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Surge explicit nonlinear model predictive control using extended Greitzer model for a CCV supported compressor

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ABSTRACT

This paper indicates the use of explicit nonlinear model predictive control method to prevent and control of compressor surge using extended Greitzer model. To encompass the effects of fast transients, nonlinear extended Greitzer model is considered for modelling and control of the compressor system. This model covers the effects of different parts, especially the compressor system pipeline, as the most important factor in the occurrence and acceleration of the surge. Also, the mathematical description of the flow dynamic of compression system is derived in the presence of close-coupled valve (CCV) as the most commonly used actuator. For controlling the compressor system surge instability, explicit nonlinear model predictive control (NMPC) is applied, whose allows efficient online implementation, significantly enlarges the operating region of the compressor and enhances the authority of the control system. Numerical simulations show that the proposed control system is able to meet the desired specifications in avoiding and active control of surge, in the presence of different types of disturbances occurring along the pipeline.

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1. Introduction

Compressors are essential machines in modern industrial processes for the pressing and transportation of gases and fluids. They are vital to the operation of key energy sectors, such as the oil and gas, nuclear, and hydroelectric. In addition, compressors are central components in the heating, ventilation, and air conditioning systems for homes and commercial buildings. With the current focus on improving the efficiency of energy usage in buildings and manufacturing systems, the compressor is an important link in the energy generation/consumption chain in our modern society that needs to be carefully studied. Surge is one of the main dynamic instability limiting the operation of centrifugal compressors [1] and it is an asymmetrical oscillation of the flow through the compressor and is characterized by a limit cycle in the compressor characteristic. This event forces the flow back toward the compressor inlet and initiates the surge limit cycle that affects the entire system. Surge can cause extensive structural damage in the machine because of the violent vibration and high thermal loads that generally accompany the instability [2].

Furthermore, any flow unsteadiness or periodic excitation in a centrifugal compressor station piping system

can significantly decrease the compressor's surge margin. Both acoustic resonance and system impedance are functions of the entire piping system connected to the compressor, including pipe friction, interface connections, valve/ elbow locations, pipe diameter, valve coefficients and etc. can move the centrifugal compressor operating point into a surge or stonewall condition.

It is critically important to consider the impact of the station's piping system on the compressor dynamic behaviour. According to the findings of Brun and Spark [3,4], the entire piping system connected to the compressor will have the following effects on the compressor system:

- (1) operating point variation
- (2) amplification and speed up of surge occurrence
- (3) fluctuations transfer in the system
- (4) limit the stability range of compressor

Thus, a careful evaluation of acoustic and impedance effects over compression system should be performed to avoid impacting the operating range of the machine and to properly design the surge control system.

Having an appropriate model from a compressor system is the first step in studying the mentioned effects

over surge instability and its controlling. There are several mathematical models developed over the years to describe the dynamics of the flow in compression systems, and extensive reviews of these models can be found in references [5,6]. Among these models, the most frequently referenced is the lumped parameter model introduced by Greitzer [7,8] for axial compressors, and later demonstrated to be applicable to centrifugal compressors by Hansen et al. [9].

Furthermore, as pointed out in reference [10], the Greitzer model alone is not adequate to predict the dynamics associated with fluid flow in distributed systems, such as acoustic waves and flow pulsations in pipelines. Because the Greitzer model only covers the slow poles of the compressor system, it is unable to capture fast transient modes and analysing the behaviour of the compression system. Consequently, the model does not capture the fast dynamics of the system and therefore it is not suitable for fast control.

The dynamics of compression systems for different configurations of the inlet and the exhaust piping were studied in Refs [4,11,12]. Van Helvoirt and de Jager [10] proposed to implement a transmission line model, first introduced by Krus et al. [13] for modelling hydraulic and pneumatic line systems, to describe the effects of the pipeline dynamics between the compressor and the plenum volume in the pressure oscillations during the deep surge.

The mathematical model for the compression system that captures the effect of the piping acoustics during both the stable and unstable operating conditions, along with the dynamics in transition between these two states is expanded in [14]. Reference [14] also has the problem of using active magnetic bearings (AMBs) to control the compression system, while most compressor systems are already equipped with a CCV actuator.

Motivated by the potential benefits of controlling the surge, various measures have been introduced during the last few decades [15]. A surge avoidance system is a widely used passive method, which possesses good reliability. In the system, a surge avoidance line is defined, which is located on the right of the surge line on the compressor map.

Compressors are not allowed to operate in the region between the surge line and the surge control line. One reason of interest to study the surge phenomenon is that high efficiency operating points are usually around the surge line.

The reliability of surge avoidance systems is obtained through the sacrifice of efficiency and operating range [5,16]. Especially, the distance between the surge line and the surge control line is always conservative due to various and severe working conditions.

A totally different solution to the surge problem is called active surge control, which was first presented by Epstein et al in 1989 [17]. Active surge control systems

do not attempt to avoid surge by limiting the operating range of the compressor but by feedback control stabilize surge itself. Several types of actuators could be used in active control systems. An air injector [18], drive torque [19], closed-coupled control valve [20], movable plenum wall (including piston) [21], and bleed valve [22] are reported frequently in the literature. Both the linear and nonlinear methods can be used to design active surge controllers [23–26].

In nearly all cases, a compression system dynamic model is needed to develop an active surge controller. Although some achievements have been acquired by employing advanced algorithms like LQR, the Lyapunov method, sliding mode variable structure control, and model predictive control [27,28], however, none of these controllers have considered the effects of pipe that is a negative point in model accuracy for the stability of active controller. Based on a recently presented enhanced compression system model with variable impeller tip clearance and pipeline acoustics, a surge controller is designed [29], but there are two objections in this case including the linear controller is not capable in capturing strongly nonlinear dynamics of compression system and also common actuators are the close-coupled valve [30] and the throttle valve [31]. Considering optimization of control signal, as well as restrictions on actuators and states are another important topics that should be considered in the subject of controller design.

Overcoming the problems, we present new numerical results in the compressor surge explicit NMPC using the CCV as an actuator in the compression system.

A surge controller is designed for a centrifugal compressor system based on the enhanced compression system model recently presented in [14], that supports fast and slow transients, also captures the effects of pipe. In addition, it allows high-speed control of the compressor system using the CCV actuator. In addition, explicit NMPC is used to control the compressor system in the presence of disturbances that provide real-time control.

Both the avoiding and active control of surge considered in the simulation and the ability of the proposed controller is shown in the stabilization of compressors operation. Therefore, this article is classified as follows. In the second part, the compressor system model taking into consideration the effects of pipeline and CCV actuator will be detailed. Reservations about an explicit nonlinear predictive control and the simulation results for various cases will be discussed in third part.

2. System model

The dynamic model of the compression system considered here was presented in details in [14] and [32]. The compression system in the study of surge consists of three main components.

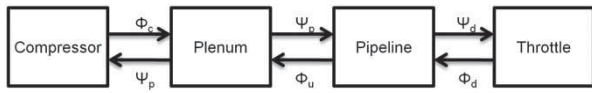


Figure 1. Block diagram of the compressor system model with pipeline dynamics.

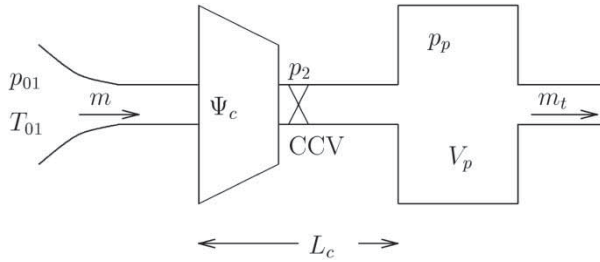


Figure 2. Compressor with close-coupled valve [5].

These are the compressor/plenum adding and storing the energy in the system, the throttle valve controlling the average flow rate, and the piping transporting the compressed gas/fluid. The nondimensional pressure rise ψ and mass flow rate Φ are defined as the following functions of the mass flow rate m and the absolute pressure P :

$$\Phi = \frac{m}{\rho_{o1} U A_c} \quad (1)$$

$$\psi = \frac{P - P_{o1}}{\frac{1}{2} \rho_{o1} U^2}$$

The constants in (1) are the impeller velocity U , the cross-sectional area of the compressor duct A_c , the inlet absolute pressure P_{o1} , and the density at the inlet ρ_{o1} .

The Greitzer model can capture the surge limit cycle for a compression system, and it serves as a good basis for the development of surge control laws. As described in [14], the acoustic resonance from the compressor pipeline is added to the model. The resulting block diagram for the compression system with pipeline dynamics is shown in Figure 1.

2.1. Compressor and plenum

The basis of the equations describing the flow in the compressor and the plenum volume comes from the Greitzer model in [7,8]. Assumptions for this model are low compressor inlet match number, low-pressure rise compared with ambient pressure, isentropic compression process in the plenum with uniform pressure distribution, and negligible fluid velocities in the plenum. The use of a close-coupled valve for surge control was accurately modelled and presented in [5,33]. The approach introduces a valve close to the plenum volume in the compressor. CCV means that there is no mass storage of gas between the compressor outlet and the valve, as can be seen from Figure 2.

Based on the mentioned statement, the pressure increase in the compressor and the pressure drop across

the valve can be combined into an equivalent compressor. This is needed in order to enable the valve for controlling the characteristic of the equivalent compressor.

The outlet mass flow from the compressor through a close-coupled valve is given as follows:

$$\Phi_r = c_r u_r \sqrt{(\Psi_p - \Psi_c)} \quad (2)$$

where c_r is valve constant, u_r is the CCV opening percentage and Ψ_p , Ψ_c are plenum pressure rise and compressor pressure rise, respectively. The range value of u_r is from 0 to 100%.

Finally, according to [5,33], equations for describing the pressure and mass flow in the constant speed centrifugal compressor can now be formulated as follows:

$$\dot{\Phi}_c = B \omega_H (\Psi_{c,ss} - \Psi_p) \quad (3)$$

$$\dot{\Psi}_p = \frac{\omega_H}{B} (\Phi_c - \Phi_p - \Phi_r)$$

where B is the Greitzer stability parameter and ω_H is the Helmholtz frequency [7].

The state variables of the model are the compressor mass flow rate Φ_c and the plenum pressure rise Ψ_p , which are nondimensionalized as shown in (1). The value of the plenum mass flow rate Ψ_p is dependent on the dynamics of the piping. Finally, the steady-state compressor pressure rise $\Psi_{c,ss}$ is obtained from the compressor characteristic curve as a function of the compressor mass flow rate

$$\Psi_{c,ss}(\Phi_c) = \begin{cases} A_1 \Phi_c^3 + B_1 \Phi_c^2 + D_1, & \text{If } \Phi_c > \Phi_s \\ A_2 \Phi_c^3 + B_2 \Phi_c^2 + D_2, & \text{If } \Phi_c \leq \Phi_s \end{cases} \quad (4)$$

The characteristic curve $\Psi_{c,ss}$ is divided into the stable and unstable part by the surge point at Φ_s . The coefficients A_1 , B_1 , and D_1 of the characteristic curve correspond to the stable flow region of the compressor, whereas the coefficients A_2 , B_2 , and D_2 meet unstable flow region, as [29].

2.2. Piping

To describe the dynamics of the piping system, a modal approximation of the transmission line dynamics [34] has been included in the compression system equations [14]. The single-mode state space representation of the piping equations can be expressed in the following form:

$$\begin{bmatrix} \dot{P}_{th} \\ \dot{Q}_p \end{bmatrix} = \begin{bmatrix} 0 & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \begin{bmatrix} P_{th} \\ Q_p \end{bmatrix} + \begin{bmatrix} 0 & B_{12} \\ B_{21} & B_{22} \end{bmatrix} \begin{bmatrix} P_p \\ Q_{th} \end{bmatrix} \quad (5)$$

for some matrix coefficients $A_{ij} \in \mathbb{R}$ and $B_{ij} \in \mathbb{R}$, where P_p and Q_p are the upstream (plenum) pressure and volumetric flow rate, respectively. In the same way, P_{th} and

Q_{th} are the downstream (throttle duct) pressure and flow rate.

The above piping model represents the system in terms of the absolute pressure and volumetric flow rate. With the assumption that the change of density ρ of gas in the pipeline because of the pressure and temperature fluctuation caused by the piping acoustics is small, the states of the piping model can be nondimensionalized as described in (1). After the coordinate transformation, the resulting piping equation with nondimensional states is expressed as follows:

$$\begin{aligned} \begin{bmatrix} \dot{\Psi}_{th} \\ \dot{\Phi}_p \end{bmatrix} &= \begin{bmatrix} 0 & \frac{2A_{12}A_c}{\rho U} \\ \frac{A_{21}\rho U}{2A_c} & A_{22} \end{bmatrix} \begin{bmatrix} \Psi_{th} \\ \Phi_p \end{bmatrix} \\ &+ \begin{bmatrix} 0 & \frac{2B_{12}A_c}{\rho U} \\ \frac{B_{21}\rho U}{2A_c} & 0 \end{bmatrix} \begin{bmatrix} \Psi_p \\ \Phi_{th} \end{bmatrix} \\ &+ \begin{bmatrix} 0 \\ \frac{\rho P_{o1}}{\rho_{o1}UA_c}(A_{21} + B_{21}) \end{bmatrix} \end{aligned} \quad (6)$$

2.3. Throttle valve

The flow rate through the throttle valve is a function of the pressure drop across the valve. Here, it is assumed the dynamics at the throttle duct section are much faster than the rest of the system, and only the steady-state behaviour is captured. The relationship between the pressure and the mass flow rate in the throttle valve section is given by

$$\Phi_{th} = c_{th} u_{th} \sqrt{\Psi_p} \quad (7)$$

where c_{th} is the valve constant and u_{th} is the throttle valve opening percentage.

2.4. Equations set

The equations for the dynamics of the complete compression system are obtained by combining (3), (6), and (7). The resulting nonlinear system has the compressor mass flow, the plenum pressure rise, the throttle section pressure rise, and the plenum mass flow rate as state

Table 1. Model parameters for the compression system.

Parameter	Symbol	Unit	Value
Comp. duct length	L_c	m	1.86
Comp. duct cross area	A_c	m ²	0.0082
Char. curve A_1 coeff.	A_1	–	–4.717
Char. curve A_2 coeff.	A_2	–	–172.6
Char. curve B_1 coeff.	B_1	–	–2.859
Char. curve B_2 coeff.	B_2	–	36.88
Char. curve D_1 coeff.	D_1	–	1.193
Char. curve D_2 coeff.	D_2	–	1.029
Gritzer Stab. parameter	B	–	0.44
Helmholtz frequency	ω_H	Rad/s	80.1
Impler Tip speed	U	m/s	213.24
Inlet pressure	P_{o1}	Pa	101,325
Inlet gas density	ρ_{o1}	Kg/m ³	1.165
Plenum volume	V_p	m ³	0.049
Pipeline length	L	m	6.5
Throttle constant	C_{th}	–	1.8569
CCV constant	C_r	–	0.5

variables. The state space equations of the assembled system are as follows:

$$\begin{aligned} \begin{bmatrix} \dot{\Phi}_c \\ \dot{\Psi}_p \\ \dot{\Psi}_{th} \\ \dot{\Phi}_p \end{bmatrix} &= \begin{bmatrix} 0 & -B\omega_H & 0 & 0 \\ \frac{\omega_H}{B} & 0 & 0 & -\frac{\omega_H}{B} \\ 0 & 0 & 0 & \frac{B}{2A_{12}A_c} \\ 0 & \frac{B_{21}\rho U}{2A_c} & \frac{A_{21}\rho U}{2A_c} & \frac{\rho U}{A_{22}} \end{bmatrix} \\ &\times \begin{bmatrix} \Phi_c \\ \Psi_p \\ \Psi_{th} \\ \Phi_p \end{bmatrix} + \begin{bmatrix} B\omega_H(\Psi_c, ss) \\ 0 \\ 0 \\ \frac{\rho P_{o1}}{P_{o1}UA_c}(A_{21} + B_{21}) \end{bmatrix} \\ &+ \begin{bmatrix} 0 & 0 \\ -\frac{\omega_H}{B} & 0 \\ 0 & \frac{2B_{12}A_c}{\rho U} \\ 0 & B_{22} \end{bmatrix} \begin{bmatrix} \Phi_r \\ \Phi_{th} \end{bmatrix} \end{aligned} \quad (8)$$

The parameters of the theoretical model are summarized in Table 1. Also, the coefficients of the characteristic curve correspond to the stable and unstable operating regions of the compressor are given in this table. The corresponding matrix coefficients of the piping equation in (5) are found to be $A_{12} = 3.7 \times 10^6$, $A_{21} = -1.92 \times 10^{-3}$, $A_{22} = -8$, $B_{12} = -3.7 \times 10^6$, $B_{21} = 1.92 \times 10^{-3}$, and $B_{22} = 7.98$.

3. Explicit NMPC

In the compressor system control problem, we encounter with nonlinear system subject to physical and operational constraints on the input and state. Well-known systematic nonlinear control methods such as feedback linearization [35–37] and constructive Lyapunov-based methods [38,39] lead to very elegant solutions, but they depend on complicated design procedures that do not scale well to large systems and they are not developed in order to handle constraints in a systematic manner. Although the control signal of the

MPC controller is restricted, the MPC controller does not take the saturation in the compression system into account and cannot optimally predict therefore [28].

It is required that nonlinearities and constraints are explicitly considered in the controller for satisfying environmental and safety considerations, rewarding physical and operational constraints, and operating compressor system on the tighter performance specifications. Nonlinear predictive control, the extension of well-established linear predictive control to the nonlinear world, appears to be a well-suited approach for this kind of problem.

However, the solution of an on-line nonlinear optimization problem is often computationally complex and time-consuming and the real-time NMPC implementation is usually limited to slow processes where the sampling time is sufficient to support the computational needs. The on-line computational complexity can be circumvented with an explicit approach to NMPC, where an explicit approximate representation of the solution is computed using multi-parametric Nonlinear Programming (mp-NLP). The benefits of an explicit solution, in addition to the efficient on-line computations, include also verifiability of the implementation (which is an essential issue in safety-critical applications) and the possibility for designing embedded control systems with low software and hardware complexity. For nonlinear MPC, the prospects of explicit solutions are even higher than linear MPC, since the benefits of computational efficiency and verifiability are even more important. This may lead to a significant reduction in the requirements to real-time embedded computer hardware and formal software verification being a feasible practical tool. Therefore, we use the explicit NMPC [40] for surge control in the compression system.

3.1. Explicit NMPC formulation

Consider the 4nd-order state space equations of compressor system in (8), as mentioned above, with Φ_c as compressor mass flow rate, the plenum pressure rise Ψ_p , the throttle section pressure rise Ψ_{th} , the plenum mass flow rate Φ_p as state variables and Φ_r is mass flow through a close-coupled valve in series with the compressor. The coefficients of the characteristic curve are given in Table 1.

The control objective is to avoid surge, i.e. stabilize the system. This may be formulated as

$$J(u[0, T], x[0, T]) \triangleq \int_0^T l(x(t), u(t), t) dt + S(x(T), T) + Rv^2 \quad (9)$$

where

$$l(x, u) = \alpha(x - x^*)^T(x - x^*) + \kappa u^2$$

$$S(x) = \beta(x - x^*)^T(x - x^*)$$

with $\alpha, \beta, \geq 0$ and the set point x^* corresponds to an unstable equilibrium point, subject to the inequality constraints for $t \in [0, T]$

$$u_{min} \leq u(t) \leq u_{max}$$

$$-x_2 + 0.6 \leq v$$

$$-v \leq 0$$

and the ordinary differential equation (ODE) given by

$$\frac{d}{dt}x(t) = f(x(t), u(t))$$

that initial condition $x(0) \in X \subset R^n$. Valve capacity requires the constraint $0 \leq u(t) \leq 0.4$ to hold, and the pressure constraint $x_2 \geq 0.6 - v$ avoids operation too far left of the operating point. The variable $v \geq 0$ is introduced in order to avoid infeasibility and $R = 10$ is a large weight. The input signal $u[0, T]$ is assumed to be piecewise constant and parameterized by a vector $U \in R^p$ such that $u(t) = \mu(t, U) \in R^r$ is piecewise constant.

3.2. Simulation results

The operation of closed-loop system under suggested controller is simulated with sampling time $T_s = 1 \text{ ms}$. In this simulation, $\alpha = 1$, $\beta = 0$, and $\kappa = 0.08$ are considered and Euler integration with T_s (size of every step) is applied to solve the ordinary differential equation. It should be remarked that the constraints on u and v are linear, such that any quadratic programming solution is feasible for the nonlinear programming.

Two operating scenarios are used to demonstrate the ability of the controller. In the first scenario, simulation of compression system equipped with CCV by using of parameters in Table 1 is given as follows. A compressor initially operates in a steady state where the throttle valve openings equal to 20%. At time $t = 8\text{s}$, the throttle is closed to 10%, such that the compressor is interred into surge if the controller had not activated, as shown in the Figure 3.

In this case, the compressor will experience the deep surge limit cycle. When the CCV and controller are activated, the CCV is opening at the time of closing throttle, therefore plenum fluid is recycled to the inlet that decreases the plenum pressure and accelerate the compressor mass flow. Finally, the operating point acquires the stable area as shown in Figure 4.

The control signal is also shown in Figure 5. In the second scenario, initially compressor is interred to the deep surge instability, and this situation continues until $t = 8\text{s}$. In this time, the controller is allowed to operate and the CCV starts to move, as shown in Figure 6. Therefore, plenum fluid is returned to the inlet that

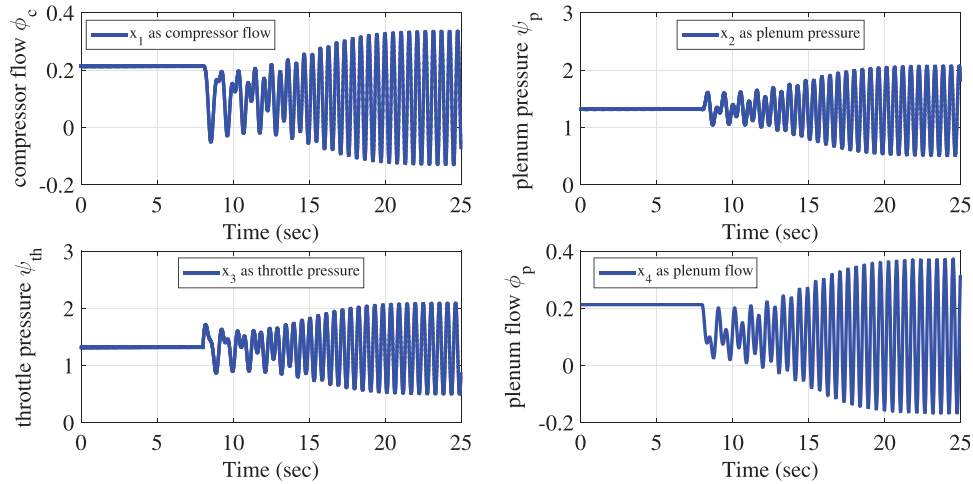


Figure 3. States of system when surge controller inactivated in scenario 1.

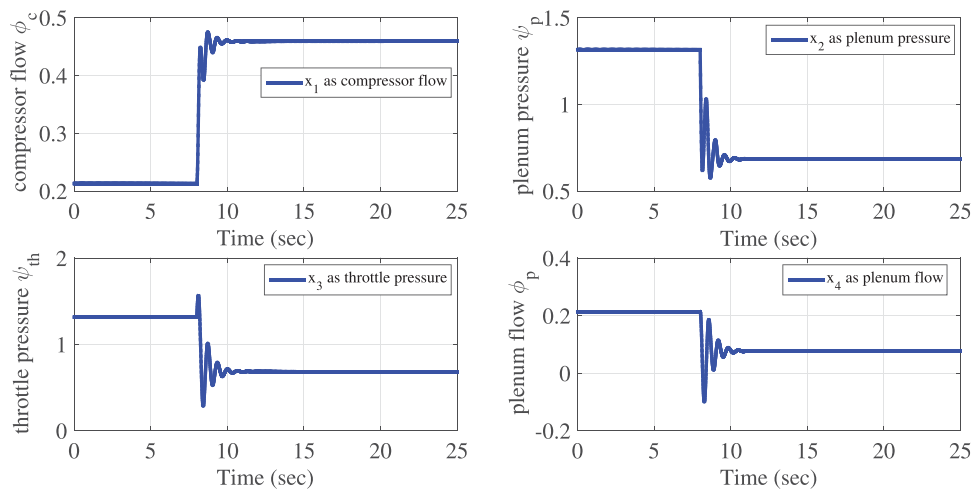


Figure 4. States of system when surge controller activated in scenario 1.

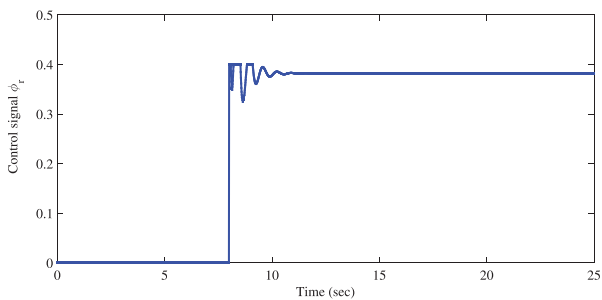


Figure 5. Control signal of surge controller in scenario 1.

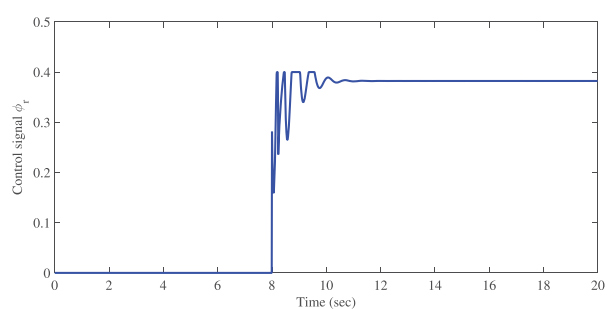


Figure 6. Control signal of surge controller in scenario 2.

causes reduction of plenum pressure and growth of the compressor mass flow. Finally, compressor stable operating point is achieved instead of initial unstable states. The transient response of the compressor system is also shown in Figure 7.

The first scenario shows controller capability to comprehend the effects of pipe, throttle and the whole of the compressor system, and the latter is after the surge occurrence to demonstrate the ability of controller.

Using of an explicit nonlinear predictive control and taking into account nonlinear dynamics of pipe, we will be able to stabilize the compressor system under different conditions to prevent from the surge occurrence and also enlarging the range of compressor by the CCV as actuator.

For enhancing efficiency in the safe operation of the compressor, the different working conditions can be covered by using the controller.

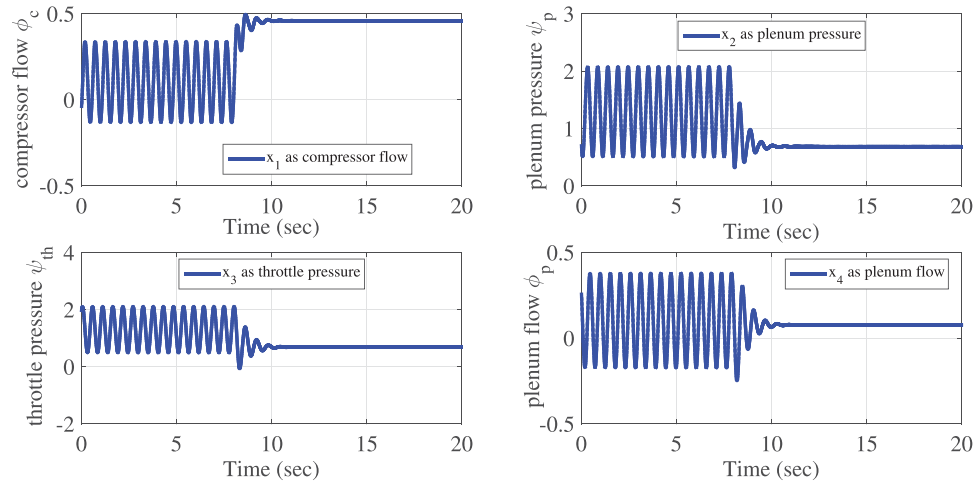


Figure 7. States of system in scenario 2.

4. Conclusion

The design of a CCV-based surge controller for centrifugal compressors was discussed. Many solutions for surge control have been proposed in the literature, but their effectiveness has been rare because of not considering the effects of the pipe. In this paper, based on the model of the compression system with piping acoustics, we derived the model that supports close-coupled valve as the most common actuator, and presented the application of explicit nonlinear model predictive control to perform stabilization of surge limit cycle. Proposed nonlinear model predictive control covers all of limitations over the states and CCV actuator. The controller was tested with the extended proposed model in different types of simulation scenarios at different conditions and its capability in rejection the flow disturbances related to surge instability and stabilizing the compressor system was shown. Robust model predictive control against system uncertainties and other disturbances can be investigated in the future studies.

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