



Accepted Manuscript

Title: Estimating Refrigeration Costs at Cryogenic Temperatures

Author: William L. Luyben

PII: S0098-1354(17)30139-4

DOI: <http://dx.doi.org/doi:10.1016/j.compchemeng.2017.03.013>

Reference: CACE 5763



To appear in: *Computers and Chemical Engineering*

Received date: 14-1-2017

Revised date: 15-3-2017

Accepted date: 17-3-2017

Please cite this article as: & Luyben, William L., Estimating Refrigeration Costs at Cryogenic Temperatures. *Computers and Chemical Engineering* <http://dx.doi.org/10.1016/j.compchemeng.2017.03.013>

This is a PDF file of an unedited manuscript that has been accepted for publication. As a service to our customers we are providing this early version of the manuscript. The manuscript will undergo copyediting, typesetting, and review of the resulting proof before it is published in its final form. Please note that during the production process errors may be discovered which could affect the content, and all legal disclaimers that apply to the journal pertain.

Submitted to Computers and Chemical Engineering

Estimating Refrigeration Costs at Cryogenic Temperatures

William L. Luyben

Department of Chemical Engineering
Lehigh University
Bethlehem, PA 18015
USA

January 14, 2017

Revised March 15, 2017

WLL0@Lehigh.edu; 610-758-4256; FAX 610-758-5057

Highlights

- Multi-stage compression refrigeration systems are studied.
- Flowsheets are designed to remove heat at levels from -25 °C down to -190 °C.
- A correlation is developed to estimate refrigeration costs at cryogenic temperatures.
- Predicted costs are higher than literature values.

Abstract

Conceptual process design makes extensive use of typical costs for energy sources that supply heat at various high temperatures and for energy sinks that remove heat at various low temperatures. Many books provide extensive data for energy sources such as steam, dowtherm, natural gas, coal, Bunker C and electricity over a broad range of temperatures, with energy cost naturally increasing with increasing temperature.

Much less information is available about the cost of refrigeration at various temperature levels. The purpose of this paper is to greatly expand and update the information for estimating cryogenic refrigeration costs. Process source temperatures from -25 °C to -190 °C are considered using single-stage and up to four-stage compression refrigeration systems with different refrigerants used at each stage. Refrigeration costs vary from \$18 per GJ at -25 °C to \$360 per GJ at -190 °C.

Keywords

Compression refrigeration; multi-stage cycles; conceptual design

1. Introduction

Conceptual design is a vital step in the development of a practical chemical process after the basic chemical kinetic information and phase equilibrium data have become available. The objective of conceptual design is to explore alternative flowsheets in terms of technical feasibility and, most importantly, in terms of economic attractiveness. Many simplifying assumptions are made at the conceptual design stage,

and extensive use of practical design heuristics is required to make it possible to explore a wide range of possible flowsheets and operating conditions.

The economics of chemical processes are dominated by issues of yield and conversion since the cost of raw materials and the value of the products are typically an order of magnitude larger than all other costs. However, energy costs are usually the second largest issue. Therefore it is essential to have some realistically accurate values for the cost of energy.

All chemical engineering design textbooks^{1,2,3,4,5,6} provide extensive tables of the costs of providing heat or work energy to a process over a range of temperature levels. For example, Turton et al¹ suggests using \$7.78 per GJ for low-pressure steam (6 bar with a saturation temperature of 160 °C). The suggested cost for high-pressure steam (42 bar at 254 °C) is \$9.88 per GJ. Electricity is given a cost of \$16.8 per GJ.

Much less information is presented on the cost of refrigeration. Douglas⁵ gives no refrigeration costs. Turton et al¹ lists \$7.89 per GJ for removing energy at -20 °C and \$13.11 per GJ for removing energy at -50 °C. How do we estimate the refrigeration cost if the process operates at -190 °C?

There are significant differences and inconsistencies in the suggested refrigeration costs among the textbooks. Seider et al² lists \$8.50 per GJ at -67 °C. This is a lower cost at a lower temperature than given by Turton et al¹ (\$13.11 per GJ at -50 °C). Peters et al⁴ gives the cost for ammonia refrigeration at 1 °C with incorrect units of \$0.64 per kW (kW is power not energy). Peters et al³ gives a cost of \$6.56 per GJ at 1 °C, which is consistent with the Turton numbers.

The purpose of this paper is to provide more accurate and updated refrigeration cost data over a wider range of temperatures. The approach is to design complete compression refrigeration cycles for a range of temperatures, developing detailed operating conditions and calculating economic costs (capital investment and energy cost). In each case the energy removed from the cryogenic process is 1 MW. The total annual cost of each case is calculated, which yields a value of refrigeration cost at the selected temperature in \$ per GJ.

It is important to note that the results presented in this work predict much larger refrigeration costs than those given in the textbooks.

2. Compression Refrigeration

The refrigeration systems considered in this work use compression refrigeration in one or more stages with different refrigerants in each stage. Saturated vapor from the evaporator of each stage is compressed to a higher pressure. The superheated vapor from the compressor is fed to a heat exchanger that is the evaporator of the next stage. The vapor is de-superheated and condensed. The liquid is flashed over a valve down to a lower temperature and recycled back to the initial evaporator. The energy is ultimately rejected in the final stage to a “sink” at ambient temperature, typically cooling water or air.

Only a single stage is required to remove heat down to about -50 °C. For lower temperatures, a single stage becomes infeasible. There are two pressure restrictions for each stage:

1. The evaporator pressure (compressor suction) should be around atmospheric to avoid high compressor suction volumes and high compression ratios. Lower sink temperatures require lighter (lower boiling point) refrigerants to achieve this condition.
2. The compressor discharge pressure should be lower than the critical pressure of the refrigerant so that two-phase condensation is possible in the condenser. This limits the condenser (source) temperature in each stage, which limits the evaporator (sink) temperature of the subsequent stage that must be at a lower temperature to achieve heat transfer.

An appropriate refrigerant must be selected for each stage since the stages operate over different temperature ranges.

In the following section, a range of cryogenic process temperatures are studied. For each temperature, a compression refrigeration system is designed that removes 1 MW of heat from the cryogenic process. The designs of the multi-stage cascades of compression refrigeration are based on the important 1989 paper by Colmenares and Seider⁷.

3. Design Basis

Many parameters must be selected in the design of a multi-stage compression refrigeration process. The most important is the value used for the differential temperature driving force for heat transfer in the evaporator/condenser heat exchanger. We assume that a ΔT of 5 °C is the economic optimum in these low-temperature processes. As this specification is reduced, heat-transfer area increases, which increases capital investment in heat exchangers. However, smaller ΔT 's reduce compression requirements (smaller compression ratios and smaller refrigerant circulation flowrates). As the economic results presented in a later section demonstrate, compressor installed capital costs represent about 75% of the total capital in these systems. Compressor work represents almost 100% of the energy cost. Therefore a small ΔT of 5 °C is selected.

There are a variety of heat-transfer situations in this system. In a typical stage, the evaporator in that stage is a heat exchanger with boiling refrigerant on the cold side. The hot side stream is superheated vapor coming from the compressor of the previous lower-temperature stage. Therefore the hot side consists of two heat-transfer regimes. De-superheating occurs in the first part (gas on the hot side and boiling liquid on the cold side), so an overall heat transfer-coefficient $U = 0.28 \text{ kW m}^{-2} \text{ K}^{-1}$ is used in this section.. Condensation occurs in the second part (condensing vapor on the hot side and boiling liquid on the cold side), so an overall heat transfer-coefficient $U = 1.2 \text{ kW m}^{-2} \text{ K}^{-1}$ is used in this section.

However, in the last stage where cooling water is flowing through the cold side of the heat exchanger, the de-superheating section has gas on the hot side and liquid on the cold side, so an overall heat transfer-coefficient $U = 0.2 \text{ kW m}^{-2} \text{ K}^{-1}$ is used in this final (highest temperature) stage. Condensation occurs in the second part (condensing vapor on the hot side and liquid on the cold side), so an overall heat transfer-coefficient $U = 0.85 \text{ kW m}^{-2} \text{ K}^{-1}$ is used in this section.

In the de-superheating section of the condenser, a log-mean temperature driving force is used with the hot stream entering at the compressor discharge temperature and leaving as saturated vapor at the dewpoint temperature of the refrigerant at the compressor discharge pressure. In the condensing section, the driving force is constant at

5 °C (the difference between the hot condensing refrigerant in the stage and the cold boiling refrigerant of the subsequent stage).

If the compressor discharge temperature in any intermediate stage is greater than 50 °C, a water-cooled desuperheating heat exchanger is used so that the process stream enters the evaporator at 50 °C. This occurs in the stage below the last high-temperature stage in most of the designs. An overall heat transfer-coefficient $U = 0.2 \text{ kW m}^{-2} \text{ K}^{-1}$ is used in these exchanger with a log-mean temperature driving force. On the hot side, the superheat vapor stream enters at the compressor discharge temperature and leaves at 50 °C. On the cold side, the cooling water enters at 30 °C and exits at 45 °C.

Capital investment in heat exchanger area is estimated from Douglas⁵ using the following equation.

$$\text{Capital Cost (\$)} = 7296[\text{Area (m}^2\text{)}]^{0.63}$$

Cooling water is assumed to be supplied at 30 °C and returned to a cooling tower at 45 °C with a cost of \$14.8 per m³ as given by Turton et al¹. Compressors have an ASTM polytropic efficiency of 80%. Compressor work is supplied by electrical energy at a cost of \$16.8 per GJ given by Turton et al¹. Aspen Economics is used to estimate the capital investment (installed equipment cost) of the compressors. The cost of the refrigerant is assumed to be insignificant since it is a one-time inventorying of material

Total annual cost TAC is a widely used economic indicator in conceptual process design. For conventional chemical process in which there is a balance between capital investment and energy costs, a fairly short payback period of three years is typically used. However, in a process that is very capital intensive like a utility, a larger payback period is more realistic. A multi-stage compression refrigeration process represents this situation since the capital investment in expensive compression equipment is large compared to the energy cost. Therefore a 10-year payback period is assumed, using the capital cost as the installed equipment cost (compressors and heat exchangers in all stages) and energy cost from compression work.

$$TAC (\$/y) = \text{Energy} + \frac{\text{Installed Equipment Cost}}{10}$$

Refrigeration cost of each case is calculated by dividing TAC by the heat transferred from the low-temperature source, which in all cases is 1 MW.

$$Cost (\$/GJ) = \frac{TAC (\$/y)}{(1 MJ/s)(GJ/1000MJ)(31.54 \times 10^6 s/y)}$$

It should be noted that the results given in Table 2 are all for a refrigeration load of 1 MW. The capital and energy cost can be easily scaled up from the 1 MW case to larger industrial refrigeration loads. The conventional 0.6 power can be used for capital, while energy cost scales directly with load. An example illustrating the economy of scale is to consider a process at -190 °C with a refrigeration load of 10 MW. Scaling up the capital cost at 1 MW to 10 MW increases capital cost from \$24,370,000 to \$97,020,000. Energy cost increase from \$9,043,000 per year to \$90,043,000 per year. The refrigeration cost becomes \$316 per GJ instead of the \$368 per GJ.

4. Flowsheets

Cryogenic process source temperatures are explored ranging from a process operating at -25 °C (so the evaporator must boil refrigerant at -30 °C) to a process operating at -190 °C (so the first-stage evaporator must boil refrigerant at -195 °C). The refrigerants considered are familiar low-boiling chemical components. Nitrogen has a normal boiling point of -195.8 °C. Methane is -161.5 °C. Ethylene is -103.7 °C. Propylene is -53.7 °C. Ammonia is -33.4 °C. The critical pressures of these components are ammonia 112.8 atm, nitrogen 33.5 atm, methane 45.8 atm, ethylene 50.5 atm and propylene 45.3 atm.

Recalling the discussion in the previous section about how the evaporator pressure should be approximately 1 atm, the selection of an appropriate refrigerant in an evaporator operating at -30 °C would indicate that ammonia should be a possible candidate. Since the compressor discharge pressure should be far below the critical pressure of ammonia, this refrigerant is expected to be a viable candidate refrigerant.

Any of the lighter components could be used in the evaporator since its pressure would be above atmospheric. However, the compressor discharge pressure is limited to

about half the critical pressure of the component, which would lower the possible condenser temperature and may require multiple stages.

Flowsheets for several cases are given Figures 1 to 6. The process heat duty is 1 MW in all cases. As the process source temperature changes, the number of stages and the compressor power change.

4.1 Single Stage:

Figure 1 shows conditions for a process source temperature of $-25\text{ }^{\circ}\text{C}$ using propylene refrigerant. The evaporator pressure is 2.14 bar giving an evaporator temperature of $-30\text{ }^{\circ}\text{C}$. Compressor discharge pressure is set at 21 bar (well below the critical pressure of propylene) so that the superheated vapor can be condensed at $50\text{ }^{\circ}\text{C}$ using cooling water. Compressor work is 1.038 MW and refrigerant flowrate is 532 kmol/h of propylene, which removes the specified 1 MW from the process in the evaporator. The 1 MW load and the 1.038 MW compressor work are rejected to cooling water.

In Figure 2 ammonia is the refrigerant that boils at $-30\text{ }^{\circ}\text{C}$ when the pressure is 1.2 bar in the evaporator. Compressor discharge pressure is set at 20 bar so that the ammonia refrigerant can be condensed at $50\text{ }^{\circ}\text{C}$ using cooling water. Comparing the ammonia cycle with the propylene cycle shows that less compressor work is required despite having a higher compression ratio ($20/1.2 = 17$ for ammonia versus $21/2.14 = 9.8$ for propylene). This occurs because of the smaller refrigerant circulation rate (238 kmol/h of ammonia versus 532 kmol/h for propylene) that reflects the larger heat of vaporization of ammonia (23.4 kJ/mol versus 18.5 kJ/mol for propylene).

If the process temperature is lowered to $-50\text{ }^{\circ}\text{C}$, as shown in Figure 3, the evaporator pressure must be reduced to 0.715 bar when propylene refrigerant is used since the evaporator temperature is now $-55\text{ }^{\circ}\text{C}$. Compressor work increases to 1.911 MW since the compression ratio is larger and the refrigerant circulation increases to 665 kmol/h.

4.2 Two Stages:

For a process temperature of $-100\text{ }^{\circ}\text{C}$, two stages are required with ethylene in the first and propylene in the second. Figure 4 shows the flowsheet. The first evaporator has boiling ethylene at $-105\text{ }^{\circ}\text{C}$. The evaporator pressure is 0.955 bar (the vapor pressure of ethylene at $-105\text{ }^{\circ}\text{C}$). The compressor requires 1.239 MW to compress 511 kmol/h of saturated vapor ethylene to 25 bar. Since the compressor discharge temperature is $119\text{ }^{\circ}\text{C}$, a water-cooled heat exchanger is installed that cools the gas to $50\text{ }^{\circ}\text{C}$. It is still superheated when it enters the next evaporator in which it is de-superheated to saturated vapor and condensed to saturated liquid ethylene at 25 bar, which corresponds to a temperature on the hot side of the next heat exchanger (the evaporator in the next stage) of $-20.8\text{ }^{\circ}\text{C}$.

The cold side of the second-stage evaporator must be $5\text{ }^{\circ}\text{C}$ colder than the hot side, so its temperature must be $-25.8\text{ }^{\circ}\text{C}$. The refrigerant in the second stage is propylene, so the pressure is 2.5 bar. The heat transferred in this evaporator (1.713 MW) is the sum of the heat transferred from the process into the first evaporator (1 MW) plus the first-stage compressor work (1.239 MW) minus the heat that is removed in the water-cooled heat exchanger (0.5295 MW). The required circulation rate of propylene is 690.5 kmol/h.

The second-stage compressor discharge pressure is 21 bar so that the propylene can be condensed at $50\text{ }^{\circ}\text{C}$ using cooling water. The heat transferred in the final condenser (2.837 MW) is the sum of the work in the two compressors ($1.239 + 1.125\text{ MW}$) plus the process load (1 MW) minus the energy removed in the water-cooled heat exchanger (0.5295 MW).

These results demonstrate the increase in compressor power as the process source temperature is lowered. For the same 1 MW process load and using propylene refrigerant, a temperature of $-25\text{ }^{\circ}\text{C}$ requires 1.038 MW, a temperature of $-50\text{ }^{\circ}\text{C}$ requires 1.911 MW and a temperature of $-100\text{ }^{\circ}\text{C}$ requires 2.364 MW.

4.3 Three and Four Stages:

Using the same procedure described above for each stage, appropriate refrigerants are selected for each stage that satisfy pressure and temperature criteria. Flowsheets with process temperature of $-150\text{ }^{\circ}\text{C}$ and $-190\text{ }^{\circ}\text{C}$ are shown in Figures 5 and 6. Three stages

are required for the -150 °C case with methane refrigerant used in the first stage, followed by ethylene and propylene in stages 2 and 3. As shown in Figure 5, total compressor work grows to a large 6.256 MW in the -150 °C case. Notice that the compressor work and the refrigerant flowrate increase from stage to stage as we move up in temperature level.

Figure 6 shows that four stages are required for the -190 °C case with nitrogen refrigerant used in the first stage, followed by methane, ethylene and propylene. Total compressor work now is a whopping 17.06 MW. Remember that the process load is only 1 MW in the first lowest-temperature stage. Table 1 gives details of operating conditions and equipment sizes for each stage.

These examples clearly demonstrate the large compressor work requirements required to provide refrigeration at cryogenic temperatures. Compressors are expensive to purchase and to operate since they require high-level energy to provide the power to their motor or steam-turbine drive. The economic effects are quantified in the following section. The objective is to see how the cost of providing refrigeration depends on the cryogenic temperature level.

5. Economic Results

Table 2 summarizes the results for the various cases. Figure 7 shows how compressor capital, total capital investment, energy cost and annual capital cost vary with the process temperature. The dominant effect of compressors is clearly evident. Keep in mind that a 10-year payback period is used in these calculations.

As expected, refrigeration cost increases as the temperature of the cryogenic process decreases. This dependence is exponential as Figure 8 demonstrates. A correlation that should be useful for conceptual design purposes is given in the following equation.

$$Cost(\$ / GJ) = \exp[2.452 - 0.01863 T(^{\circ}C)]$$

Several specific issues can be seen in the results given in Table 1. Single-stage systems can be used down to -50 °C, two-stage to -100 °C and three-stage to -150 °C.

Below this temperature level, four or higher stages are required. Capital investment is dominated by compressor costs, which are about 75% of total capital cost.

Using a 10-year payback period, total annual cost is about 50% capital and 50% energy. Using smaller payback periods would significantly increase refrigeration cost because the refrigeration system is quite capital intensive.

A 1-stage propylene system at $-50\text{ }^{\circ}\text{C}$ requires more compressor work (1.911 MW) than a 2-stage ethylene/propylene system at $-75\text{ }^{\circ}\text{C}$ (1.698 MW). However, the compressor capital cost per MW of work of the 2-stage system (\$1,572,000 per MW) is larger than the 1-stage system (\$1,455,000 per MW) because two compressors are required.

The ammonia system at $-25\text{ }^{\circ}\text{C}$ requires less compressor work, but the compressor capital cost is essentially the same as the propylene compressor (\$1,375,000). This is due to the higher compressor discharge pressure in the ammonia case ($252\text{ }^{\circ}\text{C}$ versus $79\text{ }^{\circ}\text{C}$). An ammonia two-compressor process was studied to see if it would improve the economics. The first compressor discharge pressure was 4.9 bar, giving a discharge temperature of $93\text{ }^{\circ}\text{C}$ and requiring 0.2911 MW. An intermediate cooler reduced the temperature to $50\text{ }^{\circ}\text{C}$ before feeding the second compressor whose discharge pressure was 20 bar, discharge temperature was $201\text{ }^{\circ}\text{C}$ and power 0.3869 MW. So the total work was reduced from 0.7143 to 0.6780 MW by using a 2-stage compressor with intermediate cooling. However, the capital cost of the two compressors was larger (\$2,470,000) than the cost of the single compressor (\$1,500,000) due to the economy of scale.

6. Conclusion

Cryogenic refrigeration costs have been quantitatively analyzed to update and expand estimations for purposes of conceptual design. Results are show that refrigeration costs are significantly higher than the suggestions given in the literature. A design correlation is developed.

Detailed example flowsheets are provided for multi-stage compression refrigeration processes requiring one to four stages with different refrigerants in each stage. Heat removal systems for processes operating over temperatures ranging from $-25\text{ }^{\circ}\text{C}$ down to $-190\text{ }^{\circ}\text{C}$ are presented.

References

1. Turton, R., Bailie, R. C., Whiting, W. B., Shaeiwitz, J. A., Bhattacharyya, D. *Analysis, Synthesis and Design of Chemical Processes*, 4th Ed. Prentice Hall (2012) p. 237.
2. Seider, W. D., Seader, J. D., Lewin, D. R. *Product & Process Design Principles* 2nd Ed. Wiley (2003) p. 566.
3. Peters, M., Timmerhaus, K. D., West, R. E. *Plant Design and Economics for Chemical Engineers*, 5th Ed. McGraw-Hill (2003) p. 898.
4. Peters, M., Timmerhaus, K. D. *Plant Design and Economics for Chemical Engineers*, 4th Ed. McGraw-Hill (1991) p. 815.
5. Douglas, J. M. *Conceptual Design of Chemical Processes* McGraw-Hill (1988) p. 568.
6. Ulrich, G. D. *A Guide to Chemical Engineering Process Design and Economics* Wiley (1984) p.325.
7. Colmenares, T. D., Seider, W. D. "Synthesis of Cascade Refrigeration Systems Integrated with Chemical Processes" *Comput. Chem. Eng.* 13 (1989) 247-258.

Figure Captions

- Figure 1 – Single stage for -25 °C process; propylene refrigerant at -30 °C
Figure 2 – Single stage for -25 °C process; ammonia refrigerant at -30 °C
Figure 3 – Single stage for -50 °C process; propylene refrigerant at -55 °C
Figure 4 – Two stage for -100 °C process; first-stage refrigerant at -105 °C
Figure 5 – Three stage for -150 °C process; first-stage refrigerant at -155 °C
Figure 6 – Four stage for -190 °C process; first-stage refrigerant at -195 °C
Figure 7 – Effect of process temperature on design parameters
Figure 8 – Effect of process temperature on refrigeration cost

Fig. 1 - Single Stage for -25 °C Process; Propylene Refrigerant at -30 °C

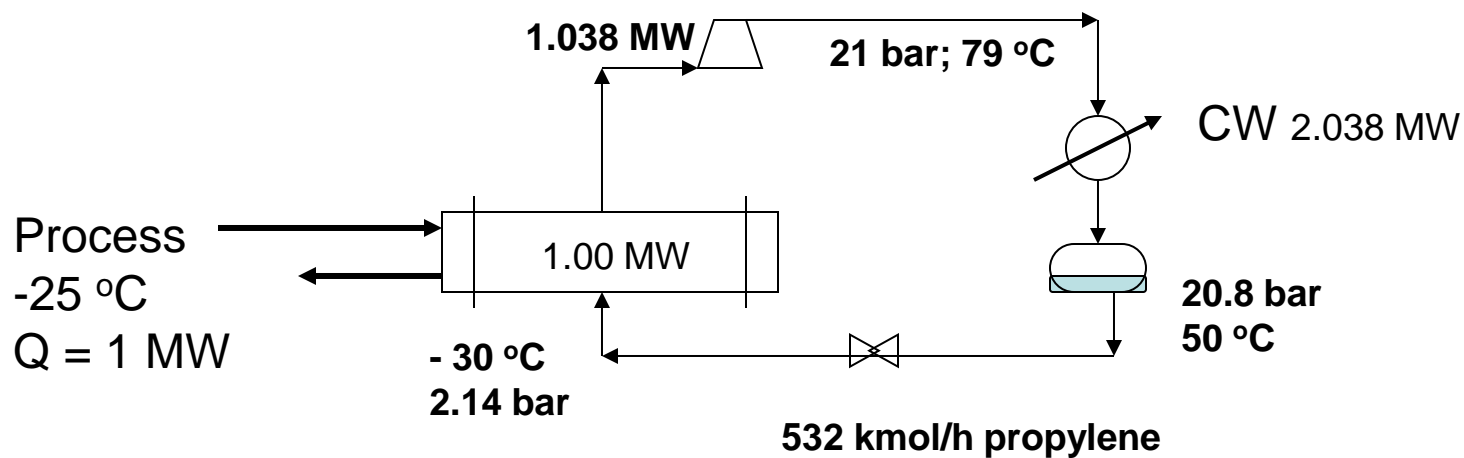


Fig. 2 - Single Stage for -25 °C Process; Ammonia Refrigerant at -30 °C

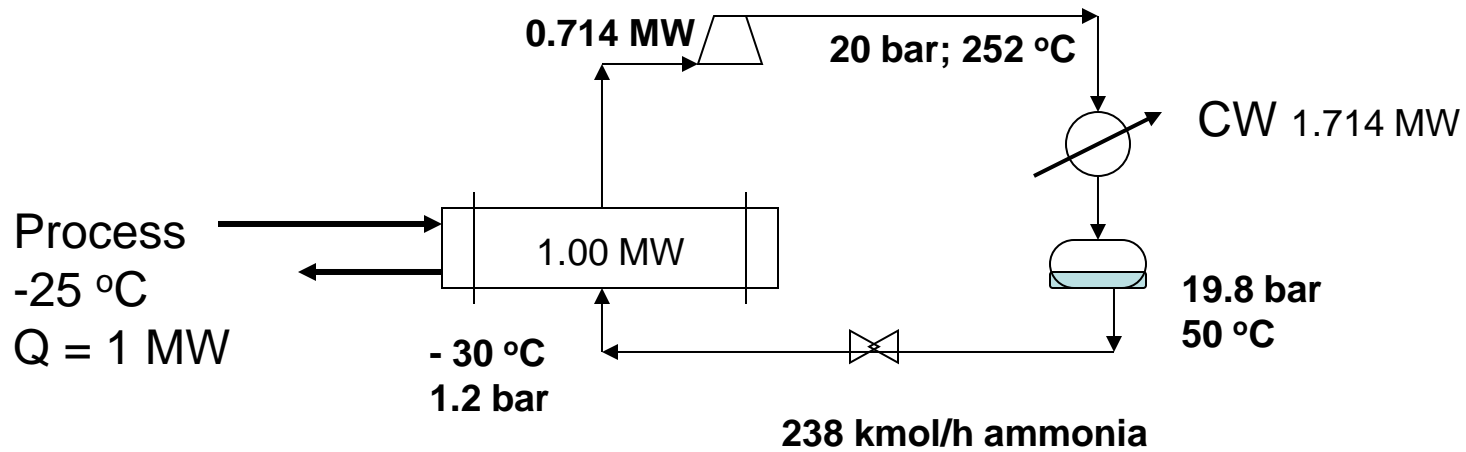


Fig. 3 - Single Stage for -50 °C Process; Propylene Refrigerant at -55 °C

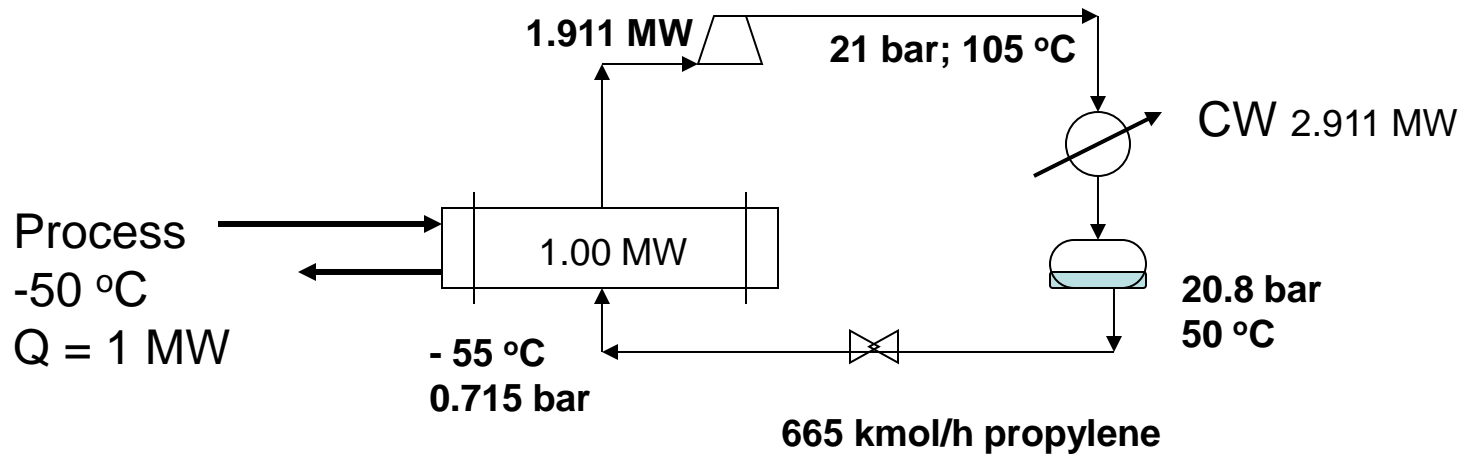


Fig. 4 - Two Stage for -100 °C Process; First-Stage Refrigerant at -105 °C

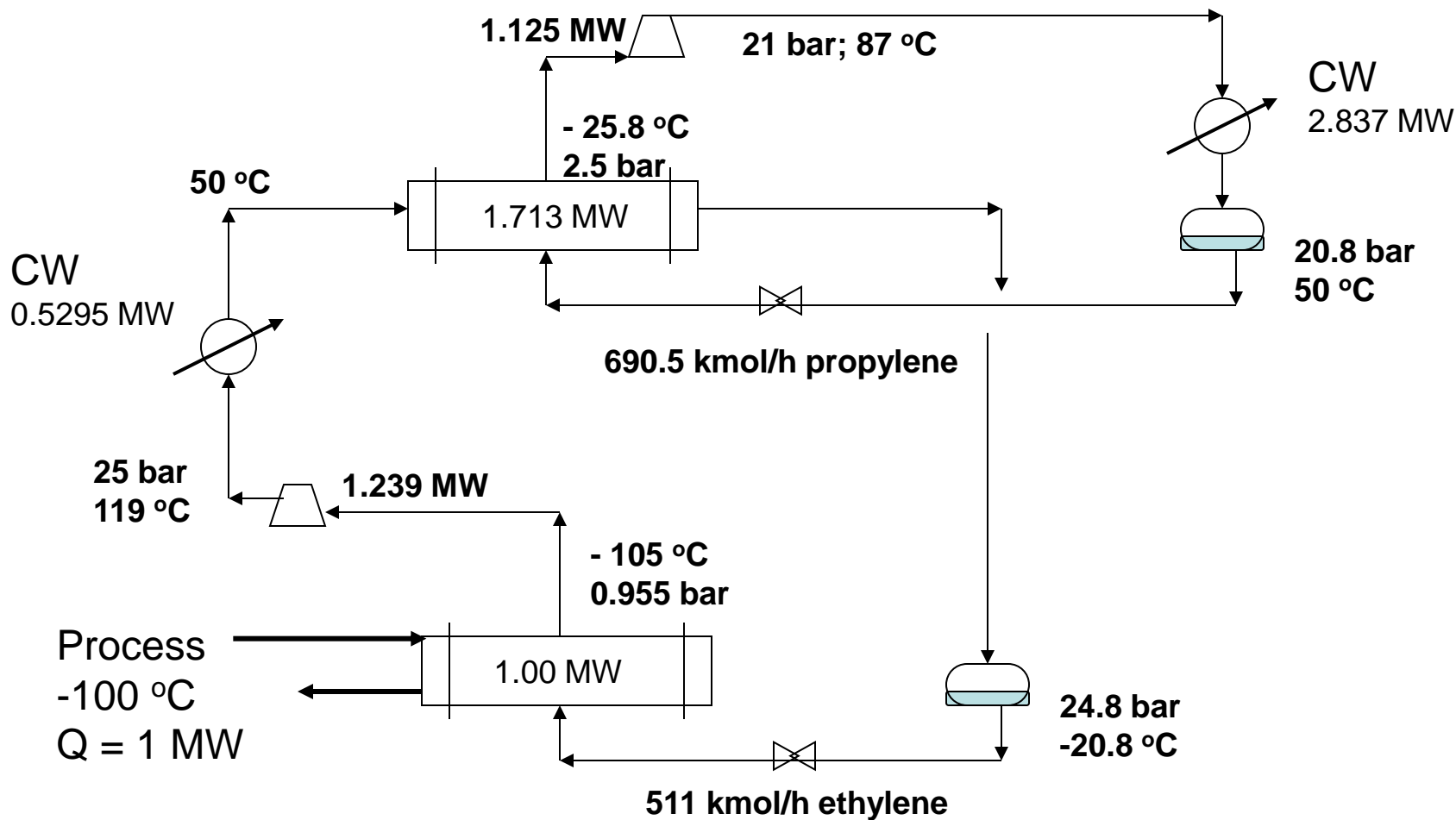


Fig. 5 - Three Stage for -150 °C Process; First-Stage Refrigerant at -155 °C

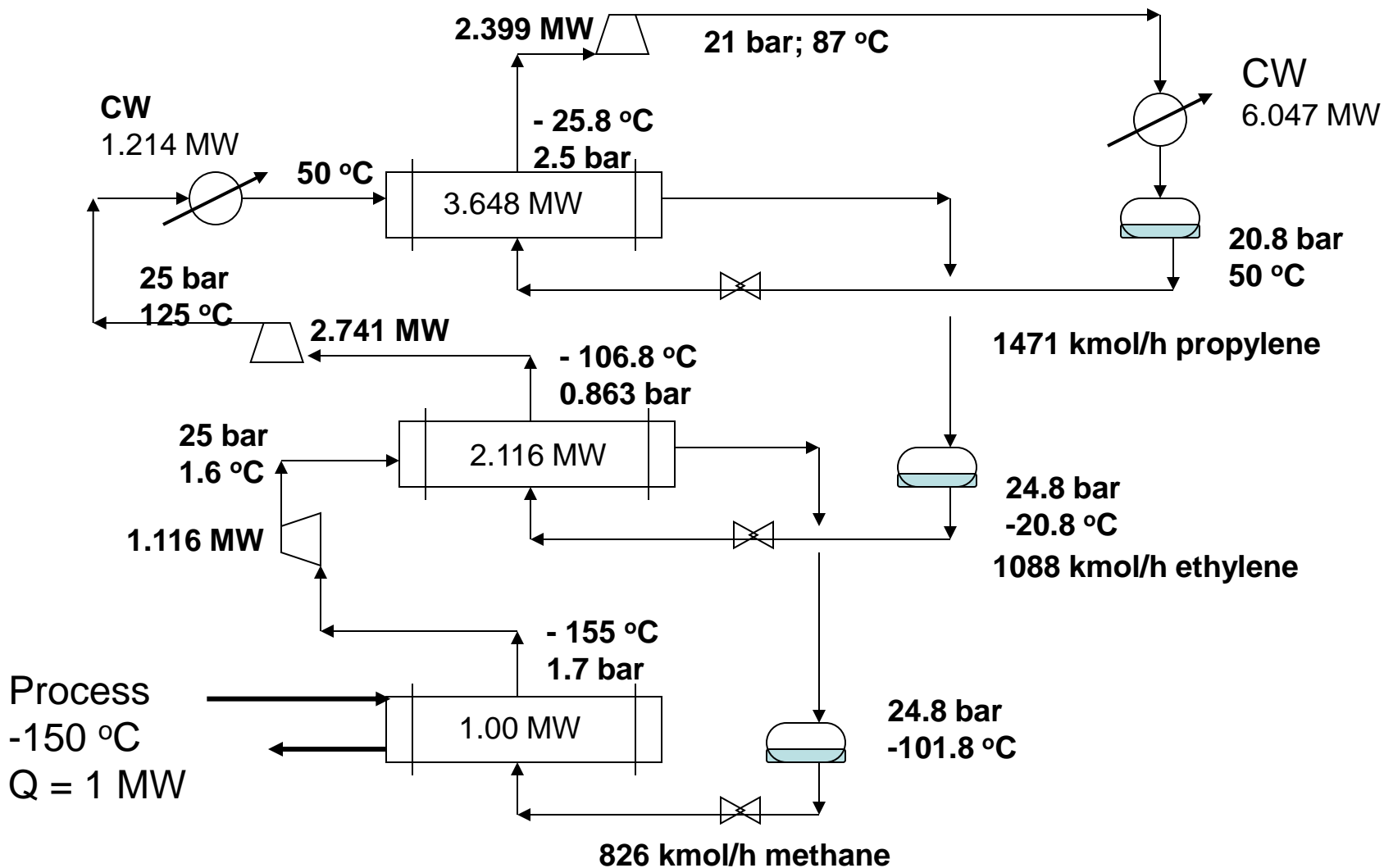


Fig. 6 - Four Stage for -190 °C Process; First-Stage Refrigerant at -195 °C

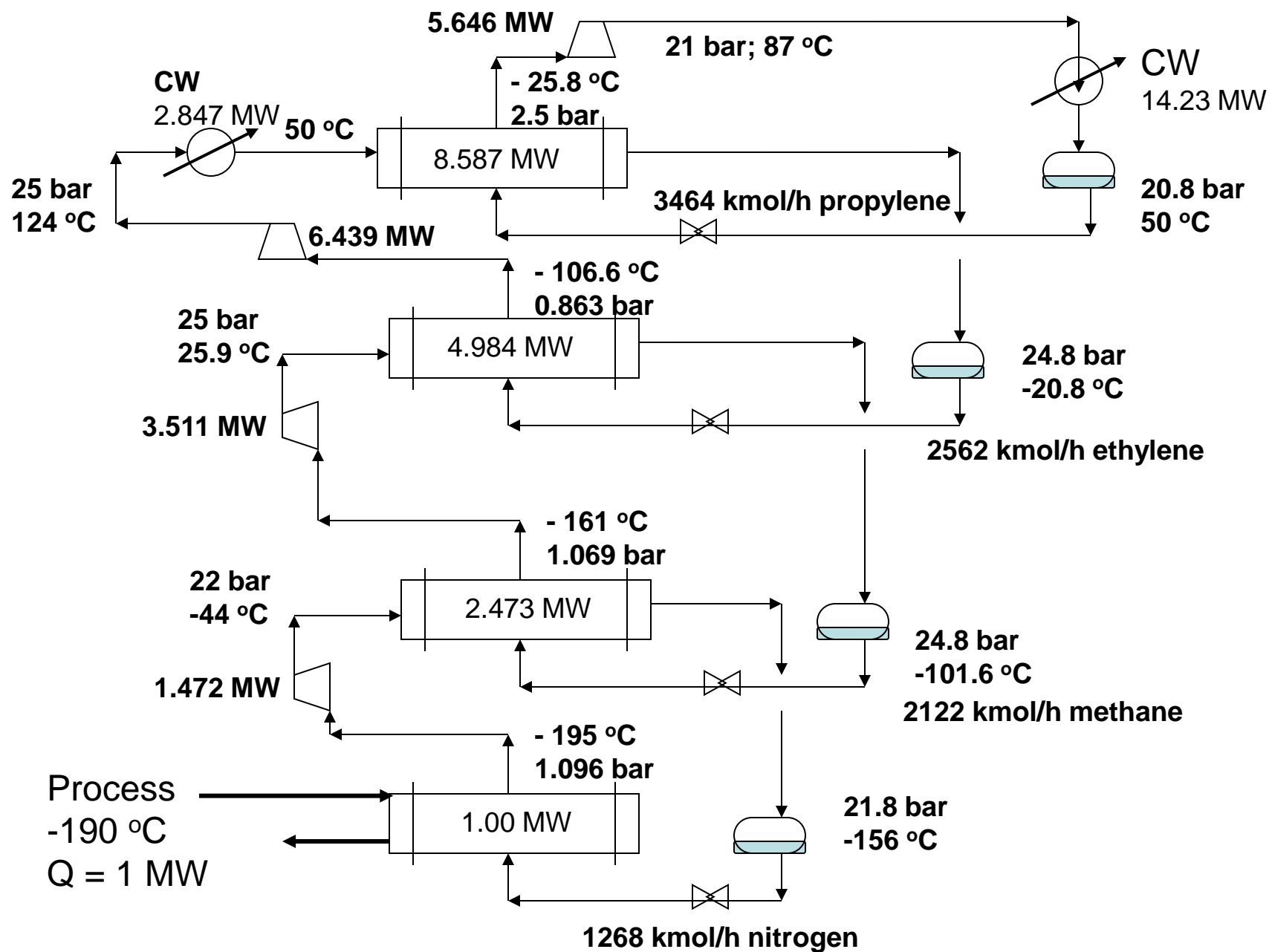


Fig. 7 – Effect of Process Temperature on Design Parameters

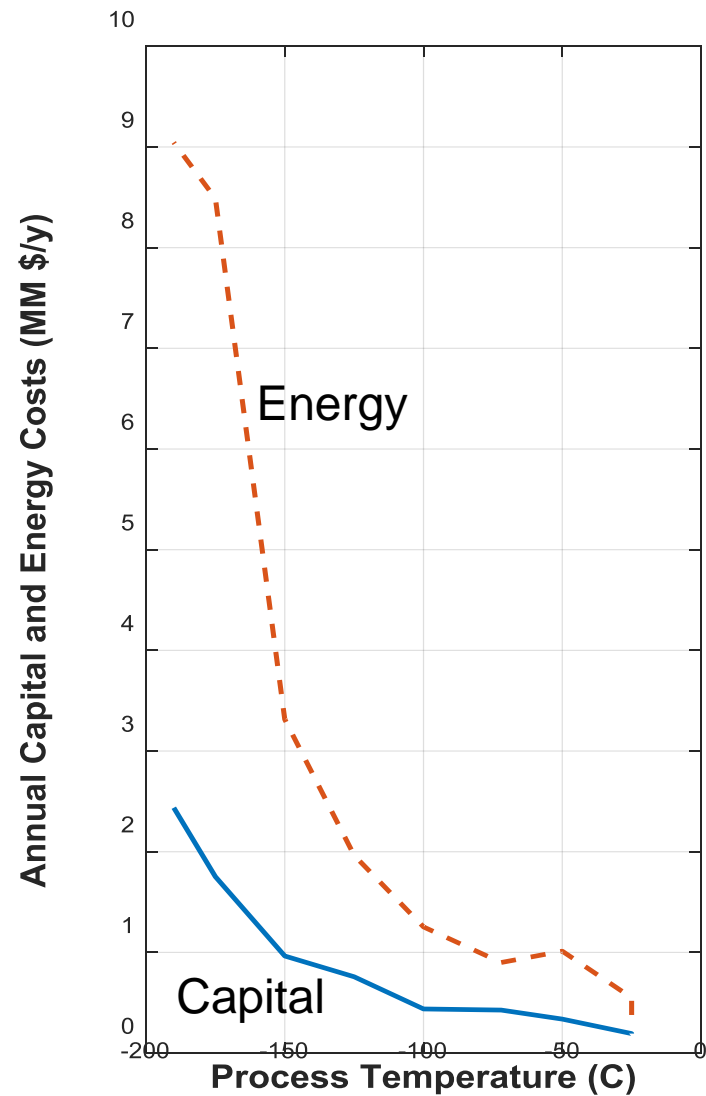
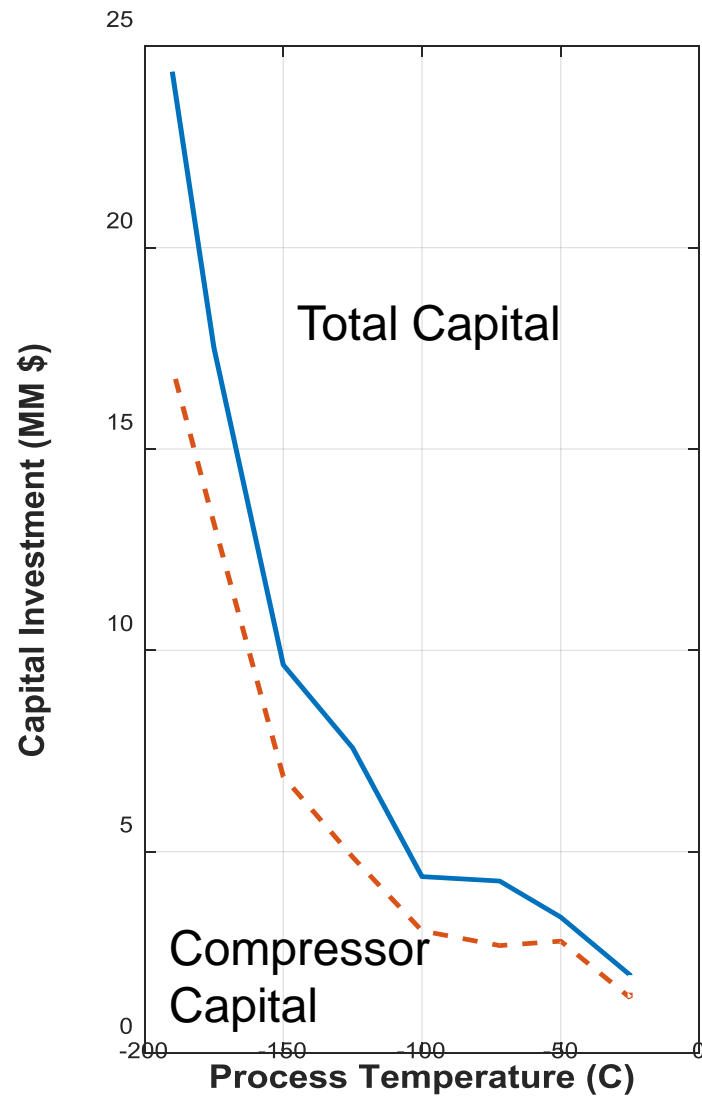


Fig. 8 – Effect of Process Temperature on Refrigeration Cost

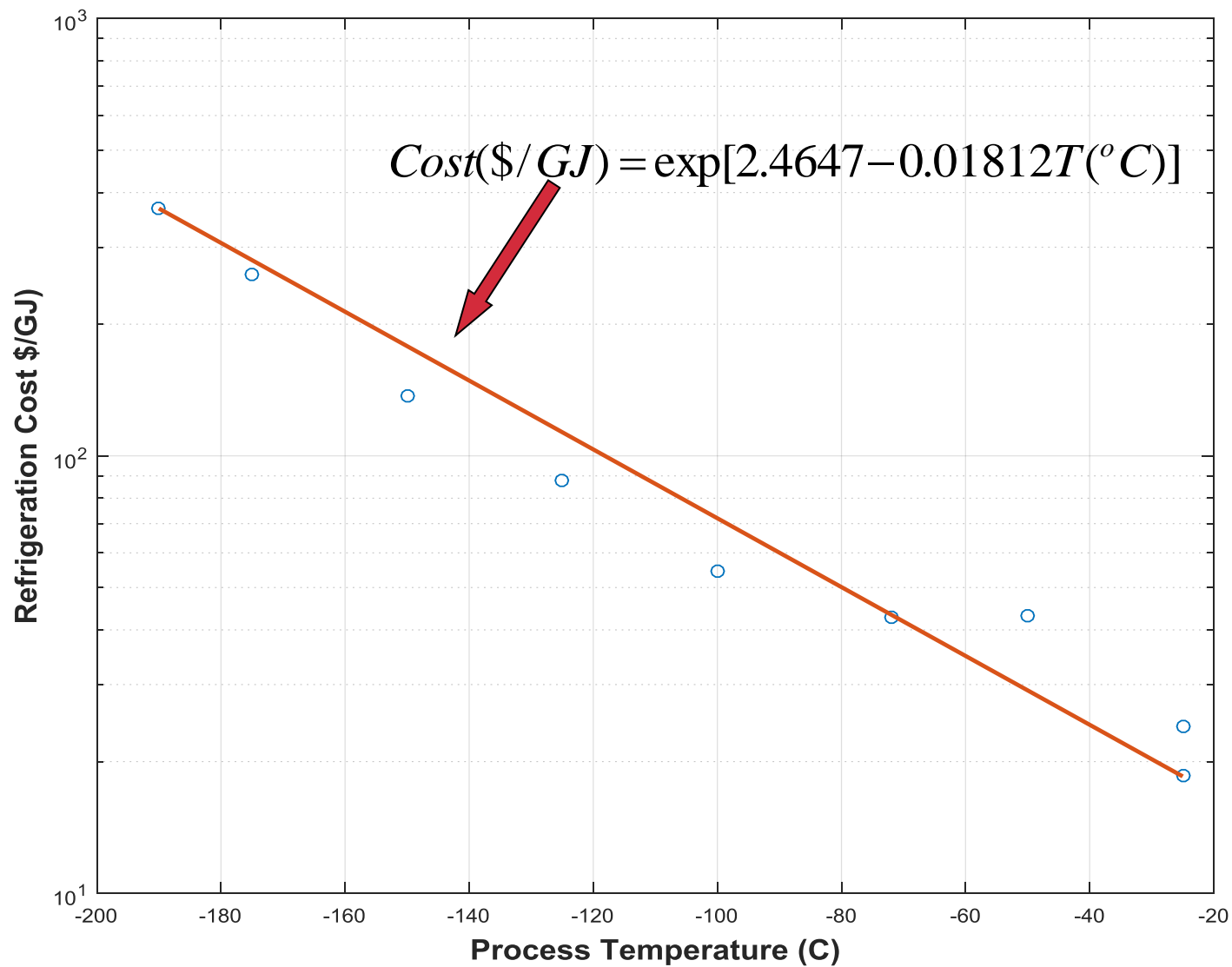


Table 1 – Process Details for -190 °C Four-Stage Process

		Stage 1	Stage 2	Stage 3	Stage 4	Condenser
Coolant		N ₂	CH ₄	C ₂ H ₄	C ₃ H ₆	CW
Flow	kmol/h	1268	2122	2562	3464	42,000
Temp	°C	-195	-161	-106.2	-25.8	50
Pressure	bar	1.096	1.069	0.863	2.6	20.8
Q	MW	1.00	2.473	4.984	8.586	14.23
Evap Area	m ²	167	321	772	1681	1661
Evap Cost	10 ⁶ \$	0.2029	0.3955	0.6959	1.300	1.150
Compressor						
Power	MW	1.472	3.511	6.639	5.646	-
T _{in}	°C	-195	-161	-106.6	-25.8	-
T _{out}	°C	-44	25.9	124	87	-
P _{in}	Bar	1.096	1.069	0.863	2.5	-
P _{out}	Bar	22	25	25	21	-
Cost	10 ⁶ \$	1.714	3.610	8.110	3.610	

Table 2 – Economic Results for 1 MW Refrigeration Load

Tcold (°C)	Refrigerant	No. Stages	Power (MW)	Capital Total (10 ⁶ \$)	Capital Comp. (10 ⁶ \$)	Energy (10 ⁶ \$/y)	TAC* (10 ⁶ \$/y)	Refrig. Cost (\$/GJ)
-25	NH ₃	1	0.7143	1.936	1.500	0.3784	0.5849	18.5
-25	C ₃ H ₆	1	1.038	1.937	1.375	0.5500	0.7568	24.0
-50	C ₃ H ₆	1	1.911	3.375	2.780	1.013	1.363	43.2
-75	C2/C3	2	1.698	3.994	2.669	0.8996	1.318	41.8
-100	C2/C3	2	2.365	4.094	3.035	1.253	1.686	53.5
-125	C1/C2/C3	3	3.707	6.974	4.868	1.966	2.710	85.9
-150	C1/C2/C3	3	6.256	8.841	6.868	3.315	4.250	135
-175	N2/C1/C2/C3	4	16.02	16.29	13.16	6.369	8.090	256
-190	N2/C1/C2/C3	4	17.07	21.89	17.04	9.043	11.35	360

* 10-year payback period