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THE USE OF TANDEM EJECTOR PUMPS IN AN INTERMITTENT BLOWDOWN TUNNEL

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ABSTRACT

A tandem ejector pumping system has been applied to an intermittent blowdown tunnel for cascade testing to achieve sub-atmospheric exit pressures and extend the operating range substantially. The ejectors are run from the same supply as the tunnel itself, but because they are only used at low Reynolds numbers when the cascade mass flow is small the overall running costs are kept low. A conventional one-dimensional ejector theory is developed in a new way for such an application as this, where the driving mass flow needs to be known for constant Mach number of the driven stream (the cascade exit Mach number). Several ejector geometries were tested in various configurations in a one-tenth scale model before the prototype ejectors were developed. It is demonstrated that by suitable grouping of terms it is possible to correlate both model and prototype ejector performance, and that this performance can be predicted sufficiently accurately by the theoretical model to justify its use as a design tool.

The method of operating two ejectors in tandem depends on the interaction of the exit stream of the first (forming the driven stream of the second) and the driver stream of the second. This is not immediately obvious, and is discussed fully in the light of the achieved performance.

Symbols

A - area, m^2

A^* - choked flow area, m^2

A_j - ejector jet area ratio

C_L - loss coefficient

D - diameter, m

L - length, m

M - Mach number

MFP - mass flow parameter

m - mass flow rate, kg/s

P - pressure, Pa

PP - primary parameter

Re - Reynolds number

SP - secondary parameter

T - temperature, K

V - velocity, m/s

γ - ratio of specific heats

ρ - density, kg/m^3

Subscripts

o - total conditions

1-7 - locations (see Figure 2)

atm - atmosphere

ej - ejector pumping flow

INTRODUCTION

Many of the advances in high-temperature, high-pressure gas turbine aerodynamics in recent years have come about as a result of testing cascades of blades in high-speed wind tunnels to measure aerodynamic losses and blade surface pressure distributions, and to understand the structures of the boundary layers and the wakes. For simulating the correct operating conditions, such testing depends on reproducing the prototype dimensionless quantities

Mach number and Reynolds number. An aeroplane gas turbine engine, in passing through a complete flight cycle, will be required to operate over wide ranges of conditions. This makes it highly desirable that the dimensionless parameters of the model cascade, or equivalently for adiabatic flow, the upstream and downstream pressures, be independently and continuously variable.

From this point of view the best tunnels in the past have usually been of the closed-loop type, which run continuously [1,2]. Their disadvantage is that they are very expensive to run. The power required to drive such a tunnel increases with the cascade mass flow, and even for modest blade sizes (typically 100mm chord, 300mm span, 7 blades) is in the megawatt range. The alternative of using smaller blades is rarely attractive because of the loss of definition this implies.

Intermittent tunnels are much cheaper to operate because they can rely on an air reservoir which is pumped to pressure over a period of time using relatively small plant. Such tunnels are usually either of the suckdown type, open to atmosphere at inlet and venting to a reservoir at sub-atmospheric pressure, or of the blowdown type, supplied from a high pressure reservoir and venting to atmosphere. In both cases one of the bounding pressures to the cascade is constrained. It may be possible to arrange for some variation by choking or diffusing but such arrangements are cumbersome and are limited in scope.

In an effort to overcome these limitations and design a tunnel which would be relatively inexpensive to run but would have a wide operating range, the authors recognised that the exhaust duct of a blowdown tunnel could be equipped with ejector pumps, driven from the same air reservoir as the cascade, to pump the exit pressure down to sub-atmospheric levels. Since these would correspond to low Reynolds number conditions where the cascade mass flow is low, there would be air available to drive the ejectors. Some preliminary studies based on expected blade operating conditions and sizes indicated that two stages of ejection, with diffusion between them, would be required. These studies also showed that the running costs of such a system (Figure 1) could be as much as two orders of magnitude lower than an equivalent continuous tunnel.

A description of the tunnel and its commissioning has been given in reference [3]. The ejector pumping system is one of the most important features of the tunnel, and its application has considerable novelty, both for high-speed wind tunnels, and for multi-staging ejectors to pump compressible airflows at high mass flow rates over large pressure ratios.

The application of single-stage ejectors to pump high-speed wind tunnels goes back many years [4,5], but to the authors' knowledge none operate in the low pressure regime proposed here. Howell [6] used a single-stage pump to achieve pressures as low as 0.06 bar in a hypersonic blowdown tunnel, but with extremely low main tunnel mass flow rates. There appears to be no available literature relating to the use of tandem ejectors under these circumstances.

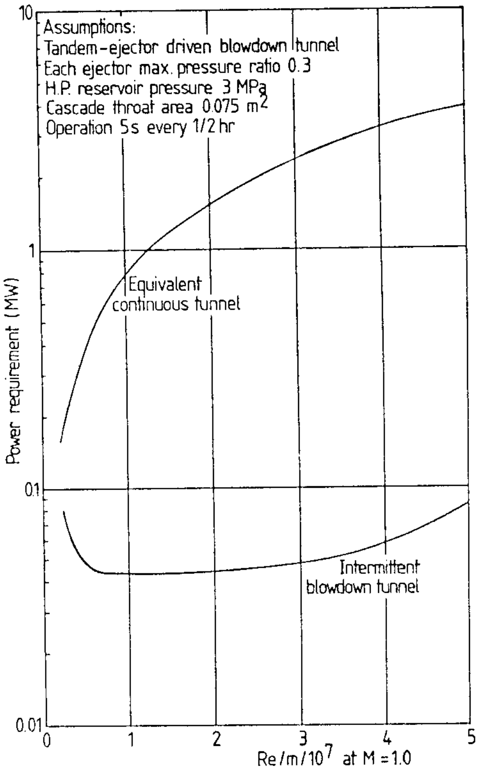


Fig. 1 Running cost comparison

In view of the general lack of knowledge in this area and the absence of guidelines for design, it was recognised that considerable development work was necessary to ensure that the prototype ejectors would achieve the desired performance. The authors therefore have sought to develop, first, a design procedure which would be relatively simple to use in this application; second, small scale models of proposed tandem ejector systems to check their performance and also the accuracy of the model; and finally, the prototype system. Full descriptions of all three aspects are given here.

2. THEORY OF EJECTORS

The theory of one-dimensional, compressible flow in an ejector pump has been developed in several papers, of which [7] is a good example. There are limitations inherent in such analyses, and the approach adopted by Whitaker [8] has been to introduce loss coefficients based on experimental data. Increasingly the limitations of the one-dimensional approach are being overcome by more detailed, two-dimensional analyses, for example, by Hoggarth et al [9]. Bonnington, King and Hemmings [10] give a comprehensive review of the literature.

It was felt that the extra sophistication of a two dimensional approach would not be justified in view of the additional computational labour involved, especially as the study was essentially a design exercise. The simplified analysis by Whitaker [8] was therefore reworked to include the diffuser in the first stage ejector, and to make it suitable for the present purposes.

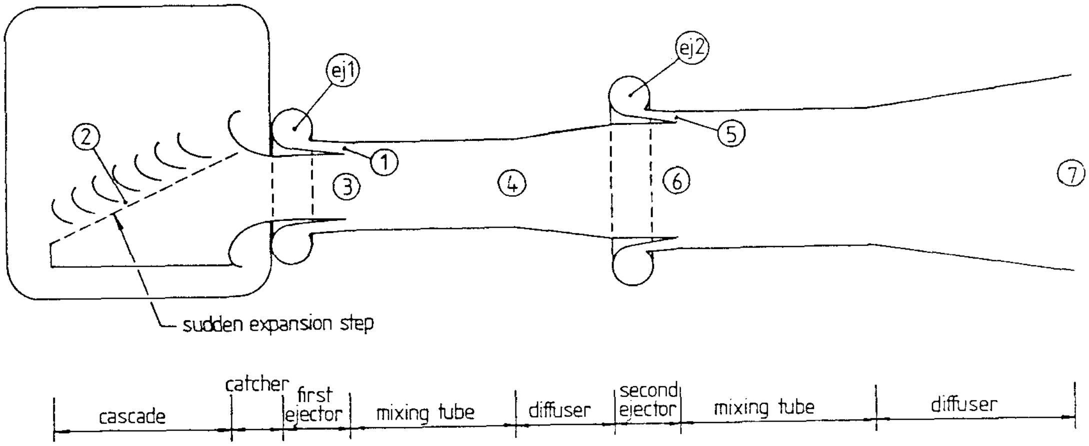


Fig. 2 Schematic diagram of tandem ejector system

Figure 2 is a sketch of the cascade and the exhaust duct containing the two ejectors, and also defines the stations referred to in the text. The basic analysis applies equally to both ejectors, but since the boundary conditions imposed on them are slightly different it is convenient to develop the equations initially as they apply to the first ejector, and subsequently to note the differences between them.

2.1 First Ejector

The first ejector consists of the ejector nozzle, mixing tube and diffuser. The diffuser is treated as perfect and any diffuser losses are combined with friction losses in the mixing tube. Thus the diffuser exit pressure P_6 is assumed to be equal to the total pressure at inlet, P_{04}.

The conservation equations for the control volume bounded by stations 1, 3 and 4 and the parallel sided mixing tube are:

$$\dot{m}_1 + \dot{m}_3 = \dot{m}_4 \tag{1}$$

$$(P_1 + \rho_1 V_1^2)A_1 + (P_3 + \rho_3 V_3^2)A_3 - C_L \tfrac{1}{2}\rho_4 V_4^2 A_4 = (P_4 + \rho_4 V_4^2)A_4 \tag{2}$$

$$T_{01} = T_{03} = T_{04} \tag{3}$$

where C_L is the combined mixing tube and diffuser loss coefficient. With the application of the compressible flow equations, equations (1) and (2) take the dimensionless form

$$PP \frac{M_1}{\left(1 + \frac{\gamma-1}{2} M_1^2\right)^{\frac{1}{2}\left(\frac{\gamma+1}{\gamma-1}\right)}} + SP \frac{M_3}{\left(1 + \frac{\gamma-1}{2} M_3^2\right)^{\frac{1}{2}\left(\frac{\gamma+1}{\gamma-1}\right)}} = \frac{M_4}{\left(1 + \frac{\gamma-1}{2} M_4^2\right)^{\frac{1}{2}\left(\frac{\gamma+1}{\gamma-1}\right)}} \tag{4}$$

$$PP \frac{1 + \gamma M_1^2}{\left(1 + \frac{\gamma-1}{2} M_1^2\right)^{\frac{\gamma}{\gamma-1}}} + SP \frac{1 + \gamma M_3^2}{\left(1 + \frac{\gamma-1}{2} M_3^2\right)^{\frac{\gamma}{\gamma-1}}} = \frac{1 + \gamma(1 + \frac{1}{2}C_L)M_4^2}{\left(1 + \frac{\gamma-1}{2} M_4^2\right)^{\frac{\gamma}{\gamma-1}}} \tag{5}$$

which introduce:

$$\text{the primary parameter, } PP = \frac{P_{01}A_1}{P_{04}A_4} = \frac{P_{01}}{P_{04}} A_j \tag{6}$$

$$\text{the secondary parameter, } SP = \frac{P_{03}A_3}{P_{04}A_4} = \frac{P_{03}}{P_{04}} (1 - A_j) \tag{7}$$

$$\text{the jet area ratio, } A_j = A_1/A_4 \tag{8}$$

The mass flow ratios of the driven to driver flows is given by the mass flow parameter:

$$MFP = \frac{SP \dfrac{M_3}{\left(1 + \frac{\gamma-1}{2} M_3^2\right)^{\frac{1}{2}\left(\frac{\gamma+1}{\gamma-1}\right)}}}{PP \dfrac{M_1}{\left(1 + \frac{\gamma-1}{2} M_1^2\right)^{\frac{1}{2}\left(\frac{\gamma+1}{\gamma-1}\right)}}} \tag{9}$$

Normally, the ejector designer wishes to maximise MFP for a given pressure rise and hence, a given SP. M_1 and A_j are set by the driving nozzle design and the equations are then solved iteratively to give a graph of MFP against SP.

For the blowdown cascade, however, the problem is to design an ejector which can vary the cascade exit pressure, and hence the Reynolds number, for a constant cascade exit Mach number. In this case M_3 is also constant, and so, by solving equations (4) and (5) explicitly for PP and SP in terms of M_3 and M_4 over a range of values of M_4, a graph of P_{03}/P_{04} vs P_{01}/P_{04} is obtained, and this predicts the performance for a constant cascade exit Mach number.

In the proposed scheme the cascade exit flow travels through a bellmouthed "catcher" into the first stage ejector, rather than exhausting first into a plenum, as is commonly done. In this way much of the exit dynamic head is saved, which eases the pumping requirements of the ejector system. A sudden expansion step is also arranged at the cascade exit to ensure that the flow separates cleanly from the cascade (Figure 2). The ratio of cascade exit static pressure to ejector driven stream pressure P_2/P_{03} is then calculable from a simple control volume analysis of the step expansion, with allowance for the pressure recovery of the catcher. Finally, then, it is possible to arrive at the pressure P_2/P_{04} as the ordinates for the ejector performance curves. This is a convenient parameter, since P_2 is the required pressure and P_{04} can readily be deduced. If only the first stage ejector is to be used, then $P_4 = P_{atm}$ and P_{04} can be calculated knowing the total mass flow (or for adequate diffusion, $P_{04} \simeq P_4 = P_{atm}$). If both stages are to be used, then P_{04} becomes the independent variable which determines how much each ejector will contribute to the overall pumping.

For the abscissae it is convenient to use not P_{01}/P_{04}, but $(P_{01}A^*_{ej1})/(P_{04}A_4)$, where A^*_{ej1} is the driver flow throat area, since in the proposed scheme A^*_{ej1} can be varied as well as P_{01} to change the ejector pumping.

2.2 Second Ejector

The analysis of the second stage ejector is very similar. The ejector is located immediately downstream of the diffuser, and exhausts to atmosphere. It is assumed that the dynamic head of the driven stream after it has been diffused is negligible. The performance is therefore plotted as P_6/P_{07} vs. $(P_{05}A^*_{ej2})/(P_{07}A_7)$, where $P_6 = P_{04}$ and $P_{07} = P_{atm}$.

3. DESCRIPTION OF MODEL EJECTORS

To test the feasibility of a tandem ejector pump system a one-tenth scale model was constructed. Five different ejector configurations were used in the experimental programme (Figure 3):

(1) a single stage ejector, in which the high pressure pumping air is introduced into the mixing tube by four discrete jets

(2) a single stage ejector, in which the high pressure pumping air is introduced into the mixing tube by an annular jet

(3) a modified form of (2), with a longer mixing tube

(4) a modified form of (3), in which the throat area of the pumping jet is reduced and the driving pressure increased

(5) a two-stage ejector which uses (4) as the first stage; the second stage is a 1.5 x linear scale version of (4).

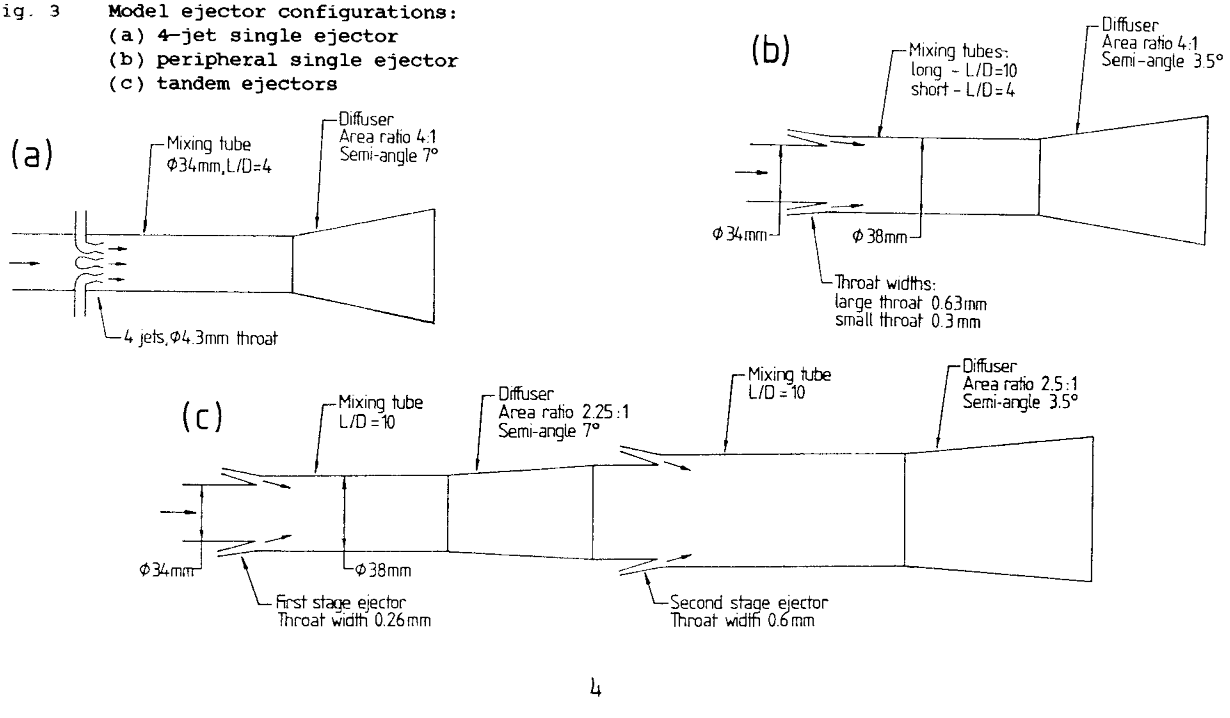
4. PERFORMANCE OF THE MODEL EJECTORS

The performance of the four different single-stage ejectors is summarised in Figure 4. The results show that the peripheral ejector with the long mixing tube pumps to the lowest pressure, pressure ratios of 5:1 being achievable. The parameters derived in section 2 collapse the data for different ejector throat areas and pumping pressures well, and in fact there is remarkably little difference between the curves for no cascade flow and sonic cascade exit conditions.

The results indicated clearly that there was a performance limit for each configuration, and that the ultimate pressure obtainable by the ejector depends on the length of the mixing tube and on the pressure of the driving stream. A long mixing tube is necessary at high delivery pressures to ensure that the supersonic jet from the ejector nozzle mixes down to a subsonic flow before entering the diffuser (Figure 5). The superior performance of the ejector with the smaller throat area driven at a high pressure can be explained by the fact that at the higher pressures the momentum influx per unit mass flow is higher.

The testing of the tandem ejector system was limited to ensuring that overall pressure ratios as low as 0.1 could be achieved, and that no interaction occurred between the two stages. Figure 6 shows that both of these aims were met. With no cascade flow the ultimate pressure ratio is 0.05, and in a separate test with cascade flow a pressure ratio of 0.1 was achieved.

Fig. 3 Model ejector configurations:
(a) 4-jet single ejector
(b) peripheral single ejector
(c) tandem ejectors



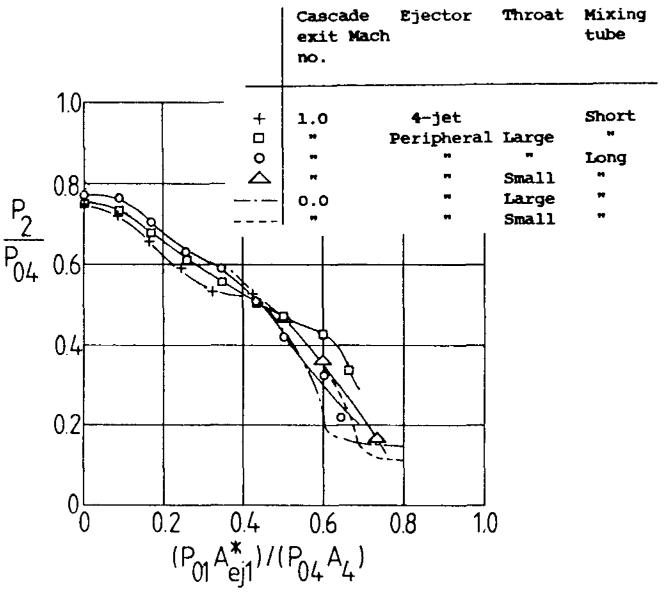


Fig.4 Comparison of performance of model first-stage ejectors

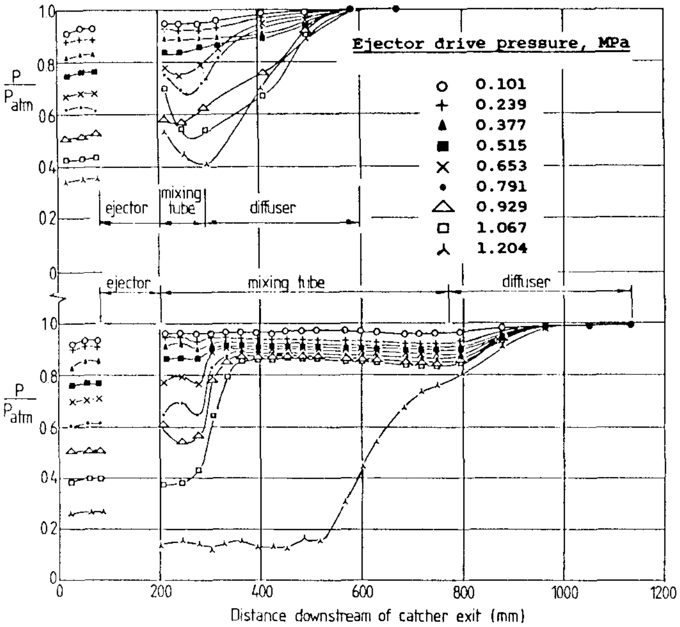


Fig. 5 Comparison of static pressure distributions downstream of a single ejector for two lengths of mixing tube

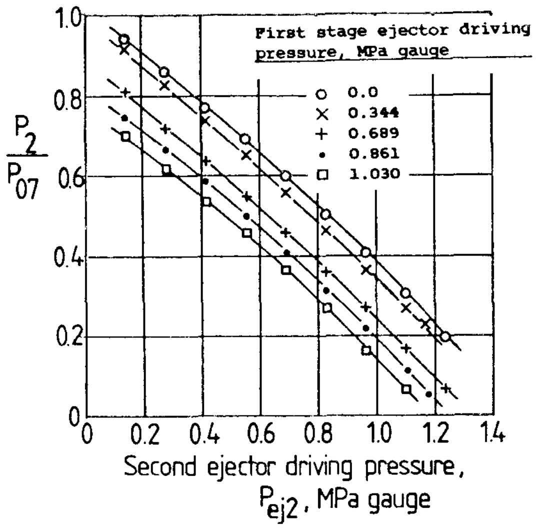


Fig. 6 No-flow performance of model tandem ejectors

5. DESCRIPTION OF THE PROTOTYPE EJECTORS

With the experience gained from the model testing the prototype ejectors were designed. Figure 7 shows the principal dimensions of the system and details of the two ejectors. The first stage is a peripheral ejector with a throat area which can be varied by means of an inner sleeve moved axially on a screw thread. Unfortunately it also proved to be a very expensive component, and to save both time and money the second stage was constructed in the form of four discrete jets ejecting into the main flow at a shallow angle. Each jet may be switched in or out individually, so that this ejector is incrementally rather than discretely variable. This discrete adjustment is not a disadvantage, as will be shown when the method of operating two stages in tandem is discussed (section 7).

6. COMPARISON OF RESULTS

6.1 No Cascade Flow

The no-flow performance of both prototype ejector stages is shown in Figure 8, and for comparison, the equivalent model performance and two performance predictions. The limit of pumping at a pressure ratio of about 0.2 is apparent, as are the consequences of increasing the ejector driving pressure past this limit, when the mixing process is blown out of the mixing tube and a rather unstable supersonic flow is established in the diffuser. The performance of both prototype ejectors is relatively insensitive to mode of operation: whether run individually or in tandem. The second ejector appears to be somewhat less efficient that the first at the lower pressure ratios, presumably because of its less favourable geometry.

The way in which model and prototype data and theory all collapse, if not to a line, at least to a narrow band, is encouraging. The performance scales well with size, and while a one-dimensional treatment will always have limitations, it is quite adequate here as a design tool.

6.2 First Stage, With Flow

The performance of the first stage ejector with cascade flow, and again with model and theoretical comparisons, is shown in Figure 9. Here the agreement is not quite as good. The prototype ejector appears to be slightly more efficient than the model or the theoretical prediction, but the lowest pressure ratio achievable was only 0.25. There would appear to be a case here for a longer mixing tube, but unfortunately site limitations did not permit this. During the prototype tests it was not always possible to maintain a constant cascade exit Mach number, and hence driven flow Mach number, but the performance is clearly only a weak function of this.

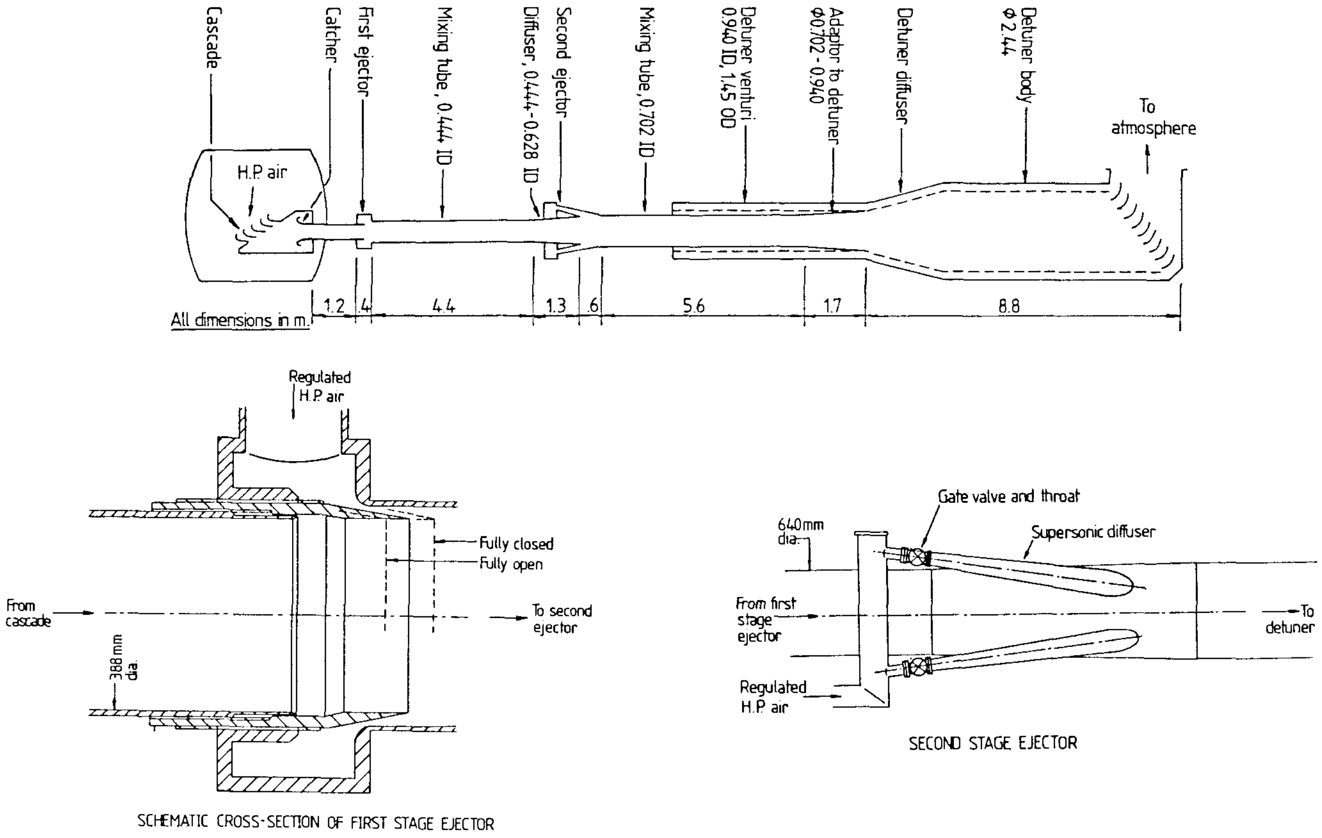


Fig. 7 Prototype tandem ejector layout and details

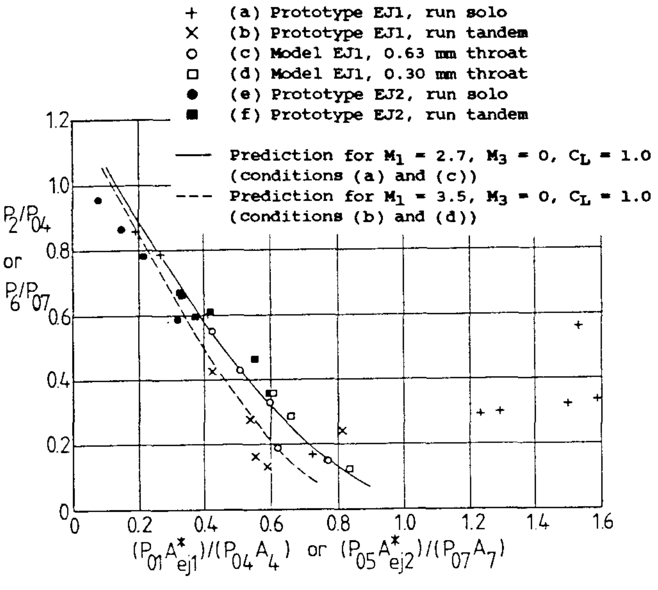


Fig. 8 No-flow ejector performance

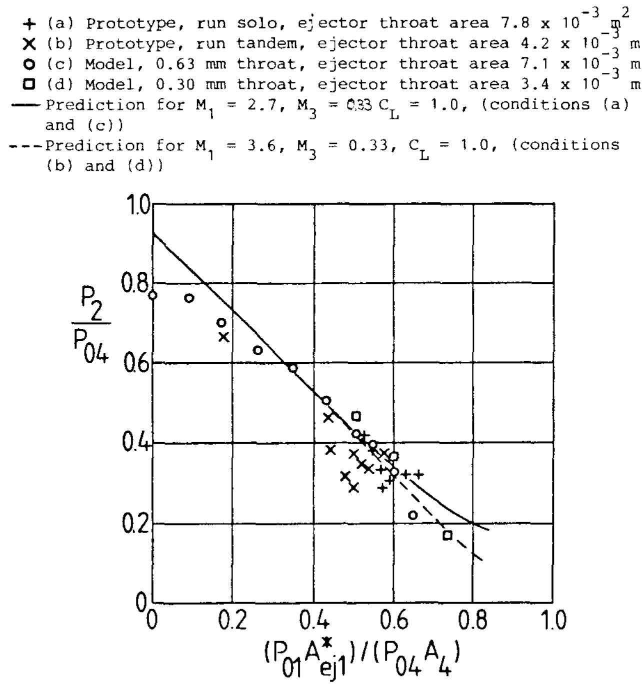


Fig. 9 First stage ejector performance with cascade flow

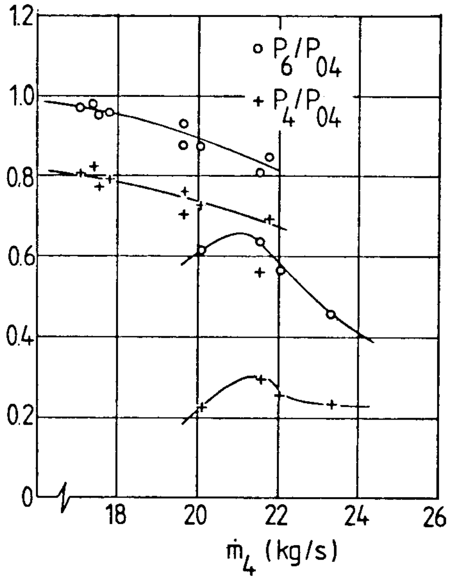


Fig. 10 Mixing tube and diffuser pressures, showing subsonic and supersonic regimes

Figure 10 demonstrates clearly what happens when the pumping limit is reached. As this point is approached the diffuser pressure recovery is reduced, and then quite suddenly the flow switches from subsonic to supersonic. The fact that there is some overlap between the two regions suggests that the point at which this occurs is very sensitive to small changes in the ejector flows and the processes of mixing.

6.3 Second Stage, with Flow

In the initial design studies it was assumed that the diffuser is perfect. In reality, of course, there is always some residual velocity at exit, and hence in the driven stream of the second ejector. The comparison of theory and experiment (Figure 11) suggests that this is indeed so (but will clearly depend in magnitude on the cascade and first stage flow rates).

The similarity of the theoretical curves bears out the statement made in section 6.2 about insensitivity to driven flow Mach number. The data in Figure 11 were measured for a variety of numbers of nozzles turned on, and clearly the mixing process does not suffer if there is a circumferential asymmetry in the driving jets.

7. METHOD OF OPERATION OF TANDEM EJECTORS

A comparison of Figures 8, 9 and 11 reveals that at the higher pressure ratios the first ejector, with flow, is markedly more efficient than the second ejector, or the no-flow performance of either. This is because when there is flow through the cascade the first ejector gains the benefit of the pressure recovery of the step expansion and catcher described in section 2. If the second ejector is operated alone this benefit is largely lost in the diffuser. When only modest amounts of pumping are required, therefore, it is the first rather than the second

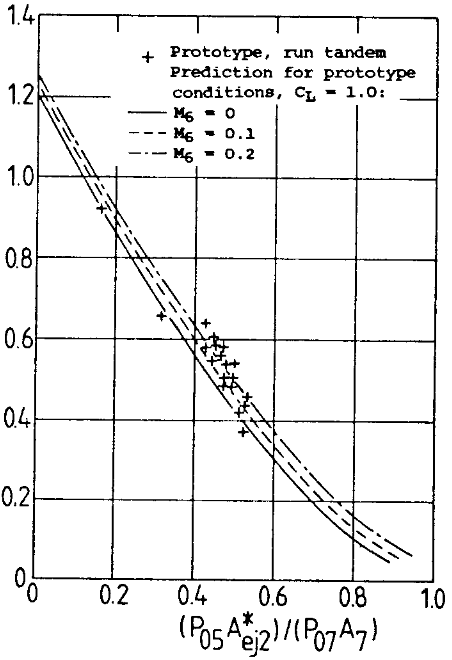


Fig. 11 Second-stage ejector performance with cascade flow

ejector which is used. An additional advantage is that the first ejector is continuously-variable and can be "fine-tuned".

As the pumping requirement is increased the first ejector eventually reaches its limit and the second ejector has to be brought in. However, if this ejector is simply opened at this point, it will have the effect of reducing P_{04} and hence increasing $(P_{ej1}A^*_{ej1})/(P_{04}A_4)$, taking the first ejector past its pumping limit and inducing supersonic flow in the diffuser. Qualitatively, this can be visualised as sucking the mixing process out of the mixing tube. Furthermore, the driver flow of the first ejector becomes the driven flow of the second ejector, so that any increase in driving flow to the first ejector requires a corresponding increase in driving flow to to the second ejector in order to realise the same pressure ratio. When running tandem ejectors, then, the technique is to run the second ejector as hard as necessary (which usually means full open, because of the overlap in performance of the two stages), and then open the first ejector only as far as necessary to achieve the overall pressure ratio. Because of this, an incrementally variable second stage is quite acceptable.

A simple multiplication of the ultimate limits of the two ejectors (Figures 9 and 11) suggests that it is possible to achieve pressure ratios below 0.1 with the existing system. In fact this has not been possible because of air supply limitations – the lowest pressure ratio recorded to date with flow is 0.18. In time the authors hope to realise the full potential of the system.

8. CONCLUSIONS

A tandem ejector pumping system has been applied to an intermittent blowdown cascade tunnel to achieve the running cost advantages of such a tunnel but retain the flexibility of independent and continuous variation of Mach and Reynolds numbers normally associated only with closed loop, continuous tunnels. The equations of flow governing the ejection process have been reworked to produce a simple design procedure which, although one-dimensional and ultimately subject to inherent limitations, has been very effective in these studies.

A scale model ejector system was constructed to demonstrate the feasibility of the concept and explore different geometries. The experience gained was used in the design of the full-size prototype. By suitable grouping of terms into non-dimensional parameters, it was possible to collapse prototype and model performance data and theoretical predictions, so that the ejector performance is readily predictable.

The model ejectors, used together, were able to pump pressure ratios as low as 0.1 (10:1). The full performance of the prototype system has not yet been realised because of a limit on the air supply to the ejectors, but nevertheless a pressure ratio of 0.18 has been achieved.

ACKNOWLEDGEMENTS

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