Single-cycle mixed-fluid LNG process
Part I: Optimal design

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Abstract

In this paper, we discuss the design optimization of a relatively simple LNG process; the PRICO process. A simple economic objective function is stated and used on eight different cases with varying constraints. Important constraints are discussed and the results are compared with the commercial PRICO process and with other publications.

Keywords: PRICO, LNG, design

1 Introduction

Stebbing and O'Brien (1975) reported on the performance of the first commercial PRICO plants in operation. Price and Mortko (1996) from the Black & Veatch company discussed the process and gave some key values for several of their plants. With respect to academic work, Lee et al. (2002) used the PRICO process as one of their case studies for testing their approach to design optimization. The same group later published some updated results (Del Nogal et al., 2005).

Process description: Figure 1 shows a simplified flowsheet of the PRICO process. *Natural gas* is fed to the main heat exchanger (NG HX) after some pretreatment (removal of water, CO_2 etc.) which is not included in this paper. In heat exchanger NG HX, the natural gas is cooled, liquefied and sub-cooled by heat exchange with cold refrigerant.

The *refrigerant* is partially condensed in the sea water (SW) cooler (condenser) and is fed to the NG HX and is cooled together with the natural gas stream. The refrigerant is a sub-cooled liquid at the outlet of NG HX and is expanded to the

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low pressure (P_l). The resulting two-phase mixture provides the cooling in NG HX by vaporization. The outlet from the heat exchanger (NG HX) is slightly super-heated, partly to avoid damage to the compressor.

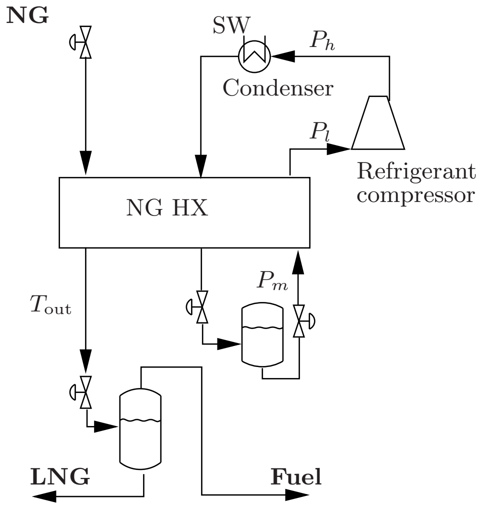

Figure 1: Simplified flowsheet of the PRICO process.

Model: The process is modelled using the gPROMS software with the accompanying Multiflash package for thermodynamic calculations. The SRK equation of state is used for thermodynamic calculations both for the natural gas and the refrigerant. The main heat exchanger is a distributed system, which for modelling purposes has been discretized into 100 cells using forward and backward finite difference method, for the cold and hot streams respectively. The resulting optimization problems are also solved using gPROMS.

1.1 Optimal design

The process structure is given and we wish to optimize the design parameters such as heat exchanger areas (A_{HOT} - heat transfer area used to cool warm refrigerant and A_{NG} - heat transfer area used to cool natural gas in the main heat exchanger) and compressor size, temperatures and pressures. This problem is quite complex, so one often uses simplified approaches. For heat exchanger design, a common approach is to minimize only the operating cost, but subject to a minimum approach temperature in all heat exchangers (Lee et al., 2002; Del Nogal et al., 2005), that is, $\Delta T_i \geq \Delta T_{\min}$ is introduced as a design constraint. However, this approach may give sub-optimal designs, see Jensen and Skogestad (2008), so instead we use the simplified total annual cost (sTAC) method presented in Jensen and Skogestad (2008). We wish to maximize the LNG production for a given compressor design, and consider the following optimization problem:

$$\min \left(-\dot{m}_{\text{LNG}} + \hat{C}_0 \left(A_{\text{HOT}}^{0.65} + A_{\text{NG}}^{0.65} \right) \right) \quad (1)$$

$$\begin{aligned} \text{subject to} \quad & W_s \leq W_s^{\max} \\ & m_{\text{fuel}} \leq m_{\text{fuel}}^{\max} \\ & c \leq 0 \end{aligned}$$

where the cost factor $\hat{C}_0$ is an adjustable parameter. For example, it may be adjusted to get a desired ΔT_{min}. $\hat{C}_0$ represents the trade-off between lower operating cost (favoured by a small $\hat{C}_0$) and lower capital cost (favoured by a large $\hat{C}_0$. The minimization is performed with respect to the design parameters (A_{HOT} and A_{NG}) and operating parameters (flows, pressures, splits etc.). The set c is additional constraints described below.

1.2 Design constraints

In design it is necessary to impose some constraints for the optimization to assure that a feasible solution is found.

Pressure: A large conventional centrifugal compressor has a maximum discharge pressure in the order of 30 to 40 bar while a vertical split centrifugal compressor may have outlet pressure up to about 80 bar (General Electric Oil and Gas, January 2007).

Compressor suction volume ($\dot{V}_{\text{suc}}$): The current maximum limit for a single flow centrifugal compressor seems to be 380000 $\text{m}^3\,\text{h}^{-1}$ (General Electric Oil and Gas, January 2007).

Compressor head: A simple correlation for the maximum head (or specific enthalpy rise) per compressor wheel is for a centrifugal compressor (see Equation 1.73 on page 37 in Lüdtke, 2004):

$$\text{Head} = \Delta h = s \cdot u^2 \qquad [\text{kJ kg}^{-1}] \quad (2)$$

where $s \approx 0.57 - 0.66$ is the work input factor and $u\,[\text{m s}^{-1}]$ is the velocity at the wheel tip.

Table 1: Maximum head, $\Delta h\,[\text{kJ kg}^{-1}]$ (Equation 2 with $s = 0.57$). The last column shows the total head for a compressor with one of each of the wheels in the table.

	Wheel diameter						
Rotational speed	1.7 m	1.6 m	1.5 m	1.4 m	1.3 m	1.2 m	**Sum 6 wheels**
3600 RPM	59.6	52.8	46.4	40.4	34.8	29.7	**263.6**
3000 RPM	41.4	36.6	32.2	28.0	24.2	20.6	**183.0**

Compressor shaft work (W_s^{max}): A maximum value for the compressor shaft work may be imposed for example due limitations the compressor itself or in the available power supply.

2 Results for optimal design

The numerical results from the optimization for eight cases are reported in Table 2. Boldface numbers indicate specifications or active contraints. We have adjusted $\hat{C}_0$ in Equation 1 to obtain $\Delta T_{\min} \approx 2.0\,°\mathrm{C}$ in the main heat exchanger (NG HX) for all cases. We have assumed $10\,°\mathrm{C}$ super-heating at the compressor inlet ($\Delta T_{\mathrm{sup}} = 10\,°\mathrm{C}$) except in Cases 3 and 4. Note from the results that it is optimal in all cases to have no C_3H_8 in the refrigerant.

Table 2: Optimal design results for eight different cases

Case	1	2	3	4	5	6	7	8
$\dot{m}_{\mathrm{feed}}$ [kg s^{-1}]	52.2	45.0	45.3	44.8	49.6	49.6	76.1	80.8
$\dot{m}_{\mathrm{LNG}}$ [kg s^{-1}]	44.6	41.7	42.0	41.5	46.3	46.3	71.1	75.8
$\dot{m}_{\mathrm{fuel}}$ [kg s^{-1}]	7.7	**3.33**	**3.33**	**3.33**	**3.33**	**3.33**	**5.0**	**5.0**
$\dot{m}_{REF}$ [kg s^{-1}]	478	475	472	443	251	298	611	617
T_{out} [°C]	**-144**	-156	-156	-156	-157	-157	-157	-156
ΔT_{sup} [°C]	**10.0**	**10.0**	11.6	**25.7**	**10.0**	**10.0**	**10.0**	**10.0**
$\Delta T_{\min}$ [°C]	1.96	1.97	1.95	2.03	1.97	1.94	2.00	2.04
η [%]	82.8	82.8	82.8	82.8	82.8	82.8	82.8	82.8
W_s [MW]	**77.5**	**77.5**	**77.5**	**77.5**	**77.5**	**77.5**	**120**	**120**
P_h [bar]	**22.0**	**22.0**	**22.0**	**22.0**	50.4	**30.0**	**30.0**	**30.0**
Pressure ratio [-]	**5.5**	**5.5**	**5.5**	**5.5**	22.5	16.6	7.3	7.2
P_l [bar]	4.0	4.0	4.0	4.0	2.24	1.81	4.11	4.16
Head [kJ kg^{-1}]	134	135	136	145	256	216	162	161
V_{suc} [m^3 s^{-1}]	84.3	83.3	84.0	83.9	75.1	**106**	**106**	**106**
UA_{HOT} [MW °C^{-1}]	38.4	40.9	41.3	39.8	18.7	22.9	51.8	52.2
UA_{NG} [MW °C^{-1}]	4.8	4.4	4.4	4.6	5.7	5.5	8.0	8.2
UA_{tot} [MW °C^{-1}]	43.1	45.3	45.7	44.4	24.4	28.4	59.8	60.4
$\hat{C}_0 \cdot 10^{-3}$ [kg s^{-1} m$^{-1.3}$]*	**110**	**120**	**130**	**107**	**37**	**51**	**3000**	**3000**
Refrigerant composition:								
x_{CH_4} [mole-%]	33.3	32.3	32.3	32.5	31.1	29.2	32.5	33.2
$x_{C_2H_6}$ [mole-%]	35.3	33.2	33.4	34.7	32.3	32.9	32.9	33.5
$x_{C_3H_8}$ [mole-%]	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$x_{n-C_4H_{10}}$ [mole-%]	25.0	24.6	24.3	22.8	26.7	30.3	23.4	23.5
x_{N_2} [mole-%]	6.4	9.9	10.0	10.0	9.9	7.6	11.2	9.8

*$\hat{C}_0$ adjusted to obtain $\Delta T_{\min} \approx 2.0\,°\mathrm{C}$

Case 1 *Nominal design using data from Price and Mortko (1996).*

We specify the LNG temperature at the exit of NG HX (T_{out}) to $-144\,°\mathrm{C}$ (Price and Mortko, 1996). The LNG production ($\dot{m} = 2.52\,\mathrm{kmole\,s^{-1}} = 44.6\,\mathrm{kg\,s^{-1}}$) is slightly larger (3.7 %) than that reported by Price and Mortko (1996) ($\dot{m}_{\mathrm{LNG}} = 2.43\,\mathrm{kmol\,s^{-1}}$*), but note that the feed composition is different and that we have neglected the removal of heavy components. On the other hand, we have not included the heating of fuel gas before re-compression to turbine fuel which would have further increased the LNG production by providing some cooling for free.

The resulting fuel gas is $7.7\,\mathrm{kg\,s^{-1}}$ and will produce about 230 MW of energy by combustion in a gas turbine (assuming 60 % efficiency and $50\,\mathrm{MJ\,kg^{-1}}$). In the remaining cases with 77.5 MW compressor we have limited the amount of fuel gas to $3.33\,\mathrm{kg\,s^{-1}}$ for (about 100 MW), which replaces the specification on T_{out}. For

*Calculated from $4.71\,\mathrm{MNm^3\,day^{-1}}$

the two last cases we have assumed 120 MW compressor so we use the specification of 5.0 kg s^{-1} fuel flowrate.

Case 2 *Constraint on the amount of fuel gas;* 3.33 kg s^{-1}

Note that the temperature out of the heat exchanger (T_{out}) is reduced from -144 to $-156\,^{\circ}$C to reduce the amount of flash gas. This results in a 6.0 % reduction in production compared with Case 1 ($\dot{m}_{\text{LNG}}$ drops from 44.6 kg s^{-1} to 41.7 kg s^{-1}). This is because we are unable to cool as much natural gas and this is not compensated for by the increased liquid fraction after expansion.

Case 3 *Optimized super-heating.*

We find by removing the constraint on super-heating, that ΔT_{sup} increases from 10.0 °C to the optimal value of 11.6 °C. This gives an 0.8 % increase in LNG production compared with Case 2. This illustrates, as discussed by Jensen and Skogestad (2007), that the optimal super-heating is not zero for a system with internal heat exchange.

Case 4 *Higher super-heating.*

In this case we specify a higher super-heating (25.7 °C, compared to the optimal of 11.6 °C). This gives only a 1.3 % reduction in LNG production compared to Case 3, which shows that the optimum is "flat" in terms of super-heating. With 0.22 °C super-heating we get a reduction of 2.3 % in LNG production compared with Case 3.

In reality, we expect that the heat transfer coefficient is lower in the super-heating section than in the vaporization section. This suggests that the optimal degree of super-heating will be lower than what we find with constant heat transfer coefficients.

Until now we have fixed the two refrigerant pressures. Specifically, the discharge pressure P_h is fixed at 22 bar and the pressure ratio, $Pr = P_h/P_l$ at 5.5 (Table 2). However, some authors have published optimization results with discharge pressure much higher than 22.0 bar (Lee et al., 2002; Del Nogal et al., 2005). They also claim that the refrigerant flowrate should be about 3-4 times the flowrate of natural gas on mole basis. For the cases up till now we have obtained a ratio of about 6, which is about 50 % higher. These two observations are closely related as the amount of refrigerant depends on the pressure ratio (Pr).

Case 5 *No pressure constraints*

Here we optimize the process without the constraint on discharge pressure and pressure ratio. We see that with the same compressor work, the production of LNG is increased from 41.7 kg s^{-1} to 46.3 kg s^{-1} (11 %) while the refrigerant amount is reduced from 14.7 kmole s^{-1} to 7.64 kmole s^{-1} (which gives a ratio 2.9 between refrigerant and LNG flowrate). To achieve this, the high pressure is increased to $P_h = 50.4$ bar and the pressure ratio is $Pr = 22$.

Some other interesting results to note are; i) the compressor suction volume actually decreases and ii) the necessary heat transfer area for the hot refrigerant stream is less than half, UA is 18.8 MW/°C compared to 40.9 MW/°C for Case 2. Both these effects are related to the fact that much less refrigerant is needed, but how can this be explained? The cooling duty per kg of refrigerant is closely related to the compressor head, [kJ kg^{-1}], which again is closely related to the pressure ratio. So increasing the compressor head (and pressure ratio) will increase the cooling duty per kg of refrigerant and thus decrease the required amount of refrigerant.

There is a potential problem with this design. A high pressure ratio usually requires more compressor stages (casings) and this may not be desirable, although some of the extra capital cost related to the extra compressor casing and higher pressure will be offset by the reduction in heat transfer area.

We wish to use only one compressor casing, which may not be feasible with the high pressure ratio of 22 in Case 5. To get a realistic design we next use performance specifications for the MCL1800 series compressor from General Electric Oil and Gas (January 2007).

Case 6 *MCL1800 series compressor.*

MCL1800 is a centrifugal compressor with casing diameter of 1800 mm. The reported maximum suction volume is 380000 m^3 h^{-1} or about 106 m^3 s^{-1}, the maximum discharge pressure is 30 bar and the maximum shaft work is 120 MW (General Electric Oil and Gas, January 2007). In this case we keep 77.5 MW as the maximum compressor shaft work to compare with the other cases, and specify a maximum compressor suction volume of 106 m^3 s^{-1} and maximum pressure of 30 bar.

Interestingly, the results show that we are able to almost match the production and pressure ratios obtained in Case 5 with realistic specifications and one compressor casing. The total head in the compressor may be achieved with one compressor casing with 5 wheels and a rotational speed of 3600 RPM, see Table 1.

Note that the suction volume V_{suc} is an active constraint for the three last cases in Table 2 where actual compressor data are utilized.

Finally, we would like to find the maximum train capacity limit for the PRICO process with a single compressor casing.

Case 7 *Again we utilize the larger MCL1800, but we allow for more shaft power, namely* 120 MW

We find that we may produce 71.1 kg s^{-1} LNG in a single PRICO train with one compressor casing using realistic design data. Note that the required compressor head is reduced from 216 kJ kg^{-1} to 162 kJ kg^{-1} compared to Case 6 so for this case we may use a slower driver (see Table 1), for example a Frame 9 gas turbine with rotational speed of 3000 RPM (General Electric Oil and Gas, January 2007).

Note that the cost factor $\hat{C}_0$ on heat exchanger areas is increased tenfolds compared to the other cases. This was necessary in order to increase $\Delta T_{\min}$ up to 2.0 °C.

Table 3: Optimal design with fixed $C_0 = 110 \cdot 10^3\,\mathrm{kg\,s^{-1}}\,\left(\mathrm{m^{-2}}\right)^{0.65}$

Alternative Case	1	2	3	4	5	6	7	8
$\dot{m}_{\mathrm{LNG}}$ [kg s^{-1}]	44.6	45.4	45.8	44.3	36.5	42.6	77.2	81.4
$\Delta T_{\min}$ [°C]	1.96	1.96	1.86	2.04	2.89	2.51	1.48	1.65
UA_{HOT} [MW °C^{-1}]	38.4	41.7	42.3	39.7	8.0	18.4	64.2	62.8
UA_{NG} [MW °C^{-1}]	4.8	4.6	4.7	4.5	2.9	4.1	10.3	10.4
UA_{tot} [MW °C^{-1}]	43.1	46.4	46.9	44.2	10.9	22.5	74.6	73.2
Change compared with Table 2								
$\Delta\dot{m}_{\mathrm{LNG}}$ [%]	0.0	8.9	9.0	-1.1	-21.2	-8.0	8.6	7.4
ΔUA_{tot} [%]	0.0	2.4	2.6	-0.45	-55.5	-21.0	24.7	21.2

There seems to be no reasons for this large increase in the cost factor and an alternative design optimization is given below.

The LNG expansion valve and the refrigerant expansion valve shown in Figure 1 may be exchanged with a combination of liquid turbine and valve. Ideally, one would prefer to do the entire expansion in a turbine, but two-phase turbines are to our knowledge not in use so it is necessary to use the combination of a liquid turbine and a valve (Barclay and Yang, 2006). The liquid turbine will then take the pressure down to slightly above the saturation pressure and the expansion valve will take care of the two-phase expansion.

Case 8 *A liquid turbine is included in the expansion of the natural gas* ① *and in the expansion of the refrigerant* ②.

The production is further increased by 6.6 % compared with Case 7. The total heat transfer area is increased by 1.0 %. Note that also for this case we had to specify a high $\hat{C}_0$ to obtain $\Delta T_{\min} \approx 2.0\,°\mathrm{C}$.

In summary, we see that some design constraints strongly affect the optimal solution. These constraints are related to the compressor performance; maximum suction volume, maximum discharge pressure, maximum head and maximum shaft work. The changes given by these constraints are illustrated in cases 5 to 7. Other constraints have less influence on the optimal solution; these are the degree of super-heating and the amount of flash gas. The changes given by these constraints are illustrated by the first four cases.

2.1 An alternative design optimization

Above we adjusted the cost factor $\hat{C}_0$ to achieve $\Delta T_{\min} \approx 2.0\,°\mathrm{C}$. For cases 7 8 this resulted in a cost factor unrealistically much larger than for the other cases. We suspect that this is due to the non-linear behaviour of $\Delta T_{\min}$. A better and simpler approach is to fix $\hat{C}_0$. Here, $\hat{C}_0 = 110 \cdot 10^3\,\mathrm{kg\,s^{-1}}\,\left(\mathrm{m^{-2}}\right)^{0.65}$ is used for all cases.

Table 3 shows key results for all eight cases with the same specifications as before.

For Case 7 we get an increase by 8.6 % in the LNG production compared to the corresponding case in Table 2 where $\hat{C}_0 = 3000 \cdot 10^3\,\mathrm{kg\,s^{-1}}\left(\mathrm{m^{-2}}\right)^{0.65}$. This is achieved by increasing the total heat transfer area by 24.7 % and ΔT_{min} is reduced from 2.0 °C to 1.5 °C. A similar increase is achieved for Case 8. For cases 5 and 6 the production is reduced compared to the results in Table 2.

3 Conclusion

The PRICO process is optimized for several different constraints and compared with the commercial process and other publications. Using compressor specifications found online we are able to increase the LNG production compared to the commercial process. Important constraints, especially concerning the compressor feasibility, are discussed.

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