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COMPRESSOR SURGE CONTROL WITH AMB ACTUATION

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ABSTRACT

We examine the stabilization of surge for an unshrouded centrifugal compressor by modulating its impeller tip clearance using Active Magnetic Bearings (AMBs). The thrust AMB actuates on the position of the compressor impeller, thus changing the clearance between the impeller and the shroud. This affects the performance and efficiency of the centrifugal compressor. First, using only the pressure ratio information for feedback, we derive a nonlinear control law that stabilizes the compression system with the AMB dynamics ignored. Second, we include the AMBs into our analysis, and we observe from simulations that the derived nonlinear control is not able to stabilize surge due to the extra dynamics introduced by the magnetic bearings. A similar phenomenon occurred in our simulation when the controller designed [1], which used both pressure ratio and mass flow but also ignored the AMB dynamics. We are then motivated to design control laws that are more robust to AMB dynamics. We rely on linear robust control theory to derive a stabilizing H_∞ controller in the presence of uncertain actuator dynamics. Simulation shows that the H_∞ controller is able to stabilize the compression system in the presence of the AMB dynamics.

INTRODUCTION

Surge is a dynamic instability that occurs in compression systems and is characterized by large-amplitude low-frequency oscillations of the pressure ratio and flow rate. Surge occurs as the mass flow through the compressor is reduced to a critical point where the flow pattern becomes unstable, with even flow reversal in severe cases. This can cause high exhaust temperature, heavy impeller load and serious compressor damage, restricting the stable operational range and performance of centrifugal compressors.

Over the past few years, much effort has been made by the research community in the area of surge control. Greitzer in [2] introduced a lumped parameter model of the compression system consisting of compressor, plenum, and throttle valve. This model has now become the standard for designing active and passive surge and stall controllers. The Greitzer model was adopted in many published works in control of surge using linear [3, 4] and nonlinear systems and control theory [5, 6]. Jungowski [7] and Hagino [8] latter showed that this model is also applicable for system with the exhaust pipe acting as plenum volume.

Arnulfi et. al presented in [9] a brief review of recent published works on both active and passive surge control. Some of the most common actuators for active surge control are close-coupled valves, as presented by Gravdahl and Egeland [10], and throttle valves as shown in [11]. Senoo in [12] and Senoo and Ishida in [13] studied the effect of varying the clearance between the compressor impeller and the shroud on the performance of centrifugal compressors. Sanadgol [1] proposed to stabilize the compression system in the presence of surge by modulating the compressor impeller tip axial clearance using active magnetic bearings (AMBs). However, the derived control law relies heavily on flow rate feedback, which is not readily available on many industrial compressors with the bandwidth and precision required for controls. Additionally, the author gave limited consideration to the dynamics of the magnetic bearings when proving the stability of the compression system, despite that it plays the important role of actuating the impeller position.

Here, we revisit the problem of surge stabilization through tip clearance modulation as considered in [1] and offer possible solutions to the issues discussed above. First, we propose a control law that relies only on pressure feedback to control surge for the ideal compression system, where the actuator dynamics are ignored. In this case, a nonlinear controller is derived to stabilize the plant in the presence of surge. Next, we study the stability of the compression system in the presence of the AMBs. We observe that the derived nonlinear controller cannot stabilize the augmented system. Indeed, even with both flow rate and pressure ratio available for feedback, we show in simulation that the controller of [1] would fail to stabilize surge when the AMB dynamics are present. Therefore, we include the dynamics of the AMBs as uncertainties acting on the actual impeller position. Relying on linear robust control theory, we derive a pressure feedback H_∞ controller that stabilizes the augmented Compression-AMBs system. Finally, performance and robustness of both controllers are compared to the results in [1] through simulation.

COMPRESSION SYSTEM MODEL

In order to study the effects of surge and the active control of this dynamic instability, a full size compressor test rig is being built at the Rotating Machinery and Controls (ROMAC) Laboratory at the University of Virginia. The test rig, shown in Figs. 1 and 2, consists of an unshrouded single-stage centrifugal compressor on active magnetic bearings and a modular exhaust ducting system. The throttle valve controls the mass flow and pressure ratio in the compression system, and it can be moved along the exhaust piping in order to simulate a variable size plenum volume. The axial position of the compressor impeller, and thus the clearance between the impeller tip and the compressor shroud, can be controlled by the magnetic thrust bearing. Table 1 summarizes the design and performance parameters of the centrifugal compressor.

Table 1: Compressor design parameters

Parameter	Value
Design speed (RPM)	23,000
Design mass flow rate (kg/sec)	0.833
Design pressure ratio	1.68
Inducer hub diameter (mm)	56.3
Inducer tip diameter (mm)	116.72
Impeller tip diameter (mm)	250

The mathematical model derived by Greitzer [2] of a compression system composed of compressor,

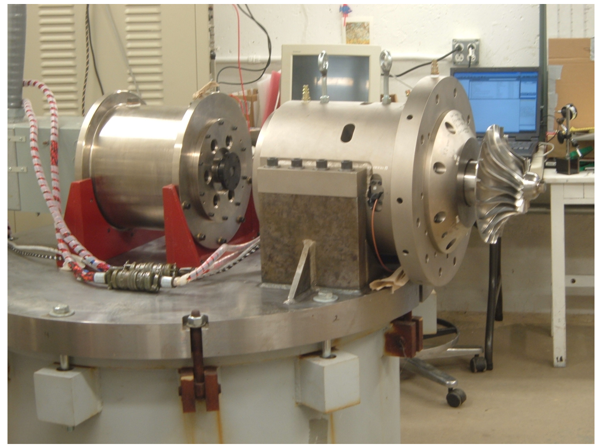

Figure 1: Single stage unshrouded centrifugal compressor on active magnetic bearings, under construction at the ROMAC Laboratory, University of Virginia.

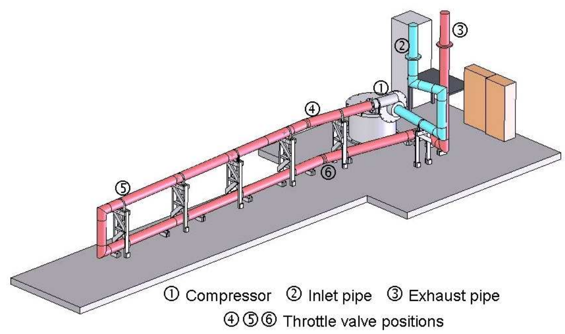

Figure 2: The active surge control test rig consists of a single-stage unshrouded centrifugal compressor and modular ducting system acting as a variable size plenum volume.

plenum volume and throttle valve as shown in Fig. 3 is used to represent the dynamics of our compression system. The compression system dynamics are represented by the following set of equations,

$$\dot{\Psi}_p = \frac{\omega_H}{B}(\Phi_c - \Phi_{th}) \quad (1a)$$

$$\dot{\Phi}_c = B\omega_H(\Psi_c - \Psi_p) \quad (1b)$$

where B is the Greitzer parameter and ω_H is the Helmholtz frequency for the compressor. The states Ψ_p and Φ_c are the non-dimensional plenum pressure ratio and compressor mass flow rate, Ψ_c is the non-dimensional compressor pressure ratio and Φ_{th} is the non-dimensional throttle mass flow rate.

A modification of this model was introduced in [1] to include the effects of tip clearance variation. The

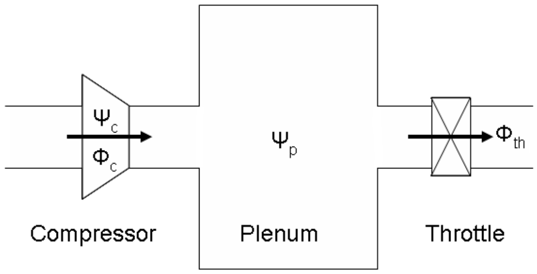

Figure 3: Compression system with compressor, plenum and throttle valve.

resulting plant equation is given as follows,

$$\dot{\chi} = \frac{\omega_H}{B}(\xi - \tilde{\Phi}_{th}) \quad (2a)$$

$$\dot{\xi} = B\omega_H(\tilde{\Psi}_{c,ss} + k_{cl}\delta_{cl} - \chi) \quad (2b)$$

The constant k_{cl} is the tip clearance coefficient and is given by the geometry and operating condition of the compressor. The states χ and ξ are the variations of the plenum pressure ratio Ψ_p and compressor mass flow rate Φ_c from their respective equilibrium values found on the compressor characteristic curve. The control input δ_{cl} is the variation of the impeller tip clearance from its design value and its magnitude is limited by the maximum possible displacement of the compressor impeller. Finally, $\tilde{\Psi}_{c,ss}$ and $\tilde{\Phi}_{th}$ are the differences of the steady state compressor pressure ratio and throttle mass flow rate Φ_{th} from their respective equilibrium values.

PRESSURE FEEDBACK CONTROL IN THE ABSENCE OF AMB DYNAMICS

In this section we consider the stabilization of the ideal compression system, where the dynamics of the magnetic bearings are ignored. The objective of the active surge controller is to maintain the operating condition of the compression system on its characteristic curve. The compression system is subject to unknown downstream flow disturbances, which are modeled as changes in the percentage throttle opening u_{th}. They set the equilibrium operating point of the compressor, which in turn affects the throttle mass flow rate and the steady state compressor pressure ratio. Thus $\tilde{\Psi}_{c,ss}$ and $\tilde{\Phi}_{th}$ are approximated as

$$\begin{aligned} \tilde{\Psi}_{c,ss}(\xi, u_{th}) &= \Delta_c(u_{th})\xi, \\ \tilde{\Phi}_{th}(\chi, u_{th}) &= \Delta_{th}(u_{th})\chi \end{aligned}$$

where Δ_c and Δ_{th} are norm bounded uncertainties, whose values depend on u_{th}.

Using the system output χ and its derivative as states, a local coordinate transformation as described in [14] leads to the following system

$$\begin{aligned} \dot{\zeta}_1 &= \zeta_2 \quad (3) \\ \dot{\zeta}_2 &= \omega_H^2 \left[\Delta_c\left(\frac{B}{\omega_H}\zeta_2 + \Delta_{th}\zeta_1\right) + k_{cl}\delta_{cl} - \zeta_1\right] \\ &\quad -\frac{\omega_H}{B}\Delta_{th}\zeta_2 \end{aligned}$$

where states $\zeta_1 = \chi$ and $\zeta_2 = \dot{\chi}$ are easily accessible. The problem is thus a state feedback control problem. Implementing the backstepping design procedure on (3), we obtain the following nonlinear control law

$$\begin{aligned} \delta_{cl}(\zeta) = \frac{1}{k_{cl}} \Big[&-k(\zeta_1 + \zeta_2) \quad (4) \\ &\times \left(B^2\zeta_2^2 - \frac{1}{B^2}\zeta_2^2 - \omega_H^2\zeta_1^2 - \frac{\lambda+1}{\omega_H^2 k}\right) + \zeta_1\zeta_2\Big] \end{aligned}$$

where λ and k are controller parameters.

Define a Lyapunov function V

$$V(\zeta) = \frac{1}{2}\zeta_1^2 + \frac{1}{2}(\zeta_1 + \zeta_2)^2 \quad (5)$$

Then, the derivative of V along the trajectory of the closed loop system satisfies

$$\begin{aligned} \dot{V} \leq &-(1 + \omega_H^2)\zeta_1^2 - \lambda(\zeta_1 + \zeta_2)^2 + \quad (6) \\ &\frac{1}{4k}(\|\Delta_c\|_\infty^2 + \|\Delta_{th}\|_\infty^2 + \|\Delta_c\Delta_{th}\|_\infty^2) \end{aligned}$$

which shows that the system states can be brought to an arbitrary small region around the origin by increasing the value of k. Therefore, it shows practical stability of the closed loop system in in the presence of norm bounded uncertainties Δ_c and Δ_{th}.

Figure 4 shows the simulation results for the compression system with AMB dynamics ignored. The simulation spans over part of the stable operation region of the compressor by the opening of the throttle valve from 70% (stable) to 20% (unstable). We see that the proposed nonlinear controller with $\lambda = 3$ and $k = 1$ successfully stabilizes the compression system on the characteristic curve, and its performance is comparable to the results in [1].

We next simulate the compression system with AMB dynamics included in the model and repeat the previous simulation test. The thrust AMB dynamics, which is controlled independently by the surge controller, is approximated by a 3rd order low-pass Butterworth filter with bandwidth of 60Hz. Fig. 5 shows the simulation for the augmented system with the previous nonlinear controller. In this simulation, the compressor enters surge at around 40% throttle opening. The position error between the desired and actual impeller position introduced by the thrust bearing destabilizes the compression system. In the

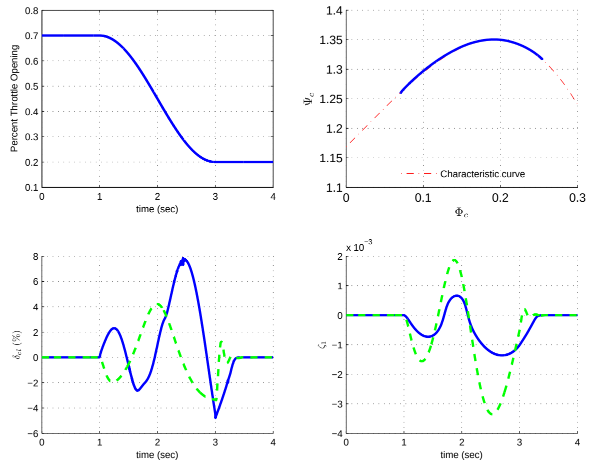

Figure 4: Throttle percentage opening, compressor characteristic curve, impeller tip clearance variation, and plenum pressure ratio variation for compression system with nonlinear pressure control (solid) vs. controller from [1] (dashed).

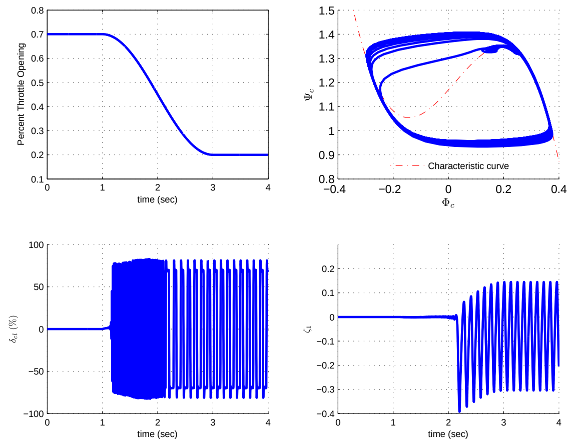

Figure 5: Throttle percentage opening, compressor characteristic curve, impeller tip clearance variation, and plenum pressure ratio variation for augmented Compression-AMB system with the nonlinear pressure control.

next section, we derive a robust active surge controller that stabilizes the compression system in the presence of impeller tracking error introduced by uncertain AMB dynamics.

ROBUSTNESS TO AMB DYNAMICS

Previously, we assume that the impeller displacement in the compressor perfectly followed the desired position calculated by the surge controller. However, the axial displacement of the impeller is actuated by the magnetic thrust bearing, which has its own limitations and uncertainties. Furthermore, it is assumed that the bearing is controlled independently by the surge controller. Thus, we need to assure that their interaction does not destabilize the compression system, which is achieved through robust control theory. In this section, we consider the AMB dynamics as a bounded model uncertainty, and study its effect on the stability of the overall system.

The compressor rotor in our test rig is radially supported by two magnetic bearings with E-core geometry and rate load capacity of 1414 N per quadrant at an operating flux density of 1.25 tesla. Also, axial support is provided by the magnetic thrust bearing, which has been designed to satisfy a rate load capacity of 6600 N and slew rate of 2×10^6 N/s. The calculated maximum steady state axial load on the impeller is 3300 N.

Robustness Condition

Linear robust control is preferred for this application since it is simpler and better understood than its nonlinear counterpart. In order to linearize the system (3), Δ_c and Δ_{th} are evaluated at a nominal value of u_{th}. Then, the system equation becomes linear, and the linearization is valid for the compressor operational range. Let the linearized compression system be represented on the frequency space by a transfer function matrix $G_C(s)$.

Assume that the thust AMB actuator is controlled independently by the surge controller, and the resulting stable closed-loop transfer function from reference to actual impeller position is given by the transfer function $H(s)$. Additionally, let $K_C(s)$ be a stabilizing linear active surge controller derived from the linearized system equation $G_C(s)$. The loop transfer function of the augmented compression-AMB system is $G_C(s)H(s)K_C(s)$, where the desired impeller clearance from $K_C(s)$ is fed to the AMB, and its output is the actual impeller position to the compression system. Then, by reordering the loop transfer function, it can be shown by the small gain theorem [15] that the augmented system is stable if and only if

$$||T_i(j\omega)(I - H(j\omega))||_\infty < 1 \tag{7}$$

where $T_i(s) \stackrel{\text{def}}{=} (I - K_C G_C(s))^{-1} K_C G_C(s)$ is the input complementary sensitivity function of the closed-loop compression system, or

$$||T_i(j\omega)W_2(j\omega)||_\infty < 1 \tag{8}$$

for a stable, proper transfer function $W_2(s)$, such that $|W_2(j\omega)| > |I - H(j\omega)|$ for all frequencies ω. Therefore, $(I - H(s))$ is the sensitivity function of the thrust AMB, and $W_2(s)$ is the bound on the maximum tolerable impeller error caused by the thrust AMB dynamics over frequency range for the closed-loop compression system to remain stable.

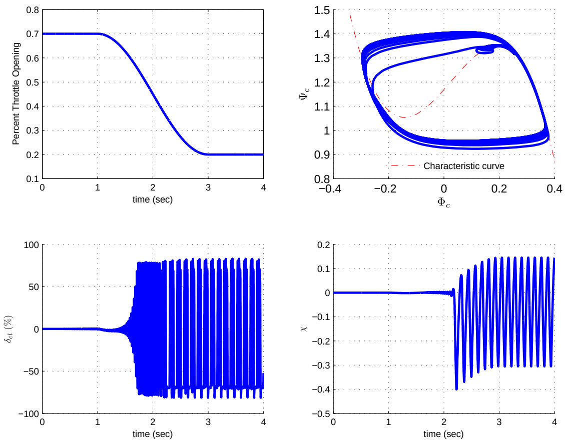

Figure 6: Throttle percentage opening, compressor characteristic curve, impeller tip clearance variation, and plenum pressure ratio variation for augmented Compression-AMB system with the controller from [1].

H_∞ Controller Synthesis

In order to find the weighting function $W_2(s)$, we approximate the closed-loop thrust bearing dynamics $H(s)$ as a 3rd order low-pass Butterworth filter with bandwidth of 60Hz, which is reasonable for our type of bearing. Define $T_o(s)$ to be the output complementary sensitivity function of the compression system,

$$T_o(s) \stackrel{\text{def}}{=} (I - G_C K_C(s))^{-1} G_C K_C(s)$$

Through the interconnection scheme described by the block diagram in Fig. 7, where $W_1(s)$, $W_2(s)$, $W_3(s)$ and $W_4(s)$ are stable and proper weighting functions, we synthesize an H_∞ controller such that it minimizes the H_∞-norm of the weighted closed-loop system $T_{zw}(s)$.

$$T_{zw} = \begin{bmatrix} -W_2 T_i W_3 & W_2(I - T_i) K_C W_4 \\ W_1(I - T_o) G_C W_3 & W_1 T_o W_4 \end{bmatrix} \quad (9)$$

Note that the stability condition (8) for the compression-AMB system appears explicitly in T_{zw} when $W_3 = 1$, and the augmented system is stable if $||T_{zw}(j\omega)||_\infty < 1$. The weighting functions $W_1(s)$ and $W_4(s)$ are selected accordingly to obtain other desired system characteristics such as bandwidth and control input magnitude limits.

Simulation Results with Robust Control

Figure 8 shows the performance of the derived robust surge controller by repeating the simulation test presented in the previous section of the throttle valve closing from 70% to 20% with the augmented compression-AMB system. The dynamics of the thrust bearing from the desired to actual impeller

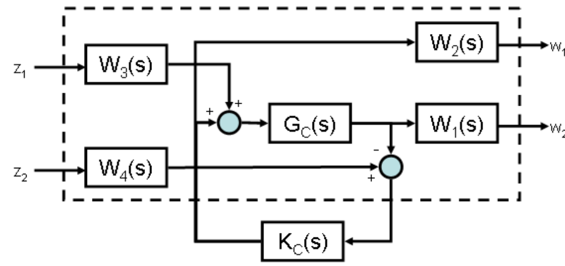

Figure 7: Block diagram of the weighted feedback system for the H_∞ synthesis, where where $G_C(s)$ is the compression system, $W_1(s)$ is the weight on plant output, $W_2(s)$ is the weight on the control input, $W_3(s)$ is the weight on the disturbance, $W_4(s)$ is the weight on the reference signal, and $K_C(s)$ is the active surge controller.

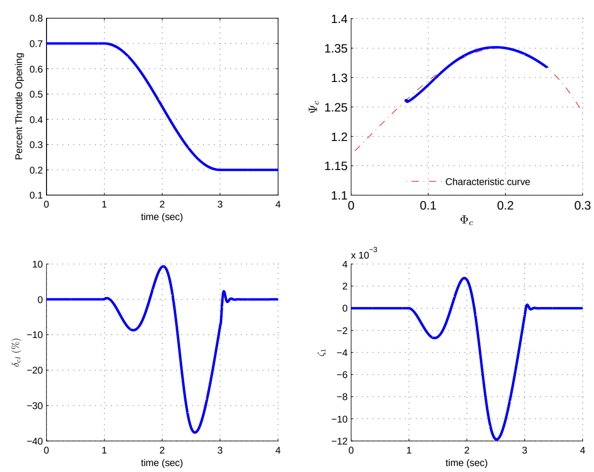

Figure 8: Throttle percentage opening, compressor characteristic curve, impeller tip clearance variation, and plenum pressure ratio variation for augmented compression-AMB system with the robust H_∞ surge control.

position is approximated using the same $H(s)$ as in the controller synthesis. Comparing to the results in Figs. 5 and 6, the H_∞ controller stabilizes the compression system in the presence of AMB dynamics, which matches our analysis. Additionally, the magnitude of the required actuation of δ_{cl} is larger than the previous nonlinear controller, but it remains within a reasonable limit of 40% of maximum allowable impeller displacement.

CONCLUSIONS

Active surge control through impeller tip clearance modulation using magnetic bearings was examined, and issues on the unavailability of the compressor mass flow rate measurement for feedback and system robustness to uncertain actuator dynamics were analyzed. For the first, a nonlinear active controller that only relies on plenum pressure measurement to control surge was derived, and we showed practical stability of the system in the presence of unmodeled downstream flow disturbance.

On the other hand, when the thrust AMB dynamics were added to the system as model uncertainty, both the derived nonlinear pressure feedback controller and the surge controller given in [1] failed to stabilize the augmented plant. Linear robust control provided us with the necessary tools to synthesize an H_∞ controller that stabilizes the augmented compression-AMB system. Simulation showed that, while both controllers derived in this work successfully stabilize the compression system without flow measurement, robust control is necessary for stabilization when the AMB dynamics are present.

Future work include implementation of various surge controllers in the test rig, and comparing the simulation results to experimental measurements. Commissioning of the compressor test rig and validation of the compression system equations (2) are important steps toward this objective. Finally, further research in nonlinear robust control scheme is necessary to fully exploit the potential of the magnetic bearings in controlling surge.

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