Proceedings of the 19th World Congress
The International Federation of Automatic Control
Cape Town, South Africa. August 24-29, 2014

Dynamic Modeling and Simulation of Compressor Trains for an Air Separation Unit

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Abstract: This paper presents dynamic models of a compressor train for an air separation unit. The model is derived from physical equations. A block library and complete models are implemented in Matlab/Simulink. Model accuracy is validated by comparing simulation results with realistic data gained from an existing industrial plant. The goal of modeling is to obtain a better understanding of the dynamic behavior in different operating modes and to get a base for design of new control strategies, *e.g.* a supervisory controller for minimizing power consumption of the total compressor train.

Keywords: Air Separation Unit; Compressor Modeling; Dynamic Simulation

1. INTRODUCTION

Cryogenic air separation units (ASU) are part of various industrial processes, *e.g.* in refinery, food, chemical or metallurgical industries. Depending on the process, the distillation unit produces high purity oxygen, argon or nitrogen. The process originally was developed by Carl von Linde in 1902. Important parts of an ASU are the compressors. They compress the filtered air from the environment to the necessary pressure. Afterwards the compressed air is cooled down to 100 K and separated into its three parts. Separation takes place in the distillation column. Due to the difference in boiling temperatures, the individual air components can be separated, each in liquid and gaseous form. This is purely a physical process. A detailed process description can be found in [Hands, 1986].

Nowadays, ASUs find new application in next generation fossil power plants, where CO_2 emission is to be reduced by using carbon capture and storage (CCS) methods. Some of the methods use oxygen instead of air for fuel combustion. This avoids N_2 and increases the CO_2 concentration in the flue gas and makes it easier to separate CO_2. CCS reduces CO_2 emissions by 90% or more, which would be an important contribution against global warming. Additional details for different CCS methods can be found in [Jansen, 2004].

The detailed structure of an ASU depends on the CCS process. Fig. 1 shows the ASU structure of interest, with two compressor stages, called main air compressor (MAC) and booster air compressor (BAC). Each of these stages contains more than one compressor, for example an axial and a radial compressor.

Integrating CCS with an ASU into a power plant will affect the efficiency of the plant, by reducing it up to 10%. [Bouillon et al., 2009] compare different solutions for a CCS power plant and conclude that an ASU is not suitable, because it would double the price of electricity.

To improve efficiency, new control approaches have been

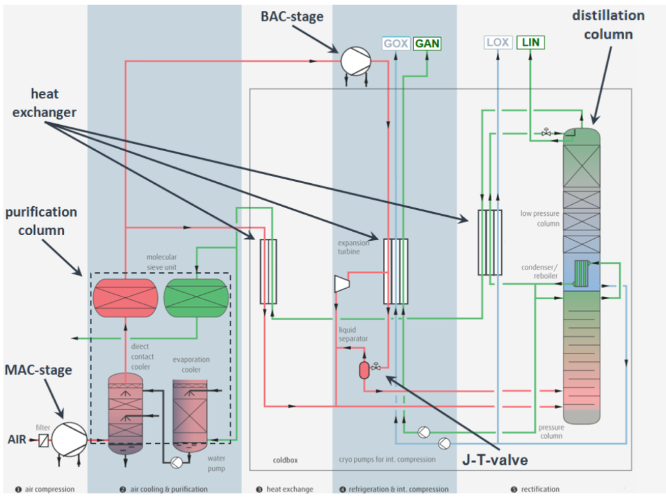

Fig. 1. Typical structure of an ASU plant, [Linde, 2007]

studied, like dynamic optimization strategies using model predictive control [Zhu et al., 2011]. Zhu's paper presents an on-line optimization of the operating conditions of the ASU distillation part. A new high performance process controller (HPPC) has been introduced in [Vinson, 2006]. It requires that the control system operates the plant at optimal efficiency over the full range of steady-state and dynamic operating conditions. The ASU has to follow the changing operating points of the power plant. To optimize these dynamic changes, a homotopy-based backtracking method is applied in [Zhu et al., 2009]. Most publications have their main focus on the distillation column.

Until recently, there have been few studies of the compressor part, which can causes high power consumption. Instead of optimizing the existing basic machine controllers and the individual components, the idea is to design a new supervisory controller for the whole compressor train, which not has been done until now.

Depending on the plant configuration, there are more

actuating signals than controlled variables. Thus, for each operating point of the power plant, the required operating conditions for the compressors can be reached by different set point combinations for the underlying basic machine controllers. This degree of freedom can be used for optimization, *i.e.* power minimization.
For design of the new optimizing controller, models of the ASU are required. Especially, a better understanding of compressor train dynamics is required by running simulations.
Section 2 outlines the derivation of the physical equations in order to build the model. Section 3 presents simulation results. Section 4 compares simulation results with measurement data from an existing industrial plant, in order to validate the models. The final section 5 contains conclusions.

2. ASU MODELING

For building the dynamic ASU models, the toolkit MATLAB/Simulink is used. The aim is to create a library with new blocks for each element of the compressor train like pipes, compressors, valves and coolers. An additional objective is adaptability for other processes like fuel gas compression or chemical production of nitric acid (HNO_3). Most of the ASU plants are powered by a steam turbines. Since models of steam turbine are readily available, this component will not be considered in this paper. Instead, models for the remaining components will be developed. In the following, the required physical equations for the mentioned elements are derived.

2.1 Pipe Model

The units of an ASU plant are interconnected by pipes with flowing air. The model equations can be gained from thermodynamics and fluid dynamics. The three basic equations are

- thermic state equation of gas,
- continuity equation,
- motion equation.

It is assumed that the cylindrical pipe lengths are much larger than their diameter, and that the flow can be regarded as a one-dimensional flow string. The pipes are fixed in space and are assumed to have no elastic deformation. The thermic state equation of a gas is

$$p = \rho \cdot R \cdot T \cdot z, \tag{1}$$

with pressure p, density ρ, specific gas constant R, temperature T, and real gas factor z. The mass flow $\dot{m}$ through a pipe cross section area A with velocity w is

$$\dot{m} = \rho \cdot A \cdot w. \tag{2}$$

The mass dm in a pipe piece of length dx is

$$dm = \rho \cdot A \cdot dx. \tag{3}$$

The change of dm with time depends on the difference between inlet and outlet flow, *i.e.*

$$\frac{\partial(dm)}{\partial t} = \dot{m} - [\dot{m} + \frac{\partial \dot{m}}{\partial x}dx] = -\frac{\partial \dot{m}}{\partial x}dx. \tag{4}$$

Applying (3) into (4) yields

$$\frac{\partial(\rho A dx)}{\partial t} = -\frac{\partial \dot{m}}{\partial x}dx. \tag{5}$$

or

$$\frac{\partial \rho}{\partial t} = -\frac{1}{A} \cdot \frac{\partial \dot{m}}{\partial x}. \tag{6}$$

Another assumption is that energy losses by heat transfer, wall friction or turbulences can be neglected, such that we can assume an isentropic flow, *i.e.* [Maier, 2008]

$$\frac{p}{\rho^{\kappa}} = const = c, \tag{7}$$

where with (1) the constant c is

$$c = \frac{p}{\rho^{\kappa}} = \frac{p}{\rho^{\kappa-1}\rho} = \frac{RTz}{\rho^{\kappa-1}} \tag{8}$$

By differentiating (7) and applying (8) and (6) we get

$$\frac{\partial p}{\partial t} = -\frac{\kappa RTz}{A} \cdot \frac{\partial \dot{m}}{\partial x}. \tag{9}$$

To derive the motion equation, we use the force balance in the considered pipe element, yielding

$$dm \cdot \frac{dw}{dt} - pA + pA - \frac{\partial}{\partial x}(pA) \cdot dx + F_R \cdot dx = 0. \tag{10}$$

where F_R means the wall friction force per pipe length. It can be described by

$$F_R = \frac{\lambda RTz}{2DA} \cdot \frac{\dot{m}^2}{p}. \tag{11}$$

The constant λ is the friction factor, which depends on the Reynolds number and the pipe wall surface roughness.
With the derivation of the velocity w over time

$$\frac{dw}{dt} = \frac{\partial w}{\partial t} + w \cdot \frac{\partial w}{\partial x}, \tag{12}$$

the formula for force balance and the continuity equation we get the motion equation

$$\frac{\partial \dot{m}}{\partial t} = \frac{2\dot{m}}{\rho\kappa RTz} \cdot \frac{\partial p}{\partial t} + [\frac{2\dot{m}^2}{\rho^2 A\kappa RTz} + A] \cdot \frac{\partial p}{\partial x} - F_R. \tag{13}$$

Integrating the continuity equation and the equation of motion along the pipe length, yields the following two equations, which we can use for the modeling

$$\frac{dp_{in}}{dt} = \frac{\kappa RzT_{in}}{AL} \cdot [\dot{m}_{in} - \dot{m}_{out}], \tag{14}$$

and

$$\frac{d\dot{m}_{in}}{dt} = \frac{2\dot{m}_{in}}{\rho\kappa RTz} \cdot \frac{dp_{in}}{dt} + [\frac{2\dot{m}_{in}^2}{\rho^2 A\kappa RTz} + A]\frac{p_{in} - p_{out}}{L} - F_R. \tag{15}$$

The entire derivation is explained in [Blotenberg, 1988]. The two following terms in equation (13) can be neglected without losing the accuracy of the result [Gronau, 1983]

$$\frac{2\dot{m}_{in}}{\rho\kappa RTz} \cdot \frac{dp_{in}}{dt}, \tag{16}$$

and

$$[\frac{2\dot{m}_{in}^2}{\rho^2 A\kappa RTz}] \cdot \frac{p_{in} - p_{out}}{L}. \tag{17}$$

Hence it is assumed that there is no heat transfer in the pipe, consequently the inlet and outlet temperature are the same. But for mixing junctions with more than one