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OPTIMAL OPERATION OF A SIMPLE LNG PROCESS

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Abstract: Considering the large amount of work that goes into the design of LNG processes there is surprisingly little attention to the subsequent operation. This probably comes from the misconception that optimal design and optimal operation is the same, but this is usually not true. In this paper we are studying optimal operation of a relatively simple LNG process, namely the PRICO process.

Keywords: Self-optimizing control, optimization, operation

1. INTRODUCTION

Large amounts of natural gas are found at locations that makes it infeasible or not economical to transport it in gaseous state (in pipelines or as compressed natural gas) to the customers. The most economic way of transporting natural gas over long distances is to first produce liquefied natural gas (LNG) and then transport the LNG by ships. LNG has approximately 600 times the density of gaseous natural gas.

At atmospheric pressure LNG has a temperature of approximately −162 °C, so the process of cooling and condensing the natural gas requires large amounts of energy. Several different process designs are used and they can be grouped roughly as follows:

- Pure fluid cascade process: Several pure refrigerant cycles are used to limit the mean temperature difference in the heat exchange
- Single mixed refrigerant: The refrigerant composition is adjusted to match the cooling curve of the natural gas. Some are designed with a separate pre-cooling cycle
- Mixed fluid cascade process: Energy efficiency is further improved by using several mixed refrigerant cycles

The process considered in this paper is a single mixed refrigerant process, namely the PRICO process (Stebbing and O'Brien, 1975) and (Price and Mortko, 1996). This is the simplest configuration utilizing mixed refrigerant, but it provides valuable insight also applicable to more complex configurations. The PRICO process is optimized in several publications ((Lee *et al.*, 2002) and (Del Nogal *et al.*, 2005)), but only with respect to design.

2. PROCESS DESCRIPTION

Figure 1 shows a simplified flowsheet of the PRICO process.

Nominal conditions:

- The natural gas enters with a pressure of 55 bar and a temperature of 25 °C after pretreatment
- Natural gas flow rate is 1 kmol s^{-1}
- Composition of natural gas: 89.7 % methane, 5.5 % ethane, 1.8 % propane, 0.1 % n-butane and 2.8 % nitrogen

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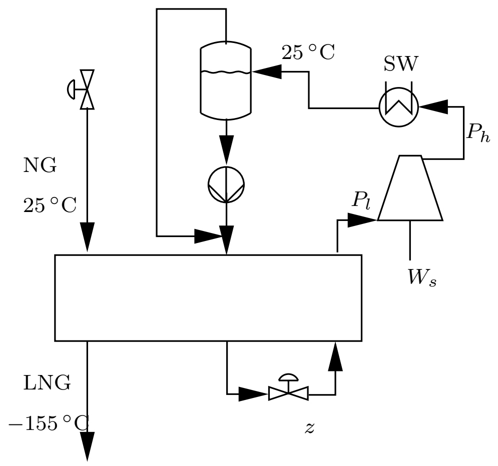


Fig. 1. Simplified flowsheet of the PRICO process

- Pressure drops:
  - · 5 bar in natural gas stream
  - · 0.1 bar in SW cooler
  - · 4 bar for hot refrigerant in main heat exchanger
  - · 1 bar for cold refrigerant in main heat exchanger
- Constant heat transfer coefficients
- The refrigerant is a mix of nitrogen (N_2), methane (C_1), ethane (C_2), propane (C_3) and n-butane (nC_4) and the composition is used in optimization
- Cooling of refrigerant to 25 °C in SW cooler
- Vapour to compressor is super-heated 10 °C
- The compressor has a fixed isentropic efficiency of 80 %
- In design the minimum temperature difference in the heat exchanger (ΔT_{min}) is 1.2 °C

Using the above conditions the cycle in Figure 1 is working as follows: After compression the mixed refrigerant is cooled to 25 °C in a sea water (SW) cooler before it is further cooled together with the natural gas through the main heat exchanger. The high pressure sub-cooled liquid is then sent through a choke valve to give a low temperature two-phase mixture which is vaporized in the main heat exchanger to provide the necessary cooling duty. The vapour is slightly super-heated (10 °C) before it is compressed back to the high pressure.

2.1 Model

The SRK equation of state is used both for the natural gas and the refrigerant. The main heat exchanger is a distributed model, which has been discretized into 100 cells.

2.2 Manipulated inputs

There are in total 9 manipulated inputs (degrees of freedom):

- Compressor power W_s
- Choke valve opening z
- Flow of sea water (SW) in SW cooler
- Flow of natural gas (can also be considered a disturbance)
- Composition of refrigerant (4 independent inputs)
- Active charge (within the heat exchangers). The active charge can be manipulated by altering liquid level in a receiver in the cycle, or by having an external filling/emptying system. Here it may be changed by the liquid pump after the refrigerant separator

2.3 Constraints during operation

There are some constraints that must be satisfied during operation.

- Super-heating: The vapour entering the compressor must be at least 10 °C super-heated
- T^{out}_{LNG}: Natural gas temperature out of the main heat exchanger must be −155 °C or colder
- Pressure: Must be within certain bounds (not considered in this paper)
- Compressor outlet temperature must be below a given temperature (not considered in this paper)
- Compressor power (W_s) maximum 20 MW

2.4 Active constraints

Using some general knowledge of the process we are able to identify constraints that will be active at optimum. In total there are 3 active constraints:

- Super-heating should be minimized (e.g. see (Jensen and Skogestad, 2005), and for this case this means controlling $\Delta T_{sup} = 10$ °C. Note that measuring the degree of super-heating directly requires knowledge of the refrigerant composition.
- Excess cooling is costly so $T_{LNG} = -155$ °C
- Maximum cooling: Assume $T = 25$ °C after SW cooler

2.5 Degrees of freedom

After implementing the three active constraints using three of the nine manipulated inputs, we are left with six degrees of freedom. For this steady state analysis the pairing of inputs and outputs is insignificant, so say we are left with the following subset of manipulated inputs:

- Pressure P_h (could correspond to the liquid pump as physical input)
- Four refrigerant compositions

- Flow of natural gas (can also be considered a disturbance)

These variables should be adjusted to optimize the operation.

3. OPTIMI ATION RESULTS

In this section we will show that the most common method for designing heat exchangers, the specification of the minimum approach temperature ΔT_{min}, has a major drawback in terms of finding the true optimum.

3.1 Design versus operation

In design it is common to specify ΔT_{min} for each heat exchanger in order to get a balance between capital costs (favored by a large ΔT_{min}) and operational costs (favored by a small ΔT_{min}). In operation however, ΔT_{min} is free to vary. This gives rise to two different optimization problems. One for design

$$\begin{aligned} &\min(W_s) \\ \text{such that} \quad &\Delta T - \Delta T_{\min} \geq 0 \end{aligned} \tag{1}$$

and one for operation

$$\begin{aligned} &\min(W_s) \\ \text{such that} \quad &A_{\max} - A \geq 0 \end{aligned} \tag{2}$$

In both cases we have as optimization degrees of freedom, the pressure P_h and four compositions. The feed rate of NG is assumed given. The A_{max} used for operation is obtained as a result of solving the design problem.

Table 1 shows the difference between design and optimal operation at the conditions listed in section 2. In design, we specify $\Delta T_{min} = 1.2\,°\mathrm{C}$ (which is the same as reported in (Del Nogal *et al.*, 2005) and (Lee *et al.*, 2002)). In operation, with areas found by design, ΔT_{min} is reduced to 0.537 °C and we are able to find a new operating point (with the same heat exchanger area) with 2.60 % less compressor power (Case I in Table 1). This is possible by altering the composition of the refrigerant and the pressure. The pressure ratio (P_h/P_l) is actually increased slightly from 5.93 to 6.87, but this is more than compensated for by the reduction in refrigerant flow rate (from 3.118 to 2.773 kmol s^{-1}). In Case II, we vary only the pressure (and fix the composition in operation to the value found in the design), and we are able to reduce the shaft work by 1.89 % compared to design. Similar results have also been reported for an ammonia cycle (Jensen and Skogestad, 2005). Note that althoug the savings depend on the value for ΔT_{min}, the fact that there are savings do not.

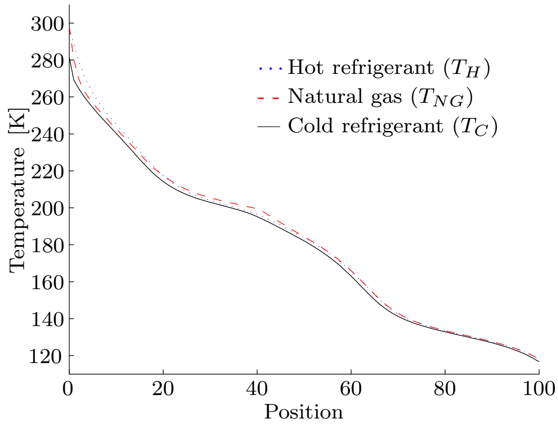


Fig. 2. Temperature profile in the main heat exchanger as function of position for optimal operation (Case I)

Table 1. Difference between design with fixed ΔT_{min} and optimal operation with free refrigerant composition (Case I) and with composition as design (Case II)

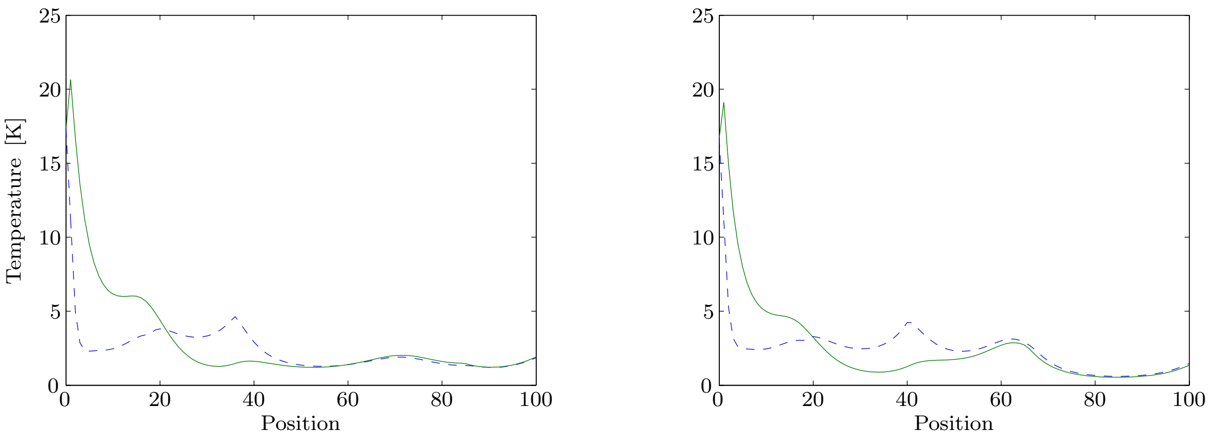
	Design *	**Case I** **	**Case II** *
$\mathbf{\Delta T_{min}}$ [°C]	1.200	0.537	0.642
$\mathbf{W_s}$ [MW]	17.404	16. 50	17.075
$\mathbf{P_h}$ [bar]	1 .124	23.617	23.811
$\mathbf{P_l}$ [bar]	3.226	3.438	3.05
Flow [kmol s 1]	3.118	2.773	2.621

* Composition [%]:
N_2: 7.72, C_1: 23.65, C_2: 3 .4 , C_3: 0.00, C_4: 2 .14
** Composition [%]:
N_2: 7.45, C_1: 25.86, C_2: 38.5 , C_3: 0.00, C_4: 28.11

We generally find a smaller ΔT_{min} in optimal operation, because the temperature difference varies more throughout the heat exchanger. Only at the limit when ΔT_{min} is zero (infinite heat transfer areas) is ΔT_{min} for design and optimal operation equal. Note that the savings actually increase with decreasing heat transfer areas (increasing ΔT_{min}).

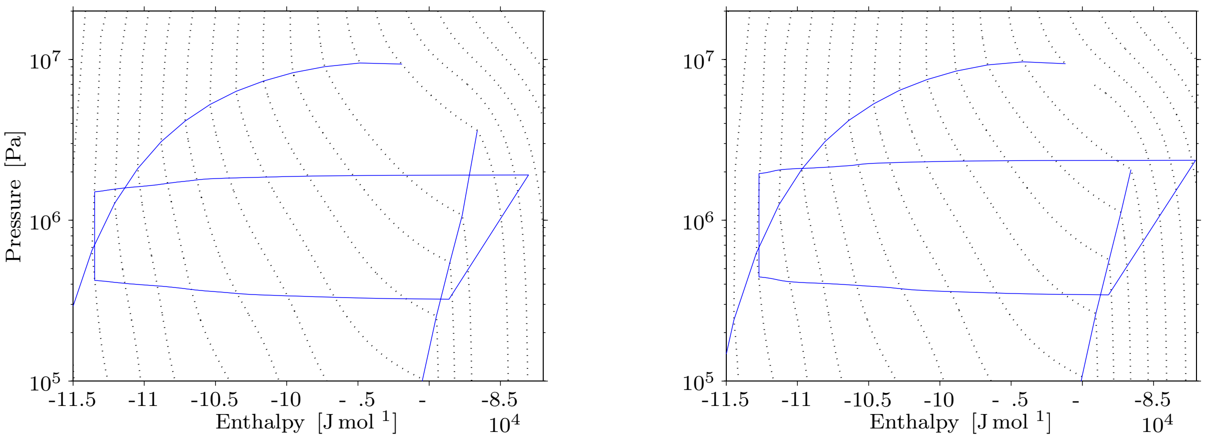
The temperature profile in the main heat exchanger is given in Figure 2. Note the very close match of the cooling and heating curves. To see this more clearly, the temperature difference profile in optimal design and optimal operation are shown in Figure 3. The two optimums are also illustrated in pressure-enthalpy diagrams in Figure 4. Note that since the composition of the refrigerant is changed, the pressure enthalpy diagrams are actually different.

The obtained values for W_s, both for design (17.40 MW) and optimal operation (16.95 MW), are better than the results reported by (Del Nogal *et al.*, 2005) 24.53 MW and (Lee *et al.*, 2002) 26.60 MW. It is unclear if this is because of differences in the optimization or in the conditions.



(a) Design (b) Case I

Fig. 3. Temperature difference profile in the main heat exchanger as function of position for design ($\Delta T_{min} = 1.2\,°\mathrm{C}$) and optimal operation (Case I). Dashed line - $T_{NG} - T_C$. Solid line - $T_H - T_C$



(a) Design (b) Case I

Fig. 4. Pressure-enthalpy diagram for both (a) design and (b) optimal operation (Case I)

4. IMPLEMENTING OPTIMAL OPERATION

We have now identified the optimum for the PRICO process, but how should we control the process to maintain close to optimal operation when the process is exposed to disturbances? This is related to selection of controlled variables, which is presented below.

4.1 Operating strategies

In general, there are two main modes of operation of a plant:

(1) Throughput (of NG) given, minimize operating cost (here W_s) (studied in previous section)
(2) Maximize throughput given operational constraints (in this case the bottleneck will be W_s^{max} where W_s^{max} is given by design).

For our case study the bottleneck (W_s) is identical to the operating cost, so optimal operation in modes (1) and (2) are identical. This follows since $\frac{\partial W_s}{\partial F} \geq 0$, so increasing F will also increase W_s (which is not possible).

4.2 Self optimizing control

Self optimizing control is when we can achieve acceptable loss with constant setpoint values for the controlled variables (without the need to re-optimize when disturbances occur) (Skogestad, 2000).

We will use the procedure in (Halvorsen *et al.*, 2003) to identify controlled variables that may result in self optimizing control. First we use a linear model (scaled both in input and output direction) to locate promising controlled variables.

The most promising candidates are then tested on the non-linear model using full disturbances.

Outline of the linear procedure:

(1) With fixed active constraints, obtain a linear model (G) from the unconstrained inputs (u) to outputs (candidate controlled variables):

$$y = Gu$$

(2) Scale the linear model in the inputs such that the effect of all inputs on the objective function is equal.
(3) Scale the linear model in the outputs so their expected variety (sum of span y and implementation error n) is equal.
(4) We are looking for controlled variables that maximize the minimum singular value of the scaled linear gain matrix.

From this point on we assume that the refrigerant composition is maintained constant and that the natural gas flow is determined by an upstream or downstream process (a disturbance). So we are left with one unconstrained degree of freedom for optimization, e.g. P_h. Since there is only one input it is not necessary to scale with respect to the input, and the procedure of finding the set of controlled variables that maximize the minimum singular value reduces to picking outputs with high linear scaled gains.

4.3 Linear analysis of controlled variables (CV)

Two obvious controlled candidates are the pressures P_h and P_l. Other candidate controlled variables can be a temperature somewhere in the main heat exchanger or after some unit. It is also possible to control a linear combination of two measurements, such as the degree of sub-cooling of refrigerant at outlet of main heat exchanger ($\Delta T_{sub} = T - T_{sat}$) or a temperature difference at some position ($\Delta T_j(i) = T_j(i) - T_C(i)$).

To get the optimal span y we consider the variance of y using 1 % of the expected disturbances (given in Table 2).

The results of the linear method for the most promising controlled variables are given in Table 3. Table 3 shows the scaled linear gain from the input (P_h) to some candidate controlled variables. Only a subset of all the variables are given We are looking for candidates with a high scaled linear gain $|G'|$ so P_l looks like a poor choice with $|G'|$=0.69. P_h is much better with $|G'|$=6.41. The theoretical loss is inversely proportional to $|G'|^2$ so controlling the high side pressure P_h instead of the low side pressure P_l would reduce the loss by a factor $(6.41/0.69)^2 = 86.3$. Other variables in Table 3 are also promising, including temperature

Table 2. Nominal, minimum and maximum values for the disturbances

	Nominal	Min	Max
NG flow [kmol s 1]	1	0.5	1.1
NG P [bar]	55	50	60
SW T [°C]	25	20	30
REF C_1 [%]	25.86	15.86	35.86
REF C_2 [%]	38.5	28.5	48.5
REF C_4 [%]	28.11	18.11	38.11
REF N_2 [%]	7.45	0.00	17.45

in the main heat exchanger, refrigerant flow and temperature out of the compressor.

Table 3. Linear analysis of controlled variables

CV	G	span y	n *	G' 1 6
P_h [Pa]	1	56036	1e5	6.41
P_l [Pa]	-3.76e-2	4175	5e4	0.6
ΔT_{sub} [°C]	-2.24e-5	2.0	1.5	6.24
$T_H(12)$ ** [°C]	-1.80e-5	1.66	1	6.78
$T_{NG}(12)$ [°C]	-1.5 e-5	1.62	1	6.0
$T_C(12)$ [°C]	-1.74e-5	1.82	1	6.18
$T_{NG}(50)$ [°C]	-2.5 e-5	11.3	1	2.10
$T_H(50)$ [°C]	-2.53e-5	11.0	1	2.11
$T_C(50)$ [°C]	-2.5 e-5	14.	1	1. 6
$F \quad w$ [mol]	-8.46e-4	73.5	10	10.13
T_o^{ou} [°C]	2.80e-5	1.51	1	11.16
W_s [MW]	8.30e-8	0.21	1e-4	0.40
$\Delta T_{NG}(41)$ [°C]	7.66e-6	0.55	1.5	3.74
$\Delta T_H(21)$ [°C]	-3.7 e-6	0.28	1.5	2.13

* Implementation error
** (i): Position in heat exchanger (see Figure 2)

This linear approach is only valid close to the nominal point so the most promising controlled variables must be checked using the non-linear model with full disturbances. This will reveal feasibility problems together with large reduction in performance caused by non-linearities.

4.4 Non-linear analysis of promising CV's

In the linear analysis we assumed composition disturbances of 10 % (absolute) for each component, but in practice this turns out to be infeasible because of excessive compressor work. From this we see that the composition when filling the system is vital to achieving good performance, and make-up should be added to stay at the desired composition.

In the following we assume that the steady state disturbance in the composition is 2 % for each component. From the linear analysis we found that P_l is a poor controlled variable, and this is verified in the non-linear brute force evaluation where we find that a constant P_l is infeasible for several of the disturbances. Temperatures within the main heat exchanger look promising from the

Table 4. Maximum loss with implementation error and disturbance; d1 - all variables (Table 2) except composition. d2 - also composition (2 %)

CV	Maximum loss	
	d1 *	d2 **
ΔT_{sub}	0.62 %	0.62 %
P_h	0.74 %	2.58 %
T_o^{ou}	0.81 %	0.88 %

* No disturbance in refrigerant composition
** 2 % disturbance in refrigerant compositions

linear analysis, but these are affected by nonlinearities and proves to be poor in practice. This is also the case for refrigerant flow. Table 4 shows the losses of the three best candidate controlled variables (CV's). We have used the disturbances from Table 2, (except that we use only $\pm 2\,\%$ in refrigant composition) and implementation errors (n) from Table 3. Simultaneous disturbances have not been considered.

Note that controlling ΔT_{sub} requires knowledge about the composition of the refrigerant, so it may be better to control the compressor outlet temperature with only 0.26 % extra loss.

4.5 Proposed control structure

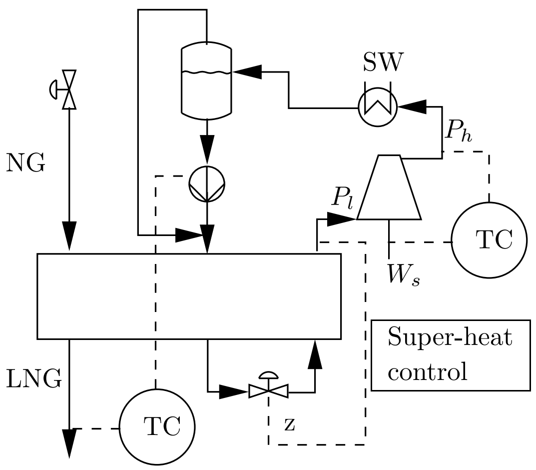


Fig. 5. Proposed control structure

First of all we should control the active constraints:

- Control T_{NG}^{out} with liquid pump
- Control ΔT_{sup} with choke valve

The remaining degree of freedom could be used to control either of the three candidates listed above (ΔT_{sub}, P_h or T_{om}^{out}).

In Figure 5 we choose to:

- Control T_{om}^{out} with compressor power W_s

Note that we in this steady state study we can only say what variables that should be maintain constant during operation.

5. CONCLUSION

We have shown that the design method specifying ΔT_{min} to design heat exchanger will result in a operating point that is not optimal. Even for a small design ΔT_{min} of 1.2 °C, the compressor power can be reduced by as much as 2.60 %. For a process that requires the large amounts of energy, such as LNG processes, this is a significant saving.

For the PRICO LNG process we have found that there will be one unconstrained degree of freedom. This degree of freedom may be used to control P_h, ΔT_{sub} or T_{om}^{out} as this will lead to self-optimizing control. Controlling ΔT_{sub} gives smaller loss, but controlling T_{om}^{out} or P_h may be easier in practice.

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