Introducing Back-up to Active Compressor Surge Control System ⋆

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Abstract: A novel method for introducing a back-up system to an active compressor surge control system is presented in this paper. Active surge control is a promising method for extending the compressor map towards and into the unstable area at low mass flow by stabilizing the surge phenomenon. The method also has potential for allowing operation at higher efficiencies. However, a failure in the active surge control system may endanger the compressor by entering deep surge as the compressor is allowed to operate in the stabilized surge area. We propose the use of a back-up system applied to the active system to keep the compressor safe should the active system fail. This paper present an active compressor surge control system with piston actuation combined with a blow off system as the back-up. Performance of the combined system is evaluated by simulating the system in situations where the piston is saturated or jammed. The combination results in a system with increased performance by taking advantage of both systems.

Keywords: centrifugal compressor, compressor surge, piston-actuated active surge control system, blow off surge control system, combined surge control system.

1. INTRODUCTION

The operating area of compressors can be described by plotting compressor pressure rise against flow for varying compressor speed. This is called the compressor map. The stable operating area is limited for low mass flows by the so-called surge line and for high mass flows by the stone wall or choke line. Operation of a compressor at flows below the surge line would drive the compression system into an instability known as surge. This is an axisymmetric oscillation of mass flow and pressure rise and is followed by severe vibrations in the compression system. The vibrations may reduce the reliability of the system and large amplitude vibrations may lead to compressor damage, especially to compressor blades and bearings.

Most industrial compressors are equipped with a surge avoidance system ensuring that the compressor does not enter the surge area. These surge avoidance systems usually work by recycling flow from downstream to upstream when the operating point reach a surge control line that is located to the right of the surge line. Such surge avoidance schemes successfully ensure safe operation, but the introduction of the surge control line reduces the usable size of compressor map, thereby restricting the compressor operational envelope.

Surge stabilization by using active control system was proposed by Epstein et al. (1989) and since then a number of theoretical and experimental results have been published. A number of different actuators and control methods have been applied, as summarized by Willems and de Jager (1999). Recent developments in this field include the work by Arnulfi et al. (2001) on hydraulic actuators as well as Bøhagen and Gravdahl (2008) on drive torque actuation. Williams and Huang (1989) proposed to employ a movable plenum wall as an actuator for the surge control problem. Their experimental results showed that the developed system, using a loudspeaker as the movable wall, was able to stabilize surge and enlarge the operating area in the low mass flow region of the compressor map. This inspires the piston-actuation surge control in the current work.

Active surge control systems have mainly been implemented in university laboratories and have not yet found wide spread use in industrial compression systems. One reason for this is safety. Although active surge control is a promising method for compressor map enlargement, the enlarged area is open loop unstable, and failure in the active system will cause the compressor to go unstable and enter surge. The introduction of a back-up system is therefore necessary.

Active surge control using a blow off valve was presented by Willems and de Jager (1998). In principle, this is quite similar to a surge avoidance system using recycle, but the blow off valve is controlled by a control law that actively stabilizes the equilibrium or operating point instead of using a control line. This approach stabilized surge and the compressor map was enlarged to the left side of the surge line. The control system was so-called one-sided as it was only able to discharge flow from plenum and not inject flow into plenum. An active surge control system with piston actuation was introduced by Uddin and Gravdahl (2011a) where the surge stabilization relies on a piston which can both draw flow from the plenum or inject flow into the

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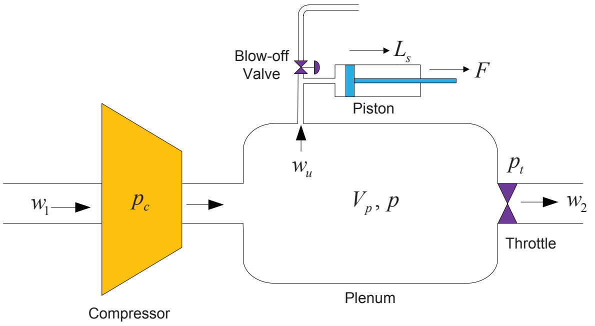

Fig. 1. Compression system equipped with a piston.

plenum. However, there are two situations that could cause this active system to fail: 1) actuator saturation caused by limited the maximum piston stroke, and 2) actuator fault such as jamming of the piston.

This paper presents an active compressor surge control system with piston actuation combined with a blow off system as back-up. Blow off system is a common method of controlling surge and piston-actuation was introduced, but the two methods are combined for the first time in this paper. Performance of the combined system is evaluated by simulating the saturation and jamming of the piston. The combination is expected to give a better system performance by taking the advantage of each method. A study case is presenting an application of the combined surge control method in a centrifugal compression system. Instability at low mass flow in a centrifugal compressor is dominated more by occurring surge than stall. Fontaine et al. (1999) presented a comparison of linear and non-linear control for axial compressor and concluded that linear control works well for compressor surge problem but not for both surge and stall. Based on their result, we are applying linear control design for the both surge control methods.

The paper consists of five sections including introduction in Section I. Section II describes compressor dynamics and compressor characteristic. Section III describes control design for piston-actuated active surge control law, blow off surge control law and active surge control including back-up. Simulation results are presented in Section IV. Finally, conclusions and future works are presented in Section V.

2. COMPRESSOR DYNAMICS

A model of a compression system equipped with a piston actuator combined with a blow off valve for surge control is shown in Fig. 1. The compression system dynamics was introduced by Greitzer (1976) and given as follows:

$$\dot{w}_1 = \frac{A_c}{L_c}\left[p_c\left(w_1\right) - p\right] \tag{1}$$

$$\dot{p} = \frac{a_0^2}{V_p}\left[w_1 - w_2\left(p\right) - w_u\right] \tag{2}$$

where w_1 is the compressor mass flow, w_2 is the throttle mass flow, w_u is the control mass flow, p_c is the compressor pressure rise, p is the plenum pressure, a_0 is the speed of sound, V_p is the plenum volume, A_c is the inlet duct cross section area, L_c is the length of the inlet duct. The control mass flow w_u is applied for surge control and will be controlled by the piston or the blow off valve.

Non-dimensionalization of the equations were done by using factors: $\frac{1}{2}\rho U^2$ for pressure, $\rho U A_c$ for mass flow, $\frac{1}{\omega_H}$ for time and L_c for length and resulting in:

$$\dot{\phi}_1 = B\left[\psi_c\left(\phi_1\right) - \psi\right] \tag{3}$$

$$\dot{\psi} = \frac{1}{B}\left[\phi_1 - \phi_2(\psi) - \phi_u\right] \tag{4}$$

where $B = \frac{U}{2\omega_H L_c}$ and $\omega_H = a_0\sqrt{\frac{A_c}{V_p L_c}}$. The notation ϕ_1 is the non-dimensional compressor mass flow, ϕ_2 is the non-dimensional throttle mass flow, ϕ_u is the non-dimensional control mass flow, ψ_c is the non-dimensional compressor pressure rise, ψ is the non-dimensional plenum pressure, B is the Greitzer's constant, U is the mean rotor velocity, ω_H is the Helmholtz resonator frequency, ρ is the fluid density and τ is the non-dimensional time. The non-dimensional throttle mass flow was defined by Gravdahl and Egeland (1997):

$$\phi_2 = \gamma_T\sqrt{\psi}. \tag{5}$$

A compressor pressure rise characteristic is modeled by a qubic function as introduced by Moore and Greitzer (1986):

$$\psi_c\left(\phi_1\right) = \psi_0 + H\left[1 + \frac{3}{2}\left(\frac{\phi_1}{W} - 1\right) - \frac{1}{2}\left(\frac{\phi_1}{W} - 1\right)^3\right] \tag{6}$$

where ψ_o is the shut-off value of the axisymmetric characteristic, W is the semi-width of the cubic axisymetric compressor characteristic, and H is the semi-height of the cubic axisymetric compressor characteristic, consult Moore and Greitzer (1986) for more detailed definition.

A compressor operating point is an intersection point between compressor pressure rise ψ_c and throttle pressure drop ψ_T. The throttle pressure drop is given by:

$$\psi_T\left(\phi_2\right) = \frac{1}{\gamma_T^2}\phi_2^2, \tag{7}$$

where γ_T is the throttle setting. Four different compressor operating points based on the compressor data given in Table 1 are shown in Fig. 2. Point A is a stable operating point with the throttle setting $\gamma_T = 0.7$. The compressor is operating at surge point when the throttle setting is $\gamma_T = 0.6$ as shown by point B. A throttle setting less than 0.6 brings the compressor into surge, for example: point C with $\gamma_T = 0.5$ and D with $\gamma_T = 0.3$. It can be shown that operating points located at positive compressor characteristic slope are unstable and thereby leading to surge, see Gravdahl and Egeland (1999).

3. SURGE CONTROL DESIGN

3.1 System State Equation

For notational convenience, define the system states as follows:

$$x_1 = \phi_1,\ x_2 = \psi, \tag{8}$$

Table 1. SIMULATION PARAMETERS

Parameter	Value	Unit	Parameter	Value	Unit
U	68	m/s	a_0	340	m/s
V_p	0.1	m^3	A_c	0.0038	m^2
L_c	0.41	m	ρ	1.2041	$\mathrm{kg/m}^3$
m_s	1	kg	A_s	0.0038	m^2
ψ_o	0.352	-	W	0.25	-
H	0.18	-			

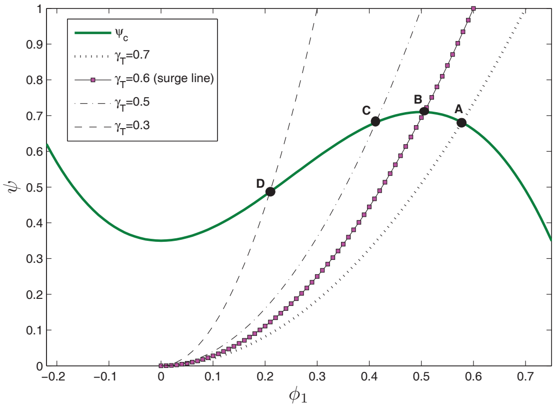

Fig. 2. Compressor and throttle characteristics.

and constants as follows:

$$b_1 = B,\ b_2 = \frac{1}{B} \tag{9}$$

such that the dynamics (3) and (4) can be written as:

$$\dot{x}_1 = b_1 \left[\psi_c\left(x_1\right) - x_2\right] \tag{10}$$

$$\dot{x}_2 = b_2 \left[x_1 - \phi_2\left(x_2\right) - \phi_u\right]. \tag{11}$$

Define deviation from an operating point as follows:

$$\tilde{x}_1 = x_1 - x_{10},\ \tilde{x}_2 = x_2 - x_{20},\ \tilde{\phi}_u = \phi_u - \phi_{u_0} \tag{12}$$

where (x_{10}, x_{20}) is the compressor operating point. The ϕ_{u_0} is assummed to be zero as the piston is idle during steady operation. The blow off valve is working only if the piston should fail. Transforming the system dynamics (10) and (11) into new state coordinates (12) results in:

$$\dot{\tilde{x}}_1 = b_1 \left[\tilde{\psi}_c\left(\tilde{x}_1\right) - \tilde{x}_2\right] \tag{13}$$

$$\dot{\tilde{x}}_2 = b_2 \left[\tilde{x}_1 - \tilde{\phi}_2\left(\tilde{x}_2\right) - \tilde{\phi}_u\right] \tag{14}$$

where the compressor pressure rise, throttle pressure drop and control mass flow in the new coordinates are defined by:

$$\tilde{\psi}_c\left(\tilde{x}_1\right) = -k_3\tilde{x}_1^3 - k_2\tilde{x}_1^2 - k_1\tilde{x}_1 \tag{15}$$

$$\tilde{\phi}_2\left(\tilde{x}_2\right) = \gamma_T\sqrt{\left(x_{20} + \tilde{x}_2\right)} - \gamma_T\sqrt{x_{20}} \tag{16}$$

with $k_1 = \frac{3Hx_{10}}{2W^3}\left(x_{10} - 2W\right)$, $k_2 = \frac{3H}{2W^3}\left(x_{10} - W\right)$ and $k_3 = \frac{H}{2W^3}$.

The compression system described in (13) and (14) is stabilized by control mass flow $\tilde{\phi}_u$ which can be generated by the piston or blow off valve depending on the chosen surge control method shown in Fig. 3. The details of the control laws are explained in the next parts.

3.2 Piston-actuated active surge control

Active surge control system using a piston was improved by using integral control in Uddin and Gravdahl (2011b). Piston-actuated active surge control is stabilizing surge by injecting flow into the plenum or drawing flow from the plenum by moving the piston. Define control mass flow generated by the piston as:

$$\tilde{\phi}_u = \tilde{\phi}_p, \tag{17}$$

which is a function of the piston velocity as

$$\tilde{\phi}_p = b_3\tilde{x}_3, \tag{18}$$

where b_3 is a constant given by

$$b_3 = \frac{1}{2B}\left(\frac{A_s}{A_c}\right) \tag{19}$$

and $\tilde{x}_3$ is nondimensional piston velocity defined by

$$\tilde{x}_3 = \frac{d}{d\tau}\left(\frac{L_s}{L_c}\right). \tag{20}$$

The notation A_s is for the piston cross section area, A_c is the inlet duct cross section area, L_s is the piston stroke and L_c is the length of the inlet duct. The closed loop piston dynamics including integral action is given as follows:

$$\dot{\tilde{x}}_3 = b_4\left[b_5\tilde{x}_2 + \tilde{u}_p\right] \tag{21}$$

$$\dot{\tilde{x}}_4 = \tilde{x}_3 \tag{22}$$

$$\dot{\tilde{x}}_5 = \tilde{x}_4. \tag{23}$$

The notation b_4 is a constant defined by $b_4 = \frac{2B^2}{M_s}$ with M_s is the non-dimensional piston mass given by $M_s = \frac{m_s}{\rho A_c L_c}$ where m_s is the piston mass. The notations $\tilde{u}_p$ is piston control force and b_5 is a constant defined by $b_5 = \frac{A_s}{A_c}$. The states $\tilde{x}_3$ is the piston velocity, $\tilde{x}_4$ is the piston stroke, and $\tilde{x}_5$ is time integral of the piston stroke. A state feedback control law for the piston control force is defined by:

$$\tilde{u}_p = -K_p\tilde{x}_p \tag{24}$$

where K_p is the control gain and $\tilde{x}_p = [\tilde{x}_1, \tilde{x}_2, \tilde{x}_3, \tilde{x}_4, \tilde{x}_5]^T$. Uddin and Gravdahl (2011b) applied linear quadratic regulator for the linearized the system around operating point A. The computational resulted in a control gain

$$K_p = [\,4303.2\ \ -2157.2\ \ 1707.9\ \ 1719.9\ \ 316.2\,] \tag{25}$$

and closed loop eigenvalues at $s_{1,2} = -1.2400 \pm 2.1987i$, $s_3 = -2.3043$ and $s_{4,5} = -0.1838 \pm 0.1824i$. All eigenvalues are located in the left half plane (LHP) such that the system locally asymptotically stable. An alternative control law using feedback from plenum pressure and piston displacement only can be found in Uddin and Gravdahl (2011a).

3.3 Blow off surge control

Blow off surge control is stabilizing surge by discharging the plenum fluid out of the compression system. The control mass flow is defined by:

$$\tilde{\phi}_u = \tilde{\phi}_b. \tag{26}$$

The blow off flow is adjusted by a blow off valve and the function is defined by:

$$\tilde{\phi}_b = \gamma_b(\tilde{u}'_b)\sqrt{x_{20} + \tilde{x}_2} \tag{27}$$

where $\gamma_b(\tilde{u}'_b)$ is the opening valve as a function of control signal u'_b with nominal value in the range of $0 \leq \gamma_b(\tilde{u}'_b) \leq 1$. For simplicity, define $\gamma_b(\tilde{u}'_b) := \tilde{u}_b$ such that

$$\tilde{\phi}_b = \tilde{u}_b\sqrt{x_{20} + \tilde{x}_2}. \tag{28}$$

Notice that this actuator model does not include actuator dynamics as in (21)-(23). The compressor dynamics with a blow off line are then defined by:

$$\dot{\tilde{x}}_1 = b_1\left[\tilde{\psi}_c(\tilde{x}_1) - \tilde{x}_2\right] \tag{29}$$

$$\dot{\tilde{x}}_2 = b_2\left[\tilde{x}_1 - \tilde{\phi}_2(\tilde{x}_2) - \tilde{\phi}_b\right]. \tag{30}$$

Linearization around an operating point results in

$$\dot{\tilde{x}}_b = A_b\tilde{x}_b + B_b\tilde{u}_b \tag{31}$$

where

$$\tilde{x}_b = [\,\tilde{x}_1 \ \tilde{x}_2\,]^T, \tag{32}$$

$$A_b = \begin{bmatrix} -b_1k_1 & -b_1 \\ b_2 & \dfrac{-b_2\gamma_T}{2\sqrt{x_{20}}} \end{bmatrix}, B_b = \begin{bmatrix} 0 \\ -b_2\sqrt{x_{20}} \end{bmatrix}. \tag{33}$$

A state feedback control for the blow off valve is given by:

$$\tilde{u}_b = -K_b\tilde{x}_b. \tag{34}$$

Linearizing the system (29)-(30) around operating point A and applying pole placement method by selecting closed loop poles at -1.09 ± 1.32 results in

$$K_b = [1.1666 - 0.9978]. \tag{35}$$

The poles was selected such that the blow off valve will not be saturated. Another possibility would be to use saturated control like in Willems et al. (2002). The pole selection was also intended to makes the closed loop system using blow off valve has greater damping and shorter natural frequency than the one using piston. It was due to the control mass flow in closed loop system with blow off valve has one direction and not bidirection as in the closed loop system with piston. It was demonstrated in Willems et al. (2002) that a compression system described by (29)-(30) is stabilizable with positive feedback blow off. The lower constraint on $\tilde{u}_b$ does not affect the stability of the linearized system but reduces the range of stabilizing control gains.

3.4 Active surge control with back-up

This active surge control system with back-up is combining the piston actuation and blow off valve to generate a control mass flow $\tilde{\phi}_u$ for surge control purpose. The piston surge control is the main system and operates by default. The blow off surge control is the back-up system and works if the main system should fail. The main system is said to have failed if the piston is saturated or the piston is jammed. The failure in the system is detected by observing the piston velocity $\tilde{x}_3$ and the piston control law output

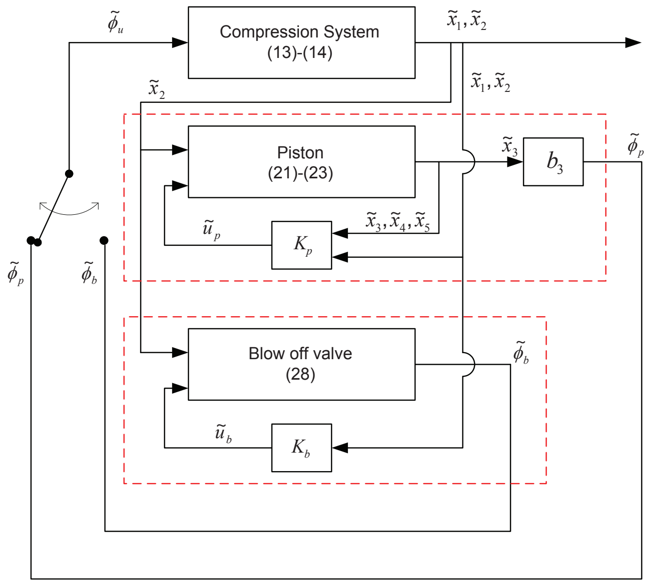

Fig. 3. Control diagram with switching operation of active surge control with piston actuation and blow off valve.

$\tilde{u}_p$. The control law for this combined system is defined as follows:

$$\tilde{\phi}_u = \begin{cases} \tilde{\phi}_p & \\ \tilde{\phi}_b & \text{for } \tilde{u}_p \neq 0 \wedge \tilde{x}_3 = 0 \end{cases} \tag{36}$$

where $\wedge$ is AND logic operator. The back-up system is only applied to save the compressor from entering surge when the main active surge control system should fail. Block diagram of the active surge control system including back-up is shown in Fig. 3.

4. SIMULATION

The performance of the active compressor surge control with back-up is evaluated by simulating piston saturation and piston jammed conditions. Four active surge control systems are simulated in both conditions for comparison. The first active surge control system called "Active I " is a piston-actuated active surge control system with piston stroke up to ± 0.35. The second active surge control system called "Active II" is a piston-actuated active surge control system with maximum stroke ± 0.15. The third active surge control system called "Active III" is a blow off active surge control system. The fourth active surge control system called "Active IV" is a piston-actuated active surge control system with maximum stroke ± 0.15 combined with blow off active surge control system as the back-up. The simulation scenario is initialized by operating the compressor at point A with throttle setting $\gamma_T = 0.7$ then at $\tau = 20$ the throttle is reduced to $\gamma_T = 0.5$ and then to $\gamma_T = 0.3$ at $\tau = 100$. The valve closing rate is 0.04 per non-dimensional time unit.

First simulation was done by simulating the system in normal condition and the result is shown in Fig. 4. The Active I controller performed well in stabilizing surge. The Active II controller was not able to stabilize surge as the control law required a piston stroke beyond the saturation limit. Moreover, closing throttle to $\gamma_T = 0.3$ causes the

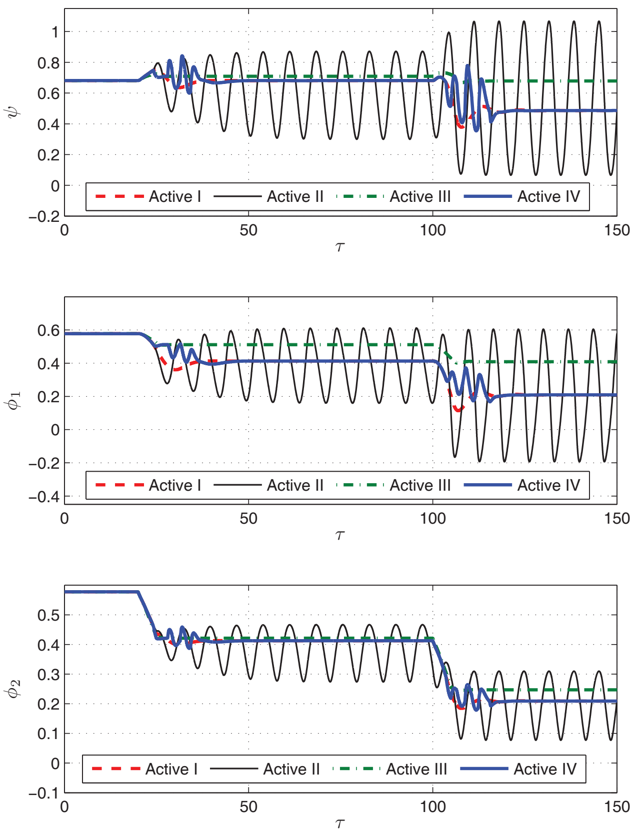

Fig. 4. Compressor states in a condition where the pistons of Active II and Active IV should be saturated.

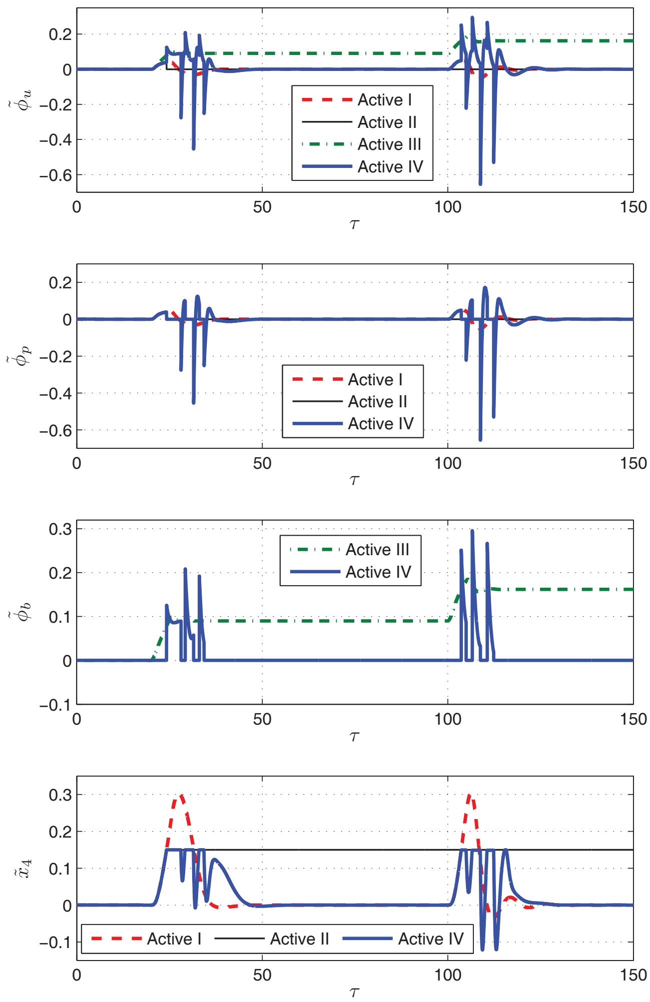

Fig. 5. Control mass flow and piston stroke in a condition where the pistons of Active II and Active IV should be saturated.

”un-actuated” compressor to enter deep surge. The Active III controller stabilized surge, but the steady state is higher than the desired due to the blow off moved the system to a new equilibrium. The Active IV controller was able to stabilize surge and blew off some flow to compensate the piston saturation. Only using Active I and Active IV, the compressor operates stable at the desired operating point in the left side of the original surge line. Fig. 5 shows the control mass flow and piston stroke of the four systems. The Active IV control mass flow is a result of switching operation between the piston and the blow off valve according to control law (36). Fig. 6 shows the piston velocity, piston position, and piston control force of Active IV.

The second simulation is done by assuming that the piston is jammed at $\tau = 112$. At that time γ_T will be 0.3. The simulation is only performed for Active I, Active III and Active IV, as Active II has been failed to stabilize surge under saturation. Fig. 7 is showing the results. The Active I was entering deep surge when the piston was jammed. The jammed piston did not affect to the Active III as piston was not used in the system. Active IV was able to keep the compressor in stable operation by blowing off flow when the piston is jammed and the compressor operating point moved to the stabilized operating point by the back-up system which is exactly the same as the operating point of Active III.

5. CONCLUSIONS AND FUTURE WORKS

5.1 Conclusions

Piston-actuated active surge control combined with blow off surge control as back-up was presented. Active surge control system with piston actuation which is able to discharge flow from plenum and inject flow into plenum showed better performance than the blow off surge control system which is only able to discharge flow from plenum. The simulation result showed that the combined system can both enlarged the compressor map and still keep safe operation even though the piston should fail. Providing longer piston stroke or applying some control method may avoid piston saturation, however a back-up system is still needed to assure safe compressor operation, for instance if the piston should jam.

5.2 Future works

This work is continued by: 1) analyzing the region of attraction of each surge control system and the combined system, 2) applying non-linear control in each surge con-

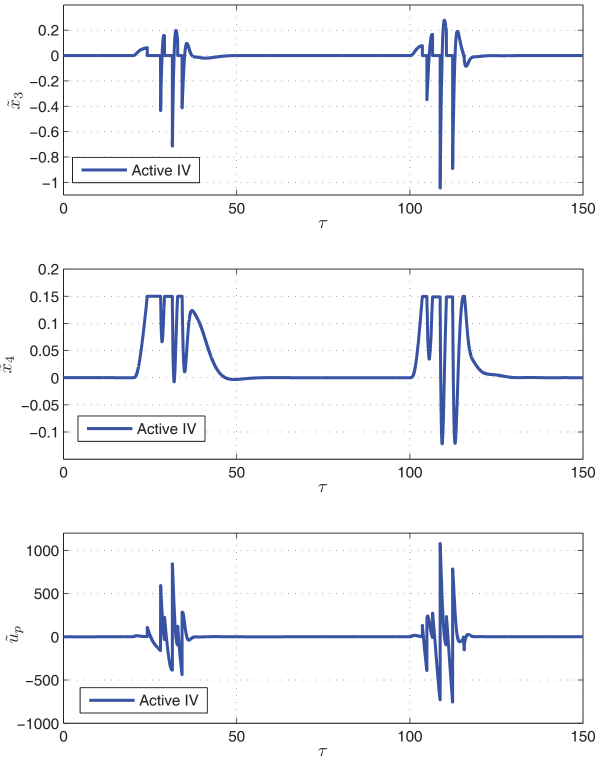

Fig. 6. The piston states and control force of Active IV in a condition where the piston should be saturated.

trol, 3) stability analysis of the switching between the two mode, and 4) experimental test in a laboratory scale.

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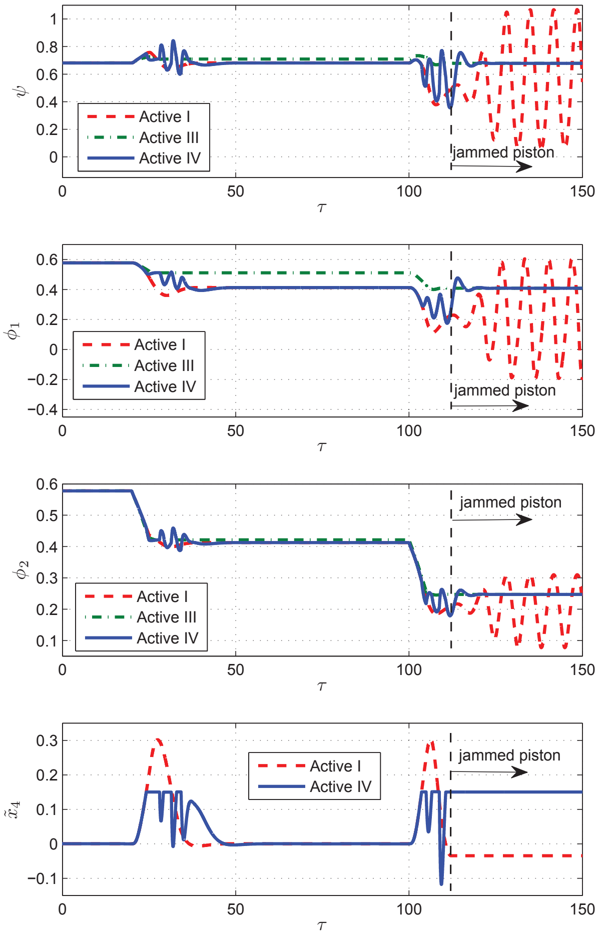

Fig. 7. Compressor states and piston stroke in a condition where the pistons of Active I and Active IV are jammed at $\tau = 112$.

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