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A Simple Approach to Calculate/Minimize the Refrigeration Power Requirements

Asma Etoumi

Additional information is available at the end of the chapter

Abstract

The vapor compression refrigeration cycle is the most common method used for removing heat from a lower temperature level to a higher temperature level using a mechanical work. At lower temperatures (typically lower than −40°C), complex refrigeration schemes, such as cascaded refrigeration cycles, may be needed, increasing the complexity of the models used to predict the power requirements. This chapter introduces a new linear refrigeration model to predict the shaft power demand of the refrigeration cycle given the cooling demand, the condensing, and evaporation temperatures. The refrigeration model is based on regression of rigorous simulation results. This chapter also proposes a new systematic optimization method for minimizing the work consumed in refrigeration system. The methodology employs nonlinear model to find the optimum refrigeration temperature levels and their cooling duties. To solve the nonlinear problem, generalized reduced gradient algorithm is used. A case study is presented to demonstrate the advantage using of the proposed methodology. Results show that the difference in power prediction between rigorous simulation and the new refrigeration model is about 10%. The work consumed in refrigeration cycle can be reduced to 9% compared to the base case when the operating conditions are optimized.

Keywords: The coefficient of performance, multi-stage cycle, cascade cycle, an ethylene cold-end process

1. Introduction

Refrigeration systems are commonly used to provide cooling to sub-ambient processes. The most common refrigeration system in use today is the vapor compression refrigeration cycle [1]. Heat is extracted from a lower temperature heat source and pumped to a higher temperature by means of the work of the compressor. This higher temperature might be to an

external cooling utility (e.g., cooling water), a heat sink within the process or to another refrigeration system [2]. The power demand of the cycle depends strongly on the temperature at which cooling is required, the temperature at which the refrigerant is condensed, as well as the type of refrigerant being used [3].

Generally, a simple refrigeration cycle (i.e., a single-stage compression cycle) cannot be used to provide cooling at very low temperature due to industrial limitation of refrigerant [4] and a complex cycle (e.g., a multi-stage cycle) or cascaded is used as an alternative. Complex refrigeration system that utilizes a multi-stage compressor presents lower energy consumption when compared to the simple cycle [2]. Although complex cycle reduces power consumption, the design and optimization of this cycle are challenging because there are a large number of design alternatives and, consequently, their design fundamental interaction.

Branan [5] presented graphs that help prediction of power requirements for simple and multi-level refrigeration cycles at various temperature ranges using propane, propylene, ethane, or ethylene as the refrigerant. In preliminary design stage and in optimization where evaluation of a large number of cycles may be required, shortcut methods may be preferred because they allow faster evaluation without needing detailed specification of refrigeration design parameters (e.g., refrigerant mass flow rate, cooling duty, and the partition temperature in cascaded cycle). A shortcut method to predict the coefficient of performance (COP) of simple vapor compression cycles for pure refrigerants under the assumption of isentropic compression was proposed by Shelton and Grossman [6]. The shortcut model predicted the COP by using system temperatures and thermodynamic data of refrigerant (i.e., specific heat capacity and molar latent heat of vaporization). This chapter proposes a new refrigeration model to predict the net power demand for various design options (refrigerants and configuration) of the refrigeration cycle. The new proposed model predicts the actual coefficient of performance as function of the ideal performance (i.e., the Carnot cycle). This chapter also addresses refrigeration integration with sub-ambient process streams and provides a systematic methodology required for operational optimization of refrigeration cycles. Inputs to the optimization model include process stream data and initial estimate of operating conditions. The outputs of optimization are evaporation temperatures and cooling duties of each level and shaft work of each stage. The proposed optimization is less complicated than recently published work of Montanez-Morantes et al. [7]. Also, it is very useful for students who do not have strong mathematical background.

This chapter is organized in the following way: first, an introduction that outlines the basic configurations on which the model is based and that illustrates how the complex cycles can be decomposed into an associated simple cycles is presented. Then, the approach that is used to develop new refrigeration models is presented. After that, two examples are introduced to illustrate the effectiveness of the new refrigeration model for predicting the power demand in multi-level refrigeration cycles. Section 3 introduces a systematic methodology for the optimization of operating conditions in multi-level refrigeration cycles. A case study is presented in Section 4 to demonstrate the benefit of the use of the optimization method proposed in this chapter. Finally, the conclusions of the chapter and the recommendations for future work are presented in Section 5.
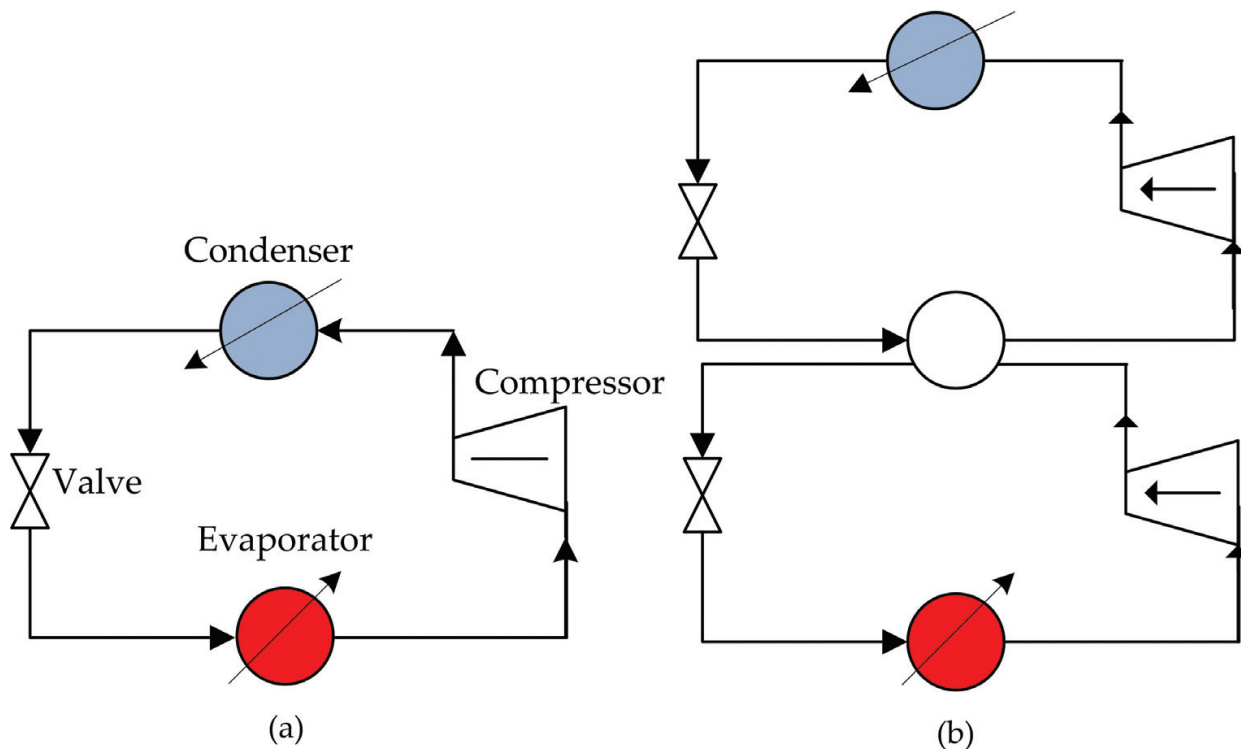
1.1. Refrigeration cycle configuration

1.1.1. Simple vapor-compression cycle

Figure 1(a) shows a simple vapor-compression cycle (VCC), which consists of four components: compressor, condenser, expansion valve, and evaporator. In a simple refrigeration cycle, heat is absorbed from a process stream through evaporation of a liquid refrigerant in an evaporator. The amount of heat transferred to the refrigerant in the evaporator is called the refrigeration load, Qevap. The evaporated refrigerant stream is compressed, and the heat from the refrigerant stream is rejected to the external heat sink. The amount of heat rejected to the external heat sink in the condenser equals the summation of the refrigeration load and the shaft work consumed in the compressor. The VCC assumes isobaric condensation and evaporation, isentropic compression, evaporation of refrigerant to a saturated vapor state, and condensation to a saturated liquid state.

1.1.2. Cascaded refrigeration cycle

Cascade cycles are usually applied in two cases [8]: (1) when the temperature range between condensation and evaporation cannot be covered by any single refrigerant and (2) when use of a single refrigerant cycle consumes more work than a cascade cycle.

Cascaded cycles are composed of two or more refrigeration cycles, where each cycle employs a different refrigerant. The low temperature cycle and a high temperature cycle, as shown in **Figure 1(b)**, are connected to each other through a heat exchanger, which acts as an evaporator

Figure 1. (a) Simple vapor-compression cycle; (b) cascade refrigeration cycle.

for the high temperature cycle and a condenser for the low temperature cycle. The partition temperature, which is the temperature of the evaporator of the upper cycle, is an important design parameter. It influences the total shaft work requirement of the cycle as will be seen in Section 1.2.1.

1.1.3. Multi-stage refrigeration cycle

A multi-stage cycle is normally implemented when the pressure ratio between the heat rejection and heat absorption pressures is high and when cooling is required at different temperature levels. For example, in a two-level cycle, illustrated in **Figure 2(a)**, the cooling duty is satisfied at two different evaporation temperatures using a single refrigerant expanded to two different pressure levels. Introducing two cooling levels reduces the refrigerant flow in the low temperature cycle that in its turn reduces the overall power requirement of the cycle [2].

1.1.4. Decomposing of complex refrigeration cycle

Decomposing a complex cycle into an assembly of simple vapor compression cycles is an alternative approach for predicting the power demand of a complex cycle. For example, a
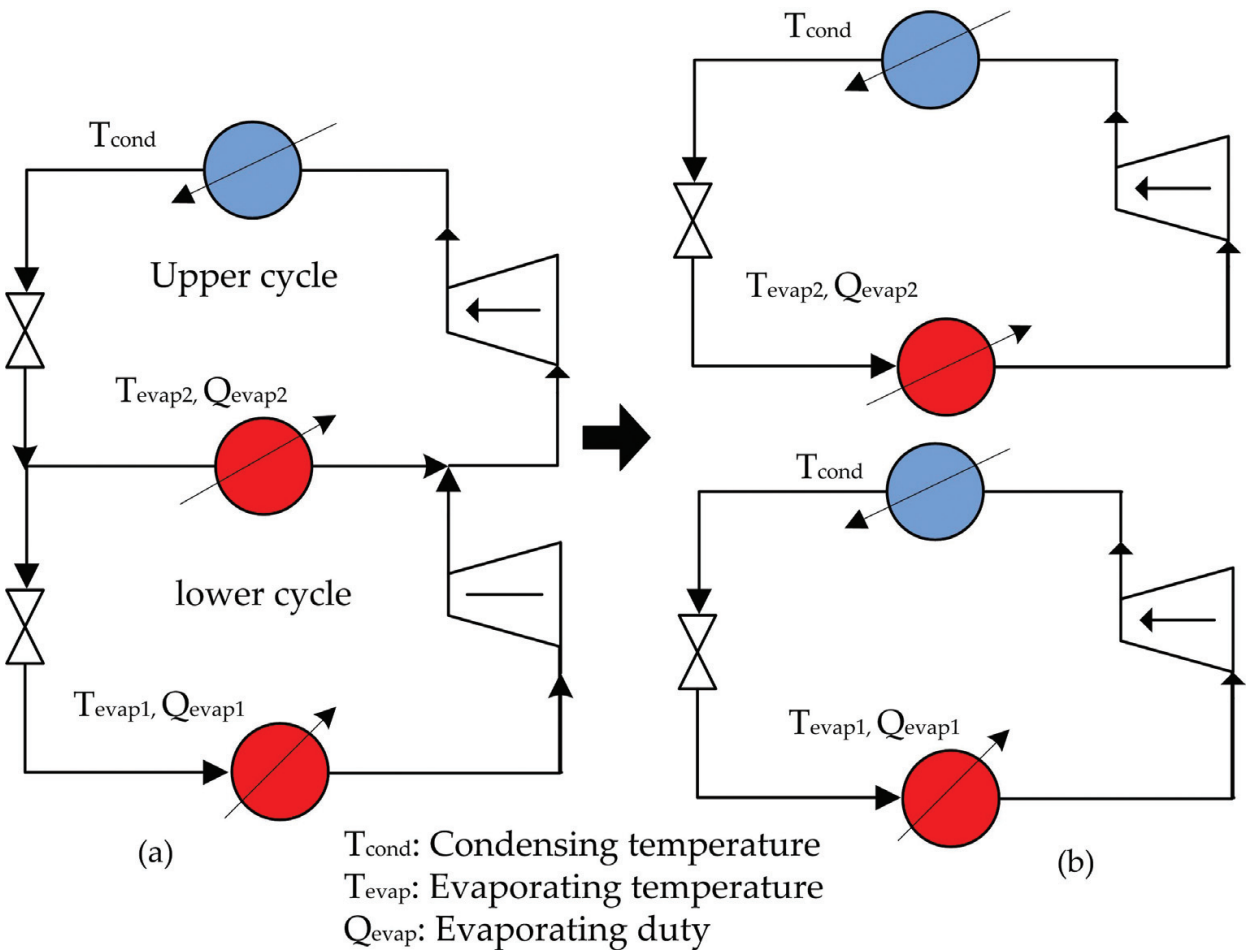
Figure 2. (a) Multi-level refrigeration cycle—two heat sources and a single heat sink; (b) decomposition into simple cycles.

multi-stage cycle with multi-refrigeration levels and a single rejection level can be decomposed into simple cycles, as shown in **Figure 2(b)**. The multi-stage cycle, such as the one shown in **Figure 2(a)**, is decomposed into two simple cycles operating in parallel. One simple cycle operates between level 1 and ambient with cooling load given by the evaporator for level 1. The other simple cycle operates between level 2 and ambient with cooling load given by the evaporator for level 2 [2].

The decomposition approach constitutes a useful for the synthesis, analysis, and design of complex refrigeration system. The main advantage of decomposing approach is that power requirement can be predicted easily using a shortcut method and various design options can be screened quickly. However, some errors might arise when the complex cycle is compared with the decomposed simple cycles. This error is caused by the mixing effects at the inlet to the compressor in the case of a cycle with multiple refrigeration levels or the inlet to the expander for a multi-rejection level cycle [9].

1.2. Thermodynamic performance of refrigeration system

The performance of a refrigeration system can be characterized by an actual coefficient of performance COP_{act} that is defined as the ratio of the heat absorbed Q_{evap} to the shaft work consumed W:

$$COP_{act} = \frac{Q_{evap}}{W} \tag{1}$$

An ideal refrigeration cycle could be based on a Carnot cycle, for which the ideal coefficient of performance COP_{id} can be defined by:

$$COP_{id} = \frac{T_{evap}}{T_{cond} - T_{evap}} \tag{2}$$

Where T_{evap} is the evaporating temperature (K) and T_{cond} is the condensing temperature (K).

As a rule of thumb, the shaft work can be approximately estimated with a coefficient of performance ratio, η, typically equal to 0.6 [1, 2].

$$W = \eta \frac{Q_{evap}}{COP_{id}} \tag{3}$$

The next section examines the possibility of using the typical value for η (0.6) to calculate the shaft work requirement of a cascade refrigeration cycle.

1.2.1. Example 1: performance and shaft work evaluation of a cascade refrigeration cycle

This example evaluates the coefficient of performance ratio and the shaft work requirement of a cascade refrigeration cycle for a range of condensing temperatures (40, 30 or 20°C), cooling duty of 523 kW, and evaporation temperature of −82°C. In the cascaded refrigeration cycle, as

shown in **Figure 1(b)**, ethylene and propylene are used in the lower and upper cycles, respectively.

In this example, the total shaft work requirement will be calculated using rigorous simulation and compared with the shaft work predicted using Carnot model (η = 0.6). Aspen HYSYS is applied for simulation of the cycles with physical properties calculated by choosing Soave-Redlich-Kwong as the fluid package. In the simulation, the partition temperature T_{part} between the two cycles is optimized to minimize the total shaft work of the cascaded cycle. T_{part} is allowed to change between the lowest temperature that the upper cycle can operate at, which is the normal boiling point for the refrigerant of the upper cycle, and the maximum temperature at which the lower cycle can reject the heat. At each partition temperature, the total shaft work of the cascaded cycles is calculated by adding the shaft work consumption of the upper cycle and the lower cycle. Then, the optimal partition temperature is identified by optimizing the shaft work as shown in **Figure 3**.
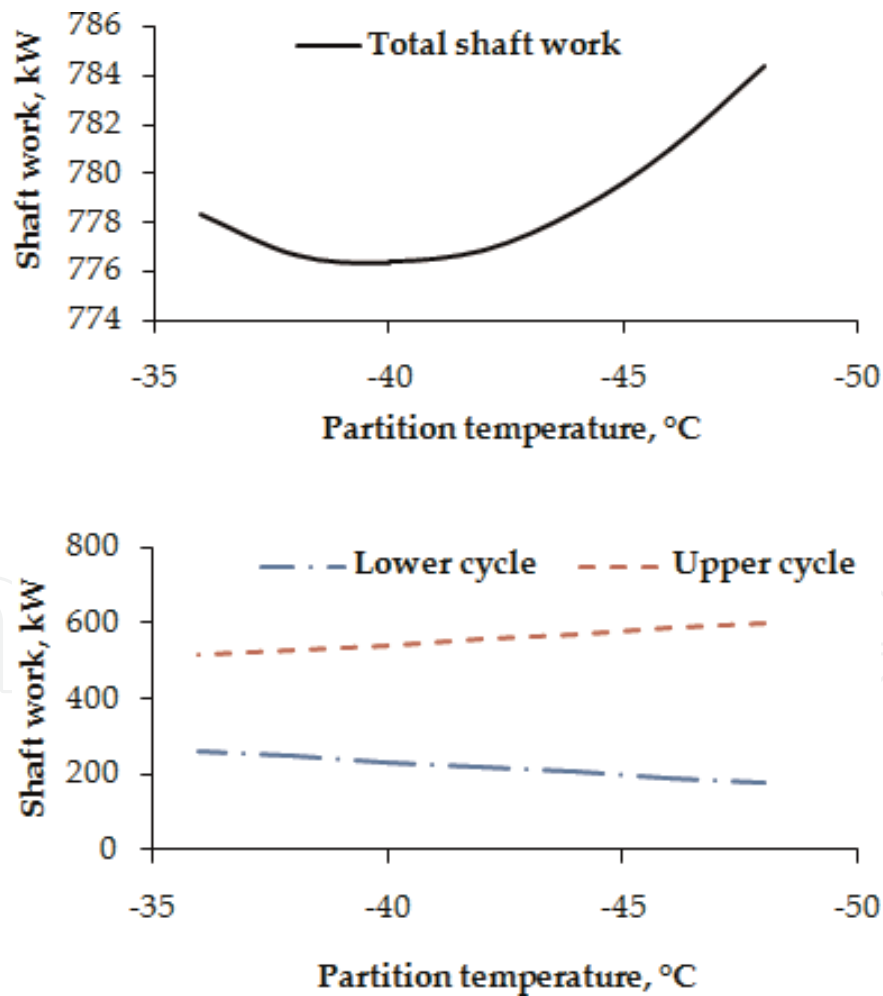
Figure 3. Calculation the optimal partition temperature of ethylene-propylene cascade cycle (T_{evap} = −82°C, T_{cond} = 40°C and Q_{evap} = 523 kW).

Q_{evap} (kW)	T_{evap} (°C)	T_{cond} (°C)	T_{part} (°C)	COP_{id}	COP_{act}	η	Shaft work (kW)		%Error
							(HYSYS)	(η = 0.6)	
523	−82	40	−40	1.57	0.67	0.43	776	556	28
		30	−44	1.71	0.79	0.46	663	510	23
		20	−47	1.88	0.92	0.49	570	464	19

Table 1. Comparison of total shaft work calculated by rigorous simulation and Carnot model using fixed value of coefficient of performance ratio, η.

Results in **Figure 3** show that the optimum partition temperature is closest to the first evaporation level of the upper cycle rather than the last evaporation level of the lower cycle; a similar conclusion was reported by Lee [8].

In **Table 1**, it may be seen that the coefficient of performance ratio (η) is below 0.6, which is the typical value for η [2]. In terms of shaft work requirement, the results in **Table 1** show that the error is significantly larger if the 'typical' ratio of 0.6 is used to predict shaft work requirement of the refrigeration cycle. Therefore, using a single value for the efficiency factor η is not the correct way to evaluate the refrigeration system performance. Section 2 proposes a new shortcut approximation of COP_{act} [3].

2. New shortcut model for refrigeration cycle

This section presents the approach that is used in building a new refrigeration model for predicting the coefficient of performance of a refrigeration cycle [3]. The first step starts with generating performance data using a rigorous simulation package, Aspen HYSYS, where it is assumed that the detailed thermodynamic and unit operation models provide a relatively realistic representation of the refrigeration cycle. Inputs to the simulation software include the refrigerant evaporating temperature, process cooling duty and refrigerant condensing temperature. Rigorous simulations of refrigeration cycles are carried out with the following assumptions: (1) Soave-Redlich-Kwong equation of state is used to calculate thermodynamic and physical properties, (2) a centrifugal compressor that has an adiabatic efficiency of 75% compresses the refrigerant, (3) let-down valves are adiabatic, (4) there is negligible pressure drop in heat exchangers and pipe work and there are no heat gains or losses, (5) the refrigerant leaves the condenser as a saturated liquid and leaves the evaporator as a saturated vapor, (6) the temperature difference between the process source stream temperature and the evaporating temperature is 5°C, and (7) the condensing temperature is variable, to account for heat rejection to ambient media or other heat sinks. The simulation outputs include the compressor power demand and the refrigerant condenser duty. The simulation is repeated for an appropriate range of operating conditions (evaporation and condensing temperatures). The inputs and outputs are then used to correlate the actual COP with the ideal COP. A simple linear relationship between COP_{id} and COP_{act} is obtained, as shown in equations below, for a simple refrigeration cycle using ethylene or propylene as a refrigerant [3].

For ethylene,

$$COP_{act} = 0.741 COP_{id} - 0.81 \tag{4}$$

For propylene,

$$COP_{act} = 0.758 COP_{id} - 0.747 \tag{5}$$

For ethylene-propylene cascade cycle,

$$COP_{act} = 0.596 COP_{id} - 0.213 \tag{6}$$

The potential benefit of the linear model is that it is the fast and easy evaluating refrigeration power demand, as will be illustrated in the following example.

2.1. Case study 1: Evaluation of multi-level refrigeration cycles using decomposition approach and the new refrigeration model

This case explores the use of the new refrigeration model for estimating the power demand in two types of multi-level cycle, with propylene as the refrigerant. The model results will be compared with HYSYS simulation results to demonstrate the usefulness of the model for estimating the net power demand in the complex refrigeration cycles. In this work, the Peng-Robinson equation of state will be used to calculate fluid and thermodynamic properties.

2.1.1. Prediction power requirement in multi-level refrigeration cycle: two heat sources and a single heat sink

The power demand in the multi-level cycle is estimated, for the case data given in **Table 2**, by representing the multi-level refrigeration cycle as a two parallel simple cycle, as shown in **Figure 2(b)**. It is clear from the results in **Table 3** that the refrigeration model is under predicting

	Temperature (°C)	Duty (kW)
Heat source 1	−40	3000
Heat source 2	−12.75	3000
Heat sink	30	–

Table 2. Case data – two heat sources and one sink.

Modelling approach	Shaft work (kW)	%Error
Multi-level cycle (HYSYS)	981 + 2203 = 3184	–
Two simple cycles (shortcut model)	1937 + 922 = 2859	10
Two simple cycles (HYSYS)	1943 + 977 = 2920	8

Table 3. Predicted shaft work requirement for multi-level cycle – two heat sources and a single heat sink using HYSYS and new shortcut model [3].

the compression shaft work requirement by 325 kW. The error between the refrigeration model and HYSYS predictions for the overall shaft work is about 10%. This error occurs because of the mixing of the saturated stream from the low-pressure evaporator with the superheated compressor outlet from the low-pressure compression stage [2]. The superheated inlet conditions to the high-pressure compression stage lead to an overall increase in the power requirement when compared with two simple cycles operating in parallel, as shown in **Figure 2(b)**.

2.1.2. Prediction power requirement in multi-level refrigeration cycle: a single heat source and two heat sinks

The single heat source and two heat sinks cycle can also be presented as a two parallel simple cycle, as shown in **Figure 4(b)**. **Table 4** shows process data for the analyzed case study. The minimum approach temperature is assumed to be 3°C.

The results in **Table 5** show that the error between the power demand predicted by the proposed refrigeration model and that predicted by rigorous simulation software is about 4%. This error comes from the mixing effects at the inlet into the throttle valve, as shown in **Figure 4(a)**. However, this scale of error should be acceptable for preliminary estimation of refrigeration power consumption.

In summary, the case study shows that although there is some error associated with the decomposition approach, the refrigeration model still can predict the power demand within reasonable accuracy. The predicted power demand is shown to be within 10% of that of more accurate simulation models. The simplicity of the refrigeration model enables its use for
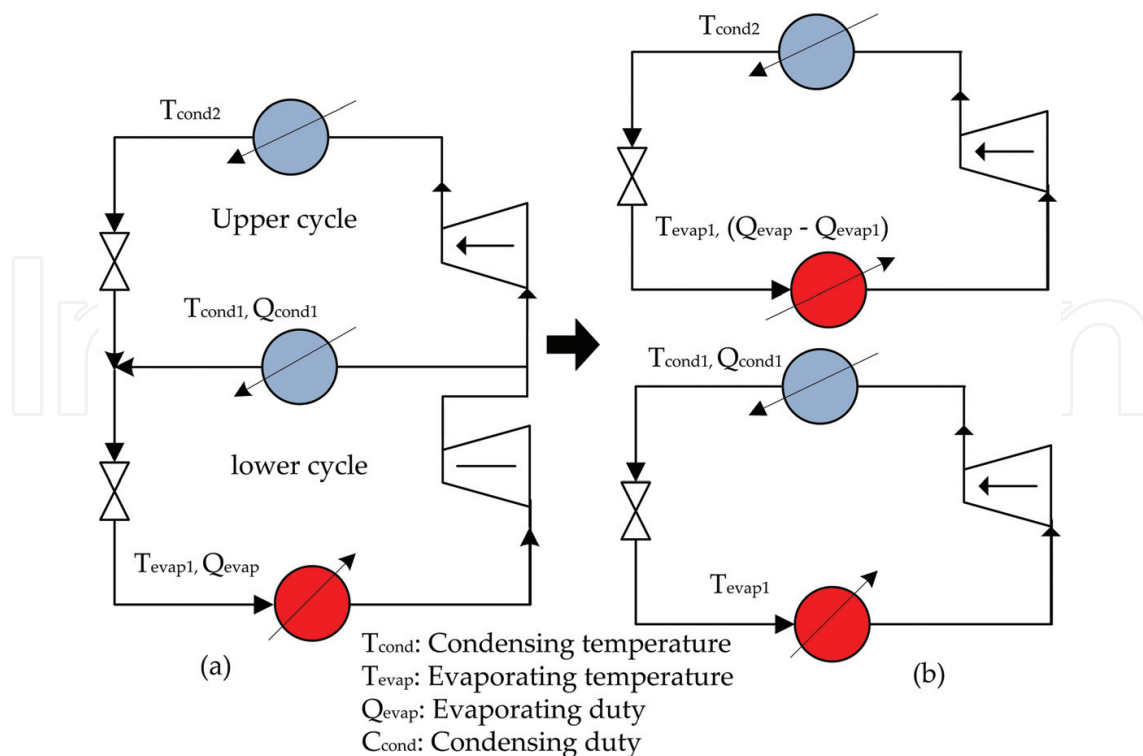
Figure 4. (a) Multi-level refrigeration cycle—a single heat source and two heat sinks; (b) decomposition into simple cycles.

	Temperature (°C)	Duty (kW)
Heat source 1	−40	3000
Heat sink 1	17	2227
Heat sink 2	37	–

Table 4. Case data – one heat source and two sink.

Modelling approach	Shaft work (kW)	%Error
Multi-level cycle (HYSYS)	1615 + 276 = 1891	–
Two simple cycles (shortcut model)	704 + 1112 = 1816	4
Two simple cycles (HYSYS)	727 + 1166 = 1893	−0.11

[1]Relative to complex cycle simulation in HYSYS.

Table 5. Predicted shaft work requirement for multi-level cycle – two heat sinks and a single heat source using HYSYS and new shortcut model[1].

optimizing the design conditions of a complex refrigeration cycle and/or the associated processing conditions, as will be seen in Section 3.

3. Heat-integrated process and refrigeration

For systems working at sub-ambient temperatures, the power demand can be very high; the lower the source temperature, the more complicated the refrigeration system design, the larger the amount of energy consumed. However, there is a great chance to reduce of the compression energy consumption if heat integration technology is applied, such that most appropriate refrigeration levels and their duties are determined to match them against grand composite curve (GCC), as shown in **Figure 5**. The GCC provides the overall source and sink temperature profiles of a process and allows the minimum hot and cold utility requirements to be identified. Also, another key feature of the GCC is that it considers the integration among the process, heat exchanger network, and refrigeration system simultaneously [8]. Therefore, in this work, the GCC has been used in the optimization approach presented in Section 3.1 to find the optimal operating conditions that minimize the compressor energy consumption.

3.1. Calculation procedure for the operational optimization of multi-level refrigeration cycles

This section describes the proposed method for the operational optimization of multi-level refrigeration cycles. In this methodology, it is assumed that the process is designed for maximum heat recovery, the refrigeration rejects the absorbed heat to external cooling utility, e.g., cooling water, the number of cooling levels is given, and the compression shaft work dominates the process economics; thus, minimization shaft power consumption in the refrigeration compressors is set as the objective for optimization.
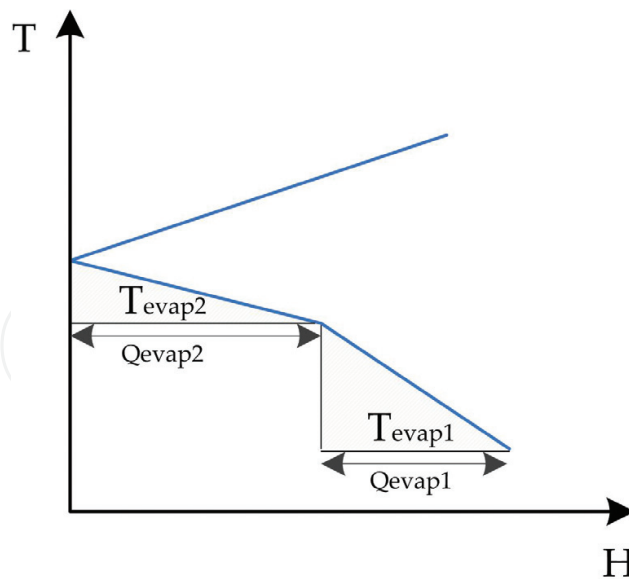
Figure 5. Matching two refrigeration levels against GCC.

The calculation procedure starts with constructing GCC, which is used to identify the total cooling duty Q_e and the lowest refrigeration temperature level. To determine the temperature level (independent variable) and the cooling duty of each level (dependent variables), the part of GCC below the pinch, which is the point where the lowest driving forces between hot and cold streams are located, is modeled as a set of linear functions,

$$H = f(T) \tag{7}$$

$$T = T_{\text{evap}} + \frac{\Delta T_{\text{min}}}{2} \tag{8}$$

$$Q_{\text{evap}_i} = H(T_{\text{evap}_i}) - H(T_{\text{evap}_{i+1}}) \tag{9}$$

$$Q_{\text{evap}_I} = Q_e - \sum_{i=1}^{I-1} Q_{\text{evap}_i} \tag{10}$$

The second step is to decompose the complex refrigeration cycle into simple cycles. This allows the shortcut model to be used for estimating the shaft work requirement of each level. Finally, a nonlinear model is applied to find the optimal cooling temperature levels and duties of each level. The objective function is to:

$$\min (W) = \sum_{i=1}^{I-1} \frac{Q_{\text{evap}_i}}{\text{COP}_i}$$

Subject to:

$$\begin{aligned}
&T_{\text{evap}_{i+1}} - T_{\text{evap}_i} \geq \Delta T_{\text{min}}, \ i = 1 : I - 1 \\
&T_{\text{cond}} - T_{\text{evap}_I} \geq \Delta T_{\text{min}} \\
&T_{\text{evap}}^{\text{lb}} \leq T_{\text{evap}_i} \leq T_{\text{evap}}^{\text{ub}}
\end{aligned} \tag{11}$$

Where W is the net power demand of the refrigeration cycle, Q_{evap_i} is the cooling duty of ith cooling level, COP_i is the coefficient of performance which is calculated from the developed shortcut model presented in Section 3, T is the shifted temperature, $\Delta T_{\min}$ is the minimum approach temperature, T_{evap_i} is the evaporation temperature of ith cooling level, T_{cond} is the condensing temperature at which the refrigerant being condensed, I is the number of cooling levels, and lb and ub represent the lower and upper bounds, respectively.

The main features of this calculation procedure are: (1) using GCC to determine the temperature level and the duty of each stage; (2) the shaft work required of each stage calculated directly without going through the detailed refrigeration calculations or rigorous simulation; and (3) the constrained optimization problem can be solved easily using a simple optimization algorithm, such that available in MATLAB and Excel (i.e., Excel's Solver). The limitations of this approach can be summarized as follows: (1) the advantage of using economizer in minimizing shaft work consumption cannot be explored because the effect of its use cannot be represented in GCC [7], (2) only pure refrigerants are considered, and (3) heat is rejected to external utility rather than process heat sink streams, so the opportunities of the matching refrigeration system with process sink streams—which can provide significant energy savings—are missing. The implementation of the proposed optimization approach for minimizing the overall shaft work requirement of a complex refrigeration cycle is illustrated in Section 4.

4. Case study 2: cold-end process of an ethylene plant

This case aims to illustrate how the new refrigeration model can be used to estimate the power demand in a multi-level cascade cycle. A second aim is to illustrate the performance of the proposed optimization model in minimizing the net power demand.

The complex refrigeration cycle in the ethylene-propylene cascade refrigeration cycle from the cold-end process of an ethylene plant is selected for analysis. Case study data are given in **Table 6**. **Figure 6(a)** shows the process flow diagram of the ethylene-propylene cascaded refrigeration cycle. **Figure 6**(b) shows the refrigeration cycle matched against the grand composite curve of the stream data presented in **Table 6**. **Table 7** gives the corresponding cascaded refrigeration cycle details, including two propylene cooling levels in the upper cycle and three ethylene cooling levels in the lower cycle.

4.1. Prediction power requirement in multi-level refrigeration cycle

For the cascaded cycle, the low temperature cycle and the high temperature cycle are treated as individual complex refrigeration cycles. The power demand in the three-level ethylene cycle is estimated by representing the three-level refrigeration cycle as a three parallel simple cycle. The power demand in the high temperature cycle is estimated by representing the two-level refrigeration cycle as a two parallel simple cycle. The refrigerant data in **Table 7** are used in the refrigeration model to predict the power demand at each compression stage.

	Supply temp. (°C)	Target temp. (°C)	Duty (kW)
H1	−14	−15	3780
H2	−22	−23	15,911
H3	27	−95	18,985
C1	23	78	7296
C2	7	8	3634
C3	−1	0	14,651
C4	−27	23	6373
C5	−27	23	185

Table 6. Process stream data for case study 2 [10].
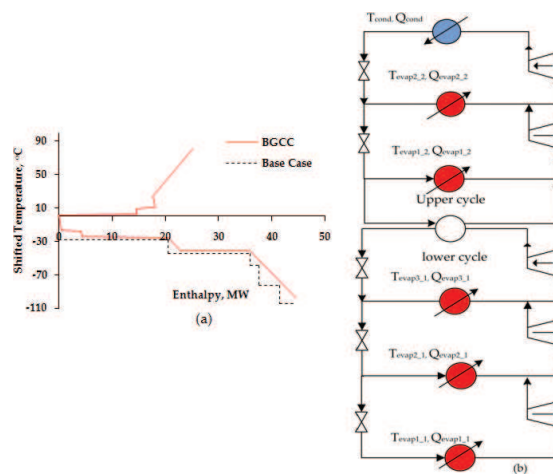
Figure 6. Balanced grand composite curve for case study 1 (a) and its refrigeration cycle (b).

	Temperature (°C)	Duty (kW)
Ethylene refrigeration level 1	−104.1	3113
Ethylene refrigeration level 2	−82.9	3890
Ethylene refrigeration level 3	−58.3	1713
Propylene refrigeration level 1	−44	15,395
Propylene refrigeration level 2	−28	20,482

Table 7. Refrigerant data for ethylene-propylene cascade refrigeration cycle.

For the purpose of comparison, the refrigeration cycle was also simulated in HYSYS software. The fluid package chosen in the simulator for determining thermodynamic properties was the Peng-Robinson equation of state. In this case, it is assumed that the compression efficiency is 80% and all the absorbed heat is rejected to cooling water at 23°C. Also, for the heat exchangers, a 5°C minimum temperature approach is specified, while it is assumed that there

is no pressure drop in both heat exchangers and piping [10]. The partition temperature, which is the temperature of the evaporator of the upper cycle, was set at $-44°$C.

Table 8 presents the results of the predicted power demand for complex cycle using HYSYS and the new refrigeration model. The error between the refrigeration model and HYSYS predictions for net power demand prediction is about 10% in the lower cycle and -2% in the upper cycle. As can be seen from the HYSYS simulation results for the three parallel cycles, this error arises mainly from the mixing effect at the inlet to the compressor.

4.2. Optimization operating conditions in multi-level refrigeration cycle

This section only explores the optimization of the evaporating temperatures in the lower refrigeration cycle presented in **Figure 6(b)**. This is because the temperature differences between the heat source temperature profile of the process (i.e., the GCC) and the evaporation temperature of refrigeration levels in the upper cycle are small, as shown in **Figure 6(a)**. This small temperature difference leads to slight improvement on the energy efficiency of the upper cycle [10].

In this work, the deterministic method (generalized reduced gradient (GRG)) will be used to search for the optimal solution of the objective function. In order to test the prediction accuracy of the proposed method, the results of the proposed method will also be compared with the published results of Oh et al. [10], in which genetic algorithm (GA) was employed to search for the optimal operating conditions of multi-level refrigeration cycle.

The decision variables manipulated for the optimization and the upper and lower bounds are listed in **Table 9**.

The optimal results for the three-level refrigeration cycle using ethylene as a refrigerant are summarized in **Table 10**. The results show that a significant decrease in the overall compression duty of the refrigeration cycle (9%) over the base case is obtained due to the reduction of the temperature lift (the temperature difference between the evaporator and condenser).

Low temperature cycle		
Modeling approach	**Power demand (kW)**	**Error**
Complex cycle (HYSYS)	780 + 1794 + 1615 = 4189	–
Three simple cycles (shortcut model)	2794 + 1620 + 230 = 4644	−10.8%
Three simple cycles (HYSYS)	2393 + 1541 + 228 = 4161	0.67%
High temperature cycle		
Complex cycle (HYSYS)	4218 + 21,810 = 26,028	–
Two simple cycles (shortcut model)	17,863 + 8669 = 26,532	−2%
Two simple cycles (HYSYS)	17,362 + 8656 = 26,018	0.04%

Table 8. Predicted power demand for cascade cycle using HYSYS and new model.

Decision variable	Lower bound	Upper bound
T_{evap_2}	−91.45	−66.13
T_{evap_3}	−76.32	−46.19

Table 9. Optimization variables and their constraints for case 2.

Low temperature cycle		Base case	Optimized case	
			Oh et al. [10] (Complex cycle)	This work (Three simple cycle)
Level 1	Pressure (kPa)	101.3	127.7	127.7
	Temperature (°C)	−104	−100	−100
	Cooling duty (kW)	3113		2221
	Shaft power (kW)	780		1717
Level 2	Pressure (kPa)	300	288.4	264.3
	Temperature (°C)	−82.9	−84.89	−85.67
	Cooling duty (kW)	3890		2847
	Shaft power (kW)	1794		1333
Level 3	Pressure (kPa)	800	649.1	574.8
	Temperature (°C)	−58.3	−66.62	−67.31
	Cooling duty (kW)	1713		3648
	Shaft power (kW)	1615		829
Net power demand (kW)		4189	3560	3828
Saving relative to base case (%)		—	15	9

Table 10. Comparison between the optimization results of the proposed model and the published results [10].

It is clear from the results that GRG has good performance (in terms of getting closer to the optimal solution) and less computationally expensive compared to GA. The results in **Table 10** show that insignificant difference between the results from the proposed model and those obtained from the published results of Oh et al. [10], where the deviation between the results is only 1% both for the evaporation temperature of cooling level 2 and for the evaporation temperature of cooling level 3. The differences between the results of the proposed model and those of Oh et al. [10] are mainly due to using the decomposition approach, which is used to predict the coefficient performance of the refrigeration cycle.

5. Conclusions

This chapter introduces a new refrigeration model and proposes a systematic methodology for operational optimization of multi-level refrigeration cycle. The methodology applies NLP model to minimize the overall power demand of the refrigeration cycle and uses GCC and

linear refrigeration model that is based on regression from rigorous simulations. The GCC is used to obtain cooling duty of each level, while the linear refrigeration model is used to predict the actual coefficient performance of the complex refrigeration cycle that is decomposed into assembly simple cycles. The refrigeration model requires only condensing and evaporating temperatures. The effectiveness of the proposed optimization approach has been demonstrated on the case study of its application to ethylene cold-end process. The results of the case study demonstrate that the refrigeration model can predict the power demand within 10% of rigorous simulation. The optimization algorithm can find a close optimum solution within very short time (less than a second). Also, the results reveal that 9% saving in shaft power demand can be achieved by optimizing the operating conditions. The difference between the optimal operating conditions (i.e., the evaporation temperatures) found by GA and GRG is 1%. Although these findings support the validity of the refrigeration model, and the reliability and computational efficiency of the optimization approach in finding a close optimal solution, there are some factors that need to be considered in the future. These factors include the trade-offs between capital and operating costs, the opportunities for rejecting heat to a cold heat sink within the process rather than an external cooling utility, and the use of mixed refrigerant (the advantage of using mixed refrigerant can be explored by using a refrigeration database that includes the power demand at various operating conditions).

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