Performance of Choked Unsteady Ejector-Nozzles for use in Pressure-Gain Combustors

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If the conventional steady flow combustor of a gas turbine is replaced with a device which achieves a pressure gain during the combustion process then the thermal efficiency of the cycle is raised. All such 'Pressure Gain Combustors' (e.g. PDEs, pulse combustors or wave rotors) are inherently unsteady flow devices. For such a device to be practically installed in a gas turbine it is necessary to design a downstream row of turbine vanes which will both accept the combustors unsteady exit flow and deliver a flow which the turbine rotor can accept. The design requirements of such a vane are that its exit flow both retains the maximum time-mean stagnation pressure gain (the pressure gain produced by the combustor is not lost) and minimises the amplitude of unsteadiness (reduces unsteadiness entering the downstream rotor). In this paper the exit of the pressure gain combustor is simulated with a cold unsteady jet. The first stage vane is simulated by a one-dimensional choked ejector nozzle with no turning. The time-mean and rms stagnation pressure at nozzle exit is measured. A number of geometric configurations are investigated and it is shown that the optimal geometry both maximizes time mean stagnation pressure gain (75% of that in the exit of the unsteady jet) and minimizes the amplitude of unsteadiness (1/3 of that in the primary jet). The structure of the unsteady flow within the ejector nozzle is determined computationally.

I. Introduction

If a stagnation pressure gain could be achieved across the combustion process in a gas turbine then its entropy rise would be reduced compared with a conventional combustor, and the exergy and availability of the exit flow would be increased. Sir William Hawthorne in the conclusions of his 1994 IGTI scholar lecture [1], captured this concisely when he said '*the largest loss of thermodynamic availability occurs in the combustion chamber. What is needed is a work producing combustion chamber.'* If such a combustion chamber could be practically realized then the rise in turbine inlet exergy and availability would result in a step increase in gas turbine thermal efficiency.

Three types of pressure gain combustor have been reported in the literature, pulse combustors, pulse detonation engines (PDEs) and wave rotors. Each has relative merits and disadvantages, however all produce a highly unsteady exit flow which has a higher exergy than would be possible with a conventional steady flow combustor. If a practical engine using such a combustor is to achieve a higher thermal efficiency, then the downstream turbine must be able to extract a significant proportion of the additional exergy as shaft work. To achieve this, a first stage turbine vane must be designed which accepts the unsteady combustor exit flow and delivers a flow to the downstream rotor which retains the maximum possible time mean exergy and stagnation pressure. It is also a requirement of the first turbine vane that the exit flow has the minimum variation in stagnation pressure. This is because unsteadiness in the nozzle exit flow will propagate into a downstream rotor and may adversely affect the efficiency.

Three conceptual methods of joining a pressure gain combustor to a downstream turbine stage have been reported in the literature. Schematics of the three are shown in figure 1. The papers are all concerned with ensuring that the increase in work output from a downstream turbine would be maximised.

In the first method, fig 1a, a plenum is located between combustor exit and turbine inlet. Such a geometry was reported by Gemmen et al [2]. They found that the plenum reduced unsteadiness in the exit flow but that the dissipation of the exit flow in the plenum resulted in a stagnation pressure gain of only 1%. These tests show that dumping the flow into a plenum at combustor exit would significantly reduce the benefit of pressure gain combustion.

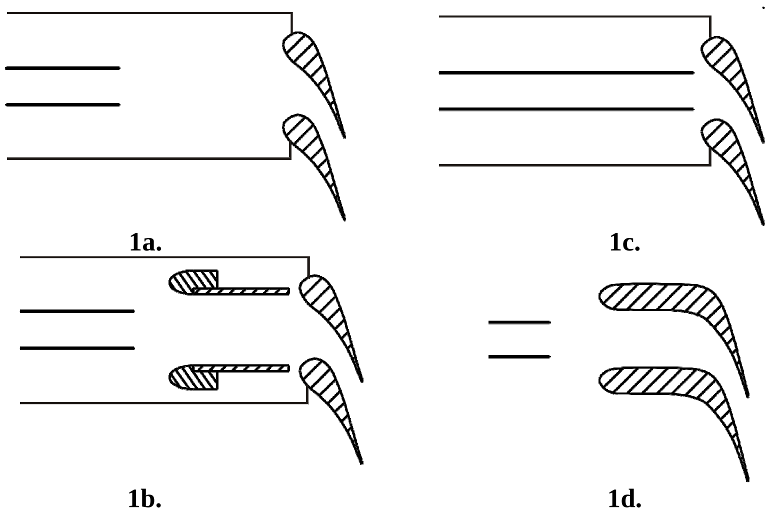


Figure 1. Possible methods of coupling a pressure gain combustor to a downstream turbine

In the second method, fig 1b, an ejector or similar geometry is mounted downstream of the pressure gain combustor and a short transition piece is used to join the ejector exit flow to the turbine inlet. Paxson and Dougherty [3] explicitly used an ejector and achieve a pressure gain of 3.5%. Kentfield et al [4] and Porter [5] used a contracting tube, similar in geometry to an ejector, to achieve a pressure gain of 4% and 8% respectfully. This should be compared to a typical 2-6% stagnation pressure drop across a conventional combustor [6]. Paxson and Dougherty also found that the unsteadiness in the exit flow from the ejector was small, 5% of the mean. These results are very promising, however the large surface area of the combined ejector, transition piece and first stage turbine nozzle make the geometry likely to incur high frictional losses and to be difficult to cool.

In the third method, fig 1c, the combustor exit is connected directly to the inlet of the downstream turbine. This is similar to the approach used in 'pulse turbocharging' of IC engines. In this method the distance between the car cylinder and turbocharger turbine is minimised so that flow with maximum exergy enters the downstream turbine. In most cases the benefit from increasing the exergy at turbine inlet will more than offset the loss in efficiency due to high levels of unsteadiness at turbine inlet [7]. To the authors' knowledge the only paper that reports a detailed experimental investigation of pulse turbocharging an axial turbine is Daneshyar et al [10]. They mount an unsteady pulse generator directly upstream of an axial flow turbine. The pulses used were characteristic of those encountered in IC engines and were much larger in amplitude than those produced by pulse combustors but are likely to be a lot smaller than those produced by PDEs. The effect of feeding the pulsed inlet flow directly into the turbine was to reduce its efficiency by 10%. It is clear from these tests that directly coupling a pressure gain combustor to the conventional turbine can result in a significant reduction in turbine efficiency.

The available literature raises the question of whether an ejector and first stage vane can be combined to form a single component which at inlet has the geometry of an ejector and at exit has the geometry of a first stage vane, shown in fig 1d. The aim of this component would be to achieve a high pressure gain whilst delivering a relatively steady flow to the turbine rotor. Such a component would have a much smaller surface area than would be obtained by using a separate ejector, transition piece

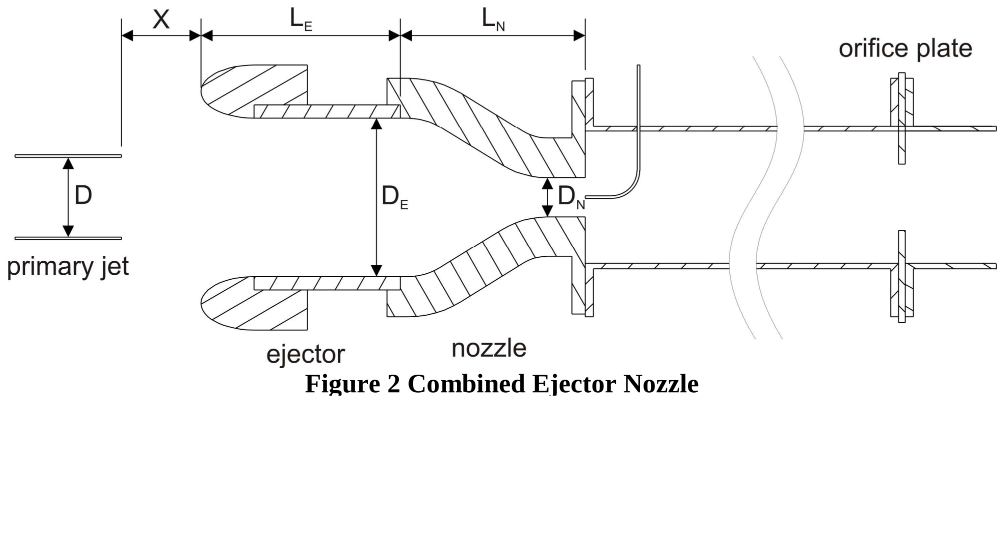


Figure 2 Combined Ejector Nozzle

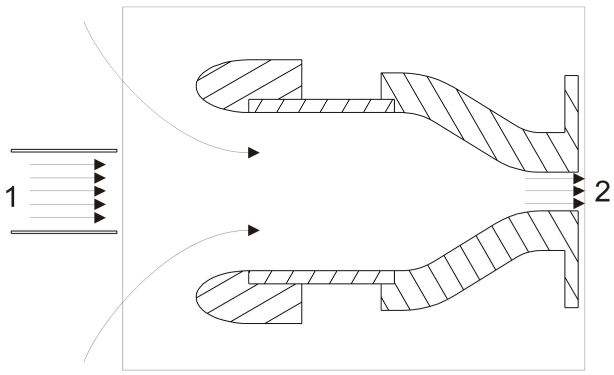


Figure 3 Control Volume

and vane and would thus produce less frictional loss and be easier to cool. This paper reports the testing of such an ejector-vane combination. To simplify the testing the turning in the vane was not simulated and the vane was modeled as a one-dimensional nozzle (fig 2). The ejector geometry chosen for the tests was the optimal geometry reported by Mason and Miller [8]. In the paper the distance between the ejector leading edge and nozzle exit was systematically varied to determine its effect on the percentage of the additional exergy created by the pressure gain combustor which is preserved to the nozzle exit and rms unsteadiness there.

II. Experimental and computational methods

The experimental setup is shown in fig. 2. The experiment was designed to allow the rapid assessment of different geometrical configurations. This was achieved using a simplified configuration. Only one primary jet and ejector-vane passage were simulated and the vane exit turning was not simulated. The pressure ratio across the ejector-nozzle, achieved with a downstream vacuum pump, was set so that the exit remained choked during testing. To allow accurate measurements of performance to be made the experiments were performed isothermally at ambient temperatures.

The primary jet is produced by a resonance tube connected at one end to a piston. The piston oscillates, exciting the ¼ wave mode of the tube. The velocity at the open end of the resonance tube is sinusoidal with a peak Mach number 0.19. Further details can be found in Mason and Miller [8] or Heffer at al [9]. The aspect ratio of each slug from the primary jet, length to diameter ratio L/D, was systematically varied from 4 to 7. This covers the range of L/D found optimal for unsteady ejectors, Mason and Miller [6].

At inlet, the geometry of the ejector-nozzle is the same as the optimal ejector reported by Mason and Miller [8]. The ratio of the ejector to primary jet diameter, D_E/D, is 2 and the ratio of the nose radius to jet diameter is, R_N/D, is 0.26. The length of the parallel part of the ejector, L/D_E, was varied from L_E/D_E = 0.8 to 4. The reason for varying this parameter was to determine how it affects the time average stagnation pressure gain and the rms unsteadiness at nozzle exit. Heffer et al. [9] reported that for unsteady ejectors the amplitude of exit unsteadiness dropped as L_E/D was raised. The distance between the primary jet and the ejector-nozzle, X/D, was varied from 1.3 to 3.3. The reason for varying this parameter was that Mason and Miller [6] reported that for unsteady ejectors there was an optimal value at X/D = 2.

One of the aims of the experiment is to measure the percentage of the exergy in the inlet flow which is retained at nozzle exit. The control volume considered is shown in figure 3. The primary jet produces a flow at state 1, and the nozzle exit flow is at state 2. The ground state for the exergy calculations, 3, is the state that which would occur if the primary jet were to mix out fully into a plenum at ambient pressure. This idealized state is the same as would occur from a steady combustor with zero pressure loss. If both the temperature of the primary jet and ambient fluid entering the control volume at inlet is constant and equal, $\dot{E}$ is given by:

$$\dot{E}_1 = \int\int \rho V_1 \left(h_1 - h_3 + T_3\left(s_1 - s_3\right)\right) dAdt = \int\int \rho V_1 T_3 \left(\Delta s_{13}\right) dAdt = \int\int \rho V_1 T_3 R \, ln\left(\frac{p_1}{p_3}\right) dAdt$$

Where h is enthalpy, T is stagnation temperature, p is stagnation pressure, ρ is density, and R is the gas constant for air. Non zero exergy flux enters the control only during outflow, time period 0 to $\tau/2$, and thus the integral can be reduced to this period. Using a truncated Taylor series to represent the natural logarithm, the inlet exergy flux, $\dot{E}$ can be written as:

$$\dot{E}_1 \approx \int_0^{\tau/2} \frac{\rho A_j V_j^3}{2} dt \tag{2}$$

It should be noted that the RHS of equation 2 represents the kinetic energy flux in the primary jet. Under the same approximations the exergy flux leaving the nozzle is given by:

$$\dot{E}_2 = \int\int \rho V_2 \left(h_2 - h_3 + T_3(s_2 - s_3)\right) dAdt = \int\int \rho V_2 T_3 (\Delta s) dAdt = \int\int \rho V_2 T_3 R \, ln\left(\frac{p_2}{p_3}\right) dAdt$$

$$\dot{E}_2 \approx \int\int \rho V_2 C p T_3 \left(\left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) dAdt \tag{3}$$

Where γ is the ratio of specific heats and Cp is the specific heat capacity at constant pressure. It should be noted that this term includes the nozzle loss as a reduction in the exergy flux of the exit flow. It should also be noted that the final term on the RHS of equation 3 represents power which could be extracted by an isentropic turbine operating between the nozzle exit and the state 3.

The ratio of equations 2 and 3 is defined as the ejector-nozzle efficiency η and is given by:

$$\eta = \frac{\dot{E}_2}{\dot{E}_1} \approx \frac{\int\int \rho V_2 C p T_3 \left(\left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) dAdt}{\int_0^{\tau/2} \frac{\rho A_j V_j{}^3}{2} dt} \tag{4}$$

Experimentally the denominator of equation 4 can be fully determined. The numerator of equation 4 is more difficult to determine as the spatial and temporal variation of V_2 and p_2 can not be easily measured. In the experiment the stagnation pressure was measured using a pitot probe. This yields a point measurement that is pneumatically averaged temporally. The time average mass flow was measured using a British standard orifice plate [10]. In practice the formula in equation 5 was used to calculate efficiency.

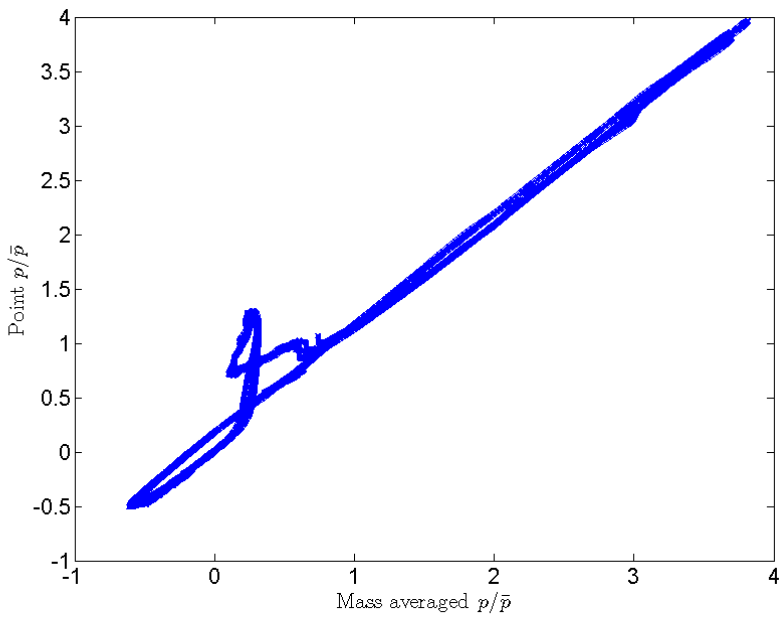


Figure 4 Comparison of mass averaged stagnation pressure and the point measurement from CFD.

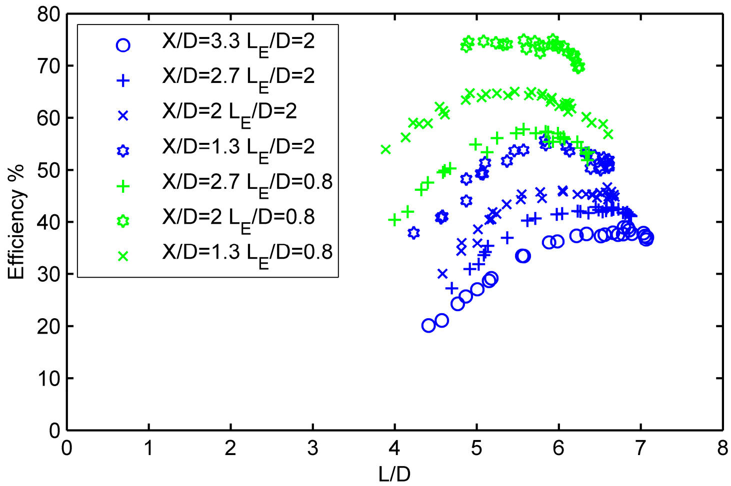


Figure 5 Efficiency of the ejector-nozzle

$$\eta = \frac{\dot{E}_2}{\dot{E}_1} \approx \frac{\bar{\dot{m}} C_P T_3 \left(\left(\frac{\bar{P}_2}{P_3} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right)}{\int_0^{\tau/2} \frac{\rho A_j V_j{}^3}{2} dt} \tag{5}$$

Where $\bar{\dot{m}}$ and $\bar{p}_2$ are the temporal average mass flow rate and centre line pressure respectively. It should be noted that because the centre line pressure was used the effect of nozzle loss on the exit exergy flux has been effectively removed. This is in practice gives a more useful measure of efficiency as a nozzle is present in both a PGC gas turbine and a normal gas turbine.

To determine whether the assumptions in the experimental measure of efficiency give realistic results a CFD solution was undertaken. To determine whether the CFD gave an accurate measure of efficiency the CFD and experimental measure of efficiency, derived by equation 5 were compared. The CFD was then used to compare the efficiency derived from equation 5 with that derived from equation 4 with the nozzle loss removed. These comparisons are shown later in the section.

The CFD solution was undertaken using Fluent 6.2.16. The solver was density-based and discretised the equations of motion to second order accuracy. The turbulence model used was the Reynolds Stresses model. The grid used was axisymmetric and contained 80,000 nodes. Experimental measurements of pressure were used at the piston face of the primary jet. The geometry, grid, boundary conditions and turbulence model were the same as those used by Heffer et al [9] to investigate unsteady ejectors. The only difference between the two geometries and their boundary conditions was the inclusion of the choked boundary condition at the exit of the nozzle. The ejector-nozzle geometry simulated had L_E/D =2 and X/D=2. The primary jet was simulated with an L/D=4.

Using the approximate measurement method, equation 5, the CFD predicts η= 31% and the experiment predicts η= 28%. This shows that the CFD and experiment are in reasonably good agreement. Using the precise measure of efficiency, equation 4 with the nozzle loss removed, the CFD predicts η= 23%. This shows that buy taking centre line time average values at nozzle exit the efficiency is over estimated by approximately 8%. The reason for this 8% over-estimation of efficiency can be seen from fig. 3. This shows time variation of the centre line pressure plotted against the pressure mass averaged over the area. For most of the period the two agree, however, for a small period the two can be seen to diverge. This is due to the vortex passing through the exit plane of the nozzle and the velocity profile changing shape.

This CFD and experimental analysis shows that the trend of the experimental efficiency should be believed but the absolute value is approximately 8% high.

III. Effect of geometry on efficiency

The effects of two geometry parameters are presented, the spacing between the primary jet exit and ejector-nozzle inlet, X/D, and the length of the ejector-nozzle, L_E/D. The first parameter is a measure of how close a combustor exit can be mounted relative to the ejector-nozzle inlet. The second

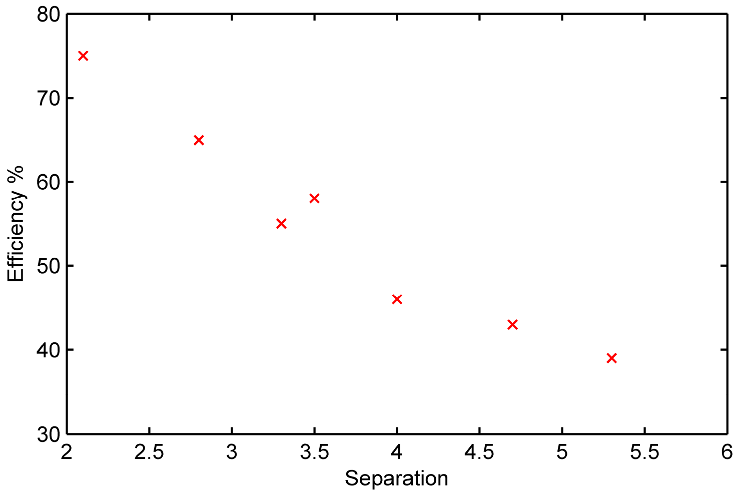


Figure 6 Variation of peak efficiency with spacing between primary jet and nozzle.

parameter is a measure of how 'compact' the ejector-nozzle can be designed. Figure 5 shows the efficiency plotted against *L/D*, the geometry of each slug of fluid ejected by the primary jet. *L/D* was chosen because it has been shown [8,10] that it is the non-dimensional parameter which fixes the operation of unsteady ejectors.

The effect of varying *X/D* from 3.3 to 1.3, at constant L_E/D=2 was to raise the efficiency. At *X/D*<1.3 the presence of the ejector-nozzle was found to affect the acoustics of the primary resonance tube reducing its performance. This implies that if an acoustically driven pressure gain combustor, such as a pulse combustor, is used then *X/D*=1.3 should be used, while if a non-acoustically driven pressure gain combustor, such as a Pulse Detonation Engine, is used then *X/D*<1.3 may give even higher efficiencies.

The effect of reducing L_E/D from 2 to 0.8 was to raise efficiency. The peak efficiency measured was 75% and was obtained at *X/D*=1.3 and L_E/D=0.8 while operating the primary jet with an *L/D* of 5.

Reducing either *X/D* or L_E/D thus increases efficiency. This implies that if the design aim is to maximize efficiency then the length from the exit of the primary jet to the throat of the nozzle should be minimized. Fig. 6 shows that the peak efficiency varies almost linearly with $(X+L_E)/D$. The cause of this variation is not understood; however, it is likely that by reducing $(X+L_E)/D$ the total dissipation of mechanical energy has been minimized. The peak efficiency of 75% is much higher that the 32% reported by Heffer et al. [9] for the unsteady ejector on which its is based. One possible cause of this difference is that in the ejector-nozzle the static pressure decreases in the region where the mixing process occurs. Denton [11] showed that if the pressure is reduced before mixing occurs that the total entropy created in the mixing process is reduced. This is because when a shear layer is subjected to a favorable streamwise pressure gradient, the transverse velocity gradient, *dV/dy*, is reduced. This is because, the slow moving fluid resides in the favorable pressure gradient for longer than the fast

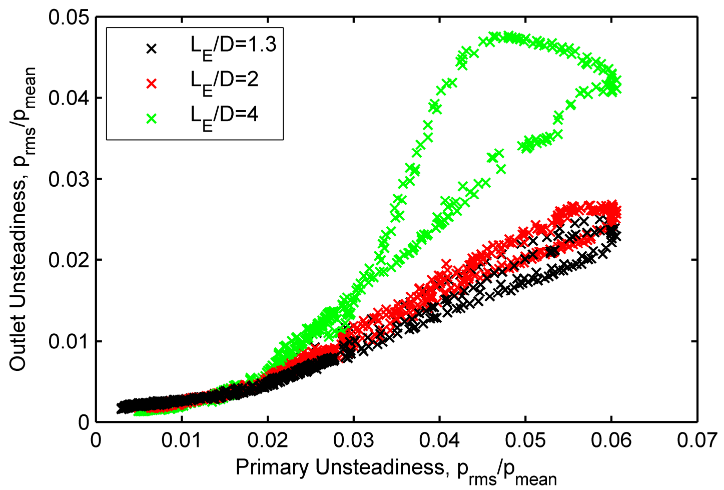


Figure 7 Unsteadiness at exit from the nozzle compared with unsteadiness in the primary jet

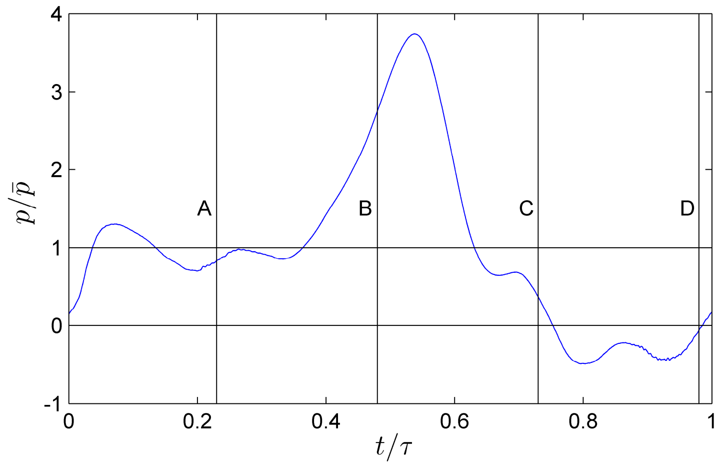


Figure 8 Stagnation pressure variation over the cycle

moving fluid, therefore the slower fluid has a larger increase in velocity and so the velocities of the two fluid elements become closer. The entropy generation rate in the shear is proportional to $\mu_{eff}(dV/dy)^2$ and therefore the total increase in entropy will be is reduced.

IV. The effect of geometry of nozzle exit unsteadiness

The exit unsteadiness was measured with X/D=2 and L_E/D=1.3, 2 and 4. Figure 7 shows the rms unsteady stagnation pressure in the exit nozzle plotted against the rms unsteady stagnation pressure in the primary jet. For short ejector nozzles, L/D_E=1.3-2, the exit unsteadiness is approximately 1/2-1/3 of that in the inlet. However as the ejector is made longer, L_E/D=4, the graphs show that the unsteadiness increases dramatically. This is in contrast to the results of Heffer et al [9], who found that as an ejector becomes longer the rms exit unsteadiness drops. The rise in unsteadiness at L_E/D=4 is thought to be due to the ejector-nozzle resonating. At an L_E/D=4 the ejector appears to be too short to resonate. However, the presence of the choked nozzle alters the local wave propagation speed. This can significantly reduce the length of component at which resonance occurs. Marble and Candle [11] showed that the component length at which acoustic effects become important is given by $(c\text{-}u)/f$, where c is the speed of sound and u is the flow velocity and f the frequency.

It is also noted that, for each geometry, the results do not collapse neatly onto one line. The reason for this is not well understood but it may be due to the primary jet changing in frequency over its operating range.

V. Structure of nozzle exit flow

The time-resolve nozzle exit stagnation pressure plotted over one period is shown in fig 8. Four time snapshots (A-D) of stagnation pressure coefficient, $(p_0\text{-}p_{atm})/(1/2\rho V_j^2)$, and entropy are plotted in fig 9. The time instants at which these occur are marked on fig 8 (A-D).

The stagnation pressure gain in fig. 8 is positive for 70% of the cycle. This is in contrast to the primary jet which has a raised stagnation pressure for only 50% of the cycle. In time snapshot A the primary jet has emerged from the end of the resonance tube and the shear layer formed between the fluid and the pipe roll up into a vortex ring. In time snapshot B the vortex ring has convected downstream towards the nose of the ejector nozzle. As the vortex reaches the ejector inlet it compresses the fluid within the ejector raising its stagnation pressure. This is a similar mechanism observed by Heffer et al [9] in unsteady ejectors. The peak value of stagnation pressure is 3.5 times the mean pressure gain. In time snapshot C the vortex ring has moved into the ejector-nozzle and the stagnation pressure starts to drop. In time snapshot D the vortex ring starts to move through the throat of the nozzle. Between time instances C and D the stagnation pressure falls below atmospheric, it is likely that this is due to a pressure wave as the high pressure in B ‘over corrects’. The results show that the unsteadiness at ejector-nozzle exit is much lower than at the exit of the primary jet.

It should be noted that in this paper the primary jet has been simulated using a cold jet. In a real combustor there will be a temperature difference between the primary jet and the surrounding fluid. Marble and Candle [11] analyzed the effect of temperature non-uniformities convecting through a choked nozzle. They indicate that a temperature non-uniformity of 0.5 of the mean level will produce pressure waves that propagate both upstream and downstream of the nozzle with a pressure ratio of 0.05 and 0.02 respectfully. In a real machine these effects should also be considered.

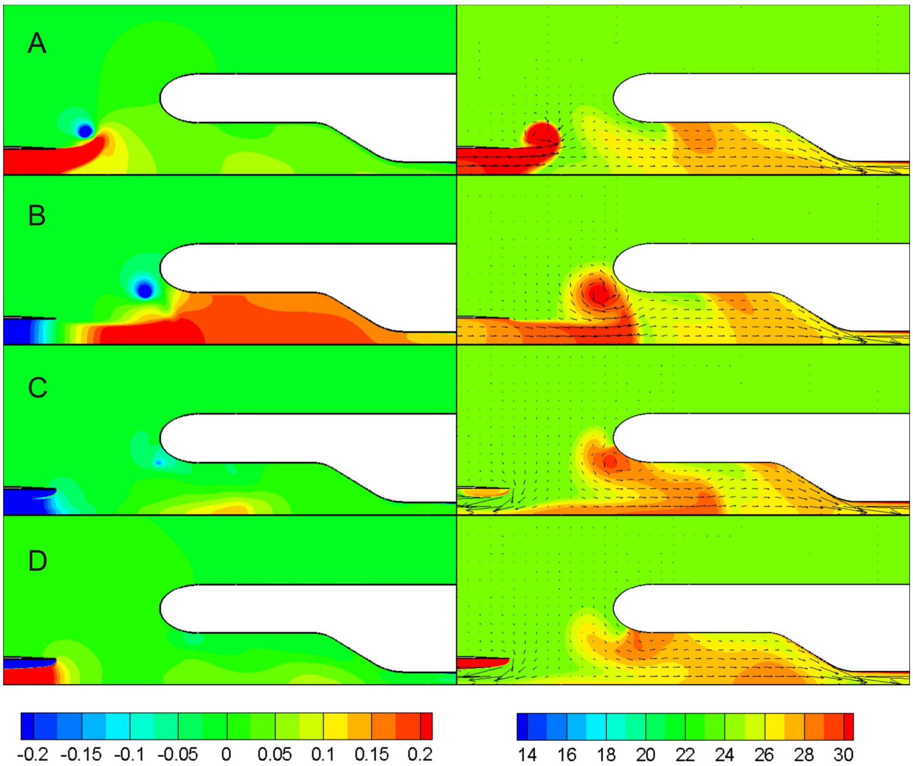
VI. Conclusions

A combined ejector-nozzle has been designed to couple the unsteady flow from a pressure gain combustor to the turbine rotor. This was found to have a high efficiency, up to 67% of the additional exergy in the exit flow from a pressure gain combustor to the inlet of the turbine rotor. This is superior to designs using a simple ejector. Ejector-nozzles should be designed to be as short as possible and the primary jet should be placed as close as possible to the ejector-nozzle provided that there is no significant interaction with the acoustics of the primary jet. This minimizes dissipation of the exergy in the jet before it is accelerated in the nozzle. Mixing in an accelerating flow reduces the dissipation and raises the efficiency. It is also found that short ejector-nozzles tend to have lower values of unsteadiness than longer nozzles. For an ejector-nozzle coupled to a pulse-combustor it is expected that the exit rms unsteady pressure will be 1/3 of the value of that occurs at combustor exit. The peak value of the ejector-nozzle exit stagnation pressure is 3.5 times the time-mean stagnation pressure gain. However in a real application the propagation of fluid with a non-uniform stagnation temperature through the nozzle will cause additional unsteady pressure variations.

An ejector-vane designed according to the finding of this paper would, in addition to having a high efficiency and low exit unsteadiness, have a relatively small wetted area and thus would have lower cooling requirements and incur lower frictional losses than previous designs which have appeared in the literature [2],[3],[4],[5].

VII. Acknowledgements

The authors would like to thank Rolls-Royce PLC for their funding and support, particularly Sam Mason for many enthusiastic discussions. The authors are also grateful to EPSRC for their assistance.



Pressure Coefficient **Entropy**

Figure 9 Time instances from the CFD

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