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Ejectors and jet pumps. Design for steam driven flow

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EJECTORS AND JET PUMPS. DESIGN FOR STEAM DRIVEN FLOW

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EJECTORS AND JET PUMPS. DESIGN FOR STEAM DRIVEN FLOW

1. NOTATION AND UNITS

A coherent set of mass, length, time and temperature units must be used in all the equations in this Data Item. Two coherent sets of units - British and SI - are given below.

Note:	1 slug	= 32.174 lb mass
	0°F	= 459.7°R
	1 Btu/lb	= 24 990 ft lbf/slug
	1 torr	= 1mm Hg abs.

		SI	British
A	cross-sectional area of component	m^2	ft^2
A_R	area ratio, A_m/A_e	—	—
A_R^*	area ratio, A_m/A_{th}	—	—
A_T	area ratio, A_e/A_{th}	—	—
a	speed of sound in fluid	m/s	ft/s
C_D	primary-nozzle discharge coefficient	—	—
C_1	correction factor for gases other than air (see Section 5.4.2)	—	—
C_2	correction factor for air temperatures above 20°C (see Section 5.4.2)	—	—
c_p	specific heat capacity at constant pressure	J/kg K	ft lbf/slug°R
C_{PR}	ratio of specific heat capacities, c'_p/c''_p	—	—
D	diameter of narrowest section of mixing chamber	m	ft
d	diameter of primary nozzle	m	ft
h_{fg}	specific enthalpy of vaporisation	J/kg	ft lbf/slug
K_d	diffuser loss coefficient (steam/liquid ejectors) (Equation (B2.3))	—	—
K_m	mixing chamber loss coefficient (Equation (B2.2))	—	—
K_s	secondary inlet loss coefficient (steam/liquid ejectors) (Equation (B2.1))	—	—

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L	length of parallel section of mixing chamber	m	ft
L_d	length of exit contraction or diffuser	m	ft
M	Mach number, U/a	—	—
$\dot{m}$	mass flow rate	kg/s	slug/s
N_1	pressure ratio, $(p_{t5} - p_{t0})/p_{t1}$	—	—
N_2	pressure ratio, $(p_{t5} - p_{t0})^{1/2} \rho'_e U_e'^2$	—	—
N_p	primary pressure ratio, p_{t1}/p_{t5}	—	—
N_s	secondary pressure ratio, p_{t5}/p_{t0}	—	—
n	number of stages in multistage ejector	—	—
p	absolute static pressure	N/m ² (Pa)	lbf/ft ²
p_t	absolute total pressure	N/m ² (Pa)	lbf/ft ²
p_v	absolute vapour pressure	N/m ² (Pa)	lbf/ft ²
R	gas constant (for steam, $R = 461.5$ J/kg K, or 4970 ft lbf/slug°R)	J/kg K	ft lbf/slug°R
R_R	ratio of gas constants, R'/R''	—	—
r_c	density ratio for steam/liquid ejectors, ρ'_1/ρ''	—	—
r_{c1}	density ratio for steam/liquid ejectors, ρ'_e/ρ''	—	—
r_m	mass flow ratio, $\dot{m}''/\dot{m}'$	—	—
s	distance from primary-nozzle outlet to beginning of constant-area section of mixing chamber	m	ft
T	absolute static temperature	K	°R
T_s	absolute saturation temperature	K	°R
T_t	absolute total temperature	K	°R
T_R	ratio of inlet total temperatures, T_{t1}/T_{t0}	—	—
U	flow velocity	m/s	ft/s
W	inverse of mass flow ratio, $1/r_m$	—	—
γ	ratio of specific heat capacity at constant pressure to that at constant volume (≈ 1.1 (average) for saturated steam; ≈ 1.315 (average) for superheated steam)	—	—

γ_R	ratio of specific heat ratios, γ'/γ''	—	—
η_d	contraction or diffuser loss coefficient, p_{t5}/p_{t4} (steam/gas ejectors)	—	—
η_s	secondary inlet loss coefficient, p_{te}''/p_{t0} (steam/gas ejectors)	—	—
ρ	fluid density	kg/m ³	slug/ft ³
ϕ_1	included half-angle of convergent section of mixing chamber	degree	degree
ϕ_2	included half-angle of diffuser	degree	degree
ΔT	degree of subcooling, $(T_1 - T_0)$	K	°R

Subscripts

c	denotes conditions leading to cavitation
e	primary nozzle exit plane
f	denotes property of saturated liquid
fg	denotes change of phase at constant pressure
g	denotes property of saturated vapour
m	mixing chamber
t	denotes total value (see Section 5.3)
th	primary nozzle throat
u	denotes multistage unit
0	secondary flow entry plane
1	primary nozzle entry plane
3	entry plane of parallel section of mixing chamber
4	mixing chamber exit plane
5	contraction or diffuser exit plane

The reference planes are defined in Sketch 1.1. For constant area mixing, planes e and 3 are coincident and have been referred to by subscript e .

Superscripts

'	refers to primary stream or primary nozzle
''	refers to secondary stream

Primary flow

Secondary inlet flow

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2. INTRODUCTION

This Item is one of a series of ESDU Data Items concerned with the design and performance of ejectors and jet pumps. Such devices are characterised by the use of the kinetic energy of one fluid stream (the primary, motive or driving flow) to drive a second fluid stream (the secondary, induced or driven flow) by direct mixing. The design parameters, requirements and methods vary considerably depending on whether the working fluids are gases, liquids, vapours or mixtures of these components. Each type is therefore considered in a separate Data Item. Derivation 25 considers ejectors in which the primary and secondary fluids are both air and Derivation 26 considers ejectors in which they are both liquids. This Item considers steam driven ejectors.

The terms “ejector” and “jet pump” are alternative names for the same device and the term “injector” is also used. Although common usage, it is not strictly correct to assume that the terms “ejector” and “injector” are used when the working fluids are gases and the term “jet pump” when they are liquids.

2.1 Purpose and Scope of This Item

This Data Item provides information for the design and performance evaluation of ejectors, jet pumps or injectors in which the primary fluid is steam and the secondary fluid is a liquid or a gas. The terms steam/liquid and steam/gas ejector are employed according to whether the secondary fluid is a liquid or gas. Techniques for the optimum design of a steam driven ejector are presented. The method for steam/liquid ejectors is based on a theoretical analysis, that for steam/gas ejectors is based on experimental data obtained on a wide range of steam driven ejectors. A performance prediction method for steam/liquid ejectors is also presented, based on the same theoretical analysis as is used in the design method. A performance prediction method for steam/gas ejectors is discussed briefly.

2.2 Layout of This Item

Section 3 of this Item discusses some applications of steam driven ejectors.

Section 4 describes the principles of ejector operation and defines the different components. These may vary considerably with application and Section 4 considers some of the configurations possible.

Section 5 describes design methods that can be used to determine the on-design operating conditions and optimum dimensions of a steam driven ejector. Some considerations regarding mechanical design are given in Section 5.5.

Section 6 describes a method by which the performance of a steam/liquid ejector may be evaluated. Performance prediction for steam/gas ejectors is also discussed.

Section 7 describes designs such as multi-stage, annular nozzle and multi-nozzle ejectors and considers their advantages in certain situations.

Section 8 discusses some of the problems that may arise when an ejector is in operation.

Section 9 presents worked examples that illustrate the application of the Design and Performance Prediction Methods.

Section 10 lists sources of information used in the preparation of this Item together with further information.

Appendix A contains a glossary of terms used in describing ejector components, design parameters and performance.

Appendix B presents detailed theoretical analyses of the fluid flow through ejectors. These analyses are based on one-dimensional flow representations of mass, momentum and energy conservation.

3. APPLICATIONS OF STEAM DRIVEN EJECTORS

Steam driven ejectors are used extensively in the power generation, chemical processing and nuclear industries. Their main advantage is that they have no moving parts and hence require little maintenance. Almost any type of liquid, gas or vapour can be pumped, and solids-bearing fluids can be handled. Compared with mechanical pumps, steam ejectors have very low efficiencies when used in normal pumping applications but when a source of waste or low grade steam is available, a steam ejector may be cheaper to operate than a mechanical pump. Steam ejectors also have many applications, such as heating, humidifying and pumping toxic and solids-bearing fluids, where a mechanical pump may be unsuitable.

In some applications, a steam driven ejector may have advantages over other types of ejector, such as air driven and liquid driven ejectors. In gas pumping applications, the primary steam flow can be separated from the secondary gas by condensation - an option which is not available for air driven ejectors. Combustible gases can be handled more safely. In liquid pumping applications, a steam driven ejector causes far less contamination of the pumped liquid than a liquid driven ejector. This is because the steam condenses within the ejector and so occupies only a small fraction of the volumetric discharge from the ejector. Offset against these advantages is the cost of the steam supply, which usually far exceeds the cost of an equivalent compressed air or liquid supply.

Some typical applications of steam driven ejectors are discussed below. Although many different uses are considered, it is worth noting that most commercially supplied steam/gas ejectors are used in vacuum pumping applications and are usually sold as multistage units. The most common uses of steam/liquid ejectors are as pumps for toxic or radioactive liquids, as liquid heaters and as feed water injectors.

3.1 Applications of Steam/Liquid Ejectors

Condensers

When a steam driven ejector is used to pump a liquid, it is inevitable that the steam condenses. In a well designed device, the discharge flow from the ejector should not contain any uncondensed steam. The condensing characteristics of steam driven ejectors make them suitable for use as maintenance-free condensers, and they have been used in this role in the nuclear industry.

Feed water injectors

Steam driven ejectors are widely used to feed water to water boilers. In this role, they are often referred to as feed water injectors. The injector is driven by a steam bleed from the boiler. The driving steam performs the dual function of pressurising and heating the water before it enters the boiler. An early use of feed water injectors was in steam locomotives.

Liquid heaters

A steam driven ejector, when immersed in a liquid, simultaneously heats and recirculates the liquid. The ejector then acts primarily as a heater and all of the thermal energy of the steam is transferred to the liquid. Steam ejectors for use as liquid heaters are supplied in several forms. When used to heat the liquid in a tank, the ejector can be immersed in the tank as required or be attached permanently to the outside of the tank. Often an air bleed line is fitted in the secondary inlet so that the ejector performs the dual role of heating and agitating or aerating the liquid. Steam ejector heaters are also used in pipeline applications, the ejector usually being supplied as a self contained Tee-piece for insertion into the pipeline.

One particular use of steam driven ejector heaters is in reducing the viscosity of sludges and other viscous liquids.

Pumping liquids and slurries

Steam driven ejectors are widely used as pumps in the chemical processing and power generating industries. They may be installed permanently in the plant or may be portable for use in tank draining and other cleaning operations. Liquids and suspended solids can be pumped. The absence of moving parts makes steam driven ejectors particularly suitable for handling highly toxic or radioactive liquids and slurries. A possible disadvantage of steam driven ejectors in liquid pumping applications when compared with mechanical pumps is that the pumped liquid or slurry is heated and contaminated by the condensed steam.

Steam desuperheaters

A common requirement in the power generating and chemical processing industries is to desuperheat (*i.e.* cool) superheated steam. Steam ejectors are often used for this purpose. The superheated steam is used to drive the ejector. Water is fed to the secondary inlet and cools the superheated steam. A temperature controller at the outlet from the ejector adjusts the water feed rate to maintain the desired steam outlet temperature.

3.2 Applications of Steam/Gas Ejectors

Gas pumps and humidifiers

Steam driven ejectors can be used to pump gases or boost gas pressures in a system. The gas may be dust bearing. Compared with air driven ejectors, the main advantage of steam driven ejectors is that most of the driving steam can be removed in an aftercondenser, leaving the gas only slightly contaminated by the driving fluid. Also, steam driven ejectors can be used in applications where use of an air driven ejector would create a combustible mixture. A common use of steam driven ejectors as gas pumps is in ventilation systems. In closed loop systems, the ejector may perform the dual role of ventilating and humidifying the recirculated gases.

Thermo-compressors

A steam driven ejector may be used to compress a low pressure vapour to a higher pressure. The term thermo-compressor is often used to describe these devices. Thermo-compressors are widely used to compress low grade steam, although almost any vapour can be compressed.

Vacuum pumps

Steam driven ejectors can be used to produce or maintain a vacuum in a gas filled vessel, or simply to evacuate a gas. In this role they are used in refrigeration units, condensers, filtration units, molten metal degassers, evaporators, dryers and in many other processes requiring sub-atmospheric pressures. Multi-staged units with intercondensers can achieve vacuums well below 1 torr. More information on this widespread application of steam/gas ejectors is given in Section 7.1.

4. THE BASIC STEAM DRIVEN EJECTOR

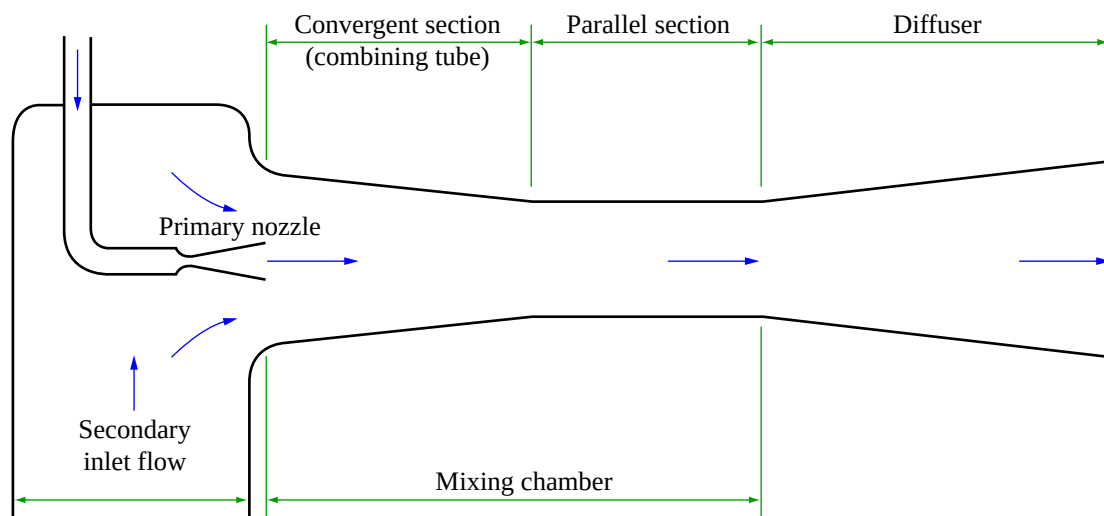
4.1 Principle of Operation

An ejector is a device in which the kinetic energy of one fluid stream (the primary fluid) is used to drive another fluid stream (the secondary fluid). The primary fluid is usually supplied through a nozzle and issues as a jet into a duct (the mixing chamber) which contains the secondary fluid.

Ejectors using steam as the primary fluid can be used to pump both gaseous and liquid secondary fluids. In a steam/gas ejector the gas is induced into motion by the turbulent mixing and entrainment at the edges of the steam jet. In a steam/liquid ejector, the steam jet and liquid move initially as an annular flow in the mixing chamber. Mixing may take place gradually, as the steam condenses, but usually occurs suddenly at a condensation shock. In both types of steam driven ejector, mixing may occur violently following a compression shock in the steam flow.

In a well designed ejector, the primary and secondary fluids should be fully mixed by the end of the mixing chamber. Often, this mixed fluid is discharged through a diffuser, which is used to obtain some further static pressure recovery.

The basic theory of the flow through a steam driven ejector, derived by considering mass, momentum and energy conservation, is described in Appendix B. Separate derivations are presented for steam/liquid and steam/gas ejectors.



Sketch 4.1 Component parts of a steam ejector with a single-central primary nozzle

4.2 Component Parts

Sketch 4.1 shows a typical configuration for a steam driven ejector. The ejector consists of four main components: the primary nozzle, the mixing chamber, the diffuser and the secondary inlet. It should be noted that this terminology, although widely used, is not unique. The mixing chamber and diffuser are often considered to form one unit called the diffuser. The converging section of the mixing chamber is sometimes called the combining tube. In air pumping applications the secondary inlet is sometimes called the air chamber.

Of the component parts of an ejector, only the mixing chamber is used in all applications and hence effectively characterises the device. In most commercially supplied devices the mixing chamber consists of a converging section (the combining tube) followed by a short parallel-sided section. Other configurations are possible, however, and are discussed in Section 5.5.3. The geometry and layout of the other component parts of an ejector are dependent on the application for which the ejector is intended. The main variations that are used are described below whilst more detailed information is given in Section 5.5 (Mechanical Design Considerations) and Section 7 (Aspects of Some More Complex Designs).

(a) Primary nozzle

Most designs incorporate a primary nozzle, since it is more economical to transport steam to the ejector at high pressure, rather than at high velocity with its inherently higher pressure losses. The nozzle may be convergent but is more usually convergent-divergent (provided that there is sufficient primary pressure available to achieve supersonic flow). Detailed information on the design and optimum position of primary nozzles is given in Section 5.5.1 and 5.5.4.

As an alternative the primary nozzle may be annular, producing either a discrete annular jet separate from the wall of the mixing chamber, or a wall jet located around the periphery of the mixing chamber. For a given steam flow rate, the surface area of an annular jet is greater than that from a central nozzle so that mixing is more rapid. Consequently an annular nozzle ejector requires a shorter mixing chamber than a central nozzle type. Annular nozzle ejectors are discussed in greater detail in Section 7.2.

(b) Multi-nozzle ejectors

Multi-nozzle units may be used in conjunction with a single mixing chamber and diffuser. They have the advantage of reducing considerably the mixing chamber length required. More information is given in Section 7.3.

(c) Multi-barrel ejectors

Multi-barrel units with a common secondary flow supply are sometimes used. The units are arranged in parallel and discharge into a common delivery tube.

(d) Diffuser

The diffuser raises the static pressure of the fluid discharged by the ejector. There need be no diffuser if the velocity of the fluid discharged from the mixing chamber is already suitable for existing pipework and the mixing chamber diameter is appropriate. A contraction may be used to connect the ejector to a duct of smaller diameter. This need not cause significant energy loss but will increase

the exit velocity and decrease the discharge pressure.

(e) Secondary flow inlet

The secondary fluid may be supplied at various angles to the primary stream, depending on the intended application of the ejector. High angles may introduce extra losses, although these may be negligible if the inlet velocity is low. Under steady-state conditions, there need be no secondary or induced flow if the ejector is to be used solely to maintain a fixed vacuum or discharge pressure.

(f) Multi-stage ejectors

Multi-stage units are widely used, particularly in vacuum applications. The ejectors are arranged in series, which extends the range of pressure and/or flow ratios over which the secondary fluid may be pumped. In vacuum applications, greater efficiency is achieved by condensing the steam efflux from each stage and providing a fresh supply of steam to the next stage. More information on this topic is given in Section 7.1.

5. DESIGN OF STEAM DRIVEN EJECTORS

5.1 Introduction

The performance of a steam driven ejector depends upon its geometry (shape, layout and dimensions), the properties of the steam and secondary fluid (density and, in the case of steam/gas ejectors, molecular weight and specific heat ratios), and the flow conditions (pressures, temperatures, mass flow rates) at the primary and secondary inlets and the diffuser outlet. The flow conditions define the performance of the device. Usually, the main aim in design is to determine the optimum geometry to obtain a given performance. Some or all of the flow conditions may be fixed by the application of the ejector; the remainder are found during the design stage.

This Section describes methods for designing a steam driven ejector to achieve a given performance. Different techniques are presented for steam/liquid and steam/gas ejectors. The method for steam/liquid ejectors is based upon a theoretical analysis of the flow in the ejector. Empirical loss coefficients are used to compensate for the idealisations made in the analysis. For steam/gas ejectors, the method used is based entirely on empirical results. Although the steam driven ejector is conceptually a simple device, the physical processes of turbulent mixing, heat transfer and condensation that occur within it are extremely complex and, to date, no method has been devised that can account fully for all of these phenomena. The design methods given here are therefore necessarily approximate but should be sufficient to provide a good first estimate of the final design. Some adjustment to nozzle size and/or axial positioning will always be required before the ejector can be commissioned.

The design methods assume that the optimum design, or on-design condition, corresponds to the operating condition at which the ejector uses a minimum of steam to achieve a specified pressure ratio. Different design pressure ratios are used for steam/liquid and steam/gas ejectors and these are defined in Sections 5.3 and 5.4 respectively. This definition of the on-design condition is of particular practical value since steam supply costs usually far exceed the capital cost of a steam ejector. Most commercially supplied steam driven ejectors are sold primarily on the basis of their estimated steam consumption.

Steam ejectors are normally designed to operate from a supply of dry steam for reasons explained in Section 8. It is assumed in the design methods that dry steam is used. The design methods also neglect the effects of certain phenomena - vapour freezing, cavitation and vapour binding - that can reduce the

performance of a steam driven ejector as discussed in Section 8. A correctly designed ejector should not be affected by these phenomena and the design methods given here assume that these problems will be eliminated by careful mechanical design. The method for steam/liquid ejectors includes checks that indicate whether cavitation or vapour binding may be a problem.

The design method for steam/liquid ejectors is presented in Section 5.3 and that for steam/gas ejectors in Section 5.4. Guidelines on detailed mechanical design (*i.e.* the shape and layout of components) are given in Section 5.5 and these should be followed once the calculations specified in Sections 5.3 or 5.4 have been completed.

It is convenient to describe the geometry, fluid properties and flow conditions in terms of non-dimensional parameters. These are defined in Section 5.2.

5.2 Dimensionless Parameters Used in Design

Geometry of the ejector

The main dimensionless parameters defining the geometry of a steam driven ejector are as follows.

- (1) The mixing chamber area ratio (minimum cross-sectional area of the mixing chamber/cross-sectional area of the primary nozzle exit),

$$A_R = A_m/A_e = D^2/d_e^2.$$

It is sometimes convenient to use an alternative area ratio A_R^* , which is defined by

$$A_R^* = A_m/A_{th} = D^2/d_{th}^2.$$

- (2) The primary nozzle area ratio (cross-sectional area of the primary nozzle exit/cross-sectional area of the n throat),

$$A_T = A_e/A_{th} = d_e^2/d_{th}^2.$$

Fluid Properties

For steam/gas ejectors, the main dimensionless parameters defining the fluid properties are as follows.

- (3) The ratio of specific heat ratios (specific heat ratio of steam/specific heat ratio of secondary gas),

$$\gamma_R = \gamma'/\gamma''.$$

- (4) The ratio of gas constants (gas constant of steam/gas constant of secondary gas),

$$R_R = R'/R''.$$

For steam/liquid ejectors, a single parameter, the ratio of the steam density to the secondary fluid density may be used

$$r_c = \rho'_{t1} / \rho''.$$

The design methods assume that the steam flow behaves as an ideal gas with constant specific heat ratio and gas constant. This assumption is reasonable if, as is usual for steam ejectors, the steam supply is saturated or superheated. Values of $\gamma' = 1.315$ and $R' = 461.5$ J/kg K for steam are assumed throughout.

Flow Conditions

The main dimensionless parameters defining the flow conditions at the ejector inlets and outlet are as follows.

- (5) The primary pressure ratio (steam supply pressure/discharge pressure),

$$N_p = p_{t1} / p_{t5}.$$

- (6) The secondary pressure ratio, or compression ratio (discharge pressure/secondary inlet pressure),

$$N_s = p_{t5} / p_{t0}.$$

- (7) For steam/liquid ejectors it is convenient to combine the pressure ratio parameters, N_p and N_s , into a single parameter, N_1 , defined by

$$N_1 = \frac{p_5 - p_0}{p_1} = \frac{N_s - 1}{N_s N_p}.$$

- (8) The mass flow ratio (secondary fluid mass flow rate/steam mass flow rate),

$$r_m = \dot{m}'' / \dot{m}'.$$

For steam/gas ejectors, it is necessary to define an additional parameter, the ratio of the total temperatures in the primary and secondary inlet flows,

$$T_R = T_{t1} / T_{t0}.$$

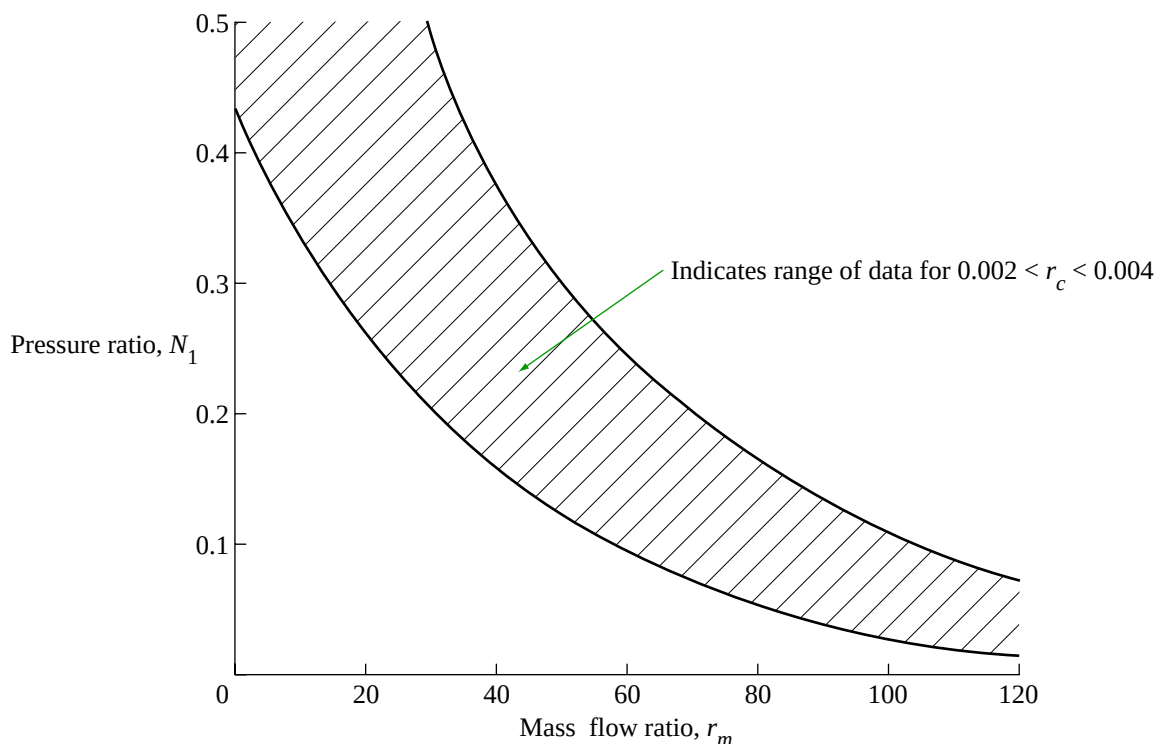
In this Data Item it is assumed throughout that flow velocities in the steam nozzle inlet, secondary inlet and diffuser outlet are small enough to be neglected so that $p_1 \approx p_{t1}$, $p_0 \approx p_{t0}$, $p_5 \approx p_{t5}$, $T_1 \approx T_{t1}$, $T_0 \approx T_{t0}$. For convenience, the subscript t is omitted in the remainder of this Data Item.

5.3 Design of Steam/Liquid Ejectors

The recommended design method for steam/liquid ejectors is based on the theoretical results that are presented in Section B2 of Appendix B. The theory is a development of the methods presented in Derivations 10 and 13. It has been validated against experimental results taken from Derivations 10, 12 and 13. In all cases, calculated values of mass flow ratio, r_m , for given pressure ratios were within 20% of

measured values. Strictly, the method is only valid for ejectors with a single central convergent-divergent nozzle and with a constant area mixing chamber. It should be used with caution for other ejector configurations.

The design charts used in the design method were derived by calculating, at each pressure ratio, N_1 , and density ratio, r_c , combinations of the area ratios, A_R and A_T , that gave the highest mass flow-ratio, r_m . These calculations were performed using typical values for the loss coefficients of $K_s = 0.10$, $K_m = 0.20$, $K_d = 0.15$, $C_D = 1.0$. It is possible that lower values for K_s , K_m and K_d could be obtained by careful mechanical design of the ejector. Ejector designs produced by this design method should therefore be regarded as typical, rather than best possible, designs. Sketch 5.1 shows a compilation of data published by manufacturers of steam/liquid ejectors. The shaded area encompasses performance data obtained on several steam/liquid ejectors for density ratios in the range 0.002 to 0.004.



Sketch 5.1 Typical performance figures for steam/ liquid ejectors

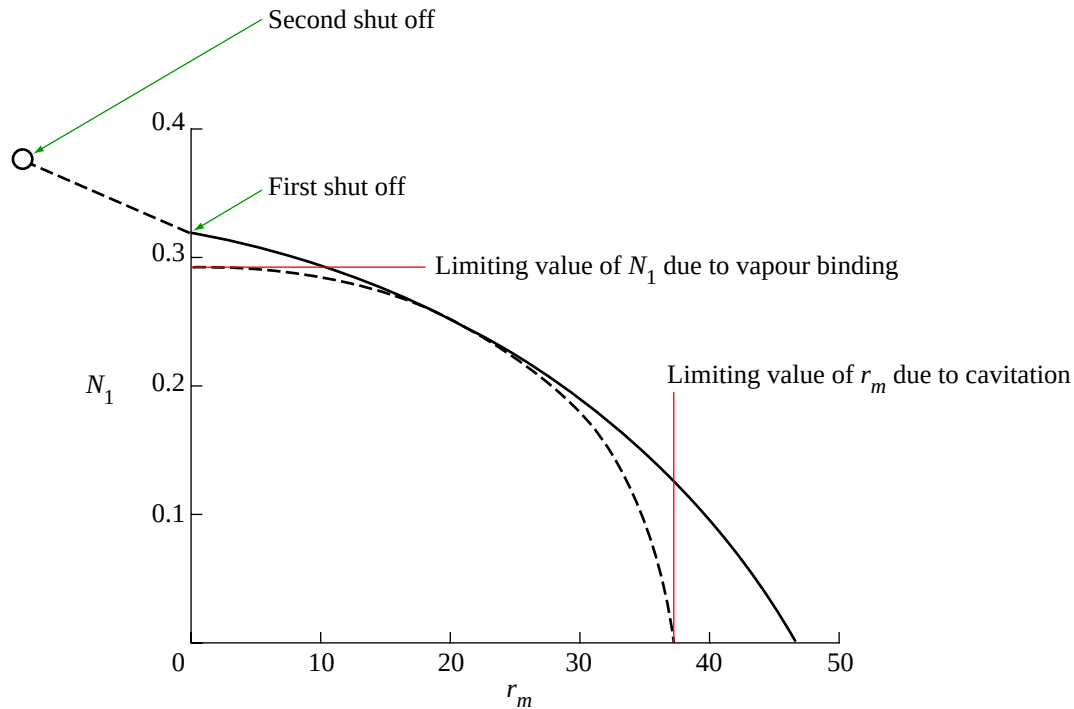
The performance of a steam/liquid ejector is limited by the phenomena of cavitation and vapour binding. These phenomena are discussed in Section 5.3.1, together with typical operating characteristics for steam/liquid ejectors. The design method is presented as a step-by-step procedure in Section 5.3.2.

5.3.1 Typical operating characteristics of steam/liquid ejectors

A typical operating curve, indicating the relationship between discharge pressure and mass flow rate over the whole operating range, is shown in Sketch 5.2.

The mass flow ratio delivered by the ejector increases as the steam supply pressure, p_1 , is increased or the pressure difference across the ejector is decreased. A maximum mass flow ratio is reached when the parameter, N_1 , equals zero. The maximum mass flow ratio available may be limited by the onset of

cavitation. High secondary mass flow rates lead to high liquid velocities with an increased risk of cavitation. A typical cavitation-induced limit is shown in Sketch 5.2. The design method given in Section 5.3.2 indicates when cavitation may be a problem. Further discussion of cavitation can be found in Section 8.2.3.



Sketch 5.2 Typical operating curve for a steam/liquid ejector

The mass flow ratio decreases as the parameter, N_1 , is increased. A point may be reached at which the secondary flow stops, while the steam flow continues to pass through the ejector. This point is often called the 'first shut-off', or 'breakpoint'. Further decreases in p_1 or increases in $(p_5 - p_0)$ may eventually reduce the outlet flow from the ejector to zero with the primary flow leaving via the secondary inlet so that there is effectively a negative secondary flow. This 'second shut-off' point is an important design parameter and is required so that the maximum possible pressure in the ejector due to a line blockage or valve failure can be assessed and components rated accordingly.

The maximum pressure ratio that can be attained may be limited by incomplete condensation of the steam flow or by boiling of the liquid flow. Collectively, these phenomena are often called vapour binding and are discussed more fully in Section 8.3.

5.3.2 Design method : steam/liquid ejectors

Stage 1 - Determine the constraints on the design

The constraints may relate to the required performance of the ejector or may be dimensional, restricting the size of the ejector. For most applications the main design constraint falls into one of the following

categories:

- (a) the secondary mass flowrate is specified (*e.g.* tank emptying duties),
- (b) the mass flow ratio is specified (*e.g.* liquid heating duties),
- (c) the pressure difference, $(p_5 - p_0)$, across the ejector is specified (*e.g.* liquid pumping duties).

From these constraints it should be possible to estimate a value for the pressure ratio N_1 or the mass flow ratio, r_m . If a range of values for N_1 or r_m is required, then the ejector should be designed to operate at the middle of that range. It is unusual for both N_1 and r_m to be constrained and that case is not considered here.

Calculate the density ratio, $r_c = \rho'_1 / \rho''$, of the supply steam and the secondary flow. The density of the steam may be obtained from steam tables or, approximately, from the ideal gas relationship $\rho'_1 = p_1 / R' T_1$, where $R' = 461.5 \text{ J/kg K}$.

Stage 2 - Obtain values for the area ratios A_R and A_T

Usually, the required ejector performance can be achieved with several different combinations of A_R and A_T . However, only one combination leads to a design with optimum performance. This design method assumes that optimum performance corresponds to the design that achieves the highest mass flow ratio (*i.e.* minimum steam mass flow rate) for a given pressure ratio, N_1 . Optimum combinations of A_R and A_T and corresponding values of r_m have been calculated using the theory in Section B2. The results are presented in Figures 1, 2 and 3 for several values of the density ratio, r_c . Results for values of r_c other than those in the Figures can be obtained by direct interpolation.

Proceed as follows, according to the design constraints.

- (a) If the pressure ratio, N_1 , is defined as one of the design requirements, find the optimum values of A_R and A_T from Figures 1 and 2. Read off the corresponding value for the mass flow ratio, r_m , from Figure 3.
- (b) If the mass flow ratio, r_m , is defined as one of the design requirements, then read off the corresponding value for the pressure ratio, N_1 , from Figure 3. For this value of N_1 read off the optimum area ratios, A_R and A_T , from Figures 1 and 2.

Stage 3 - Calculate the remaining unknown design parameters

If either the primary or secondary mass flow rate is unknown, calculate the unknown quantity using the known or estimated value of the mass flow ratio, r_m , and the value of the other mass flowrate.

Find any unknown pressures from the design requirements and the design value of N_1 .

Calculate the area of the primary nozzle throat, A_{th} , from

$$A_{th} = \frac{\dot{m}' \sqrt{T_1 R'}}{0.67 p_1}.$$

Calculate the primary nozzle exit area from the calculated value of A_{th} and the known value of A_T .

Calculate the mixing chamber area from the calculated value of A_e and the known value of A_R .

Calculate the nozzle throat, nozzle exit and mixing chamber diameters from the calculated areas.

Stage 4 - Determine whether cavitation or vapour binding is likely to occur

If cavitation and vapour binding occur they will reduce the performance of an ejector designed by this design method. It is important to estimate whether the design is likely to be affected by either of these phenomena.

Cavitation is likely to occur if the secondary inlet supply pressure, p_0 , is below the limit given by

$$p_0 \leq p_{v0} + \frac{\dot{m}''^2(1 + K_s)}{2\rho''(A_m - A_e)^2},$$

where p_{v0} is the vapour pressure of the liquid at the secondary inlet temperature, T_0 , and typically $K_s \approx 0.1$.

Vapour binding may occur in the diffuser if the temperature T_4 approaches the value of the boiling temperature of the liquid at the pressure p_4 . An estimate of T_4 can be obtained from

$$T_4 = \frac{h_{fg} + c'_{pf}T_1 + c''_p r_m T_0}{c'_{pf} + r_m c''_p}$$

and an estimate of p_4 from

$$p_4 = p_5 - \frac{(\dot{m}' + \dot{m}'')^2(1 - K_d)}{2\rho''A_m^2},$$

where typically $K_d \approx 0.15$. The criteria given above are crude, but indicate when cavitation and/or vapour binding may be a problem.

Stage 5 - Assess the final design

The final design should be assessed to determine whether it is acceptable in terms of the design constraints, size limitations and cavitation/vapour binding performance. Should cavitation or vapour binding be indicated the design should be reassessed: new values of A_R and/or A_T may be chosen, leading to poorer on-design performance, or the design constraints may be relaxed. If required the off-design performance of the ejector can be investigated by using the performance prediction procedure given in Section 6.1. The design procedure should be repeated until a satisfactory design is obtained.

Stage 6 - Carry out the detailed design of the ejector

The detailed design of the shape and layout of the primary nozzle, secondary inlet, mixing chamber and diffuser should be performed following the guidelines given in Section 5.5. This will define the following parameters:

- secondary inlet geometry,
- primary nozzle length and shape,
- mixing chamber length and shape,
- diffuser length and shape.

5.4 Design of Steam/Gas Ejectors

The recommended design method is intended to give quick, approximate estimates of the geometry of an ejector required to achieve a given performance. The method is similar to that presented in a previous Data Item (Derivation 25) and uses data obtained on a wide range of single-stage steam/gas ejectors. In all cases, the secondary gas was air at ambient temperature (nominally 20°C). Correction charts are presented that permit estimates of performance with other gases or with gases at higher temperatures. These charts are based on the Heat Exchange Institute Standards (Derivation 19), which in turn are based on a set of measurements made in the 1950's using two commercially available ejectors (Derivations 7 and 8). The Design Method applies only to single stage ejectors and individual units in multi-stage ejectors. Further information on the design of multi-stage units is given in Section 7.1.

The data used in the Design Method were obtained mostly from Derivations 2, 4, 9, 11, 14, 17 and 35 and published manufacturers' data for steam/gas ejectors. All of the ejectors tested had the following common features:

- (a) a single, central, convergent-divergent primary nozzle,
- (b) a 'constant pressure' design of mixing chamber (see Section 5.5.3), with included half-angles, ϕ_1 , in the range $2^\circ \leq \phi_1 \leq 10^\circ$,
- (c) conical diffusers with included half-angles, ϕ_2 , in the range $4^\circ \leq \phi_2 \leq 8^\circ$.

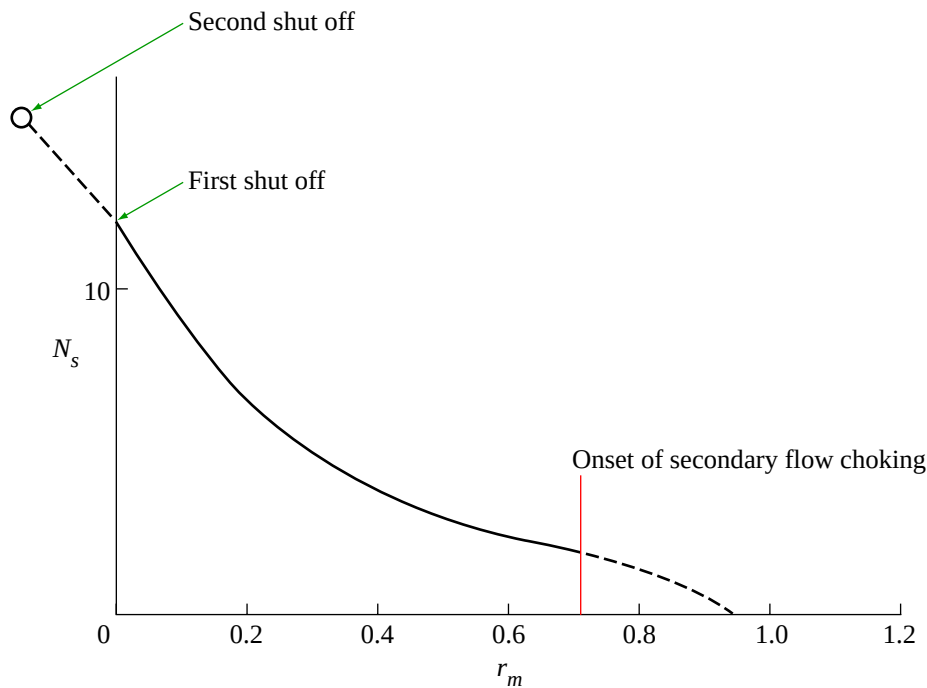
In most tests a saturated or slightly superheated steam supply was used with supply pressures typically in the range 5 to 20 bar abs. The design charts correspond to optimum operating conditions for these ejectors. In most cases, the optimum operating conditions were taken as the operating point at which the lowest value of secondary inlet pressure p_0 (highest vacuum) was measured for a given mass flow ratio, r_m . Typically, this point was determined by first adjusting the area ratios, A_R and A_T , to obtain minimum p_0 for given r_m then, in the resulting design, adjusting the primary nozzle position to improve further the performance of the ejector.

Typical operating characteristics for steam/gas ejectors are discussed qualitatively in Section 5.4.1. The design method is presented as a step by step procedure in Section 5.4.2.

5.4.1 Typical operating characteristics for steam/gas ejectors

The performance of steam/gas ejectors is usually measured in terms of the secondary pressure ratio, N_s , for a fixed steam supply pressure, p_1 . A typical operating curve for a steam/gas ejector at a fixed steam supply pressure is shown in Sketch 5.3. The curve shows the relationship between the secondary pressure ratio, N_s , and mass flow ratio, r_m , over the whole operating range of the ejector.

The mass flow ratio delivered by the ejector increases as the secondary pressure ratio is decreased. The maximum mass flow ratio that can be delivered is limited by secondary flow choking. The secondary flow is most likely to choke downstream of the primary nozzle. A typical limit on r_m due to secondary flow choking is shown in Sketch 5.3. A limiting value of the compression ratio is reached when the secondary mass flow rate is zero. This point is often called the 'first shut off', or 'breakpoint'. Further increases in N_s (by raising p_5 or lowering p_0) may eventually result in a negative mass flow ratio, with the primary flow leaving via the secondary inlet. This 'second shut off' point is an important design parameter and is required so that the maximum possible pressure in the ejector due to a line blockage or valve failure can be assessed and components rated accordingly.



Sketch 5.3 Typical operating curve for a steam/gas ejector

5.4.2 Design method : Steam/gas ejectors

Stage 1 - Determine the constraints on the design

The constraints may relate to the required performance of the ejector or may be dimensional, restricting the size of the ejector. For most applications, the main design constraint falls into one of the following categories :

- (a) the primary pressure ratio, $N_p = p_1/p_5$, is specified,
- (b) the secondary pressure ratio, or compression ratio, $N_s = p_5/p_0$, is specified,
- (c) the mass flow ratio, r_m , or the secondary mass flow rate, $\dot{m}''$, is specified.

To use the design method it is necessary to specify at least one of the other two design constraints. If a specific value is not available, it should normally be possible to specify a range of likely values. The design method should then be carried out for several selected values in this range.

Stage 2 - Determine factors C_1 and C_2 for gases other than air and temperatures above 20°C

The design method is based on data obtained for steam ejectors pumping air at 20°C. If other gases or gases at higher temperatures are to be handled, it is necessary to convert the secondary mass flowrate to an equivalent value for air at 20°C (the 20°C air-equivalent mass flow rate). This conversion is made using a correction factor, C_1 , for gases other than air and a correction factor, C_2 , for gas temperatures above 20°C. If the secondary gas is air at 20°C, $C_1 = C_2 = 1.0$. Otherwise proceed as follows:

- calculate the ratio between the molecular weight of the secondary gas and that of air. A value of 29 may be assumed for the molecular weight of air. Knowing this molecular weight ratio, the correction factor, C_1 , can be read directly from Figure 4,
- if the secondary mass flow rate is specified, divide this specified value by the product, $C_1 C_2$, to obtain the 20°C air-equivalent value. This value is then used in the remainder of the calculations,
- value for the correction factor, C_2 , can be read directly from Figure 5,
- if the secondary mass flow rate is to be calculated, multiply the value calculated in Stage 5 by the product, $C_1 C_2$.

Stage 3 - Estimate the unknown pressure ratio or mass flow ratio.

Figures 6 show the peak values of secondary pressure ratio, N_s , likely to be obtained from a well-designed ejector. These Figures are based on a compilation of more than 200 data points, together with typical operating curves published by manufacturers of steam ejectors. The data are presented in two forms. Figure 6a represents maximum achievable performance in current state of the art design whilst Figure 6b represents average performance through the data set. It is recommended that these Figures are used as follows:

- Figure 6a can be used to estimate a target performance for an ejector, to be arrived at by fine adjustments to the geometry of the ejector while it is being commissioned,
- Figure 6b should be used for conservative design or for a first attempt at a new design of ejector.

To use Figure 6a or Figure 6b, simply read the unknown parameter, N_p , N_s or r_m , directly from the appropriate curve.

Stage 4 - Estimate the optimum geometry for the ejector.

The optimum nozzle area ratio, A_T , for the given primary pressure ratio, N_p , can be read directly from Figure 7. This chart is theoretically based and represents the nozzle area ratio required in order that the static pressures in the steam and secondary flows are matched at the exit of the nozzle, i.e. $p_e' = p_e''$. The exit flow from the nozzle is then correctly expanded and the nozzle is operating as efficiently as possible. The curves on Figure 7 were derived as follows.

- Assume that $p_e'' \approx p_{te}'' \approx p_{t0} \approx p_0$.
- Assume that $p_{te}' \approx p_{t1}' \approx p_e' (1 + 0.158 M_e'^2)^{4.17}$, where M_e' is a direct function of A_T .
- Estimate the ratio $p_1/p_0 = N_p N_s$ for fixed r_m from Figure 6b.

- Set $p'_e = p''_e$ and so estimate the optimum value for A_T .

The optimum area ratio, A_R^* , for the given mass flow ratio can be estimated from Figure 8. Read a value for the parameter, A_R^*/N_p , and so calculate A_R^* . Figure 8 is based on data obtained from Derivations 4, 9, and 17; the shading indicates the range of these data. It is recommended that, for given r_m , an average of the highest and lowest values of A_R^*/N_p is taken.

Stage 5 - Calculate the remaining unknown design parameters.

If either the primary or secondary mass flow rate is unknown, calculate the missing quantity using the known or estimated value of the mass flow ratio.

Find any unknown pressures from the design requirements and the known values of N_p and N_s .

Calculate the throat area, A_{th} , of the primary nozzle from

$$A_{th} = \frac{\dot{m}' \sqrt{T_1 R'}}{0.67 p_1}$$

Calculate the primary nozzle exit area using the calculated value of A_{th} and the known value of A_T .

Calculate the mixing chamber area using the calculated value of A_{th} and the known value of A_R^* .

Calculate the nozzle throat, nozzle exit and mixing chamber diameters from the calculated areas.

Stage 6 - Assess the final design.

The final design should be assessed to determine whether it is acceptable in terms of the design constraints and any size limitations. It may be useful to investigate the off-design performance of the ejector using the performance prediction procedure given in Section 6.2.

The design procedure should be repeated until a satisfactory design is obtained.

Stage 7 - Carry out the detailed design of the ejector.

The detailed design (shape, layout) of the primary nozzle, secondary inlet, mixing chamber and diffuser should be performed following the guidelines given in Section 5.5. This will define the following parameters:

- mixing chamber length and shape,
- secondary inlet geometry,
- primary nozzle length and shape,
- diffuser length and shape.

5.5 Mechanical Design Considerations

Steam ejectors can be made from any material that can withstand the thermal stresses imposed by the steam flow. It is important that the shape and relationship of the component parts of the ejector do not vary during operation. The choice of material is usually a compromise between machinability, strength, cost, and resistance to wear and corrosion. Derivation 5 is a useful source of information on materials for use in steam ejectors.

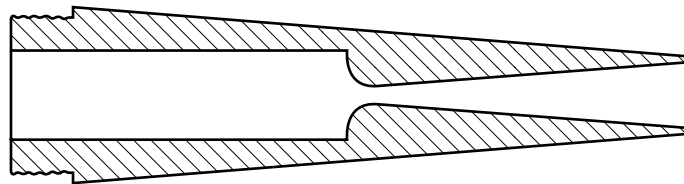
It is not practicable to specify precisely the optimum design of the component parts of a steam ejector. Different manufacturers favour different geometries and configurations and in some applications fabrication costs may outweigh any advantage to be gained in improved performance. The recommendations given here are based solely on the criterion of obtaining optimum performance and production costs are not considered. Where appropriate, steam/liquid and steam/gas ejectors are considered separately.

Optimum performance requires careful design and fabrication of the steam nozzle and also requires that all internal surfaces, including joints, are smooth.

Derivations 4, 9, 10, 11 and 17 are useful sources of information on the mechanical design of steam ejectors.

5.5.1 Primary nozzle

The primary nozzle in a steam ejector may be either convergent or convergent-divergent. For non-critical operation ($p_1/p_e \leq 1.84$) a simple convergent nozzle suffices. However, most ejector nozzles are operated under critical conditions and highest efficiency is achieved if a convergent-divergent nozzle is used. A typical design is shown in Sketch 5.4. The steam is accelerated in the short convergent section, reaching a Mach number of 1.0 just downstream of the throat of the nozzle. The steam expands and accelerates in the divergent section and leaves the nozzle as a supersonic jet. If the divergent portion of the nozzle is omitted this expansion still occurs but will be accompanied by loss-inducing compression shocks that reduce the efficiency of the ejector.



Sketch 5.4 Typical design of convergent-divergent primary nozzle for a steam ejector

Optimum values for the diameter of the nozzle throat, d_{th} , and nozzle exit, d_e , should be determined from the design methods. For both convergent and convergent-divergent nozzles, it is recommended that the converging portion of the nozzle has a circular or elliptical profile with a radius or minor axis of at least $0.3d_{th}$. If a simpler but less efficient converging cone section is used, the included angle of the cone should be about 24° . A long convergent section leads to increased friction losses, and hence lower C_D , without necessarily improving the smoothness of the flow at the throat.

The divergent portion of the nozzle is usually conical and an included angle of about 10° is recommended. For a given area ratio, A_e/A_{th} , smaller angles lead to a longer divergent section and hence higher friction losses while larger angles lead to a danger of flow separation at low pressure ratios. The throat joining the conical and divergent portions of the nozzle should be as short as possible, and should provide a smooth transition, with no discontinuity of slope or curvature.

For both convergent and convergent-divergent nozzles, the nozzle exit should have a sharp lip, as near to a feather edge as is practicable. The outer surface of the nozzle should be smooth and either parallel sided or slightly converging. These two factors together result in a narrow wake from the nozzle lip and hence increased mixing between the primary and secondary streams. All internal surfaces of the nozzle should be of high quality to reduce friction losses, particularly in the divergent portion.

The steam supply line to the nozzle should be sized so as to keep friction losses within reasonable limits; it is recommended that the steam velocity in the supply line is below 50 m/s. However, the steam supply line should not be so large that it restricts the secondary flow into the ejector.

Further information on the design of steam nozzles may be found in Derivations [9](#), [20](#), [27](#) and [52](#).

5.5.2 Secondary inlet and mixing chamber entry

In a single nozzle design, the secondary flow enters the mixing chamber via the annular gap between the primary nozzle and the body of the ejector. It is important that this flow passage is smooth, with no sharp constrictions or expansions.

For steam/liquid ejectors, a bell mouth inlet to this flow passage is recommended, with the outer surface of the primary nozzle shaped to give a constantly converging inlet passage. A conical inlet is simpler to machine but entry losses are higher. An included angle between 20° and 40° is recommended for conical inlets. For more detailed recommendations, see Derivation [9](#). Any joints in the secondary inlet should be smoothed to reduce friction losses and minimise the risk of cavitation and the inlet should be as short as possible to minimise friction losses.

Similar designs of secondary inlet can be used for steam/gas ejectors, provided the secondary flow is ducted into a plenum chamber upstream of the ejector. This configuration may not be feasible if space is limited but it has the advantage of ensuring that the supply pressure of the secondary stream is stable, which helps ensure reliable operation of the ejector. As a rough guideline, the minimum dimension of the plenum chamber should be at least ten times the diameter of the entrance to the mixing chamber.

A check should be made that the velocity of the secondary stream is within reasonable limits throughout the inlet passage. Maximum velocities between 10 and 20 m/s are recommended for steam/liquid ejectors to minimise wear and friction losses. Values below 100 m/s are recommended for steam/gas ejectors to minimise friction losses and avoid losses associated with compression shocks. For steam/liquid ejectors, the cavitation performance of the inlet should be checked as described in the design method. It is recommended that the minimum clearance between the primary nozzle and the body of the ejector is at least 1 to 2 mm.

5.5.3 Mixing chamber

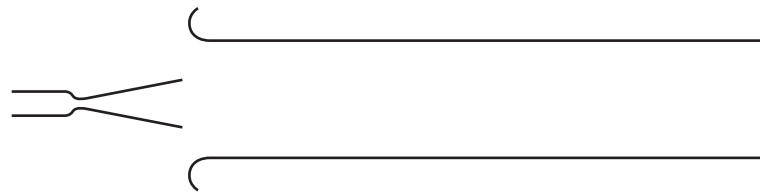
Two designs of mixing chamber are widely used : a parallel sided circular cylinder and a converging cone combined with a short parallel section.

Theoretical analyses of the flow in the mixing chamber usually assume that mixing occurs under either constant area or constant pressure conditions. This nomenclature is also used for the mixing chamber designs shown in Sketch [5.5](#), although the conical geometry is only a crude approximation to that required for constant pressure mixing.

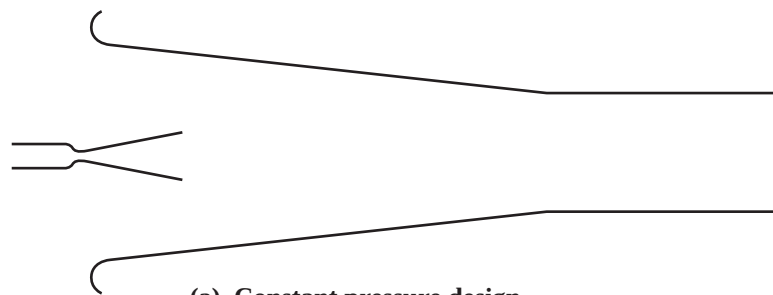
There is no reason to suppose that either design is more efficient. Steam/gas ejectors are usually equipped with constant pressure mixing chambers and steam/liquid ejectors with constant area chambers.

The function of the mixing chamber is to mix the steam and the secondary flow. In a well designed ejector momentum and energy transfer between the steam and secondary flow should be complete before the combined flow enters the diffuser. In a steam/liquid ejector this implies that the steam is completely condensed by the end of the mixing chamber. If the mixing chamber is too short, momentum and energy exchange will continue into the diffuser, resulting in an off-design performance from the diffuser with

higher friction losses and an increased risk of flow separation. Conversely, if the mixing chamber is excessively long, increased friction losses will offset any performance gains from the improved mixing between the primary and secondary streams.



(a) Constant area design



(a) Constant pressure design

Sketch 5.5 Mixing chambers for steam ejectors

The optimum length for a constant area mixing chamber is typically in the range 5 to $10D$. It is more difficult to give precise recommendations for ejectors with constant pressure designs of mixing chamber since the length available for mixing depends on the axial location of the primary nozzle in the mixing chamber. A tentative recommendation is that the mixing chamber should be sized so that the distance from the nozzle exit to the start of the diffuser is in the range 5 to $10D$. More specific recommendations are given in Derivations 4, 9 and 17.

For the 'constant pressure' design of mixing chamber, the included half-angle of the conical section, ϕ_1 , should be in the range of 2° to 10° . Some more specific recommendations are given in Derivations 4, 9, 17 and 34. It may be found advantageous to use a double taper cone in order to give a smoother transition for the flow as it enters the parallel section of the mixing chamber. For all designs, the length of the parallel section should be adjusted so that the total length for mixing is in the range recommended above. Typically, this section has a length of 2 to $4D$.

These recommendations only apply to ejectors with single, central nozzles. Multi-nozzle ejectors and annular nozzle ejectors should have shorter mixing chambers, since mixing occurs over a shorter distance in these configurations and any excess length results in increased friction losses. Further information on these configurations is given in Sections 7.2 and 7.3.

The optimum mixing chamber length is also related to the geometry of the diffuser. If the diffuser angle is larger than recommended, a longer mixing chamber should be used to produce a more uniform velocity profile at the diffuser inlet and hence help prevent flow separation. If the diffuser angle is smaller than recommended, a shorter mixing chamber may be used since some mixing within the diffuser may then be tolerated.

The mixing chamber walls should be as smooth as economically feasible since velocities and turbulence levels in the mixing chamber are high.

5.5.4 Primary nozzle position

The optimum position of the primary nozzle depends on the geometry chosen for the mixing chamber and on the application of the ejector. Early designs of steam ejectors often used movable (spindle or spear) nozzles. The nozzle could be moved to assist start up and to fine tune the performance of the ejector. However, such a nozzle removes the main advantage of a steam ejector which is its lack of movable parts. Also, it is difficult to devise a mechanism to hold a movable nozzle rigidly in position and any flexure results in variable performance and an increased risk of fatigue failure.

Modern steam ejectors normally use fixed primary nozzles, the optimum position being determined by experiment before the ejector is commissioned. Often, provision for slight adjustments is made by using shims or packing. Typically, movements as small as 1 mm can be sufficient to cause noticeable changes in the performance of even a large steam ejector.

The nozzle should be placed centrally on the axis of the mixing chamber in the case of ejectors with a single primary nozzle. For ejectors with a constant area design of mixing chamber it is recommended, as a first attempt, that the nozzle exit is placed 0.5 to $1.0D$ upstream of the start of the mixing chamber. It is difficult to give precise recommendations for the optimum nozzle position in ejectors with a constant pressure design of mixing chamber. Several correlations are given in the literature (*e.g.* Derivations 17 and 18), but all are of limited application since they were obtained on only one geometry. That there is an optimum nozzle position is best seen by considering what happens when the nozzle is moved from its optimum position. Retracting the nozzle increases the area available for the secondary flow and so increases the secondary mass flow rate at the expense of the discharge pressure, p_5 . The secondary flow may in some cases separate from the walls of the mixing chamber, resulting in high losses and unstable operation of the ejector. Moving the nozzle into the mixing chamber reduces the secondary mass flow rate and increases the discharge pressure, p_5 . The area available for the secondary flow may be so reduced that the secondary flow stops. For steam/gas ejectors it should be noted that the optimum position of the nozzle may depend on the composition of the gas being pumped, dense and/or viscous gases restricting the expansion of the primary jet and damping turbulence in the mixing region between the primary jet and secondary flow.

5.5.5 Contraction or diffuser

This component may be required to produce a specific exit pressure or velocity, or simply to connect an ejector to a downstream duct of different diameter.

The efficiency of a contraction is usually high, provided the internal wall is smooth, the contraction angle is not extreme and no turning vanes are fitted.

More care must be taken in the design of a diffuser. Flow separation and hence high losses may occur if the divergence angle of the diffuser is too high or if the velocity profile at the exit of the mixing chamber is highly non-uniform. If the mixing chamber is of optimum length, it is recommended that the included half-angle of the diffuser (ϕ_2) is in the range 3 to 4° and in no case greater than 7° (Derivations 9 and 22). For a short mixing chamber, producing highly non-uniform flow, a smaller included angle should be used. It is recommended that the area ratio of the diffuser, A_5/A_4 , does not exceed a value of 5.