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Optimal design of full order state observer for active surge control in centrifugal compressors using genetic algorithm

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ABSTRACT

This paper presents the design of a genetic algorithm (GA) based full order state observer for a compressor system. The observer measures the compressor system parameters and computes the difference between the measured output and the setpoint, which is used as feedback to the active surge controller. The proposed control structure with a GA tuned compressor characteristic is required to minimize system oscillations and energy losses induced by the piston actuation gas recycling technique. The design method for GA based full order observer begins with a state observer modelling, observer error estimation based on the magnitude of fluid friction and mechanical effects on measured parameters. The aim of the controller is to regulate the compressor driver speed such that the measured output achieves its required set point. In order to achieve a maximum compressor operating capacity, the compressor characteristics were optimized using GA, where the mass flow was maximized to improve the compressor efficiency. Simulations were carried out, and results showed the viability of GA based full order state observer compared with the piston actuation recycle method to control active surge in centrifugal compressor systems.

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1. INTRODUCTION

Certain industrial operations, such as refining, power generation and chemical processes, require that fluids flow at very high pressures. Typically, compressors increase the pressure in flowing fluids by mechanically decreasing their volumes [1], [2]. In a compressor system, fluid passes through the suction eye at a lower velocity and exits the impeller circumference faster [3]. Owing to the inherent presence of surge and other unwanted dynamics in the compressor system, an effective control scheme is required for their safe operation [1]. Therefore, the applied control must ensure efficient operation to guarantee economic sustainability. Compressor surge is an unstable operation with non-dimensional characteristics. It occurs because of mass flow and pressure fluctuations. Surge is characterized by a high frequency and can damage the compressor in a few cycles due to mechanical vibrations, oscillations and excessive temperature in the compression system. Thus, avoiding surge guarantees improved stability and safety of the entire compression system's equipment and devices [4]. The surge problem can be tackled using either a surge avoidance strategy or an active surge control scheme. A surge avoidance strategy differs from active surge control in that the latter ensures stabilization of surge oscillations while the former ensures non-occurrence of surges [5].

Active surge preventive control ensures that the compressor is protected from any form of external disturbances or unusual changes in operating conditions [6].

Compressor parameters that can be used for estimation, control and performance assessment include mass flow rate, pressure rise, rotating shaft speed and fluid temperature [7]. Amongst these, the mass flow rate can be actively explored for improved stability and active surge control [8]. To achieve this, direct feedback of the flow channel pressure and outlet tank pressure dissipations are employed to achieve a rapid controller response for surge stabilization [9]. Typically, a surge is controlled by the active deployment of proportional integral and derivative (PID) anti-surge controllers [10]. However, advanced control schemes such as model predictive control (MPC) for active surge control allow for the accommodation of constraints on fluid temperature to ensure effective system and parameter control using a variable or constant speed compressor driver [7]. In most control schemes, variable speed drive (VSD) and recycling valve actuator positions are used as manipulated variables for achieving a safe and stable operation in the compression system [4]. The recycling valve is activated when the controlled pressure exceeds a specified maximum value in the compressor map [4]. The recycling technique using piston actuation control is also employed as one of the methods for surge avoidance to keep away the compressor operating node from a surge portion [3]. The gas recycling process increases the mass flow rate at the compressor inlet to prevent any surge. This process, however, induces system oscillations [7], [11]. In order to mitigate gas recycling effects in compressor operation, there is the need for parameter estimation, which can easily be achieved using state observers [12]. State observers, reduced-order or full order, provide a systematic way of estimating the internal state variables of a system by considering the control variables and measurements of system output [13]. Usually, the design of full order observers is for error estimation, disturbance rejection or system reliability improvement. The observer can compensate for plant deficiencies by computing appropriate observer gain matrices [14]. A full order observer can compensate for the problem of measurement delay; it can also provide a means of VSD control by estimating all state variables for the proper control of mass flow and the compressor efficiency [11], [15], [16].

The compressor efficiency control in a dynamic process with a full order observer involves a complex control, where parameters such as speed, mass flow and pressure rise must be adjusted rapidly. In order to achieve improved compressor efficiency, a genetic algorithm (GA) is used to optimize the flows. GA is a computational intelligence algorithm that uses a genetics analogy to select a candidate of parameters representing the genes for solving a problem [15]. The framework for the GA based compressor control system is shown in Figure 1.

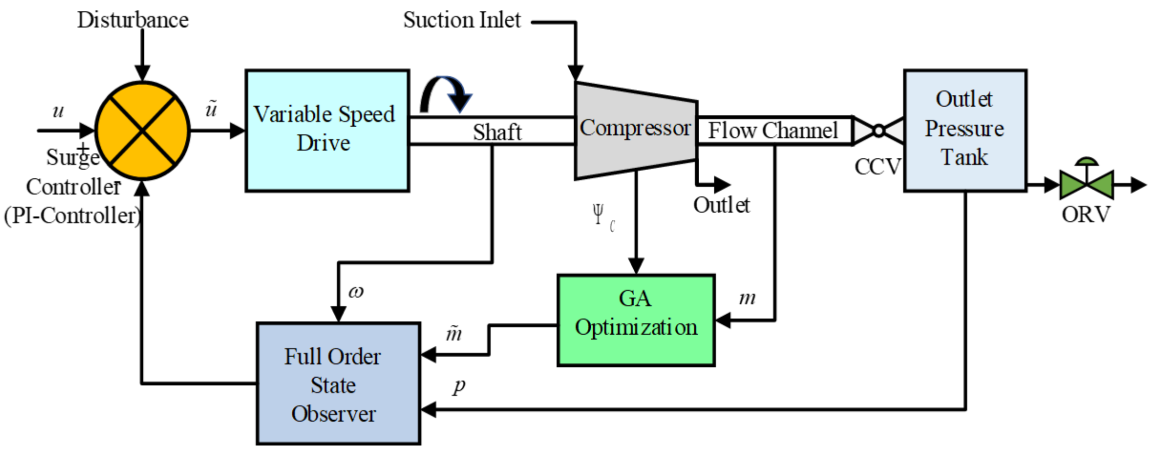

Figure 1. Framework of the compressor control system

The framework ensures a safe control system that can actively prevent surge without much energy loss, drop in compressor efficiency or operation below the designed capacity. Improving compressor operating efficiency is achieved by employing GA to optimize the formulated compressor characteristics optimization model, taking mass flow as the decision variable. The choice of GA in this paper is based on its working principle, which can easily handle multiple model optimization parameters. The GA can substantially run in a reasonable number of iterations and terminates with a chosen ending condition, for example, a fitness convergence, when individuals meet target fitness, the maximum number of generation limits, and the maximum number of stall generations. Also, the optimization model considered in this paper is a single parameter optimization problem. The GA makes it easy to encode this parameter into a chromosome string for easy manipulations.

The rest of the paper is organized as follows: the research methods detailing all the necessary fundamental information are presented in section 2. The modelling and presentation of the proposed observer design is given in section 3. Results are presented and analysed in section 4. Section 5 presented the conclusion and recommendation for future works.

2. RESEARCH METHOD

Three parameters describe the operation of the centrifugal compressor: polytrophic head, mass flow rate and rotating speed. The compressor characteristics that determine the compressor's operating efficiency are formulated in the GA optimization model. The state observer employed in this work computes the system parameters for feedback to the active surge controller. The centrifugal compressor model is derived based on interactions between the mechanical systems and parameters.

2.1. Centrifugal compressor model

The compression system model comprises a compressor, outlet pressure tank, flow channel and outlet regulator valve, which are represented by associated compressor parameters [17]-[19] as shown in (1) to (3):

$$\dot{p} = \frac{a_0^2}{V_p}\left(m - m_r(p)\right) \tag{1}$$

$$\dot{m} = \frac{A_1}{L_c}(\Psi_c(m,\omega)p_{01} - p) \tag{2}$$

$$\dot{\omega} = \frac{1}{J}(\tau_d - \tau_l) \tag{3}$$

where p is the tank pressure, m is the mass flow through the channel, ω is the rotor speed, α_0 is the ultrasonic velocity, V_p is the tank volume, $m_r(p)$ is the regulator mass flow, A_1 is the flow channel area, L_c is the channel length, $\Psi_c(m,\omega)$ is the compressor characteristic, p_{01} is the inlet pressure, τ_d is the drive torque, τ_l is the load torque and J is the inertia of the impeller shaft.

2.2. Compressor surge controller

Various surge controllers have been actively deployed for various active surge control in compression systems. MPC performed commendably well due to its good prediction capabilities in active surge control, and its functionality is relatively suitable for process plants with consistent operational timing control requirements. PID controllers are substantially employed as compressor surge controllers due to their simplicity, operational adaptability, durability and ease of maintenance [10]. As a result of the high sensitivity to noise in the derivative term, proportional-integral (PI) controllers are frequently employed in industrial process applications, which are commonly associated with noise characteristics [9].

The PI controller is a classical technique for control [9], which is commonly applied to track reference signals. Because of the noise irregularities in the compression process, an anti-windup technique is usually looped into the PI controllers [20], [21]. The anti-windup prevents the impact of chattering and enhances smooth PI control operation [20], [22]. Based on these facts, PI controllers with anti-windup are employed [10], [23] for surge control in compressors.

2.3. State observers

State observers are feedback computing devices driven by process inputs and outputs to approximate state vectors of a system and whose outputs can be used to execute feedback control laws [13]. A reduced-order observer estimates fewer state variables of a system and assumes that certain state variables are already available as measured outputs. Therefore, its inherent transformation delay in computation causes measurement delay to feedback for the surge controller [16]. All state variables are directly measured at system output in a full order observer, and no transformation is required. Therefore, it has the advantage of compensating for measurement delay to correct the system control deficiencies [16]. In active surge control, a full order observer provides reasonable parameter estimates for the system by including a speed model into observer computation for variable drive control through the surge controller's action [16]. The suggested model for observer design is the general form of an observer as given in (4) and (5) [24]:

$$\dot{x} = Ax + G\gamma(Hx) + \ell(u,y) \tag{4}$$

$$y = [y_1 \quad y_2]^T = [Cx \quad h(u,x)]^T \tag{5}$$

where $x \in \mathcal{R}^{n}$ is the state variable vector, $u \in \mathcal{R}^{m}$ is the input vector, $y_1 \in \mathcal{R}^{r_1}$ and $y_2 \in \mathcal{R}^{r_2}$ are the measured and estimated variables, respectively. This formation is such that the linear and nonlinear uncertainties of the state enter the dynamics of the compressor via a linear and nonlinear mapping of Ax and $\gamma(Hx)$ respectively.

2.4. Genetic algorithm optimization

The GA can substantially run in a reasonable number of iterations and terminates with a desired chosen ending condition, for example, a fitness converged, when individuals meet target fitness, a maximum number of generation limits and the maximum number of stall generations [25]. The GA optimization methodology in this work is derived based on Holland's notion of schemata, which states that low order and above-average schemata receive increasing trials in the subsequent generation of GA as shown in (6) [15]:

$$\varsigma(m,t+1) \geq \varsigma(m,t)\frac{f(m,t)}{F(t)}\left[1 - P_c\frac{\delta(m)}{L-1} - O(m)P_m\right] \tag{6}$$

where m is the mass flow encoded in sets of strings that represents the schema, $\zeta(m,t)$ is the number of strings in the t_{th} generation matched by schema, $\delta(m)$ defines the length of the schema, $O(m)$ is the schema order that represents the number of fixed positions, $f(m,t)$ is the schema fitness value. L is the chromosome length and $F(t)$is the average number of occurrences of m. P_m and P_c are the rate of operation of mutation and crossover, respectively.

Therefore, for the compressor characteristic optimization model's objective function $\Psi_c(m)$, a given range of mass flow, $m_{min} \leq m \leq m_{max}$, can be maximized to improve compressor efficiency. Hence, the genetic creatures of individuals are programmed to undergo a process of simulation that reflects a degree of goodness similar to [15], fitted with a potential search solution to maintain the compressor characteristic at an optimum value of pressure ratio. Detailed information on the compressor characteristics that determine the compressor's operating efficiency is presented in [26].

3. STATE OBSERVER MODEL FORMULATION

The state observer model is formulated in this section according to [17]. The fluid flow principle is used to design observer error estimation for feedback, it ensures that the mass flow is unidirectional and irreversible. In formulating the state observer model, a closed coupled valve (CCV) is considered in order to control the equivalent compressor characteristic as written in (7):

$$\Psi_e(m,\omega_1) = \Psi_c(m,\omega_1) - \Psi_v(m,\omega_1) \tag{7}$$

where, Ψ_e is the equivalent compressor characteristic, Ψ_c is the compressor characteristic, Ψ_v is the CCV characteristic, m is the mass flow through the channel and ω_1 is the rotor speed.
The mass flow of the compressor is modelled as a function of the outlet regulator valve as given in (8):

$$m_{out} = k_r\sqrt{p - p_{01}} \tag{8}$$

where, k_r is the regulator gain, p is the tank pressure, p_{01} is the inlet pressure.
The load torque developed by the compressor can be modelled as:

$$\tau_l = |m|\frac{D_2}{2}\mu\omega_2 \tag{9}$$

where, τ_l is the compressor load torque, D_2 is the tip impeller diameter, μ is the slip factor and ω_2 is the tip impeller velocity. When we consider (8) and (9), the state observer model of the compressor system can be formulated from the compressor system model in (1) to (3) as given in (10) to (12):

$$\dot{p} = \frac{a_0^2}{V_p}(m - k_r\sqrt{p - p_{01}}) \tag{10}$$

$$\dot{m} = \frac{A_1}{L_c}\left((\Psi_c p_{01}) - p\right) \tag{11}$$

$$\dot{\omega} = \frac{D_1}{2l}(\tau_d - \tau_l) \tag{12}$$

where p is the tank pressure, m is the mass flow through the channel, ω is the rotor speed, a_0 is the ultrasonic velocity, V_p is the tank volume, A_1 is the flow channel area, Lc is the channel length, Ψ_c is the compressor characteristic, p_{01} is the inlet pressure, τ_d is the drive torque, τ_l is the load torque and D_1 is the inducer diameter, I is the spool moment of inertia. Some parameter variables' errors of the compressor are defined as in [16], which implies that (10) to (12) can be written as:

$$\dot{\hat{p}} = \frac{a_0^2}{V_p}\left(\hat{m} - \hat{m}_{out}\left(\hat{p}\sqrt{|p - p_{01}|}\right)\right) \tag{13}$$

$$\dot{\hat{m}} = \frac{A_1}{L_c}\left(\hat{\Psi}_c(\hat{m},\hat{\omega}) - \hat{\Psi}_v(\hat{m},\hat{\omega})p_{01} - \hat{p}\right) \tag{14}$$

$$\dot{\hat{\omega}} = \frac{D_1}{2l}(\hat{\tau}_d - \hat{\tau}_l) \tag{15}$$

where $\hat{\tau}_d$ and $\hat{\tau}_l$ are the torques modified, let $\hat{\tau}_d = \tau_d - \tau_{l0}$, $\hat{\tau}_l = \tau_l - \tau_{l0}$ and τ_{l0} is for the equilibrium. In (13) to (15) represent the formulated state observer model for the compressor system.

3.1. Compressor efficiency optimization problem formulation

To determine the compressor efficiency, the compressor characteristic is derived as a function of mass flow and speed, which represents a pressure ratio in isentropic compression. Improving compressor operating efficiency is achieved by employing GA to optimize a formulated compressor characteristics optimization model. Considering the surge line mass flow $m_{sl}(\omega)$ as the decision variable, with a given range of maximum mass flow $m_{\max}$ and minimum mass flow m_{min}, the optimization problem of the compressor efficiency is formulated as:

$$\Psi_C(m) = max\sum_{i=1}^{N}\left[1 + \frac{\left((\mu r_2^2\omega^2) - \frac{r_1^2(\omega-\alpha m)}{2} - k_f m^2\right)}{c_p T_{01}}\right]^{\frac{k}{(k-1)}} \tag{16}$$

where, N is the total number of optimization samples, μ is the slip factor, r_1 is the mean inducer radius, r_2 is the impeller radius, ω is the rotor speed, α is the incidence loss, k_f is the fluid friction constant, k is the ratio of specific heats, c_p is the specific heat at constant pressure and T_{01} is the inlet temperature. The optimum solution to (16) is determined subject to the constraint given as:

$$m_{min} \leq m \leq m_{max} \tag{17}$$

Since the problem in (16) is a single parameter optimization problem, a suitable heuristic algorithm like GA is employed to efficiently optimise the compressor's efficiency. The GA and the PI controller parameters are given in Table 1. The values of the parameters given in Table 1 were determined after a thorough experimental analysis. Even though there are no standards for selecting the population and other GA parameters, we employed the parameters that give us the optimum solution over 50 independent runs.

Table 1. Compressor controller design parameters

S/N	Description	Parameters	Values
1	Proportional gain	K_p	120
2	Integral gain	K_i	5
3	Integral time	T_i	24s
4	Anti-windup gain	K_b	42×10^{-3}
5	Number of chromosome	*nPop*	50
6	Dimension	nVar	1
7	Crossover	Cr	0.7
8	Mutation	Mu	0.2

4. RESULTS AND ANALYSIS

This section presents and discusses results obtained from all simulations for performance evaluation. The simulations were carried out using MATLAB, and the figures demonstrated simulation results. The resulting analysis was carried out by comparing the performance of the developed GA based full order observer and the traditional piston actuation recycle approach.

4.1. Compressor parameter estimates for ga based full order observer model

The plots of parameter estimate for genetic algorithm based full order observer outputs are presented. The estimated parameters include suction pressure, outlet pressure, mass flow, pressure ratio, feedback error and speed, as shown in Figure 2 to Figure 7. In line with previous works, these parameters were used as metrics to measure the performance of the developed method.

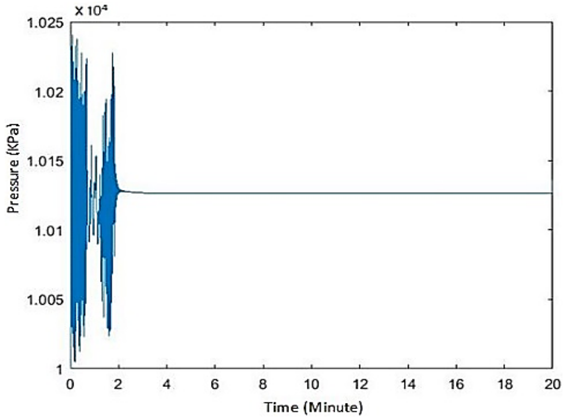

Figure 2. Plot of suction pressure

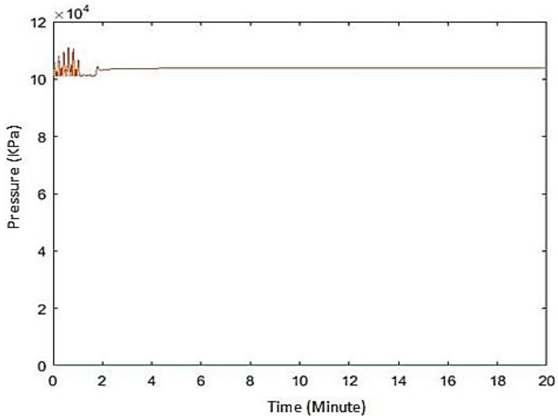

Figure 3. Plot of outlet pressure

Figure 2 shows the suction pressure plot through the compressor due to centrifugal propulsion. In the beginning of the plot, it can be noted that the suction pressure varies between a minimum of about 10,000 KPa and maximum of 10,225 KPa, which stabilizes at 10,125 KPa. The pressure variations are due to centrifugal force dynamic characteristics of rotating impellers acting on the suction pressure during compressor start-up. Eventually, the impact is extended to other system parameters. In practice, the parameters oscillation due to the start-up phenomenon, by implication, neither have a safety effect on compressor performance nor the control system. Figure 3 shows the estimates of outlet pressure of the compressor, which stabilizes at 10.25×10^4 KPa. Figure 4 presents the plot of estimates of the mass flow through the compressor channel, which stabilizes at 0.31 Kg/s. Figure 5 gives a plot of estimates of the pressure ratio, which stabilizes at 0.9247 (p.u). This pressure ratio is expressed in terms of compressor efficiency, given as 92.47%.

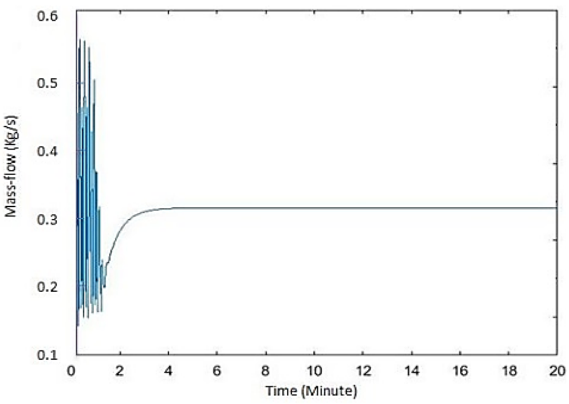

Figure 4. Plot of mass-flow

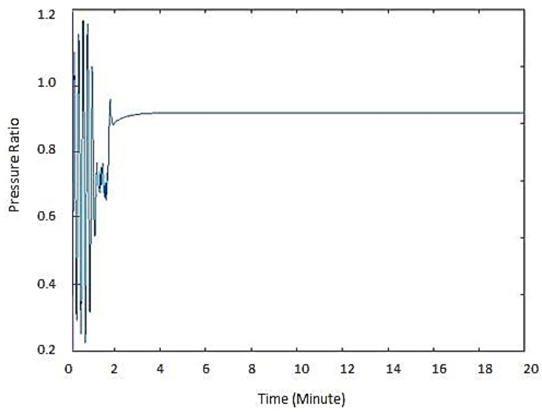

Figure 5. Plot of pressure ratio

Figure 6 is the estimated feedback error, which tends to zero as time goes to infinity $(e \to 0, t \to \infty)$, and stabilizes at an absolute value of $|-20|$ on the error scale. This stability shows a satisfactory controlled, steady-state error computed by a full-order observer for feedback to the surge controller. Figure 7 shows the estimates of the controlled speed of the compressor, which stabilizes at 2000 rpm.

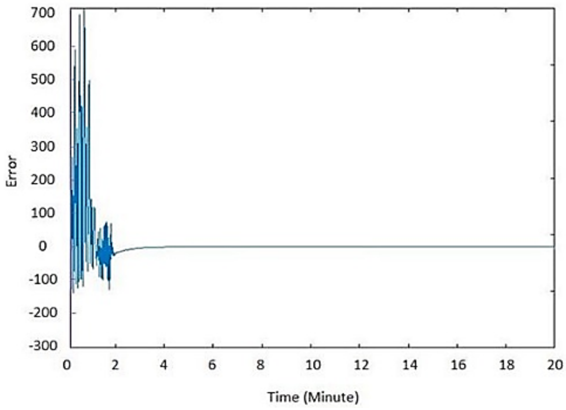

Figure 6. Plot of feedback error

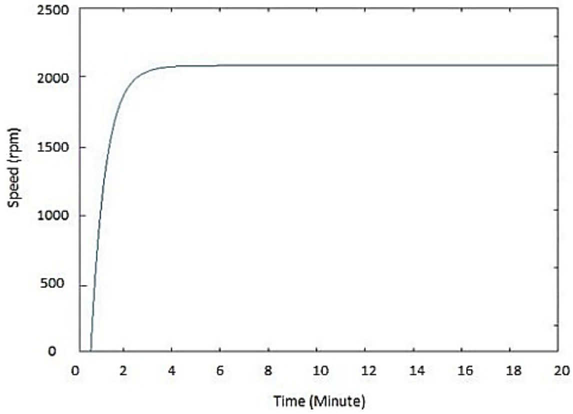

Figure 7. Plot of speed

4.2. Compressor Parameter Estimates for Piston Actuation Recycle Method

The plots of parameter estimate for the piston actuation recycle method with only pressure feedback are presented. The measured parameters include suction pressure, outlet pressure, mass flow, pressure ratio, feedback error and speed, as shown in Figures 8 to 13. Figure 8 presents the plot of suction pressure through the compressor. At the beginning of the plot, the suction pressure varies between a minimum of about 1.012×10^4 KPa and a maximum of 1.175×10^4 KPa. However, the suction pressure finally stabilizes at 1.0650×10^4 KPa.

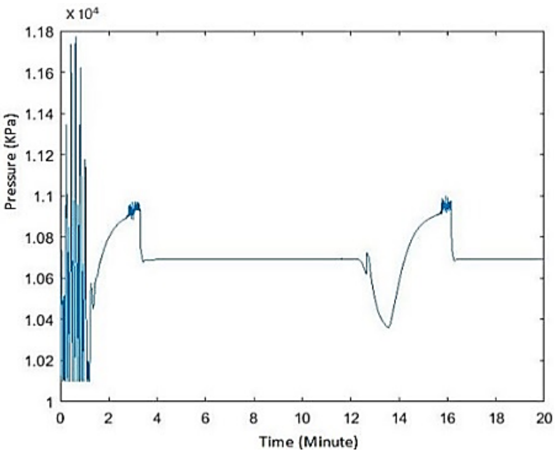

Figure 8. Plot of suction pressure

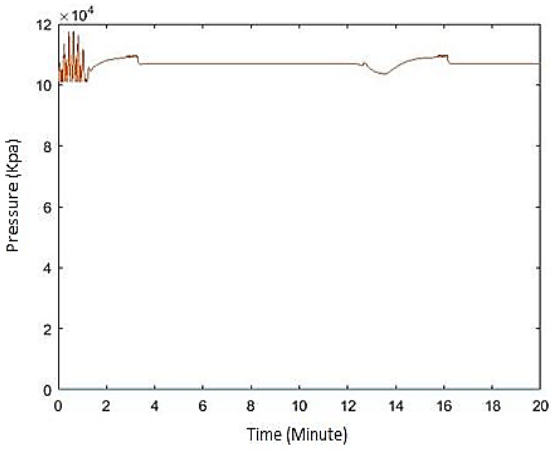

Figure 9. Plot of outlet pressure

The initial oscillations are due to centrifugal force dynamic characteristics of the rotating impellers acting on the suction pressure as the compressor spins during start-up. Incidentally, the impact is extended to other system parameters. These parameter oscillations are due to start-up operation. By implication, it neither has a safety effect on the compressor performance nor the control system. Figure 9 shows the outlet pressure of the compressor, which stabilizes at 10.25×10^4 KPa.

Figure 10 presents the mass flow through the channel of the compressor. From the figure, it can be observed that the mass flow stabilizes at 0.26 Kg/s at about 4 minutes. The compressor maintains this stable flow for about 12 minutes before a surge is experienced at around 16 minutes, then returns to the same stable mass flow afterwards until the end of the experiment. Figure 11 shows the plot of pressure ratio, which stabilizes at 0.8100 per unit (p.u), expressed in terms of compressor efficiency given as 81.00%. Figure 12 shows the plot of feedback error, which deviates from zero with an increase in magnitude in the positive direction to stabilize at an absolute value of $|60|$ on the error scale, however, it is affected by heavy swings of oscillation due to the piston actuation recycling control effect. Figure 13 is the plot of the speed setting of the compressor, which is controlled at 2000 rpm.

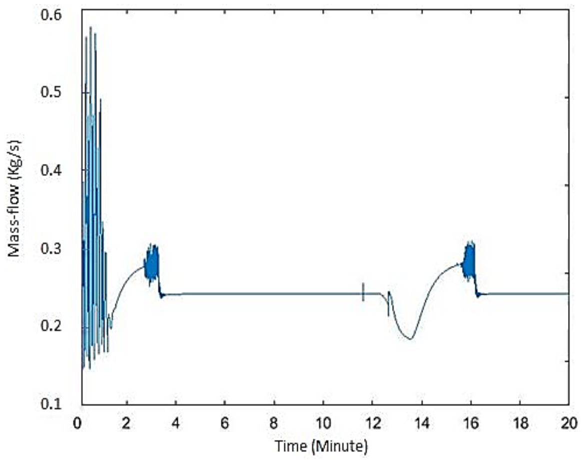

Figure 10. Plot of mass-flow

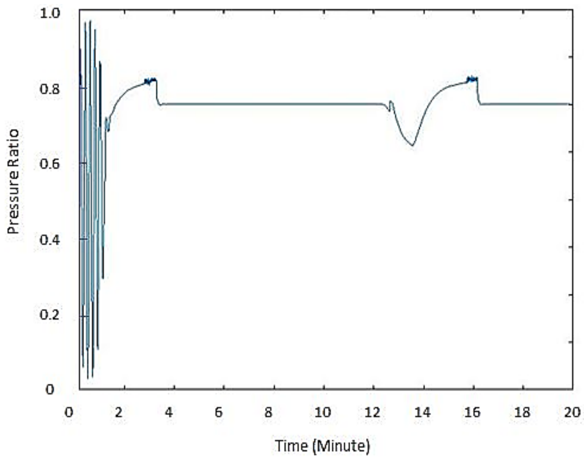

Figure 11. Plot of pressure ratio

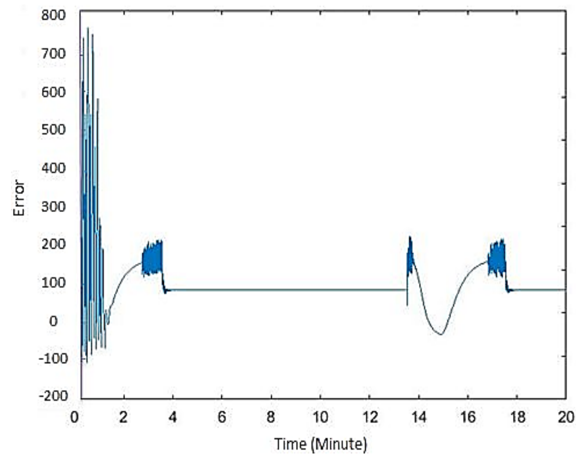

Figure 12. The plot of feedback error

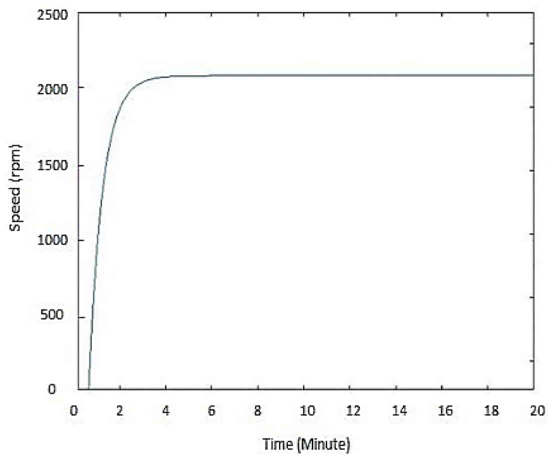

Figure 13. The plot of speed

4.3. Comparison of GA based full order observer with piston actuation recycle methods

This section compares the estimated parameters between the developed GA based full order observer with the piston actuation recycle methods. The comparision was done using pressure ratio and mass flow as performance metrics. For performance evaluation of the two systems, plots of two pressure ratios estimates and plots of two mass-flows estimates in Figure 14 and Figure 15, respectively, are shown concurrently for comparison.

Figure 14 and Figure 15 compare the pressure ratios and mass flow of the developed GA-based full order observer and piston actuation recycle methods, respectively. It can be observed that the GA based method maintain a stable pressure ratio at 0.9247, which is expressed as 92.47% in terms of compressor efficiency. The plot shows a fair approximation of feedback error by GA based full order observer with satisfactory speed regulation by an active surge controller. At the same time, the pressure ratio of the piston actuation recycling method stabilizes at 0.8100, which is expressed as 81.00%. Although surge was avoided, the control action led to an immediate drop and rise of the compressor efficiency in a few time intervals.

Figure 15 is the plot comparison of mass flows between the developed GA based full order observer and piston actuation recycle methods. It can be seen that the mass flow for the developed system has improved to stabilize at 0.31 Kg/s over the mass flow of the piston actuation recycle method, which stabilizes at 0.26 Kg/s. However, the latter’s plot is further disturbed by the recycling pressure effect, causing oscillations in the system after a few time intervals. Given the results obtained from simulation plots, the evaluation revealed that the developed GA based full order observer outperformed the piston actuation recycle method in terms of improved compressor efficiency, mass flow and reduction of the absolute value of feedback error, as elaborated in Table 2. It can be deduced, from Table 2, that the developed GA based full order observer outperformed the piston actuation recycle method by: 14.16% in terms of compressor efficiency, 19.23% in terms of mass flow and 66.66% in terms of reduced feedback error (absolute values) to surge controller.

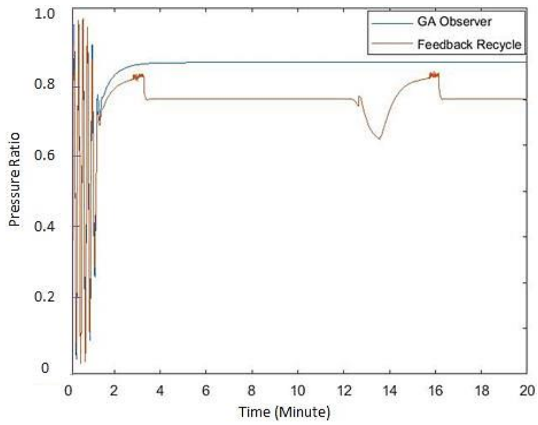

Figure 14. Plots of pressure ratios

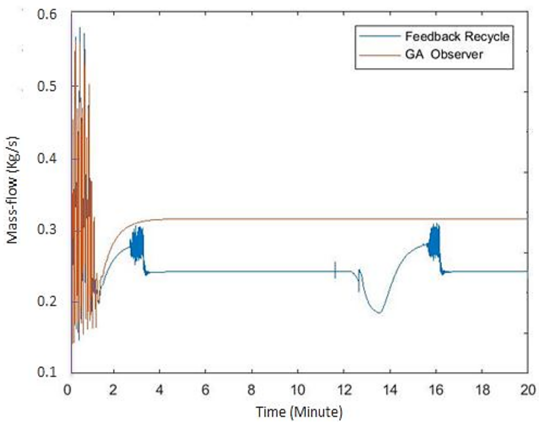

Figure 15. Plots of mass-flows

Table 2. Comparison of estimated parameters from simulation plots

SN	Parameters	Piston actuation recycle method	Full order observer system	Minimum limit	Maximum limit
1	Pressure ratio (P.U)	0.8100	0.9247	0.4000	1.0000
2	Efficiency (%)	81.00	92.47	40	100
3	Mass flow (Kg/s)	0.26	0.31	0.14	0.35
4	Feedback error	$\|60\|$	$\|-20\|$	___	0.000
5	Suction pressure (KPa)	1.0650×10^4	1.0125×10^4	___	___
6	Outlet pressure (KPa)	10.25×10^4	10.25×10^4	___	___
7	Speed (rpm)	2000	2000	800	2000

5. CONCLUSION

The optimal design of a GA based full order state observer for active surge control has been satisfactorily realized. The state observer model was formulated and applied in the full order state observer design for active surge control in a centrifugal compressor system. Principles of fluid flow friction and mechanical effects on measured parameters through the compressor were employed in the design for observer error estimation. The strategy actualizes unidirectional mass flow of fluid, which prevents backflow corresponding to surge. Due to the requirement for efficient and improved operating condition of the compressor system, compressor characteristic was formulated into optimization problem to maximize the mass flow using GA optimization technique, which guaranteed optimized compressor efficiency. Simulations of parameter estimates by the developed GA based full order observer were carried out using MATLAB.

The results obtained were promising and, when compared with the piston actuation recycle method, yielded better performance in terms of increased compressor efficiency, mass flow, and a reduced absolute value of feedback error. We will employ an extended state observer (ESO) to establish control of the start-up parameter estimates' oscillations in our future work. Other researchers can also implement the developed GA based full order state observer with MPC for active surge predictive control in compressors.

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