

Article

## Thermodynamic Modelling of an Ejector with Compressible Flow by a One-Dimensional Approach

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**Abstract:** The purpose of this study is the dimensioning of the cylindrical mixing chamber of a compressible fluid ejector used in particular in sugar refineries for degraded vapor re-compression at the calandria exit, during the evaporation phase. The method used, known as the “integral” or “thermodynamic model”, is based on the model of the one-dimensional isentropic flow of perfect gases with the addition of a model of losses. Characteristic curves and envelope curves are plotted. The latter are an interesting tool from which the characteristic dimensions of the ejector can be rapidly obtained for preliminary dimensioning (for an initial contact with a customer for example). These ejectors, which were specifically designed for the process rather than selected from a catalog of standard devices, will promote energy saving.

**Keywords:** thermodynamic model; static compression; ejector; 1D modelling; compressible fluid; design

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## Nomenclature

### Symbols

$C_p$	Specific heat at constant pressure, J/(kg.K)
$F_3$	Pressure loss coefficient in the diffuser
$H$	Total enthalpy, J
$I$	Impulse or dynalpy, J
$P$	Total pressure, Pa
$P_{t3}$	Total pressure at mixing chamber exit, Pa
$P_{r3}$	Total outlet pressure in diffuser exit, Pa
$p$	Static pressure, Pa
$q$	Mass flow, kg/s
$r$	Mayer constant, J/(kg.K)
$S_{col}$	Throat section of driving flow nozzle, m <sup>2</sup>
$S_1$	Inlet mixing chamber section of driving flow, m <sup>2</sup>
$S_2$	Inlet mixing chamber section of induced flow, m <sup>2</sup>
$S_3$	Section of the cylindrical mixing chamber, m <sup>2</sup>
$T$	Total air temperature, K
$t$	Static temperature, K
$V$	Mean velocity of the various flows, m/s

### Subscripts

1	primary driving flow at the exit of the driving nozzle, mixing chamber inlet
2	secondary induced flow at the mixing chamber inlet
3	mixed flows at the mixing chamber outlet

### Greek Letters

$\rho$	Density, kg/m <sup>3</sup>
$\gamma$	Fluid $c_p/c_v$ ratio

## 1. Introduction

A jet ejector is a device which uses the kinetic energy of a driving fluid, injected under pressure by a convergent or convergent-divergent nozzle into a zone of lower pressure, to suck in and entrain a secondary fluid with low pressure (of a comparable or different nature) and to compress the mixed flow thus obtained to the desired intermediate pressure [1].

Some specialized companies produce tailor-made devices for the food processing industry (distillation plants, sugar refineries), for the desalination of sea water, and for oil. Manufacturers have more or less detailed catalogs of standard devices. In addition to basic theoretical methods, they possess the requisite know-how and experience to obtain the desired performance.

In this study, we present a method to define the ejectors so as to enhance their adaptation to the process concerned and thus to achieve increased energy saving. The application considered here is the sugar refinery process. We will focus more particularly on compressible fluid ejectors with a cylindrical mixing chamber and we will present their dimensioning by the method known as the integral method.

## 2. Brief Literature Survey

The literature on ejectors is extensive. Eames [2], for example, proposed a new formulation of the equation of momentum applied to the modeling of a supersonic jet ejector pump. Eames [3] also conducted an experimental and analytical study of a steam ejector refrigeration machine. Fan *et al.* [4] proposed a compressible fluid CFD model of a jet in an ejector pump. Rogdakis and Alexis [5] carried out a 1D analytical study to characterize the performance of an ejector system used for air conditioning, and in a complementary study [6] they investigated an ejector design using a two-phase mixture  $\text{NH}_3\text{-H}_2\text{O}$  based on the theory of Keenan *et al.*

A major contribution was made by Gilbert and Hill [7] in their 2D characterization of a variable geometry jet exhaust in terms of pressure and flow velocity distribution. Deberne *et al.* [8] proposed a numerical description of a 1D steam injector comprising a supersonic layer in the jet. Sobieski [9] developed a CFD analysis based on the compressible fluid description of a 2D air flow, but noted that the simulation results were not fully validated by experimental results (due to the lack of appropriate models of turbulence and mixing in the code used).

Finally, the thesis by Watanawanavet [10] mainly concerns a CFD analysis of energy conversion efficiency in an ejector as a function of its geometry. However, the literature remains sparse for the application considered in this article. This broad study is therefore devoted to sugar refineries and based on a craft approach validated by several hundred cases.

## 3. Context

The integral method used in this study is based on the model of a one-dimensional isentropic flow of perfect gases. It considers the characteristics of the driving and induced fluids at the mixing chamber inlet and the characteristics of the mixture at the mixing chamber exit, while including initial assumptions [11,12].

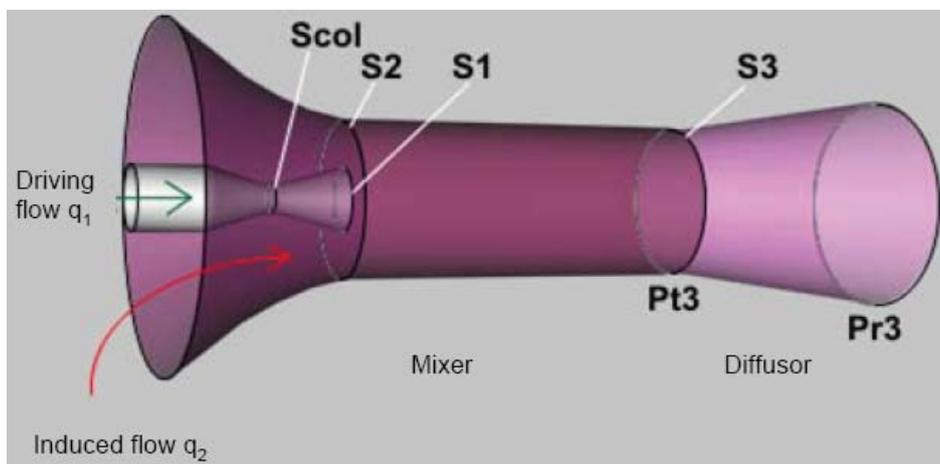
When ordering an ejector, the customer specifies the total pressures  $P_1$  of the motive fluid and  $P_2$  of the induced fluid, the associated total temperatures  $T_1$ ,  $T_2$ , the entrainment ratio  $q_2/q_1$  (driving flow/induced flow) or the total outlet pressure  $P_{r3}$  (Figure 1). The designer defines the Mach number  $M_2$  of the induced flow to obtain the best performance.

For certain applications, the customer requires a given outlet pressure  $P_{r3}$ , in which case the designer calculates the entrainment ratio  $q_2/q_1$  or *vice versa*. In this study, we have imposed the entrainment ratio  $q_2/q_1$ .

We will therefore have to calculate the outlet pressure  $P_{r3}$  to which a model of pressure loss  $F_3$  (related to the design and scale of the device considered) will be attributed by the designer based on his experience.

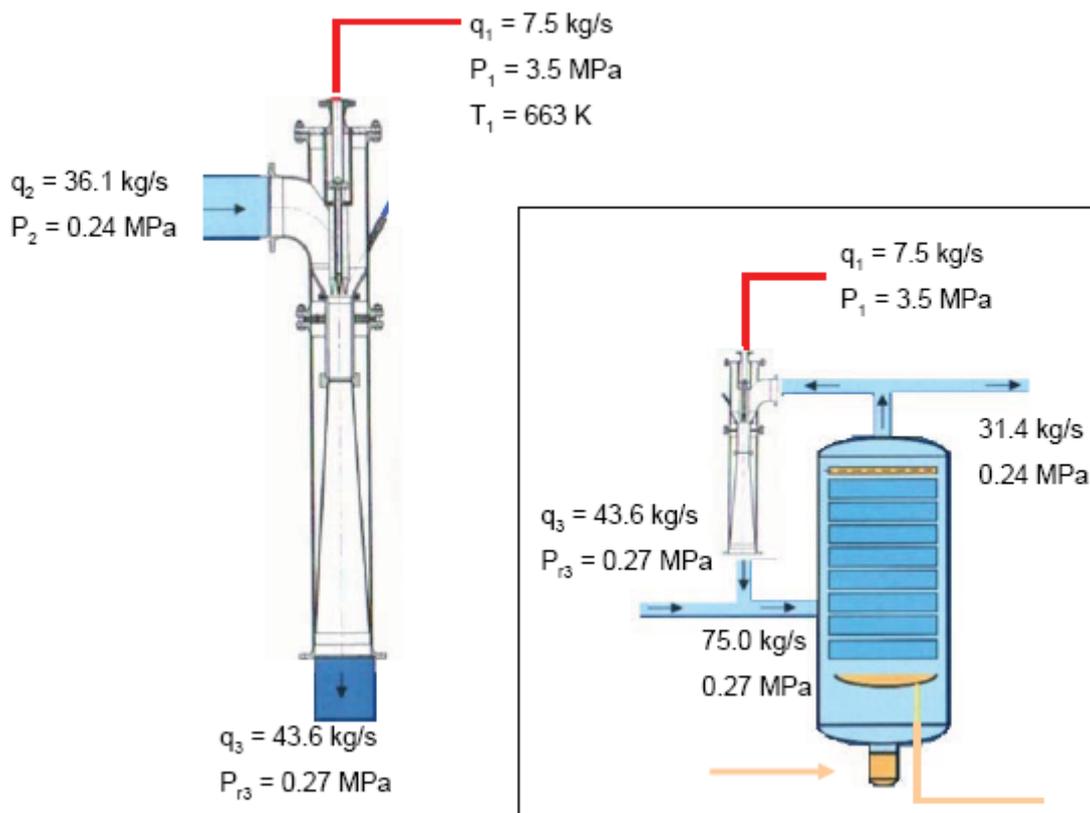
The calculation of  $P_{r3}$  will then enable the dimensions of the mixing chamber to be determined by calculating its section  $S_3$  as well as the other dimensions of the ejector, which are beyond the scope of this study. The calculated section  $S_3$  of the mixing chamber is a function of the ejector operating conditions, in particular of  $P_1$ ,  $T_1$ ,  $P_2$ ,  $T_2$ , but also of  $q_1$ ,  $q_2$  and  $P_{r3}$ .

**Figure 1.** Diagram of the studied ejector specifying the notations at the various points.



As a concrete example, Figure 2 shows a tailor-made steam-ejector designed for the sugar refinery in Wanze (Belgium) with the following characteristics.

**Figure 2.** Example of a tailor-made ejector.



**Driving stream**

Total pressure:  $P_1 = 3.5$  MPa

Total temperature:  $T_1 = 663$  K (390°C)

Driving flow:  $q_1 = 7.5$  kg/s (27 ton/hour)

**Induced stream**

Total pressure:  $P_2 = 0.24$  MPa

Induced flow:  $q_2 = 36.1$  kg/s (130 ton/hour)

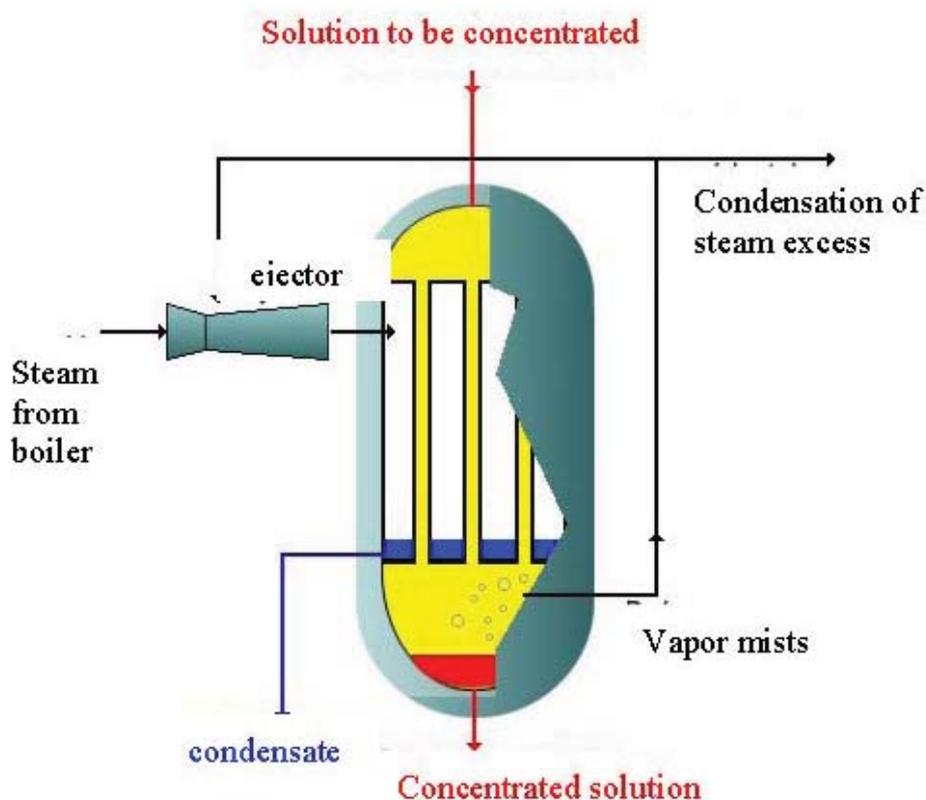
Entrainment ratio (driving flow/induced flow):  $q_2/q_1 = 4.814$

**Mix**

Total outlet pressure:  $P_{r3} = 0.27$  MPa (absolutes)

These devices are thus adapted to the specific conditions of each company. In sugar refineries, the ejectors are used during the evaporation stage (Figure 3) to increase the pressure level of the vapor mists produced at the exit of the calandrias, thus reducing the energy requirement. It is static compression.

**Figure 3.** General diagram of an evaporator with use of an ejector.



## 2. Theoretical Thermodynamic Model

The integral calculation method is based on the laws of dynalpy conservation ( $I = pS + qV$ ) in the mixing chamber, of mass and energy conservation between the streams inlet and the mixed flow outlet.

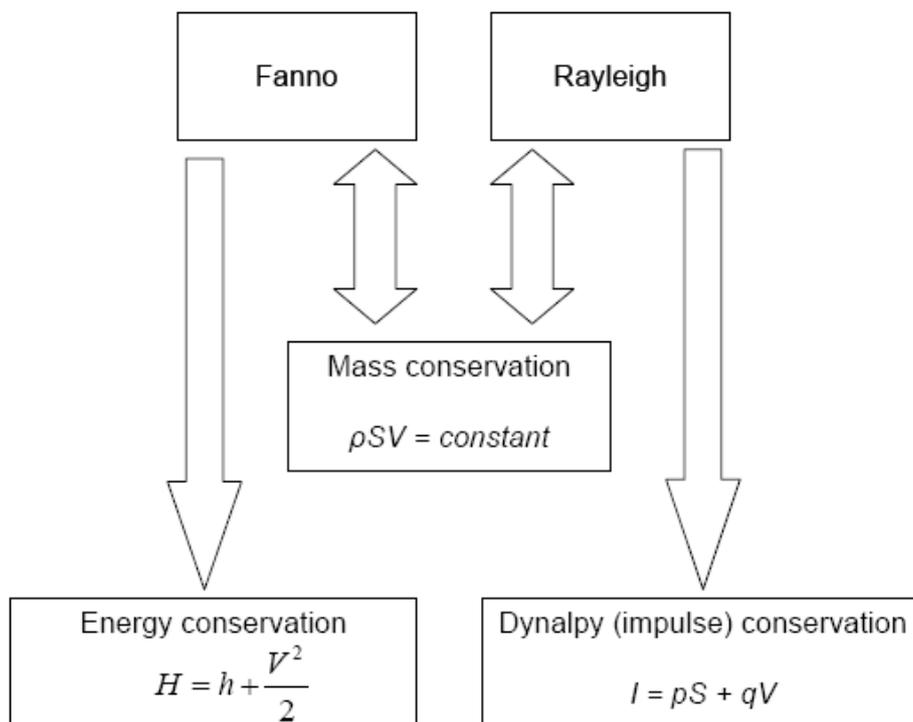
The mixture is assumed to be complete on the outlet side of the mixing chamber and a total pressure loss corresponding to that of the mixing chamber and the diffuser is taken into account. This leads to a

model of pressure loss ( $F_3$ ) which is dependent on the design and scale of the device considered. This method, which involves implementing Fanno and Rayleigh curves [13], is the most direct but requires experience with various devices to determine the pressure loss model  $F_3$ .

2.1. Basic Equations Used

The basic equations used to determine the computation formulae are presented on Figure 4 and are detailed hereafter. They are valid in compressible and incompressible fluids.

Figure 4. Basic equations used.



Subscripts 1 and 2 are used respectively for the driving and the induced fluids in the mixing chamber inlet, and subscript 3 for the mixture on the outlet side of the mixing chamber. Total quantities are in capital letters and static quantities in lower case.

2.1.1. Continuity Equation:

$$q_3 = q_1 + q_2 \tag{1}$$

with  $q = \rho SV$

2.1.2. Momentum Equation (or dymalpy conservation):

$$p_1 S_1 + q_1 V_1 + p_2 S_2 + q_2 V_2 = p_3 S_3 + q_3 V_3 = I \tag{2}$$

### 2.1.3. Energy Equation

$$q_1 H_1 + q_2 H_2 = q_3 H_3 \quad (3)$$

The fluids are now assumed to be perfect gases:

$$q_1 c_{p1} T_1 + q_2 c_{p2} T_2 = q_3 c_{p3} T_3 \quad (4)$$

$$q_1 \left( c_{p1} t_1 + \frac{V_1^2}{2} \right) + q_2 \left( c_{p2} t_2 + \frac{V_2^2}{2} \right) = q_3 \left( c_{p3} t_3 + \frac{V_3^2}{2} \right) \quad (5)$$

### 2.2. Calculation Assumptions

The fluids behave as perfect gases. However the driving can take place only for viscous fluids. The flow is one-dimensional and isentropic to initialize the calculation. It is then supplemented by a model of pressure loss based on experience. The model of pressure loss is deduced from the following relation where  $P_{t3}$  is the total pressure at the mixing chamber exit.  $P_{r3}$  is the total pressure at the diffuser exit:

$$P_{r3} = P_{t3} - F_3 (P_{t3} - p_3) \quad (6)$$

At the mixing chamber inlet, the static pressures  $p_1$  and  $p_2$  are equal. The mixture is complete at the mixing chamber outlet [2].

From the parameters fixed by the customer, *i.e.*, pressure, flow and temperature,  $P_1$ ,  $q_1$ ,  $T_1$ , for the driving fluid and  $P_2$ ,  $q_2$ ,  $T_2$ , for the induced fluid, as well as a loss ratio  $F_3$  fixed by the designer (constant value for a given ejector), we established computation formulae so as to obtain the following parameters in this order:

$t_2$ ,  $p_2$ ,  $V_2$ ,  $\rho_2$ ,  $S_2$  for the induced fluid

$T_1$ ,  $\rho_1$ ,  $V_1$ ,  $S_1$  for the driving fluid

$S_3$ ,  $T_3$ ,  $V_3$ ,  $\rho_3$ ,  $P_{t3}$ ,  $P_{r3}$  for the mixture.

All these formulae were computed then a validation of the computed code was conducted using the steam ejector described below.

### 2.3. Validation of the Computer Code

Measurements were carried out on a 12 ton/hour driving flow device, commissioned in the marketing year 2005/2006 (sugar refinery in Roye, France). This device had been defined for the following operating conditions (Table 1):

**Table 1.** Operating conditions defined for a 12 ton/hour device.

$P_1$ (MPa)	$T_1$ (K)	$P_2$ (MPa)	$T_2$ (K)	$q_2/q_1$
4.1	673	0.27	402.5	3.62

Based on the long-standing experience of the designer and co-author (C. Chacoux),  $F_3$  was imposed [see Equation (6)] and the Mach number chosen for the induced fluid was  $M_2 = 0.75$  (the choice of this value is reconsidered below).

### 2.3.1. Results of Calculations (Table 2)

**Table 2.** Results of calculation.

$P_{r3}$ (MPa)	$T_3$ (K)	$H_1$ (kJ/kg)	$H_2$ (kJ/kg)	$H_3$ (kJ/kg)	$S_3$ (cm <sup>2</sup> )
<b>0.34</b>	<b>454.8</b>	3 154	2 719	2 809	322

### 2.3.2. Results of Measurements

The total outlet pressure  $P_{r3}$  finally obtained was 0.32 MPa for 0.34 calculated, the observed entrainment ratio  $q_2/q_1$  being slightly different from that proposed (Table 3).

**Table 3.** Results of measurements for  $S_3 = 322$  cm<sup>2</sup>.

$P_1$ (MPa)	$T_1$ (K)	$P_2$ (MPa)	$T_2$ (K)	$q_2/q_1$
4.5	675.5	0.254	401	<b>3.58</b>
$P_{r3}$ (MPa)	$T_3$ (K)	$H_1$ (kJ/kg)	$H_2$ (kJ/kg)	$H_3$ (kJ/kg)
<b>0.31/0.32</b>	<b>454.5</b>	3 213	2 717	2 825

Calculation predicted a backpressure of  $P_{r3} - P_2 = 0.34 - 0.267 = 0.073$  MPa, whereas experimentation gave  $P_{r3} - P_2 = 0.32 - 0.254 = 0.066$  MPa. This slight difference can be explained by the fact that  $P_2$  is a little different in the two cases (2.67 MPa calculated, 2.54 MPa measured, *i.e.*, a discrepancy of 0.5%). Similarly, the driving pressure was 4.5 MPa instead of 4.1 previously considered during the design process, resulting in a drop in the driving rate. All these results are coherent and satisfactory.

## 3. Use of the Ejector in Conditions Different from the Nominal Point

The ejector is determined for a nominal operating point. However, the process conditions may change, in which case the user must be able to determine new values of the parameters (for instance the new driving flow for the same induced flow).

For this purpose, characteristic curves may be of assistance. If the range of operation is too wide, several ejectors must be considered in order to operate with optimum efficiency. To achieve this, an extension of the computed code is possible by imposing the ratio of the mixing chamber section to the throat section of the driving flow nozzle  $S_3/S_{col}$  as the reference criterion.

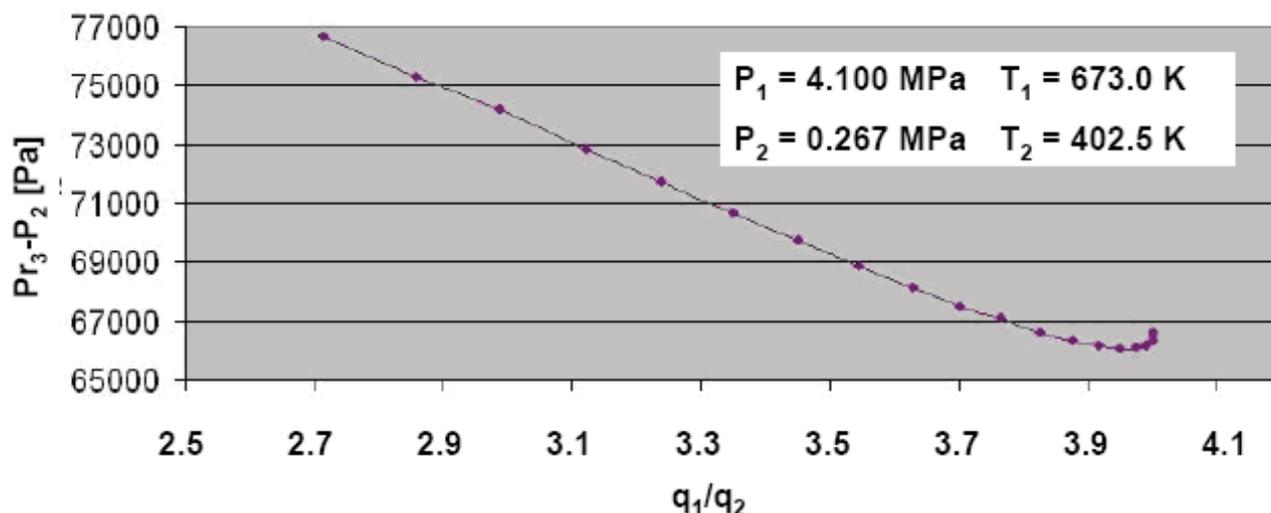
For  $S_{col}$  fixed, depending on the  $S_3$  section—called the origin section herein—The induced flow  $q_2$  is adjusted (for various values of the Mach number  $M_2$ ) until  $S_{3\text{ calcul}} \approx S_{3\text{ origin}}$  is obtained.

Using the various values of the induced flow  $q_2$ , the various values of the outlet pressure  $P_{r3}$  are calculated and the backpressure ( $P_{r3} - P_2$ ) is plotted against the driving rate  $q_2/q_1$ , the driving flow  $q_1$  and the induced flow total pressure  $P_2$  being fixed.

### 3.1. Plotting a Characteristic Curve

Figure 5 plots the characteristic curve of a given geometry ( $S_3/S_{col}$ ) related to the primary and secondary characteristics of the above steam ejector ( $q_1$  fixed).

**Figure 5.** Characteristic curve of the ejector ( $q_1$  fixed).



Note that the bent part of the curve on the right of the graph is not a true representation because of the transonic effects in this zone. With such a curve, if the discharge pressure imposed by the process changes, the user will be able to determine the new driving flow  $q_1$  required for the same induced flow  $q_2$ .

In addition, as mentioned above, for each point on this curve there is a corresponding Mach number  $M_2$ . The efficiency calculation at each point shows that the Mach number is maximum at the minimum point of the curve (see Section 4.1).

### 3.2. Plotting Several Characteristic Curves

The farther from the nominal (or design point or point of greatest efficiency) the point is, the more the ejector efficiency drops. Thus it is not interesting to widen the operating range too much. It is preferable to consider another ejector when the conditions  $P_{r_3} - P_2$  or  $q_2/q_1$  differ considerably. If the installation undergoes strong variations, several ejectors can be laid out in parallel, gauged for suitable flow ranges. A series of characteristic curves is thus constructed.

## 4. Envelope Curves: Geometry Definition for Optimum Efficiency

To quickly obtain the initial elements of a dimensioning for the first contact with a customer, it is advisable to know the envelope curves of a family of devices. Given a geometry  $S_3/S_{col}$  and the driving, induced and outlet pressure and temperature conditions, these curves can then be used to define the most efficient geometry of the device(s).

4.1. Definition of an Envelope Curve

The envelope curve is a curve (C) defined by a whole set of curves (C<sub>i</sub>), such that each curve is tangent in at least one point of (C) and that in any point of (C), a curve (C<sub>i</sub>) tangent with (C) exists. In fact in our study, curves (C<sub>i</sub>) are the characteristic curves.

The points of tangency of the envelope curves with the characteristic curves are the points of the characteristic curves located at the level of the minimum point. These points are the optimal operating points of the ejectors, *i.e.*, where the total efficiency is maximum.

The total effectiveness of an ejector results in the total efficiency which can be defined by the ratio between the compression energy recovered on total flow and the energy spent on the driving flow expansion in the nozzle. Its expression is:

$$\eta_g = \frac{q_3(\gamma_1 - 1)T_3\gamma_3r_3 \left[ 1 - \left( \frac{P_2}{P_3} \right)^{\frac{\gamma_3 - 1}{\gamma_3}} \right]}{q_1(\gamma_3 - 1)T_1\gamma_1r_1 \left[ 1 - \left( \frac{P_2}{P_1} \right)^{\frac{\gamma_1 - 1}{\gamma_1}} \right]} \tag{7}$$

Optimum Example of Operating Point on the Characteristic Curves

From the X-coordinates q<sub>2</sub>/q<sub>1</sub>, we calculated the ordinates P<sub>r3</sub> – P<sub>2</sub> of certain points around the minimum point of the characteristic curve presented on Figure 5. We also calculated the corresponding total efficiency η<sub>g</sub> for each point and drew up Table 4.

**Table 4.** Ejector global efficiency.

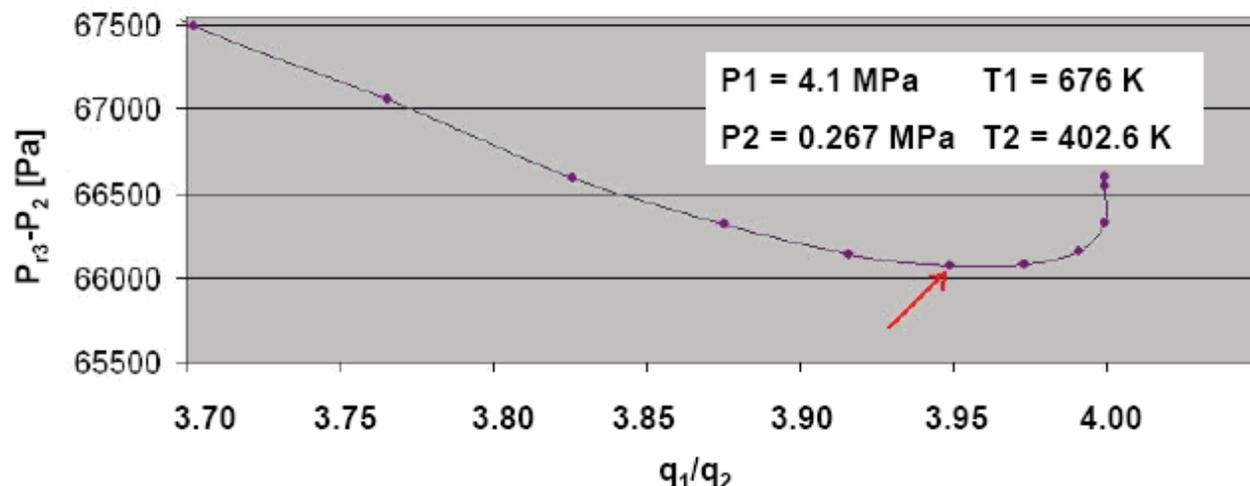
q <sub>2</sub> /q <sub>1</sub>	M <sub>2</sub>	P <sub>r3</sub> – P <sub>2</sub> (Pa)	η <sub>g</sub> (%)
2.715	0.43	76 662	34.41
2.988	0.49	74 198	35.62
3.239	0.55	71 738	36.58
3.875	0.79	66 313	38.72
<b>3.949</b>	<b>0.85</b>	<b>66 069</b>	<b>44.22</b>
3.974	0.88	66 086	39.15
3.991	0.91	66 162	39.25
4.000	0.94	66 325	39.21

It can be seen that this minimum point is the point of the coordinates q<sub>2</sub>/q<sub>1</sub> = 3.949 and P<sub>r3</sub> – P<sub>2</sub> = 66 069 Pa which corresponds to the best performance η<sub>g</sub> = 0.4422.

The calculation of the following points corresponding to higher driving rates is no longer valid because of the transonic effects not taken into account by this calculation: the losses in the diffuser can no longer be regarded as constant.

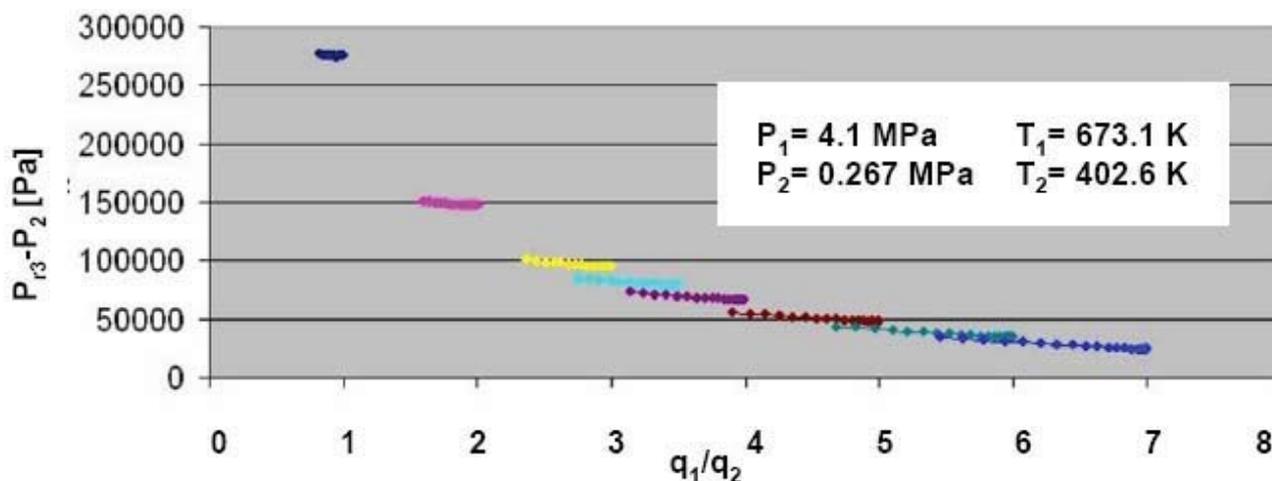
Figure 6 shows an expanded view of the characteristic curve plotted on Figure 5. The minimum point of the characteristic curve is indicated by an arrow.

**Figure 6.** Expanded view of the characteristic curve, showing the point of maximum efficiency ( $q_1$  and  $S_3$  fixed).



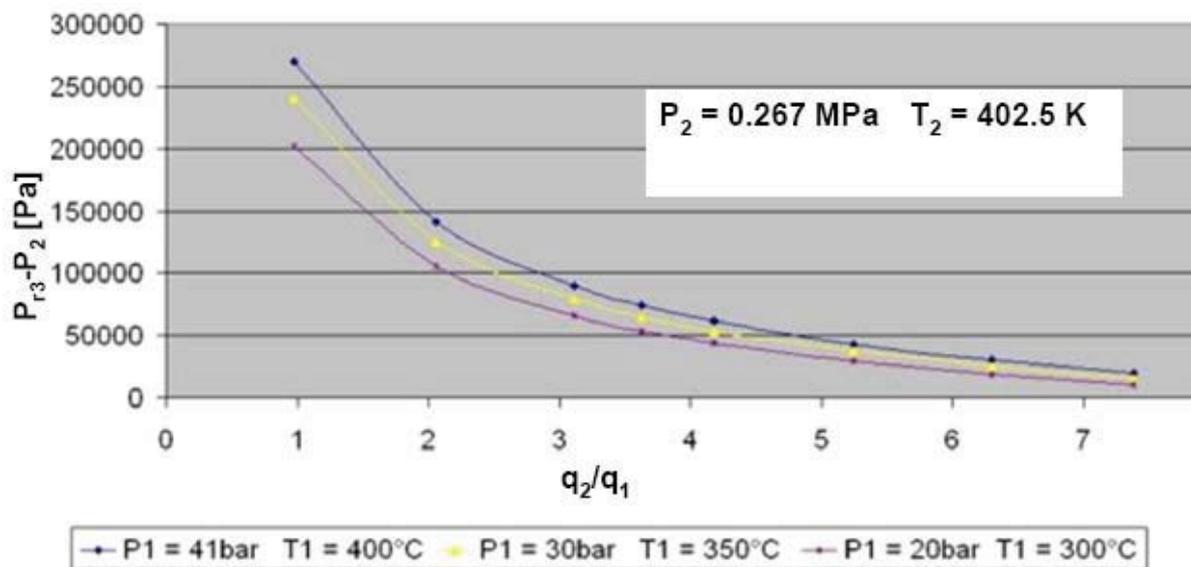
From the whole set of these optimum operating points (from each curve on Figure 7) the envelope curve valid for a single value of the total driving pressure  $P_1$  and total driving temperature  $T_1$  can be plotted.

**Figure 7.** Characteristic curves of ejectors of various sections ( $q_1$  fixed).



When the values of  $P_1$  and  $T_1$  are modified, the other parameters  $P_2$ ,  $T_2$  and  $q_1$  remaining unchanged, a new envelope curve is obtained (Figure 8).

**Figure 8.** Envelope curves for various values of  $P_1$  and  $T_1$  ( $q_1$  fixed).



4.2. Validation of the Envelope Curve

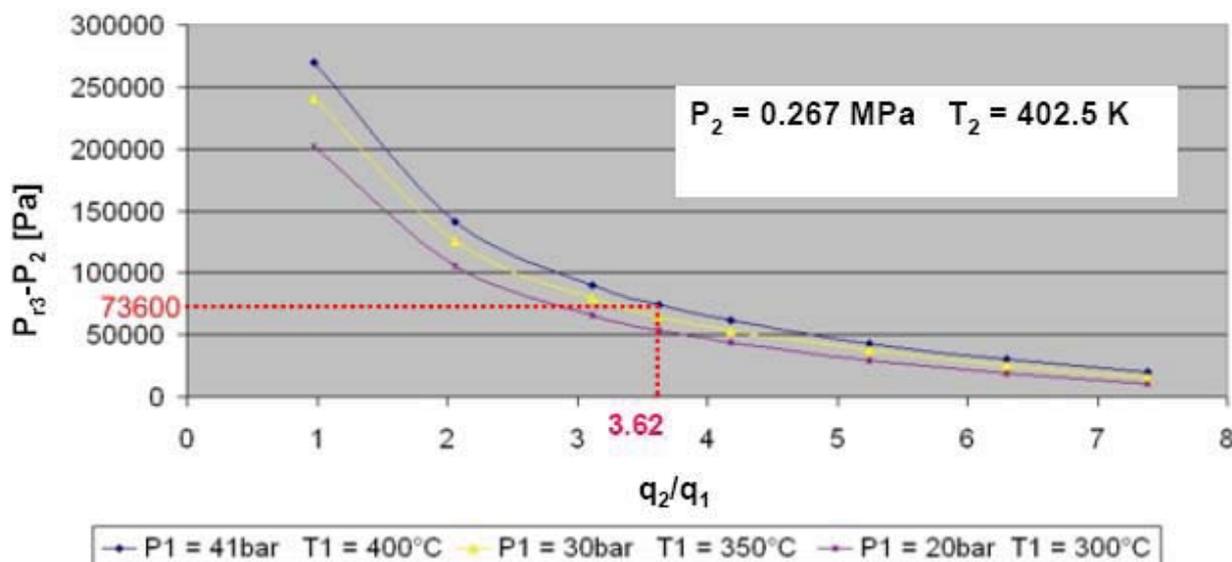
The previous example (Section 2.3) of an ejector defined for the following conditions is again taken (Table 5).

**Table 5.** Operating conditions defined for a 12 ton/hour device.

$P_1$ (MPa)	$T_1$ (K)	$P_2$ (MPa)	$T_2$ (K)	$q_2/q_1$
4.1	673	0.267	402.5	3.62

The matching envelope curve is the curve of characteristics  $P_1 = 4.1$  MPa and  $T_1 = 673$ K (400 °C). For  $q_2/q_1 = 3.62$  the following values can be read off the graph:  $P_{r3} - P_2 = 73\ 600$  Pa,  $P_2 = 0.267$  MPa, which gives  $P_{r3} = 0.3406$  MPa. This point can be seen on Figure 9.

**Figure 9.** Envelope curves ( $q_1$  fixed).



We recap below the results of measurements (Table 6).

**Table 6.** Results of measurement for a 12 ton/hour device.

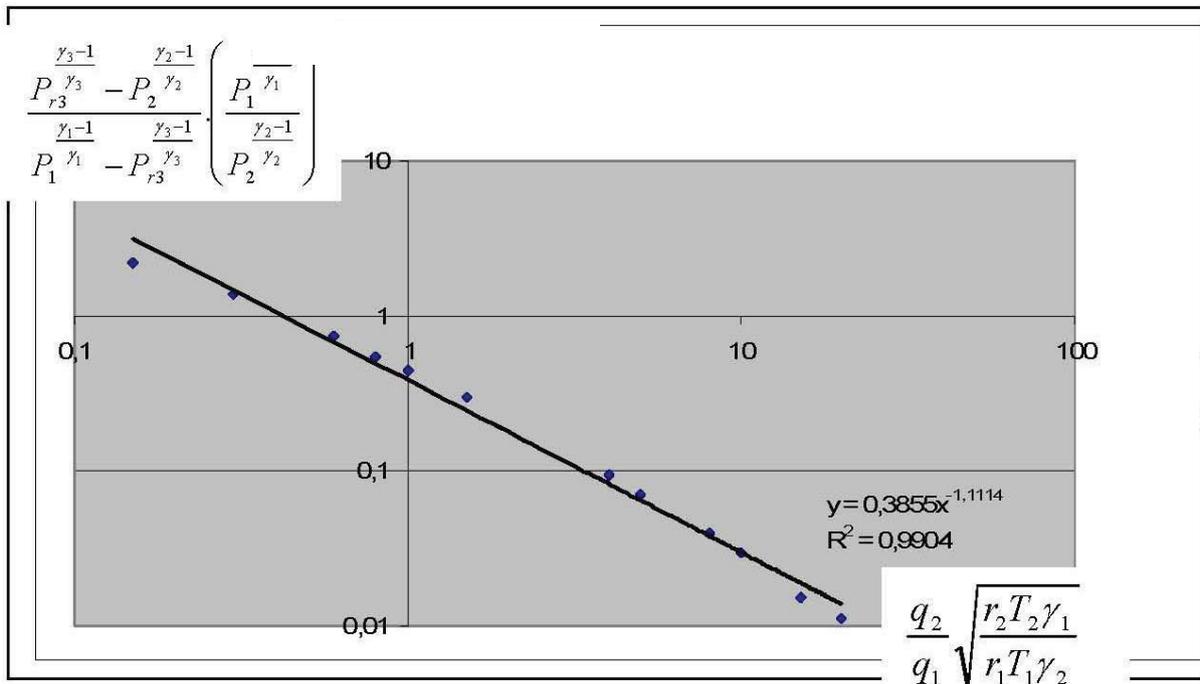
$P_1$ (MPa)	$T_1$ (K)	$P_2$ (MPa)	$T_2$ (K)	$P_3$ (MPa)
4.5	675.5	0.254	401	0.32

In light of these results, it can be considered that the use of the envelope curve during an initial customer contact provides a very good idea of the device to be dimensioned. These envelope curves depend, however, on many parameters. For other values of total pressure and temperature of the induced fluid  $P_2$  and  $T_2$ , as well as the driving fluid flow  $q_1$ , it will therefore be necessary to have new envelope curves. Likewise, it will be necessary to interpolate between these curves for values of  $P_1$  and  $T_1$  that are different from those plotted.

The situation is greatly improved by using reduced co-ordinates which allows the aggregation of these various curves into a single one (but always given  $P_2$ ,  $T_2$  and  $q_1$ ) as shown on the following example of Figure 10:

$$\frac{P_{r3}^{\frac{\gamma_3-1}{\gamma_1}} - P_2^{\frac{\gamma_2-1}{\gamma_1}}}{P_1^{\frac{\gamma_1-1}{\gamma_1}} - P_{r3}^{\frac{\gamma_3-1}{\gamma_1}}} \cdot \left( \frac{P_1^{\frac{\gamma_1-1}{\gamma_1}}}{P_2^{\frac{\gamma_2-1}{\gamma_1}}} \right) = f\left(\frac{q_2}{q_1} \sqrt{\frac{r_2 T_2 \gamma_1}{r_1 T_1 \gamma_2}}\right) \tag{8}$$

**Figure 10.** Condensed envelope curve (for  $T_2$ ,  $P_2$ ,  $q_1$  fixed).



## 5. Conclusions

This paper has presented:

1. A study of the dimensioning of compressible fluid ejectors with a cylindrical mixing chamber, used, amongst other applications, in sugar refineries during the evaporation stage. The computed code, worked out thanks to the integral method, was validated on more than three hundred steam ejectors with a discrepancy between calculation and measurements of less than 5%.
2. The thermodynamic model used is based on the model of the one-dimensional isentropic flow of perfect gases with the addition of a model of losses. It helps to make appropriate technological choices when designing new installations or during the rehabilitation of existing ones, by favouring energy economy.
3. This method could be exploited for the dimensioning of compressible fluid ejectors with a conical mixing chamber. The study would however be more complex because the driving of secondary flow in this type of mixing chamber is supersonic.

## Acknowledgements

In memory of our colleague and co-author, Claude Chacoux.

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