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A NEW VORTEX TRIODE FOR PROPORTIONAL CONTROL

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SUMMARY The vortex triode is a versatile control device and in the experimental investigation which has just been completed a successful attempt has been made to improve greatly its control characteristic by means of design development. It is thought that the proportional modulation which has been achieved now offers good prospects for extending the use of vortex triodes to a wide range of applications.

1. INTRODUCTION

The configuration of a conventional triode is shown in Fig. 1 and comprises a cylindrical chamber with three ports. The primary flow enters the chamber in a radial direction and, in the absence of control flow, passes directly to the centre of the chamber from which it discharges with little resistance through the axial exit tube. Control is achieved without the use of mechanical moving parts by the high pressure injection of a small quantity of secondary fluid through the small tangential port causing the primary jet to be deflected, setting up a swirling action in the chamber and greatly increasing the flow resistance. If sufficient secondary fluid is injected, the primary flow can be shut off completely and in this event the only fluid which continues to discharge from the device is the secondary fluid itself.

The throttling effect occurs because the angular momentum of the swirling flow is largely conserved as the exit port is approached causing a progressive increase in the circumferential component of the velocity. There is a corresponding increase in the centrifugal force and it is this which opposes the approach to the port.

A typical characteristic of the conventional triode is shown in Fig. 2, indicating that the shape is far from ideal for control purposes. It reveals that a significant amount of secondary fluid has to be injected before any effect is felt and, when it eventually occurs, the result is dramatic with an almost step decrease in primary flow accompanying further injection.

In the present work the objective has been to improve greatly the linearity of the control characteristic while retaining the facility to shut off the primary flow. A comprehensive programme of tests has been undertaken in which a number of geometrical parameters were varied, including multiple supply and control ports of different sizes and in different locations and it will be shown that an unexpected configuration was eventually discovered which yielded the almost ideal characteristic shown in Fig. 2.

Guidance regarding which parameters were likely to be most important was provided by the results of a previous investigation (1979) in which the geometry and performance of **axial** vortex valves had been studied. In the axial configuration the primary flow no longer enters the valve in a radial direction through discrete ports but enters uniformly in an axial direction through an annular gap between the outer wall of the vortex chamber and a centre-body. The principle of operation is the same however, in that the primary flow is deflected by the tangentially injected control flow and passes to the central exit port with progressively increasing swirl.

For axial flow, it was found that when multiple control ports were incorporated there was no improvement in performance to compensate for the increased complexity of the design. Regarding their **size**, this governed the amount of injected power required to achieve a particular degree of flow control and a clearly defined optimum diameter was found to exist for which the power was a minimum. On the other hand the corresponding amount of momentum requiring to be injected was independent of the size of the control port(s). Other geometrical parameters which were examined (such as the size and shape of the centre-body) were found to have very little effect on performance.

Other previous investigations of confined swirling flows have been comprehensively reviewed by Lewellen (1971) and here it suffices to mention that several notable papers on vortex triodes were presented at the Cranfield Fluidic Conferences. In none of these cases however, was an optimisation programme like the present one undertaken.

2. THE TEST RIG

The rig was essentially the same as that used for the axial valves and, since a photograph and description were provided in Ref. 1, it is sufficient here merely to provide descriptions of the radial valve and the rig modifications.

A further objective of the experimentation was to gain an improved insight into the nature of the flow within the valve and to this end a relatively large model was employed in which the vortex chamber and exit port had diameters of 100 mm and 25 mm respectively. Other features were that the model was made from perspex for flow visualisation and a large number of tappings were set into the base (See Fig. 3) allowing the pressure field in the chamber to be determined for various flow situations. The radial supply ports were also of 25 mm dia., up to four being available equally spaced around the periphery, and associated with each one was a control port the size of which could be varied from 1.6 to 5 mm dia. by means of a number of interchangeable inserts. Other possible arrangements included such geometrical configurations as the use of one of the supply ports in conjunction with one or multiple **distant** control ports, so that altogether more than a hundred combinations were available.

Water was used as the working fluid and from the valve it passed vertically upwards to a constant head tank via a diffuser which enabled some of the discharging velocity head to be recovered and the overall efficiency improved. The supply and control circuits were independent but similar, each incorporating a centrifugal pump, a manifold settling chamber and needle valves for fine control of flow-rate. The major difference, of course, was in the levels of pressure and flow-rate required, the maxima in the supply circuit being about 110 kPa and 2.5 l/s, and those in the control circuit being about 400 kPa and 0.15 l/s respectively. Note was taken of any tendency towards instability in the valve's behaviour but for the most part attention was confined to steady-state performance for which it was sufficient to use orifice plates and standard manometers for flow measurement and Bourdon gauges for the measurement of pressure.

The smallest dimension in the path of the swirling flow was of course the exit port diameter of 25 mm and the local mean velocity was 1.7 and 5.1 m/s for supply pressures of 20 and 110 kPa respectively and for zero control flow. The corresponding values of Reynolds Number were 39,000 and 120,000.

3. DISCUSSION OF RESULTS

Mention has already been made of the fact that more than a hundred geometrical combinations were available and indeed almost all were tested even though intuitively one might have expected a poor valve performance in many cases. (To permit a preliminary comparison to be made of the merits of the different configurations the control characteristic was determined for a particular pressure difference across the valve of 30 kPa). As intimated earlier, it transpires from the results that the optimum is an unexpected one and if the blanket coverage of all the possibilities had not been undertaken the desired geometry might not have been discovered.

In contrast to the comprehensiveness of the experimentation, space limitations permit only a small selection of results to be presented in this paper, in each case the total flow Q_o through the valve being plotted against the control flow Q_c (both in litres/second). The optimum configuration will be described first and data will then be given to indicate the nature and magnitude of the loss of performance when the geometry departs from it.

As shown in Fig. 4, the optimum design has one primary port and two control ports, the latter being 2 mm dia. and displaced from the primary by 90° and 180°.

The control characteristic associated with it, for the constant supply pressure of 30 kPa, is shown in Figs. 5,6,7 and 8 and serves as a comparitor for the other results. Clearly it is virtually ideal for control purposes being linear throughout the range from Q_o = 1.057 l/s when there is no control flow to 0.139 l/s when the primary flow is completely shut off. Other features are the absence of significant scatter in the measured data and the fact that even the smallest injection of secondary fluid has an immediate modulating effect on Q_p. The magnitude of the Turn Down Ratio is 7.6.

Space does not permit the inclusion of graphs to illustrate the effects of multiple **supply** ports (two, three or four) but an obvious consequence of increasing the total port cross-sectional area is to increase the primary flow-rate for zero control flow. More interesting however is the loss of linearity of the control characteristic for all of the multiple supply port configurations. As the number of ports increases, the shape of the curve becomes progressively more like that for a conventional vortex triode with the loss of linearity being particularly conspicuous at the beginning and the end of the operating range. Another feature is that as the shut-off condition is approached, the characteristics for the 2,3 and 4-port designs tend to merge together. Of greatest importance however is the clear result that a single port gives the best performance, and if one were to opt for the expense and complexity of a multiple port arrangement it would be counter-productive.

The effects of multiple **control** ports is shown in Fig. 5, the characteristic for two control ports being the optimum. The curve for a single control port is clearly less satisfactory because of the considerable scatter of the results and the loss of linearity manifest as a progressively decreasing gradient although a compensating benefit would be the smaller secondary flow requirement to reduce the primary flow by a certain amount. Linearity is also conspicuously absent for the designs having three and four control ports but in contrast the gradient now rises progressively as substantial quantities of secondary fluid have to be injected before any diminution of Q_p is observed. Summarising, the optimum configuration is markedly superior to the others that were considered.

Regarding the **size** of the twin control ports, Fig. 6 again shows that the optimum diameter of 2 mm clearly gives the best performance for control purposes. This is so with respect to both the proportionality of the flow modulation and the fact that secondary injection has an immediate effect on reducing Q_o. Indeed the departure from these desirable features becomes steadily more marked as the port size d_c increases, no doubt due to a corresponding decrease in momentum for a given amount of control fluid. [The momentum is given approximately by $\rho Q_c \frac{Q_c}{\pi d_c{}^2/4}$ (where ρ is the density) and is therefore inversely proportional to $d_c{}^2$.] As mentioned in the description of the test rig, other control port sizes were tested, down to d_c = 1.6 mm, but no further improvement in performance was detected.

Frequent reference has been made to the degree of linearity of the control characteristic which is clearly important but it is also useful to quantify the performance of the valve for a condition where flow control is partially completed rather than where the primary flow has been stopped. In the present programme, the particular degree of flow control which has been examined has been the case where the supply flow has been reduced to **half** of its initial value. Corresponding values of the control flow rate and the control pressure P were measured and used to calculate the momentum and the control power $Q_c P_c$, the values being as follows for the data presented in Fig. 6.

d_c (mm)	Q_c (l/s)	P_c (kPa)	$Q_c P_c$	Q_c/d_c
2	0.068	118	8.02	0.034
3	0.115	64.7	7.44	0.038
4	0.145	50.9	7.38	0.036
5	0.189	44.3	8.37	0.038

It was shown earlier that Q_c/d_c is proportional to √(momentum) and the table reveals further that the ratio has approximately the same magnitude for each of the four diameters. This indicates that to reduce the primary flow by a certain fraction there is a particular momentum requirement which does not depend on the size of the control ports. Another feature of the tabulated results is that the product $Q_c P_c$ passes through a minimum value when d_c is about 3 to 4 mm implying that if the optimisation criterion had been to minimise the power required to attain the specified degree of flow control (rather than to have proportional modulation etc.) control ports of this size would have been selected. However, the power requirement for the chosen configuration with d_c = 2 mm is only about 9% higher than the minimum and this is considered to be a small penalty which only partially off-sets the many substantial benefits. One of these benefits which has not been mentioned specifically before but which is evident in the table is that since $Q_c \propto d_c$ the secondary flow requirement itself is very small for the chosen size of control port.

Another possible departure from the optimum configuration concerns the **location** of the twin 2 mm control ports relative to the single supply port and, as Figs. 7A - 7F show, there is a total of six combinations. Fig. 7A relates to the optimum and has the best characteristic with respect to both linearity and low scatter, although 7E is inferior only by a small margin. It is thought that the absence of scatter for the optimum geometry may occur because the separation distance between the control ports and the supply port serves as a settling length for the secondary jets. Any unsteadiness which might be present initially is perhaps dissipated as the secondary jets develop allowing them ultimately to mix more smoothly (but with largely undiminished momentum) with the primary jet.

All of the results presented hitherto have been for a supply pressure of 30 kPa and, referring to Fig. 8, control characteristics are now considered for other pressures in the range 20 to 110 kPa applied to the optimum configuration. The figure shows that unfortunately the proportional modulation is confined to a narrow range in the vicinity of 30 kPa. If the pressure is lower, the curve is initially steep but becomes less so, whereas if the pressure is higher, the steepness of the curve increases as Q also rises. It therefore appears that the linearity at 30 kPa is a Reynolds Number effect and a manifestation of the part played by the fluid viscosity in the mixing process and elsewhere.

Having determined the optimum configuration and the conditions required for proportional control, attention was then focussed on the internal pressure distribution with the objective of gaining an improved insight into the nature of the flow. Experiments were performed at three levels of supply pressure (20, 30 and 50 kPa) and at three degrees of flow cut-off (zero, 50 and 100%). For complete cut-off, the pressure distribution tends to that of a forced vortex and the dimensionless pressure falls rapidly from the outer wall of the chamber to the central drain. Also, there are three distinct curves, one for each value of the supply pressure. In contrast the supply pressure appears not to affect the shape of the distribution profile for 50% cut-off and a single curve with little scatter was obtained for the three pressures. A feature of it is its relative flatness, particularly in the vicinity of the outer wall, where the primary and secondary jets mix together, suggesting that the mixing process does not itself incur any significant change of pressure. In fact the pressure coefficient $C_p[= (P_{12}- P)/(P_{12}- P_1)$ with subscripts defined in Fig. 3] rises to only 0.2 at the drain periphery, indicating that merely 20% of the overall pressure difference occurs upstream of the drain while the remaining 80% occurs between tappings 3 and 1. It can also be shown that not surprisingly the pressure distribution for 50% cut-off differs substantially from that for a free vortex, suggesting that as the swirling flow approaches the drain both the total pressure and the angular momentum diminish significantly.

Although the pressure drop between r_{12} and r_3 is very small for zero injection, due to the flow in the vortex chamber being radially inward and without swirl, in the circuit as a whole the pressure distribution is useful as a comparator for the corresponding distribution for partial cut-off at the same supply pressure. The calculated values given below provide such a comparison for a typical case and relate to zero and 50% cut-off, each for a constant supply pressure of 50 kPa. In brackets are the various pressure drops expressed as percentages of the overall pressure difference $P_p - P_D$ between the supply and downstream tanks.

cutoff	(P_p-P_{12})kPa	$(P_{12}-P_3)$kPa	(P_3-P_D)kPa	Q_o l/s
zero	33.25 (79.3%)	0.74 (1.8%)	7.91 (18.9%)	1.61
50%	10.75 (25.2%)	9.02 (21.2%)	22.87 (53.6%)	0.905

The comparison reveals that injection dramatically changes the pressure distribution in the circuit, for example by producing an unexpectedly large pressure drop in the drain (53.6%). Simultaneously the pressure drop in the vortex chamber rises from its initially very small value to 21.2% but is still the smallest of the three parts of the overall pressure difference. Meanwhile the total flow-rate falls from 1.61 to 0.905 l/s so that, perhaps contrary to expectations, the increase in pressure difference across the chamber and its drain is associated with a reduction of flow-rate rather than an increase. Of course, the inverse nature of the relationship occurs because of the swirl, but it is also indicative of the complexity of the flow.

The complexity is also believed to be manifest as a complicated flow pattern in the vortex chamber although further experiments require to be undertaken to substantiate this. In previous work on cyclone separators, which are in some respects similar to vortex triodes, it has been found that most of the discharging flow, passes through the boundary layer which forms on the wall of the device whereas the bulk of the fluid in the chamber tends only to recirculate. Furthermore, the boundary layer effect causes the heavy particles in the separating mixture to pass to the throat whereas one might have expected the centrifugal forces on the particles to cause them to congregate at the larger end of the cyclone chamber. Another instance of swirling flows incurring substantial and complicated secondary effects.

4. CONCLUDING REMARKS

The design of the vortex triode illustrated in Fig. 4 has been shown to be the optimum of the large number of possible configurations studied in the present programme. It is almost ideal for most control purposes offering proportional modulation throughout the operating range, the facility to stop the supply flow completely, and a consistent performance as evidenced by the absence of scatter in the measured data.

There is not a unique optimum for all situations however, and the best design for one application may differ from that for another. For example, if a high turn down ratio is more important than a linear control characteristic, the present ratio of about 7.6 could be improved upon. Similarly the fact that linearity is limited to a fairly narrow range of Reynolds Numbers could result in the design being unsuitable for some applications.

On the other hand the authors believe their discovery to be a significant one and to offer the prospect of the use of vortex triodes being extended to a wide range of applications where proportional flow control is important and where such control is sought without the use of moving mechanical parts.

5. REFERENCES

DUGGINS, R.K. and FRITH, P.C.W. (1979) The geometry and performance of vortex valves. Conf. on Control Eng., Melbourne, IE Aust. 75-79.

LEWELLEN, W.S. (1971) A review of confined vortex flows. NASA CR-1772.

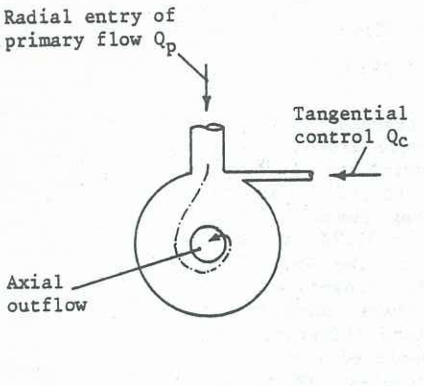


Fig. 1. Conventional triode configuration.

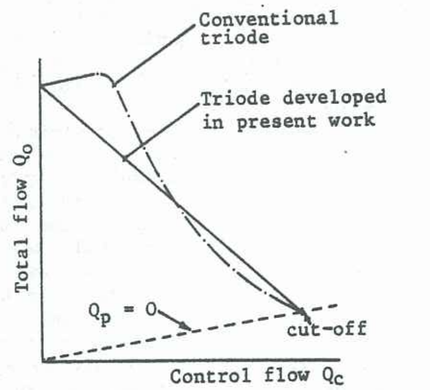


Fig.2. Control characteristics.

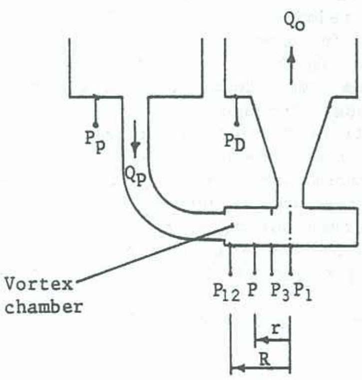


Fig.3. Location of pressure trappings

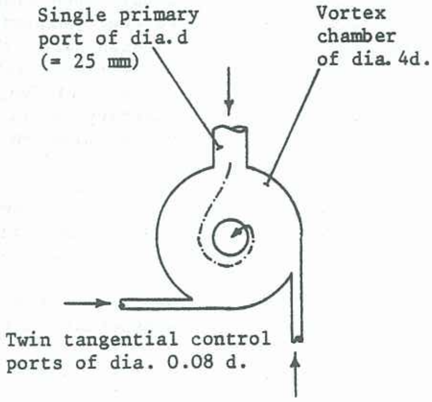
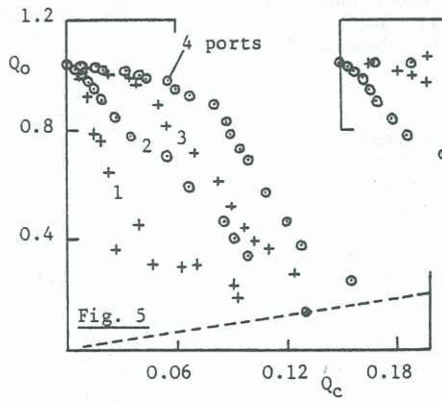
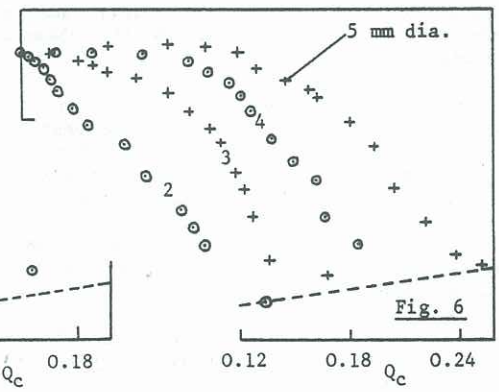


Fig.4. Optimum configuration





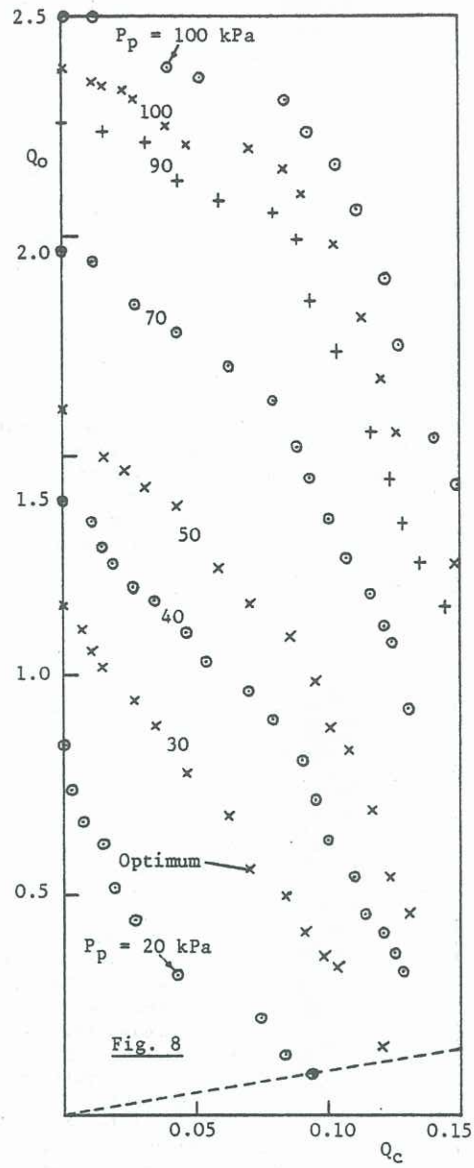


Fig.5. Effects of departure from the optimum configuration : Multiple control ports.

Fig.6. As above but twin control ports of different diameters.

Fig.8. Effects of supply pressure for the optimum configuration.

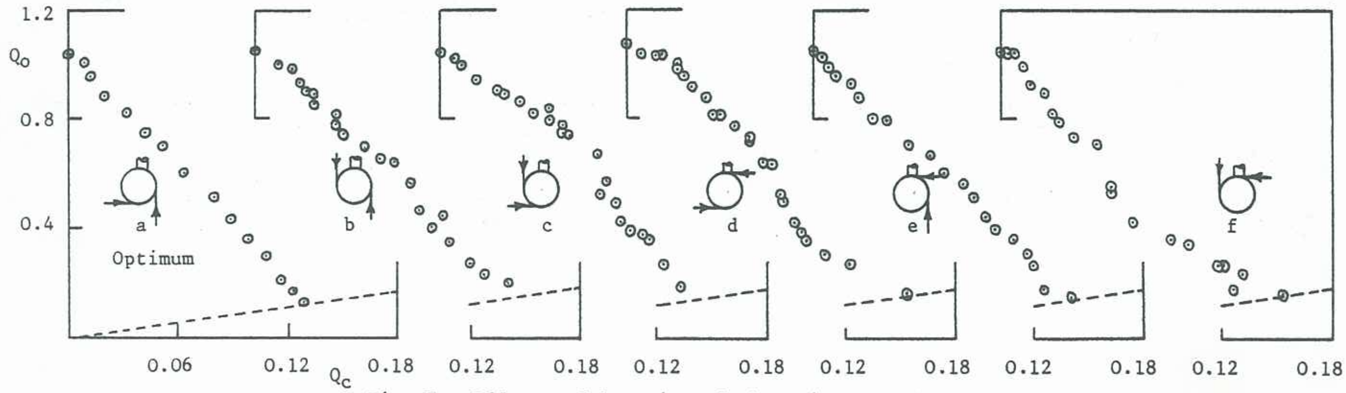


Fig. 7. Effects of location of the twin control ports