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On numerical simulation of cavitating flows under thermal regime

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ABSTRACT

In this work, we investigate closure laws for the description of interfacial mass transfer in cavitating flows under thermal regime. In a first part, we show that, if bubble resident time in the low pressure area of the flow is larger than the inertial/thermal regime transition time, bubble expansion are no longer monitored by Rayleigh equation, but by heat transfer in the liquid phase at bubbles surfaces. The modelling of interfacial heat transfer depends thus on a Nusselt number that is a function of the Jakob number and of the bubble thermal Péclet number. This original approach has the advantage to include the kinetic of phase change in the description of cavitating flow and thus to link interfacial heat flux to interfacial mass flux during vapour production. The behaviour of such a model is evaluated for the case of inviscid cavitating flow in expansion tubes for water and refrigerant R114 using a four equations mixture model. Compared with inertial regime (Rayleigh equation), results obtained considering thermal regime seem to predict lower local gas volume fraction maxima as well as lower gradients of velocity and gas volume fraction. It is observed that global vapour production is closely monitored by volumetric interfacial area (bubble size and gas volume fraction) and mainly by the Jakob number variations. It is found that, in contrast with phase change occurring in common boiling flow, Jakob number variation is influenced by phasic temperature difference but also by density ratio variation with pressure and temperature ($Ja \propto (\rho_L/\rho_G)\Delta T$).

1. Introduction

Cavitation is involved in various flow applications such as hydraulic turbines, pumps, rocket turbopump inducers, fuel injectors, marine propellers, underwater bodies, etc. In most of cases, cavitation is an undesirable phenomenon, significantly degrading performance, resulting in lower pressure head of pumps, asymmetric load on turbomachinery blades, vibrations, noise and erosion. In industrial applications, cavitating flows usually take form as a turbulent vapour polydispersed bubbly flow with phase change, bubble break-up and coalescence. In the literature, various gas–liquid mixture or two-fluid models have been developed to investigate isothermal and non-isothermal cavitating flows. According to the assumptions made, those models differ on two main points: equations solved and description of phase change.

Among cavitation models, different approaches can be found to describe phase change due to cavitation: barotropic model [1], short relaxation time model [2], velocity divergence model [3] and model based on inertial [4] or thermal bubble growth [5]. To estimate locally vapour volume fraction, one first approach is to assimilate the gas–liquid mixture to a barotropic fluid. In other words, the density of the gas–liquid mixture is considered to be a function of the local static pressure in the flow. For the simulation of a cavitating flow through a venturi, [1] proposed a sinus barotropic law considering a direct link between the gas volume fraction, phasic densities, local pressure and vapour saturation pressure. For the simulation of cavitating flows in tubopump inducers of spatial rockets, [6] proposed a sinus barotropic law with a vapour saturation pressure calculated from local temperature in the flow. Those robust approaches have provided interesting results for the simulation of hydrofoils [7], venturies [8], turbopump inducers [9–11], pump-turbines [12] or fuel injectors [13]. Although the simplicity of this modelling approach, this model enable to study complex industrial applications. However, the adaptability of such model for thermosensitive liquids, where temperature gradients are significant, seem to suffer from a lake of physical descriptions of mass and heat transfers induced by phase change at bubbles surfaces.

One second approach is to express explicitly mass and heat transfer terms and to consider that interfacial transfers are instantaneous. In that case, as proposed by [2] or [14], by introducing infinite relaxation parameters (or infinite global transfer

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Nomenclature

List of symbols

a_I	volumetric interfacial area, m^{-1}
B	B factor
C_{pk}	specific heat capacity of phase k at constant pressure, $J\,kg^{-1}\,K^{-1}$
C_{vk}	specific heat capacity of phase k at constant volume, $J\,kg^{-1}\,K^{-1}$
c_w	Wood's speed of sound, $m\,s^{-1}$
c_{EOS}	speed of sound based on sinus equation of state, $m\,s^{-1}$
c_k	speed of sound of phase k, $m\,s^{-1}$
d_b	mean Sauter diameter of the bubble size distribution, m
D_k^{th}	thermal diffusivity of phase k, $D_k^{th} = \lambda_k/(\rho_k C_{pk})$, $m^2\,s^{-1}$
e_k	specific internal energie of phase k, $J\,kg^{-1}$
E_k	total energy of phase k, $J\,kg^{-1}$
E_m	gas liquid mixture total energy, $J\,kg^{-1}$
H_L	heat transfer coefficient in phase k, $W\,m^{-2}\,K^{-1}$
h_k	specific enthalpie of phase k, $J\,kg^{-1}$
J_{loc}	local mass flux, $kg\,m^{-2}\,s^{-1}$
Ja	Jakob number, $Ja = \rho_L C_{pL}\Delta T/(\rho_G L)$
l	mean length of bubble path in the low pressure area where $p < p^{sat}$, m
L	latent heat of vaporisation $L = h_G - h_L$, $J\,kg^{-1}$
N_{uL}	Nusselt number
N_{uL0}	Nusselt number wihout wlip
p	pressure, Pa
p_k^{∞}	pressure reference in stiffened gas equation of state, Pa
Pe	thermal Péclet number, $Pe = Ud_b/D_L^{th}$
Pe_c	critical thermal Péclet number,
Pr	Prandtl number, $Pr = \nu_L/D_{L^{th}}$
q_k''	interfacial heat flux from phase k, $J\,m^{-3}\,s^{-1}$
R	bubble radius, m
Re	bubble Reynolds number, $Re = \rho_L v_r d_b/\mu_L$
s	surface tension of the fluid, Nm^{-1}
S_b	bubble surface, m^2
t	time, s
T_k	tempreature of phase k, K
u	stretching velocity, $m\,s^{-1}$
V_b	bubble volume, m^3
v_k	velocity of phase k, $m\,s^{-1}$
v_m	gas liquid mixture velocity, $m\,s^{-1}$
v_r	mean relative velocity between phases, $v_r = v_G - v_L$, $m\,s^{-1}$

Greek symbols

α_k	volume fraction of phase k
γ_k	heat capacity ratio for phase k, $\gamma_k = C_{pk}/C_{vk}$
Γ_k	mass transfer term of phase k, $kg\,m^{-3}\,s^{-1}$
λ_k	conductivity of phase k, $W\,m^{-1}\,K^{-1}$
μ_k	dynamic viscosity of phase k, Pa s
ν_k	kinematic viscosity of phase k, $m^2\,s^{-1}$
π_k	internal energy reference in stiffened gas equation of state, $J\,kg^{-1}$
ρ_k	density of phase k, $kg\,m^{-3}$
τ	time for inception of thermal regime, s
τ'	time for inception of thermal regime with effect of the relative velocity, s
τ_{res}	resident time of bubbles in the low pressure area, s

Supercripts

I	at the bubble surface
sat	at saturation
T	at triple point

Subscripts

G	gas phase
k	phase k
L	liquid phase
m	gas liquid mixture
0	initial value

coefficients), momentum, mass and heat transfers can be evaluated considering very short equilibrium relaxation times between phasic pressures, velocities, temperatures, Gibbs free energy. This approach, initially devoted for the simulation of diphasic detonation waves [15], is considered to be valid for cavitating flow at very high velocity. As shown recently by [3], very similar results can be achieved considering simply that the mass transfer term is proportional to the mixture velocity divergence. This approach has been recently used for non isothermal cavitation by [16] for the 2D simulations of cavitating flow through a venturi.

To describe cavitation, one last approach is to consider finite rate mass transfer and to express explicitly mass transfer exchange term due to phase change. In the literature, a large number of such cavitation models consider that vapour production in a cavitating flow is only driven by inertial controlled growth of vapour bubbles. For example, considering the equation of [17], a large number of authors [18,4,19,20] have proposed a mass transfer exchange term proportional to the square root of saturated vapour pressure and liquid pressure difference ($\sqrt{p^{sat} - p_L}$). Similarly, [21,22] considered that vapour production depends on phasic pressure difference ($p^{sat} - p_L$) and on the convective characteristic time scale of the flow. As shown by [23], for some cases, cavitation models of [19,20,22] seem to provide very similar results. Inertial growth models have been also massively employed for the simulation of isothermal cavitating flow through turbopump inducers [24], propellers [25], centrifugal pumps [26,27], hydrofoils [28–30] and fuel injectors [31]. For the simulation of cavitating flow in cryogenic fluids, [32,33] use the formulation proposed by [21].

Various authors have attempted to take into account the effect of liquid phase thermal gradients in the flow on cavitation. For barotropic approach [34,6] as well as for inertial controlled growth model [33,35,36], it has been done meanly by calculating the saturated vapour pressure as a function of the local temperature ($p^{sat}(T)$) and by estimating the bubble temperature variation using energy balance at the bubble scale. In the same time, a few numerical works evoked that vapour production can be driven by thermal controlled growth of vapour bubbles [37–39,5]. In [37], authors recall that bubble growth follow two steps. The increase of bubble volume is initially controlled by the liquid inertial (inertial growth) and then is controlled by heat transfer at bubble surface (thermal growth). One of the authors conclusions is that in future works "suitable bubble growth law to model the so-called thermally controlled growth has to be implemented". Conserving short relaxation time for phasic pressures and velocities equilibrium, to take into account finite rate phase change, [38] modified the relaxation model proposed by [2] by including finite rate heat transfer between the dispersed phase and the continuous phase. Later, considering the strong similarity with boiling flow simulations, [39,5] proposed to describe vapour production in cavitating flow assuming only thermal controlled bubble growth. In that approach, the vapour production at bubble surfaces depends on heat flux brought by the liquid and the gas phase.