Anton A. Kiss, Edwin Zondervan, Richard Lakerveld, Leyla Özkan (Eds.)
Proceedings of the 29th European Symposium on Computer Aided Process Engineering
June 16th to 19th, 2019, Eindhoven, The Netherlands.

Control structure design for a CO_2- refrigeration system with heat recovery

Adriana Reyes-Lúa[a], Glenn Andreasen[b], Lars F. S. Larsen[c], Jakob Stoustrup[b] and Sigurd Skogestad[a*]

[a]*Norwegian University of Science and Technology, Department of Chemical Engineering (NTNU), 7491 Trondheim, Norway*
[b]*Aalborg University, Department of Electronic Systems, Fr. Bajers Vej 7C, 9220 Aalborg, Denmark*
[c]*Danfoss A/S, Refrigeration and Air conditioning, 6430 Nordborg, Denmark*
**sigurd.skogestad@ntnu.no*

Abstract

In this work, we analyze a generic supercritical CO_2-refrigeration system with parallel compression, based on systems used for supermarket use. In order to maximize energy efficiency, this system has a "heat-recovery" function, in which part of the heat rejected at high pressure and temperature can be recovered to provide heating. Operating conditions and active constraints are strongly affected by seasonal requirements and ambient temperature. Thus, it is necessary to find a control structure that satisfies operational constraints and maintains (near-)optimal operation with different sets of active constraints. In this paper, we use a systematic procedure to define such control structure.

Keywords: constrained operation, self-optimizing control, PID control, control structure

1. Introduction

An appropriately designed control structure should maintain (near-) optimal operation, also when there are disturbances which cause the system to operate under conditions different than the design point. Optimal operation of a process in the presence of disturbances could be maintained using optimization-based control. However, in some cases, it is possible to design and implement advanced PI(D)-based control structures that also maintain optimal operation when constraints are reached (Skogestad, 2000; Reyes-Lúa et al., 2018). The advantage of such a PI(D)-based control structure compared to optimization-based control is simpler tuning and independence of an explicit model for every system (Forbes et al., 2015).

CO_2-refrigeration systems with parallel compression is environmentally attractive. Finding optimal design and operating conditions is an ongoing area of research (Gullo et al., 2018). In order to maximize energy efficiency, some systems have a "heat recovery" function, in which part of the heat rejected at high pressure and temperature can be recovered to provide heating (e.g. district heating or tap water) (Sawalha, 2013). The available energy can be increased by operating the cooler at a higher pressure, at the expense of a higher compression work. In this work, we design a PI(D)-based control structure for the studied CO_2-refrigeration system.

2. Description of the CO_2-refrigeration system with heat recovery

A flow diagram of the analyzed CO_2-refrigeration cycle with parallel compression and heat recovery is shown in Fig. 1, and the pressure-enthalpy diagram is shown in Fig. 2. The main function of this system is to provide cooling ($\dot{Q}_{ev}$) and maintain the desired cabinet temperature (T_{cab}) via heat exchange in the evaporator, which operates at low pressure (P_l). Low-pressure CO_2 in vapor phase is compressed to high-pressure (P_h) and temperature supercritical CO_2, which may be used to heat tap water in the heat recovery section. Excess heat ($\dot{Q}_{gc}$) is rejected to the ambient air in the gas cooler.

High-pressure CO_2 is expanded to an intermediate (sub-critical) pressure (P_{IP}) in the high-pressure valve (V_{hp}). Vapor and liquid CO_2 are separated in the liquid receiver. The evaporator valve (V_{ev}) regulates the flow of liquid CO_2 from the receiver to the evaporator. By opening and closing V_{hp} and V_{ev}, we regulate the refrigerant charge (mass) at the high and low pressures. Vapor CO_2 from the liquid receiver is recycled to the high-pressure side either via parallel compression ($K2$) or the intermediate pressure valve (V_{IP}) and the main compressor ($K1$). The total compression work can be reduced by utilizing the parallel compressor instead of the intermediate pressure valve and the main compressor.

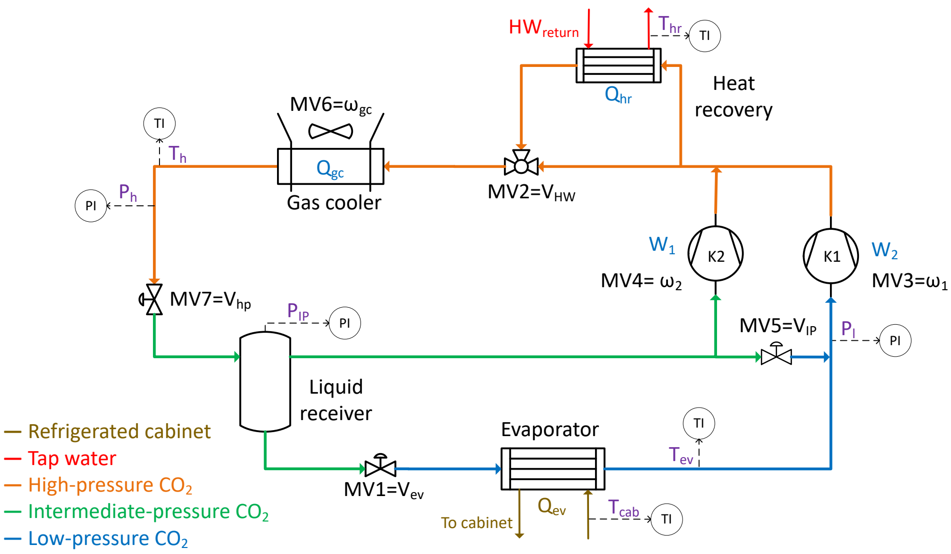

Figure 1: CO_2-refrigeration system with parallel compression and heat recovery. There are seven available manipulated variables (MV).

2.1. High-side pressure

In the supercritical region, there is no saturation condition and the pressure is independent of the temperature. From the control point of view, this means that it is necessary to control the high pressure (P_h), since it influences the gas cooler exit enthalpy (and evaporator inlet enthalpy). In other words, P_h will determine specific refrigeration capacity. As P_h is determined by the relationship between refrigerant charge, inside volume and temperature in the high-pressure side, we can actively control it using V_{hp} (Kim et al., 2004).

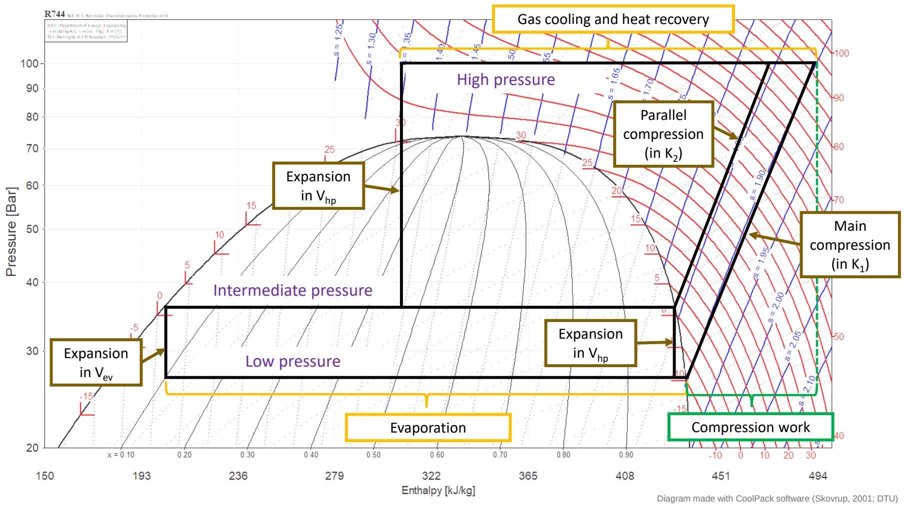

Figure 2: Pressure-enthalpy diagram of CO_2-refrigeration system with parallel compression.

It is relevant to analyze the effect of high pressure on the coefficient of performance (COP). In the case of a refrigeration system, it is defined as the ratio between cooling and compression work ($COP = \dot{Q}_{ev}/W_s$). As the isentropic compression line in the pressure-enthalpy diagram (blue lines in Fig. 2) is linear, compression work will linearly increase as P_h increases. On the other hand, in the supercritical region the isotherm (red lines in Fig. 2) becomes steeper with pressure, reducing the capacity enhancement from a given increase in pressure. For this reason, the COP reaches a maximum above which the added capacity no longer fully compensates for the additional work of compression. Thus, there is an optimal high pressure that maximizes COP (Nekså, 2002).

We should also note that in the supercritical region, at a fixed pressure, a small change in refrigerant exit temperature can produce a large change in gas cooler exit enthalpy (and evaporator inlet enthalpy), making COP very sensitive to the gas cooler refrigerant exit temperature. Previous studies (Liao et al., 2000; Jensen, 2008; Sawalha, 2013) are in line with this and have shown that the optimal set-point for the high pressure (P_h) should be corrected by the outlet temperature of the gas cooler (T_h).

2.2. Heat-recovery section

Part of the heat rejected at high pressure can be recovered to provide hot water in the heat recovery section. Heat is rejected at gliding temperature, as supercritical CO_2 is cooled. This way, the temperature profile of the CO_2 matches the heating-up curve of water, giving reduced thermodynamic losses and high efficiency (Kim et al., 2004). As it can be deduced from Fig. 2, increasing the high pressure increases the available heat for recovery, at the expense of a higher compression work. Additionally, the available heat for recovery in the supercritical region is much higher than with sub-critical CO_2.

3. Design of the PI(D)-based control structure

In this section, we apply part of the systematic plantwide control procedure proposed by Skogestad (2000) to design a self-optimizing control structure which maintains (near-) optimal operation, also in the presence of disturbances. The first step of the procedure is to define the operational objective. Here, we want to maximize the coefficient of performance (COP), subject to the system itself and operational constraints:

$$\min_u \; -COP(u,x,d) = -(\dot{Q}_{ev} + \dot{Q}_{hr})/(W_1 + W_2)$$

$$\text{s.t.} \quad f(u,x,d) = 0 \qquad \text{system equations (model)} \tag{1a}$$

$$g(u,x,d) \leq 0 \qquad \text{operational and physical constraints} \tag{1b}$$

$$e(u,x,d) = 0 \qquad \text{set-points} \tag{1c}$$

where x are the internal states, u are the degrees of freedom, and d are the disturbances.

Physical constraints in Eq. (1b) are related to pressure (P_i), motor velocities (ω_i) and valve openings (z_i), specifically: $P_i \leq P_i^{max}\ \forall i$, $(P_{IP} - P_l)^{min} \leq (P_{IP} - P_l)$, $\omega_j^{min} \leq \omega_j \leq \omega_j^{max}\ \forall j$, and $z_k^{min} \leq z_k \leq z_k^{max}\ \forall k$. The most important set-point in Eq. (1c) is to supply enough cooling ($\dot{Q}_{ev}$) to maintain $T_{cab} = T_{cab}^{sp}$. Additionally, we would like to supply enough heating ($\dot{Q}_{hr}$) to maintain $T_{hr} = T_{hr}^{sp}$.

The next step is to determine the steady-state optimal operation. In order to do this we:

- *Identify steady-state degrees of freedom:* The analyzed system has seven available manipulated variables, MVs in Fig. 1: $u = [\omega_1,\ \omega_2,\ \omega_{gc},\ z_{Vev},\ z_{Vhp},\ z_{V_{HW}},\ z_{V_{IP}}]^T$. These degrees of freedom can be used to achieve optimal operation. Note that ω_2 and $z_{V_{IP}}$ are not independent, as either would have a similar effect in P_{IP}.
- *Identify important disturbances and their expected range:* In this case study, important disturbances (d) are cooling demand ($\dot{Q}_{ev}$, corresponding to T_{cab}^{sp}), and heating demand ($\dot{Q}_{hr}$, corresponding to T_{hr}^{sp}). The range for both is $\dot{Q}_i^{min} \leq \dot{Q}_i \leq \dot{Q}_i^{max}$.
- *Identify active constraints regions:* Once that the disturbances and their range are specified, the active constraints regions are found. This can be done by optimization or using engineering insight (Jacobsen and Skogestad, 2011). There are three relevant operating regions:
  1. *"Unconstrained" case*: corresponding to spring/fall operation.
  2. *Maximum heating*: corresponding to winter, when $\dot{Q}_{hr} = \dot{Q}_{hr}^{max}$.
  3. *Maximum cooling*: corresponding to summer, when $\dot{Q}_{ev} = \dot{Q}_{ev}^{max}$.

In every case, cooling requirements ($T_{cab} = T_{cab}^{sp}$) must be met. If possible, heating requirements ($T_{hr} = T_{hr}^{sp}$) should also be met. We do not consider $\dot{Q}_{ev} = \dot{Q}_{ev}^{min}$, as it corresponds to shut-down. $\dot{Q}_{hr} = \dot{Q}_{hr}^{min}$ is included in the "unconstrained" and maximum cooling cases. Then, we will design a control structure that works for the three relevant cases mentioned above. Fig. 3 shows the proposed control structure. The procedure to design this control structure is explained below.

For each region, each steady-state degree of freedom (MV) needs to be paired with a controlled variable. First, we pair active constraints. Then, for the remaining degrees of freedom, we identify self-optimizing controlled variables. These are usually a combination of measurements found by optimization. When designing control structures for systems with changing active constraint regions, it is useful to organize constraints in a priority list (Reyes-Lúa et al., 2018). Physical constraints (Eq. (1b)) have the highest priority. Regarding set-points (Eq. (1c)), $T_{cab} = T_{cab}^{sp}$ has a higher priority than $T_{hr} = T_{hr}^{sp}$.

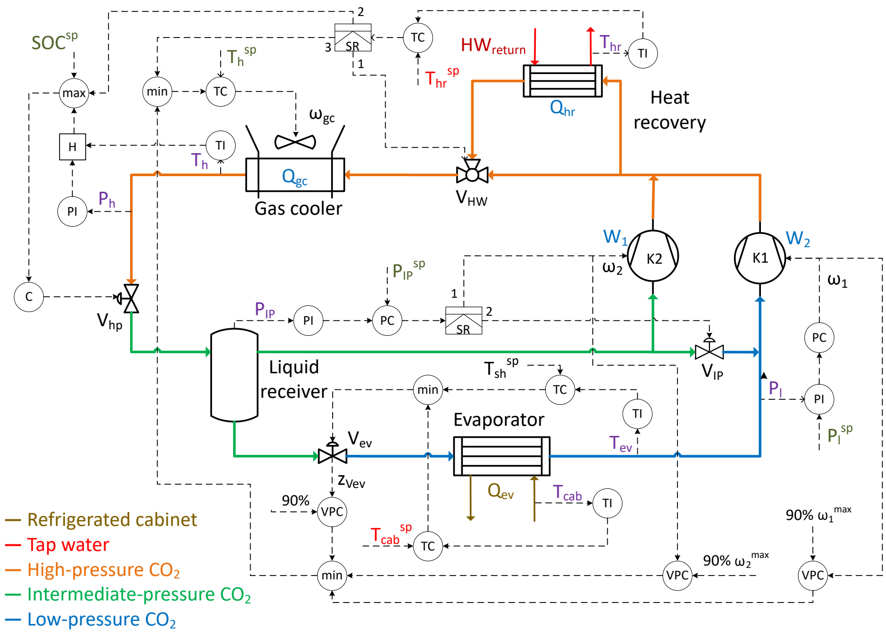

Figure 3: Control structure for the CO_2-refrigeration system with heat recovery.

3.1. "Unconstrained" case

This is the base case and we can satisfy every constraint. We use MV1=V_{ev} to control T_{cab}, and MV2=V_{HW} to control T_{hr}. The set-points are given by the operator. In order to assure that the evaporator is not over-flooded, we include a controller for the evaporator outlet temperature (T_{sh}). We have five remaining unconstrained degrees of freedom, two of which are not independent (ω_2 and V_{IP}). We pair these degrees of freedom as follows:

1. MV3=ω_1 controls P_l. P_l^{sp} is found by optimization (self-optimizing variable).
2. The parallel compressor (MV4=ω_2) and MV5=V_{IP} are used to control the pressure in the liquid receiver (P_{IP}). The set-point defined by optimization and may be a self-optimizing variable. Normal operation is using ω_2, but when the flow is too low, we use V_{IP}. We can implement this with a split-range controller.
3. MV6=ω_{gc} controls T_h (outlet of the gas cooler). T_h^{sp} is defined by optimization.
4. MV7=V_{hp} controls P_h. As explained in Section 2.1, the set-point is a linear combination (H) of P_h and T_h, which is a self-optimizing variable (Jensen, 2008).

3.2. Maximum heating

When V_{HW} becomes fully open, we must switch the manipulated variable to continue controlling T_{hr}. This is handled using split-range control with selectors. First, we switch to V_{hp} as manipulated variable and increment the available heat for recovery by increasing P_h. To implement this, we include a selector for the set-point of the high-pressure controller. Once we reach P_h^{max}, we get additional capacity for the heat-recovery section by increasing T_h, using ω_{gc} as manipulated variable. This will increase mass flow through the compressors and, as consequence, the discharge temperature.

If we continue to increase T_h, at some point liquid in the low-pressure section may be insufficient and V_{ev} will reach its maximum opening. Alternatively, the compressors could reach maximum capacity due to the increased mass flow. To prevent this, we implement valve-positioning controllers (VPC) with a *min* selector, which will prevent T_h from increasing in such a way that either the valve or the compressors ($z_{V_{ev}}$, ω_1 or ω_2) saturate.

3.3. Maximum cooling

As cooling requirements increase, z_{Vev} will open and reach z_{Vev}^{max}. The valve positioning controller for V_{ev} will adjust T_h (and indirectly P_h) such that the system reaches Q_{ev}^{max}.

4. Final remarks

Using a systematic procedure, we designed a PI(D)-based control structure for a CO_2-refrigeration system, that maintains (near-)optimal steady-state operation, also with changes in the set of active constraints. We should point out that pairing on the low-pressure side could be different (e.g. controlling T_{cab} with the main compressor, and P_l with V_{ev}). The final decision would consider system dynamics. It is important to mention that we can usually reach the same steady-state control objectives we reach with split-range controllers by using valve-positioning control or different controllers with different set-points.

References

M. G. Forbes, R. S. Patwardhan, H. Hamadah, R. B. Gopaluni, Jan 2015. Model Predictive Control in Industry: Challenges and Opportunities. IFAC-PapersOnLine 48 (8), 531–538.

P. Gullo, K. M. Tsamos, A. Hafner, K. Banasiak, Y. T. Ge, S. A. Tassou, Dec 2018. Crossing CO_2 equator with the aid of multi-ejector concept: A comprehensive energy and environmental comparative study. Energy 164, 236–263.

M. G. Jacobsen, S. Skogestad, Oct 2011. Active Constraint Regions for Optimal Operation of Chemical Processes. Industrial & Engineering Chemistry Research 50 (19), 11226–11236.

J. B. Jensen, 2008. Optimal operation of refrigeration cycles. Phd thesis, NTNU, Norway.

M.-H. Kim, J. Pettersen, C. W. Bullard, 2004. Fundamental process and system design issues in CO_2 vapor compression systems. Progress in Energy and Combustion Science 30 (2), 119–174.

S. Liao, T. Zhao, A. Jakobsen, Jun 2000. A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles. Applied Thermal Engineering 20 (9), 831–841.

P. Nekså, Jun 2002. CO_2 heat pump systems. International Journal of Refrigeration 25 (4), 421–427.

A. Reyes-Lúa, C. Zotică, S. Skogestad, Jul 2018. Optimal Operation with Changing Active Constraint Regions using Classical Advanced Control. In: 10th ADCHEM. IFAC, Shenyang, China.

S. Sawalha, Jan 2013. Investigation of heat recovery in CO_2 trans-critical solution for supermarket refrigeration. International Journal of Refrigeration 36 (1), 145–156.

S. Skogestad, Oct 2000. Plantwide control: the search for the self-optimizing control structure. Journal of Process Control 10 (5), 487–507.