

Introductory Note

Note added February 1999. This theory refers to the fundamentals defining the main dimensions of the ejector. It requires the calculation (or the obtention by experimental procedures) of the boiler characteristics, a serious matter in itself. It also presupposes that a large number of details coming from a long experience are to be respected. It does not include the swirl of both of the steam jet and the gas intake. Finally, a still pending serious problem is that it assumes that the flow is steady, non-pulsating, a field open to future investigation. It is not a "kitchen recipe" guaranteeing good results without a good tuning up with measurements. However, the reader may try, provided that if success crowns his trial and error, the merit is to be credited to the theory. If not, the Author expects that the failure is not to be credited to the theory, but to the user.

THEORY OF THE LEMPOR EJECTOR AS APPLIED TO PRODUCE DRAUGHT IN STEAM LOCOMOTIVES

Summary

This is a revision of the Porta-Taladiz theory of 1957, incorporating the effect of friction in the mixing chamber and other corrections. The theory is based on the principle of conservation of energy as applied by Strahl in 1913, with due corrections for the case under consideration. It serves for the design of an optimum ejector leaving a minimum correction of the blast pipe to be carried out in practice as usual since Stephenson's times. It serves also to predict the behavior of a given ejector.

As a matter of course, the accuracy of either the calculated or the measured performance of the boiler conditions the optimization of the ejector design. Fortunately there seems to be a reasonable latitude in the dimensions around the optimum.

The Lempor ejector resembles the Le Maître and is thermodynamically equivalent to the Kylpor, the latter resembling the Kylchap. However, its internal losses to friction in the mixing device are smaller.

Successful experiments would show 100% improvement against the best known draughting device.

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Theory of the Lempor ejector

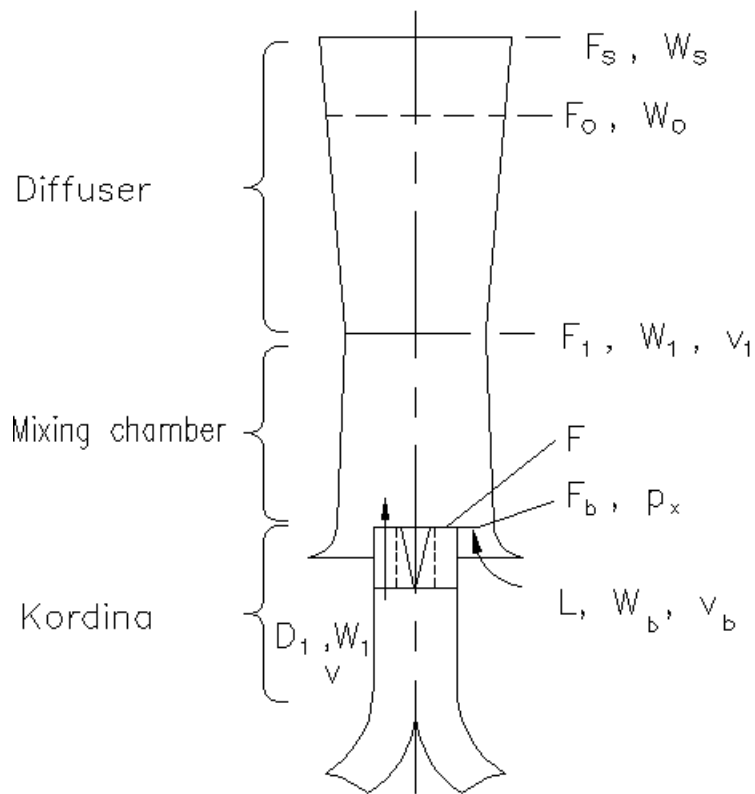


Fig 1 The Lempor system

The principal parts of the ejector are (Fig 1)

- A. The blast pipe or tuyère made up of four nozzles with converging-diverging sections so as to account for the supersonic flow prevailing during the exhaust beats.
- B. The mixing chamber of slightly converging section whose length is a compromise between the evening-out of exit velocity and available total height.
- C. The diffuser, having the maximum possible diverging angle, yet with good diffusing efficiency
- D. The Kordina, which is really an ejector in itself disposed so that each steam puff creates a vacuum in the other cylinder.

The interesting cross-sections are:

Symbol	Units	Description
F_s	$[m^2]$	The chimney exit area
F_1	$[m^2]$	Chimney throat area
F_b	$[m^2]$	Gas inlet area at the bottom of the mixing chamber
F	$[m^2]$	Steam tuyère area
F_0	$[m^2]$	Ideal diffuser outlet area

Let also be:

L	$[kg\ s^{-1}]$	Gas mass (rate) to be expelled
D	$[kg\ s^{-1}]$	Mass rate of steam flowing through the tuyère
W_0	$[m\ s^{-1}]$	Mixture velocity at ideal outlet
W_1	$[m\ s^{-1}]$	Mixture velocity at chimney throat
W_b	$[m\ s^{-1}]$	Gas velocity at gas inlet section of area F_b
W	$[m\ s^{-1}]$	Gas velocity at tuyère exit plane
p_x	$[N\ m^{-2}]$	Static pressure (absolute) at the tuyère exit plane
p_0	$[N\ m^{-2}]$	Atmospheric pressure
p_c	$[N\ m^{-2}]$	Smokebox pressure
$p_0 - p_c$	$[N\ m^{-2}]$	Draught
p_1	$[N\ m^{-2}]$	Pressure at chimney throat
v_1	$[m^3\ kg^{-1}]$	Specific volume of the steam-gas mixture
ξ	-	Coefficient expressing the pressure drop due to friction in the mixing chamber
ξ_b	-	Coefficient expressing the pressure drop at the gas entrance
λ	-	$\frac{1}{2} \left[1 + \left(\frac{F_1}{F_0} \right)^2 \right]$
F_0	$[m^2]$	Ideal diffuser outlet section = $\frac{1}{\sqrt{\frac{1-\eta}{F_1^2} + \frac{\eta}{F_s^2}}}$
η	-	Diffuser efficiency (e.g. 10)

Assuming a uniform velocity distribution at every cross-section, the fundamental equation of the ejector is obtained by applying the principle of the conservation of energy to the fluid streams:

Equation 1

$$(L+D)(p_0 - p_x)v_1 + \frac{1}{2}D(W - W_1)^2 + \frac{1}{2}L(W_1 - W_b)^2 + \frac{1}{2}(L+D)W_0^2 - \frac{1}{2}DW^2 - \frac{1}{2}LW_b^2 = 0$$

1
2
3
4
5
6

In this energy balance, there are 6 terms:

- 1 Mixture compression work which is considered to be carried out at constant specific volume v_1 and from the pressure prevailing at the tuyère exit plane
- 2 & 3 Shock losses obtaining in the mixing chamber
- 4 Loss by kinetic energy of the gas having a non zero velocity at the ejector outlet. The outlet section is taken smaller than the actual one so as to account for energy losses within the diffuser.
- 5 Useful steam energy
- 6 Useful gas energy

Expanding Equation 1, simplifying and dividing by $2(L+D)$ and grouping the quadratic terms, we have:

$$(p_0 - p_x)v_1 - \frac{D}{L+D}WW_1 + \frac{W_1^2}{2} \left[1 + \left(\frac{W_0}{W_1} \right)^2 \right] - \frac{L}{L+D}W_bW_1 = 0 \quad \text{Equation 2}$$

The continuity equations can be written:

$$W = \frac{D}{F} v \quad W_1 = \frac{L + D}{F_1} v_1 \quad W_b = \frac{L}{F_b} v_b \quad \frac{W_0}{W_1} = \frac{F_1}{F_0}$$

Introducing these values into Equation 2, operating and setting λ as above,

$$p_0 - p_x = \frac{D^2 v}{F F_1} - \frac{(L + D)^2 v_1}{F_1^2} \lambda + \frac{L^2 v_b}{F_1 F_b} \quad \text{Equation 3}$$

This expression gives the vacuum at the tuyère exit plane. But it is more convenient to work with the vacuum (pressure) prevailing in the smokebox, this related to the working requirements of the boiler. It is

$$p_c - p_x = \frac{1}{2} \left(\frac{L}{F_b} \right)^2 v_b$$

Therefore it is

$$p_0 - p_x = (p_0 - p_c) + \frac{1}{2} \left(\frac{L}{F_b} \right)^2 v_b \quad \text{Equation 4}$$

Introducing Equation 4 into Equation 3, the fundamental expression giving the smokebox draught is obtained:

$$\text{Draught} = (p_0 - p_c) = \frac{D^2 v}{F F_1} - \frac{(L + D)^2 v_1 \left(\lambda + \frac{\xi}{2} \right)}{F_1^2} + \frac{L^2 v_b}{F_1 F_b} - \frac{1}{2} \left(\frac{L}{F_b} \right)^2 v_b (1 + \xi_b) \quad \text{Equation 5}$$

At this point, the friction at the mixing chamber and gas entrance are immediately (have been) introduced. ξ is a coefficient introduced to express the pressure drop as a fraction of the velocity head at the throat $\left(\frac{L + D}{F_1} \right)^2 v_1$ and ξ_b that corresponding to the gas entrance.

The whole of this treatment assumes that the problem is unidimensional, i.e. that the gas and steam streams are coaxial. Equation 5 can be easily solved numerically, giving arbitrary values to F_1 and F_b and choosing the pair making F a maximum. While it is possible to operate mathematically the expression so as to have a formula giving F directly, the former procedure has the advantage of showing the sensitivity of the design to a change in the different variables.

It can be written:

$$F = \frac{D^2 v}{(p_0 - p_c) F_1 + \frac{(L + D)^2 v_1 \left(\lambda + \frac{\xi}{2} \right)}{F_1} - L^2 v_b \frac{1}{F_b} + \frac{F_1 L^2 v_b (1 + \xi_b)}{2 F_b^2}} \quad \text{Equation 6}$$

Equation 6 can be written

$$F = \frac{A}{B + \frac{C}{F_0} + \frac{D}{F_0^2}} \quad \text{Equation 7}$$

in which

$$A = D^2 v$$

$$B = F_1(p_0 - p_c) + \frac{(L + D)^2 v_1 \left(\lambda + \frac{\xi}{2} \right)}{F_1}$$

$$C = -L^2 v_b$$

$$D = \frac{1}{2} L^2 v_b (1 + \xi_b) F_1 = -\frac{C}{2} (1 + \xi_b) F_1 \approx -\frac{C}{2} F_1 \quad \text{since } \xi_b \text{ is small.}$$

The optimal F_0 that makes F a maximum is when

$$\frac{dF}{dF_b} = 0 = \frac{-A \left[-\frac{C}{F_b^2} - 2 \frac{D}{F_b^3} \right]}{\left[B + \frac{C}{F_b} + \frac{D}{F_b^2} \right]^2}$$

i.e. when

$$\frac{C}{F_b^2} + 2 \frac{D}{F_b^3} = 0$$

whence

$$F_b = -2 \frac{D}{C} = \frac{-2 \left(-\frac{C}{2} F_1 \right)}{C} = F_1$$

This result shows that the optimal mixing chamber is not a constant section one. If this result is introduced into Equation 5, we get

$$\text{Draught} = (p_0 - p_c) = \frac{D^2 v}{F F_1} - \frac{(L + D)^2 v_1 \left(\lambda + \frac{\xi}{2} \right)}{F_1^2} + \left(\frac{L}{F_1} \right)^2 v_b - \frac{1}{2} \left(\frac{L}{F_1} \right)^2 v_b (1 + \xi_b)$$

$$\text{Draught} = \frac{D^2 v}{F F_1} - \left(\frac{L + D}{F_1} \right)^2 v_1 \left(\lambda + \frac{\xi}{2} \right) + \frac{1}{2} \left(\frac{L}{F_1} \right)^2 v_b (1 - \xi_b) \quad \text{Equation 8}$$

and

$$F = \frac{D^2 v}{F_1 (\text{Draught}) + \frac{(L + D)^2}{F_1} v_1 \left(\lambda + \frac{\xi}{2} \right) - \frac{1}{2} \frac{L^2}{F_1} v_b (1 - \xi_b)} \quad 1.01 \quad \text{Equation 9}$$

This equation can be solved graphically giving tentative values to $F_1 = F_b$ so as to get the maximum for a tuyère area giving the lowest back pressure.

This theory is based on Strahl's theory [Strahl, 1913] with convenient modifications. A numerical check against the application of the momentum theorem to the mixing chamber showed coincident results. Later, it was possible to demonstrate a rigorous equivalence.

The factor 1.01 accounts for a flow coefficient of 0.99 deduced experimentally from data taken at the Rugby test plant.

As a typical value for ξ , the figure of 0.10 can be taken, but a more exact estimation is possible if the theory given in (1) {Strahl?} {Porta L.D., Heat transfer and friction in ejector mixing chambers, 1974?} is accepted. $\xi_b \approx 0.04$ can be considered a typical value for a carefully designed bell mouth.

Equation 5 allows the prediction of the performance of any ejector satisfying the assumptions stated before, while Equation 9 gives the optimum dimensions. It is clear that the design, or the prediction of performance, cannot have better accuracy than the boiler data on which are based L , D , draught, v_b , etc. and also the steam conditions at the blast pipe tip. These can be forecasted with reasonable accuracy by means of laborious calculations, but undoubtedly actual test data are far more desirable. However, this is not enough: a locomotive ejector, the very heart of the machine, must incorporate due allowances for maintenance standards, fuel quality variations, service required from the locomotive, unavoidable steam and gas leaks, etc. Besides, it cannot be claimed that the present theory contemplates all the phenomena occurring in the ejector in full detail. Therefore it is to be used as the best available approach to an optimum, the final adjustment is to be carried out in practice by actual train running and fitting various tuyères of different area as it has been the universal practice since the days of George Stephenson. Fortunately, the main chimney dimensions are given by very flat curves and therefore it is possible to play within a fairly large latitude. To account for these phenomena, the ejector should be designed for 5% extra gas quantity and 10% more draught

The theory can take into account compressibility effects, which are actually influenced at the highest rate of working in engines operating with very high draught requirements (up to 700 mm H₂O in the SNCF 141R (see Appendix A4). Calculations should be carried out solely for the maximum rating since experience shows that the automaticity of an ejector so designed is satisfactory and not giving too large an excess of air at low working, while- and as a matter of course- no front end limit is apparent.

The calculations are to be repeated if the engine is called upon the work at very high heights above sea level, the atmospheric pressure becoming much lower (53% of sea level pressure at 4775 m above sea level on the Tocompa (?) line, Argentina). As it is well known, combustion phenomena change appreciably, yet a very limited experience seems to indicate that the same ejector behaves satisfactorily with independence of the height. But this does not prove that the optimum dimensions for sea level working remain also optimum at higher heights.

The whole design is a compromise because of the limited height available. The mixing chamber and the diffuser dispute the space on hand, and therefore the tuyère should be placed at the lowest possible position in the smokebox. So far as the written experience is concerned involving the gas producer combustion system, no fears arise of "burning too much at the front or the back" as many times has been said, but no evidence has been produced.

When carrying out the computations, due allowance should be made to account for leakage steam not passing through the blast pipe, air leakage into the smokebox, stoker jets or burner steam (increasing the gas quantity to be sucked), stoker engine steam, oil heater steam, vacuum ejector steam, lighting generator steam, steam to ashpan for the gas producer combustion system, etc. which are far from being negligible.

Another point to be considered is the one concerning building up inaccuracies whose influence can in no case be favouring, and therefore the actual design should incorporate all available means to provide adequate lining up and also least perturbations in the flow.

An allowance of 1% extra section should be considered on the value of F to account for the contraction coefficient.

Finally, last but not least, nothing is known about the influence of pulsating conditions obtaining in locomotives. So far, ejectors designed on the steady flow hypothesis have behaved satisfactorily and blast pipe area so predicted required little adjustment in practice and within $\pm 5\%$

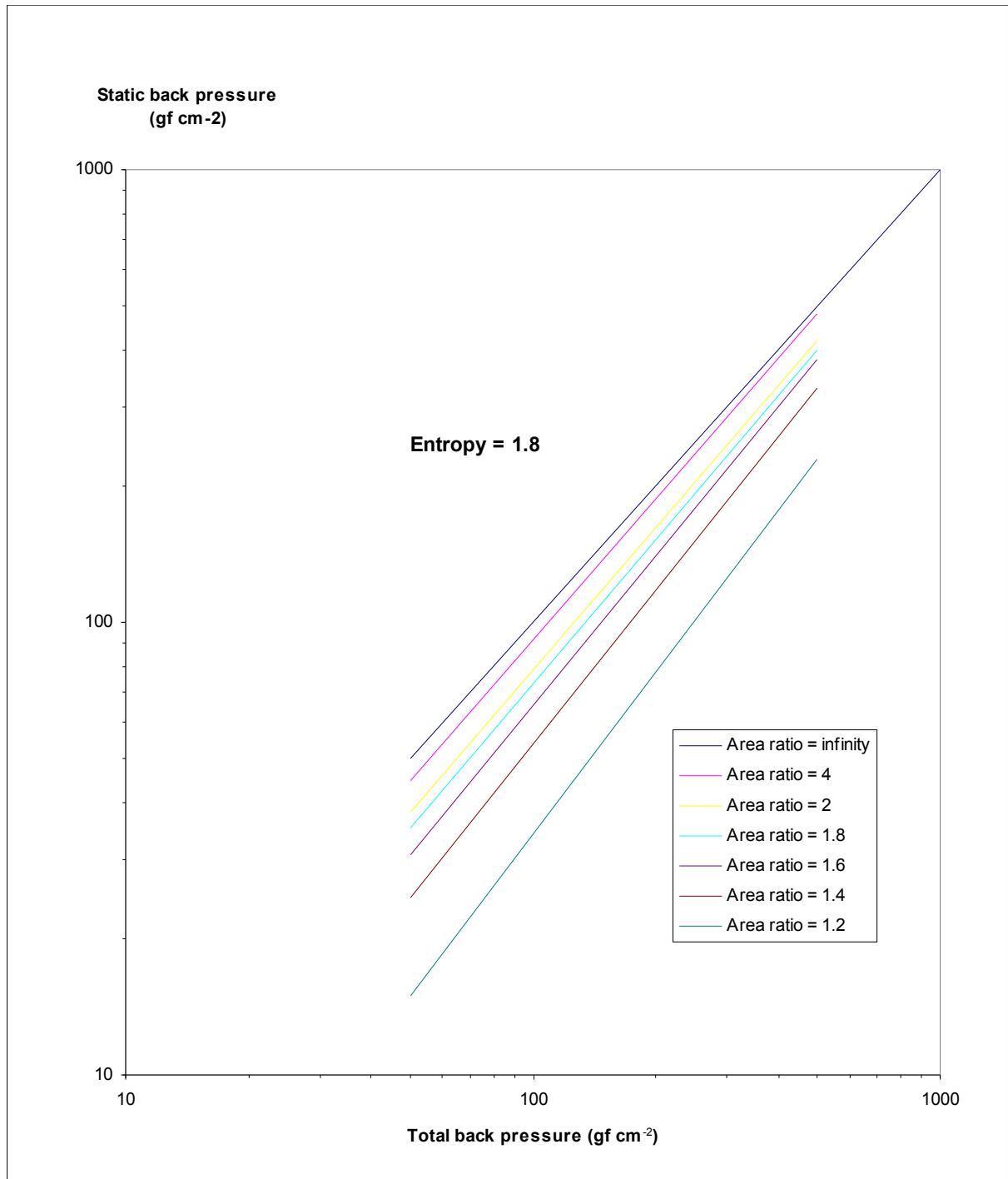


Fig 2: Relation between static and total back pressure

Comparative measures of ejector performance

Since most locomotives operate with similar relations between gas and steam mass ($L/D \approx 2$) and all of them have high superheat temperature and not too different internal cylinder efficiency, the gas and steam conditions do not differ too much. Therefore the usual back pressure - draught curve is a fair measure of ejector efficiency. The application of thermodynamic principles lead to consider the total back pressure which can either be measured with a Pitot tube in the exhaust stand (at $\frac{3}{4}$ radius of the tube) or corrected according to the graph given in Fig 2 if the static pressure has been measured.

Fig 3 gives comparative curves. Those reported to be ejectors designed by the author refer to Kylpor ejectors built some 15 years ago and therefore giving a somewhat worse performance of what could be expected from a today's design. It is seen that performance is some 40% better than any other type of apparatus, possibly increasing to 50% if the present knowledge is applied.

Another comparative parameter, easier to determine, is the relation given in Table 1.

$$\left(\frac{\text{tuyere_area}}{\text{gas_area_through_the_boiler}} \right)^2$$

The back pressure is proportional to the (tuyère area)² while the boiler draught is proportional to (gas area)². Due allowance should be made when comparing ejectors, to the fact that oil-burning and "gas producer" engines are so designed that part of the steam produced does not pass through the blast pipe.

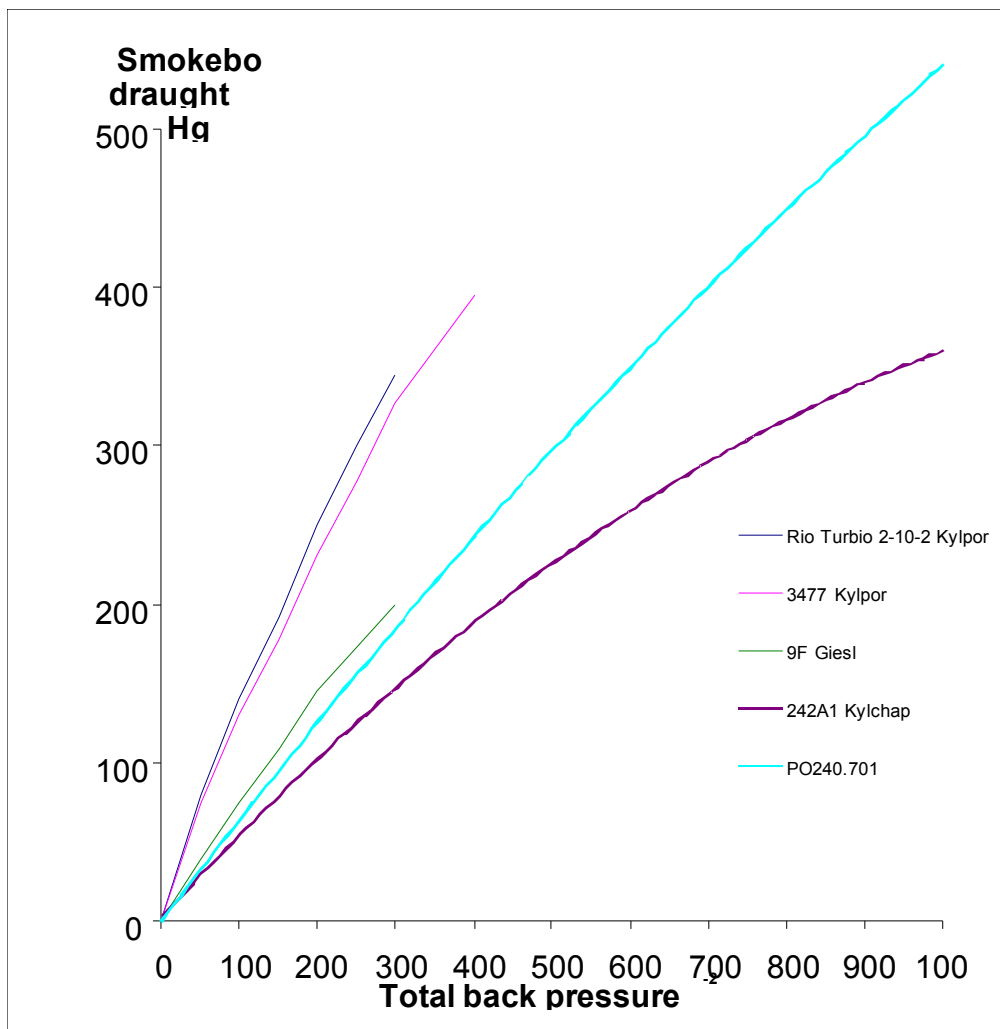


Fig 3 Comparative performance curves.

Table 1 $\left(\frac{\text{tuyere_area}}{\text{gas_area_through_the_boiler}} \right)^2$

Engine	Tuyère area F cm ²	Gas area Ω cm ²	F/Ω	(F/Ω) ² •10 ⁴	Ejector	Fuel
PO240.701	200	4942	0.0405	16.4	Kylchap	
242A1	246	5593	0.044	19.3	Kylchap	
141R	179	5091	0.035	12.4	Kylchap	Fuel oil
141R	179	5910			Kylchap	
141R	229	5910	0.039	15.0	Kylchap	Coal
141R469	199	5910	0.034	11.3	American	Coal
9F	195	5091	0.035	14.7	Giesl	Good coal
1802-FC6B Argentine	216	5000	0.043	18.0	Improved existing Kylchap	Oil burning
4674FC6B Argentine	110	2078	0.054	29.0	Kylpor	Gas producer combustion system, high ash coal
4674FC6B Argentine	115	2078	0.057	32.0	Kylpor	Gas producer combustion system, high ash coal

The back pressure

One of the ultimate goals of ejector design is to make a least demand to the engine in terms of back pressure or either to produce the maximum draught with a given back pressure. High draught produced economically means ample latitude to play with boiler design, gas areas, grate design, etc. essentially resulting in a smaller and lighter boiler, dispensing extra carrying wheels, good turbulence in the firebox, clean tubes, less smoke and happy crews because of an adequate steaming reserve. No effort should be spared in designing the ejector for maximum efficiency, otherwise one would fall into the trap of the “good enough engineering” which would come short of the hard competition obtaining in a near future.

The meaning of the back pressure goes further than the simple horsepower evaluation obtained after introducing its figure in the IHP formula: it is the lower(?) boundary of the steam cycle and its importance can be clearly shown when comparing its heat drop and the engine heat drop on the Mollier and of course its value is sensible at high powers and reduced cut-off.

Back pressure is usually measured with a manometer connected to the exhaust stand, but this is thermodynamically not correct because solely the static back pressure is reported: no account is given to the energy of the steam at the measuring point which is extracted from the pistons. Figure 2 gives the proper correction factor, which is not great in usual circumstances, but not so when the area ratio of stand pipe/tuyère is near unity. This happens especially in improved engines, not to say in author’s designs in which the exhaust stand area equals the tuyère area: the static pressure manometer always reads zero.

The normal design of the exhaust stand is such that each puff of one cylinder affects the piston on the other and therefore the time average back pressure, static or Pitot, is reflected in the indicator diagram. In the author’s designs the exhaust stand is so designed that the indicator line shows a lower back pressure than the manometer reading.

The time average back pressure can be obtained from the Mollier after an allowance for a flow coefficient of 0.99 which has been deduced from test data obtained at Rugby. This coefficient accounts for the actual reduction resulting from friction in the tuyère, the pulsating nature of the flow, the supersonic puffs, etc. One allowance should be made on the pressure around the tuyère, which is lower than the atmospheric.

Final comments

The theory here presented involves an accurate mechanical realization which is somewhat difficult given the kind of apparatus working in a hostile environment and handled by usually rough boilersmiths. It is felt that there is room for improvement and development work, particularly in considering the effect of pulsations.

Since the back pressure - draught experimental curves show a definite progress which it is hoped to still improve, it is felt that this will lead to a reduction of the boiler size for a given heat duty, therefore making a contribution to the increase of the power to weight ratio.

The present paper is a draught for a book (in preparation) to be entitled "*Modern Steam Locomotive Engineering*".

Appendix A1: The equivalent ideal diffuser

The conversion of kinetic energy into pressure in the diffuser is accompanied by considerable losses which must be accounted for.

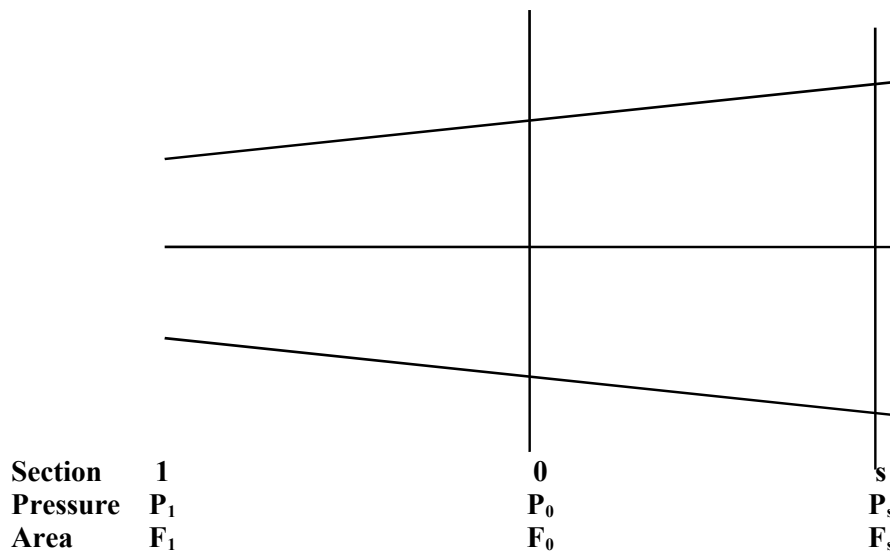
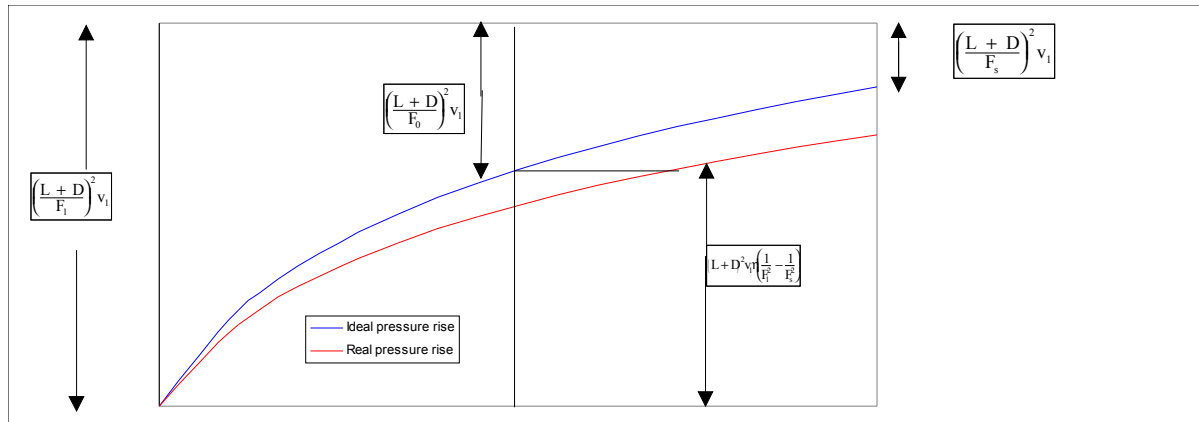


Fig 4 Equivalent ideal diffuser

The equivalent ideal diffuser is one in which the outlet section F_0 is calculated so that the sum of the kinetic energy loss is equal to the kinetic energy loss at section F_s plus diffuser friction.

it is

$$p_s - p_1 = \frac{1}{2} (L + D)^2 v_1 \eta \left(\frac{1}{F_1^2} - \frac{1}{F_s^2} \right) \quad \text{Equation 10}$$

and, in the ideal diffuser, where $\eta=1$

$$p_s - p_1 = \frac{1}{2} (L + D)^2 v_1 \left(\frac{1}{F_1^2} - \frac{1}{F_0^2} \right)$$

Hence

$$\eta \left(\frac{1}{F_1^2} - \frac{1}{F_s^2} \right) = \left(\frac{1}{F_1^2} - \frac{1}{F_0^2} \right)$$

Therefore

$$F_0 = \frac{1}{\sqrt{\frac{1-\eta}{F_1^2} + \frac{\eta}{F_s^2}}}$$

It is most difficult with the present state of knowledge existing in this particular problem, to give definite values for the diffuser efficiency η , especially because it is not known what are the inlet conditions on which that efficiency depends so much. Besides the whole question of the behavior of a diffuser under pulsating conditions, the following points should be considered to favor the efficiency of kinetic into pressure energy.

- Higher Reynolds number, of the order of $1 \cdot 10^6$
- Very high turbulence resulting from the previous mixing process.
- The application of boundary layer suction.
- Smoothness of the surfaces.

Against good efficiency, it can be set:

- The limited length available.
- The surface roughness obtaining in working conditions, especially in oil-burning engines.
- Some unavoidable inaccuracy of the whole draughting apparatus; it is a boilersmith construction operating in a hostile environment.

As a tentative figure, it is proposed to set $\eta=0.8$ up to 0.85 under favourable circumstances (short smooth accurate diffusers). With an area ratio of 4, the corresponding fraction of the kinetic energy converted is $0.80(1-1/4^2) \approx 0.75$, which is very high.

Appendix A2 The Kylpor ejector

The Kylpor differentiates from the Lempor by the fact that it makes use of the Kylala four lobe mixing device and therefore resembling the Kylchap of the type 1K/1T. Both types are thermodynamically equivalent except that the Kylpor has greater friction in the mixing chamber, this leading to the written preferences for the Lempor in recent designs. Nevertheless, the design is worthy of application in the case of a double concentric tuyère. In this design, the inner smaller tuyère is connected to the cylinder during the release phase, while both- the inner and the larger- gives an extra area during the exhaust phase. This larger area determines a very low back pressure on the exhaust line and the explanation is to be sought in the fact that actually most of the work is done by the puffing steam through a very small tuyère and thereby extracting a good deal of work of the incomplete expansion toe of the indicator diagram.

It can be shown that the upper mixing chamber of the Kylpor must be cylindrical while the bottom gas section and the tuyère must be equal to that of the tip of the Kylala petticoat. This is to comply with the requirement $F_i=F_b$.

Unlike former designs, there is no interest in having a uniform distribution of velocities at the Kylala exit and therefore this element should be as short as possible to reduce the unavoidable friction.

Appendix A3 Complementary notes

For usual locomotive practice

$$v \approx v_b \approx v_1 \approx 1.9 \text{ m}^3 \text{ kg}^{-1}$$

$$L/D \approx 2 ; F_1/F \approx 7; F_b/F \approx 7; W/W_b \approx 3.5; W_1/W \approx 0.43; W/W_b \approx 1.5$$

$$(\text{Dynamic pressure at throat})/(\text{dynamic pressure at bell mouth(gas)}) \approx 2.25$$

$$(\text{Dynamic pressure at throat})/(\text{dynamic pressure at blast pipe}) \approx 0.185$$

Recovery in diffuser = 75% of dynamic pressure at throat

$$d_1/d_f \approx 2.65 \text{ (} d_f = \text{equivalent diameter of the 4 nozzles)}$$

$$d_f/d_1 \approx 0.38 ; (F_0+F)/F \approx 8; d_b/d_1 \approx 1.07 \text{ (} d_b = \text{diameter of the bell mouth)}$$

$$d_f/d_b \approx 0.35; d_b/d_f \approx 2.8;$$

$$(\text{Height of mixing chamber/diameter of mixing chamber}) \approx 2.5$$

$$(\text{Chimney outlet diameter})/(\text{Throat diameter}) \approx 2; \text{Diffuser included angle} \leq 12^\circ$$

$$(\text{Draught at tuyère plane})/(\text{smokebox draught}) \approx 1.4$$

Specific heat of steam $\approx 0.47 \text{ cal kg}^{-1} \text{ K}^{-1}$; Specific heat of gas $\approx 0.27 \text{ cal kg}^{-1} \text{ K}^{-1}$; Specific heat of steam/gas mixture $\approx 0.34 \text{ cal kg}^{-1} \text{ K}^{-1}$

$$\text{Steam-gas mixture temperature} = 0.5(\text{steam temp.} + \text{gas temp.})$$

Appendix A4 Specific volume of steam, gas and steam-gas mixture

The steam specific volume is defined by engine performance. The latter can either be measured or, in the Mollier, the steam evolution in the cylinder represented starting from steam chest steam conditions and drawing the enthalpy drop line. Due allowance should be taken for the internal thermodynamic efficiency obtaining in the cylinder. The latter should be taken as 0.8 for modern single expansion engines and 0.88 for compounds as an average figure at high powers. Should compressibility be accounted for, the pressure at the tuyère can be obtained deducting some 1.4 smokebox draught from the atmospheric pressure. This correction may reach 10% and due account should be taken of the adiabatic temperature drop.

The gas specific volume v_0 at STP (0°C, 760 mm Hg) is obtained by calculation of the performance of the boiler, better if actually measured in controlled road or plant tests. If compressibility effects are accounted for, the same corrections as for steam are applied.

The specific volume of the gas mixture v_1 can be obtained by a weighted average of both the specific volume of steam v and gas v_b . An additional correction is that due to the temperature increase obtaining because of friction and shock losses in the mixing chamber. These losses are:

$$\frac{1}{2} D(W - W_1)^2 + \frac{1}{2} L(W_1 - W_b)^2 + \frac{1}{2} (D + L) W_1^2 \xi \quad [\text{kg m}^2 \text{ s}^{-3}]$$

$\xi = 0.1$; and are to be reported to unit mass of mixture. This gives a temperature increase from which a corrected value of v_1 can be obtained.