# Active Suppression of Aerodynamic Instabilities in Turbomachines

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**In this paper, we advocate a strategy for controlling a class of turbomachine instabilities, whose primitive phases can be modeled by linear theory, but that eventually grow into a performance-limiting modification of the basic flow. The phenomena of rotating stall and surge are two very different practical examples in which small disturbances grow to magnitudes such that they limit machine performance. We develop a theory that shows how an additional disturbance, driven from real-time data measured within the turbomachine, can be generated so as to realize a device with characteristics fundamentally different than those of the machine without control. For the particular compressor analyzed, the control increases the stable operating range by 20% of the mean flow. We show that active control can also be used to destabilize a compressor in an undesirable state such as nonrecoverable stall. Examination of the energetics of the controlled system shows the required control power scales with the square of the ambient disturbance level, which can be several orders of magnitude below the power of the machine. Brief mention is also made of the use of structural dynamics, rather than active control, to enhance stability.**

## Nomenclature

| | |
|---|---|
| $a$ | = speed of sound |
| $A_c$ | = compressor (or pump) flow through area |
| ANMP | = average net mechanical power (excess over steady state) due to perturbations |
| $A_T$ | = throttle area |
| $B$ | = system stability parameter; $B = U/a\sqrt{V/A_cL_c}$ |
| $C_x$ | = axial velocity |
| $L_c$ | = compressor (or pump) effective length |
| $M$ | = blade Mach number, $= U/a$ |
| $n$ | = harmonic number of disturbance |
| $p$ | = pressure |
| $p_t$ | = total pressure |
| $P$ | = nondimensional pressure difference, $= (p - p_{\text{ambient}})/\rho U^2$ |
| $r$ | = mean radius of turbomachine |
| $R$ | = modulus of control parameter |
| $t$ | = time |
| $U$ | = blade speed at midheight |
| $V$ | = plenum volume |
| $x$ | = axial coordinate |
| $Z$ | = control parameter, $= Re^{i\beta}$ |
| $Z_A$ | = throttle control parameter |
| $Z_\xi$ | = plenum control parameter |
| $\alpha$ | = disturbance growth rate |
| $\beta$ | = phase of control parameter |
| $\xi$ | = fractional change in plenum volume, $= \delta V/V$ |
| $\delta(\ )$ | = perturbation quantity |
| $\delta C_x$ | = axial velocity perturbation at compressor |
| $\delta C_{x1}$ | = axial velocity perturbation at compressor with no controller |
| $\delta C_x'$ | = axial velocity perturbation due to controller |
| $\Delta P$ | = compressor (or pump) pressure rise |
| $\lambda$ | = inertia parameter for compressor stators |
| $\mu$ | = inertia parameter for total compressor blading |
| $\bar{\eta}$ | = mean compressor efficiency |
| $\rho$ | = density |
| $\phi$ | = axial velocity parameter, $= C_x/U$ |
| $\psi$ | = nondimensional compressor (or pump) pressure rise, $= \Delta P/\rho U^2$ |
| $\theta$ | = circumferential coordinate |
| $\omega$ | = frequency |
| $(\bar{\ })$ | = evaluated at mean operating point |
| $\langle\rangle$ | = time average (for perturbation quantities) over a cycle |

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## Introduction

THE continuing microelectronic revolution opens new doors to the designer of fluid and mechanical systems[1] who can now consider replacing basically open-loop devices with ones employing integral electronic feedback control at relatively low cost. This engineering approach is fundamentally different from the traditional one that emphasized simplicity, but the potential for increased performance, functionality, maintainability, and lowered development costs has made integrated control machines attractive in a number of applications. Many of these are flight critical in the sense that malfunction of the control system could result in loss of the vehicle, requiring extreme confidence in the control system.

We are interested in the use of integral feedback control to improve the performance of pumps and compressors by increasing the stability of the machine through suppression of surge and rotating stall. These aerodynamic instabilities are intrinsic to a wide variety of turbomachines,[2] and often stand as absolute limits to their performance. The increased stability would thus give a potential for a large increase in machine performance-operating range and pressure rise.

Because of its importance, stall and surge control have been under investigation for a number of years. The general scheme has been to obtain performance increases by reducing surge margin, detecting the onset of rotating stall or surge, and then lowering the compressor operating point when required. This approach was largely empirical and did not prove totally successful, due to problems associated with detection of instability onset and necessity of large control forces required to move the compressor operating point.[3]

The approach proposed herein is fundamentally different in that: A) it is aimed at increasing the aerodynamic damping of the instabilities through active control, thus allowing stable compressor operation in a previously unstable (and thus forbidden), high-performance region, and B) operates on small amplitude disturbances, thus requiring relatively little control authority. The approach is illustrated conceptually on the compressor map shown in Fig. 1. Point A represents a conventional operating point without control and point B a new operating point in the region (shaded) stabilized by active flowfield control past the "natural" stall line.

In this paper, we discuss strategies for controlling a broad class of turbomachinery instabilities whose initial phases can be modeled by linear theory. Rotating stall is just one such example, in which small disturbances evolve into a finite amplitude limit cycle, where the strength of this "rotating stall cell" is set by nonlinear effects. Surge, which is a more global system instability, is another example.

We present a simple analysis to show how a controller can be employed to stabilize such situations. The analysis is not intended to be more than an introductory discussion of the topic, but the conclusions are extremely encouraging. We examine how to generate an additional disturbance using a controller processing data measured inside the turbomachine. The controlled system now constitutes a machine of fundamentally different characteristics, in which stable operation can be achieved under conditions that previously implied breakdown of the flow.

In the following, we formulate the relationship between controller behavior and the onset of local and global turbomachinery instability and show that there can be a substantial effect using realistic levels of control actions. A related topic, the active destabilization of a fully developed stall cell, is also briefly described.

## Basic Linear Problem: Active Suppression of Flowfield Instability (Rotating Stall) in an Axial Compressor

We assume rotating stall to be the mature state of the phenomenon whose linear behavior is described (see, for example, Refs. 4 or 5). This is a disturbance essentially confined to the compressor, which is steady in a reference frame rotating at a fraction of the wheel speed. The compressor, which is taken to be of high hub-tip ratio so that a two-dimensional analysis is appropriate, is fed a flow of uniform total pressure and discharges axially to a stream of eventually constant delivery pressure. Upstream and downstream terminations of the ducting are taken to be far enough away that the evanescent disturbances of the evolutionary stall cell have negligible amplitude there, and the disturbance waves are not reflected or scattered by those terminations.

The compressor is modeled as a machine whose steady delivery pressure is locally (transiently) modified by the pressure difference required to overcome the inertia of the unsteady flow in the blade channels, assumed as incompressible and one-dimensional.[4,5] The local perturbation in pressure rise across the compressor is then, in the notation of Ref. 5,

$$\frac{\delta\Delta p}{\rho U^2}=\overline{\left(\frac{\mathrm{d}\Psi}{\mathrm{d}\phi}\right)}\delta\phi-\frac{\partial\delta\phi}{\lambda\partial\theta}-\frac{r\mu}{U}\frac{\partial\delta\phi}{\partial t} \tag{1}$$

where $\overline{(\mathrm{d}\psi/\mathrm{d}\phi)}$ is the slope of the nondimensional compressor pressure rise characteristic (exit pressure minus inlet total static pressure) evaluated at the steady-state operating point, and $\phi$ is the flow coefficient $C_x/U$. Positive numbers $\lambda$ and $\mu$ represent the inertia of the fluid inside the machine that has mean radius $r$.

We represent the axial velocity perturbation in the machine as

$$\delta C_x = a_n U e^{in\theta} e^{i\omega t} e^{\alpha t} \tag{2}$$

In a linearly disturbed irrotational flow at compressor entry, the total pressure variations $p_t$ must satisfy Laplace's equation, and that element matching the compressor velocity disturbance and a boundedness condition at $x = -\infty$ must be proportional to

$$e^{in\theta} e^{i\omega t} e^{\alpha t} e^{|n|x/r} \tag{3}$$

Using this in the axial component of the momentum equation gives a relation between the total pressure and axial velocity fluctuations

$$\frac{|n|}{r}\delta p_{t1} = -(\alpha+i\omega)\rho\delta C_{x1} \tag{4}$$

at compressor entry, $\delta C_{x1}$ being the increase in axial velocity at the compressor face over that in the entry flow.

Many different control strategies for the compressor are possible. We do not attempt to discuss the topic of actual implementation in an engine, but rather show several obvious techniques that could be carried out in a research environment and that illustrate the central concepts of the stabilization procedure. For example, a vortical velocity perturbation could be put in far upstream using unsteady jets or wiggling stators [(1) in Fig. 2], the inlet guide vanes wiggled (2), or a downstream perturbation imposed (3) using loudspeakers or pulsing the combustor.[6]

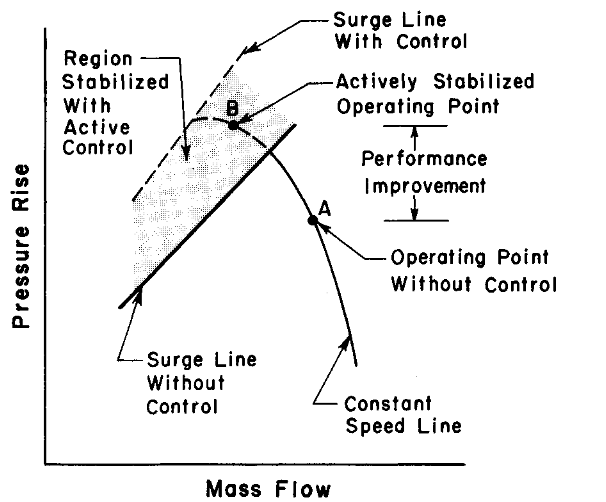


**Fig. 1 Schematic representation of the effect of active stabilization on compressor performance.**

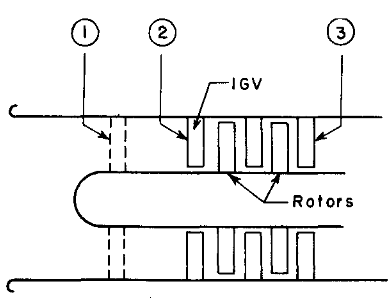


**Fig. 2 Some possibilities for applying control flow perturbation, including 1) upstream vortical disturbance, 2) wiggling inlet guide vanes (IGV's), and 3) downstream pressure perturbation.**

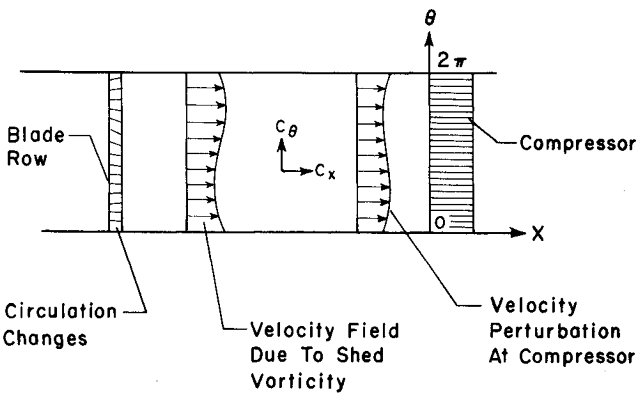


**Fig. 3 Flow geometry showing vortical flow perturbation used in example.**

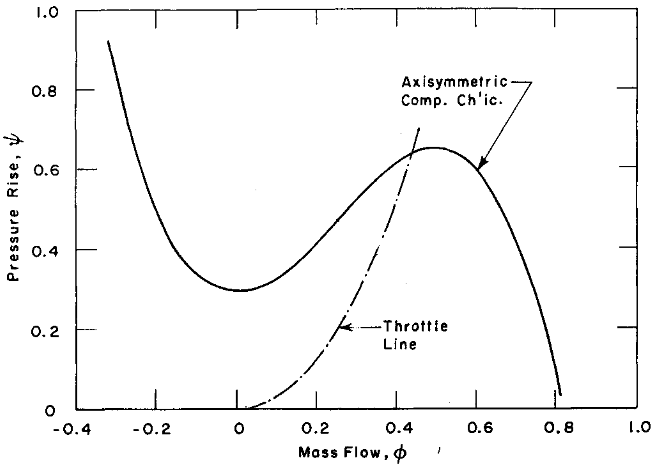


**Fig. 4 Nondimensional compressor and throttle characteristics for an axial compressor.**

In the example worked here, the control is exercised through an upstream set of vanes that are operated to create a vortical disturbance field. The situation is diagrammed in Fig. 3, where this upstream row of blades is oscillated so that the circulation around each one varies with time and a shed vorticity field, which is convected downstream, is produced. This vortical perturbation is associated with an axial velocity $\delta C_x'$, which supplements $\delta C_{x1}$, the potential disturbance in the inlet flow, so that

$$\delta C_x = \delta C_{x1} + \delta C_x' \tag{5}$$

The total pressure perturbation at compressor entry is thus

$$\delta p_t = \rho \bar{C}_x \delta C_x' - (\alpha + i\omega)\rho(\delta C_x - \delta C_x')\frac{r}{|n|} \tag{6}$$

where $\bar{C}_x$ is the mean axial velocity.

For small amplitude disturbances, the pressure in the downstream flow satisfies Laplace's equation, and must be of the form

$$e^{in\theta}e^{i\omega t}e^{\alpha t}e^{-|n|x/r} \tag{7}$$

Using continuity and the condition of constant exit angle (axial), the pressure gradient at compressor exit can be related to the velocity perturbations by

$$\frac{\partial \delta p}{\partial x} = -\rho\frac{\partial \delta C_x}{\partial t} \tag{8}$$

Therefore, from Eqs. (2), (7), and (8) the compressor exit the static pressure perturbation $\delta p$ is given by

$$\delta p = \frac{r}{|n|}\rho(\alpha + i\omega)\,\delta C_x \tag{9}$$

The difference between Eqs. (9) and (6) can now be substituted for $\Delta p$ in Eq. (1), i.e.,

$$\delta\phi\left[\frac{(\alpha + i\omega)}{U}\frac{r}{|n|}(2 + \mu|n|) + in\lambda - \left(\overline{\frac{\mathrm{d}\psi}{\mathrm{d}\phi}}\right)\right] = \delta\phi'\cdot\left[\bar{\phi} + \frac{r}{|n|}\frac{(\alpha + i\omega)}{U}\right] \tag{10}$$

where $\delta\phi = \delta C_x/U$, and $\delta\phi' = \delta C_x'/U$.

Rotating stall onset (instability) occurs when the disturbance amplitude is steady, i.e., $\alpha = 0$. For the no control situation, $\delta\phi' = 0$ (i.e., $\delta C_x' = 0$), this is[4]:

$$\left(\overline{\frac{\mathrm{d}\psi}{\mathrm{d}\phi}}\right) = 0 \quad \text{(critical slope for instability)} \tag{11a}$$

$$\frac{\omega r}{U} = \frac{-n|n|\lambda}{2 + \mu|n|} \quad \text{(phase speed of propagating disturbance)} \tag{11b}$$

Our initial control strategy is to arrange that $\delta\phi'$ is a linear function of the perturbation in compressor flow coefficient

$$\delta\phi' = \frac{\delta C_x'}{U} = Z\delta\phi \tag{12}$$

in which case Eq. (10) gives the growth rate $\alpha$ of the perturbation as the solution of:

$$\frac{(\alpha + i\omega)}{U}\frac{r}{|n|}\{2 - Z + \mu|n|\} + in\lambda - \left(\overline{\frac{\mathrm{d}\psi}{\mathrm{d}\phi}}\right) = Z\bar{\phi} \tag{13}$$

$Z$ being selected to give the most desirable machine performance.

The stability boundary of the "controlled" compressor occurs when $\alpha = 0$, at which condition

$$\frac{i\omega r}{U|n|}\{2 - Z + \mu|n|\} = Z\bar{\phi} + \left(\overline{\frac{\mathrm{d}\psi}{\mathrm{d}\phi}}\right) - in\lambda \tag{14}$$

## Some Qualitative Features of the Controlled System: Active Stabilization

If $Z$ is real, Eq. (14) amounts to the constraints that

$$\left(\overline{\frac{\mathrm{d}\psi}{\mathrm{d}\phi}}\right) = -Z\bar{\phi} \tag{15}$$

$$\frac{\omega r}{U} = \frac{-n|n|\lambda}{2 - Z + \mu|n|} \tag{16}$$

Negative real values of $Z$ thus allow neutrally stable oscillations on the positive-sloped part of the compressor characteristic, with a lower phase speed than in the uncontrolled case. As far as we are aware, this is the first indication that useful control, i.e., suppression of the growth of rotating stall, may be achieved in this way. Note that operation in this part of the characteristic regime would also call for the active control of the axisymmetric mode; we address this subsequently. The above control, however, will inhibit altogether the condition of rotating stall when the compressor is operated in the surge-free condition.

If $Z$ is purely imaginary, ($Z = iZ_i$, where $Z$ is real), the stability boundary would occur when

$$\left(\frac{\overline{d\psi}}{d\phi}\right) = \frac{\omega r}{U|n|} Z_i \tag{17a}$$

$$\frac{\omega r}{U|n|} = \frac{\bar{\phi} Z_i - n\lambda}{2 + \mu|n|} \tag{17b}$$

that is at the compressor characteristic slope,

$$\left(\frac{\overline{d\psi}}{d\phi}\right) = Z_i\left(\frac{\phi Z_i - n\lambda}{2 + \mu|n|}\right) \tag{18}$$

## Numerical Results for Rotating Stall Control

For general values of the controller parameter $Z$ it is useful to examine numerical results. In doing this, one must first specify a compressor characteristic curve. The one used in the present examples is shown in Fig. 4, based on a low-speed, three-stage compressor.[7] The axes are nondimensional pressure rise $\psi[=(p_{\text{out}} - p_{t\text{in}})/\rho U^2]$, and nondimensional mass flow $\phi(=C_x/U)$. While the quantitative features of the curve are obviously individual to the particular machine, the qualitative features reflect the generic behavior of a wide class of compressors. The point of operation of the system is set by the intersection of the (downstream) throttle line with this curve and a typical throttle line is sketched.

With no controller, the instability point is at the peak of the curve $\phi = 0.5$. With a controller, however, the point of instability can be shifted. Figure 5 is a plot of mass flow $\phi$ at instability vs $\beta$, the phase of the control parameter $Z$. The separate curves are for different values of $R$, the modulus of $Z$;

$$Z = Re^{i\beta} \tag{19}$$

$R = 1$ corresponds to a level of control perturbation equal to that of the ambient fluctuations. Even for these small values of $R$ there is a substantial movement of the stall point. If one considers the controlled disturbance to be put in by a row of wiggled far-upstream vanes, the local flow angle variation required for a 1% (vortical) velocity perturbation is roughly 1.5 deg. Thus, the $R$ values indicated are definitely possible.

These initial results of the studies are extremely encouraging. There is a major effect on the mass flow at which the compressor can operate while avoiding the rotating stall instability. This appears to be true, not only for the upstream vane-induced control signals, but also for the other strategies illustrated in Fig. 2, such as a wiggly inlet guide vane, or a downstream pressure perturbation.

## Destabilization of Rotating Stall and Stall Recovery

The calculations that have been discussed refer to the problem of stabilizing a given system. However, there are also situations in which it can be useful to move the system from a given operating point, i.e., to destabilize the motion. An important example of this occurs in the phenomenon of nonrecoverable stall. In this situation, a gas turbine engine operates with a fully developed rotating stall cell extending through the compressor. This stalled state can be extremely difficult to recover from because of the large amount of hysteresis that exists between the onset of stall and the stall cessation as the throttle is opened.[2]

Recent analyses[7] have led to computational procedures (pseudospectral methods) for tracking the evolutions of the stalled flow transients in a turbomachine, in particular the nonlinear evolution from small amplitude disturbance to fully developed rotating stall. An example of the results of such calculations is given in Fig. 6, which shows the growth of the

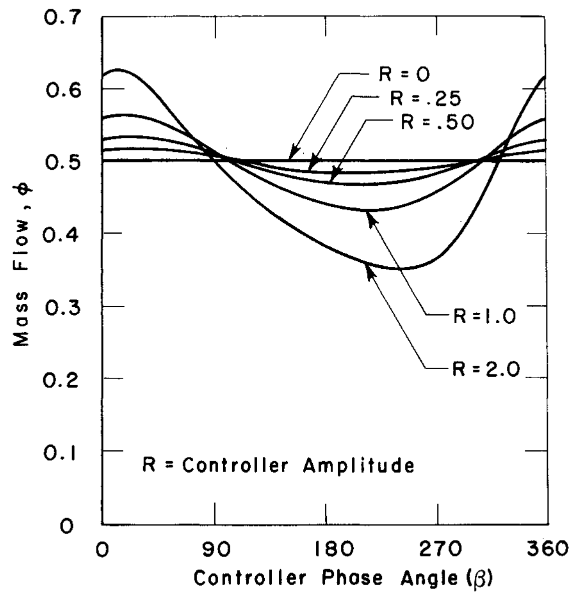


**Fig. 5 Change in nondimensional mass flow $\phi$ at onset of instability with controller phase, for different values of controller amplitude $R$.**

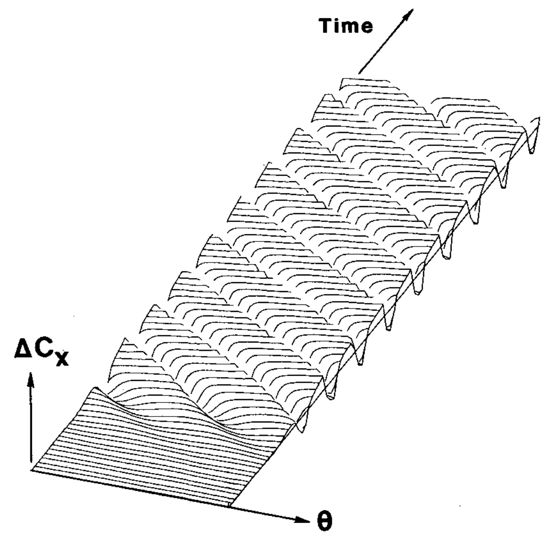


**Fig. 6 Calculated time evolution of rotating stall cell (nonlinear axial velocity wave).**

axial velocity wave that becomes the stall cell.[8] The horizontal axis is the circumferential position ($\theta$) around the annulus, the height of the wave is the axial velocity nonuniformity ($\Delta C_x$), and time is shown in the third axis. The lines of constant phase of the disturbance can be seen to be at an angle to the $\theta$ axis so that the disturbance is propagating around the circumference. The amplitude of the stall cell is roughly the same as the average flow velocity so the disturbances are large and have a major effect. Although the initial perturbation was sinusoidal, the final velocity profile is rich in harmonics due to the strong nonlinearity of the flowfield.

Destabilization of a stable stalled compressor flowfield has been explored using these time accurate computations. The control perturbation adopted was

$$\delta\psi = \delta(\text{pressure rise}) = Z\,\delta C_x$$

Figure 7 shows the results for the most unstable eigenmode, which has a harmonic distribution the same as the nonlinear

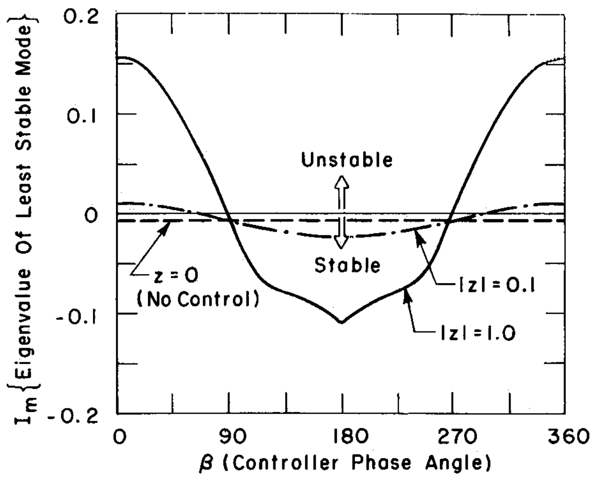


Fig. 7 Using active control to destabilize a stalled compressor: imaginary part (growth rate) of eigenvalue of least stable perturbation to nonlinear axial velocity wave (rotating stall cell).

wave and is very weakly damped. The figure gives the imaginary part of the eigenvalue (the growth rate; negative values imply stability and positive values instability) as a function of controller phase angle $\beta$ for $Z = 0$ (no control), $|Z| = 0.1$, and $|Z| = 1$. For both the latter values of $Z$, the rotating stall is destabilized over a range of values of $\beta$.

The implication of the calculations is that the stalled state can be made unstable. One outcome could be a return to operation on the axisymmetric unstalled part of the compressor performance curve. A more likely result, however, is a new operating point somewhere up the throttle line, i.e., a change in the amplitude of the limit cycle that defines the rotating stall. The magnitude of the change and the position of the new operating point can only be computed using the nonlinear evolution calculation. We have not yet done this, but the central point is that the undesirable stall mode may be susceptible to destabilization by appropriate control action.

## Simple Control Model for Global Compression System Instability (Surge)

We must also examine the global stability of the compression system (i.e., surge). The basic analytical model has been reported in several places.[2,9,10] The departure here is the concept of surge control. We do not address actual implementation, but use two basic strategies to illustrate potential gains.

It is useful to put the problem in perspective with respect to the different phenomena of interest. For a multistage axial compressor, both local (compressor) and global (compression system) instabilities are important. Attempting to attack only the latter will generally not succeed due to the strong effect of the former on the overall pumping characteristic.[2] For a centrifugal compressor, however (and perhaps for a single stage axial fan as well), the local instability leading to rotating stall is often of much less serious consequence than the system instability. In this case, one may deal only with the latter. Although there are other ways to stabilize centrifugal machines, such as close-coupled flow resistances[11] and variable inlet guide vanes, these have associated inefficiencies that the active control may avoid. In addition, the range extension offered by the active stabilization is potentially larger.

With this as background, we discuss the stability of the compression or pumping system shown in Fig. 8, which is viewed essentially as a Helmholtz resonator in much the same way as in Refs. 2, 9, or 10. In addition to the usual system components—compressor (or pump), compressor duct, throttle, plenum—we have now indicated that the wall position

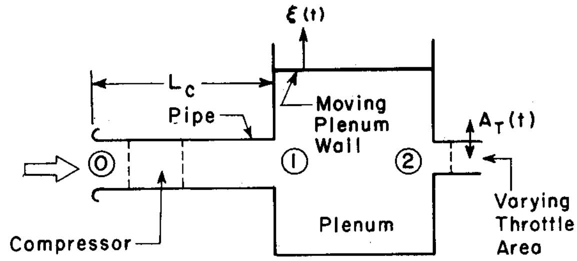


Fig. 8 Basic pumping system with control(s).

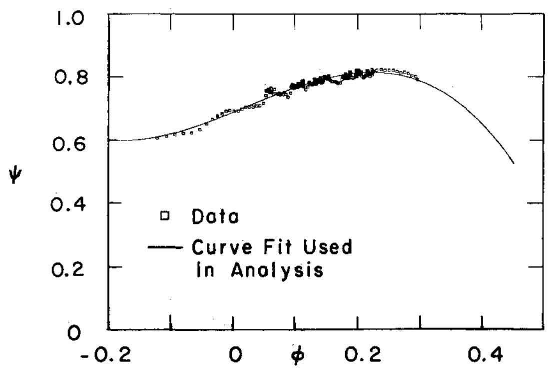


Fig. 9 Centrifugal compressor pumping characteristic used in numerical calculation.

and the throttle area of the plenum can undergo fluctuations in position. Both passive and active control schemes can be invoked, i.e., the motions could be driven directly by the fluctuations in the plenum pressure ($\delta p$) rather than by external control, although we consider only the latter here.

The plenum wall position and the throttle area are taken to be controlled according to

$$\xi = Z_\xi \, \delta P \qquad \text{(plenum wall position control)} \qquad (20)$$

$$\frac{\delta A_T}{A_T} = Z_A \, \delta P \qquad \text{(throttle area control)} \qquad (21)$$

respectively, where $\delta P$ is the nondimensional perturbation in plenum pressure ($=\delta p/\rho U^2$) and $\xi$ and $\delta A_T/A_T$ are the fractional change in plenum volume and throttle area as indicated in Fig. 8. As with the rotating stall examples, significant changes in stable flow range result, even though the control strategies are quite simple, and more sophisticated strategies could certainly be employed.

The detailed steps leading to the set of linearized equations describing the dynamics of this lumped parameter system model will not be repeated here. The compressor and throttle performance is linearized about a time mean state, denoted as before by an overbar ( ). Inertance in the throttle, which is generally small in practice, is neglected. The resulting linearized equations for the perturbations in pressure and mass flow are:

$$\frac{\mathrm{d}\,\delta P}{\mathrm{d}t} = \frac{\delta\phi_1 - \delta\phi_2}{B(1 + Z_\xi/M^2)} \qquad (22)$$

$$\frac{\mathrm{d}\,\delta\phi_1}{\mathrm{d}t} = B\left[\left(\overline{\frac{\mathrm{d}\psi}{\mathrm{d}\phi}}\right)\delta\phi_1 - \delta P\right] \qquad (23)$$

$$\delta P = \left[\frac{\left(\overline{\dfrac{\partial T}{\partial \phi}}\right)}{1 + \left(\overline{\dfrac{\partial T}{\partial \phi}}\right)\bar{\phi} Z_A}\right]\delta\phi_2 \qquad (24)$$

The quantities $\overline{\mathrm{d}\psi/\mathrm{d}\phi}$ and $\overline{\partial T/\partial\phi}$ are derivatives of the compressor and throttle pressure flow characteristics at the steady-state operating point, and $B(=U/a\sqrt{V/A_cL_c})$ is a system stability parameter.

Assuming solutions of the form $e^{st}$ and substituting into Eqs. (22) to (24) leads to an equation for the growth rate $S$:

$$S^2+S\left\{\frac{\left[1+\left(\frac{\overline{\partial T}}{\partial\phi}\right)\bar{\phi}Z_A\right]}{B(1+Z_\xi/M^2)}-B\left(\frac{\overline{\partial\psi}}{\partial\phi}\right)\right\}+\frac{1}{(1+Z_\xi/M^2)}\cdot\left\{1-\frac{\left(\frac{\overline{\partial\psi}}{\partial\phi}\right)}{\left(\frac{\overline{\partial T}}{\partial\phi}\right)}\left[1+\left(\frac{\overline{\partial T}}{\partial\phi}\right)\bar{\phi}Z_A\right]\right\}=0 \tag{25}$$

Equation (25) is, operationally, a statement that instability will occur when the mass flow is decreased such that the slope of the compressor pumping curve, $(\mathrm{d}\psi/\mathrm{d}\phi)$, reaches a critical positive value. However, if either $Z_A$ is positive and real or $Z_\xi$ is negative and real, for example, dynamic instability will be suppressed. The use of control is thus clearly of benefit in these cases. The optimum controller, however, is not necessarily with $Z_A$ or $Z_\xi$ real, and to see the general trends it is more useful to examine numerical results directly.

### Numerical Results for Surge Control

To illustrate the magnitude of the stability changes that are described, calculations have been carried out for a set of parameters representative of a centrifugal compressor. The pumping characteristic used in the computations is shown in Fig. 9, with the data on which this curve is based (taken from transient measurements) also indicated.[12]

The results for both control strategies are given in Fig. 10, which shows nondimensional mass flow at instability vs controller phase angle for different control amplitudes. The top set of curves is for control of the throttle area and the lower set refers to controlling the plenum volume.

The control has a significant effect on the stability point for the parameters examined, and it is worthwhile to consider the magnitudes of the actual control actions that are being applied. For the throttle control, a value of $Z_A=1$ means that the percent throttle area changes are equal to the percent changes in nondimensional plenum pressure. If the plenum pressure fluctuations are several percent of $\rho U^2$, we must change the throttle area also by several percent. This certainly appears feasible.

Similar considerations apply to the control by moving the plenum wall. In short, there do not appear to be serious constraints for stabilization of surge type oscillations. Preliminary confirmation of these stabilization schemes has actually been achieved by Huang (at Cambridge)[13] using a loudspeaker for the "moving wall", and by Pinsley (at Massachusetts Institute of Technology) using a downstream throttle.[14] Stabilizing the surge type of mode (in a centrifugal compressor) is easier than suppressing rotating stall in an axial compressor because of the spatial uniformity and the low frequency of the oscillations, as well as the relatively slow growth rate of the instability.[12]

### Energy Balances, Surge Suppression Mechanisms, and Controller Power Flow

The mechanism by which the controllers suppress the growth of oscillations can be discussed by examining the energy production and dissipation in the system. The instabilities of practical interest are dynamic rather than static, and the growth of oscillations thus necessitates a net (above the steady value) mechanical energy input by one of the system components when undergoing small amplitude motions.

Three elements in the basic system can feed mechanical energy into the system oscillations: 1) the compressor or pump, 2) the throttle, and 3) the moving wall of the plenum. [Within the present approximation that neglects local, as opposed to convective, acceleration effects in the throttle passage (due to the short length and, hence, low-reduced frequency), work put into the throttle control, i.e., into changing the throttle area, does not affect the system energy balance.] In the situation with no control, the flow in the throttle is only dissipative and the component responsible for instability is the compressor (or pump),[2] which must be operating on a positively sloped part of its characteristic.

In the general case, the average (over a cycle) net mechanical power (ANMP) produced at the compressor, the throttle, and the moveable plenum walls is given by

$$\left(\frac{2\pi\langle\mathrm{ANMP}_c\rangle}{\rho U^3A_c}\right)=\left(\frac{\overline{\mathrm{d}\psi}}{\mathrm{d}\phi}\right)\int_0^{2\pi}(\delta\phi_1)^2\,\mathrm{d}t \quad \text{(compressor)} \tag{26}$$

$$\left(\frac{2\pi\langle\mathrm{ANMP}_T\rangle}{\rho U^3A_c}\right)=\int_0^{2\pi}\left\{-\left(\frac{\overline{\partial T}}{\partial\phi}\right)_A(\delta\phi_2)^2-\left(\frac{\overline{\partial T}}{\partial A}\right)Z_A(\delta P\delta\phi_2)\right\}\mathrm{d}t \quad \text{(throttle)} \tag{27}$$

$$\left(\frac{2\pi\langle\mathrm{ANMP}_\xi\rangle}{\rho U^3A_c}\right)=\frac{1}{B}\frac{V}{A_cL_c}\int_0^{2\pi}\left[\delta P\frac{\mathrm{d}}{\mathrm{d}t}(Z_\xi\,\delta P)\right]\mathrm{d}t \quad \text{(plenum)} \tag{28}$$

respectively. All quantities within the integrals are nondimensional, and the bracket denotes an average over a period.

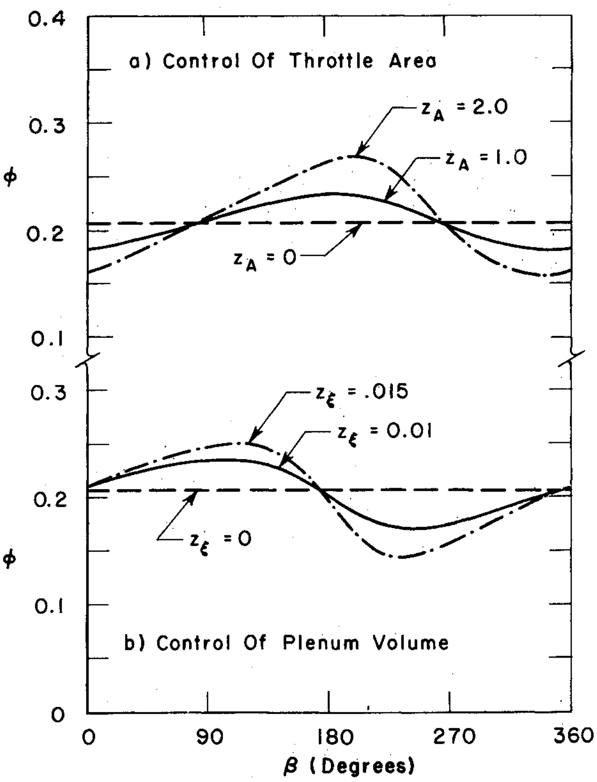


**Fig. 10 Effect of control on onset of instability (surge) in the centrifugal compressor of Fig. 9 as a function of controller amplitude $Z$ and phase $\beta$, $B=1.0$, $M=0.2$.**

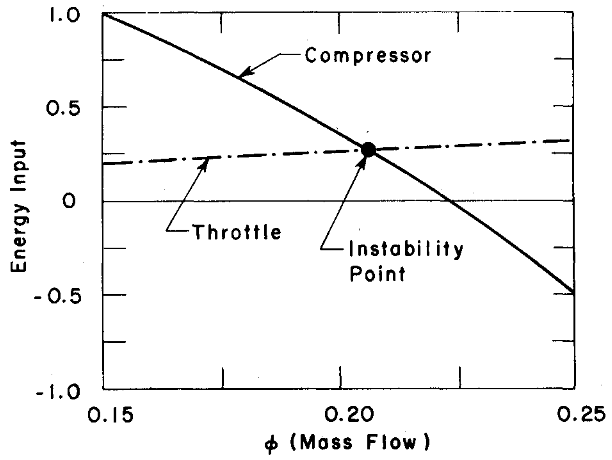


Fig. 11 Average net mechanical power (energy input) for small amplitude oscillations for system as in Fig. 10: no control.

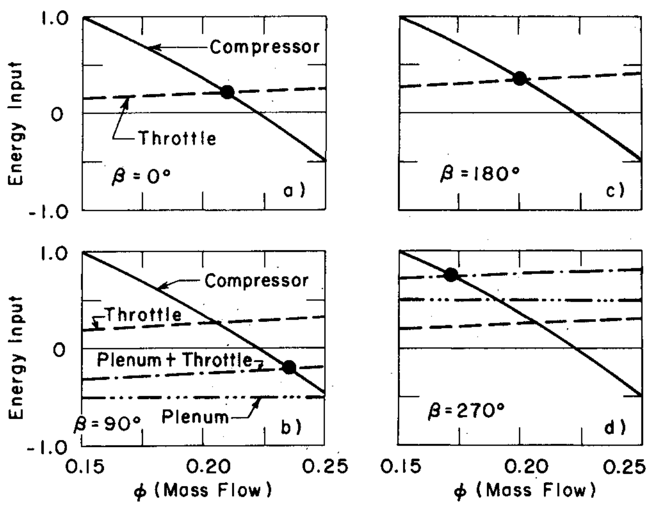


Fig. 12 Average net mechanical power (energy input) for small amplitude oscillations for system as in Fig. 10: moving wall controller ($Z_\xi = 0.01$) at four phase angles.

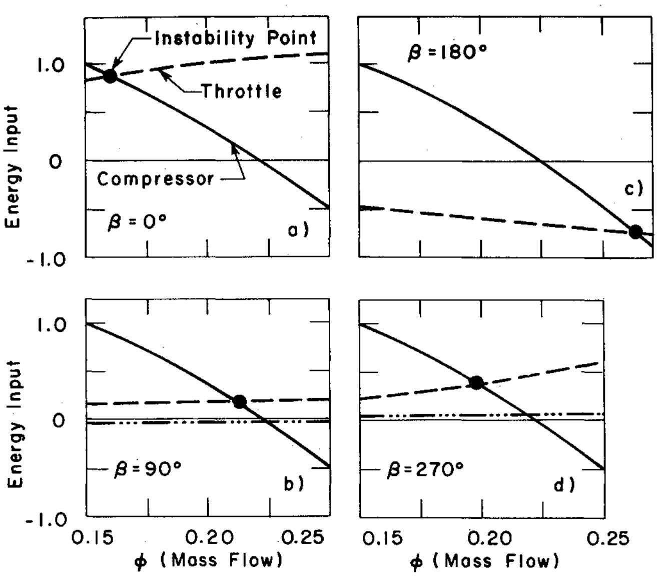


Fig. 13 Average net mechanical power (energy input) for small amplitude oscillations for system as in Fig. 10: variable throttle controller ($Z_A = 2$) at four phase angles.

The ANMP that must be expended to control compressor instability will be equal to, or less than (if there is other dissipation), the net average power produced by the compressor. The ratio of this latter quantity ($\langle \mathrm{ANMP}_c \rangle$) to the mean power needed to drive the compressor is

$$\frac{\langle \mathrm{ANMP}_c \rangle}{\text{mean compressor power}} = \frac{\bar{\eta}}{2\pi} \int_0^{2\pi} \left(\frac{\delta\psi}{\bar{\psi}}\right)\left(\frac{\delta\phi}{\bar{\phi}}\right) \mathrm{d}t \qquad (29)$$

where $\bar{\psi}$, $\bar{\phi}$ and $\bar{\eta}$ are the mean compressor or pump pressure rise, flow coefficient, and efficiency. This shows clearly that, depending on perturbation size, there can be several orders of magnitude difference between controller power and mean turbomachine power. Experimental demonstration of the large differences between device and controller power has been shown, for example, in the use of active control of afterburner oscillations.[15]

The ANMP for the different components depends on system parameters, operating conditions, and control parameters $Z_\xi$ and $Z_A$, since all the perturbation quantities are dynamically linked. Although purely sinusoidal oscillations will actually occur only at the neutral stability point, for simplicity we can envision a situation in which the perturbations in the compressor are driven sinusoidally at other operating points, and examine the energy needed to maintain these oscillations.

Figure 11 shows the ANMP input, over and above the steady-state value, for an uncontrolled system undergoing constant (small) amplitude sinusoidal oscillations. The power is evaluated at a frequency $\omega$ equal to the system natural frequency at instability. Input is indicated as positive for the compressor or pump, while dissipation is plotted as positive for the throttle.

At high mass flow rate, the oscillatory flow leads to net dissipation in both compressor and throttle; but at low flow rates, there is a net mechanical energy input due to the compressor. The energy input and dissipation (averaged over a cycle) balance at the neutral stability point.

The control can affect the power balance in two ways. It can absorb or produce power directly. It can also affect the system dynamics so that the balance between compressor and throttle energy production is changed.

Figures 12a–12d present the ANMP production (or dissipation) for the moving plenum-wall plenum control, for controller amplitude of $Z_\xi = 0.01$, and for phases of 0, 90, 180, and 270 deg, respectively. For 0 and 180 deg, the wall velocity and the pressure perturbations are in quadrature, and there is no net power input. For 90 deg, there is substantial power flow to the controller, while for 270 deg, there is substantial power flow from the controller. Thus, the primary mechanism for stabilization with the moving wall is absorption of power by the controller, since the relative amounts of energy from compressor and throttle are not greatly changed from the no control situation.

A different situation is found if we examine the power flow with the throttle controller. These are plotted in Figs. 13a–13d for $Z_A = 2$, with the same four controller phase angles. As stated, work done by the throttle controller does not impact the system mechanical energy balance, but for reference we have indicated the power input for a case where the throttle volume is 0.01 of compressor volume. The control functions by changing the balance between energy production (in the compressor) and dissipation (in the throttle). This is critically affected by the controller phase, as is clearly seen by comparing Figs. 13a and 13c. (These are actually for cases of no net controller work because wall velocity and pressure are in quadrature.)

Figure 13a shows a high level of dissipation in the throttle, with a consequent increase in system stability. In Fig. 13c, however, the throttle motion is seen to be destabilizing, and instability occurs prior to that with no control, at a condition where the compressor is on the negatively sloped part of its characteristic. The throttle control thus functions differently from the moveable plenum wall control, working instead of increasing the dissipation in an existing dissipative element.

## Discussion and Conclusions

We have examined the use of active control for the suppression of aerodynamic instabilities that limit the useful range of both axial and centrifugal turbomachines. Both local and global instabilities (i.e., incipient rotating stall as well as surge) have been analyzed and shown to be amenable to basic control strategies. In both cases, the stable flow range can be considerably extended.

For axial compressors, rotating stall is the first limiting instability encountered. The onset of the local instability (rotating stall) however, can rapidly trigger the global instability (surge), usually in only a few rotor revolutions. Experiments using passive flow control (screens) have demonstrated that suppression of rotating stall can result in a 15 to 20% increase in stable flow range as well as an increase in peak pressure rise. The results in this paper suggest that this advantage can be realized with relatively little control authority and control power required (several orders of magnitude below the poser levels of the machine under control). However, the control must be properly distributed and implemented.

We have also shown how such a control system might be used to destabilize a machine "trapped" in a stable but undesirable rotating stall—aiding the stall recovery process.

In centrifugal machines, surge is the primary concern, and the analysis in this paper shows how several approaches may be employed for control of this type of instability.

Examination of the energetics of actively controlled compression systems shows that controller power required is determined not by the power level of the machine under control but rather by the strength of the ambient disturbances. Further, the controller need not add or subtract net work from the system but has only to change the balance between the energy production and dissipation process already at work in the device.

Demonstrations of the suppression of turbocharger surge have been conducted using a simple control on oscillation phase for both of the schemes outlined here. These show, we believe, the robustness of the basic theory and the potential for practical application.

Furthermore, preliminary analysis indicates that properly designed structural dynamics may also provide flow stabilization in much the same way as active control.[16]

The control of rotating stall in an axial compressor is more challenging. The frequencies of the instabilities are higher, and the system is mechanically more complicated. control actuation might be the first hurdle. Several approaches look promising, ranging from modification of boundary-layer behavior (using suction, heated strips, acoustics, etc.), tip clearance modification, jet injection, and flaps, to gross ($1 \sim 3$ deg) angular variation of the blading. It is important to note that commercial servo motor technology is adequate for test stand demonstration of the concept, even for the high frequency ($100 \sim 500$ Hz) blade motion.

We believe that the next step is experimental investigation of the concepts presented herein. Questions of signal identification, optimal actuation strategies, impact on machine efficiency, and the actual level of performance increase realizable can only be answered experimentally.

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