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Ejector design and performance prediction

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ABSTRACT

This paper proposes a 1-D thermodynamic model for determining the critical pressure ratio, the mixing efficiency and all the dimensions of an optimum ejector providing the highest possible compression ratio for fixed inlet conditions and mass flowrates of the motive and suction fluids. The maximization of the back pressure is obtained subject to constraints imposed by the 2nd law of thermodynamics and the requirements that the flow must be subsonic at the diffuser entrance, that the mixing efficiency must be positive but smaller than one and that the length to diameter ratio for the constant area duct must be between fixed limits recommended in previous studies. The paper also describes a method for determining the off-design performance of a fixed geometry ejector which reproduces the experimental relations between the entrainment ratio, the compression ratio and the inlet conditions of the two fluids. The model uses a fixed polytropic efficiency (rather than the fixed isentropic efficiency used in previous studies) to simulate the acceleration and deceleration processes thus taking into account the effect of the pressure ratio during off-design operation. Examples of its application for isentropic and irreversible acceleration/deceleration of a perfect gas are provided and their results are analysed and compared.

1. Introduction

An ejector is a simple apparatus using the low pressure created by the accelerated stream of a primary (or motive) fluid to aspirate and compress a secondary (or suction) fluid. Ejectors are used in steam power plants to create the vacuum in the condenser as well as to remove ash from the boilers and the flue gas, in boiling water nuclear reactors to circulate the coolant, for the handling of granular materials, for pumping turbid water and slurries, for medical uses (suction of bodily fluids) and to improve the performance of certain desalination plants. Ejector refrigeration systems, which were very popular in the early 1930s, are also receiving renewed interest since they can be activated by low temperature thermal energy from renewable sources or thermal wastes thus reducing the use of fossil fuels or improving the efficiency of their usage. Although in some applications one of the two fluids can be a liquid and the two fluids may be dissimilar, the present study focuses on the more usual cases where the two fluids are identical gases or vapours.

Fig. 1 shows the main parts of an ejector and can be used to describe qualitatively its operation. The converging–diverging nozzle serves to accelerate the primary fluid from its stagnation conditions (P_{p0}, T_{p0}, $V_{p0} = 0$) to sonic conditions at the throat p2 and to supersonic conditions with a very low pressure at its exit p3. This low pressure aspirates and accelerates the secondary fluid from its stagnation conditions (P_{s0}, T_{s0}, $V_{s0} = 0$) to the low-pressure high-velocity state s3. In general at section 3 the pressure, temperature and velocity of the two streams are not the same [1]. The two streams then mix and the flow can be considered fully mixed at section 5, somewhere in the constant area duct. Normally the flow is subsonic at the inlet of the diffuser (section 6) and decelerates towards the outlet stagnation conditions (P_{70}, T_{70}, $V_{70} = 0$) in the diffuser. The relative position of the different cross-sections is specified by their axial distance “X” from cross-section p1; thus $X_{P1} = 0$ while X_7 represents the total length of the flow field. As can be seen from this description the ejector has no moving parts; therefore it does not require lubrication and suffers negligible wear.

Ejector performance has been discussed in a great number of published articles presenting experimental results and models of the flow field. Their main differences are due to the geometry of the ejector (ex: constant area mixing, i.e. $A_3 = A_4 = A_5 = A_6$), the nature of the fluids (identical or different primary and secondary fluids, perfect gas or various real fluids such as steam or natural and synthetic refrigerants) and the assumptions concerning the flow field (with or without losses during acceleration, mixing and deceleration). Three extensive review articles [2–4] have described

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Nomenclature

A	Cross-section area
a, b	Cross-sections before, after normal shock
C_p	Constant pressure specific heat
C_v	Constant volume specific heat
D	Diameter
F	Force
Fr	Fraction
f	Friction factor
$\dot{m}$	Mass flowrate
N	Number of steps in iteration procedure
P	Pressure
PR	Compression ratio ($PR = P_{70}/P_{s0}$)
R	Gas constant ($R = C_p - C_v$)
R_m	$R_m \equiv (P_6 - P_{s3})/P_{p3}$
s	Specific entropy
T	Temperature
V	Velocity
X	Axial position of a cross-section
y	Elementary pressure/temperature ratio for acceleration/deceleration processes

Greek letters

γ	Ratio of specific heats ($\gamma = C_p/C_v$)
ε	$\varepsilon = \gamma/(\gamma - 1)$
η	Isentropic (or overall) efficiency
η^*	Polytropic (or elemental) efficiency
ρ	Density
ω	Entrainment ratio ($\omega = \dot{m}_p/\dot{m}_s$)

Subscripts

0, 1, …	Thermodynamic states (Fig. 1)
a	Conditions before normal shock
b	Conditions behind normal shock
d	Diffuser
in	Control volume inlet
m	Mixing
out	Control volume outlet
p	Primary
s	Secondary

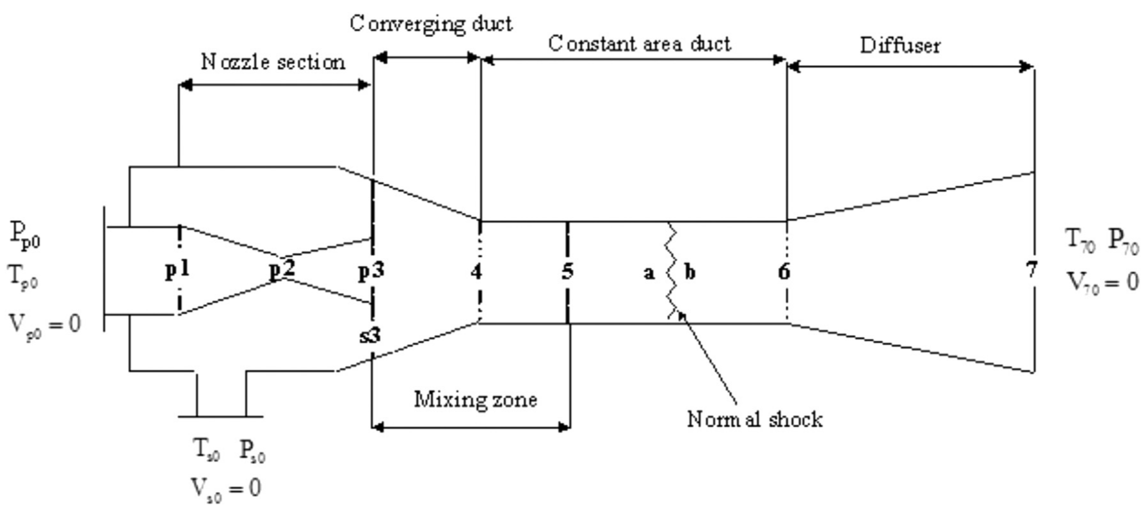

Fig. 1. Schematic representation of ejector geometry.

their methodology and conclusions. The results of some published articles are discussed and used in appropriate sections of the present text.

2. Performance characteristics, design considerations and model foundations

Typical experimental results [5–7] are qualitatively illustrated in Fig. 2. Fig. 2a shows that for a given geometry and fixed inlet conditions the entrainment ratio ($\omega = \dot{m}_s/\dot{m}_p$) is independent of the back pressure (P_{70}) when the latter is below a critical value P^*; for such conditions the primary and secondary flows are choked so that any variations of the back pressure have no influence upstream of section 3. For back pressures higher than P^* the secondary flow is subsonic and its flowrate $\dot{m}_s$ decreases rapidly as the back pressure increases. It becomes zero when the back pressure reaches the limiting value P_{lim}; for back pressures higher than P_{lim} the ejector malfunctions, i.e. part of the primary flowrate $\dot{m}_p$ is diverted and exits through the secondary inlet. If the pressure of the motive fluid P_{p0} is increased the maximum entrainment ratio decreases while the critical and limiting back pressures increase. Fig. 2b illustrates the effect of P_{p0} on the entrainment ratio for fixed inlet conditions of the secondary fluid and a fixed back pressure. Operation with values of P_{p0} up to the one corresponding to the maximum entrainment ratio is in the double choked mode while for P_{p0} values beyond this threshold only the primary flow is choked. If the secondary inlet pressure is increased the entrainment ratio increases while the primary pressure corresponding to the maximum value of ω decreases. These figures show that the performance of a fixed geometry ejector is defined by the boundary conditions at the two inlets (p0, s0) and the single outlet (70).

When designing or choosing an ejector for a particular application both the entrainment ratio ω and the compression ratio ($PR = P_{70}/P_{s0}$) must be considered. This fact can be easily justified in