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Design and Simulation of an Optimized Anti-Surge Adaptive Control System for Centrifugal Compressors in Gas Facilities

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ABSTRACT
This research enhances an adaptive control system for centrifugal compressors using MATLAB Simulink software, concentrating on gas facilities such as refineries and natural gas processing plants. Centrifugal compressors, crucial for these processes, are susceptible to surges induced by variable gas demands and operational disruptions, which, if not managed properly, might lead to compressor damage. An improved anti-surge adaptive control system was developed and simulated, using sophisticated optimization methods and adaptive control algorithms to monitor operational circumstances, predict surge occurrences, and implement preventative actions. Simulation results demonstrated that the proposed control system effectively mitigates surge occurrences, ensuring stable compressor operations across diverse conditions, while exhibiting substantial enhancements in the response time of the anti-surge valve, as well as improvements in gas compression processes, safety, reliability, and efficiency. The study advocates for more investigation into sophisticated control methodologies, such as the incorporation of machine learning and model predictive control, to improve flexible functionalities.

Keywords: Adaptive control system; anti-surge; centrifugal compressor;gas facility;MATLAB.

I. INTRODUCTION

Gas facilities, including refineries, petrochemical factories, and natural gas processing plants, depend significantly on efficient and dependable centrifugal compressors for gas stream compression. Centrifugal compressors are vital for achieving the necessary gas flow rates and pressures required for the effective operation of these facilities. Fluctuations in gas demand, abrupt pressure variations, and other operational disruptions may result in surges that compromise performance and cause damage (Al-Qahtani et al., 2021; Maghrabi et al., 2023; Ndinojuo et al., 2016). To alleviate the risks linked to surges and guarantee the secure and efficient functioning of gas facilities, the design and execution of an optimized anti-surge adaptive control system are essential. This control system continually analyzes the operating characteristics of the compressor, anticipates possible surges, and implements proactive measures to avoid surge occurrence (Hashmi et al., 2023; Ma et al., 2023). This paper presents an improved anti-surge adaptive control system for a centrifugal compressor, designed to efficiently detect and react to variable circumstances, hence ensuring stable compressor performance despite disturbances that may induce surges. This study enhances the current understanding of surge avoidance in centrifugal compressors and provides a basis for developing effective anti-surge control systems in operational gas plants. This investigation aims to improve the performance of the anti-surge adaptive control system for compressors while providing insights and solutions applicable to analogous gas-gathering systems and demanding industrial environments worldwide, thus contributing to the progress of gas facility technology and environmental sustainability.

Centrifugal compressors are vital equipment in the oil and gas industry. The compressors provide a steady fluid flow at the appropriate pressure. Nonetheless, a significant disadvantage of the compressor is surge. An increase in compressors is an unstable condition resulting from inadequate or diminished incoming flow to the compressor, requiring anti-surge management interventions. The current control systems for centrifugal compressors lack the flexibility and optimization to effectively prevent surges, leading to operating inefficiencies, increased energy consumption, and potential equipment damage. The anti-surge controllers are designed to accurately manage stage flow under low flow and turndown conditions, while preventing surge events via the use of specialized algorithms. When the operating point of a compressor stage diverges positively from the surge control line, the anti-surge controller will close the associated anti-surge valve. When the compressor stage operating point falls below the surge control line, the anti-surge controller will engage the appropriate anti-surge valve (Jones, 2016; Smith, 2018). The proliferation of compressors may result in damage; thus, it is essential to develop and simulate an optimal and adaptive anti-surge control system capable of intelligently regulating compressor operations to provide stable and dependable performance while preventing surge(Eludu & Ndinojuo, 2016; Jones, 2018; Smith, 2017).

II. LITERATURE REVIEW

2.1 Evolution of adaptive control system

The evolution of adaptive control system has had a remarkable trend since the mid-20th century, with some remarkable/significant milestones and technological advancement. As far back as the 1950's, the concept of adaptive control emerged, primarily designed for use by autopilot in supersonic aircrafts. The design system was such that the aircraft could adjust to changing flight conditions whenever the need arises. Key theoretical foundations laid, include Bellman's dynamic programming and Feldbanm's dual control, which emphasized the need for controllers to both direct and probe the system (Wittenmark, 2011). Around mid-20th century, the advancement of adaptive control technologies saw the development of the Model Reference Adaptive Controller (MRAC) and the Self-Tuning Regulator (STR). Advances in computational power and algorithms led to more robust and efficient adaptive control system. This era saw the introduction of adaptive filtering and neural networks being integrated into control systems. Adaptive control in the 21st century has had significant improvements, as the integration of machine learning and artificial intelligence has revolutionized and improved adaptive control, such as adapt in real time, handling complex and non-linear dynamics more effectively, robotics and advanced manufacturing.

2.2 Fundamentals of centrifugal compressors and working principle

Centrifugal compressors are essential in several industrial applications for effectively increasing the pressure of gas streams. Comprehending the principles of centrifugal compressors is essential for the proper design and operation of these devices. This section offers a comprehensive examination of the essential concepts, elements, and operating features that characterize the basics of centrifugal compressors. Centrifugal compressors function by transforming kinetic energy into potential energy (Devold, 2013). The procedure entails pulling gas via an axial intake and enhancing its kinetic energy by accelerating it radially. The kinetic energy is then transformed into potential energy when the gas is decelerated in the diffuser, leading to a rise in pressure. This dynamic transition is essential to the compressive function of centrifugal compressors.

2.2.1Types of centrifugal compressors

Centrifugal compressors are categorized according to their design and the number of impellers. Various kinds of centrifugal compressors in the market are contingent upon the purpose or desire.

2.2.1a Single stage compressors

This compressor operates using a single-stage compression process. Its design is less intricate than those of other sophisticated compressor types. It consists of an air intake, through which air is pulled into the cylinder via an air intake valve, the piston that compresses the air or fluid, and the discharge, where the compressed air or fluid is released for use or export.

2.2.1b Multi-stage compressors

Multi-stage compressors consist of several impellers stacked in sequence to attain higher pressures and enhance efficiency. The multi-stage compressors have many stages for the compression of air or fluids, together with a cooling system. This research focuses on the multi-stage compressor system, which consists of the first stage, intermediate stage, and final stage. The pressure regime is elevated from 5.5 bar to about 80 bar before the gases intermingle with Non-Associated Gases (NAG) and go directly to treatment prior to export.

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2.2.1c Integral gear compressor

This category of compressors includes multi-stage compressors that use a central bull gear to drive several pinion shafts, each equipped with its own impeller.

2.2.1d Horizontal split casing centrifugal compressor

This centrifugal compressor has a horizontally split casing, allowing convenient access to interior components for maintenance and inspection. It consists of the casing, rotor, impellers, bearings, and seals. They are mostly located in petrochemical plants, LNG facilities, and air separation units.

2.3 Surge protection of centrifugal compressor

Amin and Maqsood's (2021) paper examines the important topic of surge protection in centrifugal compressors, with the objective of averting expensive damage to the machinery and reducing production losses due to process disruptions. Surge, a kind of dynamic instability, presents a substantial risk to compressor functionality, requiring efficient control methods. The HYSYS software was used to model the compressor and assess the efficacy of a sophisticated Anti-Surge Control (ASC) system. This technique enables a virtual assessment of the control system's efficacy in mitigating surges across diverse operating situations. The research demonstrates the advantages of advanced ASC compared to a traditional PID control method. Despite this notable contribution, the literature indicates an insufficient examination of the practical issues related to measurement inaccuracies, delays, and computing loads in the implementation of sophisticated control systems.

III. MATERIALS AND METHODS

3.1. System design approach

This study proposes a new method to improve the anti-surge control system for centrifugal compressors by integrating a fuzzy logic controller as a backup for potential PID (Proportional Integral & Derivative) controller failure. The principal control mechanism employs a standard PID algorithm to manage the compressor's intake valve position, using data from pressure and flow rate sensors at both the inlet and output. An adaptive mechanism is included into the anti-surge control system to provide successful surge prevention (Araki &Sugie, 2019; Yen &Langari, 1999). This device perpetually observes and modifies the controller's gain in response to changing operating circumstances, as seen in Figure 1.

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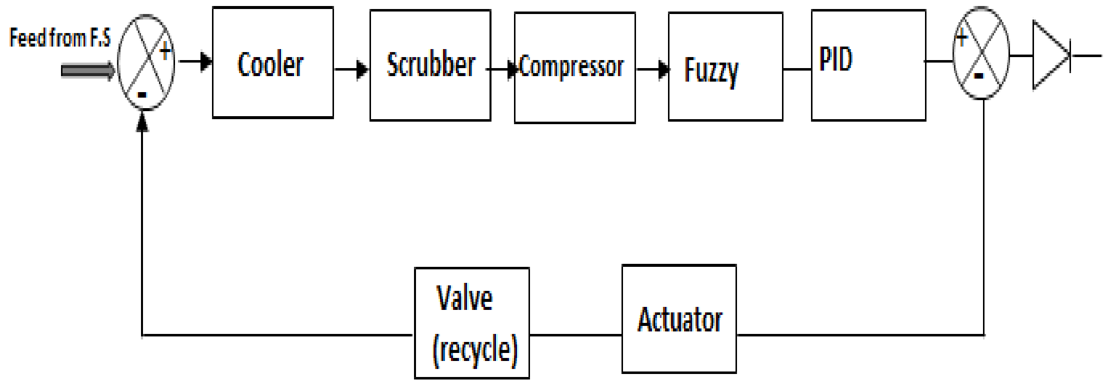

Figure 1: Schematic diagram of anti-surge control valve for a centrifugal compressor

Figure 2 illustrates a conventional compressor setup. The upstream vessel signifies a comparable upstream process and is supplied with a mass flow. The flow from the upstream vessel is directed into the cooler. The flow of the cooling medium is regulated by a controller that aims to maintain a consistent temperature of the processing gas outflow. Following the cooling process, liquid is extracted from the gas in the scrubber prior to its entry into the compressor. The compressor elevates the pressure, and a recycle line at the output facilitates the return of processed gas to the cooler, ensuring sufficient flow through the compressor. The portion of the flow that is not recycled exits the system via the check valve into the downstream vessel, which signifies the downstream process. The check valve inhibits reverse flow from the downstream vessel into the compressor. The

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recycling line has a control valve known as the anti-surge valve, which is regulated by the anti-surge controller. This controller inhibits the compressor from entering surge conditions. The compressor governor is powered by either an electrical motor or a gas turbine.

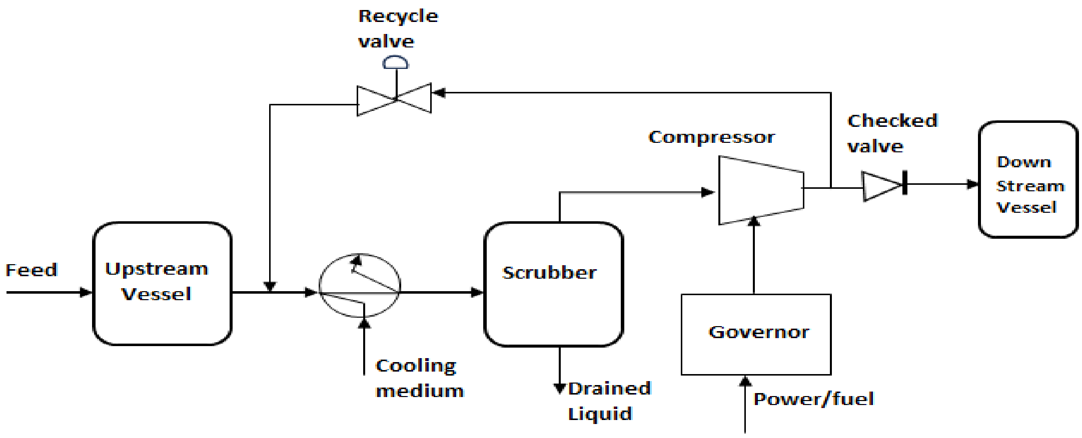

Figure 2. The compressor system

3.1.1. The Compressor Performance Map

Compressor manufacturers generate operating characteristic maps, often known as compressor performance graphs, for each compressor they build. These maps illustrate the surge limit line (SLL), indicating the flow at which the compressor's functioning becomes unstable at a certain speed. For steady operation, the compressor must function to the right of the surge limit line (Smith & Johnson, 2015). An operational line is proposed, usually positioned at a safe distance from the surge zone. The required surge offset, or margin, between the surge limit line and the operating line is contingent upon many parameters, including the velocity of instruments, actuators, and the rapidity of disturbances that must not activate the anti-surge open-loop backup system.

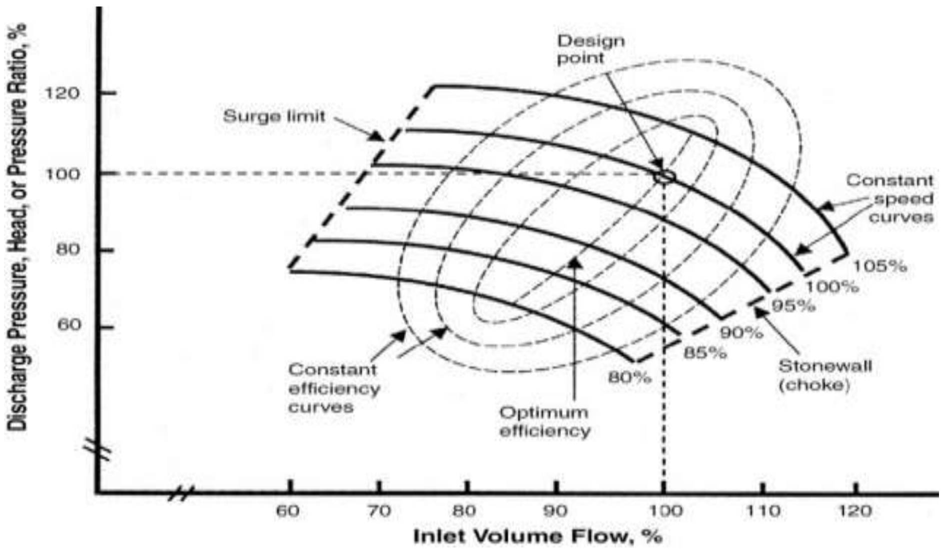

Figure 3. Compressor map of a compressor

3.4 Proportional-Integral-Derivative (PID) Controller

A PID controller, an abbreviation for Proportional-Integral-Derivative controller, is a feedback-driven control mechanism used in diverse control systems. It modifies the control signal according to the discrepancy between the target set-point and the current measured value. The PID controller has three elements: the proportional component (P), the integral component (I), and the

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derivative component (D). The proportional term delivers an instantaneous control response that correlates directly with the discrepancy between the set-point and the measured result. It facilitates the rapid minimization of error and aligns the system with the intended set-point. The integral component mitigates steady-state faults by accumulating the error over time and implementing remedial control measures. It perpetually modifies the control signal to achieve the intended set-point despite disruptions or biases.

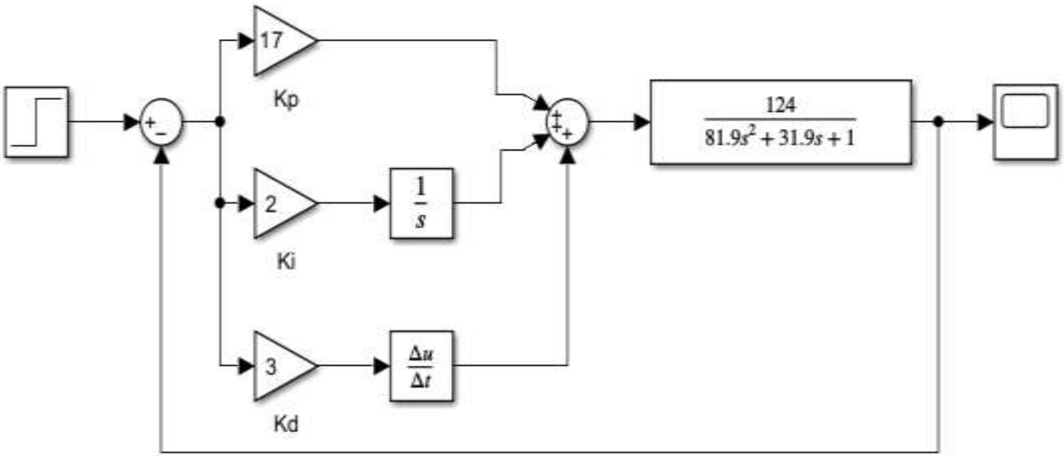

Figure 4. Block Diagram of PID controller

The Parameters used for the Simulink modelling of the PID controller are;
(a) Proportional Gain (K_p)
The proportional gain determines the proportional relationship between the error signal and the control output. In this case, the value of (K_p) used is 17.
(b) Integral Gain (K_i)
The integral gain determines the influence of past error values on the control output. It is used to eliminate any residual steady-state error. The value of (K_i) used is 2.
(c) Derivative Gain (K_d)
The derivative gain determines the rate of change of the error signal and helps to anticipate future behaviour of the error. It is used to reduce overshoot and oscillations in the system. The value of (K_d) used is 3.
These parameters are used to configure the PID controller to achieve the desired response and stability in the control system.

3.4.1 Mathematical Modelling of the PID Controller

Models depict the dynamics and behavior of the centrifugal compressor and other system components. They provide the simulation of the system's reaction to various operating situations and assist in the creation of the control algorithm. During the modeling phase, some assumptions were established to streamline the mathematical model and concentrate on the essential elements of anti-surge adaptive control. This mathematical model incorporates the dynamics of the compressor, and the control system is designed to sustain steady compressor operation while safeguarding against surge.

i. Neglecting some minor losses and inefficiencies in the compressor system to simplify the model and focus on the surge phenomenon and control system design.
ii. Assuming compressible flow through the compressor, which is appropriate for high-speed centrifugal compressors.
iii. Temporarily assuming a constant temperature throughout the system to simplify the analysis, assuming that the impact of temperature changes on the control system can be addressed separately.
iv. Assuming that the compressor operates at a steady-state condition and that the flow rate and pressure at the inlet and outlet of the compressor are constant. In this scenario, the compressor operates on a characteristic curve that relates the pressure ratio to the flow rate

Let us so take a basic anti-surge control system with a flow sensor at the compressor input and a pressure sensor at its output. By varying the compressor's speed, the control system maintains a continuous flow rate through it. By monitoring the pressure at the compressor output and adjusting the compressor speed suitably, the control system also prevents surge.

The anti-surge control system can be modeled using a feedback control system.
Let the plant be the centrifugal compressor,
let the input to the plant be the control signal= u(t) which adjusts the speed of the compressor.
Let the output of the plant be the pressure at the outlet of the compressor =P(t).
Let the reference signal be the desired pressure set-point= P_{ref}
The control system can be modeled as follows:
P(t) = G(u(t))
where G is the transfer function of the compressor.
The anti-surge control system can be modeled as a feedback control system:

$$e(t) = P_{ref} - P(t) \quad (1)$$

$$u(t) = K_P e(t) + K_i \int e(t)\,dt + K_d \frac{de(t)}{dt} \quad (2)$$

Again, assuming that the compressor dynamics can be described by the following first-order linear transfer function:

$$G(s) = \frac{K}{(Ts+1)} \quad (3)$$

Where
s=the Laplace variable
K= compressor gain
T=compressor time constant.
The control system can be modeled as a proportional-integral-derivative (PID) controller with the following transfer function:

$$C(s) = K_P + \frac{K_i}{s} + K_d s \quad (4)$$

where
e(t) =the error signal
$\frac{de(t)}{dt}$ = derivative of the error signal.

Kp, Ki, and Kd are the proportional, integral, and derivative tuning parameters, respectively.
Let's assume that the input to the plant is the square of the compressor speed u(t) and the output of the plant is the compressor pressure p(t)
The plant output can be derived by applying the transfer function G(s) to the input signal:
P(s) = G(s) u(s)

$$= \frac{K}{(Ts+1)(\frac{u^2}{s})} \quad (5)$$

Using the PID control system, the control signal can be derived from the error signal e(t), which is the difference between the reference signal and the measured compressor
e(t) = P_{ref} - P(t)
Taking the Laplace transform of the error signal, we get:

$$e(s) = P_{ref} - p(s)$$

$$= P_{ref} - \frac{K}{(Ts+1)(\frac{u^2}{s})}$$

The control signal u(s) can be derived from the error signal using the PID controller transfer function:

$$u(s) = C(s)\, e(s) \quad (6)$$

$$= (K_P + \frac{K_i}{s} + K_d s) e(s)$$

Substituting the expressions for e(s) and p(s) in the above equation, and rearranging terms, we obtain:

$$u(s) = (K_P + \frac{K_i}{s} + K_d s)(P_{ref}\, s + \frac{K}{(Ts+1)(\frac{u^2}{s})})$$

Solving for u(s), we get:

$$u(s) = \frac{K(k_p s^2 + k_i s + k_d s^3)}{k_p s^2 (Ts+1) + k_i s(Ts+1) + k_d s^3} \quad (7)$$

Taking the inverse Laplace transform of u(s), we obtain the control signal in the time domain:

$$u(t) = \frac{K\left(k_p \frac{d^2 e(t)}{dt^2} + k_i \int e(t)dt + k_d \frac{de(t)}{dt}\right)}{Ts+1} \quad (8)$$

In this streamlined approach, the control signal modulates the compressor speed according to the error signal, which indicates the discrepancy between the target reference pressure and the actual compressor pressure. The PID controller regulates control effort by altering the proportional, integral, and derivative gains. The control signal is used to modify the square of the compressor speed, hence maintaining steady operation and averting surge in the centrifugal compressor. This equation delineates the mathematical model for the anti-surge adaptive control system of centrifugal compressors, wherein the control signal modulates the compressor speed to sustain a constant flow rate and prevent surge by monitoring the pressure at the compressor outlet. The control system perpetually modulates the compressor's speed in accordance with the error signal. The integral component guarantees the eradication of steady-state error. The derivative term aids in predicting changes in pressure prior to their occurrence.

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IV. RESULTS AND DISCUSSION

4.1 Modeling in MATLAB

4.1.2 Various blocks and their functions

I. **Input block:** Represents the compressor's inlet conditions (e.g., pressure, temperature, mass flow rate) and any external disturbances. The step input block is used to generate the step function signal. It is used to test system responses to sudden changes in input. Parameters such as differential pressure (flow rate), pressure and temperature are fed into the input.

II. **Compressor model:** This core block simulates the compressor's behavior, often using performance maps, analytical equations, or empirical correlations. It determines the outlet pressure, temperature, and other relevant variables based on its internal logic and the input conditions. The compressor is used to increase the pressure of the gas by converting its kinetic energy into potential energy.

III. **PID controller:** This block implements the proportional-integral-derivative control algorithm. It takes the difference between the desired outlet pressure (setpoint) and the actual outlet pressure (measured by the sensor) as input and calculates an error signal. The controller then adjusts the control signal sent to the actuator based on this error signal and its internal parameters (proportional gain, integral gain, derivative gain).

IV. **Fuzzy logic controller:** Employs linguistic variables and fuzzy logic principles to make control decisions, offering robustness to nonlinearities and uncertainties.

V. **Actuator block:** Represents the physical mechanism (e.g., variable speed drive, inlet guide vanes) used to adjust the compressor's operation based on the control signal

VI. **Sensor block:** Measures the actual outlet pressure of the compressor and feeds it back to the PID controller.

VII. **Output block:** Represents the compressor's outlet conditions (pressure, temperature, mass flow rate, etc.).

VIII. **Performance calculation block:** Estimates key performance metrics (e.g., efficiency, power consumption, surge margin) for analysis and optimization.

IX. **Disturbance block:** Injects external disturbances (e.g., changes in inlet pressure, ambient temperature) to simulate real-world operating conditions.

4.2 Mathematical Equation of PID and FLC controllers.

A centrifugal compressor is a dynamic compressor that employs spinning impellers to enhance the pressure and flow rate of a gas or fluid. It functions according to the concept of centrifugal force, whereby the gas is expelled outward by the revolving impeller blades, thus transforming kinetic energy into elevated pressure. The primary mathematical formulae used to characterize the performance of a centrifugal compressor are:

A mathematical model for a centrifugal compressor was derived using basic concepts of fluid dynamics. We will examine a simplified model that presumes perfect gas behavior and disregards losses from friction and heat transmission, as represented by the mathematical equation in equation 9**.**

Let's denote the following parameters:

P_1 = Inlet pressure of the compressor
P_2 = Outlet pressure of the compressor
m = Mass flow rate of the gas passing through the compressor
N = Speed of the compressor impeller
R = Gas constant
T_1 = Inlet gas temperature
T_2 = Outlet gas temperature
γ = Specific heat ratio (ratio of specific heats at constant pressure and constant volume)

Now, using the ideal gas equation

$$P_1V_1/T_1 = P_2V_2/T_2 \quad (9)$$

We can assume that the gas density remains constant across the compressor.

Therefore, we can express the volumetric flow rate (Q) as:

$$Q = m/\rho \quad (10)$$

Where ρ is the constant gas density.

Next, we can use the Euler's equation for centrifugal compressors:

$$\Delta H = c_2^2/2 - c_1^2/2 + U_2 - U_1 \quad (11)$$

Where:

ΔH = Change in enthalpy (H) across the compressor
c_1 = Inlet gas velocity (m/s)

International Journal of Pharmaceutical Research and Applications
Volume 9, Issue 5 Sep - Oct 2024, pp: 858-866 www.ijprajournal.com ISSN: 2456-4494

c_2 = Outlet gas velocity(m/s)
U_1 = Inlet specific energy (J)
U_2 = Outlet specific energy (J)

The specific energy can be calculated using Bernoulli's equation:
$U = h + c^2/2$

Where h is the specific enthalpy and c is the gas velocity.
Finally, we can express the change in enthalpy (ΔH) as:
$$\Delta H = Cp(T_2 - T_1) \quad (12)$$

Where Cp is the specific heat capacity at constant pressure.
Combining all the equations, we can derive the mathematical model for the centrifugal compressor:
$Q = m/\rho$
$c_1 = Q/A_1$
$c_2 = Q/A_2$
$U_1 = h_1 + c_1^2/2$
$U_2 = h_2 + c_2^2/2$
$\Delta H = Cp(T_2 - T_1)$
$$\Delta H = c_2^2/2 - c_1^2/2 + U_2 - U_1 \quad (13)$$

We will now need more equations linking the compressor design parameters, gas properties, and impeller characteristics in order to get a comprehensive mathematical model. Efficiency formulae, impeller power curves, and other pertinent factors particular to the compressor being modeled may all be included in these equations. With a compressor duct length L, plenum volume v, a throttle and a drive unit producing a torque on the compressor then;

$$\dot{p}_p = \frac{a_p^2}{v_p}(w - w_t) \quad (14)$$
$$\dot{w} = \frac{A}{L}(\psi_C(\omega,\Omega)p_{01} - p_p) \quad (15)$$
$$\dot{\Omega} = \frac{1}{J}(\tau_P - \tau_C) \quad (16)$$

The basic equations above are a replica of Greitzer model. For a control that uses PID and PI fuzzy logic controllers, a variable speed is required. In the derived equations, the variables and parameters represent the following:
pp =Change in the plenum pressure with respect to time
ap=Throttle area
w= Mass flow rate through the compressor duct
wt=Target mass flow rate
A/L=Duct area to length ratio
ΨC(ω,Ω)=PI fuzzy logic controller
p01=Pre-compressor pressure
Ω =Change in compressor speed with respect to time
J=Moment of inertia
Tp=Torque provided by the drive unit
τC=Torque required by the compressor
To incorporate a PID controller and a PI fuzzy logic controller into the model, we can modify the basic equations as follows:
$$p_p = (ap^2)(\omega - \omega_t) + K_{PP}\left(p_p - p_{pset}\right) + K_{ip}\int(p_p - p_{pset})dt + K_{dp}\frac{d(p_p - p_{pset})}{dt} \quad (17)$$
Where, Kpp, Kip, and Kdp are the proportional, integral, and derivative gains for the PID controller respectively, and ppset is the desired plenum pressure set-point.
$$\omega = \frac{A}{L}\left(\psi_C(\omega,\Omega)p_{01} - p_p\right) + K_{Pw}(\omega - \omega_{set}) + K_{iw}\int(\omega - \omega_{set})\,dt \quad (18)$$
Where:
Kpw, Kiw are the proportional and integral gains for the PI fuzzy logic controller respectively, and wset is the desired mass flow rate setpoint.
$$\Omega = \frac{1}{J(\tau_P - \tau_C)} + Kp\Omega\ (\Omega - \Omega set) + Ki\Omega \int (\Omega - \Omega set)dt \quad (19)$$
Where
KpΩ, KiΩ are the proportional and integral gains for the PID controller respectively,
Ωset is the desired compressor speed set-point.

PID and PI fuzzy logic controllers can provide suitable control signals to maintain desired plenum pressure, mass flow rate, and compressor speed by changing the gain values in the controllers. The integration terms in the controllers assist to manage any bias or steady-state mistakes; the PID controller's derivative term enhances the responsiveness to dynamic changes. It is noteworthy that the particular needs and features of the centrifugal compressor under control will determine the exact tuning of the controllers gains. Iterative changes and performance help one to reach ideal tune.

V. CONCLUSION

Showing some overshoot and oscillations that may cause the compressor's wings to break, the PID controller stabilizes the mass flow in 9.98x10^-3 seconds. As shown in this work, the combination of PID methods with fuzzy logic helps to control surge instability, thus improving the compression system performance. Without overrun, the mass flow stabilised at 4.83 x 10^-3 seconds after the feed flow anti-surge valve closed. Designed to adapt to fluctuations in differential flow to the compressor's input and changes in differential pressure potentially causing compressor surge, the fuzzy logic controller Linked with PID controller input, the fuzzy logic controller's output improves the system's response to surge situations, therefore enabling a faster opening of the anti-surge valve to prevent such occurrences. Unlike the PID controller, the response time to surge control is much longer, occasionally leading the compressor to enter trip mode under surge circumstances. Little disturbances might maybe start surge events. The quick changes make it difficult for controllers to respond fast to prevent surges. This shows how much advanced adaptive control systems are absolutely needed.

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