People's Democratic Republic of Algeria

Ministry of Higher Education and Scientific Research

University M'Hamed BOUGARA – Boumerdes

Institute of Electrical and Electronic Engineering

Department of Power and Control

Final Year Project Report Presented in Partial Fulfilment of
the Requirements for the Degree of

MASTER

In **Electrical and Electronic Engineering**

Option: **Control Engineering**

Title:

Adaptive PID Fuzzy Logic control of the recycle compression systems

Presented by:

- **Yasmine Lahbiben**
- **Hadjer Ouacif**

Supervisor:

R.Boushaki

Registration Number:………/2016

Specially dedicated to

my beloved father, mother, brother, sisters and friends for their never ending support.

Thank you for everything.

And

To the loving memory of my Sister, You will always be remembered.

Yasmine Lahbiben

To my precious Mother,

My beloved Father,

To my Sisters and brothers

To all my friends and family

Hadjer Ouacif

Acknowledgements

Firstly and foremost, we would like to express our gratitude to Almighty Allah for giving us the guidance and strength in completing this final year project with success. We also would like to extent our greatest thanks to all those who gave us the possibility to complete this project. First and foremost, We would like to express our sincere gratitude to our supervisor Dr. Boushaki for her advice, comments, guidance, support and encouragement during the completion of our study.

We also would like to thank all other parties, those who have involved directly or indirectly in making this work a very great success. Finally, to our beloved family and friends. We owe you all, the heartiest gratitude and thanks for your encouragement, inspiration and support. Thank you for all the kindness.

Finally we address a deep appreciation, and a warm thank to electronic and electrical engineering institute members; especially our teachers that have taught us throughout our education at IGEE.

Abstract

Centrifugal compressors play an integral role in the oil and gas industry, helping power the multiple processes that convert oil and gas into thousands of practical products. These types of compressors can encounter a very dangerous and detrimental problem called surge. This work deals with the study of Centrifugal compressor surge and its prevention. In order to avoid the surge occurrence and ensure compressor's operation in the stable region with maximum efficiency, appropriate techniques are considered based on PID regulator. Classical PID controllers remain one of the simplest, most effective, robust, and easily certifiable control strategies. However, this simplicity comes with a price. Design tradeoffs between integral and derivative gain in a linear PID controller often make it difficult to achieve optimal performance. To solve this problem, the design of a multistage fuzzy PID controller is proposed in this thesis. The simulation results show the effectiveness of the multistage fuzzy PID controller in maintaining the speed and stability of the system.

Table of contents

Dedication…………………………………………………………………………... I

Acknowledgements………………………………………………………………… III

Abstract…………………………………………………………………………… IV

Table of contents…………………………………………………………………… V

List of figures……………………………………………………………………… VIII

List of Tables……………………………………………………………………….. XI

Index……………………………………………………………………………… XII

General Introduction……………………………………………………………… 1

Chapter 01: Generalities on compressors and surge phenomenon

1.1. Introduction……………………………………………………………………… 2

1.2. Centrifugal compressor…………………………………………………………… 3

1.2.1. Background and principle of operation……………………………………… 3

1.2.2. Characteristics of the compression system…………………………………... 5

1.3. Surge phenomenon……………………………………………………………… 6

1.3.1. Description of the surge………………………………………………………… 6

1.3.2. Cycles of surge and stability of operating point……………………………. 7

1.3.3. Consequences of surge………………………………………………………… 8

1.3.4. Surge avoidance and control…………………………………………..…. 8

1.4. Conclusion……………………………………………………..……………… 10

Chapter 2: Modeling of the compression system

2.1. Introduction……………………………………………………………………… 11

2.2. Brief history about Centrifugal Compressor Modeling………………………. 11

2.3. Model of compression system in open loop………………………………… 11

2.3.1. Derivation of the model……………………………………………….…… 13

2.3.2. Implementation and simulation of the model using Matlab-Simulink…….. 16

2.4. Modeling of recycling compression system……………………………….. 18

2.4.1. Implementation and simulation of the model in Matlab-Simulink………… 19

2.5. Conclusion……………………………………………………………….. 21

Chapter 3: PI and PID control of the recycle compression system

3.1. Introduction……………………………………………………………….. 22

3.2. PID control of recycle valve…………………………………………………… 22

3.2.1. Principle of a conventional PID controller……………………………….. 22

3.2.2. PID controller characteristics…………………………………………….. 23

3.2.3. PID controller regulation and design for recycle compression system……… 25

3.4. Recycle valve control……………………………………………………. 27

3.4.1. Implementation and simulation of the recycle Valve Control Using PI

Controller using Matlab -Simulink……………………………………………. 27

3.4.2. Implementation and simulation of the recycle Valve Control Using PID

Controller using Matlab -Simulink……………………………………………. 32

3.5. Conclusion……………………………………………………………... 34

Chapter 4: Adaptive multistage fuzzy logic PID controller of the recycle valve

4.1. Introduction……………………………………………………………… 35

4.2. Fuzzy logic………………………………………………………………. 35

4.2.1. History and applications……………………………………………….. 35

4.3. Fuzzy Logic Controller…………………………………………………… 36

4.3.1. Why use fuzzy logic controller…………………………………………. 36

4.3.2. Structure of the Fuzzy Logic Controller……………………………………... 36

4.4. Fuzzy PID Controller Design………………………………………………….. 38

4.5. Multi Stage PID Fuzzy Logic control of the recycle valve……………………. 39

4.5.1. The proposed Structure for MSFL PID controller…………………………… 39

4.5.2. Design of Multi Stage Fuzzy Logic PID controller………………………….. 40

4.6. Simulation and Results………………………………………………………… 44

4.6.1. Implementation and simulation of the model using Matlab-Simulink………. 44

4.7. Conclusion……………………………………………………………………... 48

General conclusion……………………………………………………………….. 49

List of figures

Fig1.1. Chart of compressor types……………………………………………... 3

Fig1.2. Single stage centrifugal compressor…………………………………… 4

Fig1.3. Major components of a multistage centrifugal compressor……………… 4

Fig1.4. The working principle of a centrifugal compressor……………………… 5

Fig1.5. Compressor performance map…………………………………………… 6

Fig1.6. Compressor characteristic with deep surge cycle………………………... 7

Fig1.7. Surge margin…………………………………………………………….. 9

Fig1.8. Surge line and surge control line………………………………………… 10

Fig 1.9. Schematic active surge control………………………………………….. 10

Fig2.1. Compression system retrieved from Egeland & Gravdahl (2003)... 12

Fig2.2. Velocity triangle at the impeller exit……………………………………... 14

Fig2.3. Simulink implementation of open loop model…………………………… 16

Fig2.4. Flow in the entire compressor…………………………………………… 17

Fig2.5. Flow through throttle…………………………………………………….. 17

Fig2.6. Ambient and plenum pressures…………………………………………... 17

Fig2.7. The compressor Speed (N)……………………………………………….. 17

Fig2.8. The compressor operating point…………………………………………. 17

Fig2.9. The recycle compression system………………………………………… 18

Fig2.10. Compression system with recycle loop in Matlab-Simulink…………… 19

Fig2.11. Mass flow……………………………………………………………….. 20

Fig2.12. Flow through throttle…………………………………………………… 20

Fig2.13. Suction and discharge pressures………………………………………… 20

Fig2.14. Compressor speed………………………………………………………. 20

Fig2.15. Feed and Recycle flows………………………………………………… 20

Fig2.16. The compressor operating point………………………………………… 20

Fig 3.1. A block diagram of a PID controllr in a feedback loop………………… 23

Fig3.2. PID controller characteristics……………………………………………. 24

Fig 3.3. The PID strategy of a recycle valve……………………………………… 26

Fig 3.4. Scheme of an anti-surge PID regulator…………………………………….. 27
Fig 3.5. Simulink implementation of controlling the recycle valve using PI controller………………………………………………………………………. 27
Fig 3.6. The evolution of the compressor states under PI control where $m_{set}0.35\ kg$ sec ………………………………………………………………… 29
Fig 3.6.a. Mass Flow……………………………………………………………... 28
Fig 3.6.b. Throttle flow…………………………………………………………... 28
Fig 3.6.c. Suction and discharge pressures………………………………………. 28
Fig 3.6.d. Feed and recycle flows………………………………………………… 28
Fig 3.6.e. Control input…………………………………………………………… 29
Fig 3.6.f. Operating point………………………………………………………… 29
Fig 3.7. The evolution of the compressor states under PI control where $m_{set}0.20\ kg$ sec ………………………………………………………………… 30
Fig 3.7.a. Mass Flow……………………………………………………………... 30
Fig 3.7.b. Throttle flow…………………………………………………………... 30
Fig3.7.c. Suction and discharge pressures……………………………………….. 30
Fig 3.7.d. Feed and recycle flows………………………………………………… 30
Fig 3.7.e. Control input…………………………………………………………… 30
Fig 3.7.f. Operating point……………………………………………………….... 30
Fig 3.8. The evolution of the compressor states under PI control where $m_{set}0.25\ kg$ sec ………………………………………………………………… 31
Fig 3.8.a. Mass Flow……………………………………………………………... 31
Fig 3.8.b. Throttle flow…………………………………………………………... 31
Fig 3.8.c. Suction and discharge pressures………………………………………. 31
Fig3.8.d. Feed and recycle flows………………………………………………… 31
Fig3.8.e. Controlinput…………………………………………………………….. 31
Fig3.8.f. Operating point…………………………………………………………. 31
Fig3.9. Simulink implementation of controlling the recycle valve using PID controller………………………………………………………………………. 32
Fig3.10. The evolution of the compressor states under PID control……………... 33
Fig3.10.a. Mass Flow…………………………………………………………….. 33
Fig3.10.b. Feed and recycle flow…………………………………………………. 33
Fig3.10.c. Suction and discharge pressures………………………………………. 33

Fig3.10.d. Compressor speed……………………………………………………….. 33
Fig3.10.d. Operating point……………………………………………………….. 33
Fig4.1. Overview diagram of a fuzzy system…………………………………….. 36
Fig4.2. Block diagram of fuzzy PID controller…………………………………… 39
Fig4.3. Simulink model of Multi stage FCPID controller………………………… 40
Fig4.4.FIS editor for Kp.fis of MSFLPID controller…………………………….. 41
Fig4.5. Input membership fuction for MSFLPID controller……………………… 42
Fig4.6. Output membership fuction for MSFLPID controller……………………. 42
Fig4.7. Simulink implementation of PID and Multi Stage FPID controllers……. 44
Fig4.8. Simulation results of the closed loop system using a Multi Stage fuzzy logic PID controller…………………………………………………………. 45
Fig4.8.a. Mass Flow………………………………………………………… 45
Fig4.8.b. Throttle flow……………………………………………………… 45
Fig4.8.c. Suction and discharge pressures……………………………………….. 45
Fig4.8.d. Feed and recycle flows………………………………………………… 45
Fig4.8.e. Compressor speed………………………………………………………. 45
Fig4.8.f. Operating point…………………………………………………………. 45
Fig4.9. Simulation results of the closed loop system using a Multi Stage Fuzzy PID controller for periodic opening of the feed flow valve……………………. 46
Fig4.9.a. Mass Flow………………………………………………………… 46
Fig4.9.b. Throttle flow……………………………………………………… 46
Fig4.9.c. Suction and discharge pressures………………………………………... 46
Fig4.9.d. Feed and recycle flows………………………………………………… 46
Fig4.9.e. Compressor speed………………………………………………………. 46
Fig4.9.f. Operating point…………………………………………………………. 46

List of Tables

Table 3.1: Effect of Kp,Ki and Kd on the Performance of PID Controller.............................24

Table4.1. Rules for Kp of Multi stage FLPIDC...43

Table4.2. Rules for Ki of Multi stage FLPIDC...43

Table4.3. Rules for Kd of Multi stage FLPIDC...43

Index [9]

Parameter	**Meaning/values**	**Parameter**	**Meaning/values**
Q	The mass flow (kg/s)	Ψ_c (w, Ω)	Compressor characteristics (rise) (Pa/kg)
Q_f	The feed flow (kg/s)	ρ	Density of the fluid ($/m^3$)
Q_r	The recycle flow (kg/s)	C	Flow coefficient
Q_t	The throttle flow (kg/s)	τ_m	Drive torque ($N.m$)
p	Plenum pressure (Pa)	τ_c	Compressor torque ($N.m$)
N	Rotational velocity(rad/s)	μ	Energy transfer coefficient 0.99
a_p	The speed of sound=343 m/s	r_1	Inducer perimeter radius 0.0395m
c_p	Specific heat at constant pressure 1005(J/kg.K)	r_2	Impeller perimeter radius 0.0565m
Vp	Plenum volume(m^3)	α	Constant of incidence loss
V_1	Suction plenum volume 0.05 m^3	K_f	Friction constant
V_2	Discharge plenum volume 0.1 m^3	K	The throttle gain proportional to the opening of the throttle.
P_{01}	Ambient pressure 101325 Pa	J	The impeller inertia $5e^{-4}$ (kg m^2)
P_1	Suction plenum pressure (Pa)	$C_{\theta 1}$	The tangential velocity of the gas at the inducer exit.
P_2	Discharge plenum pressure (Pa)	$C_{\theta 2}$	The tangential velocity of the gas at the impeller exit.
T_{01}	Ambienttemperature 20 Celcius		
A	Duct area 0.07m²		
L	Duct length 2.85m		

General Introduction

Compressors are used in a wide variety of applications such as: power generation using industrial gas turbines, pressurization of gas in the process industry, transport of fluids in pipelines.

For both axial and centrifugal compressors, operation is limited, by choking at high mass flows when sonic velocity is reached in some component, and at low mass flows by the onset of two instabilities known as surge and rotating stall.

Centrifugal compressors constitute a main part of the process machinery in the oil and gas industry. The fundamental instability problem known as surge limits the operation range for compressors at low mass flows and affects the entire compression system. The instability is characterized by large limit cycle oscillations in compressor flow, and pressure rise that reduce compressor performance. This instability problem has been studied and a surge avoidance solution is established.

Traditionally, the surge has been avoided not stopped, this solution is based on keeping the operating point at the right side of the surge line. There is a potential for increasing the efficiency of compressor by allowing the operation point closer to the surge line, which is the case in industrial current systems. However, this raises the need for control techniques, which stabilize the compressor when disturbances occur or set point is changed, otherwise, the operating point may cause a crossing of the surge line leading to instability.

In this project we will first have a quick look at centrifugal compressors generalities, and an overview about the surge phenomenon. Next, in chapter two, a mathematical model is presented for both open and closed-loop (where a recycle manuel valve is added) of compression system and enriched with a simulink implementation for each.

Then, in chapter three, a recycle valve control solution is presented using a simple PI and PID controllers and simulation is also introduced.

Finally, in chapter four, a multistage PID fuzzy logic controller is designed and discussed.

1.1. Introduction :

A **gas compressor** is a mechanical gas transportation device that increases the pressure of a gas by confinement or by kinetic energy conversion which means by reducing its volume and pressing the molecules together.

Compressors are used everywhere. This includes turbojet engines used in aerospace propulsion, power generation using industrial gas turbines, turbo charging of internal combustion engines, pressurization of gas and fluids in the process industry, transport of fluids in pipelines, manufacturing and so on.

Compressors can be divided into two general categories depending on how the pressure rise is achieved: positive displacement compressors and dynamic compressors.

- **Positive displacement compression**

  In positive displacement compression, the air is drawn into one or more compression chambers, which are then closed off from the inlet. The enclosed volume of each chamber decreases through the displacement of one or more moving parts and the pressure increases, compressing the air internally. Once the pressure reaches the maximum pressure ratio, a port or valve opens; the continued reduction of volume in the compression chamber discharges the air into the outlet system. This type of compressors is sub-categorized as either reciprocating compressors or rotary compressors and each of those include a number of different types.

- **Dynamic compression (turbocompressors)**

  In dynamic compression, air is drawn between the blades on a rapid rotating compression impeller and accelerates to high velocity. The air or gas is then discharged through a diffuser, where the kinetic energy is transformed into static pressure. Most dynamic compressors are turbo compressors with an axial or **centrifugal** flow pattern, they convert kinetic energy into potential energy and are often designed for large-volume flow rates. Unlike a positive displacement compressor which works with a constant flow, a dynamic compressor works at a constant pressure. [1]

Selection of either technology depends on the application, but there is a rule of thumb: Dynamic compression technology is best suited for base load requirements, while positive displacement compression is better suited to variable load. The main types of gas compressors are illustrated and discussed below: [1] [2]

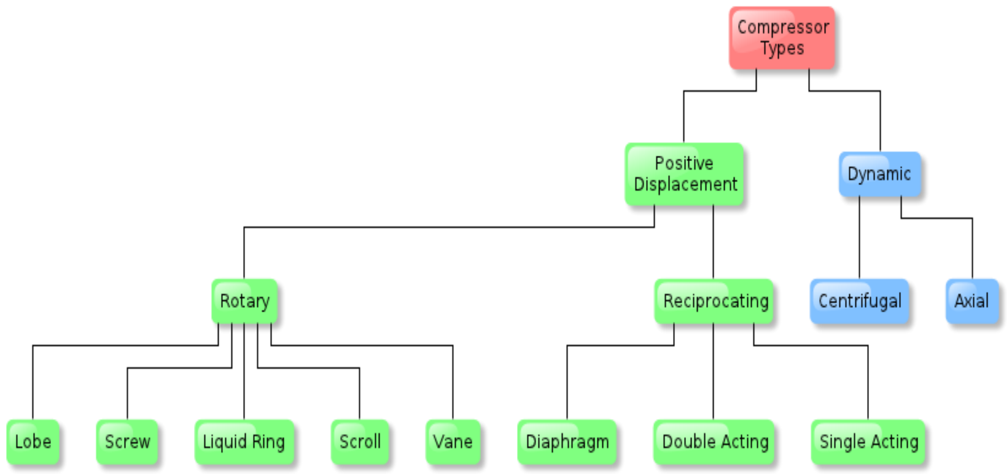


Fig1.1. Chart of compressor types. [1]

1.2. Centrifugal compressor:

1.2.1. Background and principle of operation:

Since the early 1950s, centrifugal compressors have secured a unique place in modern process plants. Also known as turbo-compressors, they belong to the roto-dynamic type of compressors. In these compressors the required pressure rise takes place due to the continuous conversion of angular momentum imparted to the refrigerant vapour by a high-speed impeller into static pressure. Centrifugal compressors that can be either a single-stage or multi-stages are steady-flow devices hence they are subjected to less vibration and noise. [3]

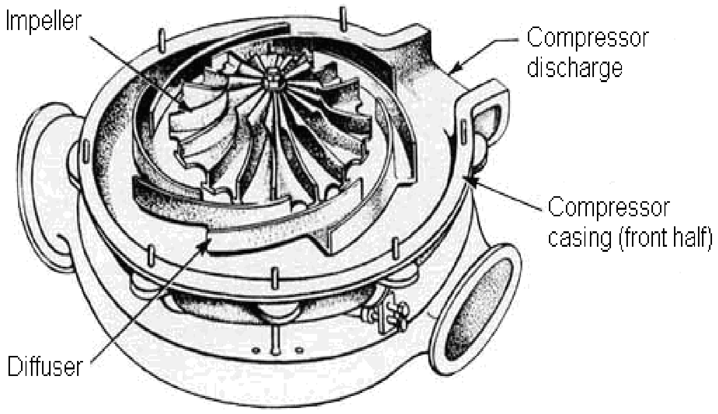


Fig1.2. Single stage centrifugal compressor. [11]

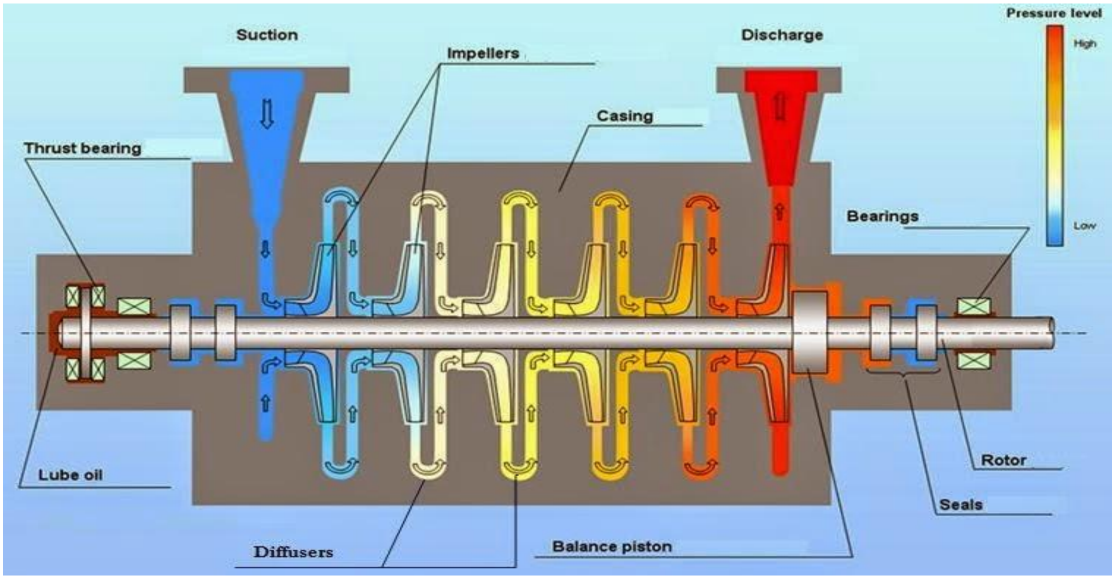


Fig1.3. Major components of a multistage centrifugal compressor. [12]

Every centrifugal compressor consists of two principal parts. The impeller and shaft, called the rotor, and the casing which the rotor is mounted in. The casing directs the flow into the impeller. The impeller can be viewed as a fan which imparts high velocity to the gas. The gas is decelerated in diverging channels with a rise in pressure. The deceleration is obtained by diffusion, and consequently the part of the casing containing the diverging channels is called the diffuser. The diffuser exit flow is collected in the volute, which most effectively leads the gas away. [3]

Figure 1.4 shows the working principle of a centrifugal compressor. As shown in the figure, low-pressure refrigerant enters the compressor through the eye of the impeller (1). The impeller (2) consists of a number of blades, which form flow passages (3) for refrigerant. From the eye, the refrigerant enters the flow passages formed by the impeller blades, which rotate at very high speed. As the refrigerant flows through the blade passages towards the tip of the impeller, it gains momentum and its static pressure also increases. From the tip of the impeller, the refrigerant flows into a stationary diffuser (4). In the diffuser, the refrigerant is decelerated and as a result the dynamic pressure drop is converted into static pressure rise, thus increasing the static pressure further. The vapour from the diffuser enters the volute casing (5) where further conversion of velocity into static pressure takes place due to the divergent shape of the volute. Finally, the pressurized refrigerant leaves the compressor from the volute casing (6). [4]

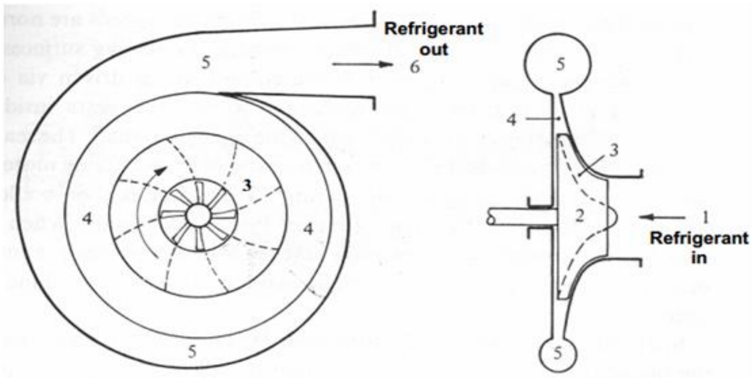


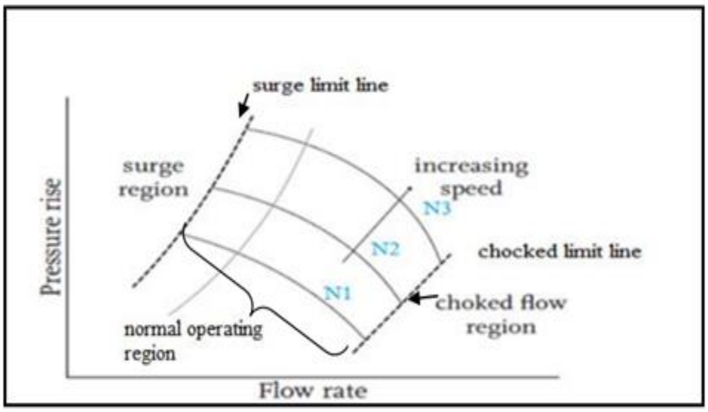
Fig1.4. The working principle of a centrifugal compressor. [4]

1: Refrigerant inlet (eye);	2: Impeller;
3: Refrigerant passages ;	4: Vaneless diffuser;
5: Volute casing;	6: Refrigerant discharge.

1.2.2. Characteristics of the compression system

Compressor performance is a graphical mapping that defines the compressor fields operation. The curves relate the rotational speed N, the pressure rise ψ across the compressor, and the mass flow Q through the compressor, figure1.5 shows the pressure –flow characteristics. The map can be divided into three regions:

- **Normal operating region:** in which the compressor works normally. It is the region confined between the surge region, and the choked region. It is mainly shown by the family of parallel curve that represents compressor operating at constant speed, where the pressure increases as the flow decreases and vice versa.
- **Surge region:** if the flow is continuously decreased, at some point the compressor operation becomes unstable causing a momentary reversal of flow, this leads to a severe vibration and damage. This phenomenon is called surge.
- **Choked flow region:** choke is an abnormal operating condition for centrifugal compressor. It occurs when the velocity of the wheel inlet reaches the sonic velocity where the stonewall (or choke) flow is the maximum flow rate that the compressor can push through.



(N1<N2<N3)

Fig1.5. Compressor performance map.

1.3. Surge phenomenon:

1.3.1. Description of the surge:

Compressor surge is a breakdown in compression which can result in a reversal of flow, and possibly a violent expulsion of previously compressed gas out through the line intake. This happens because the compressor is unable to work against the already compressed gas behind it.

1.3.2. Cycles of surge and stability of operating point:

The range of operation in turbo compressors, both axial and centrifugal, is limited, towards high mass flow, the compressor is limited by choking, when sonic velocity is reached in some component. Towards low mass flow, the stable operating region is bounded due to the occurrence of aerodynamic flow instabilities, surge and rotating stall. Traditionally, surge and rotating stall have been avoided by using control systems that prevent the operating point of the compressions system to enter the unstable region; the left of the surge line that is the stability boundary as shown in **figure1. 6**.

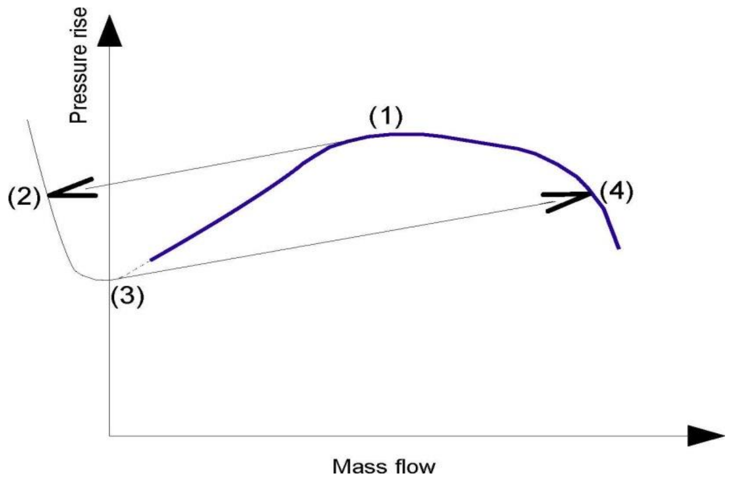


Fig1.6. Compressor characteristic with deep surge cycle.[3]

Figure 1.6 is a typical drawing of surge cycle. The cycle starts at (1) where the flow becomes unstable. Let's imagine that the throttle is closed a little bit further, lower mass flow occurs, and since we are at the surge point, there is lower discharge pressure. The "flow momentum" is decreased along with the discharge pressure being lower than the system pressure downstream. In the ease of a deep cycle this causes flow reversal (2), which ends in zero flow (3), At this point the discharge pressure is building up again, along the mass flow, to (4) and then to (1). Then, the surge cycle repeats. [3]

1.3.3 .Consequences of surge:

Surge causes overheating and damages; it may be violent enough to damage a compressor. Its cycles occur fast and can even go undetected if the applied instrumentation and controls are too slow. Giving cyclic flow, back-flow, high vibrations, pressure shocks and rapid temperature increase, So the effect of the surge can be summarized as follows:

- Unstable flow and pressure.
- Damage in sequence with rise severity to seals, bearing, impellers and shaft.
- Increase seal clearances and leakage.
- Lower energy efficiency.
- Reduce the compressor life.

1.3.4. Surge avoidance and control

The surge phenomenon represents a great problem in centrifugal compressor, therefore the only way to prevent surge is to recycle or blow off flow to keep the operating point away from the surge limit. Because compressing this extra flow entails an economic penalty, the control system must accurately determine how close the compressor is to surging and then maintain an adequate but not excessive flow rate. The combination of surge prediction and anti surge control should protect the machines with the smallest possible recycle rates.

According to de Jager (1995) and Egeland & Gravdahl (1999) there are three method to deal with surge:

- **Surge avoidance** (protection limit): in this method, the compressor is prevented from operating in the region near and beyond the surge line. This achieved by recirculation of the flow or blowing off flow through a bleed valve, and due to the inaccuracies in measurement and response time of the transmitters and valve, the anti surge control system achieves a surge control line (SCL) which determines a protection margin between the actual surge line (when the compressor will surge) and the point at which the control system will act; the compressor is then not allowed to operate in this margin. [8]

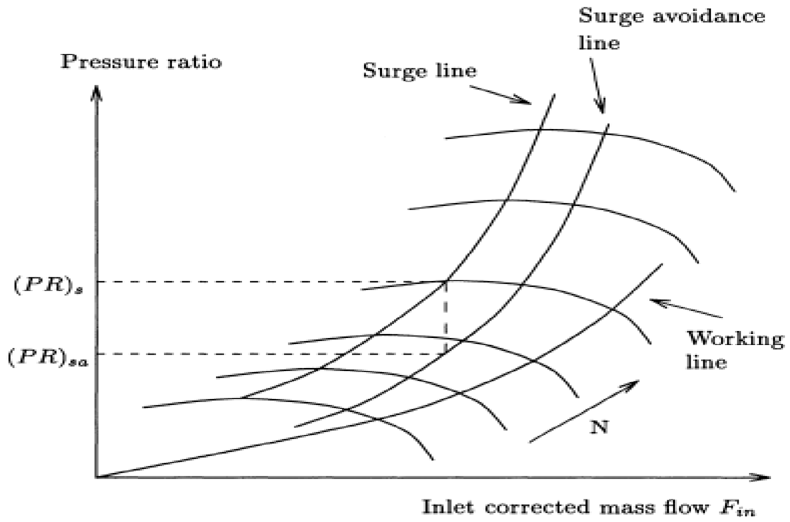


Fig1.7. Surge margin.

- **Surge avoidance and detection**: The protection margin represents a drawback for surge avoidance method since it reduces the efficiency of the compressor (maximum efficiency and pressure rise lay near SLL), this method also (surge avoidance and detection) avoids this by activation of the controller if the onset of the instabilities is detected. The main disadvantage of this method according De Jager (1995) is the problems of detecting the onset of the surge and the need of fast- acting control system. [8]

- **Active surge control:** The approach of active surge control aims at overcoming the drawbacks of surge avoidance, by stabilizing some part of the unstable area in the compressor map using feedback. This will allow for both operation in the peak efficiency and pressure rise regions located in the neighborhood of the surge line, as well as an extension of the operating range of the compressor.

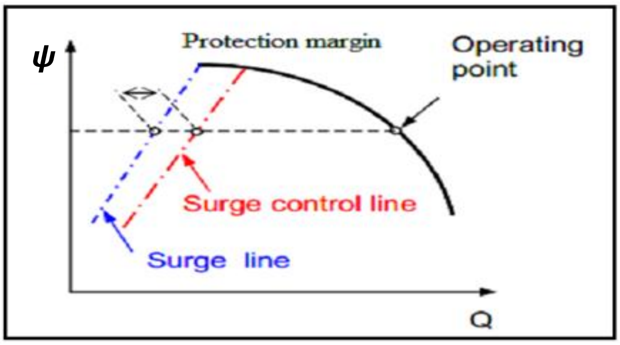


Fig1.8. Surge line and surge control line. [8]

An active surge control system comprises sensor(s) that provide information about the momentary flow conditions, actuator(s) to influence those conditions, and a controller that determines appropriate control action after comparing measured data with the desired flow conditions. A schematic representation of a general active surge control system is given in **Figure 1.9**.

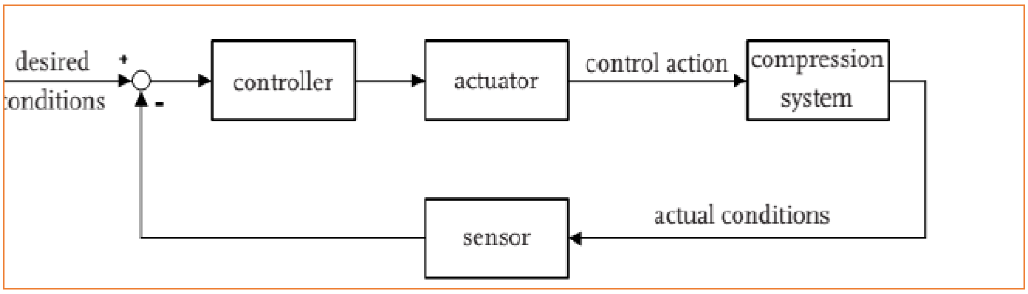


Figure 1.9. Schematic active surge control.

1.4. Conclusion:

First in this chapter, a general overview on compressors was presented, starting by giving a typical definition of a compressor, then the various domains of applications of these machines. After that, we listed the types of compressor that exist.

Then, we discussed the centrifugal compressor where a short background and a description of its main parts and of its principle of operation are given. In addition to its characteristics and performance map and operating range limits.

The last point covers the surge phenomenon through definition, description of the surge cycle, surge control methods and the consequence of surge in centrifugal compressor.

2.1. Introduction

In control system, the model is necessary for the analysis and design, it is essentially a mathematical description consisting of differential equations that can be driven using mathematical and physical laws.

Surge is an unstable operation mode of centrifugal compressors, which occurs when the operating point of the compressor is located to the left of the surge line, which is the stability limit in the compressor map. The phenomenon is manifested as oscillations of the mass flow, pressure rise and rotational speed of the compressor. The control system must be designed in such a way that the compressor remains operating in the nominal region.

In this chapter, we discuss the model describing the behavior of centrifugal compressor in two specific models of compression system. The first section will present the open loop configuration applied to describe the dynamic behavior of the compressor during the surge transient and the second section will present a recycle system with an anti-surge valve.

2.2. Brief history about Centrifugal Compressor Modeling:

In 1976, Greitzer had developed a model which is an open loop configuration that is applied to describe the dynamic behavior of the compression during the surge transients. This model is called lumped parameter model, and it is originally developed for compression systems with low pressure-ratio axial compressors but shown valid also for centrifugal compressors by Hansen et al. (1981). This model is based on a Helmholtz resonator type of model, as proposed by Emmons et al (1955).

According to Gravdahl and Egeland (1999), axial and centrifugal compressors mostly show similar flow instabilities.

In this section a multispeed model for centrifugal compressors is presented. When the speed is assumed constant, the model reduces to the dimensional model of Greitzer (Greitzer, 1976), originally developed for axial compressors, but shown valid also for centrifugal compressors by Hansen et al. (1981).

2.3. Model of compression system in open loop: [3]

The compression system is modeled as in Figure 2.1, it is represented by a duct, in which the compressor works, that discharges in a large volume called plenum. The compressed gas flows via the plenum and the compressor throttle (or blow-off valve) into the atmosphere and a drive unit imparting a torque on the compressor.

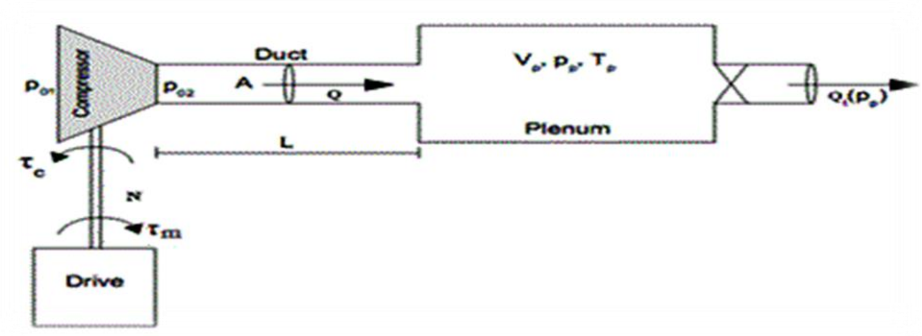


Fig2.1. Compression system retrieved from Egeland & Gravdahl (2003).[3]

In particular the model (**Fig2.1**) can be described by the following equations:

$$\dot{P}_p = \frac{a^2{}_p}{V_p}(Q - Q_t) \tag{2.1}$$

$$\dot{Q} = \frac{A}{L}(\boldsymbol{\psi}_c(Q,,N)\,P_{01} - P_p) \tag{2.2}$$

$$\dot{N} = \frac{1}{J}(\tau_m\text{-}\tau_c) \tag{2.3}$$

Where:

P_P: is the plenum pressure.

Q: is the mass flow cross compressor.

N: is the angular speed of the impeller.

a_P: is the speed of sound of the gas in the plenum.

V_P: is the volume of the plenum.

Q_t: is the mass flow through the throttle.

A: is the across sectional area of the duct.

L: is the length of the duct.

Ψ**c:** is the important compressor characteristics.

P_{01}: is the ambient pressure.

J: is the moment of inertia of the drive unit.

τ_m: is the drive torque .

τ_c: is the torque on the shaft from the impeller blades.

Remarks:

Eq 2.1 represents the dynamical compressor pressure rise, Eq 2.2 describes the dynamical compressor pressure mass flow and Eq. 2.3 represents the inertial dynamics of the compressor shaft.

The model (Eq 2.1), (Eq 2.3) was first presented in Gravdahl and Egeland (1999), and the whole model is presented in Fink et al (1992) and the two first equations are the model of Greitzer (1976).

2.3.1. Derivation of the model: [9]

a. The driver model:

The torque acts on the impeller blades and from the angular momentum balance it can be written as:

$$\boldsymbol{\tau}_c = Q(r_2 C_{\theta 2} - r_1 C_{\theta 1}) \qquad (2.4)$$

Where r1 and r2 is the radius at the inducer and the impeller exit, respectively, and $C_{\theta 1}$ and $C_{\theta 2}$ is the tangential velocity of the gas at the inducer and the impeller exit, respectively. It is customary to assume that the velocity at the impeller inlet is zero, equation (2.4) becomes:

$$\boldsymbol{\tau}_c = Q r_2 C_{\theta 2} \qquad (2.5)$$

Consider the velocity triangle at the impeller exit in Fig.2.2.

The tangential velocity at the impeller exit can, by trigonometric considerations, be written as:

$$C_{\theta 2} = U_2 - C_{a2}\cot\beta_{2b} = U_2\underbrace{\left(1 - \frac{C_{a2}}{U_2}\cot\beta_{2b}\right)}_{\boldsymbol{\mu}} \qquad (2.6)$$

Where:

μ : is the flow coefficient.

In practice the flow coefficient is less than the ideal value due to slip. The slip factor can be introduced in (2.10) as :

$$C_{\theta 2} = \underbrace{\sigma\left(1 - \frac{C_{a2}}{U_2}\cot\beta_{2b}\right)}_{\boldsymbol{\mu}} U_2 \qquad (2.7)$$

Where:

σ: is the slip factor, typically slightly less than unity.

$\boldsymbol{\beta_{2b}}$: is the rotor blade angle, accounts for rotor blades that are swept forwards or backwards. The function decreases with increasing mass flow for the usual backwards swept solution for blade design where: β_{2b} <90°

For the radial vanes**,** we have:

$\cot \beta_{2b} = 0 \Rightarrow \mu = \sigma$

With

$U_2 = r_2 N$

Then (Eq2.5) becomes:

$$\tau_c = Q r_2^2 \mu N \quad (2.8)$$

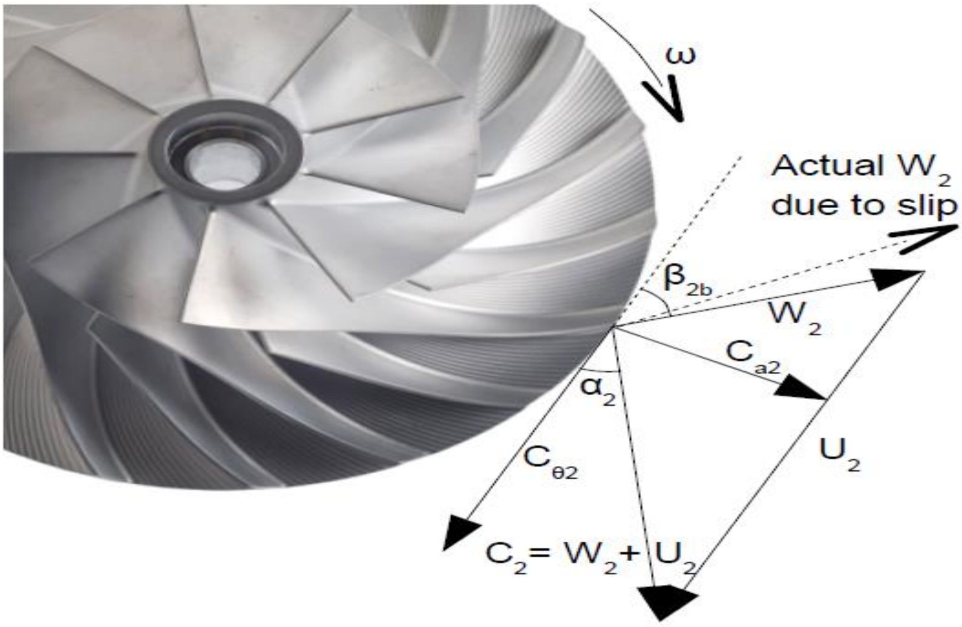


Fig2.2. Velocity triangle at the impeller exit.

In the case of compressor deep surge, we'll have a flow reversal. Finally the torque acting on the impeller blades is:

$$\tau_c = |Q|\, r_2^2 \mu N \quad (2.9)$$

b. The origin of the compressor characteristics:

The compressor characteristic **Ψ** is a property of the compressor and there are multiple ways to express it. One of them, derived from enthalpy transfer in Egeland and Gravdahl (2002), is:

$$\Psi c(Q,N) = \left(1+\frac{r_2^2\mu N^2-\frac{1}{2}r_1^2(N-\alpha Q)^2-K_f Q^2}{a_p T_{01}}\right)^{\frac{k}{k-1}} \quad (2.10)$$

The values in (2.10) are hard to find, and often an approximation based on measurement data is used for **Ψc**. This expression is also valid in the unstable area to the left of the surge line. The constant **α** determines the point of zero incidence loss, for which: $N - \alpha Q = 0$

This constant is given by:

$$\alpha = \frac{\cot\beta_{1b}}{\rho_1\ A_1\ r_1} = \frac{Nsurge\_line}{Qsurge\_line} \quad (2.11)$$

Where:

β_{1b} : The angle of the impeller blade.

c. Throttle valve model:

In order to control the flow through the valve, the orifice area A can be adjusted from 0 to 100%. The throttle flow $\mathbf{Q_t}$ depends on the plenum pressure, and can be modeled as a function of the pressure drop over the throttle and the throttle opening as suggested in Gravdahl (1998).

The final model for the flow through the throttle is:

$$Q_t = k\sqrt{(P_1 - P_2)} \quad (2.12)$$

$$\text{With : } K = CA\sqrt{2\rho} \quad (2.13)$$

Where:

$\mathbf{Q_t}$: Mass flow through the valve.

k: The throttle gain proportional to the opening of the throttle.

$\mathbf{P_1}$, $\mathbf{P_2}$: The pressures before and after the orifice.

C: Flow coefficient.

A: The orifice area.

ρ: Density of the fluid.

For the special case $P_2 < P_1$ in (2.12), the following modification is done:

$$Q_t = \mathrm{sgn}(P_2 - P_1)k\sqrt{|P_2 - P_1|} \quad (2.14)$$

The **sgn** function is not continuous and can be approximated as:

$$\text{sgn}(x)=\lim_{x\to\infty} \tanh \varepsilon x \qquad (2.15)$$

And the absolute value can be approximated as:

$$|x|=x\,\text{sgn}(x)= x \lim_{x\to\infty} \tanh \varepsilon\, x \qquad (2.16)$$

The flow through the throttle is now given as:

$$Q_t=\tanh \varepsilon(P_2 - P_1)\, k_0 \sqrt{|P_2 - P_1| \tanh\left(\varepsilon(P_2 - P_1)\right)} \qquad (2.17)$$

d. Plenum model:

The plenum is modeled by Eq (2.1) :

$$\dot{P}_p = \frac{a^2{}_p}{V_p}(Q - Q_t) \qquad (2.18)$$

2.3.2. Implementation and simulation of the model using Matlab-Simulink:

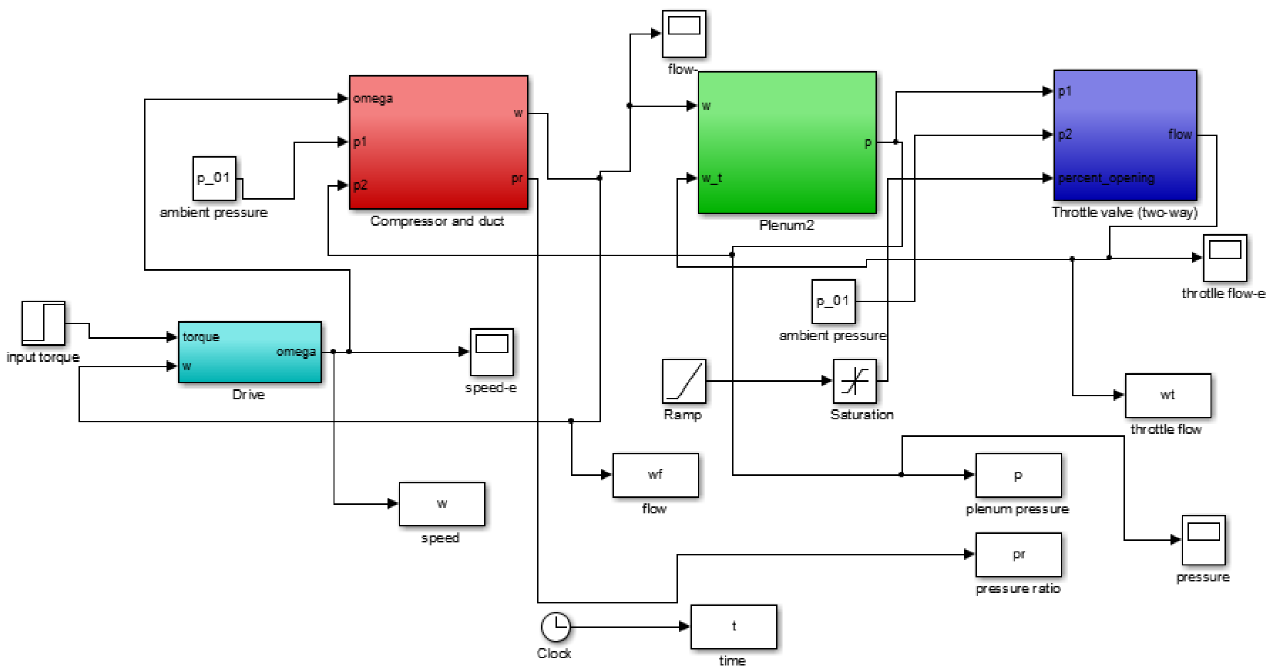


Fig2.3. Simulink implementation of open loop model.

The results of the simulation are shown in the graphs below:

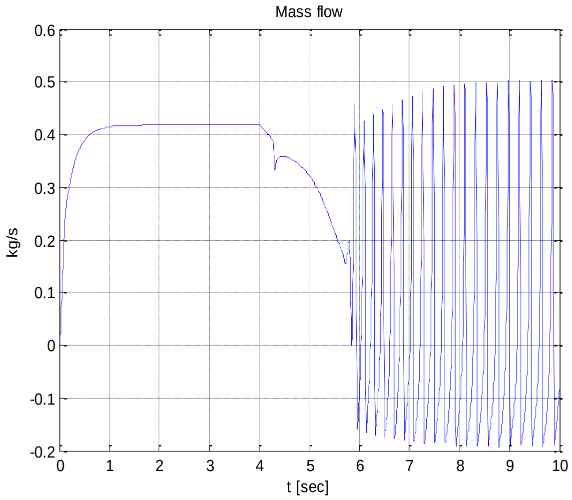


Fig2.4. Flow in the entire compressor.

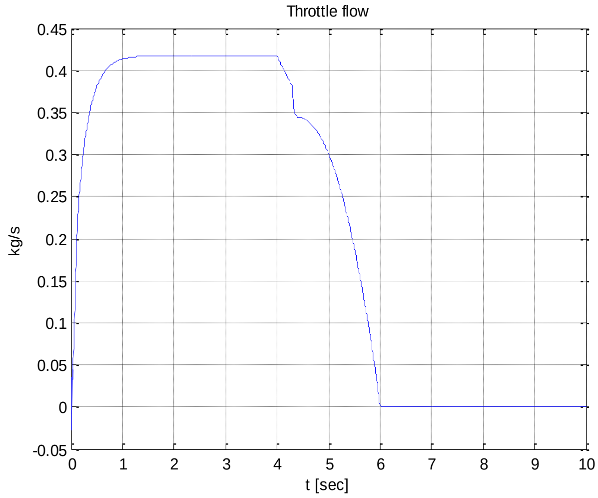


Fig2.5. Flow through throttle.

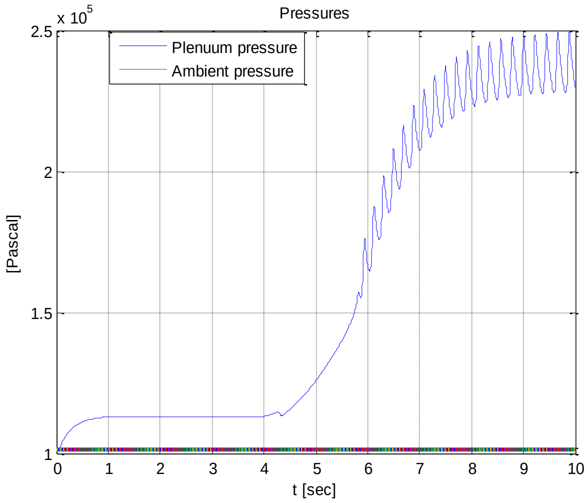


Fig2.6. Ambient and plenum pressures.

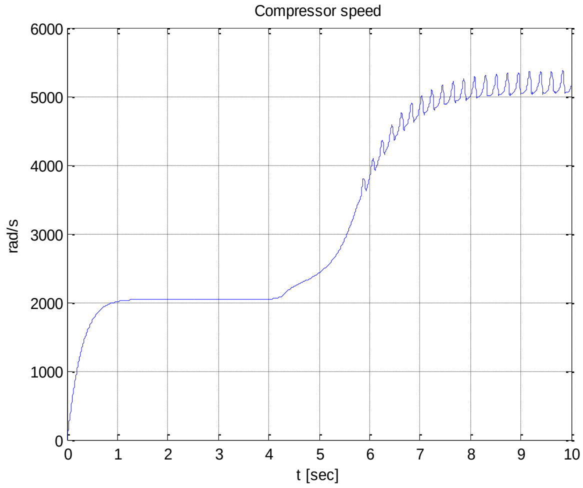


Fig2.7. The compressor Speed (N).

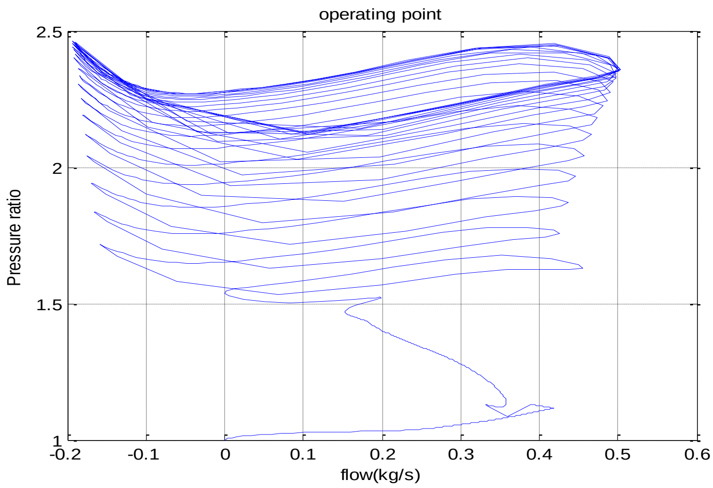


Fig2.8. The compressor operating point.

Discussion:

The **Fig.2.8** shows the behavior of the centrifugal compressor, the system is initially operating at a stable point and the throttle valve is fully open. At t= 4seconds, the throttle valve starts closing, then the operating point of the compressor moves towards lower mass flow rates but it is still stable due to the decrease of the charge on the compressor blades. As the operating point reaches the surge point at t=6seconds, the rotating speed of the compressor and the pressure ratio increase to higher values, the increase in the plenum pressure and speed is due to the small loads applied to the wheels of the compressor; the system then becomes unstable and starts oscillating.

Remark:
Normally we should not use the throttle valve to simulate the compressor flow decrease because it affects the stability of the system and that what will be changed later using the feed flow valve.

2.4. Modeling of recycling compression system: [3]

In the closed loop configuration, the throttle flow is fed back to the compressor via the recycle valve, as shown in figure 2.9.

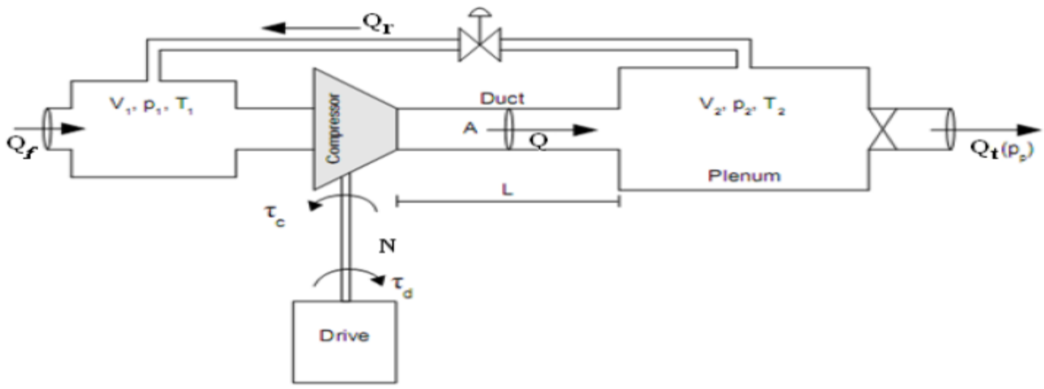


Fig2.9. The recycle compression system.

The application of the mass balance (on the first tank, and the second tank), and the calculation of momentum balance which is applied on the duct and the angular momentum relation applied on the shaft gives: [3]

$$\dot{P}_1 = \frac{a^2 p}{V_1} (Q_f + Q_r - Q) \qquad (2.19)$$

$$\dot{P}_2 = \frac{a^2 p}{V_2}\left(Q - Q_f - Q_t\right) \tag{2.20}$$

$$\dot{Q} = \frac{A}{L}\left(\psi_c(Q, , N)\, P_1 - P_2 \right) \tag{2.21}$$

$$\dot{N} = \frac{1}{J}\left(\tau_m - \tau_c\right) \tag{2.22}$$

The mass flow through the valves, the throttle and the recycle, is modeled as:

Where:

$$Q = c\sqrt{\Delta P} \tag{2.23}$$

c: is a gain proportional to the opening of the valve in the recycle loop.

ΔP: is the pressure drop across the recycle valve.

So:

$$Q_t = \tanh\left(\varepsilon(P_2 - P_{01})\right) c_t \sqrt{|P_2 - P_{01}| \tanh \varepsilon(P_2 - P_1)} \tag{2.24}$$

$$Q_t = c_t\sqrt{P_{Upstream} - P_{01}} \tag{2.25}$$

$$Q_r = c_t\sqrt{(P_1 - P_2)} \tag{2.26}$$

2.4.1 Implementation and simulation of the model in Matlab-Simulink:

The recycling model is implemented in Simulink as shown in **Fig 2.10** .

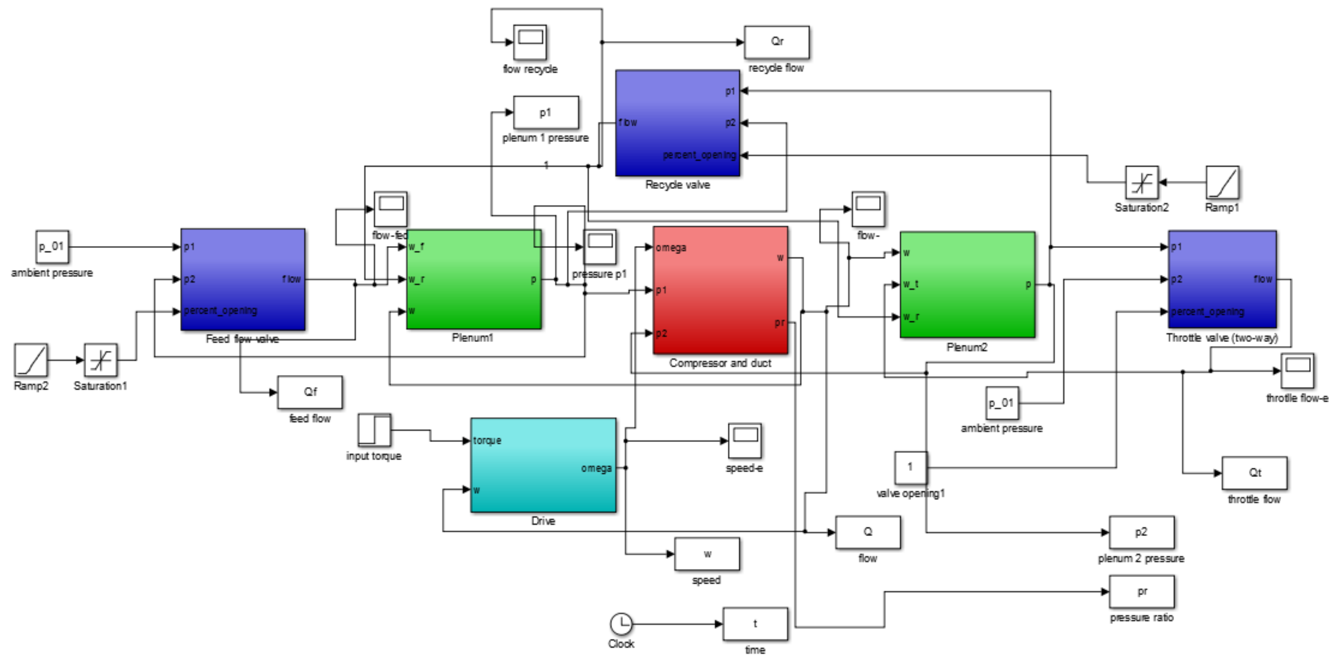


Fig2.10. Compression system with recycle loop in Matlab-Simulink.

The results of simulation are shown in the graphs below:

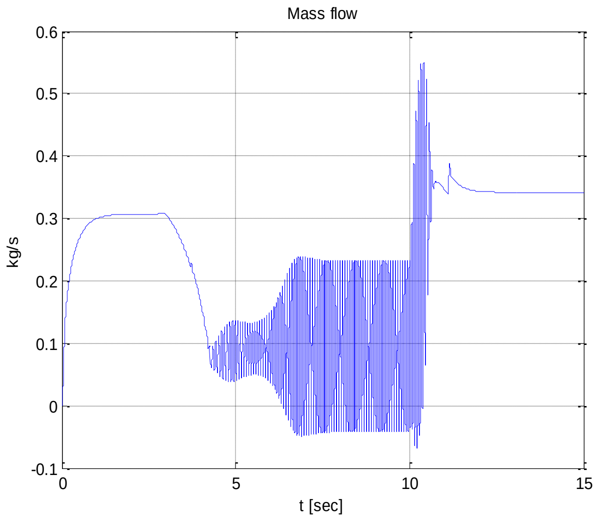


Fig2.11. Mass flow.

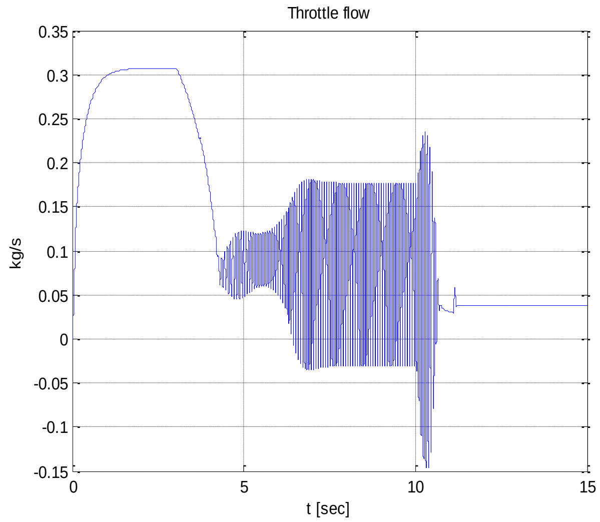


Fig2.12. Flow through throttle.

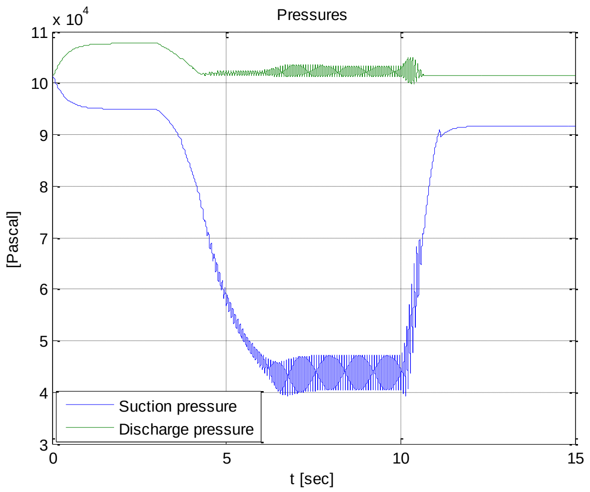


Fig2.13. Suction and discharge pressures.

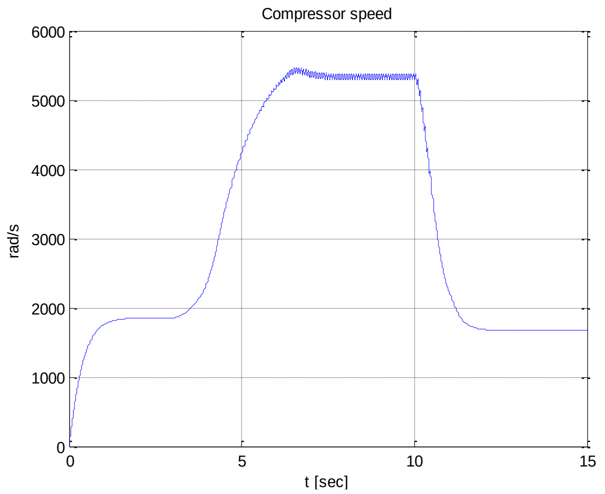


Fig2.14. Compressor speed.

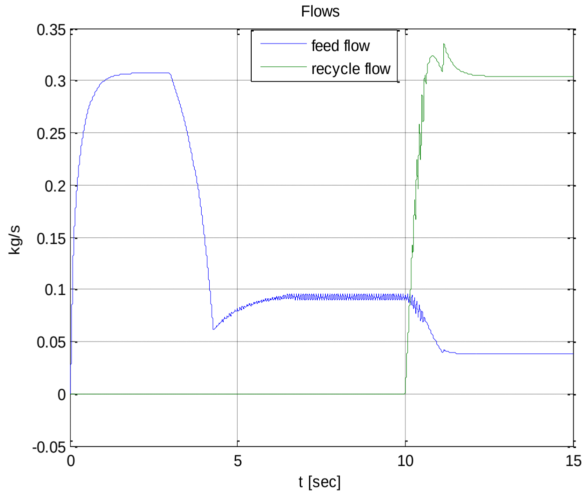


Fig2.15. Feed and Recycle flows.

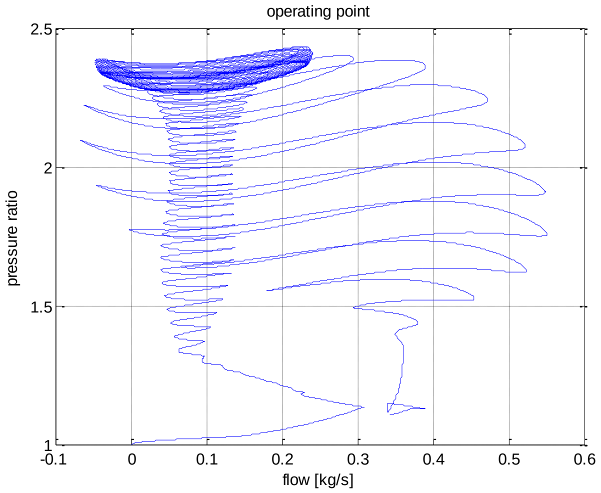


Fig2.16. The compressor operating point.

Discussion:

The simulation shows how the centrifugal compressor behaves when it is operating in both stable and unstable regions. Initially, the compressor speed and the mass flow are set to zero, the pressures in the two volumes are set to ambient pressure while the throttle and feed flow are taken at their initial states.

After starting up the system (simulation), and after its stabilization, the system will pass through two main periods:

- **The surge period**: at time t = 3s, the feed flow starts to decrease gradually creating perturbations on the compressor that eventually cause the system to enter in the unstable region (damping or surge) at time t = 4.5s where the discharge pressure decreases from 10.77 x 10^4 Pa to 10.10 x 10^4 Pa and the suction pressure decreases from 9.495 x 10^4 Pa to 4.414 x 10^4 Pa (**Fig2.13**), the mass flow enters in damping (oscillation), the rotating speed increases from 1855 rad/s to 5400 rad/s (**Fig 2.14**) (due to the small load applied on the impellers) and the operating point oscillates between negative and positive flows (**Fig 2.16**).

- **The stabilization period**: at time t = 10s the recycle valve will be opened manually (**Fig2.15**) to allow more mass flow through the system by compensation of the flow in the suction plenum (**Fig2.10**) stabilizing the system at speed of 1670 rad/s, mass flow of 0.287 kg/s, suction pressure of 9.184 x 10^4Pa and discharge pressure of 10.124 x 10^4Pa, which causes the operating point to move to the right region of positive flow, and the surge disappears.

2.5. Conclusion

The centrifugal compressor enters the instability region whenever the flow is below a certain value where its operating point oscillates between positive and negative values. To overcome this problem, a recycle valve is used to feed the flow back to the intake and maintain it above that value to ensure that the system is operating in the stable region.

From the results of the models using Matlab-simulink, we conclude that the compressors must be controlled and one way is to control the recycle valve since we could stabilize the system manually using that valve when a perturbation is done by decreasing the feed flow.

3.1. Introduction:

One of the major operational problems of centrifugal compressors is the occurrence of surge. This flow instability degrades the performance of the compressor and limits its stable operating range. The most common surge protection mechanism is flow recycling. A recycling valve opens if the operating point of the machine gets close to a predicted surge point to prevent further reduction in flow. This causes a considerable waste of energy, specifically, when the valve has to respond to sudden changes in the flow which usually results in a great deal of overshooting in the recycled flow. Unfortunately, such recycling extracts an economic penalty due to the cost of compressing this extra flow. So the control system must be able to determine accurately how close the compressor is to surging so that it can maintain an adequate but not excessive recycle flow rate. In this chapter we will use the PID controller to control the recycle valve in order to avoid surge.

3.2. PID control of recycle valve:

3.2.1. Principle of a conventional PID controller:

Proportional-Integral-Derivative (PID) controllers have been in existence for nearly two-thirds of a century. They remain a key component in industrial process control as over 90% of today's industrial processes are controlled by PID controllers. It improves the transient response so as to reduce error amplitude with each oscillation and then the output is eventually settled to a final desired value. Better margin of stability is ensured with PID controllers. [8]

PID controller is a control loop feedback mechanism (controller) commonly used in industrial control systems. It continuously calculates an error value as the difference between a desired setpoint and a measured process variable. The controller attempts to minimize the error over time by adjustment of a control variable, such as the position of a control valve. [6]

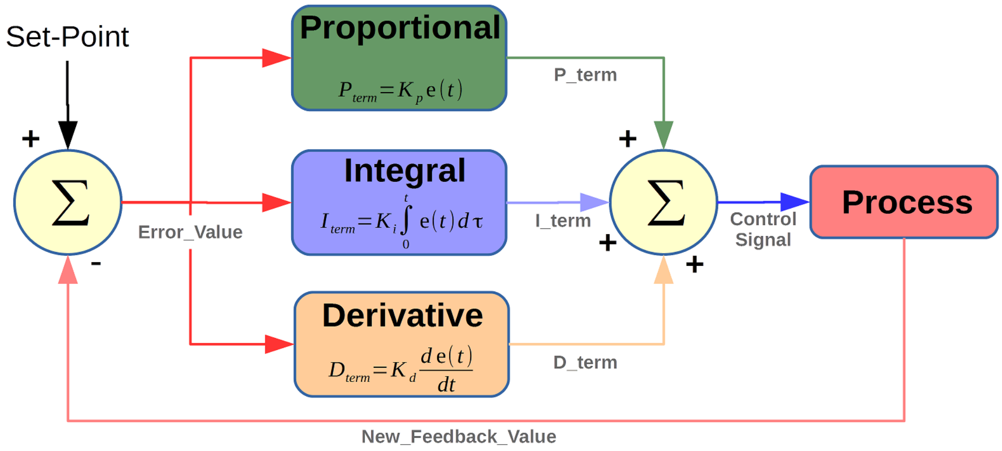


Fig3.1. A block diagram of a PID controller in a feedback loop.[6]

The input of the controller in Fig3.1 is the difference between the reference set point and the measurement signal then the output is presented by the following mathematical equation:

$$u(t) = k_p\, e(t) + k_i \int_0^t e(\tau)d\tau \; + k_d \frac{de(t)}{dt}$$

Where:

- **Error**: e(t) = Setpoint - Plant output
- **K_p, K_i,** and **K_d**, all non-negative, denote the coefficients for the proportional, integral, and derivative terms, respectively (sometimes denoted P, I, and D) .
- **P** depends on the present error.
- **I** on the accumulation of past errors.
- **D** is a prediction of future errors, based on current rate of change.

The weighted sum of these three actions is used to adjust the process via a control element.[9]

3.2.2. PID controller characteristics:

The PID parameters affect the system dynamic. Four major characteristics of the closed-loop step response are evaluated:

- **Rising time**: the time it takes for the control system output to rise beyond 90% of the desired level for the first time.

- **Overshoot**: how much the peak level is higher than the steady state.
- **Settling time**: the time it takes for the system to converge to its steady state.
- **Static (steady-state) error**: the difference between the steady-state output and the desired output.

And the following figure 3.2 gives more explanation:

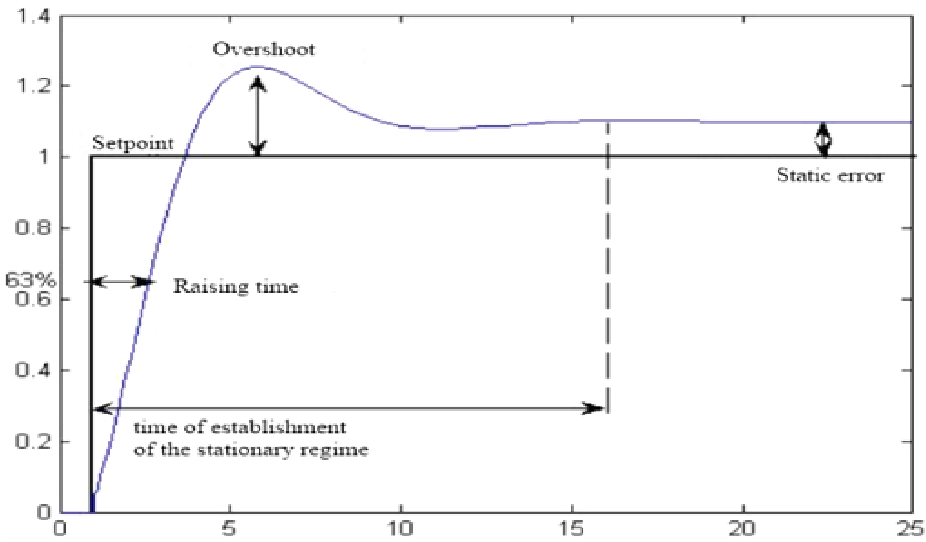


Fig3.2. PID controller characteristics

A proportional controller (K_p) will have the effect of reducing the rise time and will reduce (but never eliminate) the steady- state error. An integral controller (K_i) will have the effect of eliminating the steady- state error, but it may make the transient response worse. A derivative control (K_d) will have the effect of increasing the stability of the system, reducing the overshoot and improving the transient response. Effects of each of controller K_p, K_d and K_i on a closed loop system are summarized in the table shown below. [5]

Parameter	Rise time	Overshoot	Settling time	Steady-state error	Stability
K_p	Decrease	Increase	Small change	Decrease	Degrade
K_i	Decrease	Increase	Increase	Eliminate	Degrade
K_d	Minor change	Decrease	Decrease	No effect in theory	Improve if K_d small

Table 3.1. Effect of Kp,Ki and Kd on the Performance of PID Controller.[6]

Note that these correlations may not be exactly accurate, because Kp, Ki and Kd are dependent of each other. In fact, changing one of these variables can change the effect of the other two. For this reason, the table should only be used as a reference when determining the values for K_p, K_i and K_d. [5]

3.2.3. PID controller regulation and design for recycle compression system:

The only way to prevent surge is to recycle or blow off flow to keep the operating point away from the surge limit. The compressor is protected against surge by an anti-surge valve connecting the compressor's discharge plenum to the suction plenum, thus increasing the flow in the compressor to return it out of the surge area.

When the compressor flow rate is below the specific flow for the protection margin at a given pressure ratio, the controller should send a control signal to the valve to open. The rate of opening or closing should be based on the speed required to protect the compressor.

The surge regulation by the PID controller is intended to open this valve as soon as the flow in the compressor is getting too close to the surge flow. The PID controller acts on the error between the compressor's operating point and the surge control line, and no action is taken until the operating point moves to the left of the control line. The controller needs to be slow for normal operating conditions, but fast when needed to protect the compressor from surge.

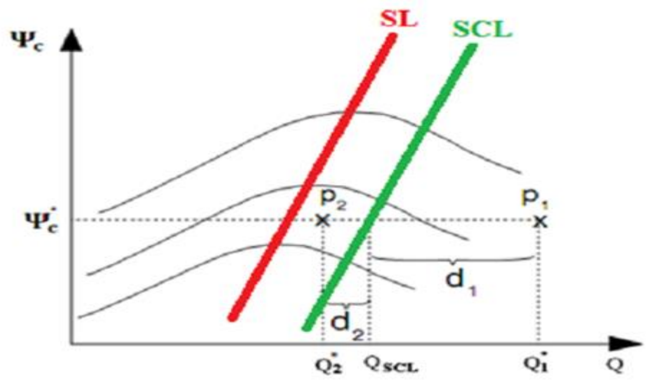


Fig3.3. The PID strategy of a recycle valve.[12]

Figure 3.3 represents the PID controller outline of a recycle valve. The set point tracking (reference) for the PID controller of the recycle valve is the surge control line SCL which is linear with an horizontal surge margin to the surge line .

$$Q_{SCL}(\psi_c) = \frac{\boldsymbol{\psi_c - b}}{\boldsymbol{a}}$$

With:

- **a :** The slope of protection line;
- **b :** The horizontal surge margin to the surge line;
- **ψ_c**: The pressure ratio of the surge control line (SCL);
- **Q_{SCL} :** The flow of the surge control line (SCL).

In fig3.4 , a condition is added to the input Error (e) of the PID controller, such as:

$$\begin{cases} e = 0 & when\ d>0 \\ e = -d & otherwise \end{cases}$$

Where: **d= Q - Q_{SCL}** represents the difference between the actual flow (the operating point) and the flow at the surge control line. [10]

When the distance is positive, the operating point is located to the right of the control line, and nothing should be done, in other words the controller must be off. When the distance is negative, the operating point is located to left of the control line, in this case, its positive value (absolute value) is used as input to the controller to turn it on.

The output of PID controller then is given by:

$$u(t) = k_p\, e(t) + k_i \int_0^t e(\tau)d\tau \; + k_d \frac{de(t)}{dt}$$

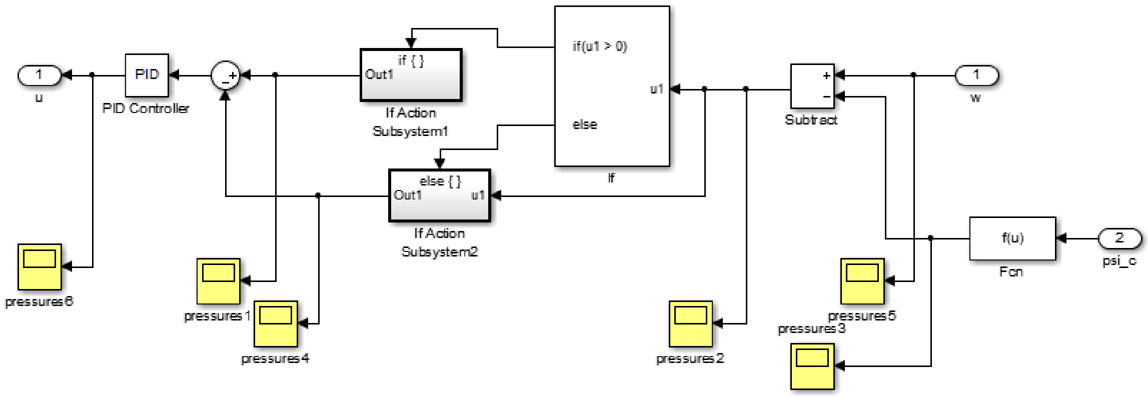


Fig3.4. Scheme of an anti-surge PID regulator.

3.4. Recycle valve control

3.4.1. Implementation and simulation of the recycle Valve Control Using PI Controller using Matlab -Simulink:

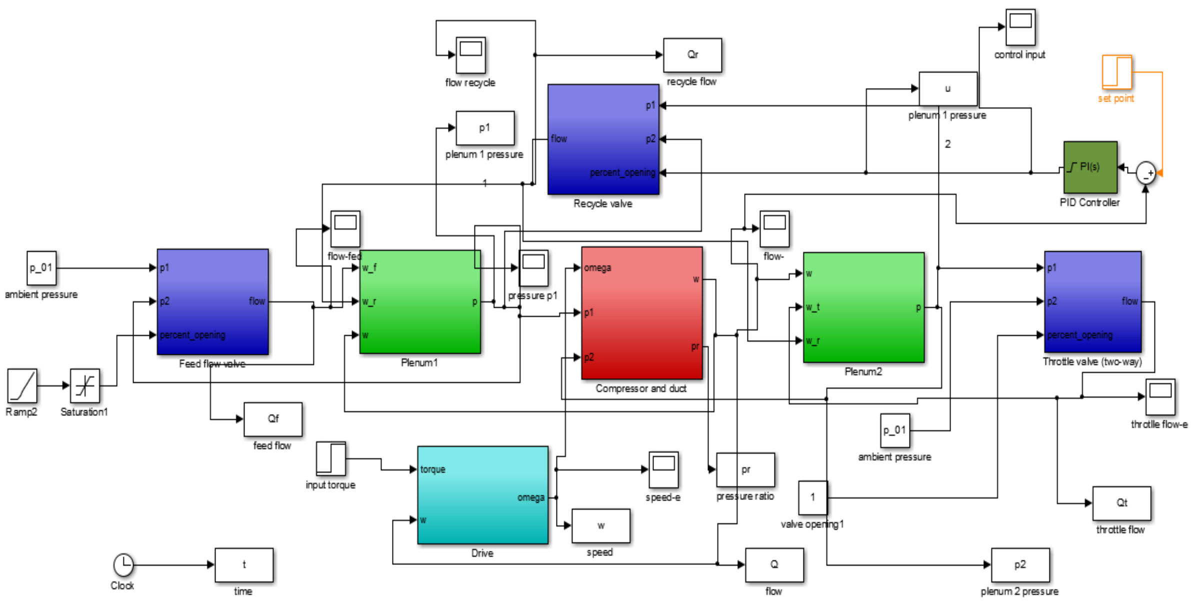


Fig3.5. Simulink implementation of controlling the recycle valve using PI controller.

The recycle valve in a compression system represents an active surge actuator. The aim of using PI controller with limited output between 0 and 1 which is the valve percentage is to control the recycle valve opening and closing during normal and surge operation of the centrifugal compressor. Testing different setpoints, two cases are considered :

- Set point in the **negative slope** region:

Since the system is in stable region, the PI controller will have no problem to control the operating point of the compressor, where the feed flow will be used to push the operating point to the left of the peak line. Figure 3.6 shows the mass flow and the output of the PI control. In this case we let the set point $m_{set} = 0.35kg/sec$ where the maximum efficiency is at the peak given by $m_{eff} = 0.308kg/sec$.

The results of the simulation are shown below:

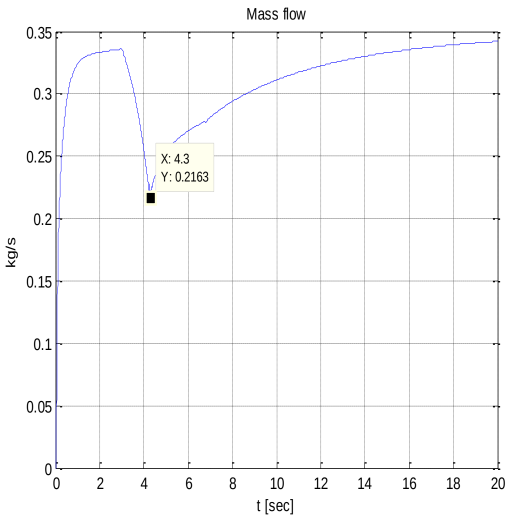


Fig3.6.a. Mass Flow.

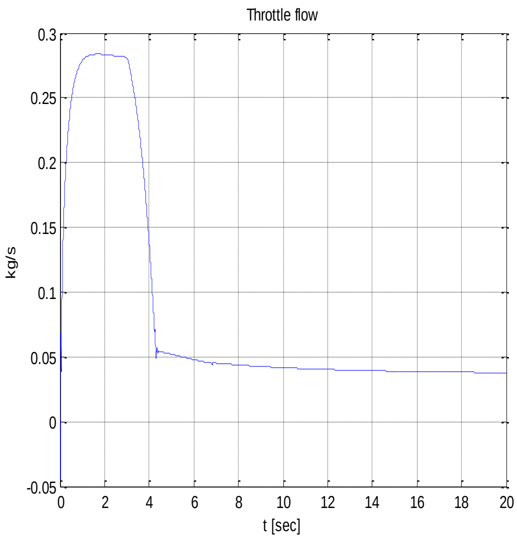


Fig3.6.b. Throttle flow

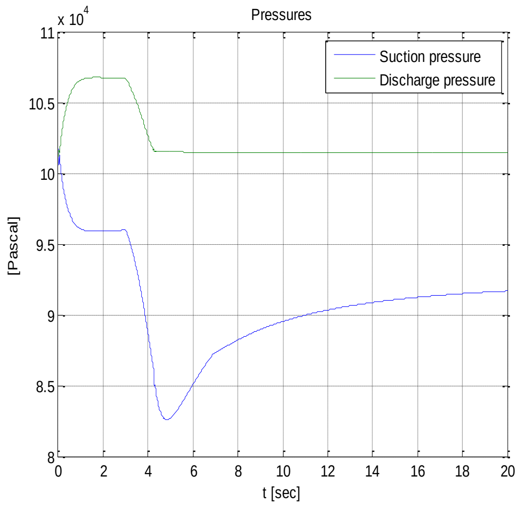


Fig3.6.c. Suction and discharge pressures

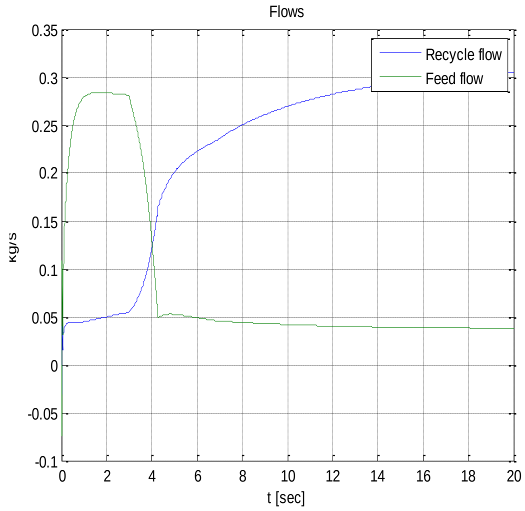


Fig3.6.d. Feed and recycle flows

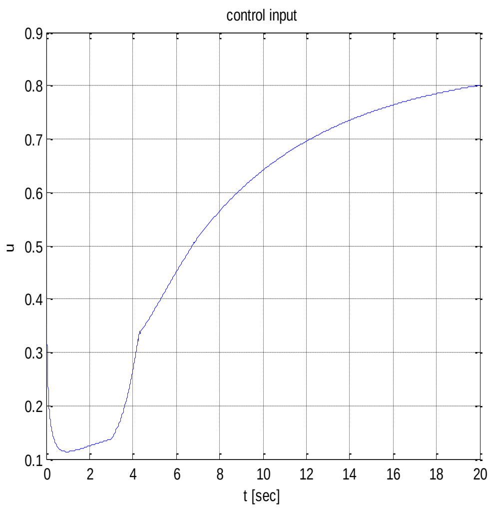


Fig3.6.e. Control input

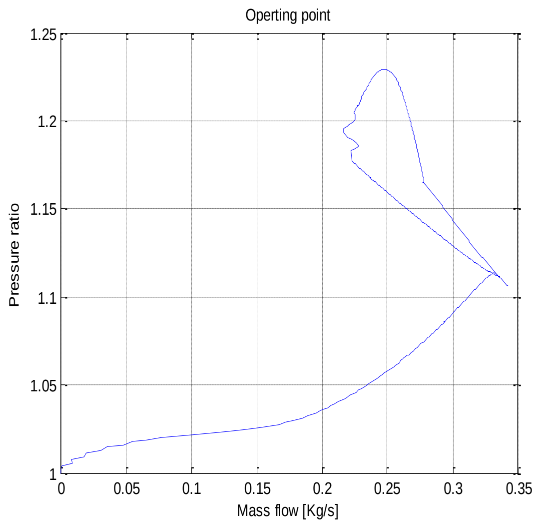


Fig3.6.f. Operating point

Fig3.6. The evolution of the compressor states under PI control where $m_{set} = 0.35\ kg/\sec$

- Set point in the **positive slope** region:

  Since the system is not in the stable region, the PI controller will have to stabilize the compressor to reach the stability region, Figure 3.7 and figure 3.8 show the mass flow and the output of the PI control. Set points from the unstable region are selected $\mathrm{m_{set1}} = 0.25\mathrm{kg/sec}$ and $\mathrm{m_{set2}} = 0.20\mathrm{kg/sec}$

The results of the simulation are shown below :

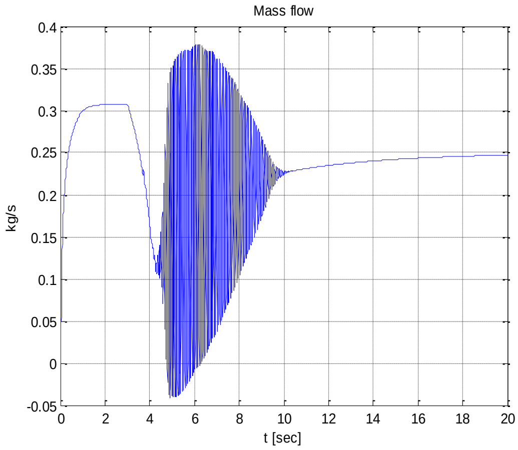


Fig3.7.a. Mass Flow.

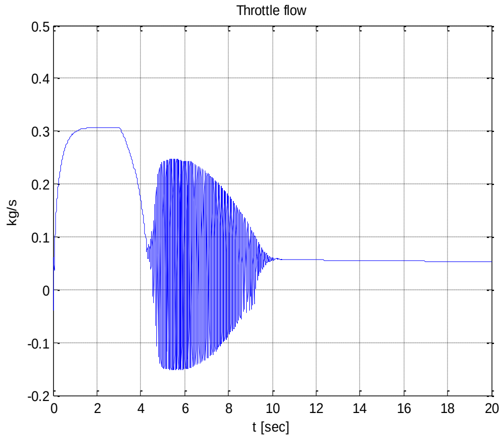


Fig3.7.b. Throttle flow

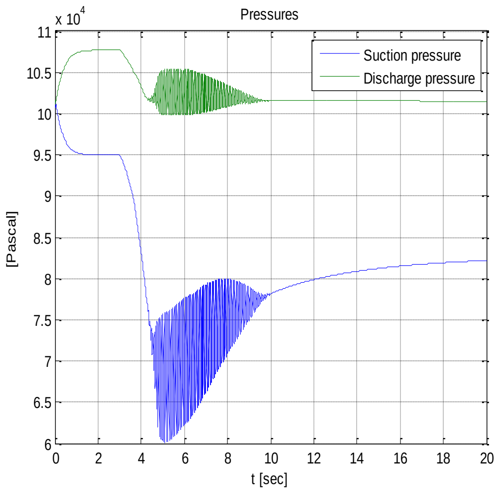


Fig3.7.c. Suction and discharge pressures

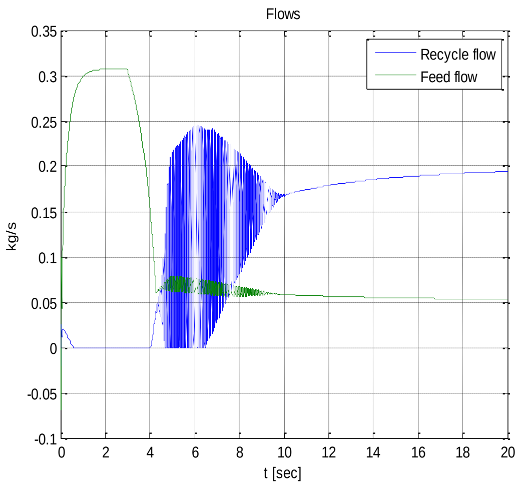


Fig3.7.d. Feed and recycle flows

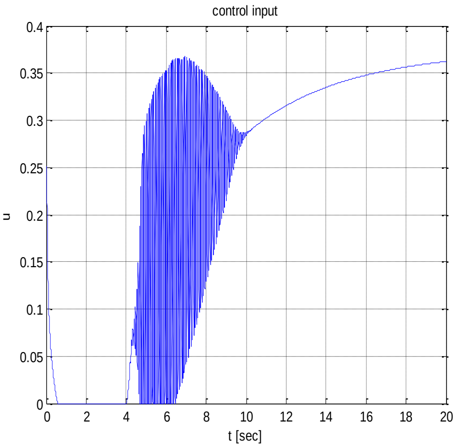


Fig3.7.e. Control input

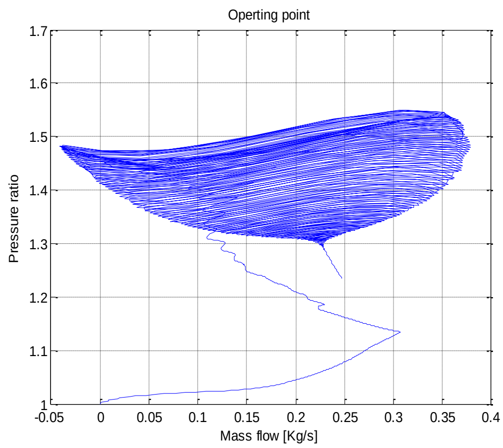


Fig3.7.f. Operating point

Fig3.7. The evolution of the compressor states under PI control where $m_{set} = 0.25\ kg/\sec$

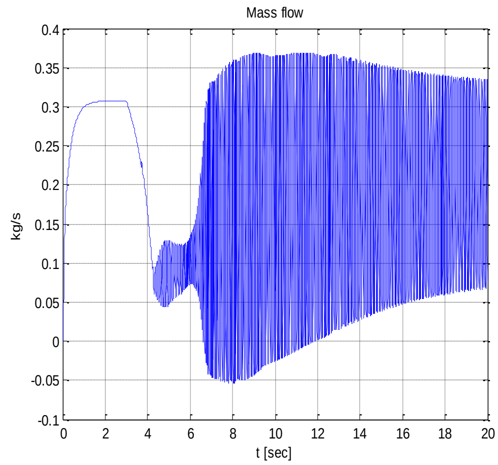


Fig3.8.a. Mass Flow.

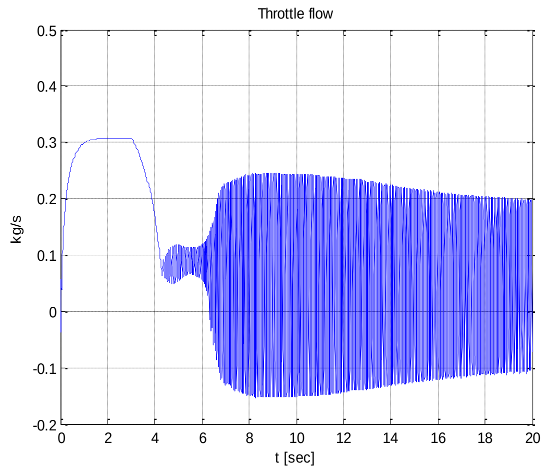


Fig3.8.b. Throttle flow

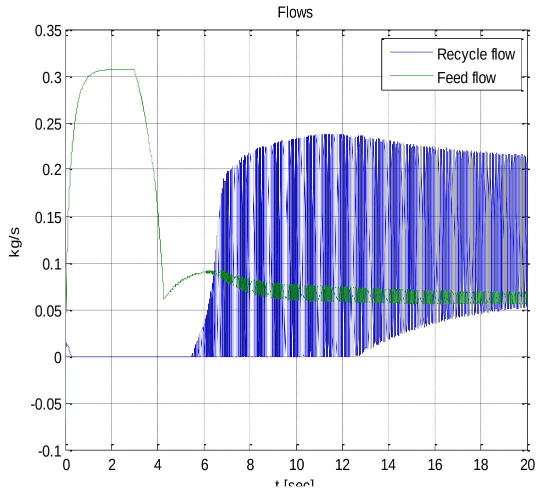


Fig3.8.c. Suction and discharge pressures

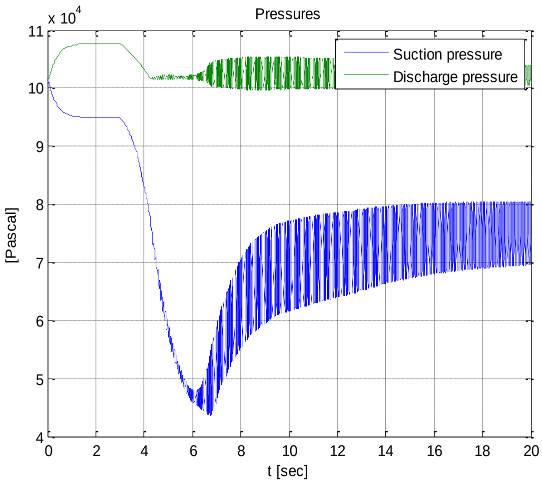


Fig3.8.d. Feed and recycle flows

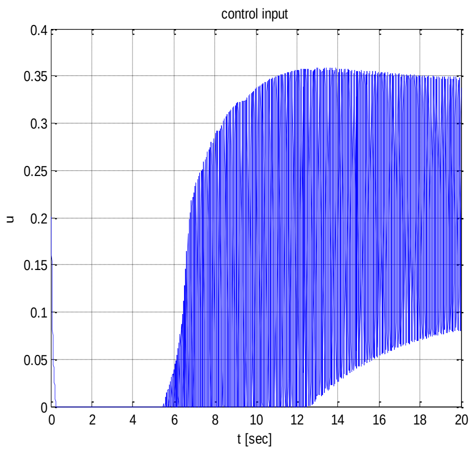


Fig3.8.e. Control input

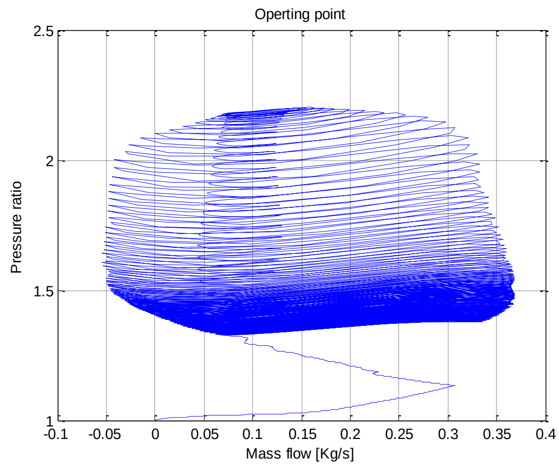


Fig3.8.f. Operating point

Fig3.8. The evolution of the compressor states under PI control where $m_{set} = 0.20\ kg/\sec$

Discussion:

From the PI control figures we can see that the PI controller in the first case has no problem to control since the region of operation is stable. In the second case even the point $m_{set} = 0.25$ kg is in the unstable region after some oscillations, the controller brought the system to stability again however as we decrease the setpoint as in fig 3.8 where m_{set} =0.20 kg/s, the PI has no effect (it can no more stabilize the compressor) because the compressor dynamics are nonlinear.

3.4.2. Implementation and simulation of the recycle Valve Control Using PID Controller using Matlab-Simulink:

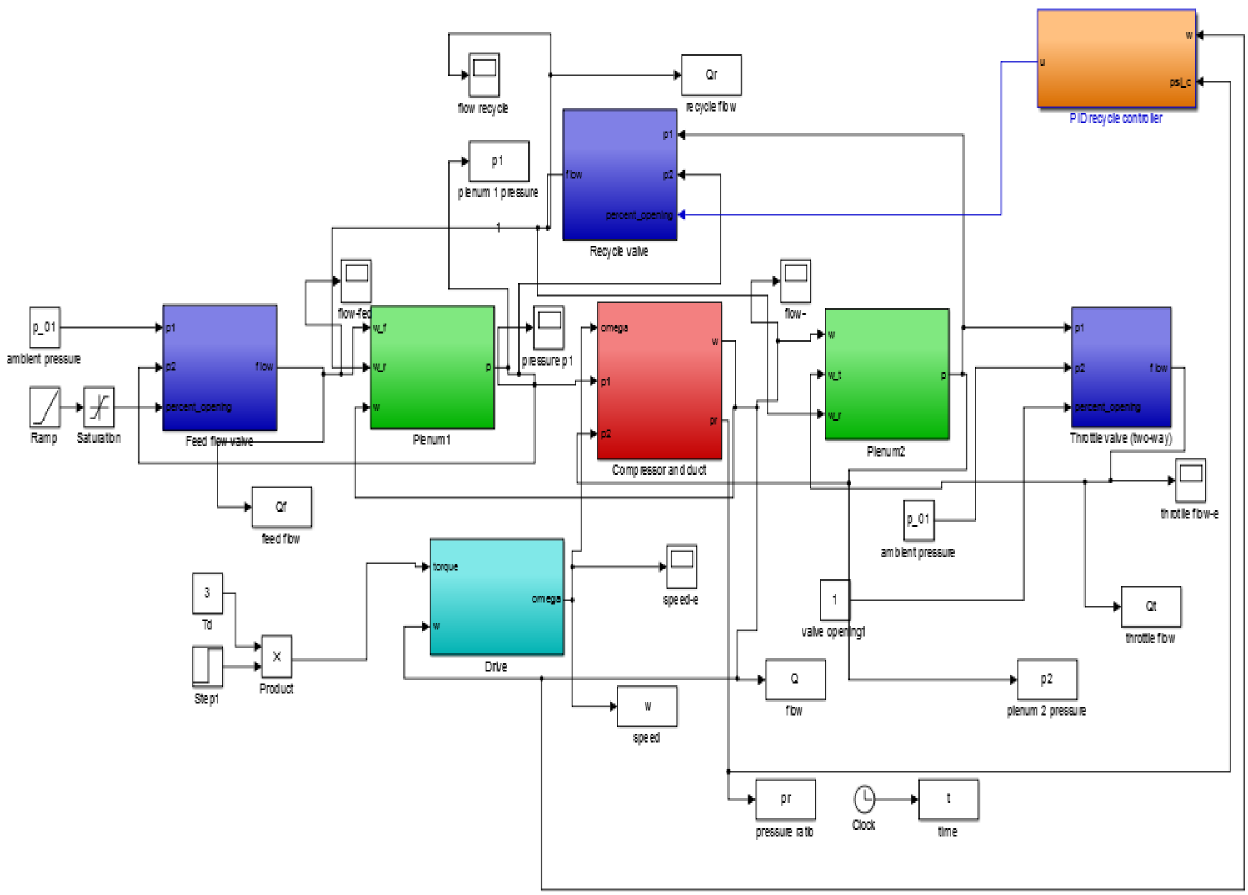


Fig3.9. Simulink implementation of controlling the recycle valve using PID controller.

The results of the simulation are shown below:

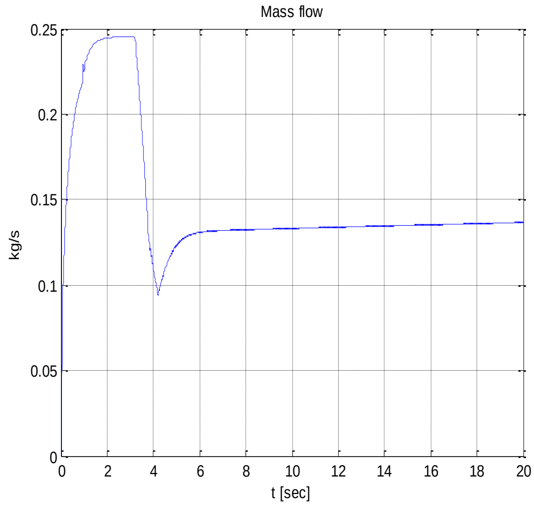


Fig3.10.a. Mass Flow.

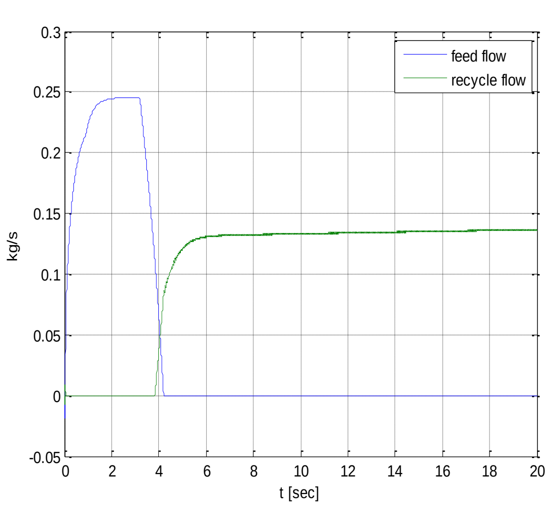


Fig3.10.b. Feed and recycle flows.

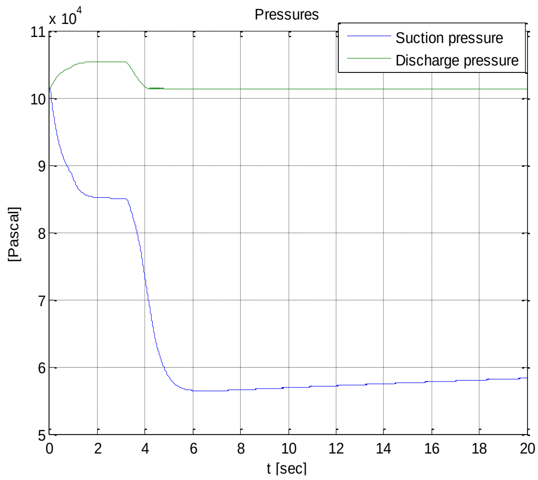


Fig3.10.c. Suction and discharge pressures

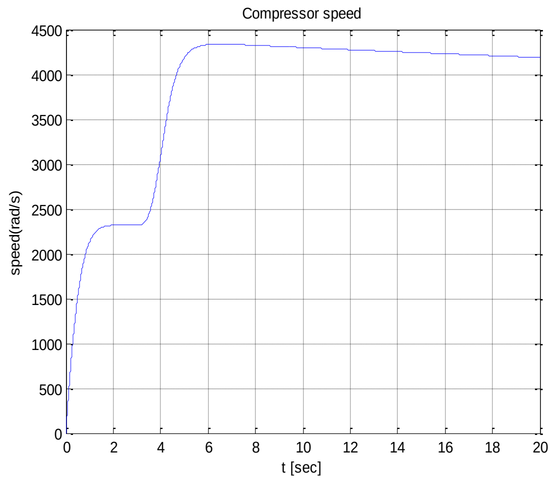


Fig3.10.d. Compressor speed

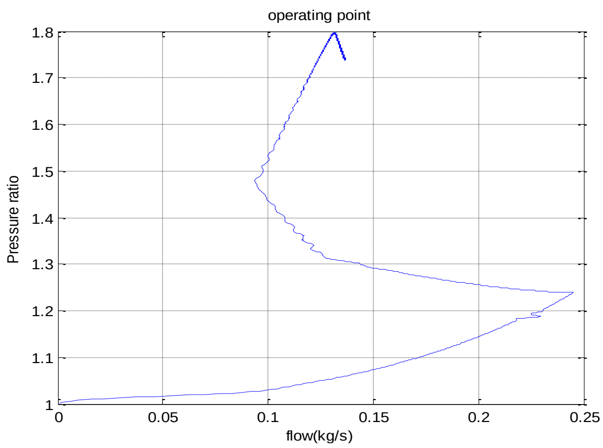


Fig3.10.e. Operating point

Fig3.10. The evolution of the compressor states under PID control

Discussion:

In order to get the influence of the conventional PID controller in controlling the opening of the recycle valve of the compression system and to demonstrate its benefit, the following steps have been used:

First, starting the system and waiting for its stabilization.

Next, closing the feed flow valve at **t** = 3seconds to disturb the system (**Fig 3.10.b**). At this moment the conventional PID controller with $\mathbf{K_p}$= 2.2, $\mathbf{K_i}$= 0.05 and $\mathbf{K_d}$= 0.005 responds and starts the gradual opening of the recycle valve to compensate the flow in the suction plenum in order to stabilize the system.

The reading results are:

- The recycle flow increases at t= 3.82 seconds, from 0 Kg/s to reach 0.1356 Kg/s (**Fig3.10.b**).
- The mass flow decreases to stabilize at 0.1362 kg/s with an undershoot of 11% (**Fig3.10.a**).
- The suction pressure decreases from 8.507 x 10^4 Pa and stabilizes at 5.826 x 10^4 Pa (**Fig3.10.c**). The discharge pressure decreases from 10.54 x 10^4 Pa and stabilizes at 10.13 x 10^4 Pa (**Fig3.10.c**).
- The rotation speed increases from 2327 rad/s at t= 3.205 seconds, and stabilizes at 4333 rad/s (**Fig3.10.d**).

3.5. Conclusion:

In this chapter, we have studied the design and tuning methods of PID controller where appropriate and effective PID parameters have been chosen for our system.

Using MATLAB, a simulation of conventional PI and PID controllers was carried out to control surge instability by controlling the opening of the recycle valve.

To summarize, we can say that there is no advantage to a PI controller over a PID controller other than simplicity; the results of both controllers do not give satisfactory solutions of surge prevention all the time and may have poor control performance for nonlinear or complex systems, that is why the next chapter will cover the regulation of the compression system using the classical regulator PID as well as the intelligent regulators, applying the technique of fuzzy logic.

4.1. Introduction:

Surge is a global nonlinear instability that affects compression systems. It limits the operating range and degrades the system performance. The work introduces a common surge avoidance techniques to extend the operation range by controlling the flow and the speed of the compressor. Because of the complexity and multivariable conditions of the power system, conventional control methods (PID, PI..) may not give satisfactory solutions of surge prevention. So, It is desirable to increase the capability of PID controllers to suit the needs of present day applications. The main focus of the chapter is to develop better methods to design PID controller because of the significant impact of the performance improvement. The chapter proposes a Multi stage fuzzy logic approach to tune the parameters of PID controller online for better system performance. It investigates a Multi stage fuzzy logic control application of a recycle compression system safety.

4.2. Fuzzy logic:

4.2.1. History and applications

The idea of fuzzy logic was invented by Professor L. A. Zadeh of the University of California at Berkeley in 1965. This invention was not well recognized until Dr. E. H. Mamdani, who is a professor at London University, applied the fuzzy logic in a practical application to control an automatic steam engine in 1974. Then, in 1976, Blue Circle Cement and SIRA in Denmark developed an industrial application to control cement kilns, That system began operation in 1982. More and more fuzzy implementations have been reported since the 1980s, including those applications in industrial manufacturing, automatic control, automobile production, banks, hospitals, libraries and academic education. Fuzzy logic techniques have been widely applied in all aspects in today’s society. [13]

4.3. Fuzzy Logic Controller:

4.3.1. Why use fuzzy logic controller?

Unlike classical control strategy, which is a point-to-point control, fuzzy logic control is a range-to-point or range-to-range control. Fuzzy logic control is considered as an alternative way to conventional control; because of its simplicity in the design, its non-dependency on the process model, its flexibility in changing control rule, and its ability to treat with nonlinearity.

Computers can only understand either '0' or '1', and 'HIGH' or 'LOW'. Those data are called crisp or classic data and can be processed by all machines. In order to enable machines to handle vague language input such as 'Somehow Satisfied', the crisp input and output must be converted to linguistic variables with fuzzy components. [14]

4.3.2. Structure of the Fuzzy Logic Controller:

In general, a fuzzy inference system consists of four modules:

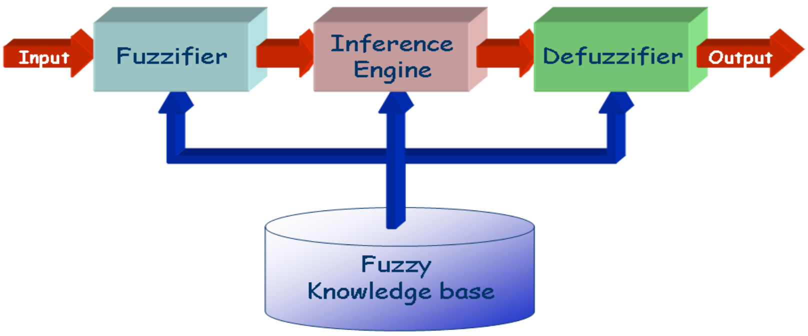


Fig4.1. Overview diagram of a fuzzy system.

The implementation of fuzzy logic technique requires the following three steps:

4.3.2.1 Fuzzification is the process of converting the classical data or crisp data into fuzzy data or Membership Functions (MFs) which have values varying between 0 and 1. This involves two steps: define the linguistic variable of input and their corresponding domain, and specify the membership function that should be used. The shape of the membership depends on the applications; it can be triangular, trapezoidal, rectangular and many other shapes.

(Fuzzy sets and fuzzy operations are given at appendix 01)

4.3.2.2 Fuzzy knowledge base and Inference engine:

The fuzzy knowledge base is the set of fuzzy rules, it stores the knowledge of how best to control the system in the form of IF-THEN rule. The IF-THEN rule describes the relationship between the input variable of the fuzzy controller and the output variable in term of the linguistic variables and fuzzy set.

The inference engine is a software system or computer program that is designed to obtain new knowledge based existing information .It evaluates which rules relevant at the current time and then decides what the input to plant should be; so it performs actually the decision making .

A fuzzy inference system (FIS) is a system that uses fuzzy set theory to map inputs (*features* in the case of fuzzy classification) to outputs (*classes* in the case of fuzzy classification). The most important two types of fuzzy inference system are Mamdani and Sugeno inferences. A FIS tries to formalize the reasoning process of human language by means of fuzzy logic (that is, by building fuzzy IF-THEN rules).

-Mamdani's fuzzy inference system: is the most commonly seen fuzzy methodology. Mamdani's method was among the first control systems built using fuzzy set theory. It was proposed in 1975 by Ebrahim Mamdani as an attempt to control a steam engine and boiler combination by synthesizing a set of linguistic control rules obtained from experienced human operators. Mamdani's effort was based on Lotfi Zadeh's 1973 paper on fuzzy algorithms for complex systems and decision processes.

Mamdani-type inference, as defined for the toolbox, expects the output membership functions to be fuzzy sets. After the aggregation process, there is a fuzzy set for each output variable that needs defuzzification.

A typical rule of Mamdani model is described as follows:

If X is **A** and Y is **B** then Z is **C**.

For example: if X is **high and** Y is **high** then Z **positive large.**

- Sugeno fuzzy inference system: It is quite similar to the Mamdani FIS. The Sugeno system lends itself to the use of adaptive techniques for constructing fuzzy models. These adaptive techniques can be used to customize the membership functions so that the fuzzy system best models the data, the Sugeno output membership is either linear or constant(the output is a crisp function) .

A typical rule in a Sugeno fuzzy model has the form:

If X is **A** and Y is **B** then Z = **f(X,Y)**.

For example: if X is **high and** Y is **high** then Z **= aX + bY +c.**

For a zero-order Sugeno model, the output level *z* is a constant ($a = b = 0$).

4.3.2.3 Defuzzification is the process of converting the fuzzy output signal of the inference engine into crisp output signal that can be used as control signal to the plant of system.

4.4. Fuzzy PID Controller Design:

Fuzzy set theory and fuzzy logic establish the rules of a nonlinear mapping. Because of the complexity and multivariable conditions of the power system, conventional control methods may not give satisfactory solutions. On the other hand, their robustness and reliability make fuzzy controllers useful for solving a wide range of control problems in power systems. In general, the application of fuzzy logic to PID control design can be classified in two major categories according to their manner of construction. [7]

1. A typical FLC is constructed as a set of heuristic control rules, and the control signal is directly deduced from the knowledge base.

2. The gains of the conventional PID controller are tuned online in terms of the knowledge base and fuzzy inference, and then, the conventional PID controller generates the control signal.

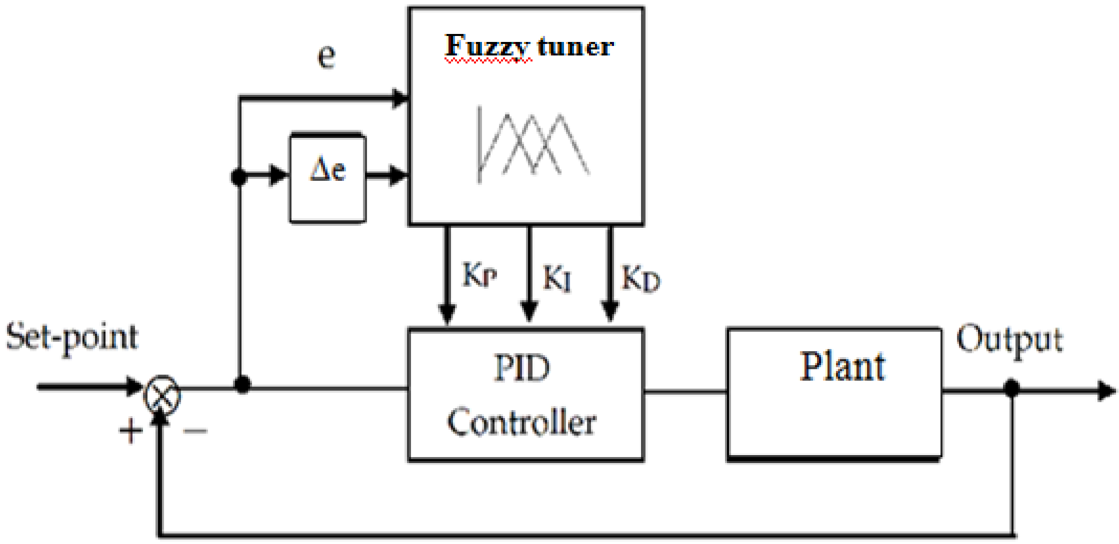


Fig4.2. Block diagram of fuzzy PID controller.

The proposed controller design shown in Figure 4.2 is a two level controller. The first level is a fuzzy network and the second level is a PID controller. The structure of the classical FPID controller in which the PID controller gains are tuned online for each of the control areas.[7]

4.5. Multi Stage PID Fuzzy Logic control of the recycle valve:

4.5.1. The proposed Structure for MSFL PID controller:

The structure of the proposed multi-stage fuzzy logic PID controller is shown in Figure 4.3. It consists of three FLCs, which tune the parameters K_p and K_i and K_d to improve the performance of the system.

These three Fuzzy Logic Controllers in FLPIDC model form the structure of the Multi stage FLPIDC. The simulink model of the designed Multi stage FLPIDC is shown in figure 4.3:

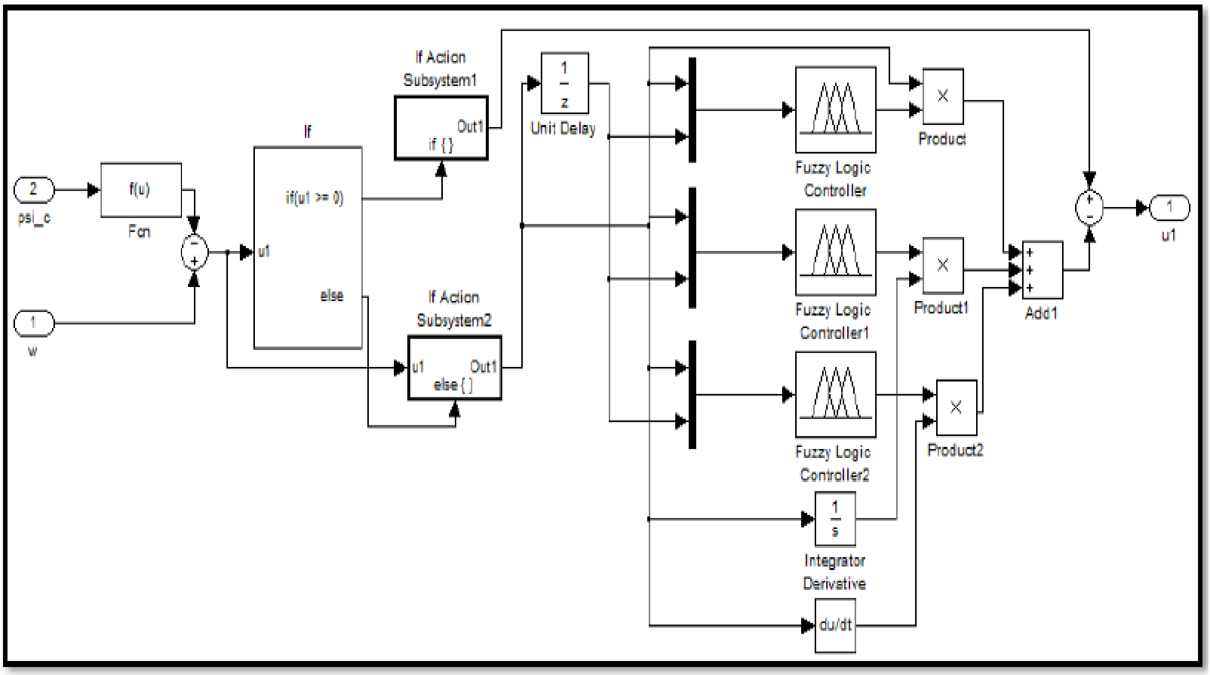


Fig4.3. Simulink model of Multi stage FCPID controller.

4.5.2. Design of Multi Stage Fuzzy Logic PID controller:

The controller is a Mamdani-type fuzzy logic controller with a typical IF-THEN rule structure. Each FLC in the proposed multi-stage fuzzy logic PID controller has two input variables: Error (E) and change in Error (delE) and one output, either K_p, K_i or K_d which are the fuzzy tuned parameters of PID controller. The tuned parameters are a Mamdani fuzzy inference model, which provide a non-linear mapping from error (E) and change in error (delE) to PID parameters K_p, K_i and K_d with seven triangular variable membership functions.[10]

The files kp.fis, ki.fis and kd.fis for the three Fuzzy Logic Controllers are created with fuzzy toolbox of Matlab. The FIS editor of the Kp**.fis** file is shown in figure 4.4.

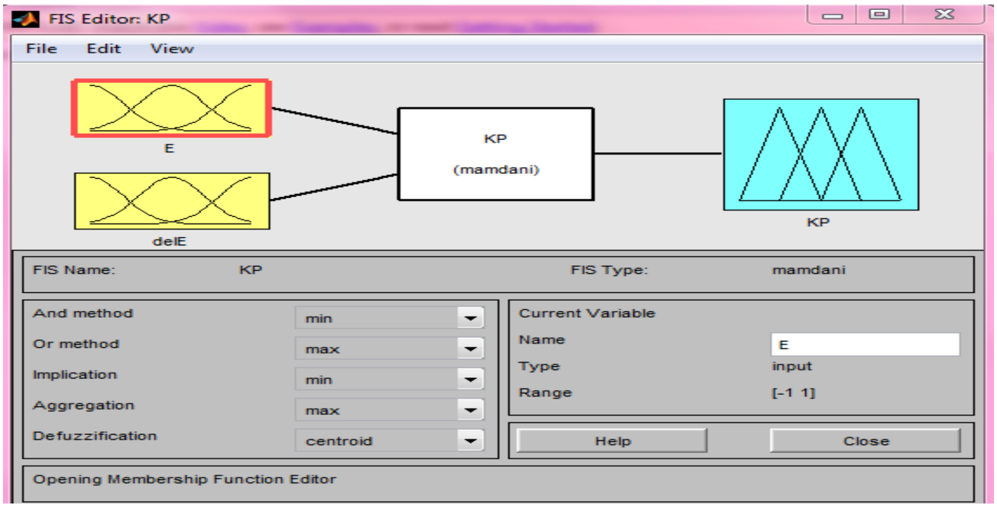


Fig4.4. FIS editor for Kp.fis of MSFLPID controller.

4.5.2.1 Fuzzification:

- ✓ For the inputs, we have defined seven linguistic terms:

  NVL: Negative Very Large.

  NLL: Negative Large Large.

  NL: Negative Large.

  NM: Negative Medium.

  NS: Negative Small.

  NSS: Negative Small Small.

  Z: Zero.

- ✓ For the output, seven linguistic variables are used:

  Z: Zero

  VS: Very Small

  MS: Medium Small

  M: Medium

MB: Medium Big

VB: Very Big

VL: Very Large

The ranges of input variables are from (-1 to 1) for both error (E) and change in error (delE). The ranges of output variables are : (0 to 8) for K_p, (0 to 2.5) for K_i and (0 to 0.4) K_d. The triangular membership functions of input and output variables are shown in Figures **4.5** and **4.6**.

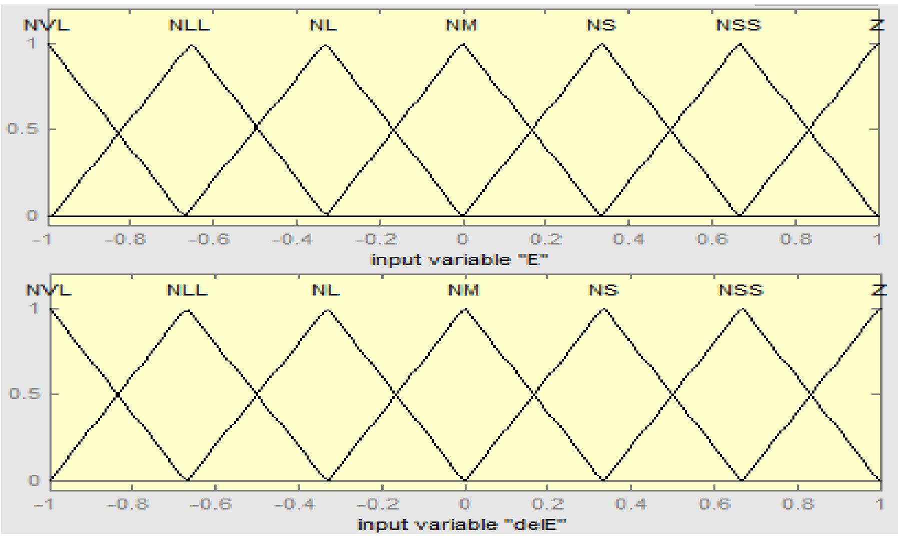


Fig4.5. Input membership function for MSFLPID controller.

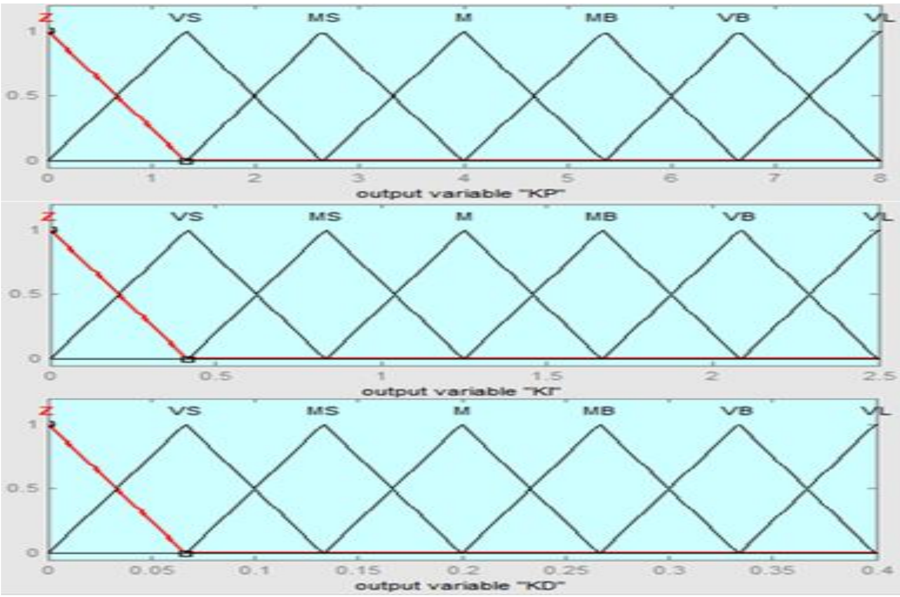


Fig4.6. Output membership function for MSFLPID controller.

4.5.2.2 Fuzzy Control Rules and Fuzzy Inference System:

The fuzzy membership values are used in the rule base in order to execute the related rules so that an output can be generated. A rule base consists of a data table which includes information related to the system.

The fuzzy rule is represented by a sequence of the form IF-THEN, leading to algorithms describing what action or output should be taken in terms of the currently observed information. [14]

The control rules for each fuzzy logic controller are shown in tables **4.1**, **4.2**, **4.3**.

E/delE	NVL	NLL	NL	NM	NS	NSS	Z
NVL	VL	VL	VB	VB	MB	M	M
NLL	VL	VL	VB	MB	MB	M	MS
NL	VB	VB	VB	MB	M	M	MS
NM	VB	VB	MB	M	MS	VS	VS
NS	MB	MB	M	MS	MS	VS	VS
NSS	VS	MB	M	MS	VS	VS	Z
Z	M	M	VS	VS	VS	Z	Z

Table4.1. Rules for K_p of Multi stage FLPIDC.

E/delE	NVL	NLL	NL	NM	NS	NSS	Z
NVL	Z	Z	VS	VS	MS	M	M
NLL	Z	Z	MS	VS	MS	M	M
NL	Z	VS	MS	MS	M	MB	MB
NM	VS	VS	MS	M	MB	VB	VB
NS	VS	MS	M	MB	MB	VB	VL
NSS	M	M	MB	MB	VB	VL	VL
Z	M	M	MB	VB	VB	VL	VL

Table4.2. Rules for K_i of Multi stage FLPIDC.

E/delE	NVL	NLL	NL	NM	NS	NSS	Z
NVL	MB	MS	Z	Z	Z	VS	MB
NLL	MB	MS	Z	VS	VS	MS	M
NL	M	MS	VS	VS	MS	MS	M
NM	M	MS	MS	MS	MS	MS	M
NS	M	M	M	M	M	M	M
NSS	MS	MB	MB	MB	MB	MB	VL
Z	VL	VB	VB	VB	MB	MB	VL

Table4.3. Rules for K_d of Multi stage FLPIDC.

The rows and columns represent two inputs, the flow input and the flow change input, and those inputs are related to IF parts in IF-THEN rules.
The output can be considered as a third dimensional variable that is located at the cross point of each row (flow) and each column (change of flow), and that output is associated with the THEN part in IF-THEN rules.

In our design we have used the Mamdani fuzzy inference type to develop the rules as set of IF –THEN statements:
IF **error** is …and the **change of error** is…THEN the **output** is …

4.6. Simulation and Results :

4.6.1. Implementation and simulation of the model using Matlab-Simulink:

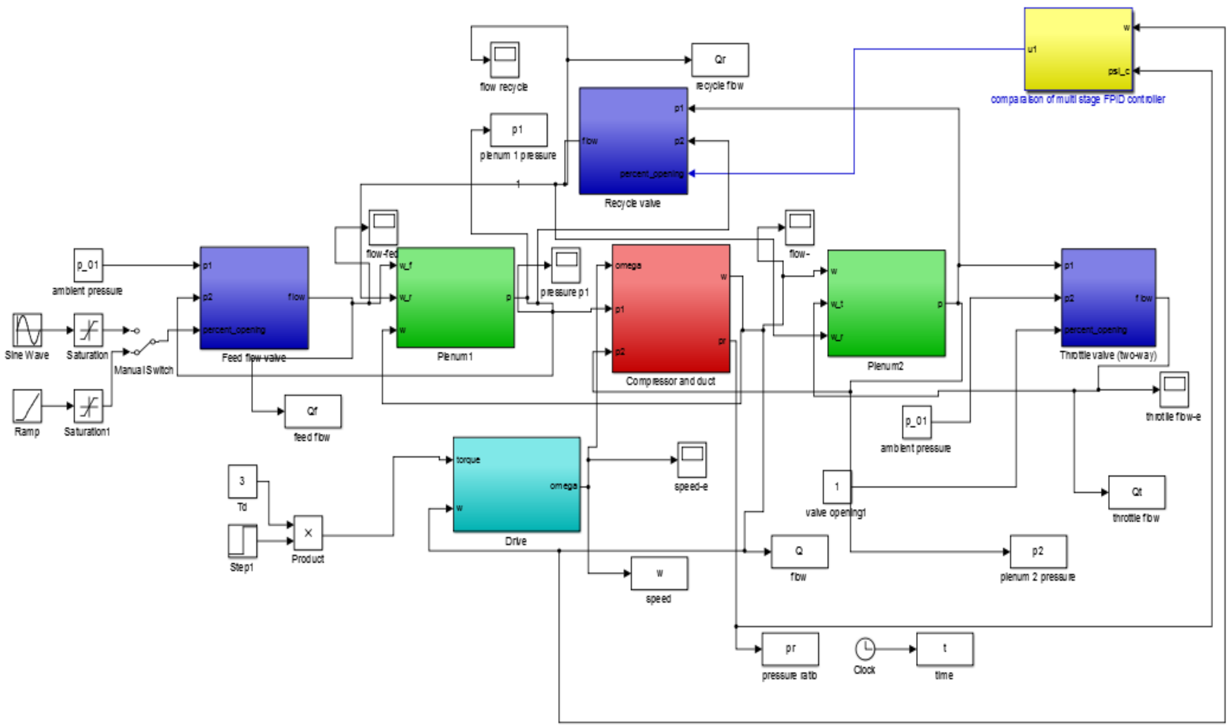


Fig4.7. Simulink implementation of PID and Multi Stage FPID controllers.

The results of Simulation of the closed loop system using a Multi Stage fuzzy logic PID controller for both **ramp** and **periodic** opening of the feed flow valve are the following:

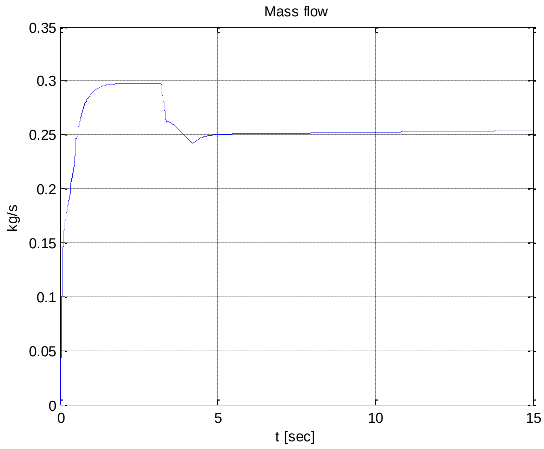


Fig4.8.a. Mass Flow.

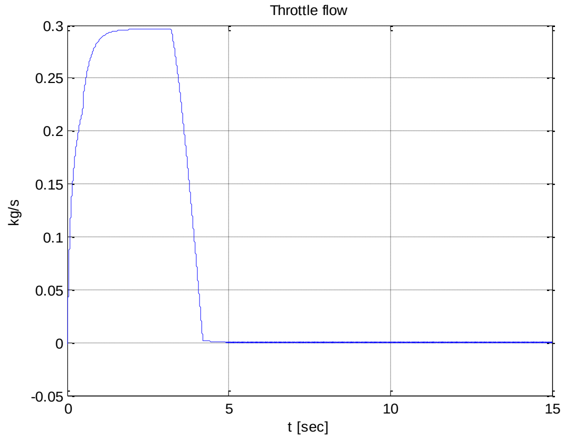


Fig4.8.b. Throttle flow.

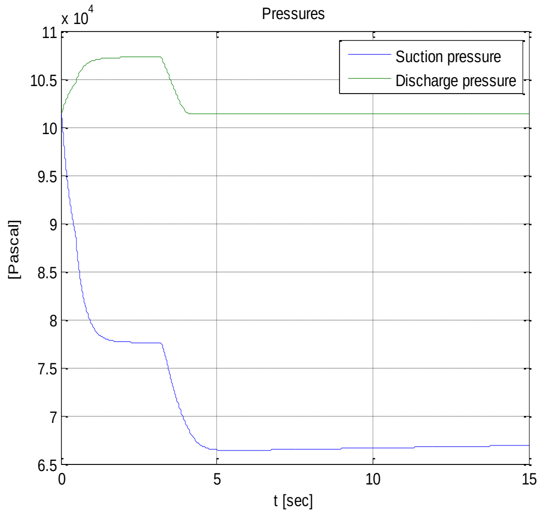


Fig4.8.c. Suction and discharge pressures

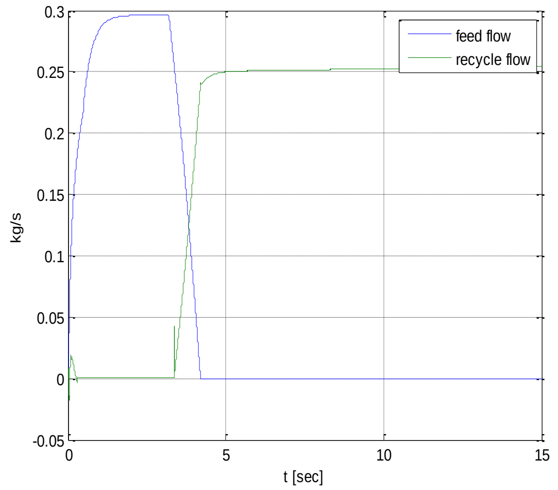


Fig4.8.d. Feed and recycle flows

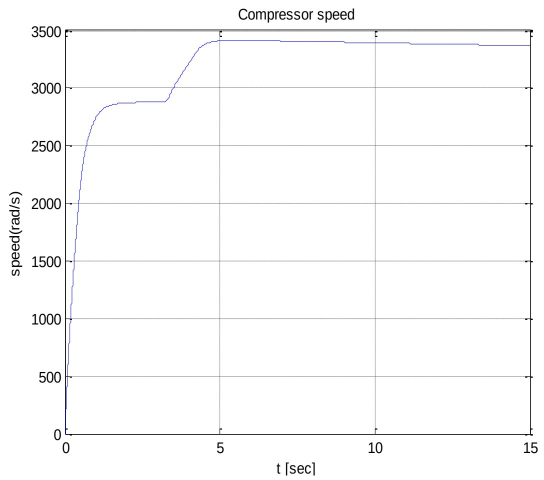


Fig4.8.e. Compressor speed

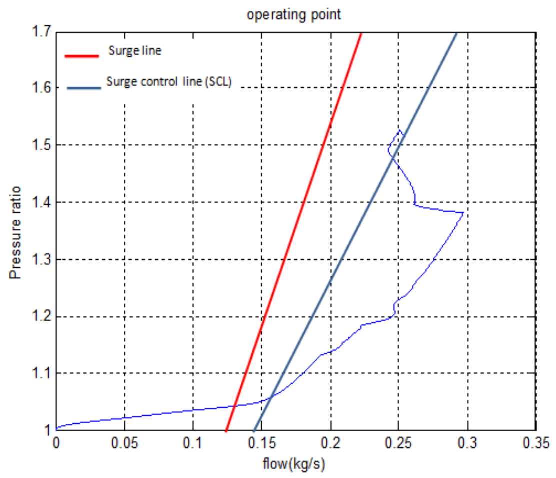


Fig4.8.f. Operating point

Fig4.8. Simulation results of the closed loop system using a Multi Stage fuzzy logic PID controller for ramp opening of the feed flow valve.

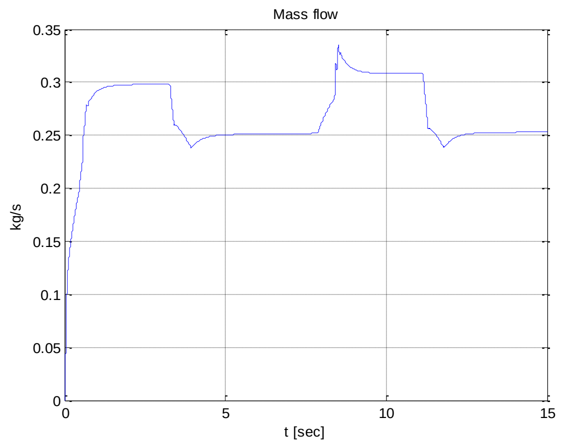


Fig4.9.a. Mass Flow.

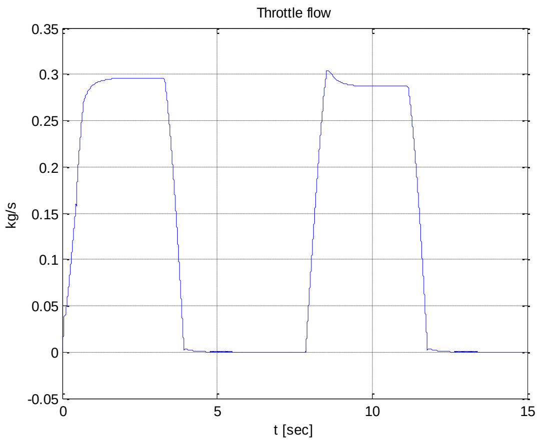


Fig4.9.b. Throttle flow.

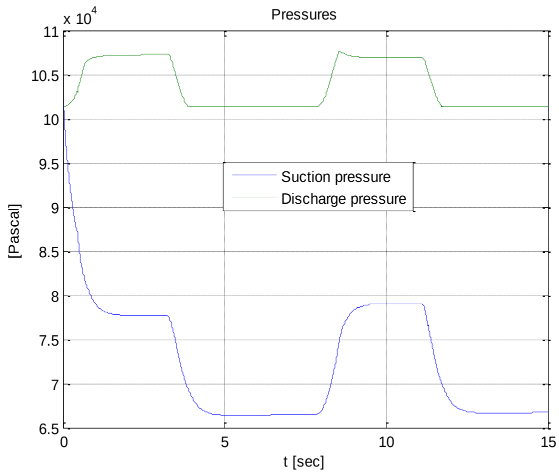


Fig4.9.c. Suction and discharge pressures

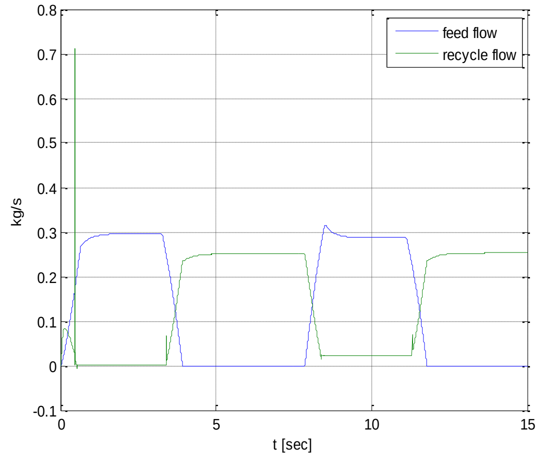


Fig4.9.d. Feed and recycle flows

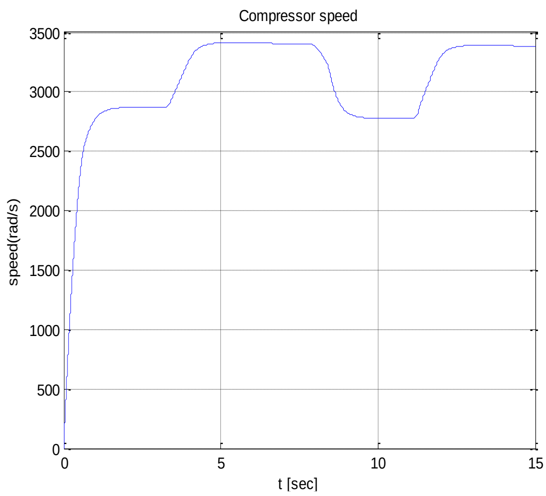


Fig4.9.e. Compressor speed

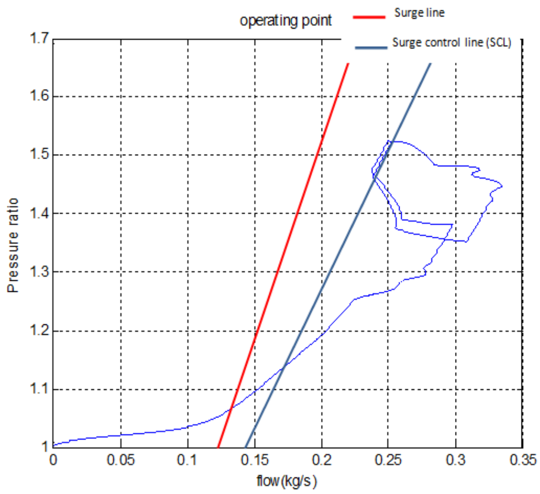


Fig4.9.f. Operating point

Fig4.9. Simulation results of the closed loop system using a Multi Stage Fuzzy PID controller for periodic opening of the feed flow valve.

Discussion:

First, for our compression system a **ramp signal** representing the opening of the feed flow valve of the compressor is the command input of the system. To demonstrate the advantages of using the Multistage Fuzzy PID controller, we have followed these steps:

- First, we start the system then wait until it stabilizes.
- After that, we close the feed flow valve at t = 3 seconds to disturb the system (Fig.4.8.c,d). At this moment the tuned Multi stage fuzzy logic PID controller responds by tuning online automatically so as to have the optimized or the most suitable values of the parameters Kp, Ki and Kd meaning that the recycle valve starts opening gradually (Fig4.8.d) to compensate the flow in the suction plenum in order to stabilize the system.

Then, the same procedure is followed using a **periodic signal** (Fig4.9).

The main remarks in the simulation output graphs are:

- ✓ **For Ramp signal :**
  - The recycle flow increases at t= 3.335 sec, from 0.001025 Kg/s to reach 0.2499 Kg/s at t= 4.999 sec (Fig4.8.d).
  - The mass flow decreases to 0.2502 kg/s at 4.497 seconds.
  - The suction pressure decreases from 7.759 x 10^4Pa at t=3.205sec to 6.671 x 10^4Pa at t=4.631sec (Fig4.8.c).
  - The discharge pressure decreases from 1.07x 10^5 Pa at t=3.259sec to 1.015 x 10^5Pa at t=4.07sec (Fig4.8.c).
  - The rotation speed increases from 2878 rad/s and stabilizes at 3411 rad/s(Fig4.8.e).

- ✓ **For periodic signal :**
  - The speed and the pressure at the discharge plenum of the compression system vary according to the amount of flow across the compressor and the progressive opening of the recycle valve from small open to fast (Fig 4.9.c,e) .

- The recycle flow is increasing when the feed flow is decreasing; and decreasing when the feed flow is increasing (Fig4.9.d) .

For both ramp and periodic signals, the operating point passes over the control line and the controller brings it back to the control line. As observed in Fig4.8.f and Fig 4.9.f, the operating point never reaches the surge line and all the time located to the right of the surge control line except during the starting of the compression process.

Remark:

Comparing the work done in this chapter with the one done in chapter 3 we can say that the conventional PID controller responds correctly to disturbances especially the changes of the feed flow; However, it is hard to obtain optimal tuning for PID controller. Besides, it may have poor control performance for nonlinear or complex systems for which there are no precise mathematical models. This motivates the interest in the Multi Stage fuzzy PID controller which responds in online, faster and with more precision.

4.7. Conclusion:

In this chapter, to exploit the beneficial sides of both PID controller and fuzzy controller and to overcome the above mentioned problems with the two methodologies, the above two controllers have been used for the design of the Multi Stage fuzzy PID that is more accurate; since it plays a significant role at higher load disturbances, by minimizing the deviations of dynamic responses of the hybrid system efficiently and brings the system to stable operation. Besides the proposed controller provides robust control and maintains the system reliable. The Multi stage fuzzy logic technique tunes the PID controller with three fuzzy logic controllers in different stages. Simulation was carried out using MATLAB to get the output response of the system to specific inputs.

Conclusion

The work addressed by this thesis has reviewed some generalities on centrifugal compressors. We discussed also, surge phenomenon, then mathematical models were presented for both open and recycle compression systems. Then, we treated the control of the recycle compression system with many different approaches to avoid this phenomenon.

In the First part, a **manual opening** of the recycle valve is suggested to overcome the surge. In the second part, an **automated opening** for the recycle valve is proposed where new approaches for controlling the nonlinear compressions system were introduced for system stabilization: PI and PID controllers, which are classical approaches based on calculating the “error” value as the difference between the flow of the operating point and the desired flow corresponding to the surge control line and attempts to minimize the error by adjusting their parameters accordingly.

For the PI controller, it needs a huge nominal gain and this high gain is impossible in the anti surge control systems because it would cause instability for the compression system.

For the PID, the results were encouraging for good stability and better performances showing an increase in the mass flow and a reduction in the rotation speed of the compressor's rotor.

Another approach has been addressed which is the **Multi Stage fuzzy logic PID controller:** a key factor that must be considered in the design of surge avoidance systems where both the parameters of membership functions of fuzzy logic controller and PID gains were proposed.

Finally, we conclude that the results show that the controllers disturbance can successfully be handled by both conventional and multi stage fuzzy logic PID controllers. However to keep our compression system stable with a good efficiency according to its robustness a MSFLPID controller that uses online parameters has been improved, and desired targets are achieved compared to the other approaches.

References

[1] Heinz P .Bloch « The Principle, Operation, and Applications of the most Important Compressor Types » Compressor selection basics positive displacement versus dynamic compression(http://www.thecompressedairblog.com)

[2] Gas compressor(https://en.wikipedia.org/wiki/Gas_compressor)

[3] Jan Tommy Gravdahl "Modeling and Control of Surge and Rotating Stall in Compressors" Report 98-6-W, Department of Engineering Cybernetics, Norwegian University of Science and Technology, N-7034 Trondheim, Norway 1998.

[4] Centrifugal Compressor Characteristics Curve (http://www.chemicalengineeringsite.com).

[4] Pooneh Gholami and Samaneh Sadeghi "PID Controller" K.N.Toosi University of Technology,Electrical Engineering Teachers Domain.

[6] https://en.wikipedia.org/wiki/PID_controller

[7] Load Frequency Control of Deregulated Power System Using Different Fuzzy Controllers CH. Malla Reddy , A. Shravan Kumar" International Journal of Science and Research (IJSR) ISSN (Online): 2319-7064 Index (2013) "

[8] Tuning Of Conventional Pid And Fuzzy Logic Controller Using Different Defuzzification Techniques Afshan Ilyas, Shagufta Jahan, Mohammad Ayyub "INTERNATIONAL JOURNAL OF SCIENTIFIC & TECHNOLOGY RESEARCH VOLUME 2, ISSUE 1, JANUARY 2013"

[9] ZAMOUM Razika, Doctoral thesis "Conception d'un Nouveau Système de Régulation Anti pompage des Compresseurs Centrifuges"IHC, University of M'hamed Bougara, June 2013.

[10] "CHAPTER 5 Novel Intelligent Multi Stage Fuzzy Logic PID Controller ".

" http://shodhganga.inflibnet.ac.in:8080/jspui/bitstream/10603/16151/10/10_chapter5.pdf "

References

[11]Single stage centrifugal compressor “ Google “

[12]Sinle stage centrifugal compressor and Major components of a multistage centrifugal compressor “Google”.

[13] Zeyad Assi Obaid, Member, IAENG, Nasri Sulaiman, M. H. Marhaban and M. N.Hamidon, Member, IAENG “Analysis and Performance Evaluation of PD-like FuzzyLogic Controller Design Based on Matlab and FPGA”.

[14] Fundamentals of Fuzzy Logic Control – Fuzzy Sets, Fuzzy Rules and Defuzzifications by Ying Bai and Dali Wang .

Appendix 01:

In this appendix we present fuzzy sets and fuzzy operations :

1. Fuzzy sets :

Fuzzy sets are sets whose elements have degrees of membership. They were introduced by Lotfi A. Zadeh and Dieter Klaua in 1965.

The definition of a fuzzy set then, from Zadeh's paper is:

> Let **X** be a space of points, with a generic element of **X** denoted by ***x***. Thus **X** ={*x*}. A fuzzy set **A** in **X** is characterized by a membership function ***f*A(*x*)** which associates with each point in **X** a real number in the interval **[0,1]**, with the values of ***f*A(*x*)** at ***x*** representing the "grade of membership" of ***x*** in **A**. Thus, the nearer the value of ***f*A(*x*)** to unity, the higher the grade of membership of ***x*** in **A**.

Membership functions for fuzzy sets can be defined in any number of ways as long as they follow the rules of the definition of a fuzzy set. The Shape of the membership function used defines the fuzzy set and so the decision on which type to use is dependant on the purpose. The membership function choice is the subjective aspect of fuzzy logic, it allows the desired values to be interpereted appropriatly. The shape of the membership function is chosen arbitrarily by following the advice of the expert or by statistical studies: sigmoid, hyperbolic, tangent, exponential, Gaussian or any other form can be used.

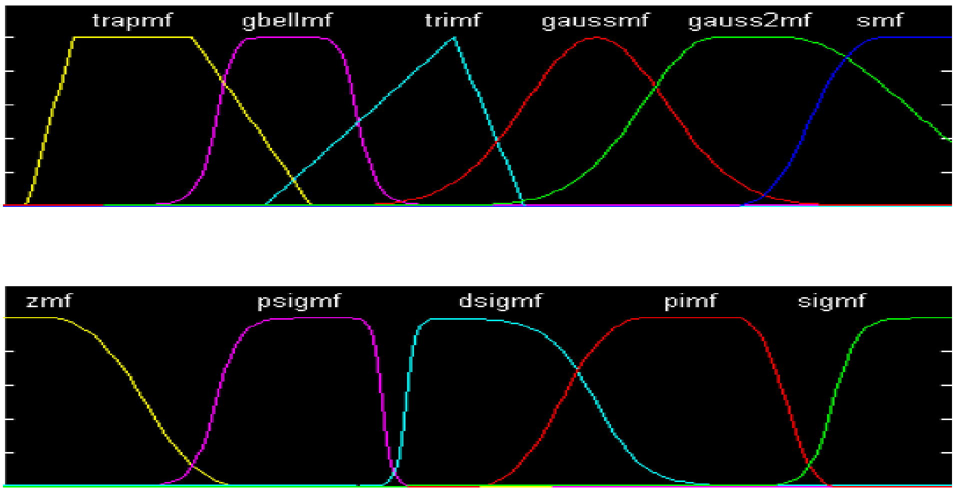


Fig3.1 : The most common membership functions.

2. Fuzzy Operations :

A and B are two sets on the universe X, $\mu_A(x)$ and $\mu_B(x)$ are their membership functions respectively. Then the possible fuzzy operations are :

2.1. Union: The membership function of the Union of two fuzzy sets A and B with membership functions μ_A and μ_B respectively is defined as the maximum of the two individual membership functions. This is called the *maximum* criterion.

$$\mu_{A\cup B}= \max(\mu_A,\mu_B) \tag{1.1}$$

2.2. Intersection (And): The membership function of the Intersection of two fuzzy sets A and B with membership functions μ_A and μ_B respectively is defined as the minimum of the two individual membership functions. This is called the *minimum* criterion.

$$\mu_{A\cap B}= \min(\mu_A,\mu_B) \tag{1.2}$$

2.3. Complement (Not): The membership function of the Complement of a Fuzzy set A with membership function μ_A is defined as the negation of the specified membership function. This is called the *negation* criterion.

$$\mu_{\text{not } A}= 1-\mu_A \tag{1.3}$$