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Numerical study on heat transfer effects of cavitating and flashing flows based on homogeneous mixture model

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ABSTRACT

A numerical study on heat transfer effects of cavitating and flashing flows was conducted by employing a compressible homogeneous mixture approach. A dual-time preconditioned method was applied to enhance the efficiency and accuracy of the computations under various flow conditions. A sensitivity analysis of the empirical coefficients used to access the predictive capability of the existing mass transfer models was assessed for cavitating flows in a converging-diverging nozzle. To confirm the use of the calibrated empirical coefficients, computations of the water flows over a cylindrical head form and a Clark-Y hydrofoil were then carried out. The results obtained indicate an overall good agreement with the experimental data. Finally, the thermal effects on the phase change process were confirmed through an examination of a flashing flow along with thermo-fluid flows. The predicted results also show a good agreement with the experimental data. In conclusion, the existing system was shown to be effective in quantitatively predicting the thermal effects of the phase change processes, and can be used to examine the hydro- and thermodynamics of multiphase flows.

1. Introduction

In cavitating and flashing flows, evaporation occurs when a partial region undergoes an extreme pressure drop in the water flow. These flows involve a transition from a liquid phase to a vapor phase. The feature distinguishing these two flows is backpressure value. Some of the pressure will be recovered if the backpressure is higher than the saturated vapor pressure, a phenomenon called cavitation. In general, cavitation occurs in a low-temperature fluid in which the vapor density is so low that the latent heat flow does not affect the cavitation phenomenon. Cavitation occurs in many hydraulic machines and components, and limits their effective range of operation.

In contrast, a flashing flow refers to the phenomenon that occurs when the backpressure is lower than the saturated vapor pressure. This phenomenon generally occurs in a high-temperature high-pressure fluid in which the difference between the saturated liquid density and the saturated vapor density decreases with an increase in the fluid temperature. The phase change is essentially dominated by the pressure energy, whereas, unlike in the case of cavitation, the flashed vapor is accompanied by a vapor volume expansion and a distinct decrease in enthalpy, thereby maintaining the energy equilibrium. This temperature depression is called a thermal effect. One of the typical engineering applications related to this phenomenon is a flashing nozzle flow, which is known to occur during a hypothetical loss of coolant accident (LOCA) in a nuclear reactor.

It is crucial to understand the physics of a two-phase flow phenomenon to improve its positive influence and alleviate its adverse impacts. In this sense, studying the cavitation and flashing dynamics in a simple geometry is one way of achieving this goal.

Among previous studies on the numerical simulation of multiphase flows, Merkle et al. [1], Kunz et al. [2], and Owis and Nayfeh [3] proposed a preconditioned dual-time incompressible multiphase flow system. This system was successfully validated through a comparison with experimental data and numerical simulations of a cavitating flow. The system was further developed by Ha et al. [4] and Jin et al. [5,6] to predict certain multiphase flow characteristics. However, an important shortcoming of this system is that it does not account directly for finite acoustic speeds or the compressible and thermal effects in a liquid/vapor mixture.

Venkateswaran et al. [7], Wang et al. [8], and Senocak and Shyy [9] proposed another incompressible multiphase flow system similar to the dual-time incompressible system described above. They also developed a generalized numerical framework for transient and multiphase problems that involve a combination of gas, vapor, and water phases. This framework accounts for changes in density by means of a pressure-based algorithm, a volume fraction transport equation, and a pressure-velocity-density coupling scheme

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used to conduct turbulent multiphase flow computations. Based on this pressure-based method, Venkateswaran et al. [10], Hosangadi et al. [11], and Ahuja et al. [12] presented an acoustically accurate form of a compressible multiphase system. This system does not require additional equations to resolve the vapor-liquid interface. In this system, the local speed of sound in a two-phase mixture is a function of the local void fraction and this local speed mimics the two-phase acoustic speed relationship according to classical analytic theory. The equations of the system are used to formulate a multiphase model for low-speed vapor/liquid mixtures by reducing the compressible system of equations to an acoustically accurate form for multi-fluid mixtures. Whereas the effects of the Mach number or temperature depression are not factored into the equations, the pressure correction equation shares common features with high-speed flows. Venkateswaran et al. [10] investigated the differences between compressible and incompressible forms of the code for two cylindrical head forms, and concluded that a compressible model captures the dynamics of the cavity more accurately than an incompressible model.

The next system of multiphase flows considered in this study is a fully developed compressibility formulation [13–17], which is based on a preconditioned dual-time incompressible multiphase flow approach. The energy conservation equation is also considered in this system and is used to obtain some additional physical properties, including the temperature and Mach number and to improve the numerical accuracy. Further, the acoustic speed of sound in multiphase mixture regions is taken into account to express the shock wave expansion systems. However, in contrast to a detailed discussion on the compressible effects, there have been few reliable reports offering detailed descriptions of the thermal effects on cavitating or flashing flows.

The mass transfer rates between a liquid and vapor were considered using another set of equations, forming a type of phase change system based on a homogeneous mixture theory. This model formulates the processes of condensation and evaporation by utilizing the phase volume fraction, phase density, local pressure, and local temperature. The present study applies two kinds of phase change models. The first is a pressure phase change model, which is a newly modified cavitation model validated by Ha and Park [4], and is based on the cavitation model originally proposed by Merkle et al. [14]. In this model, the benchmark of vaporization is the saturated vapor pressure. Therefore, it is easy to determine the instant of vapor phase generation. The other model is a thermal phase change model, which was originally proposed by Lee [18]. The benchmark of vaporization in this model is the saturated vapor temperature. Through this model, we can supplement the amount of vapor generated from water under superheated conditions. Because little thermal energy is required to induce vaporization from cold water, such energy can be neglected. In contrast, the phase change for hot water is akin to the boiling process, and therefore the thermal energy provided by an interphase heat transfer needs to be considered.

The aim of the present study is to develop and validate a computer simulation code for a numerical study of multiphase, unsteady, compressible flows with thermal effects. The operating fluids are cold water for a cavitating flow and hot water for a flashing flow. We attempted to determine both the compressible and thermal effects by utilizing numerical solutions of the combined mass, momentum, and energy equations using two types of phase change models, i.e., the pressure phase change model and thermal phase change model.

Previous works on homogeneous mixture models have shown that, with the most widely used cavitation model [2], also known as the pressure phase change model, the coefficient is varied according to the variations in the flow field. In other words, a failure to acquire the specific model coefficient for a specific flow field makes it impossible to obtain accurate numerical solutions other than those for the cavity size, cavity collapse, pressure depression, temperature depression, and so on.

Therefore, the objectives of the present work are as follows.

First, the dependence of the present pressure phase change model on its coefficients is assessed through a comparison with another well-known cavitation model. Cavitating flows in a long converging-diverging nozzle are simulated and compared using compressible and incompressible in-house codes. The purpose of doing so is twofold: to identify the empirical coefficients in an appropriate range and to examine the improvements in the present compressible system.

Second, a numerical validation is conducted through a simulation of the hydrodynamic cavitation around a cylindrical head form and a Clark-Y hydrofoil.

Third, a simulation of the flashing flows in a short converging-diverging nozzle is conducted using a combination of the pressure and thermal phase change models. This step is aimed at predicting the significant effects of latent heat transfer on the vaporization process through calculations of the temperature and pressure depressions.

2. Mathematical formulation

2.1. Governing equations

In this study, multiphase unsteady Reynolds-Averaged Navier-Stokes equations are employed to solve the problems of a multiphase flow with compressible and thermal effects. The equations consist of mass, momentum, and energy conservation equations and the two types of phase change models. Each phase has its own mass conservation equation to predict a void fraction.

$$\frac{\partial}{\partial t}(Y_L\rho_m)+\frac{\partial}{\partial x_j}(Y_L\rho_m u_j)=(\dot{m}_p^+ + \dot{m}_p^-)+(\dot{m}_T^+ + \dot{m}_T^-) \tag{1}$$

$$\frac{\partial}{\partial t}(Y_v\rho_m)+\frac{\partial}{\partial x_j}(Y_v\rho_m u_j)=-[(\dot{m}_p^+ + \dot{m}_p^-)+(\dot{m}_T^+ + \dot{m}_T^-)] \tag{2}$$

$$\frac{\partial}{\partial t}(\rho_m u_i)+\frac{\partial}{\partial x_j}(\rho_m u_i u_j)=-\frac{\partial p}{\partial x_i}+\frac{\partial \tau_{ij}}{\partial x_j}+\rho_m g_i \tag{3}$$

$$\frac{\partial}{\partial t}(\rho_m h_t - p)+\frac{\partial}{\partial x_j}(\rho_m h_t u_j)=\frac{\partial(u_j\tau_{ij}-q_j)}{\partial x_j}-h_t[(\dot{m}_p^+ + \dot{m}_p^-) + (\dot{m}_T^+ + \dot{m}_T^-)] \tag{4}$$

Here, Y, u, p, h_t, and q denote the mass fraction, velocity, pressure, total enthalpy, and heat flux, respectively. The subscripts L, V, and m indicate the liquid, vapor, and mixture phases, respectively, and subscripts p and T indicate the pressure and thermal phase change models. In addition, superscripts + and − indicate the transfer from a vapor to a liquid phase, and from a liquid to a vapor phase, respectively.

The density, total enthalpy, and heat fluxes of the mixture phase are expressed respectively as

$$\rho_m = Y_L\rho_m + Y_v\rho_m \tag{5}$$

$$h_t = \left[Y_L h_L + Y_v h_v + \frac{1}{2}(u^2+v^2+w^2)\right] \tag{6}$$

$$q_j = -\left(\frac{\mu_m}{Pr_m}+\frac{\mu_t}{Pr_t}\right)Cp_m\frac{\partial T}{\partial x_j} \tag{7}$$

where T denotes the local temperature, μ_t is the turbulent viscosity, and μ_m is the mixture viscosity. The Jones-Launder low-Reynolds