [32] Ansys, CFX V20 User's Guide, 2020.
- [33] J. Sende, Y. Hojyo and H. Fujimoto, "Modeling on atomization and vaporization process in flash boiling spray," *JSAE Review*, vol. 15, pp. 291-296, 1994.

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