ACTIVE STABILIZATION OF COMPRESSOR SURGE

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ABSTRACT

This paper describes the stabilization of compressor surge by an active method. It is known surge follows when small disturbances grow in an unstable compression system, and that small growth can be modeled through a linear stability analysis. An active element is here introduced to counter any tendency to instability and the control law governing the active stabilizer is determined from linear theory. The theory is verified in an experiment on a compression system whose plenum volume is controlled. Suppression of the flow instability was achieved by switching on the controller and the compressor was made to operate stably on a part of its characteristic beyond the natural stall line.

1. INTRODUCTION

The mean pressure rise across a compressor rotor revolving at steady speed is characterized by the pressure rise versus throughflow curve. This characteristic has a maximum, the point corresponding to incipient stall conditions where flow is becoming detached from the rotor surfaces. The angle of incidence onto the sections is reduced by increasing the throughflow, so that the right hand side of the characteristic curve marks a region where the flow is stable and the slope of the pressure/throughflow line is negative. On the left of the peak pressure point lies the unstable region where the flow is rough and unsteady. Three typical cases are illustrated in figure 1, the mean flow through the compressor/throttle system being set at the points where the compressor and throttle characteristics meet. Case 1 and case 2 in that illustration correspond to stable compressor operation while case 3 is unstable. In that case there are large amplitude oscillations in the compressor delivery pressure[Emmons(1955) and Greitzer(1976)] and that system instability is known as surge. Although the detailed mechanism of surge is complex, its principal features are contained in a very simple model[Greitzer(1981)]. The unsteady behavior of the compressor system (consisting of a compressor, a duct, a plenum volume and a throttle) is akin to that of a Helmholtz resonator in which the compressor is an energy source and the throttle a damper. The inertia of the fluid in the duct and the 'springiness' of the compressible fluid in the plenum constitute the resonant elements. When the oscillation receives more energy from the compressor than is dissipated in the throttle then any small perturbation in the flow will grow, initially exponentially with time and eventually develop into a large amplitude oscillation limited by the nonlinear characteristics of the system. The stability of a compression system to small perturbations is critical to whether or not surge will occur, and the onset of surge can be predicted by a linear stability analysis[Emmons(1955) and Stenning(1980)]. This linear model suggests that compressor surge might be suppressed by the active technique described by Epstein, Ffowcs Williams and Greitzer(1986). The main strategy of that control method is that the pressure fluctuation in the plenum is monitored, the signal phase-shifted and amplified to drive an element that varies the volume of the plenum. By means of the unsteady but controlled plenum volume, we can thus distort the natural balance between throttle and compressor flows; a controlled perturbation can alter the energy fed in by the compressor, change the dissipative energy through the throttle, and contribute to the oscillation energy directly, the precise effect depending on how the feedback signals are processed.

In this paper, we demonstrate both theoretically and experimentally that compressor surge may indeed be actively suppressed by a feedback system of that type.

2. STABILITY ANALYSIS OF A COMPRESSION SYSTEM WITH A CONTROLLED PLENUM

The compression system which we consider consists of a centrifugal compressor of a small turbocharger(Holset type H1A) which produces pressure rise and delivers mass flow to a large plenum. The discharge from the plenum is through a throttle in an exit duct whose diameter is much smaller than that of the plenum. The compression system is open to the ambient at compressor inlet and downstream of the throttle. We now introduce into our compression system a controller consisting of a surface A_3 which is part of a mass-spring system responding to the unsteady pressure fluctuation in the plenum and an externally induced control force; its displacement ξ produces a volumetric change $A_3\xi$ in the plenum. The mass stored in the plenum is thus decreased by $\rho A_3\xi$ approximately. The displacement ξ is assumed to be proportional to the driving force, since the inertia of our surface A_3 is negligible in comparison with the spring stiffness at the frequencies in our experiment. The control force is generated by a feedback system which processes the signal detected by a pressure sensor located in the plenum. The whole system is schematically illustrated in figure 2(a). Based on that model, an experimental device was arranged, shown in figure 2(b) in which a loudspeaker was used to work as that surface A_3. The dynamic equations of the system can be written as

$$
\begin{aligned}
&\chi_1\frac{dQ_1}{dt} = f_1(Q_1)-(P-P_a)-\sigma_1(Q_1-\bar{Q}_1),\\
&\frac{V}{a^2}\frac{dP}{dt} = Q_1-Q_2-\underline{A_3\rho\frac{d\xi}{dt}},\\
&\chi_2\frac{dQ_2}{dt} = (P-P_a)-f_2(Q_2),\\
&\underline{\xi = \frac{1}{K}(A_3+C)(P-\bar{P}),}
\end{aligned}
\qquad (1)
$$

The first equation and the third equation come from the momentum balance in the compressor duct and the throttle duct respectively; $f_1(Q_1)$ is the pressure rise produced by the compressor and $f_2(Q_2)$ is the pressure drop across the throttle; χ_1 is a parameter determined by geometry of the compressor duct and χ_2 the parameter determined by the throttle duct. The second equation is based on the mass conservation inside the plenum. The terms underlined ---- represent the controller action. C is some temporal operator assumed to be independent of response amplitude. We nondimensionlize the mass flow rate using the quantity $\rho A_1 U$ (U is the speed of the compressor blade tip), the pressure using $\frac{1}{2}\rho U^2$ and the time t using the characteristic time $1/\omega_H$ (ω_H is the Helmholtz resonance frequency of the compressor system) to get the nondimensional governing equations

$$\frac{d\phi_1}{d\tau} = B[F_1(\phi_1)-\Psi] - \mu(\phi_1-\bar{\phi}_1),$$
$$\frac{d\Psi}{d\tau} = (\phi_1-\phi_2)/B - (\eta+Z)\frac{d\Psi}{d\tau}, \qquad (2)$$
$$\frac{d\phi_2}{d\tau} = RB[\Psi - T(\phi_2)],$$

where $\phi_1=Q_1/(\rho A_1 U)$, $\phi_2=Q_2/(\rho A_1 U)$, $\Psi=(P-P_a)/(\frac{1}{2}\rho U^2)$, $F_1=f_1(Q_1)/(\frac{1}{2}\rho U^2)$, $T=f_2(Q_2)/(\frac{1}{2}\rho U^2)$, $\tau=\omega_H t$, $B=\frac{1}{2}U/(\omega_H A_1 \chi_1)$, $\mu=\sigma_1/(\omega_H \chi_1)$, $R=\chi_1/\chi_2$, $\eta=\frac{\rho a^2 A_3^2}{VK}$, $Z=\frac{\rho a^2 A_3}{VK}C$.

Two terms have been introduced into the governing equations by the controller. The term $\eta\frac{d\Psi}{d\tau}$ is caused by the pressure-induced motion of surface A_3; this reduces the resonance frequency of the system and is a de-stabilizing effect. The term $Z\frac{d\Psi}{d\tau}$ is our representation of the control force, by which we wish to improve the system stability.

By setting $\phi_1=\bar{\phi}+\delta\phi_1$, $\phi_2=\bar{\phi}+\delta\phi_2$, $\Psi=\bar{\Psi}+\delta\Psi$, $F(\phi_1)=\bar{\Psi}+\Psi'\delta\phi_1$, $T(\phi_2)=\bar{\Psi}+T'\delta\phi_2$, we will get the linear perturbation equations for the quantities $\delta\phi_1$, $\delta\phi_2$, and $\delta\Psi$:

$$\frac{d\delta\phi_1}{d\tau} = B(\Psi'\delta\phi_1-\delta\Psi) - \mu\delta\phi_1,$$
$$\frac{d\delta\Psi}{d\tau} = (\delta\phi_1-\delta\phi_2)/B - (\eta+Z)\frac{d\delta\Psi}{d\tau}, \qquad (3)$$
$$\frac{d\delta\phi_2}{d\tau} = RB(\delta\Psi-T'\delta\phi_2).$$

These equations have solutions of the form $e^{s\tau}$; equation(3) then reduces to the characteristic equation:

$$s^3 + a_2 s^2 + [a_1+b_1(\frac{1}{1+\eta+Z} - 1)]s + a_0/(1+\eta+Z) = 0, \qquad (4)$$

where $a_0 = R(BT'+\mu-B\Psi')$, $a_1 = 1+R+BRT'(\mu-B\Psi')$, $a_2 = RBT'+\mu-B\Psi'$, $b_1=1+R$ and Z the control law is a polynomial in s.

The system stability is now not only dependent on a_0, a_1, and a_2, which are parameters determined by naturally uncontrolled compression system, but also on the control parameter Z. The controller can obviously have a wide range of influence because the control parameter Z can be arranged to have a variety of complex forms for various complex eigenfrequencies. In our previous linear analysis, the compressor and the throttle, which are two nonlinear components in the system, act mainly as energy source and sink. If we stop the rotation of the compressor($\Psi'=0$) and close the throttle($T=0$, $R=0$), then a_0 is zero and the characteristic equation(4) reduces to:

$$s^2 + \mu s + \frac{1}{1+\eta+Z} = 0. \qquad (5)$$

We set $\alpha_0=\frac{1}{1+\eta}$ and $z=\frac{Z}{1+\eta}$ (these forms are convenient because they can be measured directly) to write that equation as

$$s^2 + \mu s + \alpha_0/(1+z) = 0. \qquad (6)$$

We now express the control parameter z in the form

$$z = Ge^{i\varphi}, \qquad (7)$$

the gain G and phase φ both being real functions of frequency. G is the amplitude ratio of input signal to output signal of the feedback loop and φ is the phase-shift in the feedback loop. The solutions of equation(6) have a form $s=\beta+i\omega$, in which both β and ω are dependent on the control parameter z. We solve (6) and (7) with a constant gain and different phase shift from 0^0 to 360^0. The calculated results indicated in figure 3 shows that in the phase range 0^0 - 180^0, the controller will reduce the system damping, even change its sign; while in the phase range 180^0 - 360^0, the controller will increase the system damping.

3. ACTIVE STABILIZATION OF COMPRESSOR SURGE

It is known(Emmons 1955; Epstein, Ffowcs Williams and Greitzer 1986), that when small fluctuations of the pressure and mass flow can grow up in a compression system, they will finally develop into a limit cycle which makes the operating point(determined by the average pressure and mass flow rates) leave the original compressor characteristic; the compressor is then in surge. On that limit cycle, the oscillation amplitudes of the pressure and mass flow rate are limited and relatively large and the oscillation frequency is usually lower than that of initial small amplitude fluctuations.

We assess our controlled compression system in the following ways. First, the controller was activated before the surge onset, the throttle was then progressively closed to make the compressor operating point move from the right of the natural surge boundary until the system surged. The measured results are illustrated in figure 4, which shows that the controller has made the compression system operate stably beyond its natural surge boundary and moved the surge boundary left. Second, we switched on the controller after the surge had happened in the system. Both pressure trace and control signal records illustrated in figure 5 show that the controller is extremely effective, the surge being suppressed almost instantaneously. This control effect demonstrates the important fact that the system can recover from surge even though surge is not a small amplitude phenomenon.

To suppress compressor surge, we must correctly choose the control parameter $z=z_r+iz_i=Ge^{i\varphi}$, i.e. choose the gain and the phase-shift. Those points, in the gain-phase plane, at which the controller can suppress surge constitute a stable area, and the points at which the controller can not suppress the surge (even makes it worse) constitute an unstable region. The two regions are divided by the stability boundary. The controlled compression system characteristic equation(4) with a solution $s=\beta+i\omega$, specifies the stability boundary according to $\beta=0$, at which condition we solve equation(4) at the throttle position just across the natural surge boundary. We thus define the stability boundary and the stable control region on the gain-phase plane; this is shown in figure 6. We found a very satisfactory level of agreement between the stability points defined in this way and the measured values in our experiment which are also shown in figure 6.

4. CONCLUSIONS

Active stabilization of compressor surge has been achieved in our experiments with a compression system incorporating a controlled plenum; the controller is able to alter the system damping and the resonance frequency. The results show that the compression system can be effectively stabilized by switching on the controller before or even after surge occurs. Our experiments indicate that the linear controller is effective even in this nonlinear aerodynamic case, and this make us believe that the stability and the performance of compression systems generally could be greatly improved through active control techniques of this type.

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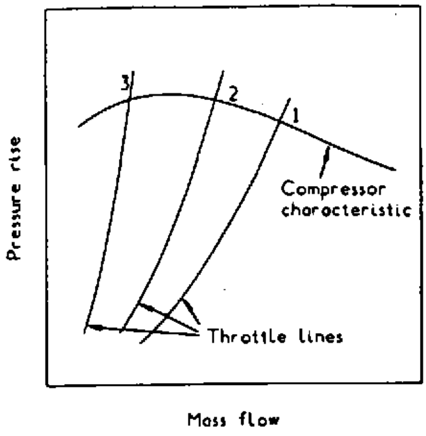

Figure 1. Typical characteristics of a compression system.

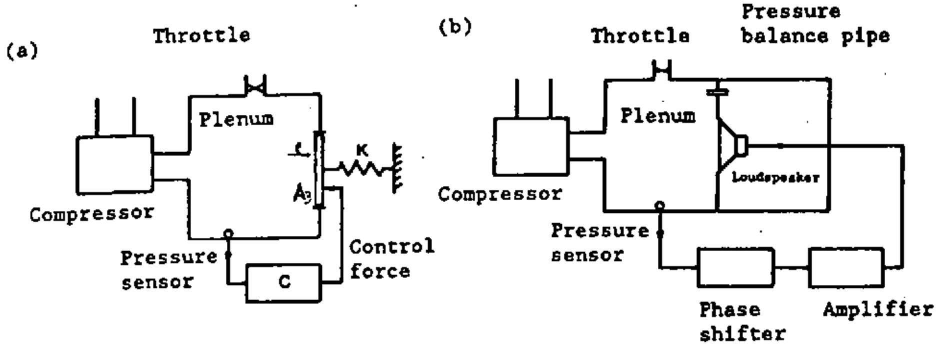

Figure 2. (a) Model of the compression system with a controlled plenum. (b) The geometry of the compression system with a controller, used in the experiments.

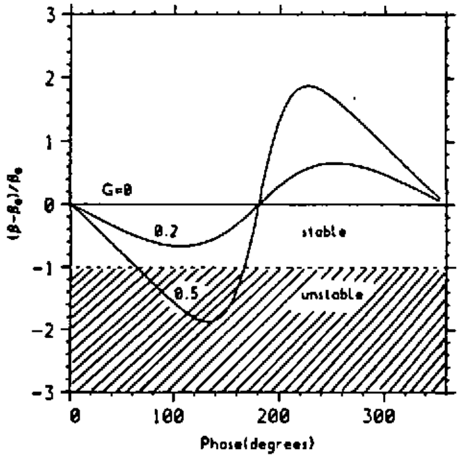

Figure 3. The system damping changed by the controlled plenum, depending on the controller gain G and the phase shift. β is the real part of eigenvalue of equation(6), β_0 being the value when G=0.

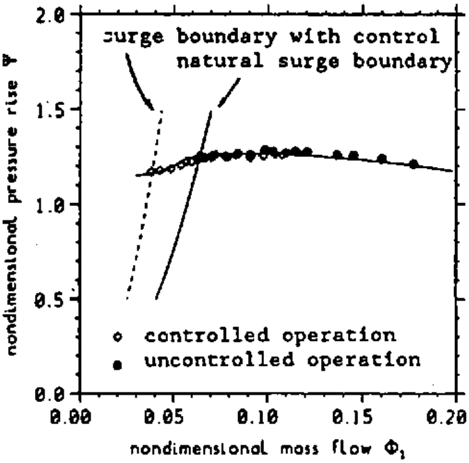

Figure 4. The effect of the active stabilisation on the compressor performance.

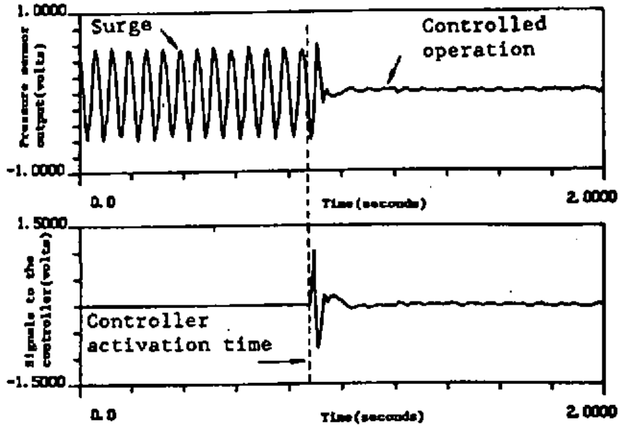

Figure 5. Time traces of the pressure fluctuations in the plenum and the signal to the control surface. The surge was completely suppressed following switching on the controller.

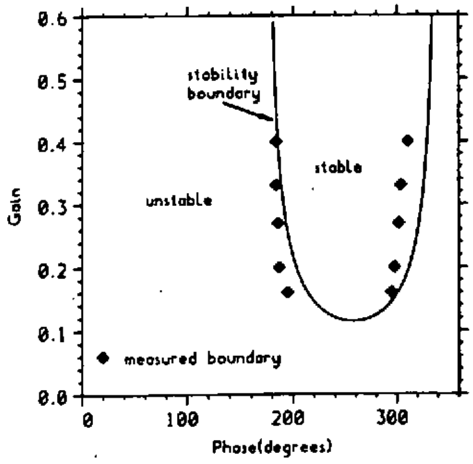

Figure 6. An illustration of stability map of the compression system with a controlled plenum. The gain is the ratio between the output and the input signals of the feedback loop, while the phase is the phase shift of the two signals. The compressor was operated just left of the natural surge boundary.