Modeling, simulation and evaluation of a centrifugal compressor with surge avoidance control

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Modeling, simulation and evaluation of a centrifugal compressor with surge avoidance control.

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Table of contents

1 Introduction  2

2 Compression system components  4

2.1 Compressor  4

2.2 Orifice  8

2.3 Controller  10

2.3.1 Differential action  14

2.4 Control valve  15

2.5 Disturbances  16

2.6 Plenum  18

2.7 Total system  19

3 Simulation  21

3.1 Approach  21

3.2 Results  22

4 Conclusions and recommendations  31

References  32

Nomenclature  33
1 Introduction

In order to obtain natural gas from a well, it is necessary to pump it to the surface and into a network. The pump normally used in this case is a centrifugal compressor. Besides the pressurization and transportation of fluids in the process and chemical industries, other applications of centrifugal compressors involve fluid compression for use in aircraft engines, in industrial gas turbines and in turbocharged combustion engines [1]. The working principle of a centrifugal compressor is to increase the kinetic energy of the fluid with a rotating impeller. The fluid is then slowed down in a volume called the plenum, where the kinetic energy is converted into potential energy in the form of a pressure rise.

Centrifugal compressors have an instable working region. In this region, a decrease of flow results in a decrease of outlet pressure. When the plenum pressure behind the compressor is higher then the compressor outlet pressure, the fluid tends to reverse or even flow back in the compressor. As a consequence, the plenum pressure will decrease, inlet pressure will increase and the flow reverses again. This phenomenon, called surge, repeats and occurs in cycles with frequencies varying from 1 to 2 Hz. Another aerodynamic instability that can occur in centrifugal compressors is stall. Both instabilities dramatically decrease the efficiency and severe surge can even cause mechanical damage to the compressor. In the case study of this internal traineeship, stall is not taken into account and all attention is focused on avoiding surge.

A commonly used method to avoid compressor surge, is to recycle part of the pressurized fluid from the compressor outlet back to the compressor inlet. This decreases the plenum pressure and increases the flow through the compressor, resulting in stable working conditions. The amount of recycled fluid is determined by the position of a control valve. A schematic flow sheet (top) and the corresponding block scheme of the signals in the total system (bottom) are shown in Fig 1.1.
The specific compression system with corresponding surge avoidance controller in this case study is designed by Siemens Demag Delaval Turbomachinery B.V. to obtain natural gas from a well. It is desirable to predict and perhaps modify the behavior of the compressor with surge avoidance control. Therefore, the objectives of this internal traineeship are:

1. Development of a simulation model of a centrifugal compressor with surge avoidance control, based on experimental data.

2. Give a qualitative evaluation of the behavior of the modeled centrifugal compressor and surge avoidance controller for various disturbances.
2 Compression system components

Before we make a model of the components, its function and working principle are considered. In this chapter the compression system components (these are: compressor, surge avoidance controller, orifice and control valve) will be treated in terms of:

- Working principle
- Function
- Modeling

2.1 Compressor

Working principle
In general, the centrifugal compressor consists of a stationary casing, containing a rotating impeller that imparts a high velocity to the fluid, and a number of fixed diverging passages in which the fluid is decelerated with a consequent rise in static pressure in the plenum. The latter process is one of diffusion, and thus the part of the compressor containing the diverging passages is known as the diffuser. Fig. 2.1(a) is a sketch of a centrifugal compressor. The impeller may be single- or double-sided as in 2.1(b) or 2.1(c) but the working principle is the same. Fluid enters the impeller eye and is whirled around at high rotational speed by the vanes on the impeller disc. The static pressure rise is obtained in the diffuser, where the very high velocity of the fluid, leaving the impeller tip, is reduced to a velocity similar to the velocity of the fluid entering the impeller eye.

Figure 2.1 Sketches of centrifugal compressors [1].
Function
The function of a compressor is to pressurize a fluid. The compressor characteristics are visualized in a so-called compressor map. In this compressor map, curves relate the rotational speed ($N$) to the outlet pressure ($P_2$) and the volume flow ($Q_v$) through the compressor. A schematic representation of a compressor map is shown in Fig. 2.2.

![Compressor Map Schematic](image)

Figure 2.2  Schematic representation of a compressor map.

At high flows, the operating region is bounded by the *stonewall line*. Here, the flow in the compressor locally reaches a Mach number of one, causing the flow to choke. The *load line* is the characteristic of the resistance that the pressurized fluid encounters behind the compressor. The intersection of the load line and the compressor characteristic for a specific $N$ is the operating point of the combination of compressor and load process. The *surge line* connects points from whereon surge can occur. Though briefly mentioned in the introduction, a more detailed description of surge with use of the compressor map is provided here. Surge is associated with a drop in delivery pressure, and with violent aerodynamic pulsations that are transmitted throughout the whole machine. It may be explained as follows. If we suppose that the compressor is operating at some point on the part of the characteristic having positive slope, then a decrease in flow will be accompanied by a fall of delivery pressure. If the pressure of the fluid downstream of the compressor is higher, the fluid will reverse and flow back in the compressor. When this occurs, the pressure ratio drops rapidly. Meanwhile, the pressure downstream of the compressor has decreased as well. The compressor will now be able to pick up again to repeat the cycle of events resulting in oscillatory behavior. As visualized in Fig. 2.2, the part left of the surge line has positive slope and this is the region where surge can occur.
A safety margin is taken into account, resulting in a new line called the **surge avoidance line**. The task of the surge avoidance controller is to prevent the compressor from operating in a point in the compressor map that is located to the left of the surge avoidance line.

**Modeling**

Although the behavior of a centrifugal compressor is not constant in time, as a first approximation a quasi-stationary model is derived. The used quasi-stationary model can be extended later, or it can be replaced by a dynamic model [2]. The aim is to develop a model with flow as input and pressure rise as output. Density and temperature are assumed to be constant throughout the whole system. Flows will be expressed as volume flows in m³/hr and pressures will be absolute pressures in bar. The compressor characteristics can be approximated with 3rd order polynomials for each line of constant rotational speed \( N \), according to:

\[
\begin{align*}
P_2(Q_v) &= a_{i,1}Q_v^3 + a_{i,2}Q_v^2 + a_{i,3}Q_v + a_{i,4} \\
\end{align*}
\]  

(2.1)

A set of manufacturer data in the form of points on predicted performance curves - the compressor is not built yet - is used to derive an approximation of the quasi-static compressor characteristic. Six curves are described by fitting a 3rd order polynomial through seven points for each curve. A curve corresponds with a specific rotational speed, expressed as a percentage of the nominal rotational speed (12200 rpm). Linear interpolation of the polynomial coefficients in (2.1) is carried out to obtain an approximation of the intermediate curves. Since the goal of the surge avoidance strategy is to stay in the stable operating region of the compressor, only this region of the compressor map is approximated. The compressor output data \( (H) \) in \( (kJ/kg) \) is converted into absolute output pressure \( P_2 \) (bar) by

\[
\begin{align*}
P_2 &= P_1 \cdot \exp \left( \frac{\ln \left( \frac{H}{c+1} \right)}{k_1 - 1} \right) \\
\end{align*}
\]  

(2.2)

with

\[
\begin{align*}
c &= 0.001 \cdot Z \left( \frac{8314.34}{M} \right) \left( 273.15 + T_i \right) \left( \frac{k_1}{k_1 - 1} \right) \\
\end{align*}
\]  

(2.3)

\[
\begin{align*}
P_1 &= 38.2 \text{ (bar)} \\
Z &= 0.92 \text{ (-)} \\
M &= 18.42 \text{ (kg/mol)} \\
T_i &= 30.0 \text{ (°C)} \\
k_1 &= 1.45 \text{ (-)}
\end{align*}
\]
The result of the 3\textsuperscript{rd} order polynomial fits is the approximated compressor map that is displayed in Fig. 2.3.

Figure 2.3  Approximated compressor map; the solid thin lines are characteristics for constant rotational speed, the solid fat line is the surge line, the dashed line is the surge avoidance line, the dashed-dotted line is the stonewall line, the dots are data points and the square is the design point.
2.2 Orifice

Working principle
The flow measurement is based on the installation of a flow restriction named ‘orifice’. The installation of the orifice causes a static pressure difference between the upstream side and the throat. The flow can be determined from the measured value of this pressure difference and from the knowledge of the characteristics of the fluid and the circumstances under which the device is being used. A technical drawing of a nozzle is shown in Fig. 2.4. The nozzle to be modeled is described in [3].

![Figure 2.4 Schematic of ISA 1932 Nozzle [3].](image)

The mass flow \( Q_m \) (kg/s) can be determined via equation (2.4)

\[
Q_m = \frac{C}{\sqrt{1 - \beta^4}} \frac{e \pi d^2}{4} \sqrt{2\Delta P \rho_i}
\]  

(2.4)

Parameter \( C \) is a function of \( Q_m \), however in this case except for \( Q_m \) (kg/s) and \( \Delta P \) (Pa), all parameters are assumed to be constant.

Function
The function of an orifice is to apply a resistance resulting in a pressure difference from which the flow can be calculated. Together with the pressure sensors and the signal processing hardware it is a flow measuring device and the measurement signal serves as the controller input.
Model
The aim is to develop a model with volume flow $Q_v$ as input, and pressure difference $dP_i$ (mbar) as output. Since density $\rho$ is constant and (2.4) is in SI units, the desired relationship between input and output is described by

$$ dP_i = \frac{0.01}{2\rho} \left( \frac{\rho \cdot Q_v \sqrt{1 - \beta^4}}{3600 \cdot \varepsilon \cdot C \left( \pi \cdot d^2 / 4 \right)} \right)^2 $$

(2.5)

The following values are obtained from [9] and [10].

$\beta = 0.41004$ (-)
$\varepsilon = 0.98217$ (-)
$C = 0.995$ (-)
$\rho = 35.769$ (kg/m$^3$)

The resulting orifice characteristic, used in the model is shown in Fig. 2.5.

Figure 2.5 Characteristic of nozzle as described in [3].
2.3 Controller

Working principle

The controller has an input signal in the form of an error and an output signal between zero and unity, that is fed into the anti surge valve. The error $e$ (controller input) is calculated by

$$e = \frac{PV - SP_{\text{final}}}{PV_{\text{max}}}$$  \hspace{1cm} (2.6)

$$SP_{\text{final}} = SP + SP_{\text{incr}}$$  \hspace{1cm} (2.7)

Here, $SP_{\text{incr}}$ is an increase in setpoint ($SP$) when an additional differential action is incorporated. This differential action will be treated in paragraph 2.3.1, for now we assume there is no differential action and $SP_{\text{incr}} = 0$. The process value ($PV$) is a function of the measured pressure difference over the orifice $dP_1$ and the compressor in- and outlet pressures $P_1$ and $P_2$. The $PV$ represents the momentary operating point on the compressor characteristic and is defined by

$$PV = \frac{dP_1}{P_2 - c_s P_1}$$  \hspace{1cm} (2.8)

Here $c_s$ is a tuning parameter with a default value of 1.1467. When the compressor is at the most right point in the characteristic, on the stonewall line, $PV$ is at its maximum value ($PV_{\text{max}}$). The value of $PV$ when the compressor is just in surge is the Surge Point Value ($SPV$). To create an anti surge margin, the value of $SP$ is determined by

$$SP = c_s \cdot SPV$$  \hspace{1cm} (2.9)

The default value for $c_s$ is 1.21. The relation between these values and the compressor map becomes clear from Fig. 2.6. From (2.8) we see that $PV$ is equal to $dP_1$ divided by a pressure. The value of $dP_1$ corresponds to a volume flow, since $dP_1$ and $Q_v$ are directly related via the orifice characteristic. Looking in the compressor map, dividing a flow by a pressure, results in $\tan(a)$ with $a$ the angle between the $PV$-line and the vertical axis, as displayed in Fig. 2.6 where the relationship between $PV$, $SPV$ and $SP$ lines and the compressor map is presented.
Figure 2.6 Schematic representation of the compressor map.

Function
The objective of the controller is to keep the compressor operating in the stable region of the compressor map, with an extra margin for safety reasons. When operating in the stable region right from the SP-line, where the error $e$ is positive, the controller output is forced to zero and integrators should be reset to avoid wind-up. As the flow decreases due to a disturbance, PV decreases as well and at a certain point where $PV < SP$, the error $e$ becomes negative. Here the controller comes into action, opening the anti surge valve. This action 'pushes' the $PV$ back to the stable region at the right-hand side of the SP-line.

Modeling
The discrete controller is designed by Siemens and described in [7]. The relation between the controller output $y$ and the input $e$ is

$$y_k = y_{k-1} + K_c \left( 1 + \frac{T_s}{INT} \right) e_k - K_c \cdot e_{k-1}$$

(2.10)

With $k$ the discrete time step, $T_s$ the sample time and $K_c$ and $INT$ are tuning parameters. In order to specify these tuning parameters, a continuous transfer function is derived. First, a $z$- transformation is applied to (2.10) resulting in

$$Y(z) = z^{-1} \cdot Y(z) + \left( K_c \left( 1 + \frac{T_s}{INT} \right) - K_c \cdot z^{-1} \right) E(z)$$

(2.11)
The discrete transfer function is

\[ C(z) = \frac{Y(z)}{E(z)} = \frac{K_c \left(1 + \frac{T_s}{INT}\right) - K_c \cdot z^{-1}}{1 - z^{-1}} \] (2.12)

To convert the discrete transfer function into a continuous one, the \textit{matched pole zero} (MPZ) method is used [11]. Therefore, (2.12) is represented in a general form according to

\[ C(z) = K_c \frac{1 - az^{-1}}{1 - \beta z^{-1}} \] (2.13)

The general continuous transfer function is

\[ C(s) = K_c \frac{s + a}{s + b} \] (2.14)

Matching the pole and the zero respectively yields

\[ \beta = e^{-\alpha T_s} \quad \text{and} \quad \alpha = e^{-\alpha T_s} \] (2.15)

Combining (2.12), (2.13) and (2.15) results in

\[ b = 0 \quad \text{and} \quad -\alpha \cdot T_s = \ln \left(\frac{1}{1 + \frac{T_s}{INT}}\right) \] (2.16)

Further evaluation of (2.16) and using a Taylor first order approximation leads to

\[ -\alpha \cdot T_s = -\ln \left(1 + \frac{T_s}{INT}\right) \approx -\frac{T_s}{INT} + h.o.t. \] (2.17)

From (2.16) and (2.17) the resulting continuous pole and zero are respectively

\[ b = 0 \quad \text{and} \quad a \approx \frac{1}{INT} \] (2.18)

The \textit{Final Value Theorem} needs to give equal results for both the discrete and the continuous transfer functions, resulting in

\[ K_s = K_z = K_c \left(1 + \frac{T_s}{INT}\right) \approx K_c \] (2.19)
Combining (2.14), (2.18) and (2.19) results in the standard expression of a continuous PI controller

\[
C(s) = K_c \frac{s + \frac{1}{\text{IN}T}}{s} = K_c \left(1 + \frac{1}{\text{IN}T \cdot s}\right) \tag{2.20}
\]

From (2.20) it becomes clear that the parameter \(K_c\) is the proportional gain and \(\text{IN}T\) is the time constant. The default parameter values used in [7] are: \(K_c = 4\), \(\text{IN}T = 25\) and \(T_s = 0.05\) (s). With these parameter values, the Bode plots of both the discrete transfer function (2.12) and the continuous transfer function (2.20) are created and shown in Fig. 2.7.

![Bode Diagram](image)

Figure 2.7 Bode plots of the discrete and the continuous controller transfer function.

The breakpoint is located at \(\sigma = \frac{1}{\text{IN}T} = 0.04\) rad/sec and the discrete transfer function is cut off at the Nyquist frequency, calculated in (2.21)

\[
\sigma_{ny} = \frac{\sigma_s}{2} = \frac{2\pi \frac{1}{T_s}}{2} = \frac{\pi}{T_s} = 62.8\ \text{rad/sec} \tag{2.21}
\]
2.3.1 Differential action

Besides a proportional and an integral action, a differential (D-) action can be applied. The D-action is incorporated by changing the setpoint due to a change in flow (dF/dt). When the flow decreases rapidly (dF/dt << 0), the danger exists of the compressor going in surge. Therefore, the original setpoint (SP) is increased with SP_{incr}. As a consequence, the PV-line passes the SP-line earlier. The controller will react earlier and have a higher output, resulting in an increased control effort.

The parameter that determines the influence of dF/dt is a weighing factor, called c_{incr}. The value of this factor depends on PV/SP, and is calculated according to (2.22). Fig. 2.8 is displayed to visualize the effect of PV/SP on c_{incr}.

\[
\begin{align*}
c_{\text{incr}} &= c_{\text{incr, def}} & \text{for } PV/SP < c_6 \\
c_{\text{incr}} &= -c_{\text{incr, def}} \left( c_7 - PV/SP \right) / \left( c_6 - c_7 \right) & \text{for } c_6 < PV/SP < c_7 \\
c_{\text{incr}} &= 0 & \text{for } PV/SP > c_7
\end{align*}
\]

(2.22)

By default, the values are:  
\[
\begin{align*}
c_{\text{incr, def}} &= 0.07 \\
c_6 &= 2 \\
c_7 &= 4
\end{align*}
\]

Figure 2.8a) Relation between PV/SP and c_{incr}.

So the smaller PV/SP, the more important the D-action will be due to a higher value of the weighing factor c_{incr}. The effect of the D-action due to a decrease in flow is the increase in setpoint, resulting in a negative error earlier than without D-action, as visualized in Fig. 2.9.

14
2.4 Control valve

Working principle
The anti surge valve is a fail open solenoid valve. This means that it needs a high signal of 20 mA to close the valve, and a low signal of 4 mA to open the valve. When some failure occurs, the valve will usually receive a low signal and it will open, which is the safe position.

Function
The control valve is the actuator of the system. When opened, it directs the pressurized fluid from just behind the compressor back to the entrance of the compressor. This prevents the flow from becoming too low at the compressor inlet and decreases the plenum pressure directly behind the compressor.

Modeling
The flow through the valve $Q_{cv}$ depends on the valve position $u$ (\textdegree) and on the pressure difference over the valve $\Delta P$ (bar). The relationship between these three parameters is given by (2.23) and visualized in Fig. 2.10.

$$Q_{cv} = \frac{3600}{\rho} \left( \frac{\pi \cdot d^2}{4} \right) \cdot u \cdot C_v \sqrt{\Delta P \cdot 10^5}$$  \hspace{1cm} (2.23)
with:
\[ \rho = 35.769 \text{ (kg/m}^3\text{)} \]
\[ d = 0.0135 \text{ (m)} \]
\[ C_v = 77 \text{ (-)} \]

Figure 2.10 Characteristic of solenoid valve at \( u = 1 \).

### 2.5 Disturbances

In order to evaluate the controller behavior, different disturbances are applied on the model. Disturbances that could be encountered in reality are a (partial) shut down of the process consuming the compressor flow or a variation of the composition and thus molecular weight of the working fluid. In this case we only consider the partial shutdown of the process behind the compressor, resulting in a change in load and in operating point of the compressor. All simulations start in the stable region, in the design point of the compressor. Three situations are described:

a) a step to a point in the stable region.
b) a step to a point between the actual surge line and the surge avoidance line.
c) a step to a point in the instable region.

The new operating points of the compressor, caused by disturbances, are shown in Fig. 2.11.
Modeling

The load line is modeled as a throttle valve with a very horizontal characteristic. The valve coefficient is varied, resulting in different load lines, with different intersections with the compressor characteristic (operating points). The relation between flow and pressure is given by

$$Q_v = C_{load} \sqrt{P_2 - P_{net}} \quad (2.24)$$

For the design point and the three cases, the coefficients are given in table 2.1.

<table>
<thead>
<tr>
<th>working point</th>
<th>design point</th>
<th>a</th>
<th>b</th>
<th>c</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{load}$</td>
<td>1100</td>
<td>600</td>
<td>500</td>
<td>400</td>
</tr>
</tbody>
</table>
2.6 Plenum

The plenum is the volume between the compressor outlet and the control valve. When the control valve is opened, the stream splits into two parts. One is being recycled into the compressor inlet, the other is going towards the throttle valve. A schematic representation of the plenum is shown in Fig. 2.12.

Figure 2.12 Schematic representation of the plenum.

In order to describe the dynamics of the fluid in the plenum, the conservation of mass of a Newtonian fluid is considered [6].

\[ \frac{1}{\rho} \frac{D \rho}{Dt} + \nabla \cdot \mathbf{u} = 0 \]  \hspace{1cm} (2.25)

Integrating over the control volume \( V_p \) and using Gauss' theorem leads to

\[ V_p \frac{d \rho}{dt} = \rho u (A_{comp} - A_{cr} - A_w) = Q_{m,comp} - Q_{m,cr} - Q_{m,\nu} \]  \hspace{1cm} (2.26)

Furthermore, isentropic compression is assumed, yielding

\[ \left( \frac{\partial p}{\partial \rho} \right)_S = a^2 \]  \hspace{1cm} (2.27)

and

\[ \frac{1}{a^2} \frac{dp}{dt} = \frac{d \rho}{dt} \]  \hspace{1cm} (2.28)

Where \( a \) is the velocity of sound

\[ a^2 = \gamma ZRT \]  \hspace{1cm} (2.29)
Combining (2.26) and (2.28) leads to

\[
\frac{dp}{dt} = \frac{a^2}{V_r} \left( Q_{m,\text{comp}} - Q_{m,\text{cv}} - Q_{m,n} \right) \quad (2.30)
\]

Converting mass flow to volume flow yields

\[
\frac{dp}{dt} = \frac{\rho a^2}{3600V_v} \left( Q_{v,\text{comp}} - Q_{v,\text{cv}} - Q_{v,n} \right) \quad (2.31)
\]

In the final equation the factor \(\frac{\rho a^2}{3600V_v}\) can be considered as \(\frac{1}{\tau}\), with \(\tau\) the time constant of this first order system. Increasing the time constant (this corresponds to a larger volume), makes the system sluggish. The value of the velocity of sound \(a\) is about 370 m/s. Density is assumed constant with a value of 35.8 kg/m³. A reasonable estimate of the total volume behind the compressor is 1 m³, this leads to a time constant \(\tau = 7.345e^{-4}\) s. When all dynamics are taken into account, with for instance the Greitzer [2] model, the time dependant behavior of the compressor can be described. Here, only the dynamics due to the plenum are considered. This means the time scale is too small in comparison to the Greitzer model and reality. Another value of the time constant is chosen to make simulations possible with a sampling time equal to that of the PLC of the controller in reality. This sampling time is 0.05 s and the time constant has a value of \(\tau = 100\) s. Although the timescale of the simulation is not correct, this approach does result in a representation of the general behavior of the total system.

## 2.7 Total System

Combining all compression system components results in the total system. A schematic representation is shown in Fig. 2.13.

![Schematic representation of total system.](image)
In paragraph 2.3 the Bode plot of the modeled controller block is presented. In Fig. 2.14 the Bode plot of the linear time invariant process model block is shown, linearised round the design point \((Q_{vo}, P_{20}) = (2476, 95.07)\)

![Bode Diagram](image)

**Figure 2.14** Bode plot of the linearised, time invariant process model

From Fig. 2.14 we can conclude that the process has first order low-pass behavior. The continuous approximation deviates from the discrete bode diagram near the Nyquist frequency, this is due to numerical effects. The bandwidth of the process is 40 rad/sec or 6.37 Hz. With models for both the controller and the process, the open-loop and closed-loop behavior can be derived from (2.32) and (2.33), yielding

\[
H_{ol} = C \cdot P
\]  

(2.32)

\[
H_{cl} = \frac{H_{ol}}{1 + H_{ol}} = \frac{C \cdot P}{1 + C \cdot P}
\]  

(2.33)

The open-loop model is used in chapter 3.2 to evaluate the stability of the closed-loop system with Nyquist plots.
3 Simulation

In this chapter we discuss the approach of the simulations and present the results with additional comment.

3.1 Approach

First the goals of the simulations have to be set. Several cases are simulated to provide an answer to these four questions:

- does the controller work?
- what is the influence of the differential action?
- what is the influence of limited control valve speed?
- how do changes in controller settings effect the behavior of the system?

The three working points (a, b, c) as a result of disturbances discussed in paragraph 2.5, will be applied in all cases. The simulated signals are:

- compressor output pressure, $P_2$
- compressor flow, $V_{comp}$
- error, $e$
- control valve position, $u$
- process value, $PV$
- setpoint, $SP$

The first two questions can be answered by simulating the shift in operating points with the differential action and without the differential action (case 1). To answer the third question, simulations for various rate limits are carried out (case 2). Finally, different settings of the controller are simulated (case 3) to provide an answer to the fourth question. An overview of the simulations is presented in table 3.1.

<table>
<thead>
<tr>
<th>case description</th>
<th>case nr.</th>
<th>PV&gt;SP</th>
<th>SPV&lt;PV&lt;SP</th>
<th>PV&lt;SPV</th>
</tr>
</thead>
<tbody>
<tr>
<td>With and without D-action</td>
<td>1</td>
<td>a</td>
<td>1a</td>
<td>1b</td>
</tr>
<tr>
<td>Various rate limits</td>
<td>2</td>
<td>2a</td>
<td>2b</td>
<td>2c</td>
</tr>
<tr>
<td>Various controller settings</td>
<td>3</td>
<td>3a</td>
<td>3b</td>
<td>3c</td>
</tr>
</tbody>
</table>
3.2 Results

Here the results are presented for the cases from which we can best observe the behavior of the compressor with surge avoidance control. In these simulations, the values of important constants are: rotational speed $N = 100\%$ of design speed (12200 rpm), inlet temperature $T_i = 23$ °C, inlet pressure $P_i = 38.2$ bar, molecular weight $M = 18.42$ kg/kmol.

Case 1

In case 1 the influence of the D-action is analyzed.

![Figure 3.1 Case 1a: with and without D-action.](image)

From Fig. 3.1 we can conclude that without D-action the time to steady state operation is less then with D-action. Notice the difference in $PV$ for both situations, in Fig. 3.2 more signals are presented of this simulation and $PV$ plays a role in these signals.
In Fig. 3.2 the difference between the two situations is better observed. Recalling the difference in $PV$ seen in Fig. 3.1, it can be concluded from Fig. 3.2 that the difference in $e$ is mainly caused by the change in $SP$. Without D-action, the value of $SP$ stays constant, $e$ remains positive and $u$ remains zero, leaving the control valve closed. With D-action, the value of $SP$ is increased as a result of a fast decrease in flow and $e$ becomes negative. From this point, $u$ has a positive output, opening the control valve. When the control valve is opened, working fluid is recycled back into the inlet of the compressor, increasing to flow through the compressor. As a result $SP$ decreases, $e$ increases and $u$ decreases as well. In the situation of a disturbance causing a shift of operating point within the stable region, the D-action only has a negative influence as it opens the control valve unnecessarily.
Figure 3.3  Case 1c: with and without D-action.

In Fig. 3.3 results are presented of a simulation with a larger shift of operating point, into the instable region. Applying the D-action in this case does prevent $Q_r$ from decreasing as much as without the D-action.
Case 2

In case 2 the influence of the maximum opening rate of the control valve is considered. Ideally, the valve is infinitely fast but in practice there is a maximum rate. From Fig. 3.4 we can conclude that until a certain rate limit, the only influence of a non-ideal valve is the damping of the spike when opening. This has practically no consequences in the behavior of the compressor. Beyond a certain rate limit the valve is too sluggish and $Q_v$ increases slowly.

Figure 3.4 Case 2c: various rate limits (%/s).
Case 3
In case 3 the effect of a change in controller settings is simulated, starting with the proportional gain $K_c$.

The default value for $K_c$ is 4 and from Fig. 3.5 we can conclude this gives stable behavior. Increasing $K_c$ is expected to make the response faster and it does. However, beyond a value of 41 the controller output is instable, resulting in an oscillating control valve. The stability is analyzed in the Nyquist plots in Fig.3.6.
Figure 3.6 Nyquist diagram for various values of $K_c$.

Although the simulations show instable behavior at $K_c$ is 41, the Nyquist plot encircles the point (-1,0) for values larger than 68. A reason for this difference could be that the process is linearised in the design point and when oscillations occur in the simulation, the process is no longer operating round the design point, but on the surge avoidance line.
Next, we change the value of integration time $INT$. The default value is 25 and decreasing $INT$ is expected to result a faster response. As shown in Fig. 3.7 decreasing $INT$ results in a faster response, until the critical value of 0.003. Simulations with this critical value show oscillatory behavior that damps out. Decreasing $INT$ beyond this critical value results in consistent instable behavior.

![Figure 3.7 Case 3c: various values for $INT$.](image)

In order to check if the closed loop system is instable according to the Nyquist criterion, the Nyquist plots are presented in Fig. 3.8.
Figure 3.8 Nyquist diagram for various values of $INT$.

From Fig. 3.8 we can conclude that the closed-loop system becomes unstable for values of $INT$ smaller than 0.0015. This corresponds with our findings from the simulations.
Finally, the effect of a change of the D-action is considered. As discussed in paragraph 2.3.1, the D-action responds to a decrease in flow. The influence of this decrease is determined by the value of $c_{incr}$. By default this value is 0.07, resulting in a stable response as observed in Fig. 3.9. Decreasing the value of $c_{incr}$ results in a slower response, increasing leads to a faster response. As $c_{incr}$ is increased above the critical value of 0.16, the system is instable, showing oscillatory behavior as can be seen in Fig. 3.9.

![Figure 3.9 Case 3b: various values for $c_{incr}$](image)

The D-action is incorporated in the process model block and a change in $c_{incr}$ will not affect the controller model. Changing $c_{incr}$ does not influence the linearised model and the Nyquist stability criterion cannot be investigated.
4 Conclusions and recommendations

We recall the objectives of this internal traineeship from the introduction:

1. Development of a simple simulation model of a centrifugal compressor with surge avoidance control, based on experimental data.

2. Give a qualitative evaluation of the behavior of the modeled centrifugal compressor and surge avoidance controller for various disturbances.

The first objective is met, however the compressor model is quasi-static and it does not represent all dynamics. We recommend making a dynamic compressor model, for instance according to [2]. The second objective is partially met. Where possible, the behavior observed in the simulations is further analyzed with the Nyquist stability criterion. The conclusions that can be drawn from this internal traineeship are:

- Surge avoidance control is necessary, it however limits the operating range of the compressor.

- Applying the differential action gives a faster response, but can result in undesirable opening of the control valve.

- If the maximum opening rate of the control valve is too low, this will increase the time to steady-state.

- Adjusting the controller settings is bounded by the stability of the system.

Recommendations for further research are:

- Development of a dynamic compressor model to evaluate transient behavior.

- Investigate the effect of varying molecular weight of the compressor working fluid.
References


**Nomenclature**

\[
\begin{align*}
A & \quad \text{area} \quad m^2 \\
c_8 & \quad \text{tuning constant} \quad - \\
c_{sp} & \quad \text{anti-surge margin} \quad - \\
C & \quad \text{discharge coefficient} \quad - \\
C_{load} & \quad \text{load line coefficient} \quad - \\
C_v & \quad \text{flow coefficient} \quad - \\
d & \quad \text{inner diameter} \quad m \\
D & \quad \text{outer diameter} \quad m \\
dP_l & \quad \text{pressure difference over orifice} \quad \text{mbar} \\
e & \quad \text{error} \quad - \\
H & \quad \text{head} \quad \text{kJ/kg} \\
INT & \quad \text{integration time} \quad s \\
k_1 & \quad \text{specific heat ratio} \quad (C_p/C_v) \quad - \\
K_c & \quad \text{gain} \quad - \\
M & \quad \text{molecular weight} \quad \text{kg/kmol} \\
N & \quad \text{rotational speed} \quad \% \\
P_1 & \quad \text{inlet pressure} \quad \text{bar} \\
P_2 & \quad \text{outlet pressure} \quad \text{bar} \\
P_{net} & \quad \text{natural gas net pressure} \quad \text{bar} \\
PV & \quad \text{process value} \quad - \\
Q_m & \quad \text{mass flow} \quad \text{kg/s} \\
Q_v & \quad \text{volume flow} \quad \text{m}^3/\text{h} \\
Q_{comp} & \quad \text{volume flow through compressor} \quad \text{m}^3/\text{h} \\
Q_{cv} & \quad \text{volume flow through control valve} \quad \text{m}^3/\text{h} \\
Q_{tv} & \quad \text{volume flow through throttle valve} \quad \text{m}^3/\text{h} \\
SP & \quad \text{set point} \quad - \\
SPV & \quad \text{surge point value} \quad - \\
T_i & \quad \text{inlet temperature} \quad ^\circ\text{C} \\
T_s & \quad \text{sample time} \quad s \\
u & \quad \text{valve position} \quad \text{(open/closed = 1/0)} \quad - \\
Z & \quad \text{compressibility factor} \quad - \\
\end{align*}
\]

**Greek**

\[
\begin{align*}
\beta & \quad \text{diameter ratio} \quad (d/D) \quad - \\
\varepsilon & \quad \text{expansion factor} \quad - \\
\rho & \quad \text{density} \quad \text{kg/m}^3 \\
\omega & \quad \text{frequency} \quad \text{rad/s} \\
\omega_{Nyq} & \quad \text{Nyquist frequency} \quad \text{rad/s} \\
\end{align*}
\]