2022

Dynamic Simulation and Anti-Surge System Design of a Demethanizer Compression Unit

Mahshid Zaresharif

Jafar Hooshmand

Dynamic simulation of a demethanizer compression unit

Mahshid Zaresharif [1*], Jafar Hooshmand[2]

1- Dublin City University, School of Mechanical and Manufacturing Engineering, Dublin, Ireland

Mahshid.zaresharif2@mail.dcu.ie

2- HATCO Compression & Separation Co., Tehran, Iran

Hooshmand@hatco.ir

* Corresponding Author

ABSTRACT

Centrifugal turbo-compressor systems are widely applied in refrigeration cycles and industrial chemical manufacturing plants. The design complexity of centrifugal compressor systems includes steady-state operation design and transient behavior analysis. Analyzing transient scenarios such as startup, shutdown, etc., is a significant process design step in safety analyses. The most threatening phenomenon in transient scenarios is the surge phenomenon. This phenomenon may occur due to abnormal circumstances in steady-state operations such as emergency shutdown (ESD), normal shutdown, coast-down operation, and startup. To keep these systems safe from the surge phenomenon, reliable anti-surge systems must be designed, which depends on many design factors in each plant. This study is conducted to build a turbo-compressor dynamic model as realistic as possible, especially in evaluating the compressor control system on its special operation condition in the demethanizer unit. This study is based on two scenarios (ESD and full-recycle) in six operating condition cases. The dynamic simulation results showed that the compressor operates safely and stable during emergency shutdown and full-recycle scenarios in all six operating conditions without a hot bypass valve. Suggested characteristics for the cold anti-surge valve are shown reliable performance to ensure compressor safety, and the anti-surge control system can handle the event from going into surge.

1. INTRODUCTION

There is a high demand for centrifugal turbo-compressor systems in different refrigeration cycles and industrial chemical manufacturing applications. Centrifugal compressor systems design includes steady-state operation design and transient behaviour analysis. Analyzing transient scenarios such as start-up, shutdown, equipment failure, or even mistakes from operators is a significant process design step that must be reviewed to ensure the safe and consistent operation of compressors. While designer and operator experience previously analyzed these critical dynamic scenarios, computational tools and dynamic simulation development provide higher safety margins with lower cost and higher speed.The surge phenomenon is the most important threatening condition that must be regarded in the transient behavior design of compressors. This phenomenon may occur due to abnormal circumstances in steady-state operations such as emergency shutdown (ESD), normal shutdown, coast-down operation, and startup. In pursuit of keeping these systems safe from the surge phenomenon, reliable anti-surge systems must be designed.

The compressor transient flow can be modelled through source terms derived from steady operation characteristics. The first effective efforts in developing a transient model to study surge and rotating stall phenomena were made by Greitzer. This model was based on the steady characteristics of the compressor, which was developed by the lumped-volume method (Greitzer,1976,1980,1986). Developing the lumped volume method for the transient modelling of the compressor continued by Davis with parametric studies for each stage of the compressor (Davis,1987). Botros (1991) studied the more advanced numerical technique was followed by BOTROS et al. l to analyze different transient scenarios: a simulation of the surge protection control process of a compressor, startup and slow transient of a compressor station. The further developer of these numerical models were Morini et al.(2007), who concentrated on the deep surge of compression systems by developing a nonlinear modular dynamic model through a calculated steady-state performance map. To this day, centrifugal compressor instabilities models and anti-surge system designs most commonly utilize Greitzer (1976) and the Moore and Greitzer (1986) (MG model). Furthermore, Botros et al. (2014) developed the mathematical model for the one-dimensional partial solver equations.

Today computational tools and dynamic simulation have been proven the most effective, affordable and reliable methods to design new industrial units. A few studies illustrate models and studies which worked on surge control systems and analysis recently (Kirill et al., 2016, Courtiade and Ottavy, 2013, Zheng and Liu, 2015, Munari et al., 2016, Marelli et al., 2014). Pak et al. (2016) studies can be mentioned as another recent study for analysis of surge control system during the ESD scenario of three CO2 compression systems and one off-gas compression system in different scenarios. Mohajer and Abbasi (2017) also developed a mathematical model for a compressor coupled with an electromotor based on the Greitzer model to evaluate the safety and flexibility of the anti-surge control system. A single nitrogen expansion cycle in a natural gas liquefaction process was also modelled by Zhan et al. (2018).

This study is conducted to build a turbo-compressor dynamic model as realistic as possible, especially in evaluating the compressor control system in a special operation condition. This particular process case is a compression station in the demethanizer unit. The case is described in detail in the next section. The objective of this study was to use a comprehensive model to evaluate the anti-surge control system of centrifugal gas compressors under critical conditions and minatory transient scenarios. The model regards main process components such as compressors and coolers as well as piping and valves. The control philosophy of the compression unit in the demethanizer unit is particularly based on column overhead pressure control.

2. Model Development

The purpose of the compressor station in the demethanizer unit is to boost the pressure of the demethanizer column overhead and send the sales gas to consumers. Besides, the responsibility of keeping the suitable pressure range of column overhead is another important role of the compression unit. Column overhead product is directed to boosters for pre-pressurizing after passing from two heat exchanger units. Afterwards, the methane-rich flow enters the main compressor header. When the suction valve is opened, the gas stream enters the station. Then it is directed through smaller piping segments assigned to compressors. There are three parallel compressors driven by the gas turbines, which have their own recycle lines with the individual cold anti-surge valves. The gas is then directed to the air coolers. After that, the gas passes through the discharge valve and is sent to the train header. Finally, the gas is directed to trim the cooler and enter the main header to send for consumers.

As mentioned above, an individual cold recycles valve directs the gas stream from the air cooler outlet to the compressor inlet. In this study, one of the compressors with maximum recycle volume is selected as the worst case to analyze. On the other hand, we should regard the arrangement of compressors to analyze the impact of compressor trips on the adjacent compressors. Since the distance of compressor suction to the header in all compressors is almost the same, max recycle volume is regarded as a benchmark of selection. The general layout of the process is illustrated in Figure 1.

This project has been developed to build an all-inclusive model based on all reliability-required aspects, including gas turbine module, volume and piping dynamic, heat transfer dynamic, shaft dynamic, actuator dynamic and fuel dynamic. The model of logic controls, including all sequences, governors, and the protective system, has also been exerted. In this study, a FORTRAN code has built a promoted modular computational model. The volume dynamic method is implemented to solve basic required equations, including conservation of mass, momentum, and energy for each control volume. Each compressor train includes nine sub-volumes which are presented in Figure 2 with the Vn tag which V represents sub-volume and n is the number of sub-volumes. The dynamic behaviour of the train results from interacting data between the sub-volumes nodes, and finally, the compression unit dynamic model is completed by connecting three turbo-compressors thorough implementing the effect of the main headers volume dynamic. All required initial values to use in the set of ordinary differential equations are provided by a FORTRAN IMSL subroutine which uses the fourth-order of Runge-Kutta pairs called IVPRK. The exclusive control design of the compression unit in the demethanizer unit is implemented, analyzed and tuned. In the following, the model presumptions and input details presenting the accuracy and complexity of the model equation system are described.

- Modified Redlich–Kwong equation of state is used as the thermodynamic basis for the gas compressor model.
- Piping system volume detail has been accurately used on the model based on isometric documents. To increase the accuracy of holding volume equation solution, piping meshing sensitive analysis has been done.
- The accurate compressor flow maps, which are polytropic head and polytropic efficiency versus actual flow rate, including a factory test verified surge control line and also all of the required information for gas compressor modelling including flow conditions such as inlet and outlet pressure, temperature, flow rate, compressibility factor, composition and other mechanical data like design speed, power, pressure ratio are imported.
- The expected volumes at the inlet and discharge points of the centrifugal compressor and gas composition (density) range have been regarded.

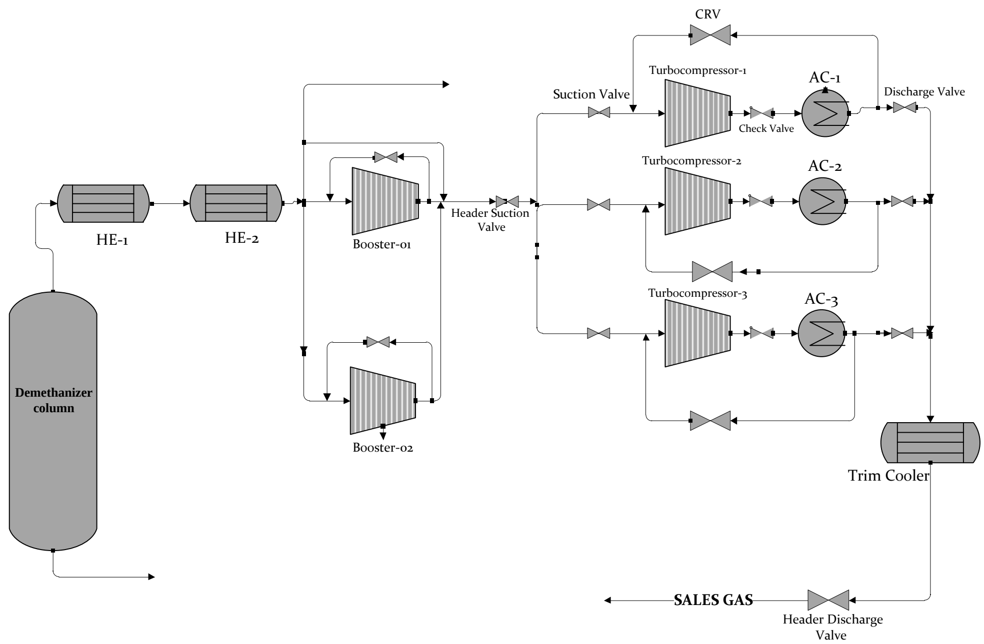

Figure 1-General layout of case study process

- Energy balance equations are utilized in this study with the continuity and momentum conservation equations.
- Compressor specification including impeller specification, friction loss factor and rotating inertia of the driver component, which is mechanically connected to the load, has been regarded. According to compressor documents, the overall moment of inertia is 53.09 kg.m2.
- General modelling of air cooler including fan specification, air heat transfer coefficient, and holding volume regarding cooler geometry has been implemented.
- Comprehensive gas turbine modelling has been carried out in normal and emergency conditions, including control system response in the transient condition (turbine unloading and the inherent time lag between the fuel shutoff and the control valve).
- The location of the check valve and estimation of valve characteristics such as time to open and time to close is assessed.
- Other valve characteristics, including stroke and additional dead times, are implemented.
- Anti-surge valve specification, including realistic tuning/configuration parameters for the surge control system, particularly values for step opening of the anti-surge valve, has been regarded.
- Heat loss to ambient is also configured in the model.
- For modelling elbows and reducers, their equivalent length has been regarded.
- First order response has been regarded as the control behaviour of On/Off valves. Valves opening and closing follow a specific ramp.
- The dynamic model contains pressure specifications in all boundary streams.
- No ancillary piping or equipment has been modelled, including control valve bypass assemblies, drain lines, vent lines, etc.
- Time step of 0.03s results from time step sensitive analysis.
- The philosophy of speed control is based on overall suction flow control. A Cascade control loop is modelled to control the speed via flow control set point. Compressor speed rate limiter of 10rpm/s is regarded.

2.1. GAS COMPRESSOR MODELING

Compressor modelling is the main part of the simulation, which is developed based on Greitzer's compressor model, and the geometric parameters are achieved by an actual turbo-compressor (Greitzer, 1976 & Moore and Greitzer, 1986).

According to the Gritzer model, the model of the centrifugal compression system is used presented in Figure 3. It is shown by a duct where the compressor discharges flow in a larger volume (plenum). The compressed fluid flows via the plenum through the throttle and control valve into the atmosphere. To describe the dynamic behaviour of the examined compression system, the Greitzer lumped parameter model is used; this model was initially designed for axial compressors and was proven to be usable for centrifugal compressors (Hansen et al., 1981). The following assumptions are made in reference: 1) the flow in the ducts is one-dimensional and incompressible 2) in the plenum, the pressure is uniformly distributed, and the gas velocity is neglected 3) the temperature ratio of the plenum and ambient is assumed to be near unity: therefore an energy balance is not required 4) the influence of the rotor speed variations on the system behaviour is neglected The dimensionless mass-flow φ, dimensionless pressure difference Ψ and dimensionless time $\tilde{t}$ are defined as:

$$\varphi = \frac{\dot{m}}{\rho_a A_c U_t} \qquad \Psi = \frac{\Delta P}{\frac{1}{2}\rho_a U_t^2} \qquad \tilde{t} = t\omega_H \tag{1}$$

with the Helmholtz frequency:

$$\omega_H = a\sqrt{\frac{A_c}{V_P L_C}} \tag{2}$$

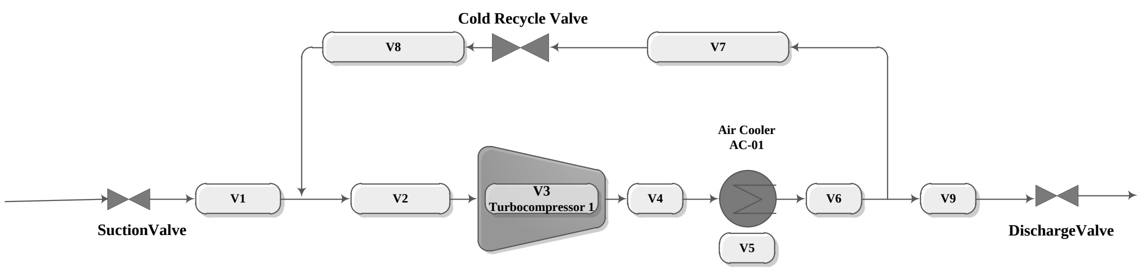

Figure 2- The 9 sub-volumes used to model single train

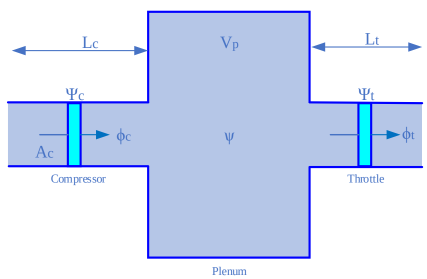

Figure 3-Compression system

$\dot{m}$ = Mass-flow
ΔP = Pressure difference between the pressure in the system and the ambient pressure
ρ_a = The air density at ambient conditions,
A_c = The compressor duct area
a = The speed of sound
U_t = The rotor tip speed
V_p = The plenum volume
L_C = Equivalent compressor duct length
The following set of dimensionless equations that describe the non-linear compression system are:

$$\frac{d\varphi_c}{d\tilde{t}} = B\,[\Psi_C - \psi] \tag{3}$$

$$\frac{d\varphi_t}{d\tilde{t}} = \frac{B}{G}\,[\psi - \Psi_t] \tag{4}$$

$$\frac{d\psi}{d\tilde{t}} = \frac{1}{B}\,[\varphi_c - \varphi_t] \tag{5}$$

$$\frac{d\Psi_C}{d\tilde{t}} = \frac{1}{\tau}\,[\Psi_{C,ss} - \Psi_C] \tag{6}$$

The equations for the behavior of the dimensionless mass flow φ_c in the compressor duct and φ_t in the throttle duct are essentially the momentum equations for each duct. Ψ_C is the dimensionless pressure rise across the compressor and Ψ_t gives the dimensionless pressure drop across the throttle. The equation for the pressure rise in the plenum ψ gives the mass conservation in the plenum. The expression for the dimensionless pressure rise across the compressor

Ψ_C is a first-order transient response model with the time constant $\tilde{\tau}$ and $\Psi_{C,ss}$ the steady-state dimensionless compressor pressure rise given in the compressor map. In these equations the Greitzer stability parameter is defined as:

$$B = \frac{U_t}{2\omega_H L_c} \quad \textbf{(7)}$$

And

$$G = \frac{L_t A_c}{L_c A_t} \quad \textbf{(8)}$$

with the throttle duct length L_t and area of A_t. Compressor performance maps should be processed, converting them based on dimensionless corrected mass flow (equation 1) and corrected speed (equation 2). Compressor map lines from 80% to 105% of design speed are imported from the compressor manufacturer. Using fitting tools, other compressor map speed lines from 70% to 5% should be produced.

$$(Corrected\ mass\ flow\ rate = \dot{m}\frac{\sqrt{T_{in}}}{P_{in}} / \dot{m}_d \frac{\sqrt{T_{in,d}}}{P_{in,d}}) \quad \textbf{(9)}$$

$$Corrected\ Speed\ = \omega / \frac{\sqrt{T_{in}}}{\sqrt{T_{in,d}}}) \quad \textbf{(10)}$$

Which the parameters are described as follows:
$\dot{m}^*$= Corrected mass flow rate
$\dot{m}$= Mass flow rate
T_{in}= Compressor Suction temperature
P_{in}= Compressor Suction pressure
$T_{in,d}$= Compressor Design Suction temperature
ω = Real Shaft Speed

It should be noted that six operating conditions are defined for turbo-compressors as below:

1. Case 1 (Summer case, normal operating condition)
2. Case2 (winter Case)
3. Case3 (C_2 rejection)
4. Case4 (Partial load)
5. Case 1 max (110% normal summer flow)
6. Case2 max (110% normal winter flow)

The dynamic model is used to model all these six operating conditions. Therefore, required information, including compressor characteristic curves and fuel conditions, are processed for all six operating conditions. Our comprehensive model solves most unit operations using lumped models, meaning that directional gradients (along with the x, y and z-axes) are neglected, and the solved property is assumed to be constant within each sub-volume. Nine sub-volumes are defined as below (Figure 2):

1. **After suction valve**: The volume of pipes from suction valve to where recycle pipeline intersects suction pipeline.
2. **Before gas compressor**: The volume of pipes from the recycling pipeline intersection to the gas compressor inlet.
3. **Gas compressor**: volume of the gas compressor.
4. **Before cooler**: The volume of pipes from gas compressor outlet to air cooler inlet.
5. Cooler: The volume of air cooler, inlet and outlet cooler header.
6. **Aftercooler**: containing the volume of pipes from trim coolers outlets to where recycle pipeline intersects discharge pipeline.
7. **Before cold recycle valve**: The volume of pipes of recycling line before cold recycle valve.
8. **After cold recycle valve**: The volume of pipes of recycling line after cold recycle valve.
9. **Before discharge valve**: The volume of pipes from "aftercooler" to discharge valve.

All these nine volumes are required for dynamic simulation and must be extracted from the plant documents. In addition, there are six main valves in this unit including the unit suction valve, pressurized valve, unit discharge valve, header suction valve, discharge header valve, cold anti-surge valve

To simulate their action, it is necessary to know the opening and closing time of actuators and CV[1] of valves. Based on this document CV of the anti-surge valve is 800, and the summation of stroke time and lag time of its actuator is 2s. Since the recycle gas to compressor has almost high speed, another principal design criterion for anti-surge control and recycle line design is maximum velocity and noise in recycle line piping. Given that, noise and velocity analyses are regarded as a part of the surge control design process.

2.2. CASE STUDIES

The double-loop cascade control system based on the compression unit's total suction flow and each compressor's speed control is proposed to control column pressure control. This control system was modelled and implemented successfully in our compression unit model. The simulation of the two most critical scenarios in compressor operation is completed in the model to test the compressor's performance: 1) Emergency Shutdown (ESD) scenario and 2) Full Recycle scenario. ESD scenario is the most severe benchmark for the anti-surge control system evaluation. These scenarios require the largest and the fastest response of the anti-surge system to maintain the compressor from going into surge. ESD events will cause the compressor to approach the surge line much quicker than a low flow operational period in normal operation. The ESD scenario can cause the turbo-compressor to cross the surge control line without effective control. The surge control line or protection margin is an allowable and safe distance from the actual surge margin that the compressor manufacturer assesses. This phenomenon becomes more intricate if the ESD occurs in a situation the compressor is operating at its maximum allowable head and consequently closer distance to the surge margin. The surge control system, anti-surge valve, and recycle line should be designed based on compressor safety issues in severe scenarios such as ESD. The surge control system is activated after receiving the trip signal with the fully open command of the anti-surge valve and then closing all gas inlet and outlet of the compressor with pressurizing or depressurizing command depending on the process condition priorities.The other scenario, the full recycle scenario, is another emergency scenario for compressor performance in the case that the discharge valve of the unit or train fails or is closed. The procedure of this scenario starts with the closing of the discharge valve with its stroke time. While the flow is reducing, the operating point senses the surge control line, which normally has a surge margin of 10%. At this point, the anti-surge control system commands to open the valve with normal speed, until the operating point surpasses the setpoint line. If flow reduction is as quick as the operating point passes the surge line, force open command is sent to the anti-surge valve. This scenario is also notably challenging for the compressor protection and surge control system which is necessary to analyze.

2.3. ESD SCENARIO

This scenario aims to evaluate emergency shutdown procedures or any other action with equivalent consequences. The scenario results must reveal the presence of surge and provide all pressure, temperature, flow rate and physical properties profiles for any stream along with the scenario run time. The emergency shutdown is simulated with the following procedure:

Table 1-ESD Scenario procedure

Gas Compressor ESD of one compressor train	
Configuration	- All controllers are in Auto mode. -ASC (Anti Surge Control) in Cascade mode with control flow. Compressor system with one ASV (Anti Surge Valve) for each train. Three compressors are running.
Actions	At time 1200 seconds of the simulation time, the turbo-compressor 1 trip occurs: - Shutdown main turbo-compressor. -Power declination rate is implemented from the gas turbine model. After 0.5 seconds – dead time -: - Close the suction valve with a stroke time of 180s - Close compressor discharge isolation valve with a stroke time of 3-5 min - Quickly open anti-surge valve via ESD action through solenoid valve ASCV by spring force (fully open in 2s)
Duration	Until settle out pressure is achieved.

[1] Coefficient flow of valve

2.4. FULLY RECYCLE SCENARIO

This scenario's objective is to evaluate the full recycle procedure as defined in the introduction or any other with equivalent actions. In this study, two scenarios can describe as a full recycle scenario: unit discharge valve failure and header discharge valve failure. The results of both full recycle scenarios are presented in the following section:

2.5. UNIT DISCHARGE VALVE FAILURE

The scenario results must reveal the presence of surge and provide all pressure, flow rate and physical properties profiles for any stream along with the scenario run time. The full recycle shutdown is simulated with the following procedure:

Table 2-Unit discharge valve failure procedure

Gas Compressor Full recycle of one compressor train	
Configuration	- All controllers are in Auto mode. -ASC (Anti Surge Control) in Cascade mode with control flow. Compressor system with one ASV (Anti Surge Valve) for each train. Three compressors are running.
Actions	**At time 1200 seconds of the simulation time, the train one Discharge valve closing occurs:** - Unit Discharge valve begins to close. - The discharge valve's stroke time is 180s, which is regarded as minimum stroke time.
Duration	Until the unit reach steady state condition.

3. VALIDATION AND SENSITIVITY ANALYSIS

The model validation tests are based on steady-state and dynamic manufacturer experimental factory data exporting from the turbo-compressor test bench. Furthermore, the turbine dynamic behaviour and interactions, notably turbine power decline rate, have been analyzed and validated through various turbo-compressor projects' actual data and experimental factory tests. The validation analysis shows that the model can simulate the turbo-compressor unit with an acceptable error. **Error! Reference source not found.** to Figure 6 present the turbine model validation curves, and Figure 7 presents ESD event data from a real compressor site compared to our compressor model results. There is good compatibility between experimental data and model results remarkably in high-speed compressor rate, which is the main concern in surge prediction studies.

Also, to analyze the sensitivity of key parameters, anti-surge CV sensitivity analysis for ESD scenario and discharge valve stroke time sensitivity analysis for full recycle scenario in normal operation cases have been carried out, presented in **Error! Reference source not found.** and **Error! Reference source not found.**.

4. RESULT AND DISCUSSION

Results of turbo-compressor unit dynamic simulation are presented for both scenarios, which are defined in the introduction. This evaluation focuses on whether the anti-surge system can handle the event without going into surge. Dynamic simulation is done for all six operating conditions.

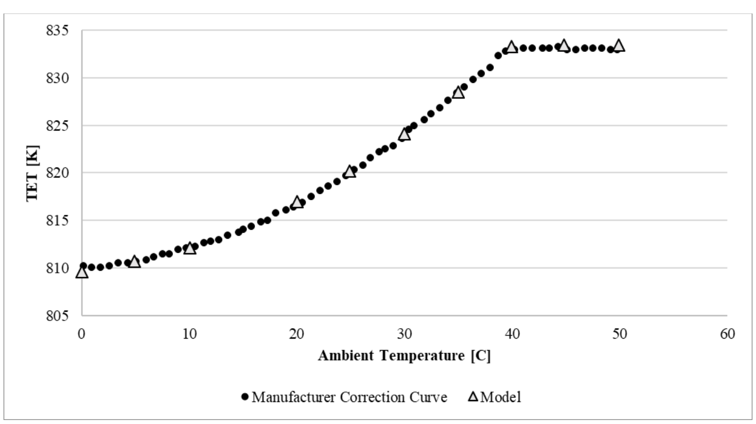

Figure 4- Turbine exhaust temperature Vs. Ambient temperature (Manufacturer Curves Vs. Model result)

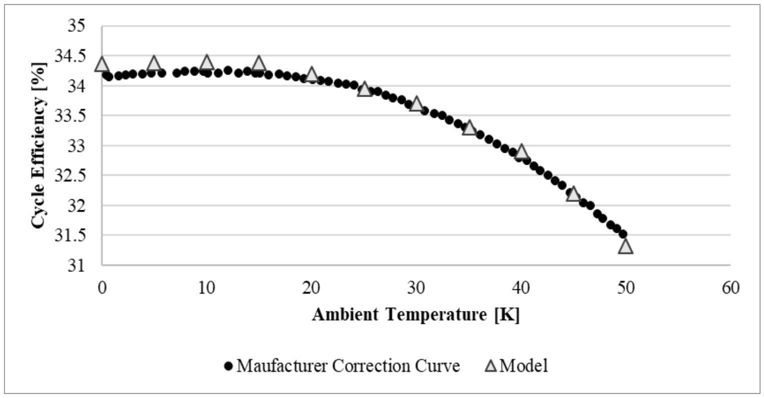

Figure 5-Turbine Efficiency Vs. Ambient Temperature (Manufacturer Curves Vs. Model result)

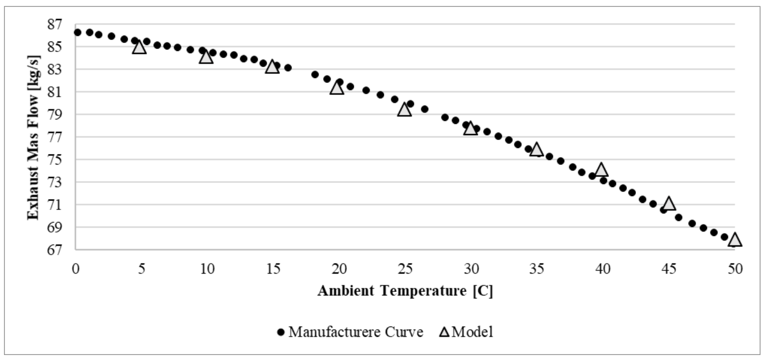

Figure 6- Exhaust Mass Flow Rate Vs. Ambient Temperature (Manufacturer curves Vs. Model result)

4.1. ESD Scenario

Under all operating conditions, dynamic simulation results for ESD Scenario have been studied and analyzed. In **Error! Reference source not found.**, the trail of the operating points during the shutdown is shown as solid lines with markers. The surge line will follow the left-most points of the compressor curves, and the operating point will travel from the full speed compressor performance curve towards the origin during a shutdown. This figure shows that the compressor has safe performance during the ESD scenario. Mass flow and pressure ratio of gas compressor after the trip are shown in **Error! Reference source not found.**. After the compressor trip, the pressure ratio drops to one, and the mass flow rate drops to zero. Variation of the speed of gas generator (GG) and power turbine (PT) is shown in **Error! Reference source not found.**.

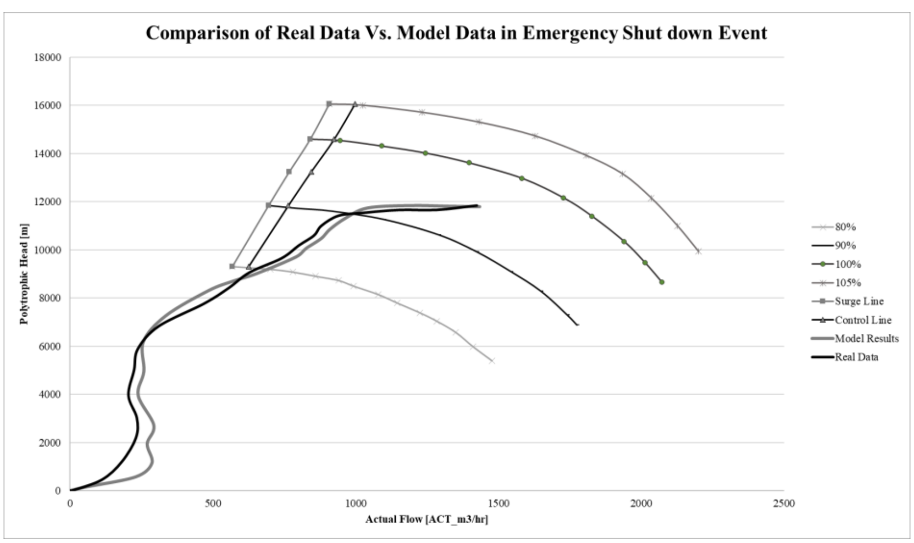

Figure 7-ESD event compressor running line (real data vs. model result)

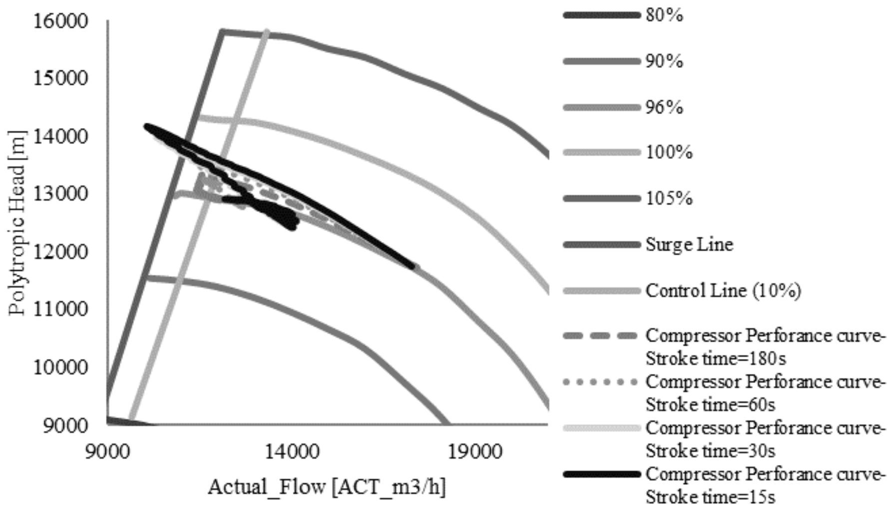

Figure 8- Stroke time sensitivity analysis during full recycle scenario

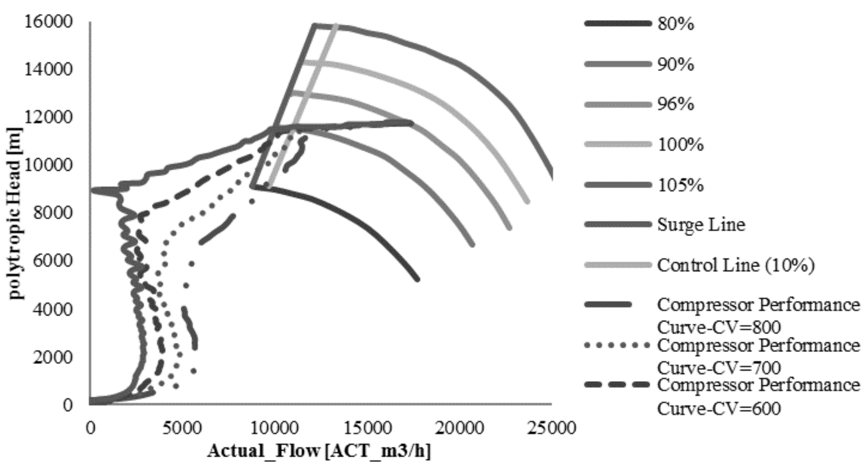

Figure 9-CV sensitivity analysis during ESD scenario

The speed profile of compressors in the compression unit is figured in Figure13. The percentage of compressor speed design presents the speed lines in the figures. The speed controller is based on flow control, as mentioned in the model parameter. When the ESD scenario occurs, consequently, header suction flow reduces proportionately. Therefore, two other compressors speed up to compensate for flow reduction. There are two limitations for the compressor in the process of speeding up: Turbine power and maximum compressor limitation. In this case, we face turbine power limitations because of high ambient temperature. In this case, the ESD scenario makes a 21% header Suction flow reduction (Figure13), affecting demethanizer overhead pressure. The pressure profile of compressors is also presented in **Error! Reference source not found.**, and settle out pressure, in this case, is calculated at 52.6 bar.

Error! Reference source not found. presents the velocity analysis of the compressor unit to noise analysis in the anti-surge line and valve. Aerodynamic noise is generated as high-velocity gas oscillates. According to noise standard equations, the maximum calculated noise generated in anti-surge flow is 47.3 dB depending on gas density, velocity and piping characteristic, which is less than maximum allowable noise (85 dB) (ISA and IEC, 2010). The maximum velocity at valves must also be under 0.3 Mach number, and the maximum velocity based on the maximum density of gas in lines must be under 44.5 m/s in case 1. As you see in **Error! Reference source not found.**, the velocities in the valve and lines are under maximum allowable value (NTC,2001).

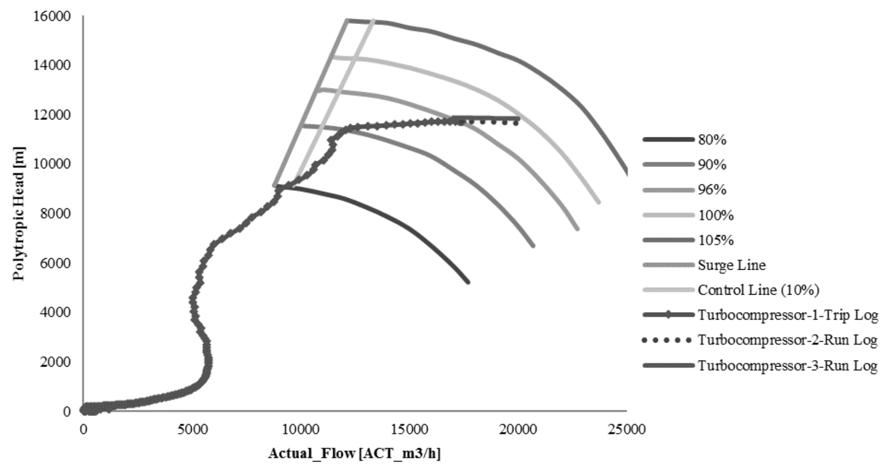

Figure 10-Running line of compressor for ESD Scenario (Case1)

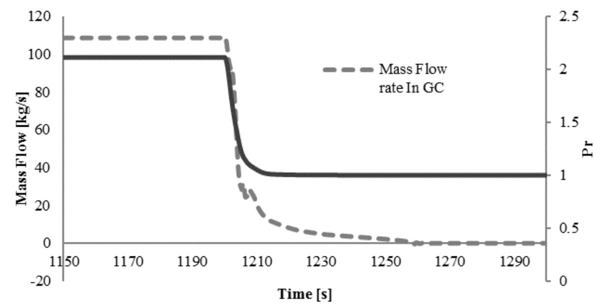

Figure11- Mass flow and pressure ratio of gas compressor for ESD Scenario (Case1)

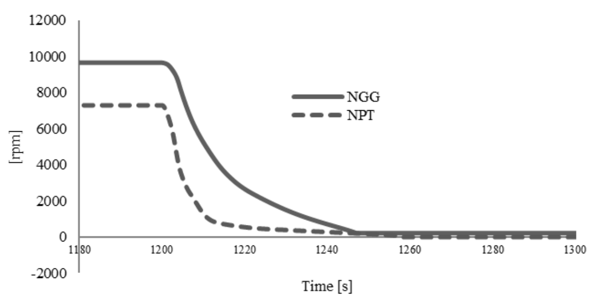

Figure12- Speed of gas generator and power turbine for ESD Scenario (Case1)

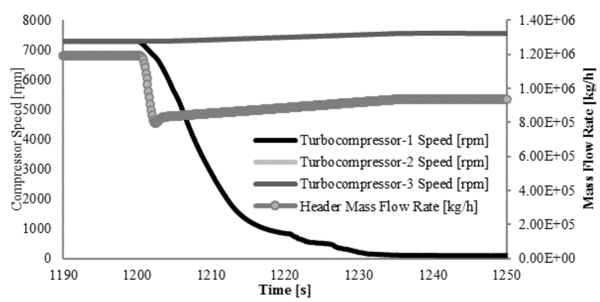

Figure13- Speed profile of turbo-compressors and header mass flow rate profile ESD scenario (Case 1)

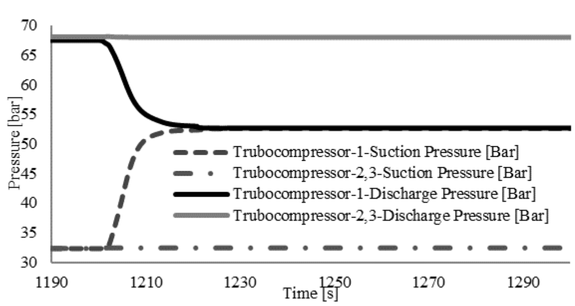

Figure14- Pressure profile of compressors in unit22 for

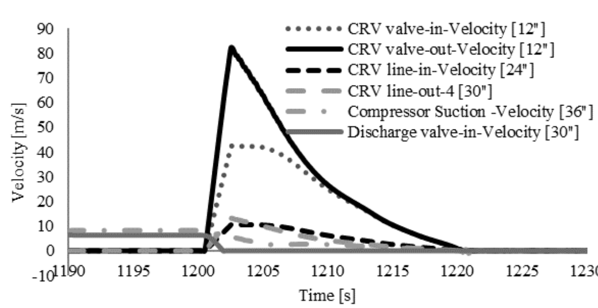

Figure 15-Velocity profile for ESD scenario (case1)

ESD scenario (Case 1)

Results of dynamic simulation in the ESD scenario were analyzed for all operating conditions cases. The results concluded that the anti-surge control system using a cold recycle valve with a CV of 800 and actuator stroke and lag time of 2s performs well enough to prevent the compressor from going into surge and keep compressor condition safe ESD scenario in normal cases. Although, there should be some additional considerations for the partial load case, case 4. Case 4 is defined as a partial load case with a 21% anti-surge valve opening. Figure16 displays the results of the ESD scenario in case 4 conditions. As presented in this figure, the operating point has passed the surge line, but with the increasing effect of the anti-surge valve, it returns to the safe zone again. This phenomenon has occurred two times at high speed of the compressor. Although the operating point passes the surge line during ESD Scenario, its flow rate doesn't fall under 30% of surge flow, and surge duration is fewer than 1s, which are the effective surge criteria. Therefore, it doesn't consider an effective and minatory surge. However, it is suggested to keep the surge margin over 20% by increasing the anti-surge percentage open as a safety margin, according to Figure 17. Two other compressors can compensate for the compressor load after the ESD scenario occurs (Figure18). After ESD occurs, two other running compressors speed up the same as before ESD scenario with 21% anti-surge valve opening.

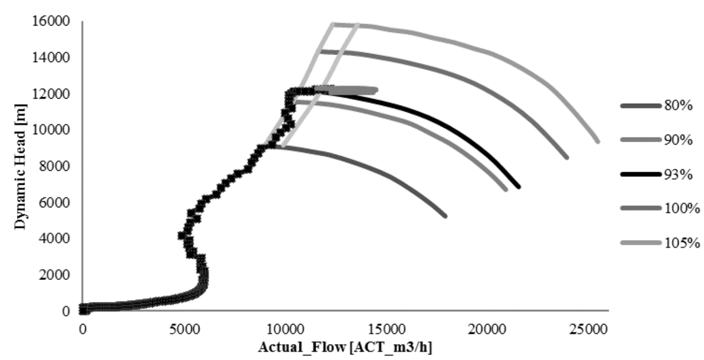

Figure16- Running line of compressor for ESD scenario (Case4)

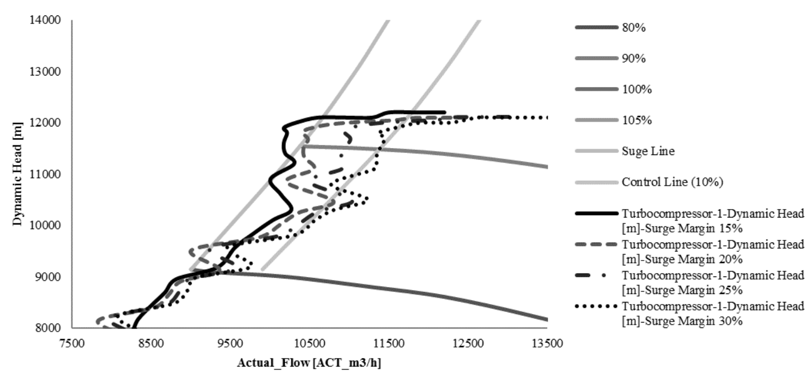

Figure 17-Surge margin analysis for ESD scenario (Case 4)

4.2. Full Recycle Scenario

Dynamic simulation results for the full recycle scenario are analyzed under all operating conditions. These results in case 1 or normal operation condition case is presented in Figure 19 to Figure 21. As shown in Figure 19, the compressor has safe performance during the Full Recycle scenario. The speed profile of compressors in the demethanizer unit has been figured out in Figure 20. In this case, we face maximum turbine power limitation to compensate for flow reduction resulting from the full recycle scenario. In this case, the full recycle scenario makes a 22% header suction flow reduction (Figure 21), which subsequently affects demethanizer overhead pressure. As demonstrated in Figure 21, the anti-surge valve should open at least 51%, providing a safe condition for compressor operation.

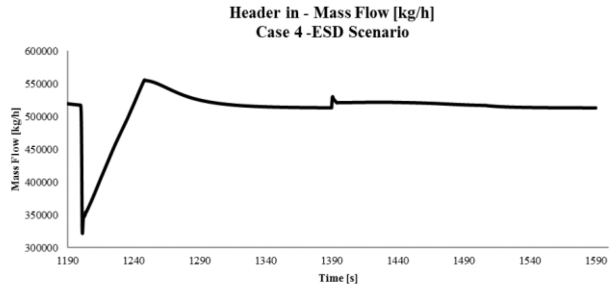

Figure18- Header-suction mass flow profile for ESD scenario (Case4)

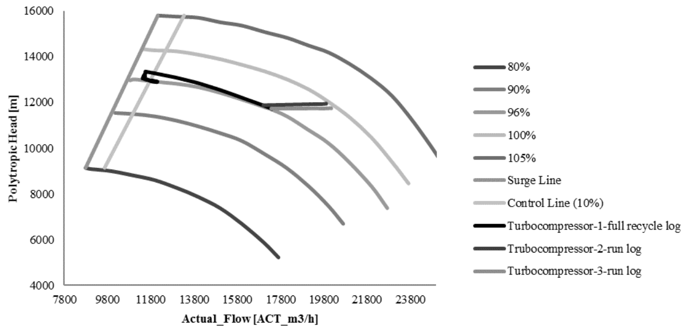

Figure 19-Running line of compressor for full recycle scenario (Case1)

Our dynamic model simulates the compressor full recycle scenario in all operating conditions mentioned in the compressor datasheet. The results show that surge doesn't happen in any conditions at this configuration. Therefore, the anti-surge control system is suitable for ensuring the compressor's safe operation.

4.3. Header discharge valve failure

Our dynamic model simulates the compressor full recycle scenario in all operating conditions mentioned in the compressor datasheet. The results show that surge doesn't happen in any conditions at this configuration. Therefore, the anti-surge control system is suitable for ensuring the compressor's safe operation.

In the following sections, dynamic simulation results for the full recycle scenario are presented under the operation condition of normal operation. Figure 22 to Figure 24 display the results of the Full Recycle scenario in normal operating conditions. As presented in Figure 22, the compressor has safe performance during the full recycle scenario. The speed profile of compressors in the demethanizer unit has been figured out in Figure 23. Within 43 seconds, the header inlet flow declines to zero (Figure 24). As demonstrated in Figure 24, the Anti-surge valve should open at least 52%, providing safe conditions for compressor operation. The minor difference between the 3 compressors is due to the differences in piping details.

5. Conclusion

In this study, the demethanizer unit is simulated using the dynamic model to evaluate the anti-surge control system and compressor safety under critical conditions, including emergency shutdown and full recycle conditions. Dynamic simulation is done for six operating conditions for two scenarios (ESD and full recycle). The model has a simulated tripped compressor and two other running compressors in the compression unit.

The dynamic simulation results presented in the previous section show that the compressor operates safely and stable during emergency shutdown and full recycle scenarios in all six operating conditions without using a hot bypass valve. Therefore, suggested characteristics for cold anti-surge valve, including the coefficient of the flow of 800 and actuator time lag of 2s, are sufficient to ensure compressor safety and anti-surge control system can handle the event from going into surge. The conclusions are presented in Table 4 summary for all operating conditions and both scenarios.

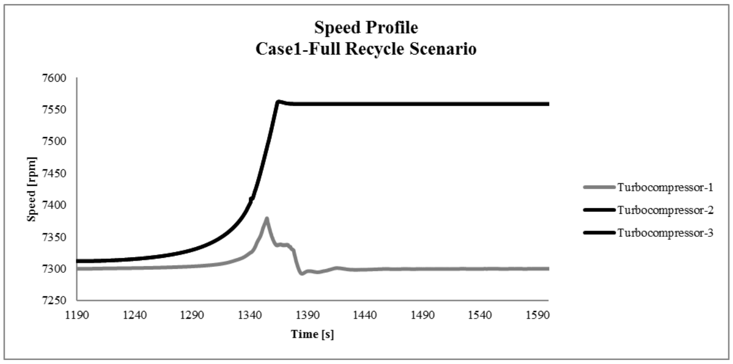

Figure 20-Speed profile of three compressors for full recycle scenario (Case1)

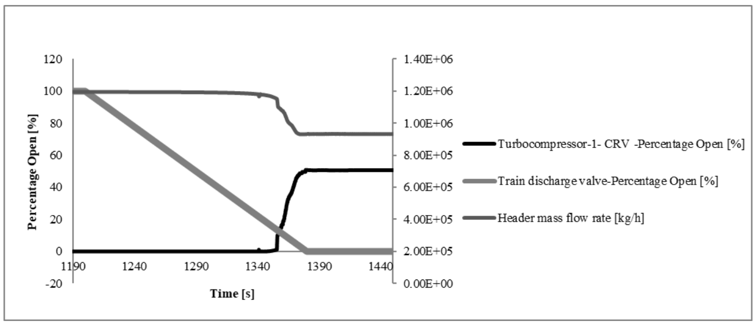

Figure 21-Header mass flow rate, anti-surge and discharge valve percentage open for full recycle scenario (Case1)

Table 3-Header discharge valve failure scenario procedure

Gas Compressor Full recycle of one compressor train	
Configuration	- All controllers are in Auto mode. -ASC (Anti Surge Control) in Cascade mode with control flow. Compressor system with one ASV (Anti Surge Valve) for each train. Three compressor are running
Actions	At time 1200 seconds of the simulation time, the Header Discharge valve closing occurs: - Header Discharge valve begins to close. -The stroke time of the discharge valve is 43s.
Duration	Until the unit reach steady state condition.

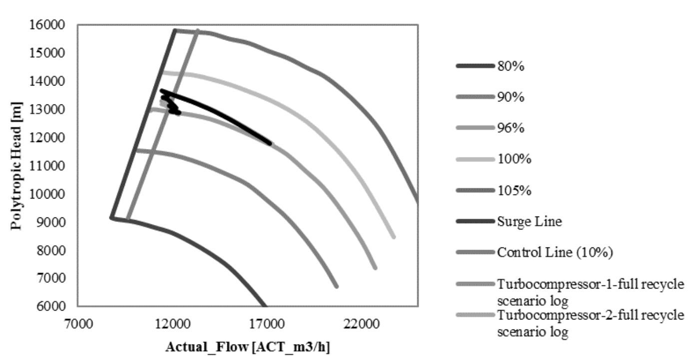

Figure 22- Running line of compressor for header discharge valve failure scenario (Case1)

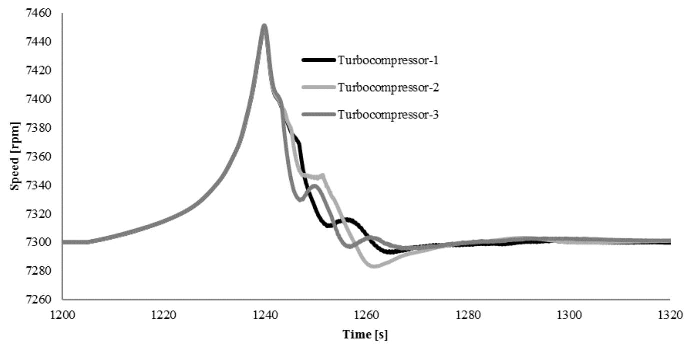

Figure 23-Speed profile of three compressors in the demethanizer unit for header discharge valve failure scenario (Case1)

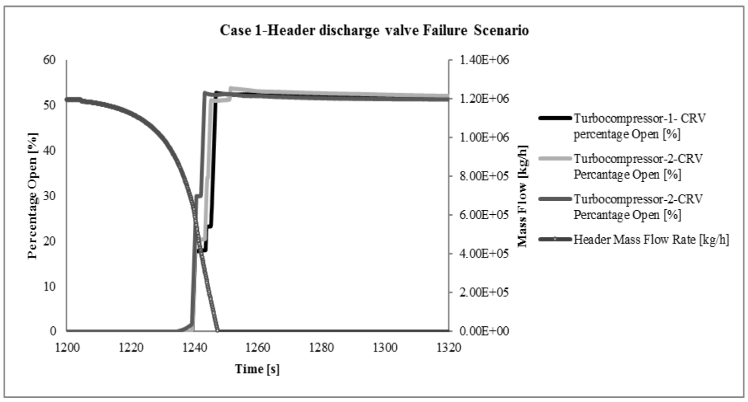

Figure 24-Anti-surge valves percentage open and header mass flow rate decline for header discharge valve failure scenario (Case1)

Table 4-the summery of all operating condition cases in two transient scenarios

No.	Scenario	Train No.			Operating Condition	Instability status	Conclusion
1	Trip	1	2	3	Case1	Stable	The anti-surge control system is suitable to handle the event and prevent the compressor from going into surge
2		1	2	3	Case2	Stable	
3		1	2	3	Case3	Stable	
4		1	2	3	Case4	Stable	
5		1	2	3	Case1max	Stable	
6		1	2	3	Case2max	Stable	
7	Full Recycle	1	2	3	Case1	Stable	
8		1	2	3	Case2	Stable	
9		1	2	3	Case3	Stable	
10		1	2	3	Case4	Stable	
11		1	2	3	Case1max	Stable	
12		1	2	3	Case2max	Stable	
13	Header Discharge Valve Failure				Case1	Stable	
Transient Maneuver					Normal Operation		

References:

K. K. Botros, P. J. Campbell, and D. B. Mah, "Dynamic simulation of compressor station operation including centrifugal compressor and gas turbine," Journal of Engineering for Gas Turbines and Power, Article vol. 113, no. 2, pp. 300-311, 1991.

K. K. Botros, B. Ruggiero, M. Szwarz, and C. Ho, "Dynamics of the recycle systems of two centrifugal compressors operating in series in a natural gas compressor station," 2014, vol. 4.

N. Courtiade and X. Ottavy, "Experimental Study of Surge Precursors in a High-Speed Multistage Compressor," Journal of Turbomachinery, vol. 135, no. 6, pp. 061018-061018-9, 2013.

M. W. Davis, Jr. and W. F. O'Brien, "A stage-by-stage post-stall compression system modeling technique," 1987.

E. M. Greitzer, "Surge and rotating stall in axial flow compressors: Part I: Theoretical compression system model," Journal of Engineering for Gas Turbines and Power, Article vol. 98, no. 2, pp. 190-198, 1976.

E. M. Greitzer, "Axial compressor stall phenomena," Journal of Fluids Engineering, Transactions of the ASME, Article vol. 102, no. 2, pp. 134-151, 1980.

E. M. Greitzer and F. K. Moore, "A theory of post-stall transients in axial compression systems: Part II—application," Journal of Engineering for Gas Turbines and Power, Article vol. 108, no. 2, pp. 231-239, 1986.

K. E. Hansen, P. Jorgensen, and P. S. Larsen, "Experimental and Theoretical Study of Surge in a Small Centrifugal Compressor," Journal of Fluids Engineering, vol. 103, no. 3, pp. 391-395, 1981.

ISA and IEC, "the international standard IEC 60534-8-3, "Industrial process control valves – Part 8-3: Noise considerations – Control valve aerodynamic noise prediction method",2010.

K. Kirill, K. Władysław, L. Grzegorz, H. Longin, and M. Radomir, "Experimental investigation of the influence of the inlet duct configuration on the unstable operation of a single-stage centrifugal blower," Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, vol. 230, no. 3, pp. 260-271, 2016/05/01 2016.

S. Marelli, C. Carraro, G. Marmorato, G. Zamboni, and M. Capobianco, "Experimental analysis on the performance of a turbocharger compressor in the unstable operating region and close to the surge limit," Experimental Thermal and Fluid Science, Article vol. 53, pp. 154-160, 2014.

A. Mohajer and E. Abbasi, "Development of compression system dynamic simulation code for testing and designing of anti-surge control system," 2017, vol. 9.

F. K. Moore and E. M. Greitzer, "A theory of post-stall transients in axial compression systems: Part 1—development of equations," Journal of Engineering for Gas Turbines and Power, Article vol. 108, no. 1, pp. 68-76, 1986.

M. Morini, M. Pinelli, and M. Venturini, "Development of a one-dimensional modular dynamic model for the simulation of surge in compression systems," Journal of Turbomachinery, Article vol. 129, no. 3, pp. 437-447, 2007.

M. Morini, M. Pinelli, and M. Venturini, "Application of a One-Dimensional Modular Dynamic Model for Compressor Surge Avoidance," no. 47934, pp. 1425-1434, 2007.

E. Munari, M. Morini, M. Pinelli, P. R. Spina, and A. Suman, "Experimental Investigation of Stall and Surge in a Multistage Compressor," Journal of Engineering for Gas Turbines and Power, vol. 139, no. 2, pp. 022605-022605-10, 2016.

NTC, "NORSOK, STANDARD PROCESS SYSTEM, P-100," ed. Norwegian Technology Centre, 2001.

S. J. Pak, J. W. Yoon, S. Kim, F. Salim, J. Lee, and I. W. Kim, "Case study of surge avoidance design for centrifugal compressor systems during emergency shut-down (ESD)," Korean Journal of Chemical Engineering, Article vol. 33, no. 3, pp. 764-774, 2016.

M. Venturini, "Development and Experimental Validation of a Compressor Dynamic Model," no. 41723, pp. 801-811, 2004.

Y. Zhan, J. Wang, W. Wang, and R. Wang, "Dynamic simulation of a single nitrogen expansion cycle for natural gas liquefaction under refrigerant inventory operation," Applied Thermal Engineering, Article vol. 128, pp. 747-761, 2018.

X. Zheng and A. Liu, "Phenomenon and Mechanism of Two-Regime-Surge in a Centrifugal Compressor," Journal of Turbomachinery, vol. 137, no. 8, pp. 081007-081007-7, 2015.