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Control of centrifugal compressors via model predictive control for enhanced oil recovery applications

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Abstract: This paper proposes a control system for integrated pressure and surge control of centrifugal compressors for enhanced oil recovery application. The proposed control system is based on linear model predictive control. A fully validated non-linear dynamic model was developed in order to simulate the operation of the compressor at full and partial load. The model of the compression system includes a main process line with the compressor and a recycle line with the antisurge recycle valve. Different disturbance and control tuning scenarios were tested and the response of the model predictive controller was analysed, evaluated and also compared with a traditional control system. Temperature effects have been taken into account in the model of the process and in the constraint formulation of the MPC optimization problem. The results show that the proposed control technique is able to meet the process demand while preventing surge and also minimizing the amount of gas recycle.

Keywords: MPC, compressor, surge, control, driver torque, recycle, carbon dioxide, supercritical.

1. INTRODUCTION

Enhanced Oil Recovery (EOR) methods are commonly used in industry to recover oil from onshore and offshore reservoirs after primary and secondary extraction (Sobers et al., 2013). Among the non-thermal gas injection methods, carbon dioxide floods have been used for EOR (Thomas, 2008). CO_2 has already been used in the past for oil recovery however this method has been recently integrated with carbon storage for the reduction of atmospheric emissions (Ravagnani et al., 2009).

For the purposes of enhanced oil recovery and carbon dioxide storage, CO_2 must be compressed to supercritical conditions. For this type of application, the phase transition takes place inside a multistage centrifugal compressor. The operation of this type of machine is limited by surge. Surge is a dynamic instability of the gas that causes flow reversal inside the machine. When the compressor is surging, the oscillatory behaviour of the gas flow causes vibrations that can damage blades, casing and bearings (Boyce, 2012). In industrial practice, surge control still relies on avoidance control. Although many solutions based on active control have been proposed (Arnulfi et al., 2006), they were not implemented on industrial-size compressors due mainly to the cost and reliability of the additional devices they require (Uddin and Gravdahl, 2012).

Avoidance control for centrifugal compressors relies on the recycle of part of the compressed gas in order to increase the inlet flow rate of the compressors. When the recycle valve opens a compressor becomes a multiple-input multiple-output (MIMO) system. Model predictive control (MPC) is considered the most appropriate control for this type of system (Seborg et al., 2004). In the literature it has already been demonstrated that model predictive control was applicable for the control of complex compression systems (Smeulers et al., 1999, Øvervåg, 2013) and for surge prevention via closed coupled valve (Johansen, 2002) and drive torque actuation (Cortinovis et al., 2012). However the minimization of the recycle flow rate and the temperature effects have not previously been taken into account.

This paper proposes the use of MPC for the integrated control of pressure and surge in centrifugal compressor applications. The amount of gas recycled for surge prevention is minimized by control tuning and the temperature constraints have been included in the MPC formulation.

The structure of the paper is the following. In Section 2 the model of the compressor is presented. In Section 3 an overview on traditional compressor control is given. It is then followed by the description of the implemented model predictive controller and its design. In Section 4 the paper includes the MPC tuning, the scenarios for the validation of the control system and the results of the dynamic simulations. Finally, Section 5 presents the conclusions of the work.

2. MODEL OF THE COMPRESSOR

2.1 Mathematical model of the compressor

The model of the compression system is a non-linear one-dimensional dynamic model that includes a main process line and a recycle line. It is represented in Figure 1. The main process line includes inlet valve, outlet valve, compressor, duct and plenum. The recycle line includes the antisurge recycle valve that is used to prevent surge occurrence. Hot gas recycle should be limited over time because it can

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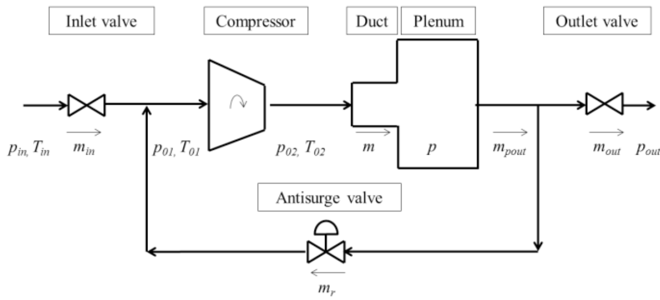

Fig. 1. Model of the compression system

overheat the machine. On the other hand it reduces the time delay of the system as a smaller amount of gas is stored along the recycle line (Botros, 2011).

The system includes also two nodes. The first node represents the physical point where the freshly fed gas mixes with the recycled gas, while the second node represents the physical point where the compressed gas splits between delivered gas and recycled gas. Variables m_{in}, m_{out} and m_r are the gas flow rate respectively through inlet, outlet and antisurge valve. m is the gas flow rate that enters the compressor and it is monitored for surge control, while m_{pout} is the gas flow rate that leaves the plenum. p_{in} and p_{out} are the inlet and outlet pressures of the system. p_{01}, p_{02} and p are respectively the compressor inlet pressure, compressor outlet pressure and plenum pressure. p is monitored for pressure control.

The model of the compressor is based on a well-established model present in the literature that includes a compressor, a plenum and an outlet throttle valve (Greitzer, 1976). This model was further developed by Fink et al. (1992) in order to include the dynamic of the rotating shaft connecting driver and compressor. Gravdahl and Egeland (1999) proposed a further modification by expressing the torque of the compressor τ_c as a function of shaft rotational velocity ω and mass flow rate m while Gravdahl et al. (2002) proposed to use the torque of the driver τ_d as input variable of the model instead of the rotational shaft speed N. This last model was the reference for this work and was modified according to Morini et al. (2007) in order to include also the recycle loop.

The equations of the model include the mass and the momentum balance of the compressor, the moment of momentum balance of the rotating shaft, the compressor torque and characteristic (Gravdahl et al., 2002). They also include the equations of the flow through inlet, outlet and recycle valve (Morini et al., 2007). The equations are the following:

$$\frac{dp}{dt} = \frac{a^2{}_{01}}{V}\left(m - m_{pout}\right) \tag{1}$$

$$\frac{dm}{dt} = \frac{A_1}{L}\left(\Psi_c p_{01} - p\right) \tag{2}$$

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

$$\tau_c = \mu r_2^2 \omega m \tag{4}$$

$$\Psi_c = \frac{p_{02}}{p_{01}} = \Psi_c(\omega, m) \tag{5}$$

$$m_{in} = k_{in}\sqrt{\rho_{in}(p_{in} - p_{01})} \tag{6}$$

$$m_{out} = k_{out}\sqrt{\rho(p - p_{out})} \tag{7}$$

$$m_r = k_r\sqrt{\rho_r(p_r - p_{01})} \tag{8}$$

where $a^2{}_{01}$ is the sonic velocity at ambient condition, V is the volume of the plenum, A_1 is the duct throughflow area, L is the duct length, Ψ_c is the compressor characteristic, J is the total inertia of the system, μ is the slip factor, r_2 is the impeller radius, k_{in}, k_{out}, k_r are the constants for respectively inlet, outlet and antisurge valve, ρ_{in}, ρ, ρ_r are the density of respectively m_{in}, m and m_r.

In this paper corrected compressor maps have been used in order to define the surge line as a function of pressure ratio, rotational shaft speed, inlet pressure and inlet temperature of the gas. The temperature of the gas entering the machine (T_{01}) has been estimated as a function of the temperature of the freshly fed gas (T_{in}), the temperature of the recycled gas (T_{02}) and the mass flow rates of these two flows (respectively m_{in} and m_r), according to the following equation:

$$m_{in}\int_{Tref}^{Tin} c_p(T)dT + m_r\int_{Tref}^{T02} c_p(T)dT = m\int_{Tref}^{T01} c_p(T)dT \tag{9}$$

where T_{ref} is the reference temperature and the heat capacity of the gas mixture c_p is evaluated at the temperature T according to:

$$c_p(T) = \sum_{i=1}^{2} x_i c_{p,i}(T) \tag{10}$$

where i is the number of components of the gas and x_i is their mass fraction. The outlet temperature of the compressor T_{02} is estimated according to the performance maps provided by the supplier of the compressor.

2.2 Case study and model validation

The case study presented in this paper refers to a multistage centrifugal compressor arranged in a single shaft