



Supercritical fluid recycle for surge control of CO₂ centrifugal compressors



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ABSTRACT

This paper presents computer-based design and analysis of control systems for centrifugal compressors when the operating fluid is supercritical CO₂.

It reports a non-linear dynamic model including a main forward compression line and two different configurations for the recycle antisurge line. Disturbance scenarios are proposed for testing the configurations and performance indicators are suggested to evaluate control performance and power consumption of the compression system.

The paper demonstrates that compared to the hot recycle, the process configuration including a cold gas recycle has better overall stability, but higher power consumption and lower values for the control performance indicators. Based on the previous considerations, the paper gives suggestions regarding the choice of the recycle configuration. Moreover it compares subcritical and supercritical compression during surge prevention and highlights the importance of the selection of the gas recycle configuration when full recycle is needed.

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1. Introduction

The compression of carbon dioxide for transportation and storage has increased in recent years. This is due to its importance within carbon capture and storage technology with large-scale transportation via pipelines (Zhang et al., 2006).

In order to increase the gas density while reducing the pressure drop along the pipeline, the gas is compressed to supercritical state via multistage centrifugal compressors. However, practical problems can arise in the operation of such a compressor, especially when dealing with gas recycle for antisurge protection.

Rotating stall and surge are two instabilities of the flow that take place inside the compressor under low flow and high pressure ratio conditions. Rotating stall involves only one sector of the cross sectional area of the compressor, where the flow recirculates instead of moving forward. During surge, the flow oscillates axially and this perturbation involves the full cross sectional area of passage. If rotating stall takes place, the efficiency of the machine decreases and this can locally damage the machine, especially the blades. If surge happens, the most likely consequence is vibration that can damage blades, casing and bearings (Greitzer, 1976).

Gas recycle is employed during start-up, shut-down and surge prevention of centrifugal compressors. During surge prevention, the gas is recycled from the outlet of the compressor to the inlet of the compressor in order to increase the inlet mass flow rate of the machine. When this practice involves the last stage of compression of carbon dioxide in a multistage configuration, the process fluid is firstly compressed from subcritical state to supercritical state inside the centrifugal compressor and then it expands from supercritical state to subcritical state along the recycle line.

Prior work on transcritical CO₂ compression operation is found from other research areas such as refrigeration plants and power generation based on a supercritical CO₂ Brayton cycle. However in the refrigeration industry the research goal is the thermal optimisation of the cycle and positive displacement machines are employed (Conboy et al., 2013; Salazar and Mendez, 2014). On the other hand, technology involving rotating dynamic compressors, such as power generation based on a supercritical CO₂ gas turbine, is still in the early stage of development (Lettieri et al., 2014). In the future there will be a need for suitable control system able to deal with the recycle of supercritical CO₂.

Therefore there is a gap in the literature regarding control of centrifugal compressors dealing with supercritical CO₂. Moreover when dealing with surge avoidance via gas recycle, common recycle configurations include hot gas recycle and cold gas recycle. The motivations for the selection of the type of recycle configuration

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Nomenclature

a_{01}	sonic velocity at ambient conditions (m s^{-1})
A_1	duct throughflow area (m^2)
c_p	specific heat at constant pressure ($\text{J kg}^{-1} \text{K}^{-1}$)
c_v	specific heat at constant volume ($\text{J kg}^{-1} \text{K}^{-1}$)
E	scaled power consumption (–)
E_{tot}	scaled overall consumption (–)
h_{01}	specific gas enthalpy at compressor inlet (J kg^{-1})
h_{02}	specific gas enthalpy at compressor outlet (J kg^{-1})
ISE_p	integral of the squared error ($\text{bar}^2 \text{s}$)
J	moment of inertia (kg m^2)
k_{ASV}	antisurge valve constant (m^2)
k_{in}	inlet valve constant (m^2)
k_{out}	outlet valve constant (m^2)
L	duct length (m)
m	inlet mass flow rate of the compressor (kg s^{-1})
m_c	corrected mass flow rate (kg s^{-1})
m_{ctrl}	control mass flow rate (kg s^{-1})
m_{in}	inlet mass flow rate of the system (kg s^{-1})
m_{out}	outlet mass flow rate of the system (kg s^{-1})
m_{pout}	outlet mass flow rate of the plenum (kg s^{-1})
m_r	recycle mass flow rate (kg s^{-1})
$m_{r,CR}$	recycle mass flow rate (cold gas recycle) (kg s^{-1})
$m_{r,HR}$	recycle mass flow rate (hot gas recycle) (kg s^{-1})
m_{surge}	surge mass flow rate (kg s^{-1})
M	scaled recycled mass flow (–)
N	rotational shaft speed (rpm)
N_c	corrected rotational shaft speed (rpm)
op_{ASV}	opening position of the antisurge valve (–)
op_{in}	opening position of the inlet valve (–)
op_{out}	opening position of the outlet valve (–)
p	pressure (bar)
p_{01}	inlet pressure of the compressor (bar)
p_{02}	outlet pressure of the compressor (bar)
p_{in}	inlet pressure of the system (bar)
p_{out}	outlet pressure of the system (bar)
p_{ref}	reference pressure (bar)
P_m	power consumption of the compressor (kW)
Q	heat flow rate (kJ)
Q_m	heat removed from the recycle flow rate (kW)
r_2	impeller radius (m)
t	time (s)
T	temperature (K)
T_{01}	inlet temperature of the compressor (K)
T_{02}	outlet temperature of the compressor (K)
T_{in}	inlet temperature of the system (K)
T_{out}	outlet temperature of the system (K)
T_{ref}	reference temperature (K)
V	plenum volume (m^3)
x_i	mass fraction of component i (–)
γ	specific heat ratio (–)
η_p	polytropic efficiency of the compressor (–)
μ	slip factor (–)
ρ	density (kg m^{-3})
ρ_{in}	inlet density of the system (kg m^{-3})
τ_d	torque of the driver (Nm)
τ_c	torque of the compressor (Nm)
ψ_c	pressure ratio of the compressor (–)
ω	angular velocity (rad s^{-1})

usually include the temperature profile of the gas along the machine and the response delay for surge protection. However this paper demonstrates that the temperature of the gas along the recycle line influences the operation of the compressor and its control. The paper represents an extension of previous work (Budinis and Thornhill, 2015) and reports on the analysis of supercritical fluid recycle with the purpose of surge prevention. It presents the model of the compression system and its control system and compares different recycle configurations during boundary disturbances. The contributions of this paper include the analysis of the influence of the recycle configuration on the stability and power consumption of the compressor. Moreover this analysis has been applied on a supercritical compressor by taking into account the effect of the pressure dependence on the thermodynamic parameters of the system.

The adopted methodology includes modelling and dynamic simulation of a single stage centrifugal compressor. Disturbance variables include the opening position of the boundary valves and the inlet pressure of the system, which is affected by the operation of the previous stages. Step disturbances have been tested and the responses of the two configurations have been recorded, analysed and compared. The simulation scenarios tested in this paper come from the literature and also from industrial practice (Dukle and Narayanan, 2003; Patel et al., 2007; Wu et al., 2007). They have been proposed in the past for the validation of compressor and anti-surge controllers. Simulation results have been analysed by means of proposed performance indicators and also by means of graphical representation of the main process variables over time.

The structure of the paper is the following. Section 2 gives a brief overview on compressor control, motivating the choice of the control system. Section 3 presents the model of the compression system, including the main process line and the recycle line according to two different configurations. Details are given regarding the performance characteristic of the compressor and the estimation of power consumption and outlet temperature. Section 4 describes the adopted control system and proposes a representation method for surge estimation. In Section 5 the case study is described together with the selection of the disturbance variables. Various performance indicators are suggested in order to evaluate the performance of the control system in terms of pressure control, gas recycle and power consumption. Section 6 presents and describes the dynamic simulations performed for the selected disturbance scenarios and analyses full recycle operation under subcritical and supercritical fluid conditions. Section 7 discusses the reported results, highlighting the factors influencing the selection of the gas recycle configuration. The paper ends with a summary and conclusion.

2. Overview of compressor control

2.1. The control system of a centrifugal compressor has two main objectives.

The *capacity controller* operates the machine in order to match either the pressure or the flow rate request from the plant. According to the controlled variable, it can therefore be called a pressure or a flow controller. Some methods to control pressure or flow include throttling discharge, throttling suction, adjusting speed, inlet guide vane manipulation and recycle manipulation (King, 2010).

The *antisurge controller* maintains the operating point of the machine inside the operating envelope, and specifically maintains the inlet mass flow rate of the compressor above the surge limit. In order to do that, it usually acts on a recycle valve, also called antisurge valve. This valve allows part of the compressed gas to be recycled back to the inlet of the compressor and therefore to increase the total inlet mass flow rate. Some methods to

measure and control surge include minimum flow upstream or maximum pressure downstream (King, 2010), compressor pressure rise versus differential across suction flow metre, pressure ratio versus actual volumetric flow, incipient surge and surge spike detection (Horowitz et al., 2006). The interaction between capacity controller and antisurge controller is a known issue in industry and is usually prevented by detuning via relative gain and decoupling (Horowitz et al., 2006).

Surge avoidance via gas recycling has been initially proposed by White (1972) and later by Botros and Henderson (1994). However recycling of gas is not the only solution in order to protect the compressor from surge. Dynamic control proposes to suppress flow perturbations by means of active control (Epstein et al., 1989), aiming at actively suppressing surge by action of an actuator, or passive control (Gysling et al., 1991), aiming at passively suppressing surge by absorbing and then dissipating flow oscillations generated by surge. More recent examples of dynamic control of compressors include Gu et al. (1999), Willems and de Jager (1998, 1999), Gravdahl and Egeland (1999), Gravdahl et al. (2002), Blanchini and Giannattasio (2002), Arnulfi et al. (2006), Boinov et al. (2006) and Bohagen and Gravdahl (2008).

In 2011 Uddin and Gravdahl (2011) proposed an active control system based on piston actuation. However their work was further extended (Uddin and Gravdahl, 2012) in order to include a back-up system in case of failure of the active control system. In this second publication, the authors commented that the active surge control system was mainly implemented in university laboratories and, although successful, it was not yet implemented in any industrial application. Among the reasons, they highlight the safety and reliability of such a system. The reason is that an active control system pushes the operating point in an area of the compressor map that is open loop unstable, i.e. on the left of the surge line. In case of failure of the active control system, the compressor would go immediately into surge. For this reason, in this paper a feedback control system based on gas recycle has been adopted for surge prevention.

3. Model of the compressor

3.1. Forward compression line

The model of the compression system presented in the paper consists of a non-linear dynamic model based on a model present in the literature (Greitzer, 1976) that has been modified by other authors in order to include the shaft dynamics (Fink et al., 1992) and the equations for the torque of the compressor and of the electric driver (Gravdahl and Egeland, 1999). The compression system includes a main forward process line and a secondary recycle line.

The layout of the main compression line includes the single stage compressor, defined as a single shaft, single casing, multi-impeller machine, with its variable speed driver, the duct connecting the compressor with the downstream plenum and the outlet throttle valve, as represented in Fig. 1. In this schematic representation of the compression line, the variables p_{01} and T_{01} are respectively the pressure and the temperature of the gas upstream of the compressor while p_{02} and T_{02} are respectively the pressure and the temperature of the gas downstream of the compressor. The condition of the gas inside the plenum is defined by its pressure p , temperature T and density ρ . The variable p_{out} is the pressure downstream the outlet valve, and in an open loop configuration $p_{01} = p_{out}$ and $T_{01} = T_{out}$.

The model of this compression system is represented by first principle equations including the mass and the momentum balance of the compressor (Eqs. (1) and (2)), the moment of momentum balance of the rotating shaft (Eq. (3)), the compressor torque (Eq. (5)) and characteristic (Eqs. (6) and (7)) and the equation of the flow

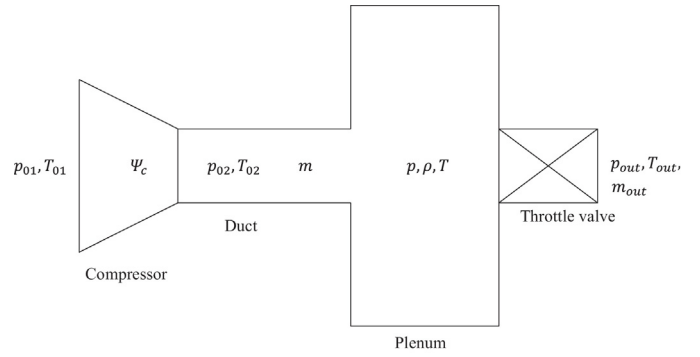


Fig. 1. Model of the main compression line.

through the outlet valve (Eq. (9)). The variables include the mass flow rate through the duct m and through the outlet valve m_{out} , the radial velocity of the shaft ω , the opening position of the outlet throttle valve op_{out} , the driver torque τ_d and the compressor torque τ_c . Parameters include the sonic velocity at ambient conditions a_{01} , the volume of the plenum V , the cross section area of the duct A_1 and its length L , the inertia of the overall driver-compressor system J , the slip factor μ , the radius of the impeller r_2 and the constant of the outlet throttle valve k_{out} .

$$\frac{dp}{dt} = \frac{a_{01}^2}{V}(m - m_{out}) \quad (1)$$

$$\frac{dm}{dt} = \frac{A_1}{L}(\psi_c p_{01} - p) \quad (2)$$

$$\frac{d\omega}{dt} = \frac{1}{J}(\tau_d - \tau_c) \quad (3)$$

$$\omega = \frac{2\pi N}{60} \quad (4)$$

$$\tau_c = \mu r_2^2 \omega m \quad (5)$$

$$\psi_c = \psi_c(\omega, m) \quad (6)$$

$$\psi_c = \frac{p_{02}}{p_{01}} \quad (7)$$

$$\eta_p = \eta_p(\omega, m) \quad (8)$$

$$m_{out} = op_{out} k_{out} \sqrt{\rho(p - p_{out})} \quad (9)$$

$$\rho = \rho(p, T) \quad (10)$$

The original configuration represented in Fig. 1 has been modified in order to include an inlet valve. Both inlet and outlet valves are not proper control valves but they rather represent throttle elements in the plant layout. The flow through the inlet valve has been estimated according to Eq. (11), where p_{in} , T_{in} and ρ_{in} define the thermodynamic state (pressure, temperature and density) of the gas upstream of the inlet throttling element while op_{in} represents the opening position of the inlet valve.

$$m_{in} = op_{in} k_{in} \sqrt{\rho_{in}(p_{in} - p_{01})} \quad (11)$$

$$\rho_{in} = \rho_{in}(p_{in}, T_{in}) \quad (12)$$

Two different configuration scenarios have been modelled. The first scenario includes a cold gas recycle where the recycled gas is cooled from the outlet temperature of the compressor T_{02} to the same temperature as the freshly fed gas T_{in} . The second scenario includes a hot gas recycle where the gas is not cooled before being recycled and therefore affects the inlet temperature of the compressor T_{01} . Both scenarios have been represented in Fig. 2.

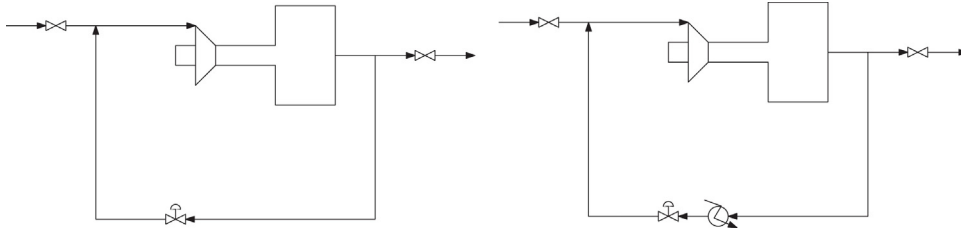


Fig. 2. Model of the compression system including main line and hot (left) or cold (right) gas recycle line.

3.2. Compressor characteristic

The compressor characteristic ψ_c represents the pressure ratio between the compressor outlet pressure p_{02} and the compressor inlet pressure p_{01} and is a function of ω and m . The compressor characteristic is usually provided by the compressor supplier. It is evaluated through steady-state tests in which the compressor runs at various rotational shaft speeds and its performance is recorded. These steady-state performance maps are only valid at the inlet reference conditions, i.e. at the same constant pressure, temperature and gas composition in which the performance tests were conducted.

When the inlet conditions of the gas change over time, the use of corrected performance maps is more appropriate. These maps take into account the influence of transient inlet conditions on the performance of the machine, according to the equation of the corrected mass flow rate m_c (13) and the equation of the corrected rotational shaft speed N_c (14).

$$m_c = \frac{m \sqrt{T_{01}/T_{ref}}}{p_{01}/p_{ref}} \quad (13)$$

$$N_c = \frac{N}{\sqrt{T_{01}/T_{ref}}} \quad (14)$$

The parameters p_{ref} and T_{ref} are respectively the reference pressure and temperature. They must be constant and can be arbitrary selected. Corrected compressor maps have been employed in the present work in order to include into the model the influence of p_{01} and T_{01} on the performance of the compressor and they have been employed in order to estimate the location of the surge line. The corrected compressor maps have been implemented in the simulation tool by means of second order polynomial functions.

3.3. Power consumption

The thermodynamic power consumed by the compressor during the compression of the gas from the pressure p_{01} to the pressure p_{02} depends on the mass flow rate and on its inlet and outlet thermodynamic state, according to Eq. (15), where h_{01} and h_{02} are respectively the inlet and outlet specific enthalpies of the gas (Camporeale et al., 2006). The value of h has been estimated as a function of both pressure and temperature of the gas. The outlet temperature of the compressor T_{02} has been estimated by means of Eq. (17), where η_p is the polytropic efficiency of the compressor and $\gamma = c_p/c_v$ is the heat ratio.

$$P_m = m(h_{02} - h_{01}) \quad (15)$$

$$h = h(p, T) \quad (16)$$

$$T_{02} = T_{01} \left[1 + \frac{1}{\eta_p} \left(\psi_c^{(\gamma-1)/\gamma} - 1 \right) \right] \quad (17)$$

3.4. Hot gas recycle configuration

In the hot gas recycle configuration, part of the compressed gas is recycled back to the inlet of the compressor without any treatment or modification. Therefore hot gas is recycled and this gives the name to this type of configuration. This configuration includes two nodes. In the first node the freshly fed gas m_{in} mixes with the recycled gas $m_{r,HR}$ while in the second node the compressed gas m_{pout} splits between recycled gas $m_{r,HR}$ and delivered gas m_{out} .

Eqs. (2)–(12) are still valid, however Eq. (1) has been substituted by Eq. (18). Eqs. (19) and (20) represent the mass balances for respectively the first and second node while Eq. (21) represents the mass flow rate through the antisurge valve characterised by the constant parameter k_{ASV} and opening position op_{ASV} .

$$\frac{dp}{dt} = \frac{a_{01}^2}{V} (m - m_{pout}) \quad (18)$$

$$m_{in} + m_{r,HR} = m \quad (19)$$

$$m_{pout} = m_{out} + m_{r,HR} \quad (20)$$

$$m_{r,HR} = op_{ASV} k_{ASV} \sqrt{\rho(p - p_{01})} \quad (21)$$

The temperature of the outlet gas can still be estimated using Eq. (17). However the compressor inlet temperature T_{01} is not constant but depends on the system inlet temperature T_{in} , inlet flow rate m_{in} , recycle flow rate $m_{r,HR}$ and recycle temperature T_r . A hot gas recycle implies that $T_r = T_{02}$. The temperature of the gas at the inlet of the compressor, T_{01} , has been estimated by means of Eq. (22), which takes into account the possibility that c_p varies with temperature, according to Eq. (23), where i represents the number of component in the gas mixture. These equations are valid only for subcritical fluid, i.e. gas.

$$m_{in} \int_{T_{ref}}^{T_{in}} c_p(T) dT + m_{r,HR} \int_{T_{ref}}^{T_{02}} c_p(T) dT = m \int_{T_{ref}}^{T_{01}} c_p(T) dT \quad (22)$$

$$c_p(T) = \sum_{i=1}^2 x_i c_{p,i}(T) \quad (23)$$

3.5. Cold gas recycle configuration

In the cold gas recycle configuration, the recycled gas is cooled to the same temperature as the freshly fed gas. In order to do that, a gas cooler is placed in the recycle line. Therefore the mass flow of gas recycle depends on the condition of the gas at the outlet of the gas cooler, according to Eq. (24), where the condition of the gas downstream of the cooler is defined by its pressure p_{HE} and density ρ_{HE} . It has been assumed that the pressure drop through the cooler is negligible and therefore Eq. (24) gives the mass flow rate through the antisurge valve.

$$m_{r,CR} = k_{ASV} \sqrt{\rho_{HE}(p_{HE} - p_{01})} \quad (24)$$

In the cold gas recycle configuration, T_{01} is constant as ideal cooling is assumed. The heat removed from the recycled mass flow rate

$m_{r,CR}$ in order to cool it from T_{02} to T_{01} is defined by Eq. (25). The heat removed from the recycled gas has a negative sign as it is removed from the system. The variable $h_{r,CR}$ is the specific enthalpy of the gas at the outlet of the gas cooler, and by assumption $h_{r,CR} = h_{01}$.

$$-Q_m = m_{r,CR}(h_{02} - h_{r,CR}) \quad (25)$$

3.6. Supercritical carbon dioxide

When a fluid is above its critical pressure and temperature, its behaviour deviates significantly from that of an ideal gas. In particular a supercritical fluid is characterised by high density and low viscosity. Therefore it has some properties in common with the gas phase and some properties in common with the liquid phase. Other physical properties that change remarkably across the critical point include enthalpy, specific heat, thermal conductivity (Kim et al., 2014) and solubility (McHardy and Sawan, 1996).

The most commonly employed equations of state for carbon dioxide in supercritical state include the Peng–Robinson equation of state (Peng and Robinson, 1976), the Soave–Redlich–Kwong equation of state (Soave, 1972) and the Span and Wagner equation of state (Span and Wagner, 1996). The Span and Wagner equation of state gives good prediction for pure or almost pure carbon dioxide (Aursand et al., 2013). For this reason it has been employed in the past for the three-dimensional computational fluid dynamic analysis of a high speed centrifugal compressor operating with supercritical carbon dioxide (Pecnik et al., 2012) and it has been selected also for the current study. This equation has been employed in order to define specific enthalpy and density of the gas depending on its pressure and temperature.

The specific enthalpy of the gas has been implemented in the simulation tool by means of a look-up table with linear interpolation. As the dimension of the table has a size limit due to computational issues, the look-up table employed covered the temperature range 330.15–523.15 K and the pressure range 40–260 bar (Anwar and Carroll, 2010). The sampling resolution of the look-up table is variable and depends on the closeness of the operating point to the critical point. The sampling resolution of the pressure varies between 1.2 bar and 10 bar, while the sampling resolution of the temperature varies between 1 K and 5 K. The highest sampling resolution is employed around the critical point.

Also the density has been implemented in the simulation tool by means of a look-up table with linear interpolation. The dimension and sampling resolution of the lookup table are the same as those of the enthalpy look-up table.

The temperature at the outlet of the compressor has been estimated by means of the map of the compressor reporting this variable as a function of inlet mass flow rate and rotational shaft speed. Also in this case, the map has been implemented in the simulation tool by means of second order polynomial function.

4. Model of the control system

4.1. Control system

The proposed control system of the compressor includes the two control loops presented above. The reason for employing a traditional multiple-loop control system is that this is the most popular control system adopted in industrial application. The capacity control loop controls the compressor outlet pressure by manipulating the torque of the driver. The antisurge control loop avoids surge by opening the antisurge valve when the compressor mass flow rate falls below its minimum value. The overall control system is represented in Fig. 3.

The capacity controller has a cascade structure with a speed controller in the slave loop and a pressure controller in the master loop.

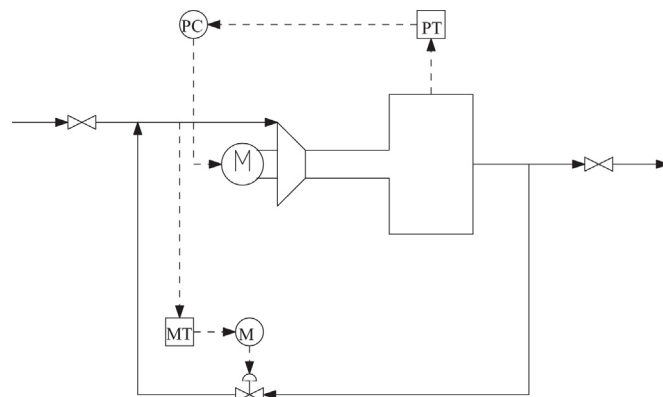


Fig. 3. Overall control system for a hot gas recycle compression station.

The manipulated variable of the capacity controller is the torque provided by the driver. The antisurge control is a feedback controller and it acts on the antisurge valve when the compressor inlet mass flow rate m is lower than the control mass flow rate m_{ctrl} . The safety margin between the surge line, represented by the surge mass flow rate m_{surge} , and the control line, represented by the control mass flow rate m_{ctrl} , is 20%. The control system has been initially tuned by means of the lambda tuning technique (Dahlin, 1968). The initial tuning parameters have been then refined by means of dynamic simulation and testing of the closed loop response of the compression system.

Pressure and surge control loops are acting on the same process system and therefore they can influence each other. The manipulation of the driver torque affects both inlet and outlet pressures, being the mass flow rate the same. At the same time a change in the upstream or downstream pressure can reduce the margin between the surge line and the operating point. The implications of the interaction are being investigated in the case study.

4.2. Representation of dynamic surge margin

A single speed compressor is a compressor running always at the same speed while a multiple speed compressor runs between its minimum rotational shaft speed and its maximum rotational shaft speed. These two boundary speeds usually depend on the driver of the compressor. For a single speed machine, the surge point is the point on the performance map with minimum flow. Forward operation takes place if the operating point of the compressor is on the right of the surge point. On the left of the surge point rotating stall and surge take place, until the compressor goes into deep surge and the flow reverses inside the compressor. For a multiple speed machine, the surge line is defined by the connections between the surge points for the different speed lines.

Surge points are defined by the compressor supplier by running the machine at constant speed and either decreasing the compressor inlet mass flow rate, increasing the compressor outlet pressure or decreasing the compressor inlet pressure, until surge can be detected by a decrease of the machine performance, noise or vibration. Usually similar tests are performed by the client in order to validate the compressor map in situ. These compressor maps represent the operation of the compressor at different speeds.

The surge mass flow rate is a function of radial speed of the shaft and inlet mass flow rate of the compressor. The distance between the surge line and the control line is defined by a control margin, as previously stated. Usually both surge and control lines are plotted on the compressor map and therefore their distance from the operating point is easily identifiable. However this type of visualisation, even if very common in both academia and industry, can

be misleading. The reason is that the surge line depends on both inlet pressure and temperature and therefore is affected by process disturbances and also by the opening of the antisurge valve.

Corrected compressor maps can be useful when the inlet conditions are different from the reference conditions, but not when they continuously change over time as happens during a process disturbance. Therefore a different way to visualise the proximity to surge is suggested in this paper. The proximity of the machine to surge is here represented as the distance between the inlet mass flow rate of the compressor m and the surge control mass flow rate m_{surge} . This distance is not constant and therefore this type of representation is a valuable qualitative and quantitative instrument for the control engineer. It allows a quick estimation of the distance between the operating point of the machine and the surge region over time. It also helps understanding the influence of inlet boundary disturbances on the surge line. Examples of this type of representation include Figs. 6 and 11, which will be presented more in detail in Section 6.

4.3. Implementation

The compression model has been implemented in MATLAB Simulink® environment. The solver ode45 is the default solver for models with continuous states and it has been selected for the resolution of the model equations. This Runge–Kutta solver is a fifth-order method that performs a fourth-order estimate of the error.

5. Case study

5.1. Description of the case study

The case study has been provided by a research partner of the project and is based on a multistage variable speed centrifugal compressor arranged in a single shaft configuration and driven by an asynchronous electric motor. The multistage compressor consists of four single stage compressors in series. There is a gear box between the second and the third stage of compression and therefore the first two single stage compressors run at the same speed while the second two single stage compressors run at another speed. The ratio between these two speeds is constant. The single stages have been approached separately however they belong to the same machine.

The fourth compression stage is the last stage of compression and is also the stage compressing the process fluid from subcritical condition to supercritical condition. The mathematical model of this stage has been employed for analysing the effect of the recycle configuration on the compressor operation when dealing with supercritical state (Sections 6.1–6.4). The mathematical model of the third stage of compression has been employed in order to compare subcritical compression and supercritical compression during full-recycle operation (Section 6.4). The third stage has the same model and runs at the same rotational shaft speed as the fourth stage of compression.

The process fluid is a mixture of carbon dioxide and water with small percentages of light hydrocarbons. This composition is typical of a gas coming from amine absorption of CO₂ from natural gas (Rufford et al., 2012).

The parameters that were included in the model are the sonic velocity at ambient conditions a_{01} , the plenum volume V , the duct throughflow area A_1 , the duct length L , the total moment of inertia of the system J , the slip factor μ , the impeller radius r_2 , the inlet, outlet and antisurge valves constants, respectively k_{in} , k_{out} , k_{ASV} , the reference pressure p_{ref} and temperature T_{ref} and the average specific heat ratio γ_{avg} . The values of these quantities are subject to a non-disclosure agreement with the partners of the project. However

Table 1

Parameter values of the model of the compressor.

Parameter	Parameter value
a_{01}^2/V (s ⁻² m ⁻¹)	0.0006
A_1/L (m)	0.0025
J (kg m ²)	2
μr_2^2 (m ²)	0.01–0.05
k_{in} (m ²)	0.1–2.5
k_{in} (m ²)	0.1–2.5
k_{ASV} (m ²)	0.1–2.5

Table 1 reports estimated parameters together with typical ranges of values for the other parameters.

The geometrical parameters of the compressor have been estimated by the geometry of the machine while the valve constants have been estimated by mean of steady state data. The composition of the gas and its average specific heat ratio were known parameters. After the analyses have been completed, the results have been scaled according to the inlet conditions of each compression stage. This is because of the non-disclosure agreement with the partners of the project.

5.2. Disturbance analysis and selection

In the next section, the proposed control system is tested under a variety of disturbance scenarios in order to demonstrate the effect on the CO₂ gas recycle. Disturbance variables include the opening position of the inlet valve op_{in} , the opening position of the outlet valve op_{out} and the inlet pressure of the system p_{in} . The types of the disturbances include negative (for the boundary valves) and positive and negative (for the inlet pressure) step changes. The magnitudes of the disturbances have been selected taking into account the variation of inlet pressure and temperature during off-load operation and the effect of the disturbance on the system, such as gas recycling and surge occurrence. Step changes have been employed for all the disturbance variables. All the simulations have been performed for 1800 s and all disturbances started at time $t = 10$ s.

5.3. Evaluation of performance

In order to estimate the performance of the controller and also the effect of the recycle configuration on the operation of the compressor, some performance indicators have been proposed and employed.

The performance of the controller in terms of pressure control has been estimated by employing the integral of the square error of the pressure with respect to its set point (ISE_p). This performance indicator has been defined according to Eq. (26), where t_{final} is the duration of the period under analysis, p is the compressor outlet pressure and p_{sp} is the set point of p . The error is the difference between the outlet pressure of the compressor p and its set point p_{sp} over time. In an ideal situation ISE_p would be equal to 0.

$$ISE_p = \int_{t=0}^{t_{final}} (p(t) - p_{sp}(t))^2 dt \quad (26)$$

The performance of the controller in terms of surge control has been estimated by graphical representation of the compressor inlet mass flow rate m and the surge mass flow rate m_{surge} . Surge took place when the lines representing these two variables crossed each other.

The power consumption of the machine has been estimated by employing the parameters M and E . The scaled mass flow M calculates the amount of gas that has been recycled over the time interval $t = 0$ to $t = t_{final}$. The amount of gas recycled is divided by the amount of gas that is compressed during 1 h of forward operation at steady state and at the design point. Variable m_r is the recycle mass flow

Table 2

Performance measures for disturbance scenarios for the inlet valve of the system.

Disturbance scenario	Hot gas recycle			Cold gas recycle		
	1	2	3	1	2	3*
Sign and magnitude (%)	–20.00	–65.00	–79.00	–20.00	–65.00	–79.00
ISE_p (bar ² s)	0.11	23.37	6220.31	0.11	23.39	9252.67
M (–)	0.00	0.00	0.02	0.00	0.00	0.63
E (–)	0.50	0.55	0.42	0.50	0.55	0.44
Q (–)	0.00	0.00	0.00	0.00	0.00	–0.28
E_{tot} (–)	0.50	0.55	0.42	0.50	0.55	0.72

rate while m is the inlet compressor mass flow rate. Because in this case the analysed time interval is 1 h, M would be 0 if there was no recycle and would be 1 in full recycle operation. The parameter is estimated by means of Eq. (27).

E is the scaled power consumption of the compressor. This parameter calculates the energy consumption of the compressor during the time interval $t=0$ to $t=t_{final}$ with respect to the power consumption of the machine during 1 h of operation at steady state and design point. The parameter is estimated by means of Eq. (28). If E is equal to 1 this means that the compressor has either been running at steady state and design point or that it is consuming as much as it would in that condition. Therefore this parameter is useful especially when used in conjunction with M . For example, if E increases but M does not this means the higher power consumption is not due to the gas recycle.

$$M = \frac{\int_{t=0}^{t_{final}} m_r dt}{\int_{t=0}^{t=1 \text{ h}} m dt} \quad (27)$$

$$E = \frac{\int_{t=0}^{t_{final}} P_m dt}{\int_{t=0}^{t=1 \text{ h}} P_m dt} \quad (28)$$

The parameter E_{tot} is defined according to Eq. (29). In the hot gas recycle configuration E_{tot} depends only on P_m while in the cold gas recycle configuration it depends on both P_m and Q_m . Q_m is always equal to 0 or negative as the heat flow is leaving the system and not provided to the system.

$$E_{tot} = \frac{\int_{t=0}^{t_{final}} (P_m - Q_m) dt}{\int_{t=0}^{t=1 \text{ h}} P_m dt} \quad (29)$$

Graphical representations have also been employed in order to compare the response of the two system configurations over time.

6. Dynamic simulations and results

6.1. Disturbance scenarios involving the inlet valve

6.1.1. Hot gas recycle configuration

Table 2 reports the results of the hot (left) and cold (right) gas recycle configurations when the inlet valve of the system closes from the fully open position to partially open position. The sign and magnitude of the disturbances are indicated in the table. The table also summarises ISE_p , M , E , Q and E_{tot} for each disturbance scenario. Scenarios indicated with * experienced surge, which took place when the compressor inlet flow rate m and the surge mass flow rate m_{surge} crossed each other. During surge, the values of the performance parameters have been reported in the tables however it has been assumed that the compressor did not fall into deep surge, which would have involved negative flow rate.

The results can be summarised in the following way. ISE_p increases with the increase of the magnitude of the disturbance. M is equal to 0 for the first two scenarios and then becomes positive when the antisurge valve opens, increasing with the increase

of the magnitude of the disturbance as more recycled gas is necessary. E increases for the first two scenarios while it decreases during the last scenario. This is because in scenarios 1 and 2 the inlet temperature of the compressor remains constant while the inlet mass flow rate decreases slightly and the compressor outlet temperature increases, therefore the power consumption increases. However during scenario 3 the mass flow rate through the compressor is much smaller than its design value and this affects P_m and therefore E . Q is always equal to 0 as the hot gas recycle configuration is adiabatic and there is no heat removal during its operation. Consequently E_{tot} is equivalent to E .

In the hot gas recycle configuration, when m and m_{ctrl} cross each other, the antisurge valve opens. When it opens, both the compressor inlet pressure p_{01} and the inlet temperature T_{01} increase. However the temperature changes significantly, while the pressure change is small and therefore this affects the surge flow rate. In particular the surge mass flow rate m_{surge} decreases and therefore the control mass flow rate m_{ctrl} decreases as well. This is because when the antisurge valve opens, both T_{01} and p_{01} increase, while $m_{surge,c}$ remains constant due to the definition of corrected compressor map and therefore m_{surge} decreases. The consequence is that m_{ctrl} becomes smaller than m and the antisurge controller closes the antisurge valve. However this action decreases p_{01} and T_{01} and when m_{ctrl} becomes bigger than m the cycle repeats.

6.1.2. Cold gas recycle configuration

The results relative to the cold gas recycle configuration can be summarised in the following way. For disturbance scenarios 1 and 2 there is no difference between the response of the cold gas configuration and the hot gas configuration as the antisurge valve does not open. However when the action of the antisurge controller opens the antisurge valve (scenario 3), the response of the two configurations differs. In particular the power consumption of the cold gas recycle configuration is higher (E_{tot} , Table 2, eighth row) and this is due to both the power consumption due to compression (E) and the heat removal required along the recycle line (Q). Moreover the cold gas recycle configuration has a lower performance in terms of pressure control and this is represented by the parameter ISE_p (Table 2, fourth row), which is higher than in the hot recycle configuration.

In the cold gas recycle configuration, when m and m_{ctrl} cross each other and the antisurge valve opens, the inlet pressure of the compressor p_{01} increases while the inlet temperature of the compressor T_{01} does not. This is because the recycled gas is cooled down from the outlet temperature of the compressor T_{02} to the same temperature as the freshly fed gas T_{in} . Therefore the compressor inlet temperature T_{01} is constant and equal to T_{in} . For this reason m_{ctrl} increases even further and therefore the antisurge valve remains open. This causes the decrease of the compressor outlet pressure p and therefore the pressure controller increases the driver torque. This action increases m until it becomes bigger than m_{ctrl} . At this point the antisurge valve closes, the compressor outlet pressure increases and the pressure controller decreases the drive torque. The consequence is that the compressor inlet mass flow rate m decreases until it crosses m_{ctrl} , the antisurge valve opens again and

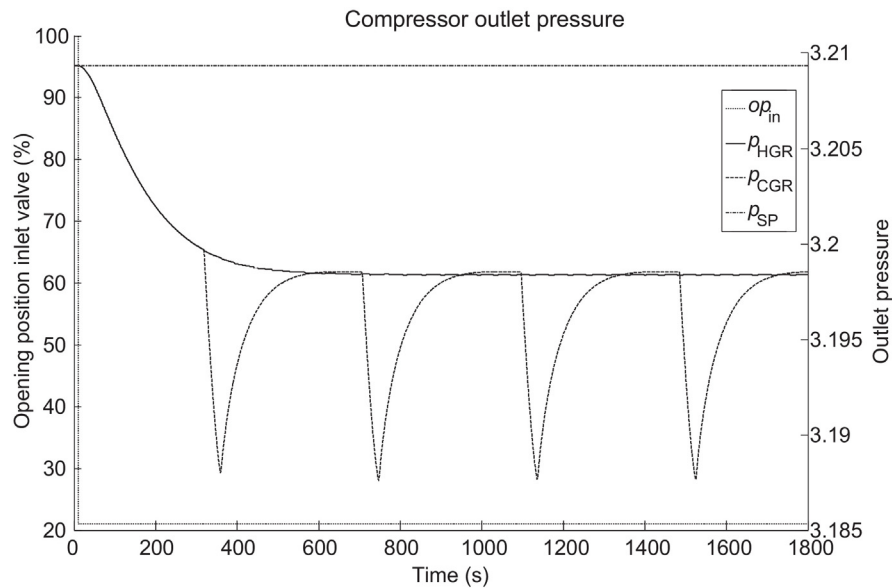


Fig. 4. Compressor outlet pressure (disturbance scenario 3).

the cycle repeats. The reason why m_{surge} increases when the anti-surge valve is closed is that m is decreasing while p is increasing. Therefore the pressure ratio is increasing and m_{surge} too.

6.1.3. Comparison of response of recycle configurations

Fig. 4 compares the compressor outlet pressure of the hot gas recycle configuration (p_{HGR}) and of the cold gas recycle configuration (p_{CGR}) during disturbance scenario 3. For the same disturbance scenario, Fig. 5 shows the manipulated variables of the system. As a summary, both recycle configurations oscillate because of the boundary disturbance and in particular the antisurge valve opens and closes over time. However the frequency of oscillation of the antisurge valve in the cold recycle configuration is lower than the frequency of oscillation of the same valve in the hot recycle configuration.

6.2. Disturbance scenarios involving the outlet valve

6.2.1. Hot gas recycle configuration

Table 3 reports the results of the hot (left) and cold (right) gas recycle configurations when the outlet valve of the system closes

from the fully open position to a partially open position. For the hot gas recycle configuration, ISE_p increases with the increase of the magnitude of the disturbance. M is equal to 0 for the first scenario and then becomes positive when the antisurge valve opens, increasing with the increase of the magnitude of the disturbance as more recycled gas is necessary. E decreases and this is mainly because m decreases over time due to the decrease of the rotational shaft speed N . Q is always equal to 0 and therefore E_{tot} is equivalent to E .

In the hot recycle configuration, m and m_{ctrl} cross each other when the action of the pressure controller decreases the torque of the driver in order to decrease the rotational shaft speed of the machine and consequently the compressor outlet pressure. The antisurge valve oscillates at high frequency and this is because of the effect of the hot recycled gas on the location of surge (Fig. 8).

6.2.2. Cold gas recycle configuration

The results relative to the cold gas recycle configuration can be summarised in the following way. For disturbance scenario 4 (Table 3, fifth column) there is no difference between the response of the cold gas configuration and the hot gas configuration as the

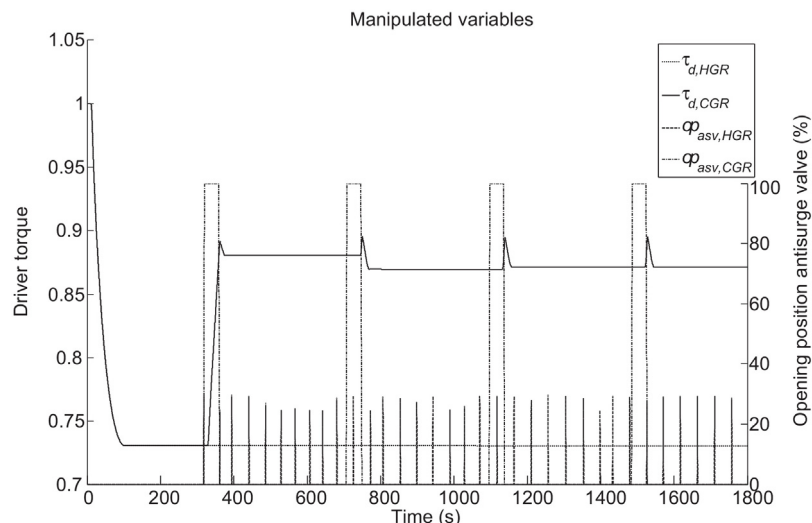


Fig. 5. Manipulated variables (disturbance scenario 3).

Table 3

Performance measures for disturbance scenarios for the outlet valve of the system.

Disturbance scenario	Hot gas recycle			Cold gas recycle		
	4	5	6	4	5	6
Sign and magnitude (%)	–15.00	–21.00	–24.00	–15.00	–21.00	–24.00
ISE_p (bar ² s)	31.39	76.61	164.83	31.39	1553.64	2026.46
M (–)	0.00	0.02	0.10	0.00	0.61	0.74
E (–)	0.42	0.39	0.37	0.43	0.43	0.42
Q (–)	0.00	0.00	0.00	0.00	–0.26	–0.32
E_{tot} (–)	0.42	0.39	0.37	0.42	0.69	0.74

antisurge valve does not open. However, when the action of the antisurge controller opens the antisurge valve (scenarios 5 and 6), the response of the two configurations differs. In the cold gas recycle configuration, ISE_p increases with the increase of the magnitude of the disturbance. M is equal to 0 for the first scenario and then becomes positive when the antisurge valve opens. It also increases with the increase of the magnitude of the disturbance as higher recycled gas is necessary. E decreases with the increasing of the disturbance magnitude while Q increases with the increase of the amount of gas recycled.

In the cold gas configuration, the opening of the antisurge valve increases m_{surge} and therefore m_{ctrl} increases. The consequence is that the antisurge valve stays open until the compressor outlet pressure p decreases (Figs. 7 and 8). When this happens, the pressure controller increases the driver torque and the consequence is the increase of m that become bigger than m_{ctrl} . Therefore the antisurge valve closes. This oscillation is represented in Fig. 6 for the disturbance scenario 5.

6.2.3. Comparison of response of recycle configurations

When comparing the hot configuration with the cold configuration, the hot recycle configuration has a better pressure control performance. It has a lower ISE_p than the cold gas recycle configuration. This is due to the oscillation of the cold recycle system due to the interaction between the control loops. The hot recycle configuration shows also smaller overall power consumption. This is due to both E and Q . The reason is that the mass flow rate through the compressor m is lower in the hot gas recycle configuration. Moreover this configuration does not require the cooling of the recycled gas. None of the tested disturbance scenarios caused the surge of

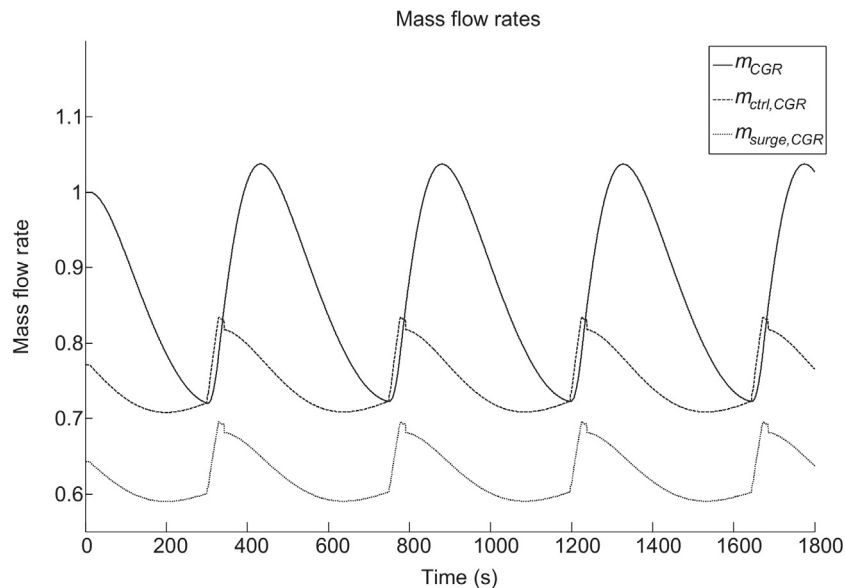
the compressor, in any of the recycle configurations. The cold configuration recycles more gas that in turn has to be cooled down from T_{02} to T_{in} . Therefore its overall power consumption, expressed by means of the parameter E_{tot} , is higher than the power consumption of the hot recycle configuration. Moreover the control performance in terms of pressure control, expressed by the parameter ISE_p , is lower than the performance of the hot recycle configuration.

Fig. 7 compares the compressor outlet pressure of the two process configurations during disturbance scenario 5. During the same disturbance scenario, Fig. 8 shows the manipulated variables of the two configurations. As a summary, it is possible to say that the disturbance scenario 5 has caused the oscillation of both hot and cold recycle configurations. This oscillation was mainly caused by the effect of the recycle on the location of the surge region. However in the cold recycle configuration it was also caused by the interaction between pressure controller and surge controller. Therefore the result is that the hot gas recycle oscillates at high frequency for a short period of time and then stabilises at a new set point. The cold recycle configuration instead continues to oscillate, however at a lower frequency.

6.3. Disturbance scenarios involving the inlet pressure of the system

6.3.1. Hot gas recycle configuration

Table 4 reports the results of the hot (left) and cold (right) gas recycle configurations when the inlet pressure of the system p_{in} changes following negative (scenario 7) or positive (scenarios 8 and 9) step changes. For the first scenario, where the inlet pressure of the system decreases, M is equal to 0 as m is always above m_{ctrl} and

**Fig. 6.** Mass flow rates (disturbance scenario 5).

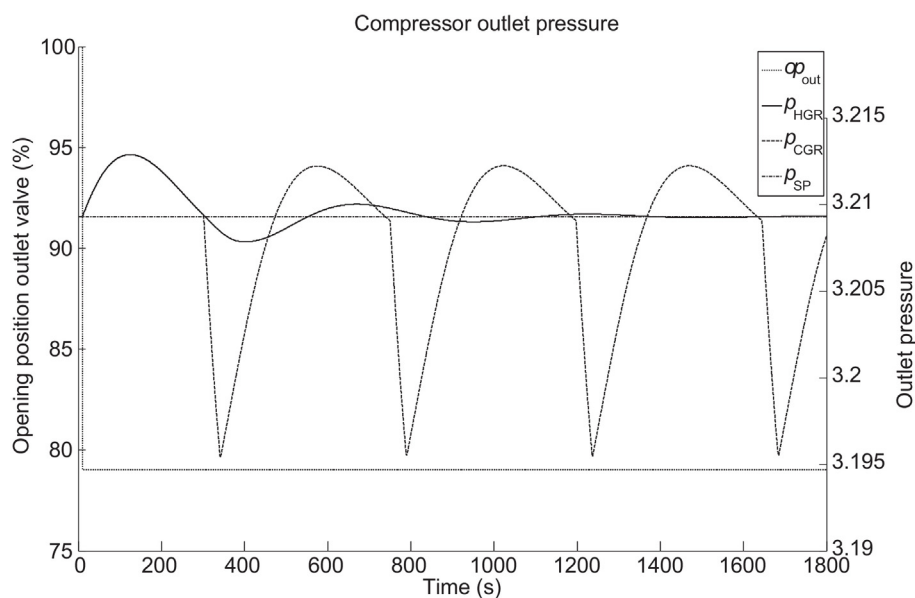


Fig. 7. Compressor outlet pressure (disturbance scenario 5).

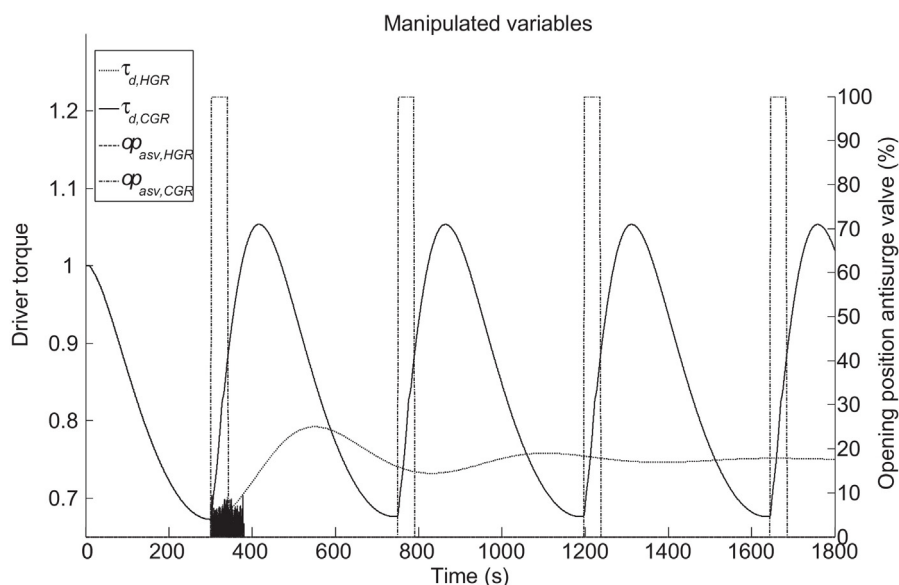


Fig. 8. Manipulated variables (scenario 5).

Table 4

Performance measures for disturbance scenarios for the inlet pressure of the system.

Disturbance scenario	Hot gas recycle			Cold gas recycle		
	7	8	9*	7	8	9*
Sign and magnitude (%)	−14.89	44.68	53.19	−14.89	44.68	53.19
ISE_p (bar ² s)	31.60	27.73	37.62	31.60	104.30	156.40
M (−)	0.00	0.02	0.04	0.00	0.11	0.14
E (−)	0.55	0.55	0.62	0.55	0.56	0.64
Q (−)	0.00	0.00	0.00	0.00	−0.04	−0.05
E_{tot} (−)	0.55	0.55	0.62	0.55	0.60	0.69

E is positive because the mass flow rate through the compressor increases with respect to operation of the compressor at its design point. For the scenarios 8 and 9, where the inlet pressure of the system increases, ISE_p increases with the increase of the magnitude of the disturbance. M is positive and increasing because gas recycling is necessary for surge prevention while E increases.

In the hot recycle configuration, when the antisurge valve opens, both T_{01} and p_{01} increase and this causes the reduction of m_{ctrl} . Therefore it becomes smaller than m and the antisurge valve closes. This causes the oscillation of the valve until the action of the pressure controller causes the increase of m (Fig. 10).

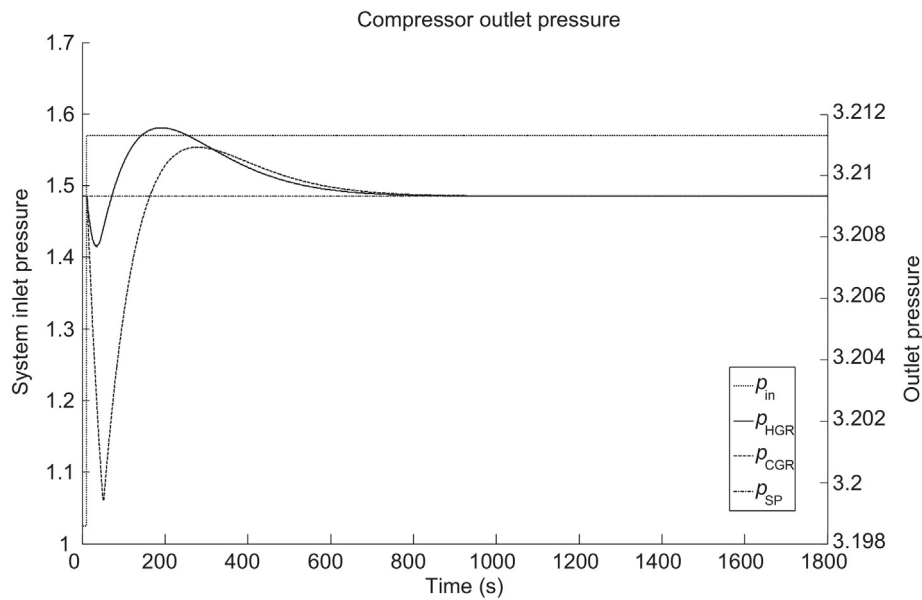


Fig. 9. Compressor outlet pressure (disturbance scenario 9).

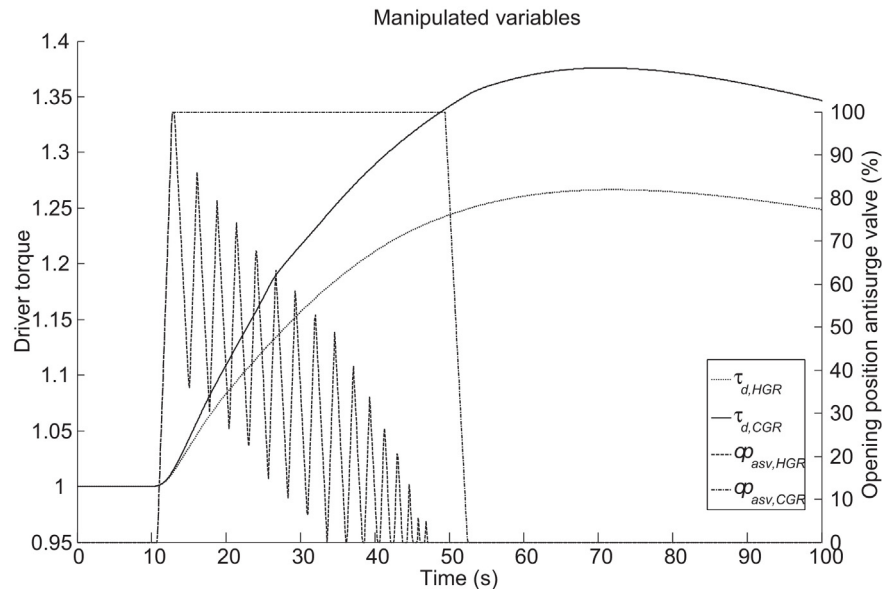


Fig. 10. Manipulated variables (zoomed view, disturbance scenario 9).

6.3.2. Cold gas recycle configuration

The results relative to the cold gas recycle can be summarised in the following way. For disturbance scenario 7 (Table 4, fifth column) there is no difference between the response of the cold gas configuration and the hot gas configuration as the antisurge valve does not open. However for scenarios 8 and 9 the cold recycle configuration presents higher ISE_p , M and E_{tot} than the hot recycle configuration.

In the cold gas recycle configuration, when the antisurge valve opens, this causes the increase of p_{01} while T_{01} remains constant. The consequence is that m_{ctrl} increases even more and therefore the valve remains open for longer and does not oscillate (Fig. 10). For this reason this configuration recycles more gas and therefore consumes more energy over time.

6.3.3. Comparison of response of recycle configurations

When comparing the hot configuration with the cold configuration, it is possible to notice that the cold recycle configuration

recycles more gas and therefore M is higher. Its power consumption is higher and therefore E_{tot} is higher.

In both configurations, the compressor entered surge during disturbance scenario 9. This scenario has been represented in Figs. 9–11 in order to explain the cause of the surge of the system under both recycle configurations. For the two recycle configurations, Fig. 9 represents the outlet pressures of the compressor while Fig. 10 represents the manipulated variables of the system.

Fig. 10 is zoomed in order to visualise the action of the rate limiter on the opening and closing of the antisurge valve in the hot gas recycle configuration. When the system inlet pressure increases, it causes the increase of the compressor inlet pressure p_{01} . Therefore this affects the value of m_{surge} that suddenly increases, following the same step dynamic as the inlet disturbance. This happens before the antisurge valve can open and this is the reason why it affects both configuration systems in the same way.

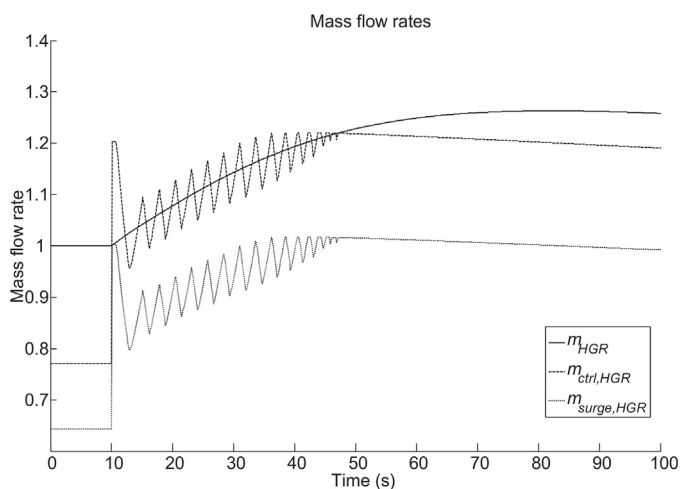


Fig. 11. Mass flow rates (zoomed view, disturbance scenario 9).

However, when the antisurge valve opens, the two configurations react in different ways. In the hot recycle configuration, the opening of the antisurge valve reduces immediately the value of m_{ctrl} that become smaller than m and therefore the antisurge valve closes again. However when this happens m becomes smaller than m_{ctrl} and therefore the antisurge valve reopens. Therefore the antisurge valve oscillates until the pressure controller acting on the driver torque increases the rotational shaft speed and therefore also the compressor inlet mass flow rate.

In the cold recycle configuration, the opening of the antisurge valve causes the increase of the flow control limit and therefore the antisurge valve stays open until m increases due to the action of the pressure controller on the driver torque (Fig. 11).

6.4. Recycle operation with supercritical CO₂

This section compares supercritical fluid recycle with subcritical gas recycle for surge prevention. In the selection between hot and cold gas recycle, the process parameters that are usually considered include time delay in the process response and machine integrity (Botros, 2011). However when dealing with supercritical compression operation it is also important to take into account the amount of gas recycled.

During the performed analysis, the third compression stage was simulated while running in full recycle mode in the hot gas recycle configuration and then in the cold gas recycle configuration. The amount of gas recycled during full recycle operation in the two configurations was compared. The fourth compressor was simulated while running in full recycle mode in the hot gas recycle configuration and then in the cold gas recycle configuration. Finally the mass of gas recycled per unit time during full recycle operation in the two configurations was compared. Results show that the subcritical compressor operating in full recycle mode is able to recycle 22.64% more gas by means of a cold gas recycle rather than a hot gas recycle. However in the supercritical compressor this amount rises to 81.50%.

Table 5 summarises the recycle pressure, temperature and flow rate during full recycle operation for the third and fourth stage in both hot and cold gas recycle configuration. The full recycle flow rate reported in the table depends on the recycle pressure and temperature upstream the antisurge valve. Therefore it is affected by the recycle configuration. All the reported parameters have been scaled according to the inlet design value of each variable at the entrance of the corresponding stage.

Table 5

Recycle variables for subcritical and supercritical compression, for hot gas recycle (HGR) and cold gas recycle (CGR) configurations.

Variable	Subcritical compression		Supercritical compression	
	HGR	CGR	HGR	CGR
Recycle pressure (–)	2.2	2.2	2.63	2.63
Recycle temperature (–)	1.25	1	1.37	1
Full recycle flow rate (–)	0.89	1.09	0.65	1.17

7. Discussion

Industrial compressor trains typically use both hot and cold gas recycle. The analysis in this paper has provided some insights about the dynamics and control issues of the two configurations. The analysis has shown the tendency for the antisurge valve to oscillate and has shown the causes of oscillation are different in the hot and cold recycle cases. Industrial implementations are not as likely to oscillate, however. The reason for this is cautious tuning of the anti-surge controller which typically will keep the recycle valve open for longer than strictly necessary.

In the selection between hot and cold gas recycle, the process parameters that are usually considered include time delay in the process response and machine integrity. In particular recycling of hot gas should be limited over time in order to guarantee the integrity of the machine. However the results reported in the paper have demonstrated that other aspects should be taken into account as well, such as system stability, energy consumption and maximum recycle mass flow rate during full recycle operation.

The adoption of the hot gas recycle configuration has some advantages. The process layout is simple and the configuration of the recycle line includes only two main items, the recycle pipeline and the antisurge valve. Moreover the time constant of the recycle line is small and this is due to the small hold-up volume along the recycle line. Therefore the system is able to respond quickly when surge may occur and gas must be recycled back in order to increase the inlet mass flow rate of the machine. The amount of gas recycled is small and this is due to oscillation of the antisurge valve. As a consequence, the energy consumption of the system is low as less gas is recycled. Moreover the recycled gas is not cooled down before being mixed with the freshly fed gas.

On the other hand, the hot gas recycle configuration has some disadvantages. These include the potential for oscillation of the antisurge valve during operation in partial recycle mode. This is due to the effect of the inlet temperature of the compressor on the location of surge. When the antisurge valve opens, the inlet temperature of the compressor increases and this reduces the value of the surge mass flow rate. As a consequence, the antisurge control system closes the antisurge valve. Moreover the machine can over-heat due to the high temperature of the recycled gas. The expansion of the gas that takes place along the recycle line causes a partial cooling that depends on the operating condition. However the temperature of the recycled gas is still higher than the temperature of the freshly fed gas and therefore the recycle of hot gas should be limited over time.

Finally during full recycle operation, the maximum amount of gas recycled is lower than if the gas was at lower temperature. This happens because the flow through a valve depends on the density of the gas and the density of the gas depends on its temperature and pressure.

The cold gas recycle configuration is the alternative to the hot gas recycle configuration. In this configuration the gas is cooled down during the recycle and therefore there are no issues related with overheating of the machine and loss of integrity for prolonged operation in recycle mode. Therefore there is no limit to this

operation. In theory the system could continuously run in partial or full recycle mode if needed. Moreover the oscillations of the anti-surge valve are limited if compared with the oscillation of the hot gas recycle configuration. This is because when the cooled gas is recycled back, the inlet pressure of the system increases while the inlet temperature does not, and this increases the surge mass flow rate.

However the cold gas recycle configuration has some downsides. The system is more complicated as it includes a gas cooler along the recycle line. Moreover the presence of the gas cooler represents both a capital and an operating cost, both of them depending on the type of exchanger. In the paper cooling has been assumed as a cost, however it could be recovered, depending on the overall process configuration. On the other hand the operation of the gas cooler would depend on the disturbances affecting the compression system and therefore should be limited.

In the cold recycle configuration, the gas cooler increases the hold-up volume along the recycle line and therefore the time constant of the system. This could be a problem when quick action is needed for incipient surge.

Moreover the increase of the surge mass flow rate due to the increase of the inlet pressure while recycling also means that this configuration may undergoes surge under small disturbances. Moreover the reduced oscillation of the antisurge valve means that the system recycles more gas if it has a cold recycle line.

Therefore the selection between the two configurations is not straightforward and all the previously mentioned aspects should be taken into account.

For an existing compression station, cold gas recycle would be more suitable as it would guarantee the possibility of running the system always in partial recycle if needed. Moreover the smaller frequency of oscillation would stress less the system. However the analysis in this paper showed a cold recycle configuration can still oscillate due to the interaction between pressure controller and surge controller, and this tendency should be taken into account when designing the control system. On the other hand there could be issues such as the cost of the cooling system and the space availability. If the compression station always runs in recycle mode but the refurbishment of the compressor would be too expensive, then heat integration could be taken into account as an option to recover some of the costs related with the cold recycle configuration.

For a new compression station, where the desire is to limit capital cost and stable operation is expected, a hot gas recycle configuration could guarantee smaller capital and operating cost and also surge protection when occasionally needed. On the other hand if the economics of the project allows that, than the ideal solution would be to install both cold and hot recycle systems. The hot recycle should be employed for urgent limited operation while the cold recycle should be employed for more long-run recycle operation.

The same idea would apply to a multistage compressor. Each stage should have a hot gas recycle connecting the output of the stage to the input of the stage. Then a cold gas recycle could cover each stage, each couple of stages or even the full multistage compressor. Each of the previous configurations would require in-depth steady state and dynamic analyses and real time optimisation in order to guarantee its safety, reliability and optimal operation.

When comparing subcritical and supercritical compression, the results reported in the paper have demonstrated the effect of the state of the fluid on the amount of gas that is possible to recycle when the antisurge valve is fully open. During subcritical compression, represented by the third stage, the cold gas recycle configuration allows the recycle of 22.6% more gas than the hot gas recycle configuration. However during supercritical compression, represented by the fourth stage, the cold gas recycle configuration recycles 81.50% more gas than the hot gas recycle configuration. This is due to the dramatic change of density of a supercritical fluid

especially when it is cooled down towards its critical point. This result is decisive when full recycle is needed and a hot gas recycle configuration may not provide a sufficient amount of gas for surge protection.

8. Conclusions

This paper reports on the effect of the gas recycle configuration for surge protection on the compression system when dealing with supercritical fluid.

The results have demonstrated that the cold gas recycle configuration has a lower performance in terms of pressure control. This is due to the higher amount of gas recycled over time. However this configuration oscillates less than the hot recycle configuration. The oscillation is due to the continuous opening and closing of the anti-surge valve, which has to react quickly in case the mass flow rate at the inlet of the compressor becomes smaller than the control mass flow rate.

The analysis shows there is an inherent dynamic tendency for oscillation with the recycle valve repeatedly opening and closing. The causes of the oscillation are different in the case of the hot and cold recycles. For the hot recycle, the main cause is the influence of both inlet temperature and pressure on the location of the surge line, while for the cold recycle it is the influence of the inlet pressure only on the location of the surge line, together with the interaction between the control loops. This finding should provide some theoretical underpinning for measures to avoid such oscillations in industrial implementations.

The high amount of gas recycled in the cold gas configuration increases the power consumption of the compressor compared to the hot recycle configuration. The higher energy consumption is due to both gas compression and recycle gas cooling.

Finally subcritical and supercritical full recycle operation has been analysed. The comparison between subcritical and supercritical compression is challenging because of the operational range of the compressor. The results of the analysis demonstrated that a subcritical compressor operating in full recycle mode is able to recycle 22.64% more gas by means of a cold gas recycle rather than a hot gas recycle. However in the supercritical compressor this amount rises to 81.50%. This result is decisive when full recycle is needed for surge protection.

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Appendix A. Supplementary data

Supplementary data associated with this article can be found, in the online version, at <http://dx.doi.org/10.1016/j.compchemeng.2016.03.012>.

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