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On the design and corresponding performance of steam jet ejectors

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HIGHLIGHTS

- 1-D model calculating all ejector dimensions is proposed and applied to base case.
- Polytropic efficiency quantifies irreversibility during acceleration/deceleration.
- Parametric studies show effect of shock position & mixing efficiency on dimensions.
- Model predicts axial evolution of pressure, temperature, Mach number, etc.

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ABSTRACT

A thermodynamic model for the design of ejectors is described, validated and applied for conditions prevailing in a multiple effect desalination system with thermal vapour compression. The model, which calculates all the dimensions of the ejector, incorporates several innovations. Firstly, it uses polytropic efficiencies to account for irreversibilities during the acceleration and deceleration processes thus taking into consideration the effect of the pressure ratio on the corresponding losses. Secondly, the proposed parametric relations between the pressure at the fictive throat and at the primary nozzle outlet constitute a generalization of particular geometries and operating conditions treated in previous studies. Thirdly, it considers mixing processes which take place with simultaneous pressure and area changes. The model has initially been used to determine the ejector dimensions and fluid properties for a base case. Parametric studies have then been conducted to demonstrate the existence of design choices which minimize the size of the primary nozzle and the length of the constant diameter duct.

1. Introduction

Jet ejectors have been known for more than a century and are used in several industrial processes as pumps, gas ejectors or vapour compressors in ejector operated refrigerators [1] and in desalination systems [2]. They can be used with gases, vapours and liquids, can be activated by low-temperature heat sources such as solar energy or waste heat and have no moving parts thus requiring practically no maintenance. However, the fluid flow and thermodynamic transformations taking place in ejectors are complex and therefore difficult to model and control.

The father of ejector technology is Charles Parsons who used them to extract the air from the condensers of steam engines and their first application in refrigeration was introduced by Maurice Leblanc in 1910 [1]. A significant early model of the phenomena taking place in the ejector was published by Keenan et al. [3] who used the one-dimensional equations of mass, momentum and energy conservation assuming perfect gas behaviour as well as isentropic expansion and diffusion. This approach does not reflect the complexities due to real fluid properties and irreversibilities. Thus, for example, it has been shown experimentally [4] and numerically [5] that the entrainment ratio of an ejector does not depend on the ratio of the primary to secondary total pressures as predicted by perfect gas theory but, rather, on the individual values of these pressures. Therefore, more recent studies use real fluid thermophysical properties [6–9] and isentropic efficiencies or friction correlations [7,8,10,11] to model the performance of ejectors. All of the above studies are based on the one-dimensional integral form of the conservation equations and assume uniform steady flow conditions.

The development of powerful and fast computers has provided the impetus for more sophisticated modelling of the flow and thermal fields in ejectors using the differential forms of the conservation equations [12,13]. This CFD (Computational Fluid Dynamics) approach requires considerable computer facilities but provides more detailed information regarding the complex phenomena (such as interactions between shock waves and boundary layers, phase changes, two- and three-dimensional effects) which take place in ejectors.

An extensive review of integral (or thermodynamic) and differential (or CFD) ejector models as well as relevant empirical correlations based on measured data has been compiled by He et al. [14]. Some newer studies are included in the references of more recent articles cited in

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Nomenclature

A	cross-section area
C	sound velocity
D	diameter
$\dot{E}$	exergy flux
F	force
f	friction factor
h	specific enthalpy
L	length
$\dot{m}$	mass flow rate
M	Mach number
P	pressure
Re	Reynolds number
s	specific entropy
T	temperature
V	velocity
x	quality of liquid/vapour mixture
y	axial coordinate

Greek letters

η	efficiency
θ_1, θ_2	converging, diverging angles
ρ	density

Subscripts

0	stagnation conditions
1, 2	thermodynamic states
d	downstream of shock
is	isentropic
m	fully mixed state
mix	mixing
p	primary flow
pl	polytropic
s	secondary flow
t	throat
u	upstream of shock

the present list of references [9,11,13]. Despite this large number of works on modelling and analysis of ejector performance further efforts are still needed in order to (among others) "study the influence of variable isentropic coefficients which are taken as constant in almost all existing thermodynamic models" [14].

In view of this situation the present study was undertaken to investigate the effects of the efficiency of the primary and secondary expansions as well as that of the diffuser on the dimensions and operating conditions of ejectors. Other objectives of the present study are to determine the effect of the pressure at the primary nozzle exit (or, equivalently, of the position of the fictive throat) and of the normal shock position on the ejector design and operating conditions. For this purpose a thermodynamic model of an ejector has been formulated and solved. It incorporates the concept of the polytropic, or infinitesimal stage, efficiency which is used extensively in the design and analysis of compressors and turbines [15] but has only once been applied to the study of ejectors [16]. The results were calculated for conditions corresponding to the operation of a multi-effect evaporation (MEE) thermal-vapour-compression (TVC) desalination system analysed in a previous study [17].

2. Description, modelling and numerical solution

Fig. 1 shows the geometry of the ejector. It also defines some of its dimensions and the Cartesian coordinate system used in this study ($y = 0$ at cross-section 1p). The primary, or motive, fluid is a vapour supplied at known stagnation conditions (P_{0p}, T_{0p}). It is accelerated to critical conditions at the throat (cross-section 2p) in the converging nozzle of length L_1 and to supersonic speeds in the diverging nozzle of length L_2. It thus creates a very low pressure at the nozzle exit plane (cross-section 3p, situated at $y = L_1 + L_2$) which aspirates the secondary, or entrained, fluid whose stagnation conditions are also known (P_{0s}, T_{0s}). The secondary fluid inlet is at $y = L_1$, i.e. opposite the throat of the primary nozzle. From $y = L_1 + L_2$ to $y = L_1 + L_2 + L_3 + L_4$ the diameter of the outer duct diminishes forming a second converging nozzle. Based on the model proposed by Munday & Bagster [18] (and used by Selvaraju & Mani [7] and Huang et al. [19] among others) we assume that the primary fluid flows out of the primary nozzle without mixing with the secondary fluid immediately. It expands in the converging section of length $L_3 + L_4$ forming a converging duct for the secondary fluid which continues to accelerate until it reaches critical conditions at 4 s, the fictive throat. At that plane, which is situated at $y = L_1 + L_2 + L_3$ upstream of the beginning of the constant–area section of the ejector (cross-section 5, situated at $y = L_1 + L_2 + L_3 + L_4$), the pressures of the primary and secondary fluids are equal ($P_{4p} = P_{4s}$). Then the mixing process begins and is completed at cross-section "m" in the constant-area section of the ejector. A normal shock occurs after complete mixing, at $y \le L_1 + L_2 + L_3 + L_4 + L_5$ (i.e. upstream of cross-section 6) causing an increase of the pressure and a decrease of the velocity from supersonic to subsonic. A further pressure increase takes place in the subsonic diffuser of length L_6 which brings the flow to stagnation conditions at cross-section 7.

The adopted geometry is more realistic than in the studies which assume that pressure equality of the two streams takes place at the exit of the converging–diverging nozzle [3,4,5,9] and than those which assume constant area mixing [4,5,9]. It is also more general than that adopted by Fang Liu & Groll [11] who assumed that the primary nozzle does not have a diverging part ($L_2 = 0$). The flow-field description is also more general than that used by Selvaraju & Mani [7] who assumed that the normal shock occurs at the end of the constant-area section.

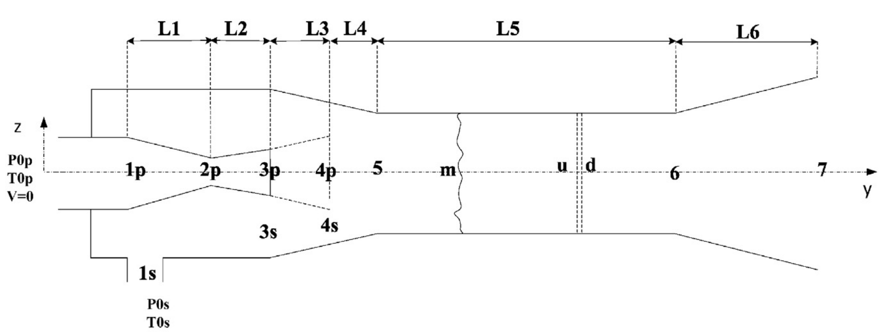

Fig. 1. Geometry of the ejector.