Model-assisted optimal control framework for industrial system coupling problems

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CHAPTER 5

Case Study of Optimization Problems under System-Working Medium Coupling Conditions

This chapter takes the safe coordinated control of a centrifugal compressor as a case study, aiming to explore optimal control strategies under strong coupling between an industrial system and its working medium. To accurately capture the nonlinear dynamic behavior and time-delay effects between the compressor system and the gas medium, a fast simulation model is developed based on the Greitzer lumped-parameter model, and an online optimization framework is constructed using the CMA-ES. By integrating the safety quantification provided by the physical model with the real-time parameter optimization of the controller via the evolutionary algorithm, a model-assisted optimal control strategy is formed, enabling the efficient and coordinated optimization of multivariate Proportional-Integral-Derivative (PID) controller parameters. This approach effectively avoids the computational burden associated with traditional optimization methods caused by strong system nonlinearities and high-dimensional parameter spaces, significantly enhancing the safety and robustness of the control process.

5.1 Background

Fluid machinery is the core equipment in the energy conversion process, playing a critical role in the energy industry. As a typical fluid machinery, the centrifugal compressor is a vital component in refrigeration, aviation, and power generation (Ju et al., 2021; Liang, 2017). It compresses working fluids by converting kinetic energy into internal energy of the gas. We show the basic structure of a representative compressor control system in Figure 1. In practice, to meet varying load and efficiency requirements, closed-loop controllers are commonly employed to tune their operating conditions **in real-time** (essentially an online optimization task) while preventing the system from the **overshooting problem** – which often leads to the surge of the compressor and hence damages the machinery. Therefore, it is of vital importance to devise a **real-time, safe control strategy** for this problem.

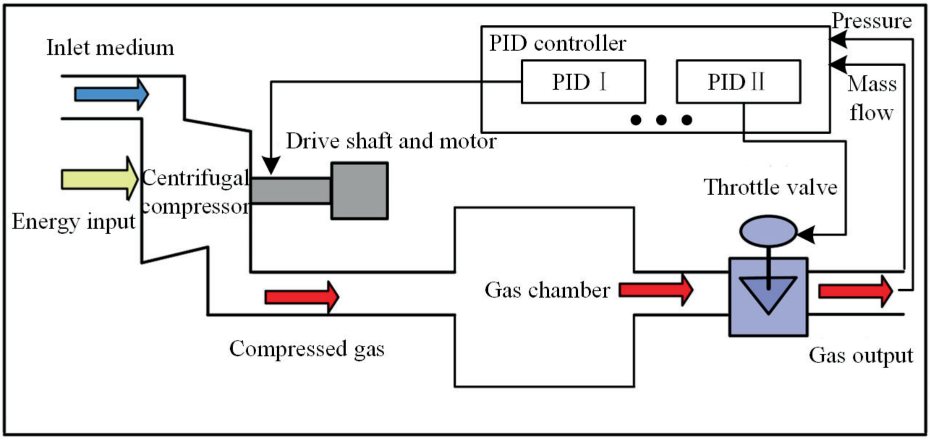

Figure 5.1: The structure of the compressor control system. Taking a gas compressor as an example, when gas enters the compressor, it is compressed by the high-speed rotation of the impeller. The compressed gas is then delivered through the gas chamber to the outlet. During this process, it is essential to implement a closed-loop control system to maintain the outlet pressure and mass flow rate at desired setpoints. A PID controller is employed to achieve this objective by measuring the outlet pressure and mass flow in real-time and dynamically adjusting key control variables, such as the rotational speed of the main shaft and the valve opening.

In the literature and practice, coordinated control has been commonly adopted, as it can simultaneously control multiple strongly coupled parameters, such as rotational speed and valve opening (Cortinovis et al., 2014). However, it can not ensure the safety of the control process, mainly due to the significant *time-delay characteristics* of the system state in reaction to the change in control parameters. The overshooting problem is a direct consequence of the time-delay characteristics, which is caused by (1) the nature of high-speed rotating machinery with considerable inertia – it takes a process to increase or decrease the rotation speed; (2) thermodynamics, such as heat exchange, which in turn delays the propagation of temperature and pressure variations along the flow path.

Overshooting problem

We show, in Figure 5.2, two examples of the transient control responses in the operation condition diagram (x-axis: mass-flow, y-axis: pressure). Each point in the chart is referred to as an operation point. In the left sub-plot, we show the control trajectory of two control processes applied one after another: from point A to B – only adjusting the valve opening, and from B to C – only controlling the rotation speed.

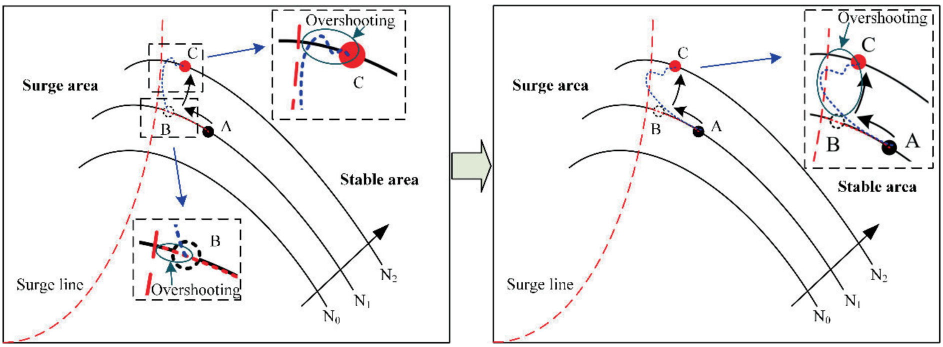

Figure 5.2: The overshoot generation process in coordinate control. Here, N_0, N_1, and N_2 represent three pressure–mass flow characteristic curves of the compressor, corresponding to different shaft speeds. The left diagram illustrates the decomposition process of operating point movement during coordinated speed-valve control, while the right diagram displays the actual trajectory of the operating point transition.

Overshoot occurs in both control processes, e.g., from B to C, the operation condition oscillates quite a bit before stabilizing to the target point C.

Also, shown in the right sub-plot, when we control valve opening and rotation speed simultaneously (called the coordinated control), the overshoot from controlling each individual parameter can accumulate and cause the overall control to deviate significantly from the original target. There exists a well-known surge area in the diagram, where excessive overshoot may induce a surge or instability, thereby posing a serious threat to the safety of the system. The key to addressing the overshoot problem in coordinated control lies in designing a control strategy that balances both the efficiency of operation point adjustment and the safety of system performance.

Issues in the current coordinated control

Traditional approaches typically employ decoupling control or model predictive control (MPC) to achieve coordination among multivariable systems (Cortinovis et al., 2014, Rodríguez et al., 2018). However, centrifugal compressors are *highly nonlinear and strongly coupled* systems, making complete decoupling difficult. Moreover, due to their computational complexity, MPC-based strategies often fail to meet real-time requirements during transient responses.

Due to its simple structure and fast response, the PID-based coordinated control strategy remains the most feasible solution for meeting the real-time requirements in centrifugal compressor control (Li et al., 2020). Two **key issues** need to be addressed in developing such a control strategy:

(1). How to establish a real-time quantitative evaluation of control safety.
(2). How to design a safety-oriented control strategy based on this measure.

To address issue (1), a fast-response model is required to predict the adjustment trajectory of the operating point and quantify surge risk. The Greitzer model, as a classical flow–dynamics model, effectively captures compressor transients and provides a solid basis for safety evaluation. For issue (2), the challenge is to optimize controller parameters in real time for the nonlinear compressor system, balancing safety and adjustment efficiency. As an advanced evolutionary algorithm, Covariance Matrix Adaptation Evolution Strategy (CMA-ES) is well-suited for strongly coupled multivariate optimization, making it a feasible solution for this task. To address the above issues, we propose:

A closed-loop control model of the centrifugal compressor was developed based on the Greitzer model to simulate the transient behavior during real-time operating condition adjustments, which allows for quantitative assessment of surge risk during the adjustment process.

A safety control strategy was established based on the CMA-ES algorithm, enabling online optimization of controller parameters to reduce overshoot and effectively prevent surge.

The optimization of PID controller parameters for coordinated control of a centrifugal compressor can be regarded as a multi-coupling parameter search problem. The CMA-ES algorithm is capable of capturing the coupling relations between variables with a covariance matrix. By learning the covariance matrix on the fly, it can determine the importance of different directions in the search space, thereby improving optimization efficiency for complex problems. This capability makes CMA-ES a suitable solution for addressing the time-delay issues in coordinated control of centrifugal compressors.

5.2 Previous Research and Main Issue

Single-parameter control

Single-parameter control remains the simplest approach for changing the operating conditions of centrifugal compressors. Ak and Cadirci (2022) demonstrated the suction flow control effectiveness of centrifugal compressors, while Kurz et al. (2021) investigated the effects of variable-speed control on the process performance and the transient and steady-state behavior of centrifugal compressors. However, single-parameter control fails to consider the interactions between variables, which may lead to deviations from the optimal operating point and limit the operational efficiency.

Coordinated control

Coordinated control methods address complex systems by decomposing them into interconnected subsystems, optimizing both operational efficiency and system safety through synergistic regulation (Peng et al., 2021, Ge et al., 2020). This approach aims to optimize multiple objectives by adjusting a set of strongly coupled system inputs. Coordinated control has been successfully applied to aerospace systems, power distribution, and synchronous motor control, achieving notable performance improvements (Shi et al., 2023, Wang et al., 2023, Xie et al., 2021). Unlike single-parameter methods, coordinated control leverages parameter coupling to adjust system variables jointly, thereby improving both optimization efficiency and accuracy. Currently, when addressing the coordinated

Decoupling control: Decoupling control aims to isolate individual control inputs to simplify the control process. However, achieving complete decoupling in centrifugal compressors remains challenging due to strong parameter interdependencies, making partial decoupling methods more prevalent. For instance, Mohammadi and Kermani (2011) treated parameter coupling effects as multistage disturbances, employing a pre-observer to predict target parameters for coordinated flow and pressure control. Similarly, Zhao et al. (2014) utilized a pre-extended observer to forecast control parameters and designed a nonlinear sliding mode controller to regulate rotational speed and valve opening. While these methods demonstrate potential, the inherent nonlinearity of centrifugal compressors prevents full decoupling. Moreover, such approaches often rely on complex nonlinear controllers with limited applicability across diverse industrial operating conditions.

MPC: MPC has recently gained significant attention for centrifugal compressor control. Alsuwian et al. (2023) implemented an MPC for ASC systems, demonstrating superior performance over conventional PID controllers in minimizing steady-state error and percentage overshoot. Daniarta et al. (2016) developed various MPC approaches, proving their effectiveness in enhancing surge avoidance capabilities. Torrisi et al. (2017) proposed an MPC that actively adjusts anti-surge valve openings to proactively prevent surge during operation. However, the transient nature of centrifugal compressor operation imposes stringent real-time requirements on controllers. Current model predictive controllers face three critical limitations: (1) computationally intensive algorithms degrade real-time performance, (2) heavy reliance on precise system models that are difficult to obtain and tune, and (3) sensitivity to timevarying parameters in practical applications. These challenges have significantly hindered industrial adoption despite ongoing academic research.

Evolutionary computation in control optimization

In centrifugal compressor control systems, the implementation of stable adjustment through relatively simple PID controllers remains preferable to both decoupled control architectures and model predictive control strategies. The optimization of control performance fundamentally depends on two critical factors: the design of coordinated control mechanisms and the precise tuning of key controller parameters. Evolutionary computation, as an established optimization methodology, has demonstrated extensive applicability and notable effectiveness in fluid machinery. It has been successfully applied to structural design, system efficiency optimization, and modeling (Fu et al., 2024, Wei et al., 2025, Sun et al., 2024). The effectiveness of evolutionary computation methods in the fluid machinery has been well established, demonstrating their advantage in handling optimization problems for systems with nonlinear characteristics. Developing a controller parameter optimization method based on evolutionary algorithms offers a feasible solution to the coordinated control challenges of centrifugal compressors.

In summary, coordinated control represents the most effective approach for achieving efficient and accurate adjustment of compressor operating points. However, the strong coupling and nonlinear characteristics of compressors, along with modeling complexity and computational inefficiency, make it difficult for traditional coordinated control methods - such as decoupling control and MPC - to achieve fast, accurate, and safe adjustment. Given the simple structure and fast response characteristics of PID controllers, developing a PID-based control strategy with adaptive parameter optimization using evolutionary algorithms offers a more practical and feasible solution.

5.3 Methodology

In centrifugal compressor operating points adjustment, the transient superposition of multiple control variables overshooting often results in system stability and safety. To address this issue, First, we developed a fast simulation model based on the Greitzer lumped-parameter model to capture the transient behavior of the centrifugal compressor, enabling quantitative evaluation of system safety during the adjustment process. Then, we propose a CMA-ES-based coordinated controller for fluid machinery that: (1) derives desired trajectories by incorporating controlled parameters into a trajectory adjustment module, and (2) employs genetic strategies to optimize controller parameters, achieving stable operating point adjustment. The methodology is structured as Figure 5.3:

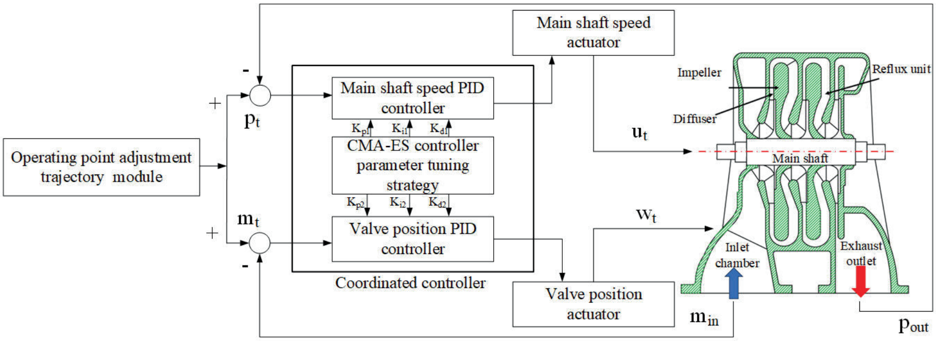

Figure 5.3: The structure of the proposed coordinated control framework on the centrifugal compressor.

5.3.1 Centrifugal Compressor Modeling

To efficiently capture the transient behavior of centrifugal compressors during adjustment, this study adopts the Greitzer lumped-parameter model. This modeling approach integrates the compressor as an embedded component within the inlet duct system rather than treating it as an isolated unit. The system configuration comprises:

1. An inlet duct embedding the compressor assembly,
2. A large-volume plenum chamber downstream of the inlet duct, and
3. Compressed gas media discharged through an outlet throttle valve to either the atmosphere or an external load.

To simulate the system's dynamic behavior, the one-dimensional momentum equation is applied to both the compressor and throttle duct, while mass conservation principles govern the plenum chamber. The dynamic compression process is modeled with a first-order time constant t. For describing transient gas compression behavior, dimensionless mass flow rate ϕ_c and pressure rise ψ are introduced:

$$\psi = \frac{\Delta P}{\frac{1}{2}\rho_\alpha U_t^2} \tag{5.1}$$

$$\phi_c = \frac{\dot{m}_c}{\rho_\alpha A_c U_t^2} \tag{5.2}$$

$$U_t = \frac{2\pi R_t \omega_t}{60} \tag{5.3}$$

where ρ_a denotes the gas density, U_t represents the blade tip speed, $\dot{m}_c$ indicates the inlet mass flow rate, R_t is the impeller outer radius, and ω_t signifies the impeller angular velocity. All dimensionless parameters in this model are calculated using this standardized method.

The first-order time constant $\tilde{t}$ is introduced as:

$$\tilde{t} = ta\sqrt{\frac{A_c}{V_p L_c}} \tag{5.4}$$

where a is the speed of sound, V_p denotes the plenum volume, A_c represents the inlet duct cross-sectional area, and L_c indicates the inlet duct length. Based on the Greitzer lumped-parameter modeling principle, the compressor system is divided into four subsystems:

1. Compressor momentum conservation subsystem (models impeller speed inertia effects on compressor behavior
2. Plenum mass conservation subsystem
3. Throttle valve dynamic characteristic subsystem
4. Approximated steady-state compressor characteristic subsystem.

The momentum conservation subsystem is expressed as:

$$\frac{d\phi}{d\tilde{t}} = B\left[\psi_c - \psi\right] \tag{5.5}$$

where B is the centrifugal compressor stability parameter, ψ_c represents the compressor dimensionless pressure rise, and ψ denotes the plenum dimensionless pressure rise.

The plenum mass conservation subsystem is formulated as:

$$\frac{d\psi}{d\tilde{t}} = \frac{1}{B}\left[\phi_c - \phi_t\right] \tag{5.6}$$

where ϕ_t is the throttle duct dimensionless mass flow rate. This subsystem characterizes the relationship between plenum pressure variations and internal mass flow rate dynamics.

Approximated steady-state compressor characteristic subsystem is represented as:

$$\frac{d\psi_c}{d\tilde{t}} = \frac{1}{\tilde{t}}\left[\psi_{c,ss} - \psi_c\right] \tag{5.7}$$

where $\psi_{c,ss}$ denotes the quasi-steady-state dimensionless pressure rise in the compressor characteristic curve. This subsystem models compressor performance across varying operating conditions. To determine the $\psi_{c,ss}$, the approximated steady-state compressor characteristic equation is utilized:

$$\psi_c(\phi_c) = \psi_c(0) + H\left[1 + \frac{3}{2}\left(\frac{\phi_c}{F} - 1\right) - \frac{1}{2}\left(\frac{\phi_c}{F} - 1\right)^3\right] \tag{5.8}$$

where parameter $\psi_c(0)$, H and F are determined by quasi-steady-state measurement of the compressor characteristic curve.

The throttle valve dynamic characteristic subsystem is described by:

$$\phi_t = c_t u_t \sqrt{\psi_t} \tag{5.9}$$

where u_t is the throttle valve dimensionless displacement, c_t denotes the dimensionless throttle parameter and ϕ_t represents the throttle duct dimensionless mass flow rate. This subsystem primarily simulates how throttle valve opening adjustments affect compressor flow and pressure characteristics.

5.3.2 Coordinated Controller Design

In compressor control systems, when the operating point changes, the difference between the target mass flow rate and the actual output mass flow rate at time instant k can be expressed as:

$$E_m(k) = \dot{m}_t(k) - \dot{m}_{act}(k) \tag{5.10}$$

where $E_m(k)$ is the mass flow error, $\dot{m}_t(k)$ represents the reference target value of mass flow at time k, $\dot{m}_{act}(k)$ defines the true value of mass flow at time k. If the mass flow m exponentially approaches the target value, its transient response during adjustment should follow:

$$\begin{cases} \dot{m} = \dot{m}_{org} + (\dot{m}_t - \dot{m}_{org})(1 - e^{-k_g(t-T)}) & t > T \\ \dot{m} = \dot{m}_{org} & t \le T \end{cases} \tag{5.11}$$

where k_g denotes the gain coefficient for mass flow adjustment, T indicates the control activation timing, and $\dot{m}_{org}$ represents the baseline mass flow rate at the nominal operating condition.

Therefore, the reference target mass flow rate at time step t should be set as:

$$\hat{\dot{m}} = \dot{m}_{org} + (\dot{m}_t - \dot{m}_{org})(1 - e^{-k_g(kT_s - T)}) \tag{5.12}$$

where T_s is the sampling period for mass flow adjustment.

Similarly, the outlet pressure must vary synchronously with the mass flow rate to ensure the operating point follows the predetermined exponential function trajectory. The outlet pressure variation during adjustment should be:

$$\begin{cases} p = p_{org} + (p_t - p_{org})(1 - e^{-k_g(t-T)}) & t > T \\ p = p_{org} & t \le T \end{cases} \tag{5.13}$$

where k_p is the gain coefficient for mass flow adjustment.

Therefore, the reference target pressure at time step k should be:

$$\hat{p}_t(k) = p_{org} + (p_t - p_{org})(1 - e^{-k_p(kT_p - T)}) \tag{5.14}$$

where T_p denotes the sampling period for the pressure signal in pressure adjustment, typically set as $T_s = T_p$.

In centrifugal compressor systems, different control variables exhibit distinct response rates. For instance, in combined valve opening-outlet pressure control, the system typically responds more sensitively to valve opening adjustments than to speed variations. This discrepancy causes actual control outcomes to deviate from design targets, resulting in higher-than-designed mass flow rates and lower-than-designed outlet pressures, thus necessitating reference target value corrections. To address this issue, we propose a control reference target selection strategy based on the error between actual and reference values. The flow reference control target selection strategy is defined as:

$$\begin{cases} \hat{\dot{m}}_t(k+1) = F^{-1}(p_{act}, \omega_{act}) & \left|\hat{\dot{m}}_t - m_{act}\right| > \delta m_{act} \\ \hat{\dot{m}}_t(k+1) = \dot{m}_{org} + (\dot{m}_t - \dot{m}_{org})(1 - e^{-k_m(k+1)T_s - T}) & \left|\hat{\dot{m}}_t - m_{act}\right| \le \delta m_{act} \end{cases} \tag{5.15}$$

where F^{-1} represents the inverse function of the compressor's pressure-flow characteristic curve, δ = 2% represents the error tolerance range during the control process. Similarly, the pressure reference control target selection strategy is given by:

$$\begin{cases} \hat{p}_t(k+1) = F(\dot{m}_{act}, \omega_t) & \left|\hat{p}(k) - p_{act}\right| > \delta p_{act} \\ \hat{p}_t(k+1) = p_{org} + (p_t - p_{org})(1 - e^{-k_p(k+1)T_s - T}) & \left|\hat{p}_t - p_{act}\right| \le \delta p_{act} \end{cases} \tag{5.16}$$

where F represents the function of the compressor's pressure-flow characteristic curve.

After obtaining the reference target pressure $\hat{p}_t(k)$ and reference target mass flow rate $\hat{m}_t$, compare them with the actual pressure value $p_{act}(k)$ and mass flow rate $\dot{m}_{act}$ respectively to obtain the following error values:

$$\hat{e}_p(k) = \hat{p}_t(k) - p_{act}(k), \hat{e}_m(k) = \hat{\dot{m}}_t(k) - m_{act} \tag{5.17}$$

where $\hat{e}_p(k)$ is the difference between the pressure value and its reference target at time step *k*, and $\hat{e}_m(k)$ represents the difference between the mass flow rate and its reference target at time step *k*.

These error values are then fed into the corresponding PID controller, whose transfer function is given by:

$$G_{pid}(s) = K_p + \frac{K_i}{s} + Kds \tag{5.18}$$

where K_p, K_i/s, and K_{ds} denote the proportional gain, integral gain, and derivative gain parameters of the PID controller, respectively.

5.3.3 CMA-ES-based Controller Parameter Optimization

In the pressure-mass flow coordinated controller, two critical PID controllers are responsible for adjusting valve opening and rotational speed. Since the coordinated control requires simultaneous adjustment of two physical parameters, and the optimal PID parameters may vary significantly across different operating points during centrifugal compressor operation, traditional PID tuning methods struggle to meet the dynamic and precise control requirements of the coordinated controller. Therefore, we propose a CMA-ES-based PID parameter optimization method. This section details the proposed approach on population generation, recombination, selection, and fitness function design. The population in the CMA-ES algorithm employed in this study is generated as follows:

$$x_k^{g+1} = m^g + \sigma^g N(0, C^g), \quad k = 1, \ldots, \lambda \tag{5.19}$$

where x_k^{g+1} denotes the $k-th$ individual in the $(g+1)-th$ generation, m^g represents the mean of the $g-th$ generation population, σ^g indicates the global step size at the $g-th$ generation, C^g stands for the covariance matrix of the $g-th$ generation, and λ specifies the population size (set to 200 in this study).

This study adopts the (μ, λ)-method for offspring generation, where only off-spring individuals participate in producing new candidates. The truncation selection criterion is based on the population's fitness function values, retaining only the top- μ individuals with optimal fitness to form the new elite subpopulation. This elite sub-population is then utilized to update the population parameters m^{g+1}, σ^{g+1}, and C^{g+1}.

The next generation's mean is obtained by computing the weighted average of the selected μ individuals from the elite subpopulation:

$$m^{g+1} = \sum_{i=1}^{\mu} w_i x_{i:\lambda}^{g+1} \tag{5.20}$$

where ω_i is weigh parameters. $x_{i:\lambda}^{g+1}$ denotes the $i-th$ optimal individual in the $(g+1)-th$ generation population, where $i:\lambda$ indicates the top-ranked $i-th$ individual satisfying condition $f(x_{1:\lambda}^{g+1}) \leq f(x_{2:\lambda}^{g+1}) \leq \ldots \leq f(x_{\lambda:\lambda}^{g+1})$, and f represents the fitness function.

This study employs both rank- μ update and rank-1 update for covariance matrix adaptation. The rank- μ update effectively integrates information across the current generation's population, while the rank-1 update captures correlation data between successive generations. In the rank- μ update, weight coefficients are introduced to incorporate sample information from multiple individuals of the previous generation, while the rank-1 update primarily captures the direction of consecutive optimal individuals to guide the search toward more promising regions. In centrifugal compressor coordinated control, the objectives are to ensure the outlet pressure and mass flow rate stabilize rapidly at target operating conditions with minimal overshoot M_p and fast settling time. The fitness function incorporates these metrics with weighting coefficients to balance their contributions, achieving both reduced overshoot and shorter transient response. The designed fitness function is:

$$f = \omega_0(\lg(Mp_p^{norm}+1)+\lg(Mp_m^{norm}+1))+\omega_1(\lg(Ts_p^{norm}+1)+\lg(Ts_m^{norm}+1)) \tag{5.21}$$

where ω_0, ω_1 are weighting coefficients, Mp_p^{norm}, Ts_p^{norm}, Mp_m^{norm}, and Ts_m^{norm} represent the normalized pressure overshoot, pressure settling time, flow overshoot, and flow settling time, respectively. The weighting factors are determined using the following strategy: Let $d(o,S)$ represent the distance from the target operating point to the surge protection curve, and let D_{bs} denote the distance from the surge protection curve to the surge line. Based on experimental experience, the weighting parameters should adhere to the following principles:

When $d(o,S) \le 0.5D_{bs}$, the safety requirement is considered more important than the speed of adjustment. The weighting factors are then set as:

$$\omega_0 = 5\omega_1 \tag{5.22}$$

When $d\left(o,S\right) > 0.5D_{bs}$, the safety requirement can be appropriately relaxed. The weighting factors are then set as:

$$\omega_0 = \omega_1 \tag{5.23}$$

5.4 Experiments

To validate the proposed controller's effectiveness, this study conducts simulation experiments comparing the method with conventional Ziegler-Nichols (Z-N) tuning. The evaluation assesses both approaches under three industrially prevalent adjustment modes:

1. Constant outlet pressure control

2. Constant mass flow rate control

3. Coordinated pressure-flow control

The superior performance in overshoot reduction and operating point trajectory optimization of the proposed method is discussed by the experimental results. The robustness of the proposed method is also validated by introducing an excitation at 30% along the trajectory.

The simulation model developed in this study is based on the G150-1.7 centrifugal compressor system as shown in Figure 5.4, with both the compressor and control system models implemented in MATLAB/Simulink. Due to the 0.1 resolution limit of the PID controller used in this study, all optimized parameters were retained with a precision of one decimal place during the experiments. The design parameters of G150-1.7 centrifugal compressor are shown in Table 5.1.

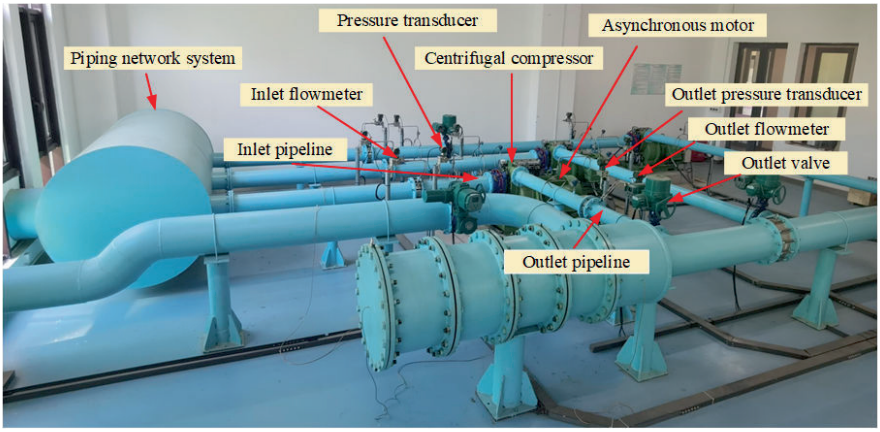

Figure 5.4: 150-1.7 centrifugal compressor system.

Table 5.1: Design parameters of G150-1.7 centrifugal compressor.

Parameters name	Value	Parameters name	Value
Inlet duct diameter	0.25 m	Hub diameter	88.0 mm
Inlet duct length	2.1 m	Impeller inlet diameter	192.0 mm
Impeller diameter	0.38 m	Blade inlet diameter	202 mm
Plenum volume	1.2 m^3	Impeller outlet diameter	350 mm
Outlet duct diameter	0.18 m	Blade inlet width	48 mm
Outlet duct length	3.5 m	Blade inlet angle	360 mm
Motor speed range	22000-25000 rpm	Blade outlet angle	500 mm

5.4.1 Constant Outlet Pressure Adjustment Experiment

The outlet pressure, mass flow, and efficiency of initial operating points for the constant outlet pressure adjustment experiment is $\left[P_{oi}, \dot{m}_{oi}, \eta_{oi}\right]$ = $\left[150.8kPa, 1.090kgs^{-1}, 71.21\%\right]$, while target operating points $[P_{ot}, \dot{m}_{ot}, \eta_{ot}]$ = $[150.8kPa, 0.792kgs^{-1}, 75.53\%]$.

The comparison of PID control parameters optimized using the Z-N method and the CMA-ES method is presented in Table 5.2. In both the constant mass flow adjustment and outlet pressure-mass flow coordinated adjustment experiments, the parameters tuned by the Z-N method remained identical to those used in the constant outlet pressure adjustment experiments.

Table 5.2: The comparison of PID controller parameters optimized by Z-N method and CMA-ES for constant outlet pressure adjustment experiment.

Method	PID controller	K_p K_i K_d
Z-N method	Mass flow controller/kPa	0.5, 0.1, 0.1
	Outlet pressure controller/kg s^{-1}	1, 0.01, 0.5
CMA-ES method	Mass flow controller/kPa	0.1, 0, 0.5
	Outlet pressure controller/kg s^{-1}	0.5, 0, 0.1

The comparison of response curves using the Z-N method and CMA-ES is shown in Figure 5.5. The operating point of the centrifugal compressor was shifted from the initial to the target condition at $t = 5s$. As evident from the results, the conventional control method causes both the valve opening and rotational speed to increase rapidly during the adjustment process. This leads to significant overshoot, with the maximum outlet pressure overshoot reaching approximately 2.6%. In comparison, the proposed controller demonstrates effective overshoot suppression. While exhibiting a marginally longer settling time for mass flow adjustment, the overall control duration remains comparable.

Figure 5.6 presents the comparison of operating point trajectories between the Z-N method and the proposed method on constant outlet pressure adjustment. Sub-plot (a) shows that the coordinated control method produces a smoother transition process than the conventional approach. This enables the system state to converge steadily to the target operating point without overshoot, which results in the operating point trajectory moving further away from the surge region. This can be seen more clearly in sub-plot (b): after applying an excitation in the mass flow direction, the adjusted trajectory obtained using the proposed method remains farther from the surge region compared to the one adjusted using the Z-N method, indicating that the proposed method exhibits stronger robustness.

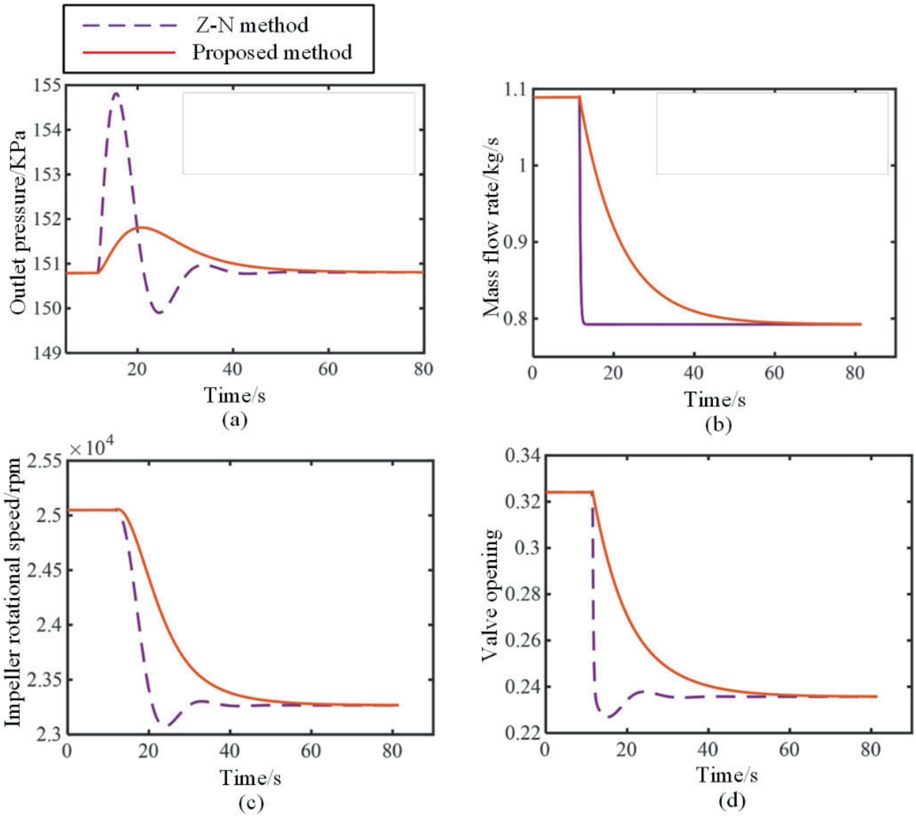

Figure 5.5: The comparison of response curves using Z-N method and CMA-ES on constant outlet pressure adjustment. (a), (b), (c), and (d) correspond to the outlet pressure, mass flow rate, impeller rotational speed, and valve opening response curves, respectively.

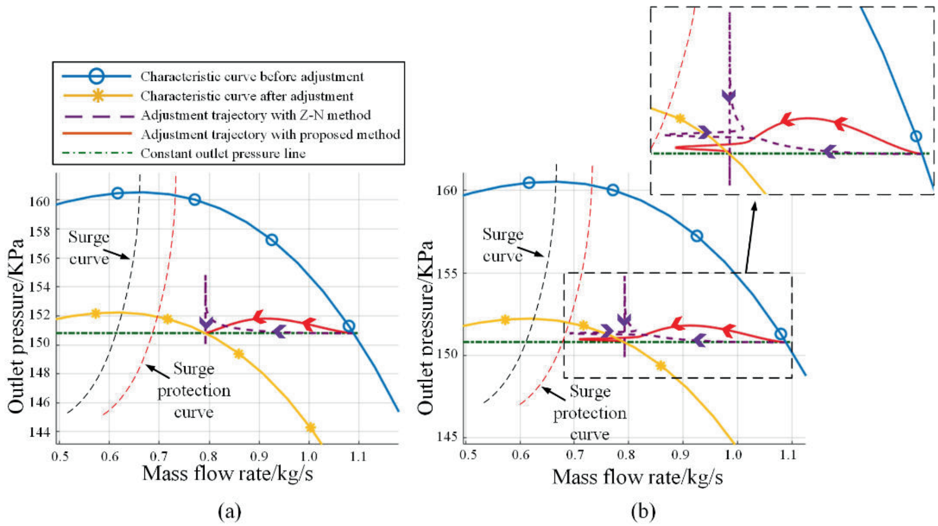

Figure 5.6: The comparison of operating point trajectories between the Z-N method and the proposed method on constant outlet pressure adjustment. (a) shows the trajectory of the operating point under normal conditions, while (b) shows the trajectory with excitation introduced on the mass flow direction during the movement of the operating point.

5.4.2 Constant Mass Flow Rate Adjustment Experiment

The initial operating points for the constant mass flow rate adjustment experiment is $[P_{oi}, \dot{m}_{oi}, \eta_{oi}] = [140kPa, 1.0kgs^{-1}, 69.72\%]$, while target operating points $[P_{ot}, \dot{m}_{ot}, \eta_{ot}] = [150kPa, 1.0kgs^{-1}, 73.14\%]$. The optimized PID control parameters are summarized in Table 5.3.

Table 5.3: Optimized PID controller parameters for constant mass flow rate adjustment experiment.

PID controller	K_p K_i K_d
Mass flow controller/kPa	0.1, 0, 0.5
Outlet pressure controller/kg s^{-1}	0.5, 0, 0.1

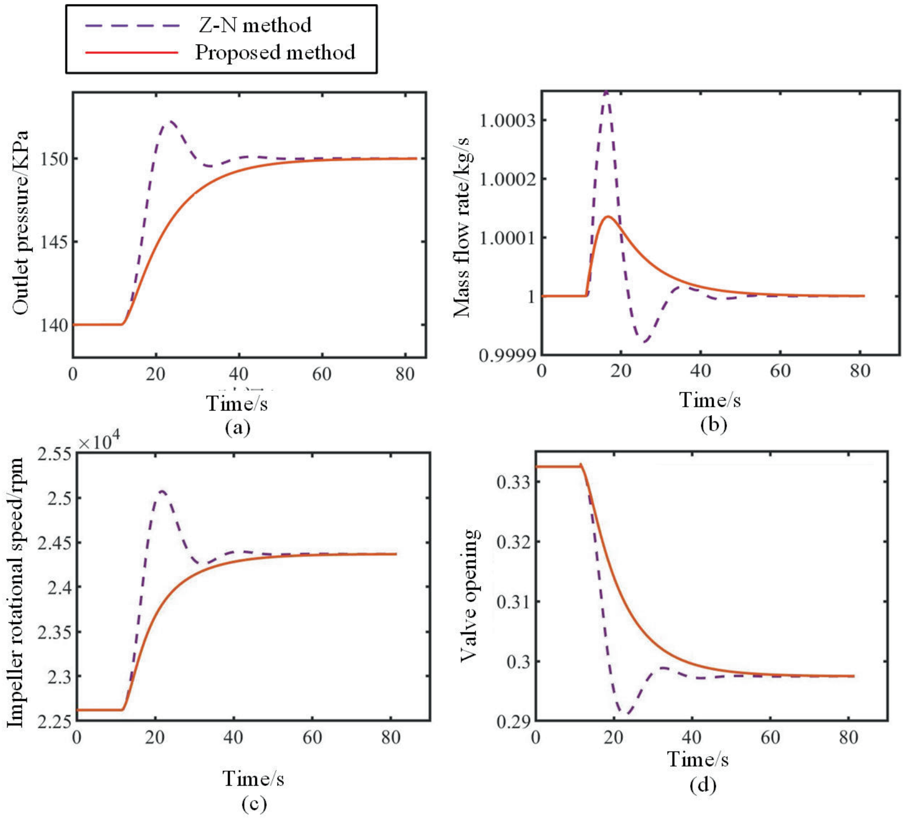

Figure 5.7: The comparison of response curves using the Z-N method and CMA-ES on constant mass flow rate adjustment. (a), (b), (c), and (d) correspond to the outlet pressure, mass flow rate, impeller rotational speed, and valve opening response curves, respectively.

Figure 5.7 compares the response characteristics of key parameters during constant mass flow adjustment for both control strategies. The Z-N tuned PID controller exhibits substantial overshoot in both outlet pressure (peak overshoot: 2.24%) and mass flow adjustment. In comparison, the proposed control strategy completely eliminates pressure overshoot while reducing mass flow fluctuations by 67.5%.

Figure 5.8 further demonstrates the improved adjustment trajectory achieved by the proposed method. Whether under normal conditions or in the presence of excitation, it is characterized by significantly reduced deviation between the operating point trajectory and the constant mass flow rate line, which is the optimal adjustment trajectory, thereby enhancing operational safety. The comparative results clearly demonstrate the superior dynamic performance in terms of safety and robustness of the proposed coordinated control approach.

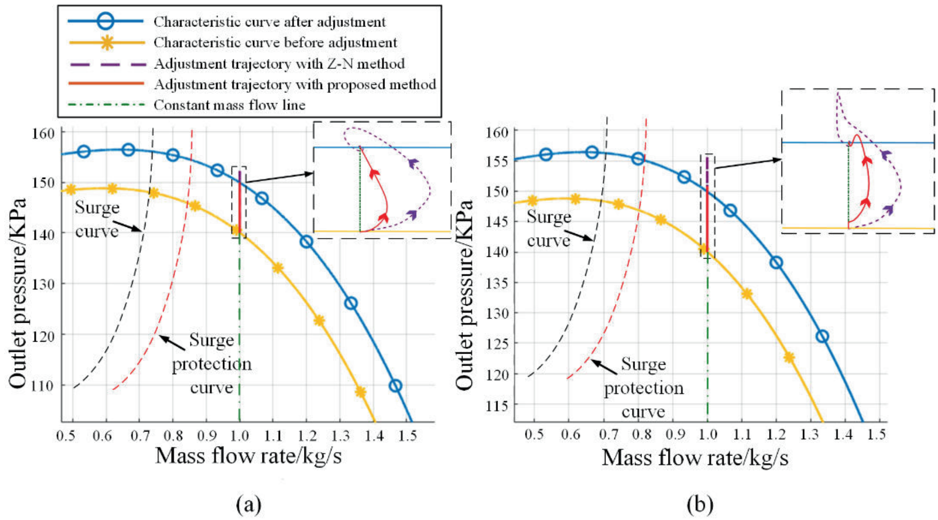

Figure 5.8: The comparison of operating point trajectories between the Z-N method and the proposed method on constant mass flow rate adjustment. (a) shows the trajectory of the operating point under normal conditions, while (b) shows the trajectory with excitation introduced on the outlet pressure direction during the movement of the operating point.

5.4.3 Coordinated Mass Flow and Outlet Pressure Adjustment Experiment

The initial operating points for the coordinated mass flow and outlet pressure adjustment experiment is $[P_{oi}, \dot{m}_{oi}, \eta_{oi}] = [112.70kPa, 1.35kgs^{-1}, 36.76\%]$, while target operating points $[P_{ot}, \dot{m}_{ot}, \eta_{ot}] = [155kPa, 1.0kgs^{-1}, 73.34\%]$. The optimized PID control parameters are summarized in Table 5.4.

The optimized PID control parameters are shown in Table 5.4. Figure 5.9 compares the response curves of relevant parameters during the coordinated mass flow and outlet pressure adjustment experiment between the two methods. When changing the centrifugal compressor' s operating point at $t=10s$, the conventional control method achieves simultaneous adjustment of both mass flow and outlet pressure, but exhibits significant pressure overshoot. In contrast, the proposed method substantially reduces overshoot without compromising the overall adjustment speed.

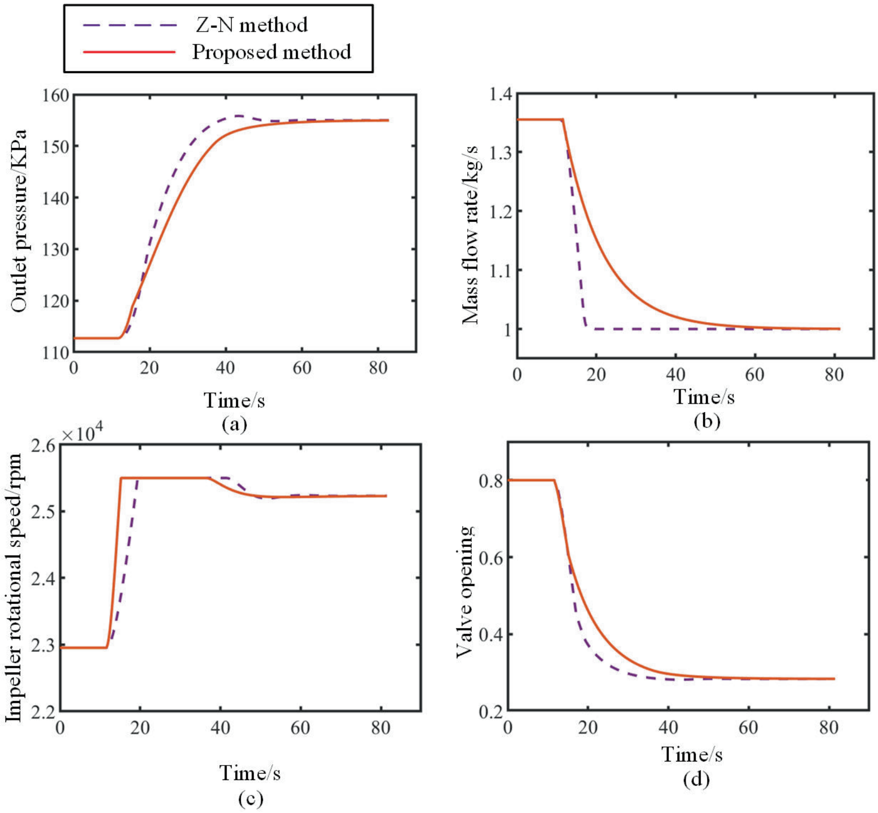

Figure 5.9: The comparison of response curves using the Z-N method and CMA-ES on coordinated mass flow and outlet pressure adjustment. (a), (b), (c), and (d) correspond to the outlet pressure, mass flow rate, impeller rotational speed, and valve opening response curves, respectively.

As shown in Figure 5.10 by the mass flow-pressure adjustment trajectory, the proposed method confines the operating point variation within a smaller range during adjustment. thereby reducing the risk of entering unstable regions. Visibly, whether under normal adjustment or with excitation applied, the operating point trajectory obtained using the proposed method remains consistently farther from the surge region and closer to the optimal adjustment curve. This clearly demonstrates that the proposed method offers superior advantages in both safety and robustness.

Table 5.4: Optimized PID controller parameters for coordinated mass flow and outlet pressure adjustment experiment.

PID controller	K_p K_i K_d
Mass flow controller	0.2, 0, 0.5
Outlet pressure controller	0.5, 0, 0.1

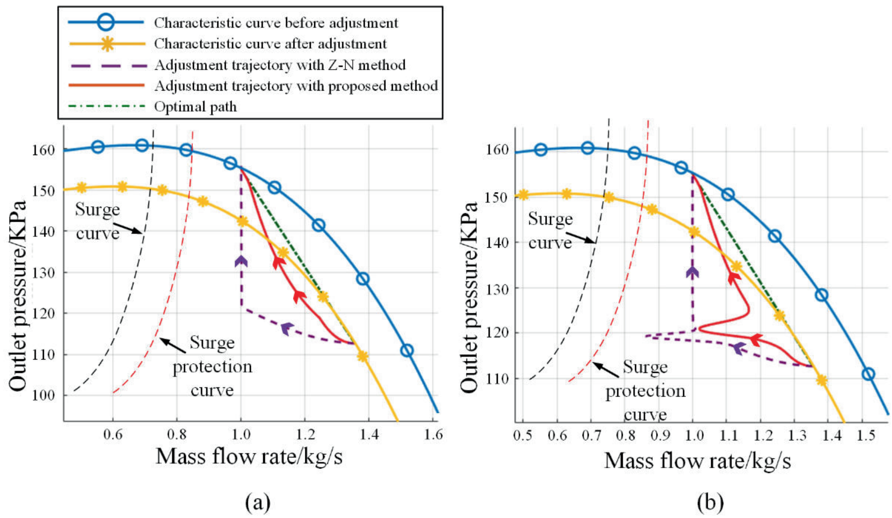

Figure 5.10: The comparison of operating point trajectories between the Z-N method and the proposed on coordinated mass flow and outlet pressure adjustment. (a) shows the trajectory of the operating point under normal conditions, while (b) shows the trajectory with excitation introduced on the outlet pressure direction during the movement of the operating point.

The three experimental cases demonstrate that the proposed coordinated controller enables the operating point to reach the target condition within a confined trajectory, confirming its effectiveness in overshoot suppression and trajectory optimization. Compared with conventional methods, the proposed approach maintains comparable adjustment efficiency while significantly enhancing operational safety.

5.5 Summary

We propose a novel coordinated control strategy for centrifugal compressors based on the CMA-ES. The strategy is designed to overcome the limitations of conventional control methods in dealing with nonlinearity, time delays, and multivariable coupling. By reducing overshoot, the proposed method enhances the safety of the operating point adjustment process. A lumped-parameter model is established, and the CMA-ES algorithm is employed to efficiently optimize multiple controller parameters without requiring system decoupling.

Experimental case studies demonstrate that the proposed controller significantly reduces flow and pressure overshoot during the operating point adjustment, thereby improving the reliability and robustness of the control system under various operating conditions.

Compared to existing methods, the proposed strategy introduces two key innovations: (1) a lumped-parameter model for rapid and quantitative evaluation of centrifugal compressor safety in operating point adjustment; and (2) a coordinated control strategy based on CMA-ES is proposed to improve control robustness and dynamic performance.

These advantages highlight the great potential of the proposed approach for practical engineering applications in system-working media coupling situation involving centrifugal compressors. Future work will focus on integrating real-time adaptive mechanisms and extending the method to a broader range of turbomachinery.

The proposed CMA-ES-based controller has several limitations. First, the optimization is performed on a transient scale (on the order of seconds) and is designed as an offline method carried out before the operating point regulation. Second, although effective, the Greitzer lumped-parameter model is unable to capture complex phenomena such as rotating stall or impeller flow dynamics. Third, the performance of the controller has only been validated through simulations; experimental verification on a physical compressor test bench is required prior to industrial deployment.