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ACTIVE CONTROL OF SURGE - DEVELOPMENT OF AN ACTUATOR

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1. INTRODUCTION

Pumping systems can exhibit many different forms of instability. One of these, found in compression systems, is surge, which can result in large scale axial velocity fluctuations in the compressor when fully developed. The surge limit is a constraint on the performance of compression systems, and significant gains could be realised if it were relaxed. The most promising way of achieving this is via active control, as has already been demonstrated by Huang and Ffowcs Williams [1].

A simple incompressible model, after the manner of Epstein et al [2], will illustrate the principles involved. In the following analysis, upper case characters will be used to denote steady state levels, and lower case characters to denote small deviations from those levels. Numerical suffices refer to the stations indicated in Figure 1, and a ' superscript represents differentiation of a function with respect to its argument.

Consider a compression system consisting of a compressor of length L and cross-sectional area A, a plenum of volume V and a downstream throttle also of cross-sectional area A, as illustrated in Figure 1. The steady state compressor characteristic is given by $(P_2 - P_{01})/\rho U^2 = \Psi(\Phi_c)$, where U is the compressor blade speed, ρ is the gas density, P and P_0 are steady state static and total pressures respectively and Φ_c is the compressor flow function, axial flow velocity divided by blade speed. The steady state throttle characteristic is taken to be parabolic, with $(P_2 - P_3)/\rho U^2 = \frac{1}{2}K_T\Phi_T^2$, where Φ_T is the throttle flow function and K_T is a constant.

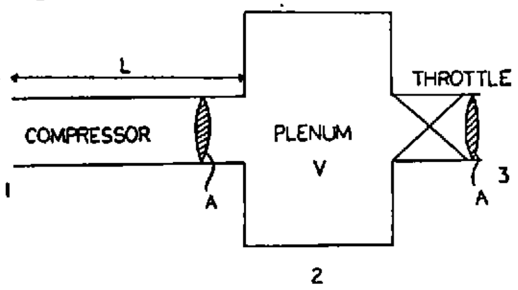

Figure 1 Compression System Model

We note first that, as the system draws air from and exhausts to atmospheric conditions, $p_{01} = p_3 = 0$. Also, as there is no steady state mass storage in the plenum, and the throttle and compressor ducts are of the same area, $\Phi_c = \Phi_T = \Phi$. We may now write down the unsteady flow equations defining the system.

The compressor pressure rise is taken to have a steady state component, a quasi-steady component resulting from the (small) change in flow function, and a component arising from the momentum equation applied to the fluid in the duct:

$$((P_2 + p_2) - P_{01})/\rho U^2 = \Psi(\Phi) + \Psi'(\Phi).\phi_c - (\lambda L/U)\frac{d\phi_c}{dt} \qquad \text{(i)}$$

(λ constant)

The gas in the plenum is assumed to undergo small isentropic pressure and density fluctuations:

$$\frac{dp_2}{dt} = a^2\frac{d\rho_2}{dt} \qquad \text{(ii)}$$

where a is the speed of sound, and (by mass conservation)

$$\frac{d\rho_2}{dt} = \frac{\rho AU}{V}(\phi_c - \phi_T) \qquad \text{(iii)}$$

Finally, the throttle is modelled as behaving in a quasi-steady manner:

$$(P_2 + p_2 - P_3)/\rho U^2 = \tfrac{1}{2}K_T\Phi^2 + K_T\Phi.\phi_T \qquad \text{(iv)}$$

The steady state levels are subtracted from either side of equations (i) and (iv), equations (ii) and (iii) are combined and the following non-dimensional variables are defined: $\pi_2 = p_2/\rho U^2$, $\tau = Ut/L$, $B^2 = (U^2/a^2)(V/LA)$.

The resulting equations are then:

$$\pi_2 = \Psi'(\Phi).\phi_c - \lambda\frac{d\phi_c}{d\tau} \qquad \text{(v)}$$

$$\frac{d\pi_2}{d\tau} = (1/B^2)\left[\phi_c - \phi_T\right] \qquad \text{(vi)}$$

$$\pi_2 = K_T\Phi\phi_T \qquad \text{(vii)}$$

Eliminating ϕ_c, ϕ_T;

$$\frac{d^2\pi_2}{d\tau^2} + \left[\frac{1}{B^2}\cdot\frac{1}{K_T\Phi} - \frac{\Psi'(\Phi)}{\lambda}\right]\frac{d\pi_2}{d\tau} + \frac{1}{\lambda B^2}\left[1 - \frac{\Psi'(\Phi)}{K_T\Phi}\right]\pi_2 = 0 \qquad \text{(viii)}$$

Applying the standard stability criterion, that all the terms must be positive, we see that the stability condition is:

$$\Psi'(\Phi) < \lambda/(B^2 K_T \Phi) \tag{ix}$$

and

$$\Psi'(\Phi) < K_T \Phi \tag{x}$$

Normally K_T is large, so these equations imply that $\Psi'(\Phi)$ should be negative for stability. Now assume that a controller is introduced that can perturb the plenum pressure such that

$$\frac{d\pi_2}{d\tau} = \frac{-K_A}{B^2}\pi_2 \tag{xi}$$

Since the system is linear, we can superpose this effect, so that equation (vi) becomes

$$\frac{d\pi_2}{d\tau} = (1/B^2)\left[\phi_c - \phi_T - K_A\pi_2\right] \tag{xii}$$

with the result that the system equation is now:

$$\frac{d^2\pi_2}{d\tau^2} + \left[\frac{1}{B^2}\left(\frac{1}{K_T\Phi} + K_A\right) - \frac{\Psi'(\Phi)}{\lambda}\right] d\pi_2/d\tau + \frac{1}{\lambda B^2}\left[1 - \frac{\Psi'(\Phi)}{K_T\Phi}\right]\pi_2 = 0 \tag{xiii}$$

and the stability condition on $\Psi'(\Phi)$ is relaxed. Effectively, the use of active control increases the damping in the system, thereby suppressing the instability. Note that, in order to prevent the instability growing, high control levels are not required. All that the actuator need do is produce pressure fluctuations of the same order as those that are detected at the onset of surge.

2. ACTUATION METHODS

There are many ways to perturb a compression system, for example modulating the inlet air flow, varying the fuel flow, or, as in the above example, fluctuating the plenum pressure. Of these strategies, the latter is intuitively the most promising, given that the plenum acts as an energy store for the instability. However, when the plenum is a combustion chamber, there are currently no readily available methods of implementing it. The high pressures involved militate against volume fluctuation and the adverse temperature conditions necessitate a remote form of actuation. Thus it is not possible to employ loudspeakers, the most commonly used actuator in active control at present. However, an established method for affecting combustion chamber steady state pressure does exist - bleeding off a small proportion of the mass flow at compressor delivery. If this bleed could be modulated, fluctuations in combustion chamber pressure would result, at the cost of the loss (to the turbine) of the mean bleed flow. In order to pursue this idea, it was first necessary to design a flow modulating actuator, which is the subject of this paper.

3. DESIGN OF THE ACTUATOR

Devices for imposing a significant modulation on a mean flow already exist - for example the electro-pneumatic transducers (sic) used for high intensity acoustic testing. Unfortunately, such equipment invariably has an electrical coil situated in, and cooled by, the gas flow. This is clearly unsuitable for modulating a combustion chamber bleed. However, the principle on which these devices work, that of modulating the flow area at a point where the flow is choked, is the most obvious means of introducing fluctuations. This principle, and the necessity of remote actuation led to the design shown in Figure 2.

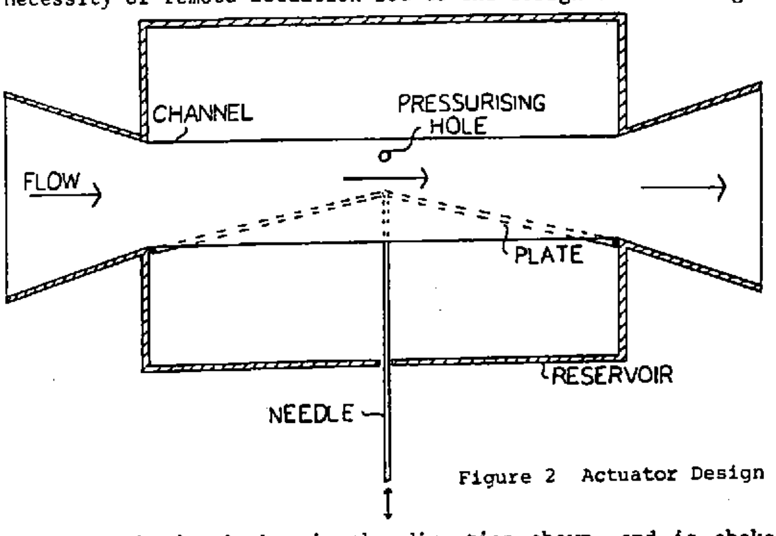

Figure 2 Actuator Design

Flow passes through the device in the direction shown, and is choked at the point of minimum area. This area may be altered by moving the hinged plates up or down inside the U-sectioned channel, and this is achieved by means of a mechanical shaker which drives the plates via a needle. In order to ensure that the steady force on the plates is minimised, the channel is surrounded by a cylindrical reservoir which is pressured to the mean pressure of the flow passing through the channel. This mean value always corresponds to the pressure at the throat, as the actuator is designed so that the supersonic/subsonic shock wave occurs downstream of the channel. It is therefore possible to pressurise the reservoir by allowing flow to escape from two holes in the side of the channel at the throat. After start up, this flow will be small, as the only loss of gas from the reservoir is through the (small) clearance between the driving needle and the cylinder wall. The use of the reservoir also has the advantage that no seal is required between the moving plates and the walls of the channel.

4. TESTING AND RESULTS

The initial testing of the device was carried out using sine wave inputs to the shaker, for two reasons, namely:

(i) When testing at a single frequency it is easier to spot system non-linearity than when using a broadband input signal;

(ii) The flow measurement was likely to contain a certain amount of turbulent noise, so that spreading input power over a band of frequencies might not result in a response that was distinguishable from the noise.

The experimental arrangement is shown in Figure 3. Air at an absolute pressure of up to 2.7 atmospheres was supplied by a compressor via a plenum chamber, and passed over a hot wire before entering the inlet nozzle. An accelerometer on the shaker provided a displacement (i.e. throat area variation) signal, and the shaker coil current was also measured.

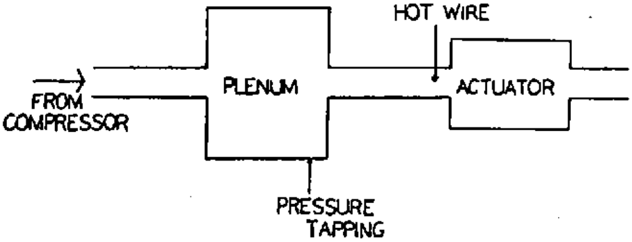

Figure 3 Experimental Arrangement

A typical set of signals taken at 40Hz is shown in Figure 4, with the trace from the hot wire linearised. Although, as expected, the flow trace shows a certain amount of noise, it is clearly being modulated by the varying throat area. The corresponding spectra for these traces are shown in Figure 5. If the spectra are taken before linearising the hot wire data, the peak on the hot wire spectrum at around 150Hz coincides exactly with those at the same frequency on the current and displacement spectra. It is therefore likely that this is noise from the equipment used to record or log the signals, rather than a genuine peak in the trace. Thus it can be seen that the system is linear in that no significant higher harmonics of the exciting frequency appear in the velocity trace.

Finally it is possible to plot an approximate transfer function between velocity fluctuation and throat area variation from the series of discrete frequency readings taken. This is done in Figure 6, for two different upstream stagnation pressures. The velocity and throat area fluctuations were calculated as fractions of the mean values. As steady, one-dimensional, compressible flow theory shows that mass flow is directly proportional to throat area, this plotting method would give a transfer function of 1 if the process were quasi-steady.

Both transfer functions depart increasingly from the quasi-steady case as the frequency increases, as would be expected. However, at any given frequency, this departure is smaller for the lower stagnation pressure case, indicating that the magnitude of the mean mass flow is significant, as well as the modulating frequency. There is also an interesting feature common to both transfer functions - three apparent anti-resonances at approximately 20, 80 and 120Hz. Whether these are a consequence of the testing arrangement (for example the plenum and piping acting as a Helmholtz resonator) or are characteristics of the actuator is not yet clear.

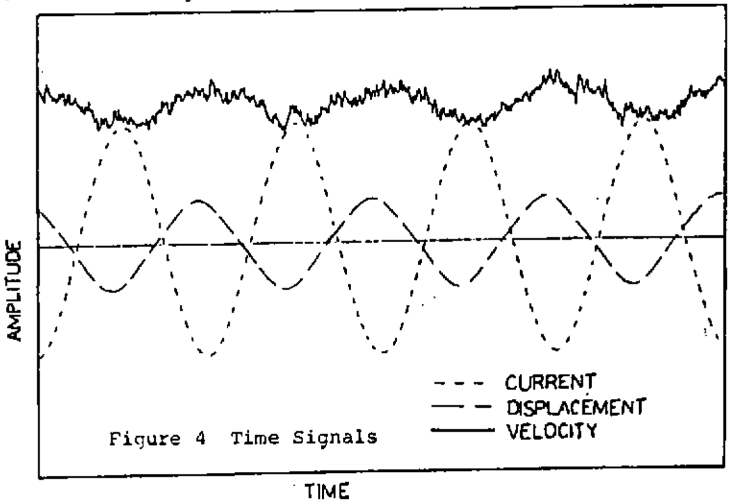

Figure 4 Time Signals

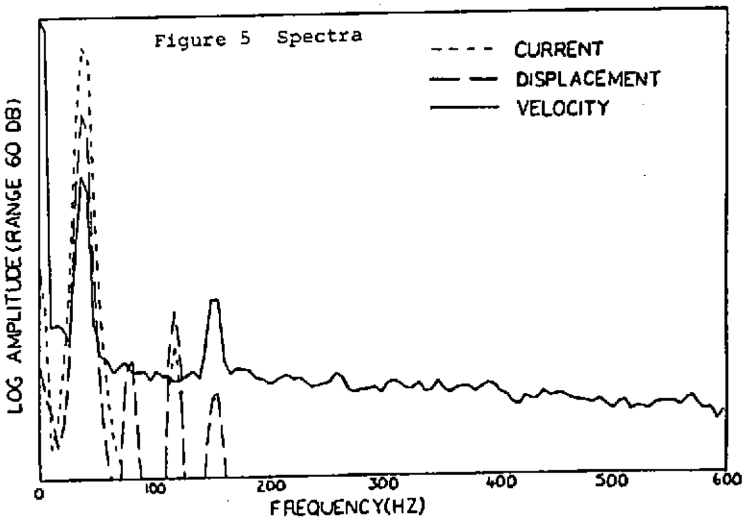

Figure 5 Spectra

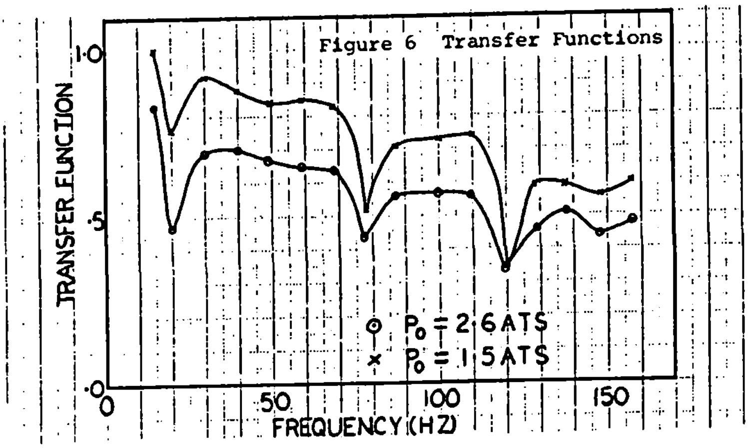

Figure 6 Transfer Functions

5. CONCLUSION

As can be seen from the results in section 4, an actuator as described here may be used to create significant percentage velocity fluctuations superimposed on a mean flow. However, it is not possible to achieve the mass flow fluctuations that would be suggested by a quasi-steady analysis. In addition the transfer function of the system is not as flat as it might be expected to be. In order to understand the processes involved more fully, an unsteady analysis of the flow and further testing will be required.

Acknowledgements

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References

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