Springer

Journal of Mechanical Science and Technology

Journal of Mechanical Science and Technology 23 (2009) 1406~1414

www.springerlink.com/content/1738-494x
DOI 10.1007/s12206-008-1125-y

A pressure output feedback control of turbo compressor surge with a thrust magnetic bearing actuator †

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(Manuscript Received March 6, 2008; Revised October 3, 2008; Accepted November 27, 2008)

Abstract

This paper presents a pressure output feedback control of turbo compressor surge using tip clearance actuation with a thrust magnetic bearing actuator. First, static and dynamic compressor models were obtained for a commercial turbocharger, and the surge point was found through local stability analysis. Then, the effect of tip clearance on the compressor pressure rise was derived, and Lyapunov analysis was used to establish a limit of stability with tip clearance modulation. After that, a linear quadratic (LQ) state feedback control was designed considering the limit established by the Lyapunov analysis. In addition, an extended Kalman filter (EKF) was designed to estimate the mass flow rate from the measured compressor pressure. Finally, the pressure output feedback controller was built by combining the LQ state feedback control and EKF. Control simulation proved the effectiveness of the output feedback controller.

Keywords: Compressor surge; Tip clearance modulation; Magnetic bearings; Output feedback controller

1. Introduction

The stable operation of compressors is degraded by the onset of two flow instabilities: surge and rotating stall. These flow instabilities reduce compressor efficiency and can cause severe damage to the compressor system. Hence, considerable research has been directed toward controlling surge/rotating stall using different schemes to both prevent damage and increase the life span of the machine [1].

Active magnetic bearings are widely used in turbomachinery applications because of their unique advantages: oil-free operation, vibration control and on-line monitoring. In addition, magnetic bearings offer the potential for active control of the compressor flow instabilities. For turbo compressors supported by magnetic bearings, the blade tip clearance can be adjusted by moving the shaft either radially or axially. Using radial magnetic bearings as servo actuators, Spakovsky *et al.* [2] obtained a 2.3% reduction of the stalling mass flow. Sanadgol and Maslen [3-5] have provided a theoretical basis to show that a magnetic thrust bearing can be used as an actuator for active surge control of unshrouded centrifugal compressors. Through simulation, they have demonstrated that compressor surge can be stabilized despite extreme changes in the throttle valve position, which results in an extended stable operating region.

† This paper was recommended for publication in revised form by Associate Editor Dong Hwan Kim

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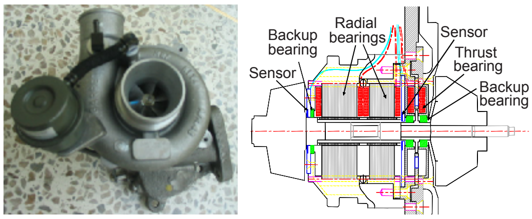
Although a mass flow rate measurement is theoretically the best choice for the control of compressor instabilities, pressure measurement is used for the proportional feedback control in most cases [1]. Compressor pressure is much easier to measure than the mass flow rate, and pressure sensors usually have wider bandwidth than flow sensors. Therefore, better control performance can be achieved if the compressor instability can be controlled by using dynamic mass flow rates estimated from the measured pressures. Hence, pressure output feedback controls have been previously studied in [6, 7]. These studies, however, dealt with axial compressors and throttle valve control without consideration of process and measurement noise in the design of the controller.

This paper presents pressure output feedback control of compressor surge with a thrust magnetic bearing actuator. First, static and dynamic compressor models were established for a commercial turbocharger and the surge point was found through local stability analysis. Then, a mathematical model of the tip clearance effect on the compressor pressure rise was derived by extending the Greitzer compressor model [8], and the stabilizing feedback gain limit of the tip clearance modulation was established by Lyapunov analysis. After that, a linear quadratic (LQ) state feedback control was designed considering the stability limit. In addition, an extended Kalman filter (EKF) was designed to estimate mass flow rate from the measured compressor pressure. Pressure output feedback controllers were constructed by combining the LQ state feedback controls with the EKF. A simulation was performed using the compressor model of a commercial turbocharger, and the result proves the effectiveness of the designed pressure output feedback controller for the compressor surge.

2. Compressor of a turbocharger

In this paper, an active control method of the compressor surge is investigated numerically by using the compressor model of a commercial turbocharger shown in Fig. 1(a). The turbocharger consists of a single-stage centrifugal compressor and a waste gate turbine. This compressor can be operated up to 150,000 rpm with floating ring journal bearings. For active surge control using axial tip clearance actuation, the journal bearings of the turbocharger are assumed to be replaced with magnetic bearings, as shown in Fig. 1(b).



(a) Turbo charger (b) AMB turbocharger test rig

Fig. 1. A commercial Turbocharger.

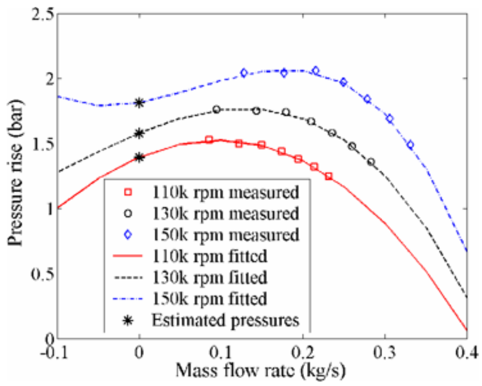


Fig. 2. Approximated compressor map of the commercial turbocharger in Fig. 1.

2.1 Static compressor characteristic

A compressor characteristic curve is approximated by a cubic approximation of $\psi_c(\phi_c)$ [9, 10].

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

where $\psi_c(0)$ is the valley point of the characteristic located on the compressor pressure rise axis and $\phi_c = 2F$ corresponds to the dimensionless compressor mass flow rate at the top point of the characteristic.

The top point could be considered as the surge point of the compressor. Then, the compressor characteristics at each rotating speed ω can be formulated as a cubic polynomial of mass flow rate:

$$\psi_c(\phi_c, \omega) = c_0(\omega) + c_1(\omega)\phi_c^2 + c_2(\omega)\phi_c^3 \tag{2}$$

It is very difficult to measure the compressor characteristic beyond the surge point. The compressor characteristic at zero mass flow rate can be approximated by using a theoretical calculation [8]. Therefore, the coefficients $c_0(\omega)$, $c_1(\omega)$, and $c_2(\omega)$ of the entire compressor map can be approximated by data measured at the stable operating points and the calculated point at zero mass flow rate, as shown in Fig. 2.

2.2 Compressor dynamics

Gravdahl derived a one-dimensional compressor system model described by Eqs. (3a)-(3c) [11]. The first two equations are basically the same as Greitzer's model [12], and the last Eq. (3c) balances the torque on the rotating shaft, which is ignored in this paper by assuming a constant speed operation.

$$\dot{p}_p = \frac{a_{01}^2}{V_p}\left(\dot{m} - \dot{m}_t\right) \tag{3a}$$

$$\ddot{m} = \frac{A_c}{L_c}\left(\psi_c\left(\dot{m},\omega\right)p_{01} - p_p\right) \tag{3b}$$

$$\dot{\omega} = \frac{1}{I_r}\left(\tau_d - \tau_c\right) \tag{3c}$$

Here, the throttle mass flow rate is given by: $\dot{m}_t = k_t\sqrt{p_p - p_{01}}$

2.3 Local stability analysis

To find where the surge points are located, local stability analysis is performed. The local stability near an equilibrium point can be investigated by using a linearized plant of the nonlinear compressor characteristics. First, an equilibrium point is determined by setting the derivatives of Eqs. (3) to zero. Then, the solution of the following equation (4) becomes the equilibrium point.

$$\dot{m}_{te} - k_t\sqrt{p_{pe} - p_{01}} = 0, \quad \psi_c p_{01} - p_{pe} = 0 \tag{4}$$

The linearized plant in the equilibrium point can be written as

$$\begin{bmatrix} \dot{\hat{p}}_p \\ \ddot{\hat{m}} \end{bmatrix} = \begin{bmatrix} -\dfrac{a_{01}^2}{V_p\sqrt{p_{01}}}\dfrac{k_t}{2\sqrt{\dfrac{p_p}{p_{01}}-1}} & \dfrac{a_{01}^2}{V_p} \\ -\dfrac{A_c p_{01}}{L_c} & \dfrac{A_c p_{01}}{L_c}\dfrac{\partial\psi_c}{\partial m}\bigg|_{m=m_e} \end{bmatrix} \begin{bmatrix} \hat{p}_p \\ \dot{\hat{m}} \end{bmatrix} \tag{5}$$

The system is controllable and observable in the entire range. According to the change of the equilibrium mass flow rate, the system pole movements are shown in Fig. 3. As expected, the system becomes unstable as the mass flow rate decreases. However,

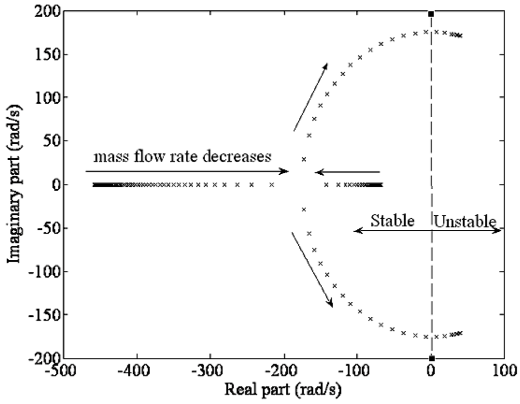


Fig. 3. Pole movements with equilibrium mass flow rate decrease.

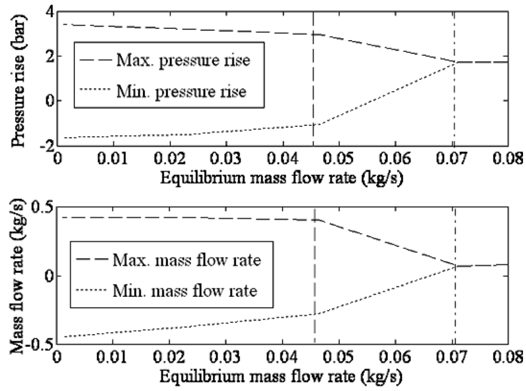


Fig. 4. Limit cycle amplitudes of compressor surges.

the compressor system has a stable limit cycle in the phase portrait even at the unstable region, which is called compressor surge.

2.4 Limit cycle of the compressor surge

The limit cycles of compressor surge can be numerically determined. The system can be modeled with a difference equation and simulated by using the Euler forward method for its simplicity. One percent process noise of the compressor model is added during the numerical simulation. As mass flow rate is decreased , the limit cycle amplitudes of compressor surges are calculated and shown in Fig. 4. The peak to peak amplitude of the limit cycle increases gradually as the mass flow decreases. The small amplitude limit cycles right after the destabilization point (between the two vertical marker lines) are called a *mild surge*. However, fully developed limit cycles arise as the mass flow decreases further after the destabilization point (left-most region in the figure), which is called a *deep surge*.

3. Control of compressor surge using tip clearance actuation

3.1 Tip clearance effect

The tip clearance between the blade tips and the adjacent shroud has a large effect on the compressor performance. The reversal air flow from the pressure side to the suction side of the compressor blade is referred to as tip leakage. As the tip clearance increases, the induced leakage reduces the energy transfer from the impeller to the fluid, which consequently produces a substantial loss in efficiency and a pressure rise. If the tip clearance at the best efficiency condition is changed, variations of the static compressor characteristics are usually expressed as loss in efficiency. For the leakage loss in centrifugal compressors, a simple model was developed by Senoo [13], shown in Eq. (6). The relative efficiency loss is nearly proportional to the ratio of the tip clearance to the blade height at the impeller outlet provided that this ratio is less than 0.1:

$$\frac{\Delta\eta}{\eta} = 0.25\frac{\Delta l}{b_2} \tag{6}$$

The compressor static isentropic efficiency can be calculated based on the perfect gas and the isentropic compression assumptions as:

$$\eta_i(\dot{m}) = \frac{T_{01}c_p\left(\left(\frac{p_p}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right)}{c_p\left(T_p - T_{01}\right)} \tag{7}$$

Applying a quasi-steady approximation for the effects of the tip clearance based on Senoo's relation of Eq. (6) and the isentropic compression efficiency, Eq. (7), Sanadgol and Maslen [3] have derived that the compressor characteristic can be expressed as:

$$\psi_c(\dot{m},\Delta l) = \left(1 + \frac{\psi_{c0}^{\frac{\gamma-1}{\gamma}} - 1}{1 + \frac{0.25\Delta l}{b_2}}\right)^{\frac{\gamma}{\gamma-1}} \tag{8}$$

The calculated tip clearance effect on the pressure rise is shown in Fig. 5. The compressor pressure rise decrease as tip clearance increases.

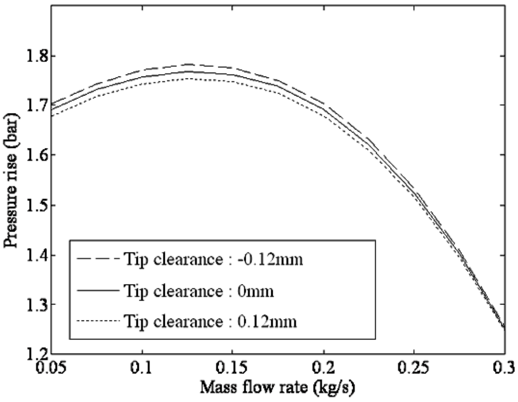


Fig. 5. Tip clearance effect on the pressure rise.

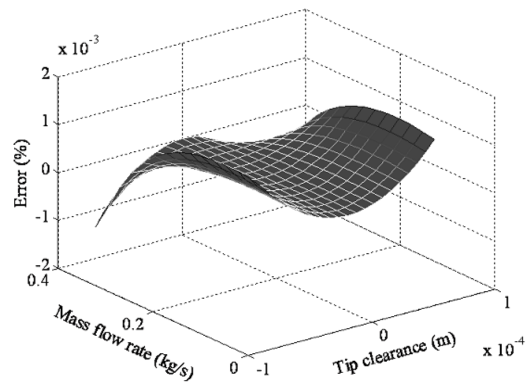


Fig. 6. Error due to the tip clearance approximation.

To establish a dynamic model for the surge control using tip clearance actuation, the tip clearance effect of Eq. (8) and Gravdahl's model of Eq. (3) are combined and the resulting compressor dynamic model is:

$$\ddot{m} = \frac{A_c}{L_c}\left(\left(1 + \frac{\psi_{c0}^{\frac{\gamma-1}{\gamma}} - 1}{1 + \frac{0.25\Delta l}{b_2}}\right)^{\frac{\gamma}{\gamma-1}} p_{01} - p_p\right) \tag{9}$$

We assume that only the axial rotor motion modulates the instantaneous compression efficiency or the compressor pressure rise. Since the tip clearance modulation is usually small, the nonlinear dynamic model of Eq. (9) can be approximated by

$$\ddot{m} = \frac{A_c}{L_c}\left(\psi_{c0}p_{01} - p_p\right) - \frac{A_c}{L_c}\frac{\gamma}{\gamma-1}\left(\psi_{c0}^{\frac{\gamma-1}{\gamma}} - 1\right)\psi^{\frac{1}{\gamma}}\frac{0.25}{b_2}p_{01}\Delta l \tag{10}$$

This model is a good approximation of the original nonlinear system as shown in Fig. 6. The linearized equation (10) has very small relative error and can confidently be used in a control design.

3.2 Lyapunov analysis for stabilizing gain limit

First, simple Lyapunov control with mass flow rate feedback is applied to control compressor surge by using tip clearance actuation. Applying the following change of coordinates to the system, the origin becomes the equilibrium under study (intersection of the compressor and the throttle characteristics).

$$\hat{\dot{m}} = \dot{m} - \dot{m}_e, \quad \hat{\dot{m}}_t = \dot{m}_t - \dot{m}_{te}, \quad \hat{p}_p = p_p - p_{pe},$$
$$\hat{\psi}_c = \psi_c - \psi_{ce} \text{ and } \hat{l} = l - l_e \tag{11}$$

The model can be written in these new coordinates as:

$$\begin{cases} \dot{\hat{p}}_p = \dfrac{a_{01}^2}{V_p}\left(\hat{m} - \hat{m}_t\right) \\ \ddot{\hat{m}} = \dfrac{A}{L_c}\left(\hat{\psi}_c\left(\hat{m}, \Delta\hat{l}\right) p_{01} - \hat{p}_p\right) \end{cases} \tag{12}$$

Now, assume the use of mass flow rate feedback for tip clearance actuation via the control law:

$$\Delta\hat{l} = -k_L \hat{m} \tag{13}$$

A sufficient condition for the equilibrium point to be globally exponentially stable is that the gain k_L is chosen such that:

$$k_L > \frac{\partial\hat{\psi}_c}{\partial\hat{\dot{m}}} \Big/ \frac{\partial\hat{\psi}_c}{\partial\Delta\hat{l}} \tag{14}$$

We will use the Lyapunov control gain as a reference to tune the LQ control gain.

3.3 LQ state feedback control

A state feedback controller is necessary in order to build an output feedback controller. A simple LQ control is used as a state feedback control in this paper. For LQ control, an optimal output feedback gain is calculated to minimize the quadratic cost function, J in Eq. (15), of the continuous-time state-space model: $\dot{x} = Ax + Bu$.

$$J = \int_0^\infty \left(x^T Q x + u^T R u\right) dt \tag{15}$$

The LQ control can be designed by using the linearized system near an equilibrium point. The LQ control gain value is tuned considering the stabilizing gain limit of Eq. (14) established by Lyapunov analysis. Although a minimum energy control that uses the least amount of control effort is applied and Q in Eq. (15) becomes zero, the mass flow rate feedback gain of the LQ control is still twice as high as the Lyapunov control gain limit.

Simulation results of both Lyapunov and LQ controls in a mild surge (equilibrium mass flow 0.055 kg/s) are shown in Fig. 7. Lyapunov control gain is 0.0168 m·s/kg, while the gain of the minimum energy LQ control is 0.0336 m·s/kg (pressure feedback gain has a very small value of 2.02e-6 m/bar). The LQ control shows faster stabilizing performance due to the higher gain. It can be seen that the dynamic characteristics of the closed-loop system are mainly determined by the mass flow rate feedback gain.

4. Output feedback comntrol

4.1 Extended Kalman Filter (EKF) [14, 15]

To build an output feedback controller, a state estimator is required to estimate the mass flow rate from the pressure measurement. Although the Kalman filter has been most widely used to estimate states from noisy data in linear systems, the standard Kalman filter cannot be applied to nonlinear systems such as the compressor model in this paper. The EKF

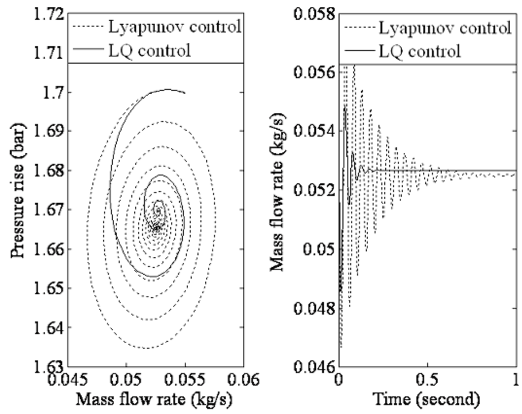


Fig. 7. LQ and Lyapunov control.

is the traditional form of nonlinear filtering and it essentially truncates the nonlinear dynamics at the first order to allow the use of linear Kalman filter equations. The EKF procedure is summarized in Eqs. (16-21). The EKF algorithm is done in two steps: prediction and correction. The prediction step involves projecting or transforming the state in forward time-step using the current knowledge of the model (Eq. (16)), as shown in Eq. (19). The second step is required to correct the prediction by using the measurement at the new time step (Eq. (17)), as shown in Eq. (20). The measurement is blended with the prediction in an optimal manner in order to minimize the mean square error of the estimate. This blending factor is called the Kalman gain in Eq. (21).

Extended Kalman filter procedure

State-update equation: $x_{k+1} = f\left(x_k, u_k, v_k\right)$ (16)

Measurement equation: $y_k = g\left(x_k, n_k\right)$ (17)

Initialize with $\hat{x}_0 = E[x_0], P_{x0} = E\left[(x_0 - \hat{x}_0)(x_0 - \hat{x}_0)^T\right]$ (18)

For $k = \{1, 2,\ldots, \infty\}$, the time-update equations of the extended Kalman filter are

$$\hat{x}_k^- = f\left(\hat{x}_{k-1}, u_{k+1}, \bar{v}\right), P_{x_k}^- = A_{k-1} P_{x_{k-1}} A_{k-1}^T + B_k R^v B_k^T \quad (19)$$

and the measurement-update equations are

$$\hat{x}_k = \hat{x}_k^- + K_k\left[y_k - g\left(\hat{x}_k^-, \bar{n}\right)\right] \quad (20)$$

$$P_{x_k} = \left(I - K_k C_k\right) P_{x_k}^-$$

$$K_k = P_{x_k}^- C_k^T \left(C_k P_{x_k}^- C_k^T + D_k R^n D_k^T\right)^{-1} \quad (21)$$

where $A_k \equiv \left.\frac{\partial f\left(x, u_k, \bar{v}\right)}{\partial x}\right|_{x = \hat{x}_k}$, $B_k \equiv \left.\frac{\partial f\left(\hat{x}_k^-, u_k, v\right)}{\partial v}\right|_{v = \bar{v}}$,

$C_k \equiv \left.\frac{\partial g\left(x, \bar{n}\right)}{\partial x}\right|_{x = \hat{x}_k}$, $D_k \equiv \left.\frac{\partial g\left(\hat{x}_k^-, n\right)}{\partial n}\right|_{n = \bar{n}}$ and R^v and R_n are the noise covariance of v_k and n_k, respectively. The noise means are denoted by $\bar{n} = E[n]$ and $\bar{v} = E[v]$, and are usually assumed zero.

A simulation in a mild surge (equilibrium mass flow 0.0525 kg/s) was performed to verify the estimation performance of the designed EKF. Both 1% state and measurement noises (average pressure noise: 0.017 bar) were introduced during the simulation.

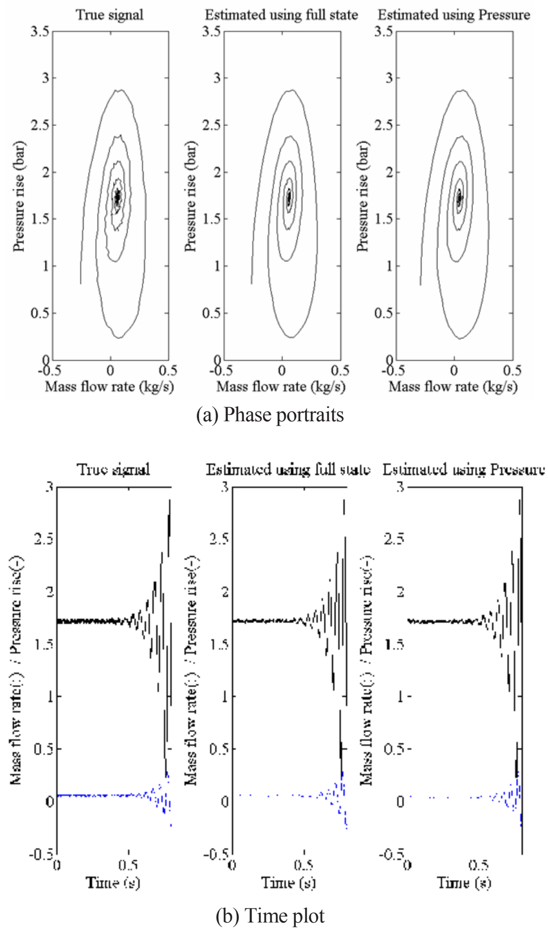


Fig. 8. Estimation of nonlinear compressor system using EKF.

Estimation using full states (both pressure rise and mass flow rate) and estimation using output (only pressure rise) signal are compared in Fig. 8. The KALMTOOL was used to implement the extended Kalman filtering [15]. Noise could be filtered significantly by the EKF. In addition, the EKF only with the pressure signal could have comparable quality estimations of both pressure rise and mass flow rate, although a surge takes place and the mass flow rate and pressure rise vibrate largely.

4.2 Simulations of output feedback controls

Control simulation of a mild surge (equilibrium mass flow is 0.0550 kg/s) was performed to test the pressure output feedback control using EKF. Both 1% process and measurement noise (average pressure noise: 0.017 bar) were introduced. The LQ control was used to stabilize the compressor surge. There is no control during the first 0.6 second and after that, the control turns on. The output feedback control using the EKF has comparable control performance to control using mass flow rate signal, as shown in Fig. 9. The thrust magnetic bearing has a limited clearance of ±0.28 mm between the bearing and the housing. Thus, tip clearance actuation is limited to a ±0.15 mm range. The tip clearance variation during the control is shown in Fig. 9(c), which illustrates that only a small variation of the tip clearance is required in order to stabilize the surge.

The simulation result shows that the designed output feedback controller can stabilize compressor surge effectively. Although a surge phenomenon begins, we can get stability of the compressor by using the output feedback control. Also, this result allows us to expect more efficiency when supporting the rotor of the turbocharger by magnetic bearings.

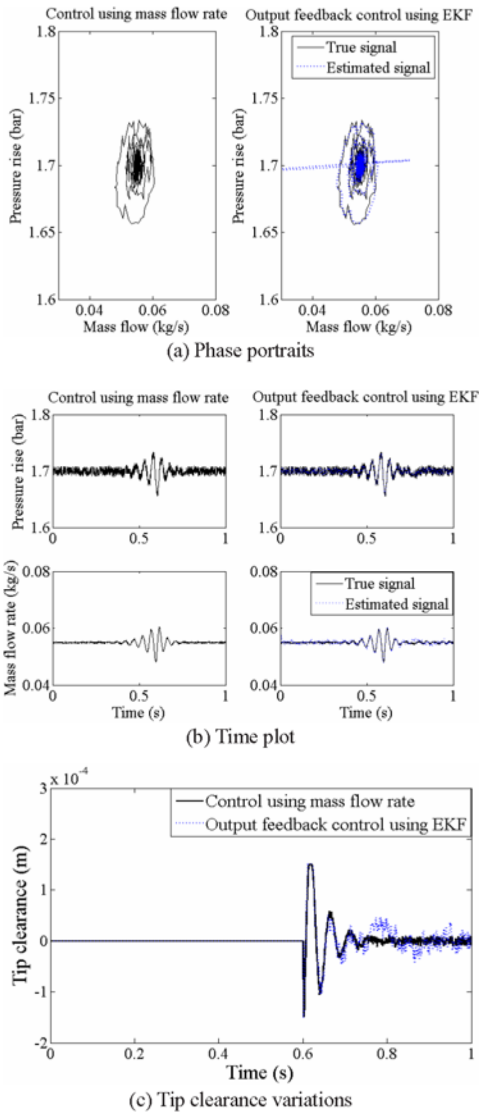


Fig. 9. The output feedback control simulations.

5. Conclusion

We have discussed pressure output feedback controls of compressor surge using tip clearance actuation with a thrust magnetic bearing actuator. First, we modeled the compressor characteristic of a commercial turbocharger and derived the tip clearance effect on the compressor pressure rise by extending the Greitzer compressor model. Based on this model, Lyapunov analysis was performed to establish the limit of the stable feedback gain. Then, an LQ state feedback controller was designed considering the stability limit of the feedback. In addition, an EKF was designed to estimate the mass flow rate by measuring the compressor pressure. Finally, pressure output feedback controllers were designed by combining the LQ feedback control and the EKF. We verified with simulation that the output feedback controller can stabilize the compressor surge effectively with axial magnetic bearing.

Acknowledgement

Authors wish to thank KOSEF and KRF for their grant of the international cooperative research program.

Nomenclature

a_{o1} : Speed of sound at the ambient condition
A and B : State equation for LQ control
A_c : Compressor duct area
A_k, B_k, C_k and D_k : Linearized discrete state and measurement equation matrices at k^{th} step
b_2 : Blade height at impeller outlet
c_0, c_1 and c_2: Curve-fitted coefficients of compressor characteristic polynomial
c_p : Specific heat at constant pressure
F : Compressor characteristic semi width
f : Nonlinear state equation for
g : Nonlinear measurement equation
H : Compressor characteristic semi height
I_r : Compressor rotor inertia
J : Linear quadratic cost function

K_k : Discrete Kalman gain at k^{th} step
k_t : Throttle parameter
k_L : Lyapunov control gain
L_c : Compressor duct length
l : Tip clearance
$\dot{m}$: Compressor mass flow rate
$\dot{m}_t$: Throttle mass flow rate
n_k : Measurement noise at k^{th} step
P_{x0} : Covariance matrix of states
p_{o1} : Ambient pressure
p_p : Plenum pressure
Q : State weighting matrix for LQ control
R : Control weighting matrix for LQ control
R_v : Process noise covariance
R_n : Meausrement noise covariance
t : Time
u :Control input for LQ control
u_k : Control input at k^{th} step
V_p : Plenum volume
v_k : Process noise
x : State variable
x_k : Discrete state variable at k^{th} step
y_k : Measurement signal at k^{th} step
ϕ_c : Compressor mass flow rate coefficient
Γ : Specific heat ratio
η : Efficiency of compressor
η_i : Isentropic efficiency of compressor
τ_c : Compressor torque
τ_d : Drive torque
ω : Rotating speed
Ψ_c : Compressor static pressure characteristic

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