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Modeling and investigation of a steam-water injector

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ABSTRACT

A one-dimensional mathematical model of a steam-water two-phase injector is presented. This model offers a method of estimating critical conditions of steam at the site of the motive nozzle throat, based on the local sound velocity in that area. Fluid thermal properties were based on a real fluid approach, where the CoolProp database was utilized. A different method was adopted to formulate governing equations for all passages of the injector based on the principles of the conservation of mass, momentum, and energy. The pressure profiles of the injector at different inlet steam pressure and inlet water pressure were used to validate the proposed model; they agreed well, with a maximum relative rate of error within 9.5%. Based on the validated model, the influence of the different area ratios and coefficients of the diverse sections on the performance of an injector used in district heating was investigated. The main inlet parameters - steam pressure and water pressure - were within the range of 0.20–0.60 MPa and 0.14–0.49 MPa. The exergy destruction rate for every steam-water injector component was also computed. The results illustrated that the injector discharge pressure increases with the throat area ratio of the motive nozzle and mixing chamber. The isentropic efficiency coefficients of the converging section and diverging section of motive nozzle affects the entrainment ratio and compression ratio differently. The main irreversibility occurs in the steam nozzle (41.34%) and mixing chamber (57.95%). The exergy efficiency of the injector decreases with the increase of the mass entrainment ratio. It also increases in coordination with the increase of inlet steam pressure, and decreases with the increase of inlet water pressure.

1. Introduction

Injector and ejector, are important devices used in many industrial applications, because they are simple, without moving parts and do not need an external energy supply system [1,2]. Generally, recovering energy and boosting pressure are the main purposes of their application. For an ejector, usually both the primary flow and secondary flow are steam or vapor. For an injector, the primary flow is usually steam or vapor, while the secondary flow is liquid. The injector is also referred to as the jet pump in many applications. Furthermore, there exists a profound difference between the ejector and injector. For example, the entrainment ratio of the ejector is generally less than 1 [1,3], while the entrainment ratio for the injector is much greater than 1 [4]. Moreover, the exit pressure of an ejector is lower than the primary flow pressure [1], while the exit pressure of an injector can be higher than the primary flow pressure [2]. The physical process inside the ejector and injector is also substantially different. Inside the ejector, a shock wave train occurs from the nozzle exit to the mixing chamber, and its structure, such as the shock wave length and expansion angle, affects the ejector's performance [5–7]. With the injector, especially the widely utilized steam-water injector, there is direct contact condensation between the steam and water, and a condensation shock occurs within the mixing chamber [2]. Moreover, the steam-water interface plays an important role for heat, mass and momentum exchange [8]. The steam jet may also transit from being stable to divergent and it exhibits diverse patterns [9]. Regarding the history of ejectors and its current applications and development, the readers may refer to review papers written by Elbel [10], Besagani et al. [1], Chen et al. [11], etc. Moreover, the injector, as a passive jet pump, is extensively used in numerous industrial applications [8]. Since it has significant heat exchange abilities, it is presently being investigated for utilization as a passive cooling system for light water reactors [2].

To further enhance the understanding of its physical process and performance, a substantial amount of studies are based on zero or one dimensional ejector modelling [1]. In the 1950s, Keenan and Neumann [12] introduced a constant-pressure mixing model, and later added a constant-area mixing model. Eames et al. [13] expounded on the friction loss inside the injector, and conducted an experiment to validate their

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Nomenclature

A	cross section area
C	speed of sound ($m\,s^{-1}$)
C_p	pressure recovery coefficient
D	diameter (m)
$\dot{E}$	exergy (kW)
h	specific enthalpy ($kJ\,kg^{-1}$)
h_L	head loss
I	exergy destruction rates (kW)
$\dot{m}$	mass flow rate ($kg\,s^{-1}$)
P	pressure (MPa)
R	compression ratio
s	specific entropy ($kJ\,kg^{-1}\,K^{-1}$)
T	temperature (K)
u	velocity ($m\,s^{-1}$)

Greek letters

η	coefficient of isentropic efficiency
ρ	density
ω	entrainment ratio
ξ	loss coefficient
β	momentum correction factor
φ	exergy efficiency

Subscripts

is	isentropic
1–8	stated points

own model. Shen et al. [14] proposed an optimization design methodology for the ejector. Munday and Bagster [15] further developed the constant-pressure mixing model by assuming that the primary fluid fans out and forms a "hypothetical -throat" prior to mixing with the entrained fluid, and offered a semi-empirical formula. Huang et al. [16] performed a 1D analysis regarding ejector performance by assuming double-choking before mixing both the primary and secondary flow, which was widely employed in later research. Zhu et al. [17] proposed a 2D expression for velocity distribution, so as to approximate the viscosity flow on the cross-section of the ejector, by introducing a "shock circle" at the entrance of the constant area mixing chamber. In all of the above modelling, the irreversible loss is usually taken by selecting isentropic coefficients for the primary flow nozzle, secondary flow nozzle, and the diffuser, with typical values ranging from 0.8 to 0.95 [16,18,19]. The irreversible loss of momentum inside the mixing chamber is also assessed by another coefficient, namely, the mixing loss coefficient, with a typical value from 0.7 to 0.9 [16,18,19]. It is assumed that critical flow is reached at the nozzle throat [16–18].

For the steam–water injector, different modelling approaches have been applied. Usually, empirical coefficients are involved, which are limited in their usage. In order to predict the exit pressure at the steam nozzle, numerous methods were proposed. Cattadori et al. [20], Yan et al. [4], and Zhang et al. [21] utilized empirical relations obtained from their experiments. Li et al. [22] assumed an isentropic process within the steam nozzle, while Trela et al. [23] multiplied a coefficient on the velocity value that was calculated from the isentropic process. Narabayashi et al. [24] assumed that critical flow is attained at the steam nozzle exit, and they employed another empirical relation to calculate the critical pressure at the steam nozzle exit.

Similar to the steam nozzle, the calculation of the water nozzle also involves empirical relations. Cattadori et al. [20] assumed the exit pressure at the water nozzle equals the steam pressure at the steam nozzle exit, which is the same as the ejector. Beithou and Aybar [25], and Trela et al. [23] also made the same assumption. Yan et al. [4] provided an empirical relation for computing the pressure at the water nozzle exit, while Zhang et al. [21] provided another empirical relation.

Since the condensation primarily occurs inside the mixing chamber, numerous models have been proposed to take this phenomena into consideration. Deberne et al. [26] developed a simple model of the mixing section and the shock wave, which requires one empirical closure equation. Beithou and Aybar [25] designed a mathematical modeling of the steam-driven jet pump without condensation shock, in which a condensation profile was utilized; however, they did not take the mixing loss into account. Yan et al. [4] adopted the same approach as Beithou and Aybar [25], yet they took the mixing loss into consideration with an empirical coefficient. Trela et al. [23] used an empirical heat transfer correlation to calculate the exit temperature of the mix nozzle. Furthermore, Deberne et al. [26] assumed that steam and liquid have the same pressure value inside the mixing nozzle; thus, they used an equivalent pressure obtained from an empirical relation of the condensation rate to calculate the mixing nozzle pressure. Li et al. [22] also utilized an empirical correlation of the condensate rate, in order to determine the fluid state inside the mixing nozzle. Other models, however, are more complex. These used a 2D approach or two phase model, with a two or three fluid approach. Narabayashi et al. [24] conducted an analytical and experimental study on water-steam injectors. The authors utilized a 2D axisymmetric and steady state formulation, where phases were treated as separate, homogeneous and immiscible. Manno and Dehbi [27] divided the mixing nozzle into two flow regimes, separated flow and dispersed flow, and developed a separate mathematical model for each. Recently Heinze et al. [28] utilized a one-dimensional three-fluid model for the direct condensation of steam jets in flowing water. In the diffuser of the steam-water injector, a single phase water flow was taken and Bernoulli's equation was adopted to model the process [2]. All the researchers included a certain loss coefficient in their models.

However, in all the above-mentioned studies on steam-water injectors, detailed analyses on the different sections of the steam nozzle are rare. Although a critical flow condition is assumed, the fluid state at the nozzle throat was not given. These studies often utilized empirical relations to compute the steam nozzle exit pressure, which is quite limited. In this paper, the converging and diverging sections of the steam nozzle will be thoroughly investigated. Additionally, the pressure and temperature at the nozzle throat will be calculated, based on the local sound velocity reached at critical flow conditions. This approach, which was used by Liu et al. when predicting an ejector [29], will provide a better method by which to calculate the nozzle exit pressure.

Furthermore, exergy analysis is crucial for evaluating the efficiency related to ejector performance enhancement [18,30,31,32]. However, limited studies are focused on the injector. Trela et al. [33] conducted as exergy analysis of a two-phase steam-water injector; they pointed out that the exergy efficiency of the injector can be quite high, from 27% to 45%.

Moreover, the injector can be utilized in district heating systems, because of its compact size and no need for external energy. Since the injector can be used as a pump in district heating systems driven with high pressure steam to replace conventional electric-driven pumps, it is a viable alternative for reducing electricity cost. Yan et al. [4] experimented on the performance of a steam-driven jet injector with a high inlet water temperature (maximum 341.15 K) for a district-heating system and analyzed the effect of the inlet steam pressure, inlet water pressure and temperature on injector performance. They ascertained that the lifting-pressure coefficient was significantly affected by the nozzle throat area of the mixing chamber. However, no other geometric parameters were discussed; neither was an exergy analysis performed.

First, a one-dimensional mathematical model of the steam-water two-phase injector was developed, in which an iterative calculation of