A Novel Approach for Active Surge Control in Multistage Centrifugal Compressor

YACINE ARIBI, RAZIKA ZAMMOUM BOUSHAKI
Signals and Systems Laboratory, Institute of Electrical and Electronics Engineering,
Université M'hamed Bougara Boumerdes,
Avenue de l'indépendance, Boumerdes,
ALGERIA

Abstract— Compressors are used in the hydrocarbon industry from upstream to downstream to boost the pressure required to transport gas into processing and transportation pipelines inside and toward processing facilities. Turbo-centrifugal compressors are subject to repetitive damages resulting from the manifestation phenomenon known as surge, observed as fluid flow reversal at low mass flows. Active surge control aims to increase the compressor operating region by allowing the compressor to operate at low mass flow in the naturally-unstable region of the compressor. Surge is avoided in the industry by forcing the operating point to the stable region of the compressor curve. Active surge control aims to control the surge phenomenon by extending the stable operating region. This work developed a novel model-based control approach for effective compression system active surge control on an automatic coupled recycle valve as a primary actuator. The feedback linearization technique is used to reduce the system complexity by providing a linear representation of the highly nonlinear system. A detailed description of the feedback linearization method for centrifugal compressors and a dual controller design was developed in this paper.

Keywords: - Centrifugal Compressor, Active Surge Control, Feedback Linearization, Nonlinear Systems, I/O Linearization, PID Controller.

Received: November 24, 2021. Revised: October 22, 2022. Accepted: December 5, 2022. Published: January 26, 2023.

1 Introduction

In Centrifugal compressors, the potential rise of the mass flow through the compressor to the nominal values often results in a decrease of inlet fluid density and hydrostatic pressure at the suction plenum, thus, resulting in an even larger intake rate from the gas storage vessels towards the centrifugal compressor, [1]. Centrifugal compressors are subject to a high oscillatory phenomenon that emerges as an intense vibration and gas flow reversal, this phenomenon is known as surge, which is a dangerous unstable operating mode that triggers inversion of the rotor's axial push. Surge prevention has been treated in the literature by anti-pumping regulation systems to preserve the system from reaching the surge line, in similar control systems, the section flow is always larger than the minimum flow that triggers the surge at any compression rise. The surge prevention control is based on a feed-backward loop that connects the discharge and suction manifolds and is controlled through a recycle valve that regulates the flow fed-back, nevertheless, this method forces the system to work at high feed flow which limits the gas production and hence the efficiency of the compressor. Since the compression system model is nonlinear, the design of an active surge controller is not a straightforward process because of the high nonlinearities present in the system and the undamped oscillatory behavior of the system.

Feedback linearization is a non-linear control design method that has attracted lots of research in recent years, [2], [3], [4], as it involves transforming the nonlinear system into a controllable linear system by using state feedback and coordinate transformations. The input-output decoupling method is used to transform the nonlinear system into a new linear-equivalent form where each output is independently and uniquely controlled by only one of the newly defined inputs.

In this work, a combined approach aims to use a speed regulator and an overall feedback linearization on a centrifugal compression system as a novel approach to tackle the active surge control problem in linear form. A linear PID controller, which is the gas production industry's first choice for the recycle actuators control, is used to control the linearized system. Furthermore, traditional PID design techniques are combined with pole placement techniques to ensure the overall system stability for a linear controller meant to control a complex and dangerous industrial process.

2 Centrifugal Compressor Model

The mathematical model adapted in this paper and used as a backbone to create the simulation model is based on the model provided by [5] and [6] and stated in equation (1), this model is widely accepted as a solid description for centrifugal and axial compressors and has found wide acceptance in literature, [5], [6].

$$\begin{cases} \dot{p}_1 = \frac{a_{01}^2}{V_1} \times \left(Q_f - Q + Q_r\right) \\ \dot{p}_2 = \frac{a_{01}^2}{V_2} \times \left(Q - Q_r - Q_t\right) \\ \dot{Q} = \frac{A}{L} \times \left(\Psi(Q,\omega) \times p_1 - p_2\right) \\ \dot{\omega} = \frac{1}{J} \times \left(\tau_d - \tau_c\right) \end{cases} \quad (1)$$

The model is realizable as per the diagram in Fig. 1.

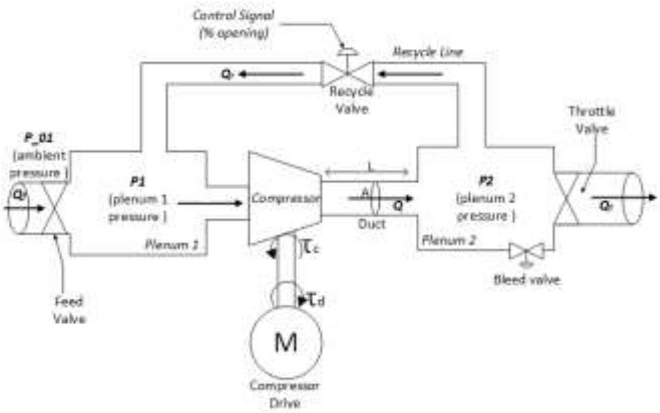

Fig. 1: Drawing for compressor with recycle line

The state space model of equation (1), with four (4) states is used to simulate the disturbance effect on the system, to define the surge limit, and to test the control system efficiency.

The compressor parameters, set points, and ambient conditions used for simulation purposes are defined in Table 1 in section 7.

To check the stability of the system in equation (1) around the system's equilibrium point, direct stability analysis shows that the uncontrolled system is stable if and only if the system is operating at $\frac{d\Psi}{dQ} \leq \varepsilon$, thus the operating point is at the negative slope, or at most, small positive slope of the $\Psi - Q$ characteristic, this conclusion is derived by analyzing at a single constant Speedline ω_0 and by considering the non-controlled compressor case, thus, setting the recycle flow Q_r to zero, then linearizing equations (1) around the equilibrium point and substituting the estimated values into the original state-space equation.

3 Feedback Linearization

Feedback linearization is a widely used control technique for nonlinear systems. This model is based on developing a transformation envelope for the nonlinear system in such a way the residual system is a linear equivalence for the original nonlinear system, this transformation is obtained via a change of variables (inputs and/or outputs) and a set of controls, both methods work to cancel the nonlinearities in the system resulting in a fully or partially linear closed-loop dynamics so that the well-known linear techniques that we have developed in linear control systems and multivariable control system courses can be applied.

Feedback linearization has two major types: input-state linearization and input-output linearization depending on whether full or partial linearization of the system is required.

Input-state linearization is focused on developing transformation for the state variables and a linearizing input that linearizes the whole system. On the other hand,

input-output linearization is a method focused on establishing the relationship between the nonlinear system inputs and outputs by defining the I/O map and then linearizing this map. If the class of nonlinear systems in equation (2) is considered a model:

$$\begin{cases} \dot{x} = f(x) + g(x)u \\ y = h(x) \end{cases} \quad (2)$$

Where: $f: D \to \mathbb{R}^n$, $g: D \to \mathbb{R}^{n\times q}$ all defined on the domain $D \subset \mathbb{R}^n$. n is the system's degree, and q is the number of outputs.

The transformation's ultimate objective is to define the control signal: $u = \alpha(x) + \beta(x)v$

Along with variables transformation $z = T(x)$ to fully or partially transform the nonlinear system, [7].

3.1 Input-Output linearization

To execute an input-output linearization that sets a relation between the output function y and the input u (the control signal), a set of repetitive differentiations is carried until the relationship is visible, then the input v, is designed to cancel the system nonlinearities, [8].

$$\frac{y(s)}{v(s)} = \frac{1}{s^r} \tag{3}$$

Nevertheless, if the system's relative degree is undefined, the controlling input v cannot be designed to cancel the system's nonlinearities.

3.2 Well-defined Relative Degree

A single input single output (SISO) system of degree n similar to the form in equation (2) is said to have relative degree r in a region Ω if, $\forall\, x \in \Omega$, [9]. where: $r \leq n$.

$$\begin{aligned} &L_g {L_f}^i h(x) = 0 \qquad 0 \leq i < r-1 \\ &L_g {L_f}^{r-1} h(x) \neq 0 \end{aligned} \tag{4}$$

Then the linearizing input can be defined as:

$$\dot{y} = \nabla h(f + g.u) = L_f h(x) + L_g h(x).u$$
$$\ddot{y} = {L_f}^2 h(x) + L_g L_f h(x).u$$
$$y^{(i)} = {L_f}^i h(x) + L_g {L_f}^{i-1} h(x).u$$

If the system has relative degree r, then we can write:

$$u = \frac{1}{L_g {L_f}^{r-1} h}(-{L_f}^r h + v) \tag{5}$$

Since: $L_g {L_f}^{r-1} h(x) \neq 0$

By replacing u from equation (5) into $y^{(r)}$:

$$y^{(r)} = v \tag{6}$$

A nonlinear system is said to have an undefined relative degree if at the operating point $L_g {L_f}^{r-1} h = 0$ but nonzero at some arbitrary point near the operating point.

3.3 Zero Dynamics

I/O feedback linearized system is composed of an external segment (input-output) and an internal segment, where the internal segment is unobservable. The observable external part of the system can be easily used to design a control signal to control the input v via negative feedback to obtain a desired output y with behavior that matches our expectations. Since the new input v does not affect the internal dynamics, their behavior must be taken into consideration. It is important to note that to design an I/O controller, the internal dynamics of the system have to be stable, [10].

By applying the notion of normal forms, when the relative degree r is defined and $r < n$, the nonlinear system (equation (2)) can be transformed into the normal form using the new states: $y, \dot{y}, \ldots, y^{(r-1)}$.

In this case, and if we define:

$\mu = [\mu_1 \mu_2 \ \ \ldots \ \ \ \mu_r]^T = [y \ \ \ \dot{y} \ \ \ldots \ \ \ y^{(r-1)}]^T$

Then in a neighborhood of the operating point Ω, the normal form of the system is:

$$\dot{\mu} = \begin{bmatrix} \mu_2 \\ \ldots \\ \ldots \\ a(\mu, \Psi) + b(\mu, \Psi)u \end{bmatrix} \tag{7}$$

With the output defined as $y = \mu_1$.

The last $(n-r)$ equations $\dot{\Psi} = w(\mu, \Psi)$ of the normal form defines the internal dynamics for the I/O linearized form.

The zero dynamics are identified by placing the restraint that the output y and all its derivatives in the multidimensional space of dimension $= (n - r)$, are all equal to zero, i.e., $y = 0$, and $y^{(i)} = 0$, for i=0,…,(n-r).

The zero dynamics describes the system's internal dynamics in a restricted $(n-r)$ -dimensional smooth surface defined by $\mu = 0$.

Calculating the zero dynamics allows us to build a conclusion on the internal dynamics' stability.

From equation (5) the input must satisfy:

$$u = \frac{-{L_f}^r h}{L_g {L_f}^{r-1} h} \tag{8}$$

By assuming that the system's initial state is located in the surface defined by $\mu = 0$, i.e., that $\mu(0) = 0$, the normal form of the system dynamics is:

$$\begin{cases} \dot{\mu} = 0 \\ \dot{\Psi} = w(\mu, \Psi) \end{cases} \tag{9}$$

The part corresponding to the internal dynamics is:
$a(\mu, \Psi) = L_f{}^r h(x)$, $b(\mu, \Psi) = L_g L_f{}^{r-1} h(x)$ and $L_g \Psi_j = 0 \quad 1 \leq j \leq n - r$

By assuming that the system in equation (1) has well defined relative degree, and the zero dynamics are asymptotically stable around the operating point, if we design a control input v that places the roots of the new linearized system inside the stable region, then the control law in equation (5) will produce an asymptotically stable closed-loop system around the operating point, [10].
For the system to obtain local asymptotic stability in the vicinity of the operating point, it is enough to ensure that the system has stable zero dynamics.

4 Feedback Linearization of the Centrifugal Compressor

4.1 Compressor Flow Design

The system has one input which is the percentage of opening the recycle valve and one main variable which is the compressor flow, hence we could rewrite our system by defining the recycle valve as an additional state to the system, based on the mass balance. In this case, the recycle flow Q_r is considerably taken into the calculations of mass and energy balances at all plenums, this will result in the complete design of equation (1) for the controlled system.

By setting the following constants:

$$k_1 = \frac{{a_{01}}^2}{V_1}, k_2 = \frac{{a_{01}}^2}{V_2}, k_3 = \frac{A}{L}, K_4 = \frac{1}{J}$$

The feed, throttle, and recycle flows are defined below

$$Q_f = k_{12} \times \sqrt{|p_{01} - p_1|}$$
$$Q_t = k_{13} \times \sqrt{|p_2 - p_{01}|}$$
$$Q_r = u \times A_t \times \sqrt{|p_2 - p_1|}$$

Where k_{12} and k_{13} are positive integers that resemble the valve opening and u is the recycle valve opening control proportional to the valve opening. The system in equation (9) can be written as:

$$\begin{cases} \dot{p_1} = k_1 \times (k_{12} \times \sqrt{|p_{01} - p_1|} - Q + u \times A_t \times \sqrt{|p_2 - p_1|}) \\ \dot{p_2} = k_2 \times (Q - u \times A_t \times \sqrt{|p_2 - p_1|} - k_{13} \times \sqrt{|p_2 - p_{01}|}) \\ \dot{Q} = k_3 \times (\Psi(Q, \omega) \times p_1 - p_2) \\ \dot{\omega} = k4 \times (\tau_d - k14 \times |Q| \times \omega) \end{cases} \tag{10}$$

Using a speed regulation module, the speed can be considered constant while operating the compressor on a constant Speedline, which allows the assumption that the speed is constant. By setting: $x_1 = p_1, x_2 = p_2, x_3 = Q$ and by substituting in equation (10) and projecting to equation (1), we get:

$$f(x) = \begin{bmatrix} k_1 \times (k_{12} \times \sqrt{|p_{01} - x_1|} - \mathrm{x}_3) \\ k_2 \times (x_3 - k_{13} \times \sqrt{|x_2 - p_{01}|}) \\ k_3 \times (\Psi(x_3) \times x_1 - x_2) \end{bmatrix}$$
$$g(x) = \begin{bmatrix} k_1 \times A_t \times \sqrt{x_2 - x_1} \\ -k_2 \times A_t \times \sqrt{x_2 - x_1} \\ 0 \end{bmatrix} \tag{11}$$
$$y = x_3$$

by calculating the respective lie derivatives, we get:

$$\mathcal{L}_g \mathcal{L}_f^0 y(x) = 0$$
$$\mathcal{L}_g \mathcal{L}_f^1 y(x) \neq 0$$

Thus, the relative degree of the system is $r = 2 < n$ and the system is **Input-Output Linearizable**.
The new control signal is then defined based on equation (6):

$$v = y^{(r)} = \ddot{x_3} \tag{12}$$

Which gives:

$$v = \ddot{x3}$$

Thus,

$$\begin{aligned} v = k_3 \times (\dot{\Psi}(x_3).x_1 & \\ & + \Psi(x_3).k_1.\Big(k_{12}.\sqrt{|p_{01} - x_1|} \\ & + u.A_t.\sqrt{|x_2 - x_1|} - \mathrm{x}_3\Big) \\ & - k_2.\Big(x_3 - k_{13}.\sqrt{|x_2 - p_{01}|} \\ & - u.A_t.\sqrt{|x_2 - x_1|}\Big) \end{aligned}$$

The linearizing input is then defined as:

$$u = \frac{-\dot{\Psi}(x_3).x_1 - \Psi(x_3)K_1\big(k_{12}\sqrt{|p_{01} - x_1|} - x_3\big) + k_2\big(x_3 - k_{13}\sqrt{|x_2 - p_{01}|}\big) + \frac{1}{k_3}v}{\sqrt{|x2 - x1|}A_t(k1 * \Psi(x3) + k2)} \tag{13}$$

Once the surge phenomenon is triggered, the mass flow experiences an increasing oscillatory behavior, depicting an instability in the mass flow and hence all other system states. The whole system can be stabilized by stabilizing the surge-triggering state, which is the mass flow.

By setting the new state variables and control signal as:

$$\boldsymbol{z} = \begin{bmatrix} z_1 \\ z_2 \\ z_3 \end{bmatrix} = \begin{bmatrix} x_3 \\ \dot{x_3} \\ \ddot{x_3} \end{bmatrix} \quad (14)$$
$$v = \dddot{x}_3$$

The linearized system is written in Brunovsky form, [11].

$$\begin{cases} \dot{z} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix} z + \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} v \\ y = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} z \end{cases} \quad (15)$$

This form is written for simplicity as

$$\begin{cases} \dot{z} = \boldsymbol{A}z + \boldsymbol{B}v \\ y = \boldsymbol{C}z \end{cases} \quad (16)$$

The output deviation signal is defined as the deviation of the actual output from the desired output which is ideally equal to zero for perfect control. The error is defined in the frequency domain for the linearized system as:

$$\boldsymbol{E}(s) = \boldsymbol{R}(s) - \boldsymbol{Y}(s)$$
$$\boldsymbol{E}(s) = \boldsymbol{R}(s)(1 - \boldsymbol{C}(s\boldsymbol{I} - \boldsymbol{A})^{-1}.\boldsymbol{B})$$

Using the final value theorem and by assuming $e(t) = 0 \; for \; t < 0$, we can write the steady state error as:

$$\lim_{s\to 0} s\boldsymbol{E}(s) = \lim_{s\to 0} s\boldsymbol{R}(s)(1 - \boldsymbol{C}(s\boldsymbol{I} - \boldsymbol{A})^{-1})\boldsymbol{B}$$

Applying the linearized compression system, we get:

$$\boldsymbol{E}(s) = -\frac{1}{s}.(\frac{1}{s^3} - 1)$$

Then the steady-state error is: Error = $\lim_{s\to 0}(s\boldsymbol{E}(s))= \infty$

Since the system is unstable, an increasing oscillation with time is incepted for the non-controlled system which leads to a theoretical infinity blow-up of gases.

5 Pole Placement and Steady-State Error Design

Taking the linearized form of equation (15), the feedback-linearized system with the external linear control is visualized in Fig. 2.

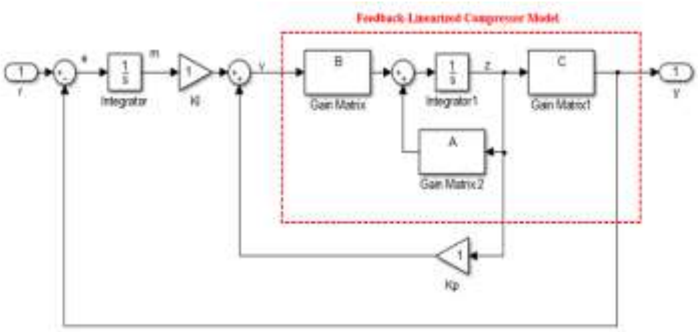

Fig. 2: Steady-state error control with PI.

The Problem has been reduced to a linear SISO system with three states. The control problem can be tackled by either pole placement, PID controller, adaptive PID controller, or simple Type I Fuzzy controller, However, hybrid PID controllers and pole placement methods can be more efficient while designing the system for a preset transient and steady-state performance. The system in Fig.2, with a PI controller, extends the state space description to:

$$\begin{cases} \dot{z} = \boldsymbol{A}z + \boldsymbol{B}v \\ \dot{m} = -\boldsymbol{C}z + r \\ y = \boldsymbol{C}z \end{cases} \quad (17)$$

The control signal from the modified PI controller will be set then to:

$$v = k_p * z(t) + k_I * \int_{t_0}^{t} e(t)dt \quad (18)$$

Substituting in the original system:

$$\begin{bmatrix} \dot{z} \\ \dot{m} \end{bmatrix} = \begin{bmatrix} \boldsymbol{A} & 0 \\ -\boldsymbol{C} & 0 \end{bmatrix} \begin{bmatrix} z \\ m \end{bmatrix} + \begin{bmatrix} \boldsymbol{B} \\ 0 \end{bmatrix} v + \begin{bmatrix} \boldsymbol{0} \\ 1 \end{bmatrix} r$$
$$y = [\boldsymbol{C} \quad 0] \begin{bmatrix} z \\ m \end{bmatrix} \quad (19)$$
$$v = \boldsymbol{K}_p z + K_I m = [\boldsymbol{K_p} \quad K_I] \begin{bmatrix} z \\ m \end{bmatrix}$$

Hence the following model:

$$\begin{bmatrix}\dot{z}\\ \dot{m}\end{bmatrix}=\begin{bmatrix}\boldsymbol{A}+\boldsymbol{BK_p} & \boldsymbol{B}K_I\\ -\boldsymbol{C} & 0\end{bmatrix}\begin{bmatrix}z\\ m\end{bmatrix}+\begin{bmatrix}\boldsymbol{0}\\ 1\end{bmatrix}r \qquad (20)$$
$$y=[\boldsymbol{C} \quad 0]\begin{bmatrix}z\\ m\end{bmatrix}$$

By defining desired performance parameters, and a desired characteristic equation, $\boldsymbol{K_p}$ and K_I are used to reach a zero steady-state error design for the response and to optimize the oscillation damping time.

5.1 Application to the Compressor Flow

To define a desired transient response as a frame of reference, the desired settling time of 0.5seconds and a percent overshoot of 5% will be adopted as desired conditions, this requires setting up a system with natural frequency = 11.59 and damping factor = 8. we get the following characteristic equation:

$$D(s)=s^2+16s+134.4 \qquad (21)$$

The third and fourth poles are set as double poles for simplicity and are set to be insignificant compared to the dominant pole. Based on this design, the characteristic equation (22) is developed to a 4th-order denominator:

$$D(s)=(s^2+16s+134.4)(s+20)(s+20) \qquad (22)$$
$$=s^4+56s^3+1174s^2+11780s+53750$$

$\boldsymbol{K_p}$ is a three-element vector defined as

$$\boldsymbol{K}_p=[K_{p1} \quad K_{p2} \quad K_{p3}]$$

while K_I is a scalar.

Two of the poles in equation (22) will be tuned by the PI controller for zero steady-state error, and the remaining poles can be placed using the pole placement method to reduce their effect on the transient performance.

Returning now to the original system, and substituting the PI controls:

$$\begin{bmatrix}\dot{z_1}\\ \dot{z_2}\\ \dot{z_3}\\ \dot{m}\end{bmatrix}=\begin{bmatrix}0 & 1 & 0 & 0\\ 0 & 0 & 1 & 0\\ K_{p1} & K_{p2} & K_{p3} & K_I\\ -1 & 0 & 0 & 0\end{bmatrix}\begin{bmatrix}z_1\\ z_2\\ z_3\\ m\end{bmatrix}+\begin{bmatrix}0\\ 0\\ 0\\ 1\end{bmatrix}r \qquad (23)$$

$$y=[1 \quad 0 \quad 0 \quad 0]\begin{bmatrix}z_1\\ z_2\\ z_3\\ m\end{bmatrix}$$

The characteristic polynomial of this system is:

$$\Delta(s)=\det(s\boldsymbol{I}-\overline{\boldsymbol{A}})$$
$$\Delta(s)=\det\left(\begin{bmatrix}s & -1 & 0 & 0\\ 0 & s & -1 & 0\\ -K_{P1} & -K_{P2} & s-K_{p3} & -K_I\\ 1 & 0 & 0 & s\end{bmatrix}\right) \qquad (24)$$
$$\Delta(s)=s^4-K_{p3}.s-K_{p2}.s-K_{p1}.s+K_I$$

By identification we get:

$$K_{p3}=-56, K_{p2}=-1174, K_{p1}=-11780, K_I=53750$$

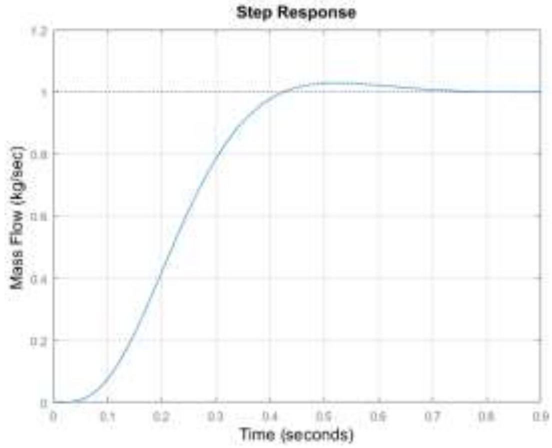

Fig. 3: Designed step response

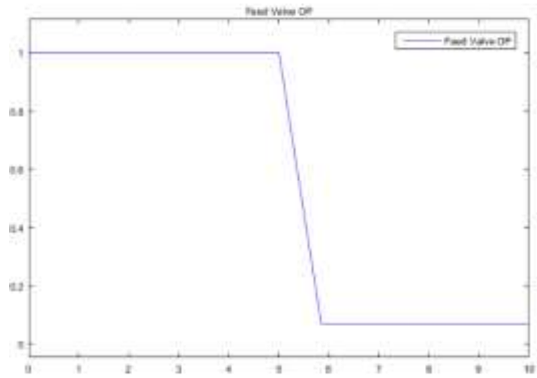

Fig. 4: Feed Flow Valve Opening (0 completely closed- 1 Completely open) vs time.

Fig. 3 Shows the desired mass flow step response with a reference gain of 53750, knowing that the amplitude is an arbitrary set.

At t=5sec, the feed flow valve is chocked to 20%, if this disturbance is applied to the throttle valve, an improvement in the system's stability will be observed. Fig. 4 and Fig. 5 show the feed flow valve closure pattern and the development of the Feed Flow vs. time.

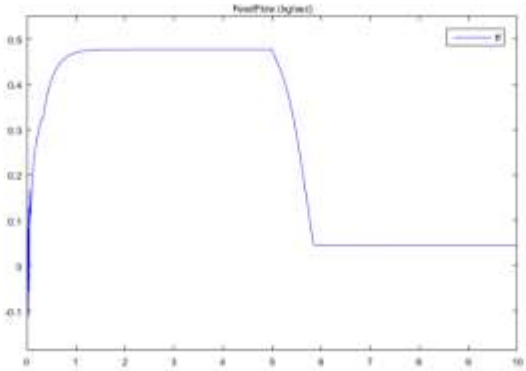

Fig. 5: Feed Flow (kg/sec) vs time (uncontrolled).

In the compressor characteristic, shown in Fig. 6, and after applying the feed flow disturbance, the operating point perfectly stabilized at the desired flow without any observable flow reversal or fluctuation at the Speedline corresponding to 30000rpm which is set as the desired speed.
In Fig. 6, while the feed flow rises into the desirable suction value, the speed regulator pushes the whole system to stabilize toward the 30000rpm Speedline, when the system reached the desired Speedline, the operating point stabilizes till the feed flow is disturbed at t= 5sec.

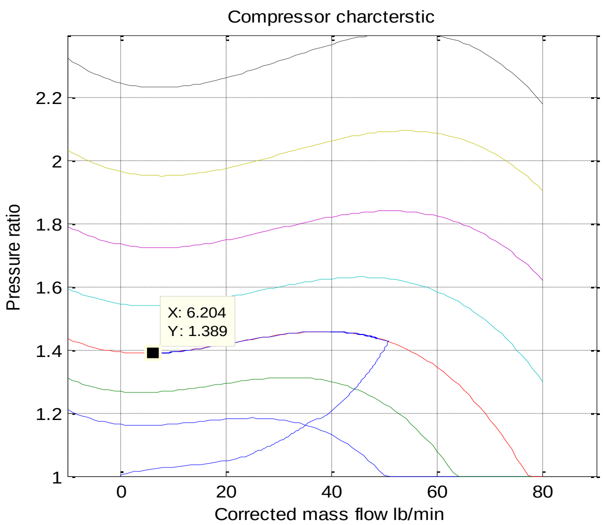

Fig. 6: Controlled Compressor map

At the time t=2 seconds, the feed flow valve is closed and reopened to ensure that the system is stabilized at the required Speedline and the controller is responding to the changes in the system, [12], [13], this also meant to push the compressor to the surge line limit through reducing the mass flow and increasing the pressure rise to preliminary assess the system behavior. The desired mass flow through the compressor is set to 0.1 kg/s once the compressor stabilizes to be able to test the system without having a negative control signal that results in a full valve closure.

In Fig. 7a and Fig. 7b, the controller is deactivated by completely closing the recycle valve to assess the effect of the controller on the stability of the system. The mass flow in Fig. 7a and the pressure ratio in Fig. 7b suffers from large oscillations when the feed flow valve is closed with the controller deactivated, the flow reversal (negative mass flow) can be observed.

The Controlled System in Fig. 8 shows more stable plenum pressures with no flow reversal when operated near the surge area, denoting that the linear controller can stabilize the system operation near the surge line and dampen the sinusoidal oscillations originally generated by the non-controlled system. The peak in the transient reached at the time t=7.2s may result in a perturbation to the physical system as it presents a significant drop in the mass flow and the plenum-1 pressures, nevertheless, this perturbation is not significantly visible in the plenum-2 pressure at the centrifugal compressor outlet since it is partially dampened via the recycle line.

From Fig. 8f, it is noticeable that as we tend to push the operating point to the Speedline's most efficient point, the control signal moves toward negative values which means a total closure for the recycle valve, which is physically impossible due to the non-realistic sense of valve "negative" closure, thus the control signal is limited to a minimum value of zero. For large speeds, the compressor can be operated on a more efficient pattern with $Q_r < Q$. Once the mass flow stabilizes in the normally-unstable region at 0.1kg/sec, the control signal decreases to a minimum that ensures zero steady-state error.

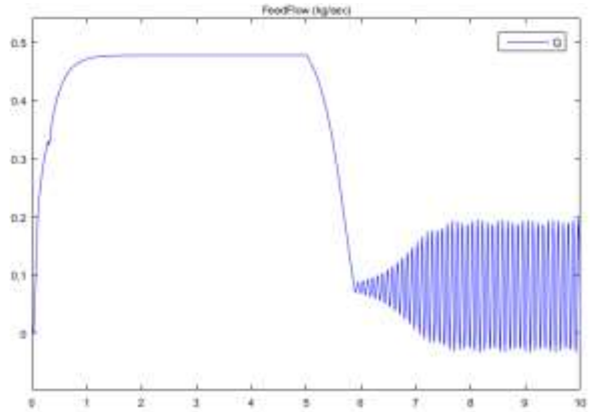

Fig. 7a: Feed Flow for under surge (uncontrolled)

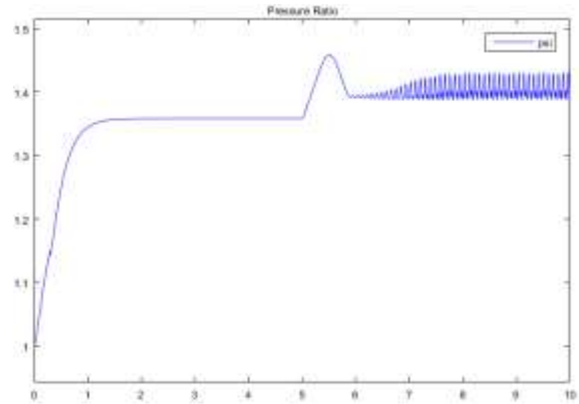

Fig. 7b: Pressure ratio for under surge (uncontrolled).

6 Conclusion

In this work, feedback linearization has been presented along with the Moor-Greitzer compressor model. A composite PI controller has been used on a linearized compression system with the aim of extending the compressor's effective working area and avoiding surge. The controller was designed to allow the compressor to operate at maximum efficiency while stabilizing the flow and the main drive speed. The linearization of the system, when the internal dynamics are stable, results in a system that is both, less complex and easier to control.

This methodology can overcome significant uncertainties in nominal models, [14], and can be generalized to a considerable class of model-based systems. The linear controllers designed using the state space approach for both the drive speed control and the mass flow control permits to development of a new approach for centrifugal compressors' active surge control.

The linearized model was simulated using the polynomial approximation, this allowed the operating point to be extended up to near-zero mass flow while the system is operating under the controller and the linearization blocks. The combination of Feedback linearization, linear PID controllers, and pole placement techniques in this paper were able to extend the operating region to cover up, theoretically, the whole right upper-hand quarter of the characteristics plane allowing the compressor to perfectly operate at the left of the surge line while preventing the natural flow reversal and the surge induced fluctuations.

The Results obtained in this work reduce the system complexity and allow for the application of other control techniques to reduce the transient time and transient peak to allow for a stable and homogeneous operating mode. Focusing on damping the transient mode in the controlled system through the controller's design enhancement can result in smooth operational curves for sensitive processes with large nonlinearities similar to the centrifugal gas compression systems. In addition to the above, simulating the system with a pulsation-damping built-in pulsation-damping system may result in smoother transient conditions.

7 Nomenclature

Table 1. Nomenclature and Abbreviations

Parameter	***Definition***	***Value (UM)***
PID	*Proportional-Integral-Derivative*	--
SISO	*Single-Input Single-Output*	--
I/O	*Input/Output*	--
PI	*Proportional-Integral*	--
P_{01}	*Atmospheric Pressure*	101325 (Pa)
P_1	*Plenum-1 Pressure*	Pa
V_1	*Plenums-1 Volumes*	0.07 (m^3)
P_2	*Plenum-2 Pressure*	Pa
V_2	*Plenum-2 Volumes*	0.12 (m^3)
Q	*Mass flow*	kg/s
A	*Duct Cross-section Area*	0.08 (m^2)
L	*Duct Length*	2.93 (m)
Ψ_c	*Pressure rise*	--
ω	*Rotation velocity*	rad/s
Q_f, Q_t, Q_r	*Feed, Throttle, and Recycle flows*	kg/s
r_1	*Inducer Radius*	0.0393 (m)
r_2	*Impeller Radius*	0.0567 (m)
τ_d, τ_c	*Drive and Compressor torques*	$N \times m$
J	*Impeller inertia*	$kg \times m^2$
a_p	*Speed of sound*	343 (m/s)
μ	*Energy transfer coefficient*	0.99
α	*Incidence loss cst*	--
C_p	*Flow coefficient*	--
T_{01}	*Atmospheric temperature*	25 (°C)
k_t, k_r	*Throttle and Recycle valves gains*	--
k_f	*Friction constant*	--

References:

[1] Hamedi, H., Rashidi, F. and Khamehchi, E., A novel approach to the gas-lift allocation optimization problem, *Petroleum Science and Technology*, 29(4), 2011, pp.418-427.

[2] Isidori, Alberto, ed, *Nonlinear control systems: an introduction*, Berlin, Heidelberg: Springer Berlin Heidelberg, 1985.

[3] Aranda-Bricaire, Eduardo, Claude H. Moog, and J-B. Pomet, A linear algebraic framework for dynamic feedback linearization, *IEEE Transactions on Automatic Control 40*, no. 1 (1995), pp.127-132.

[4] Vidyasagar, Mathukumalli, *Nonlinear systems analysis.* Society for Industrial and Applied Mathematics, 2002.

[5] Greitzer, Edward M, *Surge and rotating stall in axial flow compressors—Part I: Theoretical compression system model*, (1976), pp.190-198.

[6] Gravdahl, Jan Tommy, and Olav Egeland, *Compressor surge and rotating stall: Modeling and control*, Springer Science & Business Media, 2012.

[7] Cortés, Jorge, Lecture 3: *Feedback linearization of MIMO systems*, MAE 281B: Nonlinear Control, 2008, pp.1-11.

[8] Marquez, Horacio J, *Nonlinear control systems: analysis and design*, Hoboken^ eN. JNJ: John Wiley, Vol.161, 2003.

[9] Khalil, Hassan K, *Nonlinear systems third edition*, Patience Hall, Vol.115, 2002.

[10] Slotine, Jean-Jacques E., and Weiping Li, *Applied nonlinear control*, Englewood Cliffs, NJ: Prentice hall, Vol. 199, no. 1, 1991.

[11] Pomet, J.B., Moog, C.H. and Aranda, E., A non-exact Brunovsky form and dynamic feedback linearization, *Proceedings of the 31st IEEE Conference on Decision and Control IEEE*, 1992, pp. 2012-2017.

[12] Chetate, B., Zamoum, R., Fegriche, A. and Boumdin, M., PID and novel approach of PI fuzzy logic controllers for active surge in centrifugal compressor, *Arabian Journal for Science and Engineering*, Vol.38, No.6, 2013, pp.1405-1414.

[13] Boushaki, R.Z., Chetate, B. and Zamoum, Y., Artificial neural network control of the recycle compression system, *Studies in Informatics and Control*, Vol. 23, No,1, pp.65-76.

[14] Wang, Y., Wang, L. and Brdys, M.A., Designing robust feedback linearisation controllers using imperfect dynamic models and sensor feedback, *Cogent Engineering*, Vol.3, No.1, p.1173529.

Contribution of Individual Authors to the Creation of a Scientific Article (Ghostwriting Policy)

The authors equally contributed in the present research, at all stages from the formulation of the problem to the final findings and solution.

Sources of Funding for Research Presented in a Scientific Article or Scientific Article Itself

No funding was received for conducting this study.

Conflict of Interest

The authors have no conflicts of interest to declare that are relevant to the content of this article.

DOI: 10.37394/232016.2023.18.1

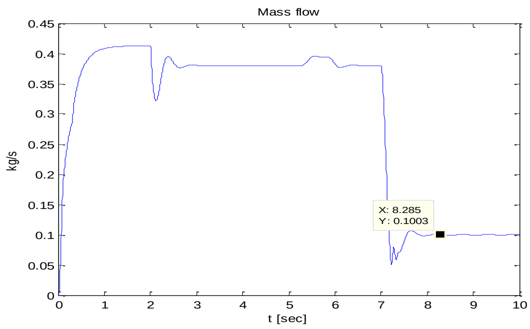

Fig. 8a: Mass Flow (kg/sec) vs time

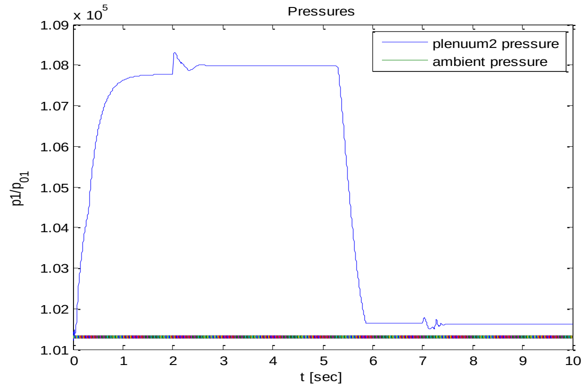

Fig. 8d: Plenum2 pressure development vs, time

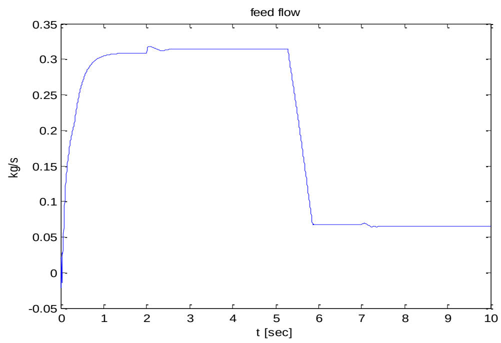

Fig. 8b: Feed flow for the controlled system vs time

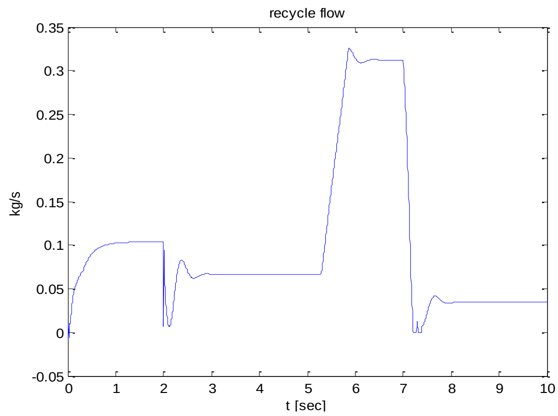

Fig. 8e: Recycle flow vs. time

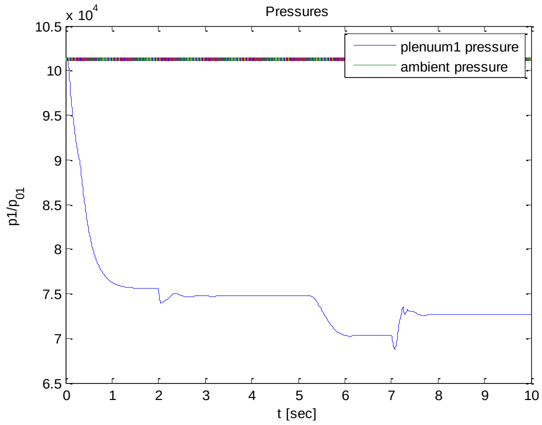

Fig. 8c: Plenum1 pressure development vs, time

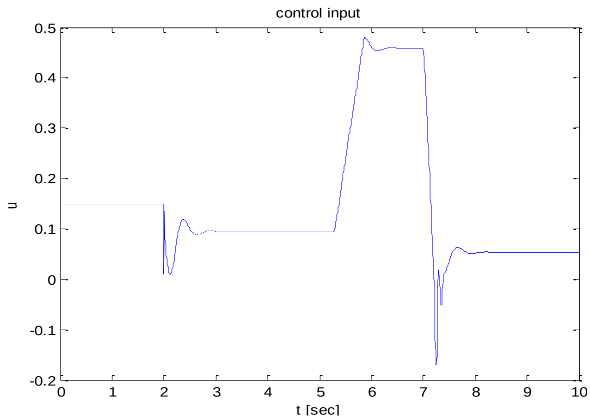

Fig. 8f: Control signal “u” vs. time.

Fig. 8: Controlled compressor parameters