International Journal of Science and Research (IJSR)
ISSN (Online): 2319-7064
Index Copernicus Value (2013): 6.14 | Impact Factor (2014): 5.611

State Observer Design for Active Surge Control in Centrifugal Compressors

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Abstract: ***A state observer for the delays found in mass flow and pressure rise measurements in active surge control of centrifugal compressor is proposed in this work. The active surge controller have proved to be mathematically stable and shown to perform adequately. However, the surge control by drive torque actuation and close coupled valve, both will face a problem regard to time delay. First, a low order centrifugal compressor model is presented where the states are mass flow, pressure rise, speed of the spool and develop the compressor characteristics on the basis of energy transfer. Based on active surge control by close coupled valve simulation, the surge is initiated under the presence of time delays in either mass flow or pressure rise measurements. As a possible solution for the measurement delays, a state observer have been implemented and tested, but with limited success.***

Keywords: Centrifugal compressor, Close coupled valve(CCV), Active surge control, State observer

1. Introduction

The operation of many centrifugal compressors, including those in turbo chargers is normally limited by the occurrence of surge[1]which is a self-exited system oscillation. Surge can be classified according to the amplitude of mass flowfluctuations, namely mild surge, classicsurge, deep surge and modified surge. The flow reversal during deep surge can make damage for compressor and large vibrations following the surge can leads to the damaging of other related equipments[3].

Many papers covering the area of surge control have been published. Close coupled valve control is considered one of the most promising scheme in this field. [4],[5],[7]. The cause of surge in compression system is that the throttle line crosses the compressor characteristics in in an area of the compressor characteristics slope. A close coupled valve utilizes the pressure drop to make the slope of the equivalent compressor to negative, and thereby stabilizing the system.

The major problem associated with CCV control is that surge occurrence due to measurement delays[7].Another drawback of CCV is that the valve introduces a pressure drop in compressor system as in [5]. The possible solution for surge occurrence due to time delays are state observers. Another method of active surge control is active torque surge control. It's control law design and proofs are presented in [7]. Drive torque actuation also faces the time delay problem. Since the compression processes are inherently nonlinear, the controller design must be able to deal efficiently with process non linearities and variations. An observer design for nonlinear system is presented in [9]. With the help of nonlinear system observer design, developed a full order observer for surge control of compression system even under the presence of time delays found in measurements.

The paper has been organised as follows. A low order model for centrifugal compressor is given in Section II. Section III explains the theory of energy transfer and compressor characteristics. Section IV contains the details about active surge control technique.State observer design is shown in section V. Simulation results and associated findings are drawn in section VI. The major conclusions and scope of this work have given in following sections.

2. Mathematical Modelling of Centrifugal Compressor

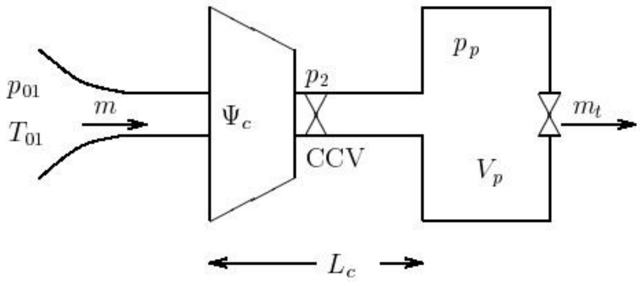


Figure 1: Compressor system with close coupled valve.

Consider a compression system consisting of centrifugal compressor, close coupled valve, compressor duct, plenum volume and throttle. The throttle can be regarded as a simplified model of turbine. The system is shown in fig(1). The model to be used for controller design is in the form:

$$\dot{P}_p = \frac{a_{01}}{V_p}(m - m_t) \quad (1)$$

$$\dot{m} = \frac{A}{L}(P_2 - P_p) \quad (2)$$

$$U = \frac{D_1}{2J}(\tau_l - \tau_c) \quad (3)$$

Where m is the compressor mass flow, P_2 is the pressure downstream of the compressor, a_{01} is the inlet stagnation velocity, L is the compressor duct length, A is area of impeller eye, J is spool moment of inertia.$\tau_l$is drive torque andτ_c is compressor torque. P_2represents the compressor characteristics. In this work, physical model based compressor[3] characteristics is presented. It is based on the energy transfer in ideal case, which is described in next section.

3. Energy Transfer and Compressor Torque

Energy transfer and compressor torque in ideal situation is explained in [5]. The ideal specific enthalpy delivered to the fluid is given by,

$$\Delta h_{oc,ideal} = \frac{W_{c,ideal}}{m} = \sigma U_2^2 \quad (4)$$

Where σ is slip factor, taken as .9 and that the compressor torque is given by the equation

$$\tau_c = m r_2 U_2^2 \sigma \quad (5)$$

Due to various losses, the energy transfer is not constant[5]. In this analysis, two major losses have been taken into account. 1) Incidence loss 2)Fluid friction loss. Other losses such as inlet casing losses, disc friction losses, leakage losses and collector losses etc. are ignored. Incidence loss is given by the equation

$$\Delta h_i = \frac{1}{2}\,(U_1 - \frac{Cot_{\beta 1b} m}{\rho_{01} A_1})2 \quad (6)$$

where, β_{1b} is the blade inlet angle and ρ_{01} is inlet stagnation density. Fluid friction loss is given by

$$\Delta h_f = C_h \frac{l}{D}\frac{W^2}{2} \quad (7)$$

Energy transfer with frictional and incidence lossfor N = 35000rpm is given in fig.(2)

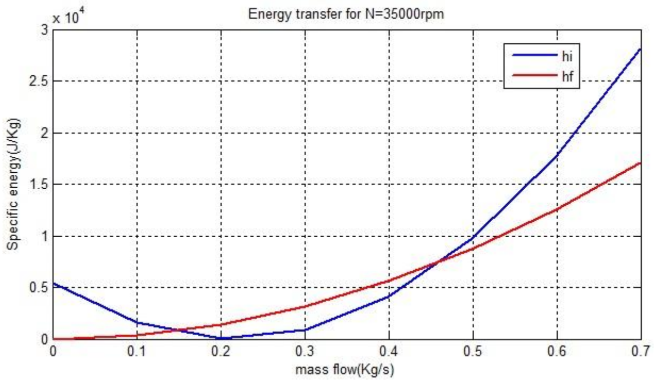


Figure 2: Energy transfer for N=35000rpm

The total specific enthalpy including the losses can be calculated by

$$\Delta h_{oc}(U_1, m) = \Delta h_{oc,ideal} - \Delta h_f - \Delta h_i \quad (8)$$

To find an expression for pressure rise, relation between pressure rise and energy transfer is need. The pressure rise is modeled as

$$P_2 = (1 + \frac{\Delta h_{oc}(U_1,m)}{T_{01} C_p})^{\frac{\gamma}{\gamma-1}} * p_{01} = \Psi_c(U_1, m) p_{01} \quad (9)$$

Where the losses have been taken into account,$\Psi_c(U_1, m)$ is the compressor characteristics for different rotor speed, which is simulated as in fig. 3

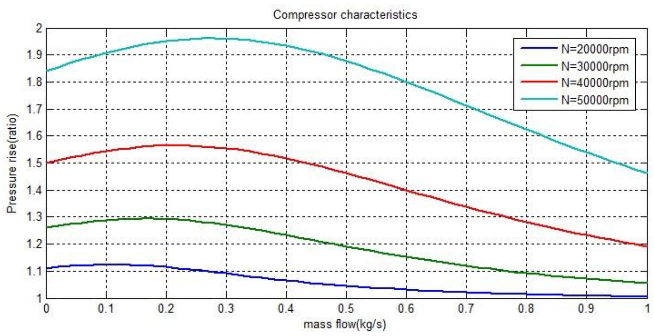


Figure 3: Compressor Characteristics

The reason for equilibria to the left of the surge line facing unstable and causing the compressor to go into surge[2], is the positive slope of the characteristic in this area. For avoiding the surge, introduce a valve in series with compressor. The pressure drop over this valve will serve as control variable and it will be used to introduce additional friction at low mass flow in order to avoid surge.

4. Active Surge Control by CCV

Active surge control by close coupled valve is given in [7]. The approach is to introduce a valve close to the plenum volume in compressor. CCV means there is no mass storage between the compressor outlet and the valve. CCV control law and its proof are given in [7] and [8].

The surge control law

$$\Psi_v = k_v \hat{m} \quad (10)$$

and the speed control law,

$$\hat{\tau}_l = -K_p \hat{U} - K_i \hat{I} \quad (11)$$

$$\dot{\hat{I}} = \hat{U} \quad (12)$$

Where,K_p>0, $K_i > 0$ and K_v>sup$\{\frac{\delta\Psi(\hat{m},\hat{U})}{\delta\hat{m}}\} + \delta_1$ makes the semi globally exponentially stable.

Another way of approach, active surge control by drive torque actuation which is presented in [8]. Active surge control is a promising way to keep the compressor out of surge. The main issue associated with active surge control is surge occurrence under the presence of measurement delay in either mass flow or pressure rise. Surge will occur even under the very small time delays (about milliseconds) in measurements, which may lead to the damage of whole compression system. Therefore, surge removal under the presence of measurement delays is very crucial for the performance of centrifugal compressors.

5. State Observer for Measurement Delays

The use of observer is a promising way for surge removal even under the presence of measurement delays in centrifugal compression system. The model used for the observer design is:

$$\dot{x} = Ax + G\Upsilon(Hx) + \rho(u, y) \quad (13)$$

$$y = [y_1 y_2]^T = [Cxh(u, x)]^T \quad (14)$$

This structure ensures that the unmeasured states that the unmeasured states enter the compressor dynamics through linear mapping Ax and nonlinear mapping γ(Hx). The observer proposed for the system in equation(13) is:

$$\dot{\hat{x}} = A\hat{x} + G\gamma\big(H\hat{x} + K_2(C\hat{x} - y_1)\big) + \rho(u, y) + K_1(C\hat{x} - y_1)) \quad (15)$$

Define $e = x - \hat{x}$. Consider the Lyapunov function candidate as

$$V(e) = e^T \text{Pe} . \quad (16)$$

The time derivative of V(e) can be formulated as

$\dot{V}(t, e) \le -2e^T Q_1 e2(v - w)^T Q_2(\gamma(v) - \Upsilon(w))$ provided that the linear matrix inequality[LMI].

$$\begin{bmatrix} P(A + K_1C) + (A + K_1C)^T P + Q - 1 & PG + (H + K_2C)^T Q_2 \\ G^T P + Q_2(H + K_2C) & 0 \end{bmatrix} \le 0 \quad (17)$$

is satisfied. The observer is from[8], modified to fit the model as:

$$\begin{bmatrix} 0 & \frac{a^2{}_{01}}{V} & 0 \\ \frac{-A}{L} & -c_2 & 0 \\ 0 & -c_3 & \frac{-K_f D-1}{2J} \end{bmatrix} \quad B=\begin{matrix} -1 & 0 & 0 \\ 0 & 1 & 1 \\ 0 & 0 & -1 \end{matrix} \tag{18}$$

$$H=\begin{matrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{matrix} \quad C=\begin{matrix} 1 & 0 & 0 \\ 0 & 0 & 1 \end{matrix} \tag{19}$$

$$\rho=\begin{matrix} 0 \\ -c_3 \quad U_1 \\ \frac{D_1}{2J}U_1 + C_4U_1 \end{matrix} \tag{20}$$

6. Simulation Results and Discussions

Based on the state equations of centrifugal compression system, simulation of centrifugal compression system without any controller have been done. In fig. (4), the effect of surge is simulated. Both the pressure and mass flow oscillates and the flow reversal occurs in the compressor. This is what all surge controllers seek to avoid since it is clear that a fluid flowing in such a manner will damage whole compression system.

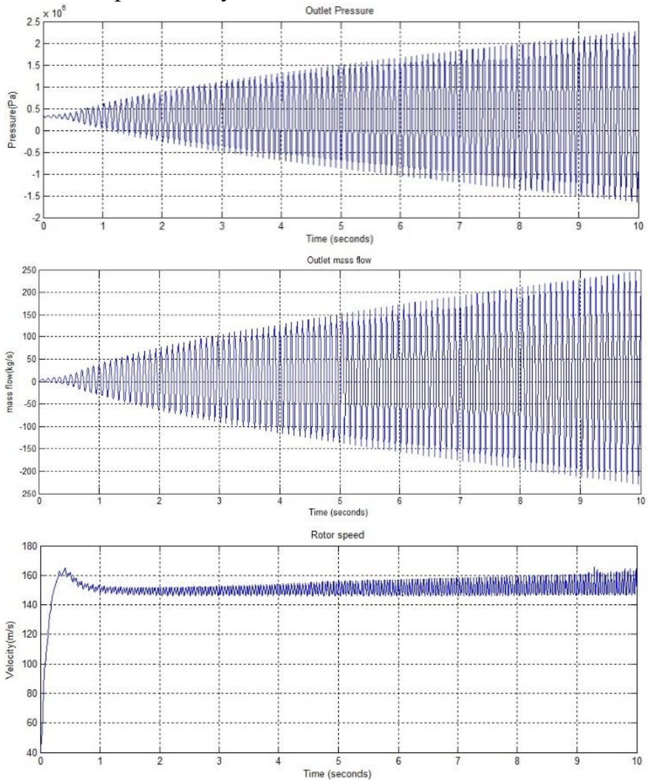


Figure 5: Simulation of centrifugal compressor without CCV

The antisurge controller contains both CCV controller and torque controller based on a PID controller. Three different simulations have been carried out. The first demonstrates the CCV controller for avoiding surge, second is same test, but without CCV. In third test, a delay in the mass flow measurement fed to surge controller is added to the system. Simulations are with a fixed throttle gain Kl=.0007. Fig.(5) shows surge control simulations with CCV.

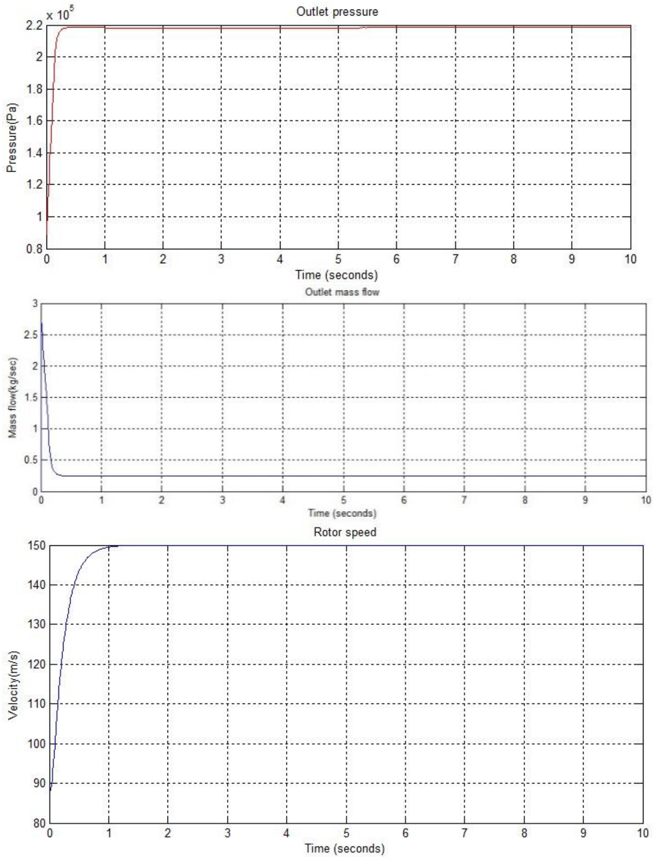


Figure 6: Surge control simulation with CCV

Due to the presence of time delay measurements, surge will be initiated. This is demonstrated in fig. (7).

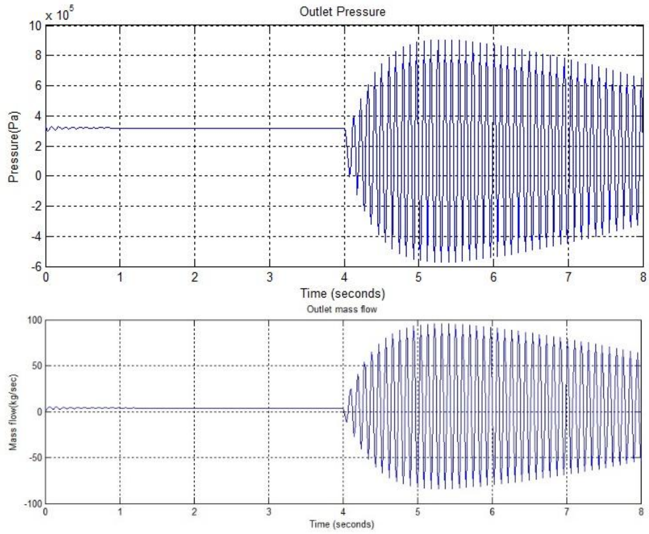
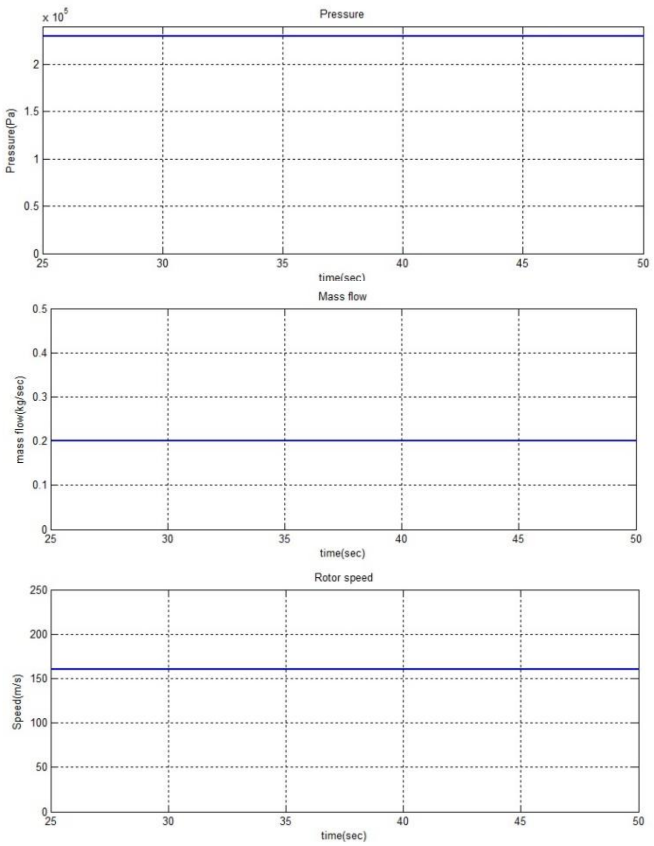


Figure 7: Simulation with time delay 4s.

A measurement delay will make the controllers unable to react fast enough to changes occurring inside the compressor which in turn will force the compressor to enter the surge. A possible solution to this problem is the development of state observers for the compression system. Fig. (7) shows the observer output for the compression system. The observer used currently only are valid in limited domain of state space.



7. Conclusion

The modelling of the compressor characteristics has been done on the basis of energy losses in the compressor stage. Through simulations, it was confirmed that the compressor can operate stable and reach desired speed in the previous unstable area to the right of the surge line in compressor map. A measurement delay will make the controllers unable to react fast enough to changes occurring inside the compressor, which in turn will force the compressor to enter the surge. A possible solution to this problem is to develop a state observer for the compressor system. The observer represents a promising solution to measurement delay problem.

8. Future Scope

Most industrial compressor systems measure volume flow, not mass flow as normally done in research work. Since, mass flow measurements require knowledge of the gas composition, which is difficult in an offshore processing system, the active surgecontroller must handle volume flow. The problem due to measurement delay is reasonable assume that an active surge controller based on volume flow measurements will face this problem. As discussed in this work, the possible solution is to use state observers. But work done on nonlinear observers for compressor systems is strongly limited and will definitely need to be further investigated.

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