ACTIVE CONTROL OF SURGE IN MULTI-STAGE AXIAL-FLOW COMPRESSORS

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ABSTRACT
This paper deals with a theoretical study of the active control of purely axial 1D instabilities in multi-stage axial-flow compressors. The paper briefly considers the development of a suitable surge model and continues with the derivation of a controller using linear optimal control theory. The controller is specifically designed to suppress the instabilities predicted by the linearised form of the surge model. The control technique involves a bleed being dynamically varied in response to fluctuations of variables. It is found that a stabilising optimal controller can always be obtained. The second part of the paper presents a non-linear simulation of the surge prediction model with the linear controller, using a 4th order Runge-Kutta scheme. This demonstrates how some non-linearities in the flow model can affect the controller operation. It is also shown that, the amplitudes and the frequency response required of the actuator to retain stability can exceed the practical limitations.

NOMENCLATURE

A: Flow area
Cₜ: Specific heat at constant temperature
Cᵥ: Specific heat at constant volume
Eₘ: Net energy input
Fₘ: Net axial momentum input
F: Optimal regulator matrix
K: Optimal observer matrix
Nₘ: Number of elements
P: Pressure
T: Temperature
U: Controlling variable: actuator
V: Velocity of air
W: Mass flow rate
WB: Bleed flow rate
X: State vector of non-dimensionalised flow perturbations
x: Axial coordinate
Y: Controlled variable: sensor
ρ: Density
r: Time-lag constant for actuator

Subscripts
a: Axial component of vector
comp: Value computed by controller
ND: Non-dimensional parameter
r: Arbitrary element within the compression system
s: Static condition
T: Total condition

Superscripts
*: Steady-state condition
δ: Small perturbation of variable
^: Estimated variable
T: Transpose of vector or matrix

1. INTRODUCTION:
The operating range of a multi-stage axial-flow compressor is limited by the onset, at low mass flows, of some internal aerodynamic instabilities. These can either develop into rotating stall or surge. Up to now, little experimental evidence has been obtained on the initial stages of compressor instabilities. In consequence, the modelling approaches have concentrated on the two end-products: the most successful surge prediction models, proposed by Elder (1984) and Davis (1987), consider a one-dimensional representation of the flow and generally split the compressor into stages to include individual stage performance and dynamic coupling between stages. The rotating stall models are based on two-dimensional flow equations and the compressor is generally modelled by a single lumped volume (Moore, 1986).

Recently, the idea of actively controlling compressor
instabilities has attracted a great deal of attention. As described by Epstein (1989), it basically consists of introducing an active feedback controller in order to increase system damping. Only a low-power controller is required as the system acts on small perturbations. This new approach has been proven on a series of laboratory models for both rotating stall (Paduano, 1991) and surge (Ffowcs Williams, 1989 & 1990 and Pinsley, 1991). Significant gains in mass flow range have been obtained but these applications were directed towards centrifugal compressors in the case of surge control and towards single-stage axial compressors for rotating stall control. The only study aimed at controlling surge in multi-stage axial-flow compressors gave disappointing results (Hosny, 1991). A common point of all the work mentioned above is the use of a simple proportional feedback as a control law.

This paper presents a new theoretical approach developed for controlling surge in multi-stage axial-flow compressors. The controller is specifically designed to suppress the axial 1D instabilities predicted by the linearised form of a surge prediction model. This model has been selected for its analytical rigor and for the satisfactory surge predictions it gives. Also, the controller is derived from linear optimal control theory which offers more scope than the basic proportional feedback.

The second part of the paper examines the influence of some critical problems largely omitted in previous studies. Although controllers are generally obtained from a linear approach, it has been suggested by Cargill (1990) that some non-linear effects can influence the initial stages of high-speed surge. It therefore seems important to study whether non-linearities in the flow model can affect the operation of the linear controller. This is carried out by using an appropriate non-linear simulation of the controlled surge model. Another problem raised by Simon (1992) concerns the amplitudes and the frequency response required of the actuator to retain stability. The values obtained in this study are confronted to the practical limitations of the actuating system considered.

2. DERIVATION OF A STABILISING LINEAR OPTIMAL CONTROLLER:

2.1. Surge Model:

The objective is to predict the onset of purely axial 1D instabilities within multi-stage compressors and the model is therefore based on the one-dimensional unsteady momentum, continuity and energy equations. It is experimentally found that the off-design mis-matching and the dynamic coupling between stages are critical features of surge. In consequence, the compressor is split into stage elements, i.e. rotor-stator pairs. The compression system modelled here is that of a test bed configuration (Fig 1) where the compressor exhausts into a plenum volume terminated by a throttle valve. The basic equations of the flow through one element can be written in a conservation form as follows:

\[ \int_{x_{r-1}}^{x_r} \frac{\partial \xi}{\partial t} \, dx = (W_{r}^{*} + P_{r}A) - (W_{r-1}^{*} + P_{r-1}A) + E_{\text{net}} \]  

where \( r \) and \( r+1 \) respectively denote the conditions at the \( r \)th element inlet and outlet. \( E_{\text{net}} \) represents the net momentum input to the flow in the axial direction resulting from the combined actions of the blading, the pressure on the walls and the frictional losses. \( E_{\text{net}} \) accounts for the net energy transfer to the flow through shaft work and heat transfer.

Solving the system of differential equations: (1), (2) and (3) for the flow through a multi-stage compressor presents two difficulties: firstly, evaluating the integral terms and secondly, expressing the net momentum and energy input. To address the first problem, the flow parameters on the left hand side of the above equations are assumed to be linearly distributed across every element of the compression system. This corresponds to a 2nd order accurate spatial discretization of the compressor model. The net axial blading force and the energy put-in by each compressor stage are obtained by using the quasi steady-state approximation introduced by Elder (1984): the flow variables at a stage outlet are obtained as those which would occur in a steady situation (i.e. using steady-state stage characteristics) with instantaneous inlet conditions. The characteristics are expressed in the form of temperature-rise and Mach number corrected pressure-rise coefficients. Also, the flow is assumed to be axial at every inter-stage plane as it corresponds to a stator exit.

The condition of clean inlet flow (i.e. no inlet disturbances) provides two boundary conditions: the inlet total pressure and temperature are set to the corresponding ambient static values. The compression system outlet plane is supposed to be choked. This assumption is justified by the value of the total pressure at the exit of a multi-stage compressor and gives the last boundary condition required.

The model is linearised about the equilibrium point of interest. This yields the following linear autonomous system of differential equations of order 2N, N being the number of elements within the compression system:

\[ H \frac{dX_{e}}{dt} = G X_{e} \]  

where \( H \) and \( G \) are constant matrices containing geometrical
therefore considered adequate for the purpose of this study.

2.3. Active Stabilisation of Surge:

The control technique developed below involves the bleed at one plane within the compressor being dynamically varied in response to fluctuations of one flow variable at the same or another plane (Fig 4). The sensed variable considered here is the total pressure but as shown later, it is not a critical choice when using optimal control theory. The bleed port is supposed to be such that the steady-state value of bleed does not modify the compressor stage characteristics (i.e. small steady-state value) and that no flow is ever injected into the compressor (i.e. oscillation amplitudes smaller than steady-state value). This assumption is not restricting in a small-perturbation analysis as the flow perturbations can always be taken as small as required. The new dynamic system obtained when considering non-dimensional bleed flow variations at plane i, $U=(W'_{B}/W'_{n})$, and non-dimensional total pressure fluctuations at plane j, $Y=(P'_{T}/P'_{n})$, is then as follows:

$$
H \frac{dX}{dt} + B_1 \frac{dU}{dt} = GX + B_2 U
$$

where $H$, $G$, $B_1$, and $B_2$ are matrices already mentioned in the initial system (4) and the $X$, $Y$, $U$, and $V$ are state variables. The above system takes a more familiar form:

$$
\frac{dX_1}{dt} = A X_1 + B U
$$

2.2. Validation of Model:

The instability predictions obtained from the surge model are confronted to the experimental surge line of two different seven-stage axial-flow compressors: the Olympus 593's low pressure compressor, typical of large size low-speed machines and the GE 36's high pressure compressor, a smaller high-speed modern compressor. The surge prediction model requires steady-state compressor stage characteristics which are properly defined in the region close to instabilities. In this study, the characteristics are obtained from experimental data but need some extrapolations beyond the experimental range. This is achieved by using a technique specifically chosen for its repeatability and rigor which involves a 2nd order polynomial best fit.

The surge prediction results obtained for the two compressor test cases are shown in Fig 2 & 3. Except for the 5800 rpm speedline of the Olympus 593, the predictions of mass flow rate and pressure ratio are within $2\%$ of the experimental values.

In general, the model seems capable of predicting purely axial 1D instability reasonably close to the actual surge line and is therefore considered adequate for the purpose of this study.

![FIG 2: SURGE PREDICTION RESULTS FOR OLYMPUS 593 LP COMPRESSOR](image1)

![FIG 3: SURGE PREDICTION RESULTS FOR GE 36 HP COMPRESSOR](image2)
(controllability) and if the variations of any flow variable can be
detected from the sensor output (detectability).

It is found that, for the two test cases mentioned earlier, the
system obtained at any equilibrium point on all compressor speed
lines (stable and unstable regions) is always controllable and
detectable. The result is valid for any location of actuators and
sensors and whatever the sensed variable considered. This seems
in agreement with the values of inter-stage Mach numbers in the
two compressors investigated: for all flow conditions, they are
always lower than 1 and so pressure perturbations can propagate
everywhere within the compression system model, ensuring the
controllability and detectability conditions.

<table>
<thead>
<tr>
<th>TABLE 1: EFFECT OF OPTIMAL CONTROLLER ON POLES OF SYSTEM</th>
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<tr>
<td>(Olympus 593 LP compressor, 4500 rpm, W=90kg.s⁻¹)</td>
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<tr>
<td></td>
</tr>
<tr>
<td>OPEN LOOP</td>
</tr>
<tr>
<td></td>
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<tr>
<td>Re(s⁻¹) Im(Hz)</td>
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<tr>
<td>----------------</td>
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<tr>
<td>-4550. +/−6230.</td>
</tr>
<tr>
<td>-710. +/−2380.</td>
</tr>
<tr>
<td>+1.24 +/−525.</td>
</tr>
<tr>
<td>+64.4 +/−319.</td>
</tr>
<tr>
<td>+38.3 +/−175.</td>
</tr>
<tr>
<td>-1610. +/−634.</td>
</tr>
<tr>
<td>-602. +/−72.5</td>
</tr>
<tr>
<td>+121. +/−48.0</td>
</tr>
<tr>
<td>-3020. 0.0</td>
</tr>
<tr>
<td>-634. 0.0</td>
</tr>
</tbody>
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Having a controllable and detectable open loop system, a closed
loop system of more attractive features can then be designed.
Firstly, the eigenvalues with a positive real part, responsible for
the instability of equilibrium points beyond the surge line, can be
shifted to the left hand side part of the complex plane
(stabilisation). When only considering axial 1D instabilities, this
allows the stable region of compressor operation to be extended
to lower mass flows without any constraints. Moreover, the use
of linear optimal control theory positions the eigenvalues in such
a way that the following expression is minimised:

\[ \int_t^\infty [X^T(\tau)R_1X(\tau) + U^T(\tau)R_2U(\tau)]d\tau \]  \hspace{1cm} (8)

where \( R_1 \) and \( R_2 \) are positive-definite symmetric weighting
matrices to be specified. This provides an optimisation criterion
between the rapidity with which any initial state, \( X_0 \), is reduced
to zero and the amplitudes of the input variable associated with it.

As an example of active stabilisation, Table 1 shows how the
poles of the open-loop system at an unstable equilibrium point
(mass flow of 90 kg.s⁻¹ at 4500 rpm for the Olympus 593 LP
compressor) are shifted by the linear optimal controller. The
controller developed here minimises the oscillating amplitudes of
the actuating variable (\( R_1=1 \)d and \( R_2=10^4 \) for both observer
and regulator problems). In this particular design, it has been shown
by Kwakernaak (1972) that the eigenvalues of the closed-loop
system are duplicate pairs of conjugate complex numbers based
on the open-loop system eigenvalues: the negative real part
eigenvalues and the positive real parts with an opposite sign.

3. NON-LINEAR SIMULATIONS OF CONTROLLED AND UNCONTROLLED SURGE MODEL:

3.1. Limitations of the Linear Approach:
In the last section, a controller was specifically designed to
suppress the purely axial 1D instabilities predicted by a linearised
dynamic flow model. The main limitation of this linear approach
is that it only considers the stability of the system to infinitely
small perturbations, i.e. asymptotic stability. In reality, however,
the axial 1D disturbances resulting from the natural unsteadiness
of the flow or those caused by stalling rows present some finite
amplitudes. In such a situation, the non-linearities of the physical
problem, which have been neglected so far, might prevent the
controller from operating satisfactorily. Also, the amplitudes
required from the actuator to suppress finite-amplitude
perturbations might be too large for the practical system
considered. It therefore appears important to simulate the non-
linear time-response of the controlled surge model to some initial
perturbations of finite amplitude.

The controller and its sub-systems described in the first part of
the paper are all ideal devices: all the physical time-lags
associated with the sensor, the controller itself and the actuator
have been neglected. In practice, the actuator bandwidth is the
most restricting parameter as a real bleed valve cannot operate at
very high frequencies (e.g. 1000 Hz). This practical limitation
might seriously affect the controller operation, especially when
some high unstable frequencies are to be controlled.

3.2. Numerical Technique:
The linearly-distributed parameter assumption used in the derivation of the surge model leads to a non-linear system of ordinary differential equations. The linear controller developed in the first part of the paper introduces additional ordinary differential equations (7). In consequence, simulating the time-response of the controlled surge model to various initial conditions only requires a time-integration technique. As shown in Table 1, the spectrum of eigenvalues for the linearised controlled surge model presents some very large imaginary values (frequencies) and widely distributed real parts (decay rates).

The numerical technique developed by Elder (1972) for simulating dynamic surge models is based on the 4th order Runge-Kutta scheme. This non-linear explicit scheme allows sufficiently large time-steps and is therefore appropriate to the simulation of the controlled surge model. All the numerical examples presented below are obtained from this technique.

3.3. Non-Linear Effects:

The set of discretized equations obtained for all elements of the compression system and the boundary conditions previously mentioned form an initial boundary value problem. In order to model flow perturbations in the compressor, finite-amplitude deviations from the steady-state matched flow conditions are introduced as initial conditions. These initial disturbances are imposed on the static pressures and velocities at all planes within the compression system.

Fig 5 shows how initial flow disturbances of 0.5% can affect the stability of the controlled surge model. For each speed line, the locations of both the actuator and the sensor are chosen in such a way that the largest stability improvement is obtained. The stable range of the controlled compressor, which extends to the extreme left of the extrapolated characteristics in the linear approach, is now dramatically reduced in this non-linear simulation. In fact, a significant gain in stability can only be obtained for the lowest speed (4500 rpm or 70% of design speed). The non-linearities in the multi-stage axial-flow compressor model therefore seem to play an important role for other speeds and badly affect the operation of the linear controller. The initial perturbations considered in this example, i.e. 0.5%, appear very small in relative terms but the absolute initial deviations in static pressures and velocities can respectively reach 2 kPa and 2 m/s. These values correspond to annulus-averaged internal flow disturbances and might be typical finite-amplitude perturbations in a multi-stage axial-flow compressor. However, little experimental data is available on internal flow disturbances and the numerical examples given in the paper must be taken as a broad indication of what might happen in reality.

Considering very small-perturbations can also limit the stability improvements obtained with the linear controller. The new surge line resulting from initial flow disturbances of 0.01% is given in Fig 6: apart from the low speed line, the extension in stable mass flow range is reduced by about half of its value in the linear approach. Except for the 4500 rpm speed line, the controlled surge model is therefore greatly sensitive to even small perturbations. Non-linear simulations of the uncontrolled surge model are described in paragraph 3.6 in order to show the origin of these non-linear instabilities.

3.4. Actuator Amplitudes:

Fig 7 shows the dynamic variations of bleed in response to initial perturbations of finite amplitude (0.5%). This is the phase during which non-linear effects can arise. Fifty milliseconds after the introduction of disturbances, the bleed variations required to stabilise the system are too large for the practical system considered. The system then converges to the zero-perturbation state. The use of optimal control theory minimises the actuator amplitudes during the transient but the maximum amplitude (2.0% here) still might be too large for the practical system considered.

The maximum bleed amplitudes are very sensitive to the initial level of disturbances within the compression system. Also, they strongly depend on how far in the naturally unstable region is the equilibrium point to be stabilised. Fig 8 and 9 respectively present the results obtained with initial perturbations of 0.5% and
0.01%. The combination of actuator and sensor locations which minimises actuator amplitudes for each speed line is indicated on the two graphs and used in the simulations. Optimum locations turn out to be close-coupled solutions in proximity of the "most heavily" stalled stage, i.e. the stage with the steepest positive slope. These optimum locations can therefore slightly vary along the compressor global characteristic as the situation of stalled stages evolves. For the two highest speeds and for the equilibrium points successfully stabilised, only the last stage is very severely stalled and optimum locations of actuators and sensors are naturally at the rear of the compressor. For the 5300 rpm speed, the third stage is the most severely stalled and the control is optimally located close to the third stage. For the lowest speed and for equilibrium points deep in the unstable region, all stages are stalled with the third and sixth stages marginally more stalled. In this case, the optimum locations of the actuators and sensors appear to be near the fifth stage. The controller weighting matrices do not have a significant effect on the results as long as $R_2$ is greater than 1 ($R_1=1d$).

The bleed amplitudes resulting from very small perturbations (0.01%, Fig 9) never attain 1% of the steady-state mass flow through the compressor. Small actions of this type do not seem to cause any serious concern for the actual system implementation. However, when considering initial perturbations of finite amplitude (0.5%, Fig 8), the maximum amplitudes of the actuating variable can reach 7% of the steady-state mass flow rate for equilibrium points deep in the unstable region. These are large values which seem to exceed the maximum amplitudes acceptable for a bleed valve.

The actuator contemplated here is a choked bleed port with a dynamically variable throat area. This solution is chosen as it seems to be a simple arrangement very appropriate to multi-stage axial-flow compressors. In this design, any small change in bleed valve area is related to bleed variations as follows:

$$\frac{\Delta A}{\Delta^*} = \frac{\delta A}{\delta^*}$$

(9)

The amplitudes of variations of this actuating system are limited by the closing of the valve ($\Delta^* = -1$). In this extreme situation, the dynamic bleed variation is equal to the steady-state bleed, i.e. $\delta^* = -\delta^*$. In consequence, the maximum bleed variation practically attainable with such a system is limited by the maximum acceptable value of steady-state bleed. This value must be kept very low for two main reasons: firstly, the steady-state bleed strongly influences the size of a bleed valve. A large valve would tend to be slow to operate and might therefore introduce very restricting frequency limitations. Secondly, a non-negligible steady-state bleed flow rate would probably incur some steady-state performance penalties and as a result, the bleed should be minimised. The practical limitations associated with the bleed valve described here might be eased by a combination of flow injection and bleed. In this case, no steady-state bleed would be needed but the practical implementation of such a system might turn out to be more difficult.

3.5. Actuator Cut-Off Frequency:

The frequency limitations of the actuating system are modelled by a 1st order time-lag differential equation as follows:
where \( WB_{\text{comp}}' \) is the bleed flow variation computed by the controller and \( WB' \) is the actual bleed flow variation. This equation introduces a simple low-band frequency filter with a cut-off frequency of \( 1/r \).

When the above equation is incorporated into the controlled surge model, the frequency response required of the actuator to retain stability varies with the operating point considered. For example, on the 4500 rpm speed line of the Olympus 593 LP compressor, the maximum cut-off frequency beyond which the controller does not operate satisfactorily is 600 Hz at a mass flow of 100 kg.s\(^{-1}\) whereas it is 5 KHz at 90 kg.s\(^{-1}\).

This situation can be explained by looking at the unstable frequencies predicted by the linearised analysis. For unstable equilibrium points close to the surge line, the dynamic flow model predicts only one undamped frequency which is of 50Hz at a mass flow of 100 kg.s\(^{-1}\). When considering equilibrium points deeper in the unstable region, some additional higher frequencies become unstable: at a mass flow rate of 90 kg.s\(^{-1}\), the model shows five unstable frequencies with the highest one of about 800 Hz.

The actuator bandwidth limitation may therefore affect the controller operation. This happens when the actuator cut-off frequency is close to an order of magnitude higher than the maximum unstable frequency. It is a major problem for equilibrium points deep in the unstable region at the very low speed where some instabilities are characterised by high frequencies which apparently have to be controlled.

### 3.6. Origin of Non-Linear Instabilities:

Non-linear simulations of the uncontrolled surge model are carried out at different equilibrium points beyond the predicted surge line and the results are then compared to an eigenvalue analysis derived from the linearised surge model. The objectives are to look at the initial stages of surge (transient of less than 50 ms) and to see when and how non-linearities appear in the non-linear system. This helps understand why the controlled Olympus 593 LP compressor is so sensitive to very small perturbations at all speeds except the 4500 rpm speed line.

For all speed lines, unstable (uncontrolled) equilibrium points in close proximity to the surge line, the dynamic flow model predicts only one undamped frequency which is of 50Hz at a mass flow of 100 kg.s\(^{-1}\). When considering equilibrium points deeper in the unstable region, some additional higher frequencies become unstable: at a mass flow rate of 90 kg.s\(^{-1}\), the model shows five unstable frequencies with the highest one of about 800 Hz.

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Figure 10 presents an example of non-linear simulation with a small initial perturbation in which a gradually developing instability will be noted. Fourier analysis of this signal accurately indicated the presence of the five unstable frequencies (from 48 Hz to 808 Hz) predicted by the linearised model.

For other speeds, the linearised model predicts only two conjugate unstable eigen-values which have decreasing frequency and rapidly increasing growth rate as the operating point considered is deeper in the stall domain (for the 5350 rpm speed this eventually leads to static instability with zero imaginary part and exponential growth of perturbations). The main feature here is the very high growth rate for perturbations which considerably increases the difficulty of the controller by quickly introducing non-linear effects.

For instance, Fig 11 shows the flow transient resulting from very small perturbations (0.01%) at 130 kg/s for the 5350 rpm speed line. The pressure fluctuations at the third plane of the compression system quickly diverge (after only 1.5 ms) with a purely exponential growth indicated by the linearised analysis. However, this fast growth of flow disturbances rapidly incurs large deviations from the steady-state conditions (5% 1 ms later, \( t=2.5 \text{ ms} \)) which then trigger some non-linear effects halting the exponential growth (after 4 ms). In this example, the non-linear effects seem to appear only 2 ms after flow perturbations become significant. The linear optimal controller described in the paper is a dynamic controller (see equation (7) for full-order observer)
and therefore presents some intrinsic "analytical" time-lags. These exist independently of the physical time-lags described in the last section. It seems very likely that this type of controller cannot cope with the fast diverging transients which quickly introduce non-linearities into the system.

Having established the origin of non-linear instabilities in the controlled surge model, it remains now to explain why the system dynamics at the very low speed (low growth rate, several undamped frequencies) are different from those at other speeds (high growth rate, single low frequency or static instability). Looking at steady-state stage characteristics for equilibrium points deep in the unstable region, it appears that for the 4500 rpm speed line, all stages are stalled but the positive slopes of pressure-rise characteristics have all moderate values. For other speeds, the situation is somewhat different as there is always one stage which presents a very steep positive slope (stage 3 for 5350 rpm, stage 7 for the two high speeds). This result illustrates how blade stall, in our model, affects the system dynamics through the slope of the steady-state stage pressure-rise characteristics. The analysis also demonstrates how high speed multi-stage compressors with large mis-match potential at off-design and restricted operating ranges due to the high Mach numbers lead to compression systems which are much more difficult to actively control.

4. CONCLUSIONS AND DIRECTIONS FOR FURTHER WORK:

A one-dimensional stage-by-stage surge prediction model has been shown to predict instabilities very close to the actual surge line. A controller was specifically designed to suppress the growth of infinitely small perturbations predicted by the linearised version of this surge model. As a result, the active control of axial 1D instabilities within multi-stage axial-flow compressors has been theoretically achieved, using a bleed flow valve actuation system and a linear optimal controller.

However, some practical problems have been shown to significantly limit the extension of stable flow range obtained from the controller. Firstly, the non-linearities present in the flow model can prevent the linear controller from operating satisfactorily. It happens at all speeds but the lowest speed and the situation worsens as the level of annulus-averaged flow disturbances increases. This problem results from the very high growth rate of instabilities which introduces non-linear effects into the system before the controller can act on disturbances. High growth rates of perturbations seem to be directly linked to the steepness of pressure-rise characteristics and therefore illustrate well the difficulties of actively controlling instabilities in multi-stage axial-flow compressors. Secondly, some large amplitudes may be required from the actuator in order to stabilise equilibrium points deep in the unstable region. These amplitudes can exceed the practical limitations of the actuation system considered. This result again depends on the actual level of internal flow disturbances within multi-stage axial-flow compressors which is not precisely known. Finally, the frequency limitations of the actuator can be too restricting: this occurs deep in the unstable region at low speed where some high unstable frequencies are predicted by the model.

When considering all these practical limitations, the compression system cannot be stabilised very far in the unstable region for both low and high speeds. The results are very dependent on the level of initial flow perturbations and how realistic is the surge model. In consequence, more analytical and experimental work is required before an active control scheme can be safely implemented on a multi-stage axial-flow compressor. The surge model has to be further validated by comparing the frequencies and transients predicted by the model to some experimental values. Also, it seems important to evaluate the actual level of internal flow perturbations within a multi-stage compressor. Finally, some analytical work involving more robust controllers and less restrictive actuating systems is also required.

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