Exergy Analysis of Cryogenic and Liquefaction Systems

Chapter Outline

15.1 Introduction 319
15.2 Energy and Exergy Analyses of Gas Liquefaction Systems 320
15.3 Exergy Analysis of a Multistage Cascade Refrigeration Cycle for Natural Gas Liquefaction 322
   15.3.1 Background 323
   15.3.2 Description of the Cycle 323
   15.3.3 Exergy Analysis 323
      15.3.3.1 Evaporators and Condensers 324
      15.3.3.2 Overall Cycle 326
   15.3.4 Minimum Work for the Liquefaction Process 326
   15.3.5 Discussion 328
15.4 Exergy Analysis of an Integrated Hydrogen Liquefaction Using Geothermal Energy 328
   15.4.1 System Description 329
   15.4.2 Analysis 329
      15.4.2.1 TEACS 329
      15.4.2.2 Binary Power Plant 330
      15.4.2.3 LH Liquefaction Cycle 331
      15.4.2.4 Utilization Factors 331
   15.4.3 Performance 331
      15.4.3.1 Performance Variations with Geothermal Mass Flow Rate 332
      15.4.3.2 Performance Variations with Geothermal Source Temperature 333
   15.4.4 Further Discussion and Findings 333
15.5 Closing Remarks 334

ABSTRACT
In this chapter, the application of energy and exergy analyses to cryogenic and liquefaction systems is described. Cryogenic engineering is the design, development, and improvement of low-temperature (usually below −150°C or 123 K) systems and components. Applications of cryogenic engineering include liquefaction and separation of gases, high-field magnets and sophisticated electronic devices that use the superconductivity of materials at low temperatures, space simulation, food freezing, medical uses such as cryogenic surgery, and various chemical processes.

KEYWORDS
Exergy; Cryogenics; Liquefaction; Gas liquefaction; Cascade refrigeration; Natural gas liquefaction; Hydrogen liquefaction; Efficiency.

15.1 INTRODUCTION
Cryogenics is associated with low temperatures, usually below −150°C (123 K). The general scope of cryogenic engineering is the design, development, and improvement of low-temperature systems and components. Applications of cryogenic engineering include liquefaction and separation of gases, high-field magnets, and sophisticated electronic devices that use the superconductivity of materials at low temperatures, space simulation, food freezing, medical uses such as cryogenic surgery, and various chemical processes.

Progress in the production of cryogenic temperatures is based on advances in thermodynamics. A solid understanding of the thermal processes within a refrigeration cycle is not possible without a sound knowledge of the laws of thermodynamics and cyclic processes. During the early years of classical thermodynamics, between 1842 and 1852, Julius Robert Mayer, James Prescott Joule, Rudolf Clausius, William Thomson, and Hermann von Helmholtz published important findings concerning the First and Second Laws of Thermodynamics. Before that time, the main approaches involved trial and error and many experiments. Even after, necessary data were often missing and only became available much later (Foerg, 2002).

The liquefaction of gases has always been an important area of refrigeration since many important scientific and engineering processes at cryogenic temperatures depend on...
liquefied gases. At temperatures above the critical point, a substance exists in the gas phase only. The critical temperatures of helium, hydrogen, and nitrogen are $-268^\circ C$, $-240^\circ C$ and $-147^\circ C$, respectively. Therefore, none of these substances exist in liquid form at atmospheric conditions. Furthermore, low temperatures of this magnitude cannot be obtained by ordinary refrigeration techniques.

To avoid heat leaks into cryogenic storage tanks and transfer lines, high-performance materials are needed that provide high levels of thermal insulation. A good understanding of thermal insulation is important for the development of efficient and low-maintenance cryogenic systems.

In today’s world, cryogenics and low-temperature refrigeration are taking on increasingly significant roles. From applications in the food industry, energy, and medical technologies to transportation and the space shuttle, requirements exist for cryogenic liquids to be stored and transferred.

In this chapter, a comprehensive exergy analysis of a multistage cascade refrigeration cycle used for natural gas liquefaction as a cryogenic system, based on a report by Kanoglu (2002b), is presented. The multistage cascade cryogenic system is described and an exergy analysis of the cycle components and the minimum work required for liquefaction are provided.

### 15.2 ENERGY AND EXERGY ANALYSES OF GAS LIQUEFACTION SYSTEMS

Several cycles, some complex and others simple, exist for the liquefaction of gases. Here, we consider the Linde-Hampson (LH) cycle, shown schematically and on a $T$-$s$ diagram in Figure 15.1, to illustrate energy and exergy analyses of liquefaction cycles. Makeup gas is mixed with the uncondensed portion of the gas from the previous cycle, and the mixture at state 1 is compressed by an isothermal compressor to state 2. The temperature is kept constant by rejecting compression heat to a coolant. The high-pressure gas is further cooled in a regenerative counterflow heat exchanger by the uncondensed portion of gas from the previous cycle to state 3, and throttled to state 4, which is a saturated liquid—vapor mixture state. The liquid (state 6) is collected as the desired product, and the vapor (state 5) is routed through the heat exchanger to cool the high-pressure gas approaching the throttling valve. Finally, the gas is mixed with fresh makeup gas, and the cycle is repeated.

The refrigeration effect for this cycle may be defined as the heat removed from the makeup gas in order to turn into a liquid at state 6. Assuming ideal operation for the heat exchanger (i.e., the gas leaving the heat exchanger and the makeup gas are at the same state as state 1, which is the compressor inlet state), the refrigeration effect per unit mass of the liquefied gas is given by:

$$q_L = h_1 - h_6 = h_1 - h_f \text{ (per unit mass of liquefaction)}$$

(15.1)

where $h_f$ is the specific enthalpy of saturated liquid that is withdrawn. From an energy balance on the cycle, the refrigeration effect per unit mass of the gas in the cycle may be expressed as:

$$q_L = h_1 - h_2 \text{ (per unit mass of gas in the cycle)}$$

(15.2)

![Figure 15.1](image-url)
The maximum liquefaction occurs when the difference between \( h_4 \) and \( h_5 \) (i.e., the refrigeration effect) is maximized. The ratio of Equations 15.2 and 15.1 is the fraction of the gas in the cycle that is liquefied, \( y \). That is,

\[
y = \frac{h_1 - h_2}{h_1 - h_f} \quad (15.3)
\]

An energy balance on the heat exchanger gives:

\[
h_2 - h_3 = x(h_1 - h_5)
\]

where \( x \) is the quality of the mixture at state 4. Then the fraction of the gas that is liquefied may also be determined from:

\[
y = 1 - x \quad (15.4)
\]

An energy balance on the compressor gives the work of compression per unit mass of the gas in the cycle as:

\[
w_{in} = h_2 - h_1 - T_1(s_2 - s_1) \quad (15.5)
\]

The last term in this equation is the isothermal heat rejection from the gas as it is compressed. Assuming air behaves as an ideal gas during this isothermal compression process, the compression work may also be determined from:

\[
w_{in} = RT_1 \ln(P_2/P_1) \quad (15.6)
\]

The coefficient of performance (COP) of this ideal cycle is then given by:

\[
\text{COP} = \frac{q_L}{w_{in}} = \frac{h_1 - h_2}{h_2 - h_1 - T_1(s_2 - s_1)} \quad (15.7)
\]

In liquefaction cycles, an efficiency parameter used is the work consumed in the cycle for the liquefaction of a unit mass of the gas. This is expressed as:

\[
w_{in \, per \, mass \, liquefied} = \frac{h_2 - h_1 - T_1(s_2 - s_1)}{y} \quad (15.8)
\]

As the liquefaction temperature decreases, the work consumption increases. Noting that different gases have different thermophysical properties and require different liquefaction temperatures, this work parameter should not be used to compare the work consumptions of the liquefaction of different gases. A reasonable use is to compare the different cycles used for the liquefaction of the same gas.

Engineers are usually interested in comparing the actual work used to obtain a unit mass of liquefied gas and the minimum work requirement to obtain the same output. Such a comparison may be performed using the Second Law. For instance, the minimum work input requirement (reversible work) and the actual work for a given set of processes may be related to each other by:

\[
w_{actual} = w_{rev} + T_0 s_{gen} = w_{rev} + e_{x, dest} \quad (15.9)
\]

where \( T_0 \) is the environment temperature, \( s_{gen} \) is the total entropy generation, and \( e_{x, dest} \) is the total exergy destruction during the processes. The reversible work for the simple LH cycle shown in Figure 15.1 may be expressed by the stream exergy difference of states 1 and 6 as:

\[
w_{rev} = ex_6 - ex_1 = h_6 - h_1 - T_0(s_6 - s_1) \quad (15.10)
\]

where state 1 has the properties of the makeup gas, which is essentially the dead state. This expression gives the minimum work requirement for the complete liquefaction of a unit mass of the gas. Exergy efficiency may be defined as the reversible work input divided by the actual work input, both per unit mass of the liquefaction:

\[
\psi = \frac{w_{rev}}{w_{actual}} = \frac{h_6 - h_1 - T_0(s_6 - s_1)}{(1/y)[h_2 - h_1 - T_1(s_2 - s_1)]} \quad (15.11)
\]

We present a numerical example for the simple LH cycle shown in Figure 15.1. It is assumed that the compressor is reversible and isothermal; the heat exchanger has an effectiveness of 100% (i.e., the gas leaving the liquid reservoir is heated in the heat exchanger to the temperature of the gas leaving the compressor); the expansion valve is isenthalpic; and there is no heat leak to the cycle. The gas is air, at 25°C and 1 atm at the compressor inlet and at 20 MPa at the compressor outlet. With these assumptions and specifications, the various properties at the different states of the cycle and the performance parameters discussed earlier are determined to be the following:

<table>
<thead>
<tr>
<th>Properties</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>( h_1 )</td>
<td>298.4 kJ/kg</td>
</tr>
<tr>
<td>( s_1 )</td>
<td>6.86 kJ/kg K</td>
</tr>
<tr>
<td>( q_L )</td>
<td>34.9 kJ/kg gas</td>
</tr>
<tr>
<td>( h_2 )</td>
<td>263.5 kJ/kg</td>
</tr>
<tr>
<td>( s_2 )</td>
<td>5.23 kJ/kg K</td>
</tr>
<tr>
<td>( w_{in} )</td>
<td>451 kJ/kg gas</td>
</tr>
<tr>
<td>( h_3 )</td>
<td>61.9 kJ/kg</td>
</tr>
<tr>
<td>( s_f )</td>
<td>2.98 kJ/kg K</td>
</tr>
<tr>
<td>( \text{COP} )</td>
<td>0.0775</td>
</tr>
<tr>
<td>( h_4 )</td>
<td>61.9 kJ/kg</td>
</tr>
<tr>
<td>( T_4 )</td>
<td>-194.2°C</td>
</tr>
<tr>
<td>( w_{in , liquid} )</td>
<td>5481 kJ/kg liquid</td>
</tr>
<tr>
<td>( h_5 )</td>
<td>78.8 kJ/kg</td>
</tr>
<tr>
<td>( x_4 )</td>
<td>0.9177</td>
</tr>
<tr>
<td>( w_{rev , liquid} )</td>
<td>733 kJ/kg liquid</td>
</tr>
<tr>
<td>( h_6 )</td>
<td>-126.1 kJ/kg</td>
</tr>
<tr>
<td>( y )</td>
<td>0.0823</td>
</tr>
<tr>
<td>( \psi )</td>
<td>0.134</td>
</tr>
</tbody>
</table>

The properties of air and other substances are obtained using EES software (Klein, 2006). This analysis is repeated for different fluids and the results are listed in Table 15.1.

We observe in Table 15.1 that, as the boiling temperature decreases, the fraction of the gas that is liquefied, the COP, and the exergy efficiency decrease. The exergy efficiency values are low, indicating significant potential for improving performance, and thus decreasing the need for work consumption. Noting that the cycle considered in this numerical example involves a reversible isothermal compressor and a 100% effective heat exchanger, the exergy efficiency figures here are optimistic. In practice, an isothermal compression process may be approached using a multistage compressor. For a high effectiveness, the size of heat exchanger must be large, meaning a higher cost. The work consumption may be decreased by replacing the expansion valve with a turbine. Expansion in
a turbine usually results in a lower temperature with respect to an expansion valve while producing work, and thus decreasing the total work consumption in the cycle. The complexity and added cost associated with using a turbine as an expansion device is only justified in large liquefaction systems (Kanoglu, 2001, 2002b). In some systems, both a turbine and an expansion valve are used to avoid problems associated with liquid formation in the turbine.

The effect of liquefaction temperature on the liquefied mass fraction and COP is illustrated in Figure 15.2, while Figure 15.3 shows the effect of liquefaction temperature on the exergy efficiency for various gases. These figures are obtained using the cycle in Figure 15.1. As the liquefaction temperature increases the liquefied mass fraction, the COP and the exergy efficiency increase.

### 15.3 EXERGY ANALYSIS OF A MULTISTAGE CASCADE REFRIGERATION CYCLE FOR NATURAL GAS LIQUEFACTION

An exergy analysis is presented of a multistage cascade refrigeration cycle used for natural gas liquefaction, based...
on a report by Kanoglu (2002b). Multistage cascade refrigeration is described and an exergy analysis of the cycle components and the minimum work required for liquefaction are provided.

### 15.3.1 Background

Natural gas is a mixture of methane (60%–98%), small amounts of other hydrocarbons, and various amounts of nitrogen, carbon dioxide, helium, and other trace gases. Natural gas can be stored as compressed natural gas at pressures of 16–25 MPa and around room temperature, or as liquefied natural gas (LNG) at pressures of 70–500 kPa and temperatures of around −160°C or lower. Natural gas can be transported as a gas in pipelines or as a liquid. In the latter case, the gas is liquefied using unconventional refrigeration cycles and then transported, often by marine ships in specially made insulated tanks. It is returned to a gaseous state in receiving stations for end use.

Several refrigeration cycles, working on different refrigerants, can be used for natural gas liquefaction including the mixed-refrigerant cycle, the cascade cycle, and the gas expansion cycle. The first cycle used for natural gas liquefaction was a multistage cascade refrigeration cycle using three refrigerants (propane, ethane, or ethylene and methane) in the individual refrigeration cycles that make up the overall cycle.

Much work is consumed to produce LNG at about −160°C from natural gas at atmospheric temperature in the gas phase (Finn et al., 1999). Reducing the work consumed in the cycle can help reduce the cost of LNG. Exergy analysis can help design, optimize, and assess such systems, identifying the locations of exergy destruction, thereby highlighting directions for potential improvement. For such work-consuming processes, exergy analysis helps determine the minimum work required for a desired result.

### 15.3.2 Description of the Cycle

Figure 15.4 shows a schematic of the cascade refrigeration cycle and its components. The cycle consists of three subcycles, each using a different refrigerant. In the first cycle, propane leaves the compressor at a high temperature and pressure and enters the condenser, where cooling water or air is used as a coolant. The condensed propane enters an expansion valve, where its pressure is decreased to the evaporator pressure. As the propane evaporates, the heat of evaporation comes from the condensing ethane, cooling methane, and cooling natural gas. Propane leaves the evaporator and enters the compressor, thus completing the cycle.

The condensed ethane expands in an expansion valve and evaporates as methane condenses and natural gas is further cooled and liquefied. Finally, the methane expands and evaporates as natural gas is liquefied and subcooled. As methane enters the compressor to complete the cycle, the pressure of the LNG is dropped in an expansion valve to the storage pressure. The three refrigerant cycles have multistage compression and expansion, usually with three stages, and, consequently, three evaporation temperature levels for each refrigerant. The mass flows in each stage are usually different.

Natural gas from the pipeline undergoes a process in which acid gases are removed and pressure is increased to an average value of 40 bar before entering the cycle.

### 15.3.3 Exergy Analysis

The exergy flow rate of a fluid in a control volume can be written, neglecting kinetic and potential energies, as follows:

\[
\dot{E_x} = \dot{n}_{ex} = \dot{n}[h - h_0 - T_0(s - s_0)]
\]  

(15.12)
where $T_0$ is the reference (dead) state temperature, $h$ and $s$ are the specific enthalpy and entropy of the fluid at the specified state, and $h_0$ and $s_0$ are the corresponding properties at the dead state. Multiplying the specific flow exergy $ex$ by the mass flow rate gives the exergy flow rate.

The specific exergy change between two states (e.g., inlet and outlet) can be expressed as:

$$
\Delta ex = h_2 - h_1 - T_0(s_2 - s_1)
$$

Some exergy is lost during the process due to entropy generation. The specific irreversibility can be written as

$$
i = T_0\Delta s = T_0s_{gen}
$$

where $s_{gen}$ is the entropy generation.

Two main causes of entropy generation in the process are friction and heat transfer across a finite temperature difference. Heat transfer is generally accompanied by exergy transfer, which is given by:

$$
ex = \int \delta q \left( 1 - \frac{T}{T_0} \right)
$$

where $\delta q$ is differential heat transfer and $T$ is the source temperature where heat transfer takes place. Heat transfer is assumed to occur in surroundings at $T_0$. If the transferred heat is lost; Equation 15.14 expresses a thermal exergy loss.

The exergy destruction and exergetic efficiency expressions are presented below for the cycle components in Figure 15.4.

**15.3.3.1 Evaporators and Condensers**

The evaporators and condensers in the system are treated as heat exchangers. There are four evaporator-condenser systems in the cycle. The first, evaporator-condenser I, is the evaporator of the propane cycle and the condenser of the ethane and methane cycles. Similarly, evaporator-condenser II is the evaporator of ethane cycle and the condenser of methane cycle. The third system is the evaporator of the methane cycle and the fourth system is the condenser of the propane cycle, where cooling water is used as a coolant. An exergy balance for an evaporator-condenser expresses the exergy destroyed in the system as the difference of the exergies of incoming and outgoing streams. For evaporator-condenser I,

$$
\dot{i} = \dot{E}_{in} - \dot{E}_{out} = \left[ \sum (\dot{m}_p e_{xp}) + \sum (\dot{m}_e e_{xe}) \right]_{in} + \sum (\dot{m}_m e_{xm})_{in} + \sum (\dot{m}_e e_{xe})_{out} \left[ \sum (\dot{m}_m e_{xm})_{out} + (\dot{m}_n e_{xn})_{out} \right]
$$

where the subscripts in, out, $p$, $e$, $m$, and $n$ denote inlet, outlet, propane, ethane, methane, and natural gas, respectively. The summations account for the fact that there are three stages in each refrigerant cycle with different pressures, evaporation temperatures, and mass flow rates.

The exergetic efficiency of a heat exchanger can be defined as the ratio of total outgoing stream exergies to total incoming stream exergies. For evaporator-condenser I,
An alternate definition for the exergy efficiency of a heat exchanger is the ratio of the increase in the exergy of the cold fluid to the decrease in the exergy of the hot fluid (Wark, 1995). For evaporator-condenser I, the only fluid with an exergy increase is propane, while the exergies of ethane, methane, and natural gas decrease. Therefore, the alternate efficiency can be written as:

\[
\varepsilon = \frac{\sum (\dot{m}_p e_{x_p})_{\text{out}} - \sum (\dot{m}_p e_{x_p})_{\text{in}}}{\sum (\dot{m}_e e_x)_{\text{in}} - \sum (\dot{m}_e e_x)_{\text{out}} + \sum (\dot{m}_m e_{x_m})_{\text{in}} - \sum (\dot{m}_m e_{x_m})_{\text{out}}}
\]  

(15.17)

The values of efficiencies calculated using these two approaches are usually similar. In this analysis, the second approach is used.

The exergy destruction rate and exergy efficiency for evaporator-condenser II are expressible as:

\[
I = \dot{E}_{\text{in}} - \dot{E}_{\text{out}}
\]

\[
I = \left[ \sum (\dot{m}_e e_x)_{\text{in}} + \sum (\dot{m}_m e_{x_m})_{\text{in}} + (\dot{m}_n e_{x_n})_{\text{in}} \right] - \left[ \sum (\dot{m}_e e_x)_{\text{out}} + \sum (\dot{m}_m e_{x_m})_{\text{out}} + (\dot{m}_n e_{x_n})_{\text{out}} \right]
\]

(15.18)

\[
\psi = \frac{\sum (\dot{m}_e e_x)_{\text{out}} - \sum (\dot{m}_e e_x)_{\text{in}}}{\sum (\dot{m}_m e_{x_m})_{\text{in}} - \sum (\dot{m}_m e_{x_m})_{\text{out}} + (\dot{m}_n e_{x_n})_{\text{in}} - (\dot{m}_n e_{x_n})_{\text{out}}}
\]

(15.19)

The exergy destruction rate and exergy efficiency can be written for the evaporator of the methane cycle as:

\[
I = \dot{E}_{\text{in}} - \dot{E}_{\text{out}}
\]

\[
I = \left[ \sum (\dot{m}_m e_{x_m})_{\text{in}} + (\dot{m}_n e_{x_n})_{\text{in}} \right] - \left[ \sum (\dot{m}_m e_{x_m})_{\text{out}} + (\dot{m}_n e_{x_n})_{\text{out}} \right]
\]

(15.20)

\[
\psi = \frac{\sum (\dot{m}_m e_{x_m})_{\text{out}} - \sum (\dot{m}_m e_{x_m})_{\text{in}}}{(\dot{m}_n e_{x_n})_{\text{in}} - (\dot{m}_n e_{x_n})_{\text{out}}}
\]

(15.21)

and for the condenser of propane cycle as:

\[
I = \dot{E}_{\text{in}} - \dot{E}_{\text{out}}
\]

\[
I = \left[ \sum (\dot{m}_p e_{x_p})_{\text{in}} + (\dot{m}_w e_{x_w})_{\text{in}} \right] - \left[ \sum (\dot{m}_p e_{x_p})_{\text{out}} + (\dot{m}_w e_{x_w})_{\text{out}} \right]
\]

(15.22)

\[
\psi = \frac{(\dot{m}_w e_{x_w})_{\text{out}} - (\dot{m}_w e_{x_w})_{\text{in}}}{\sum (\dot{m}_p e_{x_p})_{\text{in}} - \sum (\dot{m}_p e_{x_p})_{\text{out}}}
\]

(15.23)

where the subscript w denotes water.

15.3.3.1 Compressors

There is one multistage compressor in the cycle for each refrigerant. The total work consumed in the cycle is the sum of work inputs to the compressors. The minimum work input for the compressor occurs when no irreversibilities occur and exergy destruction is correspondingly zero. In reality, irreversibilities occur due to friction, heat loss, and other dissipative effects. The exergy destructions in the propane, ethane, and methane compressors, respectively, can be expressed as:

\[
I_p = \dot{E}_{\text{in}} - \dot{E}_{\text{out}}
\]

\[
I_p = \sum (\dot{m}_p e_{x_p})_{\text{in}} + W_{p, \text{in}} - \sum (\dot{m}_p e_{x_p})_{\text{out}}
\]

(15.24a)

\[
I_e = \dot{E}_{\text{in}} - \dot{E}_{\text{out}}
\]

\[
I_e = \sum (\dot{m}_e e_x)_{\text{in}} + W_{e, \text{in}} - \sum (\dot{m}_e e_x)_{\text{out}}
\]

(15.24b)

\[
I_m = \dot{E}_{\text{in}} - \dot{E}_{\text{out}}
\]

\[
I_m = \sum (\dot{m}_m e_{x_m})_{\text{in}} + W_{m, \text{in}} - \sum (\dot{m}_m e_{x_m})_{\text{out}}
\]

(15.24c)

where \( W_{p, \text{in}} \), \( W_{e, \text{in}} \), and \( W_{m, \text{in}} \) are the actual power inputs to the propane, ethane, and methane compressors, respectively. The exergy efficiency of a compressor can be defined as the ratio of the minimum work input to the actual work input. The minimum work is simply the exergy difference between the actual inlet and exit states. Applying this definition to the propane, ethane, and methane compressors, the respective exergy efficiencies become:

\[
\psi_p = \frac{\sum (\dot{m}_p e_{x_p})_{\text{out}} - \sum (\dot{m}_p e_{x_p})_{\text{in}}}{W_{p, \text{in}}}
\]

(15.25a)

\[
\psi_e = \frac{\sum (\dot{m}_e e_x)_{\text{out}} - \sum (\dot{m}_e e_x)_{\text{in}}}{W_{e, \text{in}}}
\]

(15.25b)

\[
\psi_m = \frac{\sum (\dot{m}_m e_{x_m})_{\text{out}} - \sum (\dot{m}_m e_{x_m})_{\text{in}}}{W_{m, \text{in}}}
\]

(15.25c)

15.3.3.1.2 Expansion Valves

In addition to the expansion valves in the refrigerant cycles, an expansion valve is used to reduce the pressure of LNG to the storage pressure. Expansion valves are essentially isenthalpic devices with no work interaction and negligible heat interaction with the surroundings. Exergy balances can
be used to write the exergy destruction rates for the propane, ethane, methane, and LNG expansion valves:

\[ I_p = \dot{E}_{x_{in}} - \dot{E}_{x_{out}} = \sum (\dot{m}_p e_p)_{in} - \sum (\dot{m}_p e_p)_{out} \] (15.26a)

\[ I_e = \dot{E}_{x_{in}} - \dot{E}_{x_{out}} = \sum (\dot{m}_e e_e)_{in} - \sum (\dot{m}_e e_e)_{out} \] (15.26b)

\[ I_m = \dot{E}_{x_{in}} - \dot{E}_{x_{out}} = \sum (\dot{m}_m e_m)_{in} - \sum (\dot{m}_m e_m)_{out} \] (15.26c)

\[ I_n = \dot{E}_{x_{in}} - \dot{E}_{x_{out}} = \sum (\dot{m}_n e_n)_{in} - \sum (\dot{m}_n e_n)_{out} \] (15.26d)

The exergy efficiency of an expansion valve can be defined as the ratio of the total exergy output to the total exergy input. The exergy efficiencies for the expansion valves considered here are thus:

\[ \psi_p = \frac{\sum (\dot{m}_p e_p)_{out}}{\sum (\dot{m}_p e_p)_{in}} \] (15.27a)

\[ \psi_e = \frac{\sum (\dot{m}_e e_e)_{out}}{\sum (\dot{m}_e e_e)_{in}} \] (15.27b)

\[ \psi_m = \frac{\sum (\dot{m}_m e_m)_{out}}{\sum (\dot{m}_m e_m)_{in}} \] (15.27c)

\[ \psi_n = \frac{\sum (\dot{m}_n e_n)_{out}}{\sum (\dot{m}_n e_n)_{in}} \] (15.27d)

### 15.3.3.2 Overall Cycle

The exergy destruction in the overall cycle is the sum of exergy destruction in all internal devices (i.e., condensers, evaporators, compressors, and expansion valves). This cycle exergy destruction can be obtained using the preceding exergy destruction expressions. The exergy efficiency of the overall cycle can be defined as:

\[ \psi = \frac{\dot{E}_{x_{out}} - \dot{E}_{x_{in}}}{W_{actual}} = \frac{\dot{W}_{actual} - I_{total}}{W_{actual}} \] (15.28)

where the numerator expresses the exergy difference (or the actual work input to the cycle \( W_{actual} \) less the total exergy destruction \( I \)). The actual work input to the overall cycle is the sum of the work inputs to the propane, ethane, and methane compressors as follows:

\[ W_{actual} = W_{p, in} + W_{e, in} + W_{m, in} \] (15.29)

The exergy efficiency of the cycle can also be expressed as:

\[ \psi = \frac{W_{min}}{W_{actual}} \] (15.30)

where \( W_{min} \) is the minimum work input to the cycle, which represents the minimum work for the liquefaction process.

### 15.3.4 Minimum Work for the Liquefaction Process

The exergy efficiency of the natural gas liquefaction process can be defined as the ratio of the minimum work required to produce a certain amount of LNG to the actual work input. An exergy analysis on the cycle permits the minimum work required to be determined. The liquefaction process essentially involves the removal of heat from the natural gas. Therefore, the minimum work for the liquefaction process can be determined utilizing a reversible or Carnot refrigerator. The minimum work input is simply the work input required for the Carnot refrigerator for a given heat removal, and can be expressed as:

\[ w_{min} = \int \delta q \left( 1 - \frac{T_0}{T} \right) \] (15.31)

where \( \delta q \) is the differential heat transfer and \( T \) is the instantaneous temperature at the boundary where the heat transfer takes place. Note that \( T \) is smaller than \( T_0 \) for a liquefaction process and to yield a positive work input, the sign of heat transfer must be negative since it is a heat output. The evaluation of Equation 15.31 requires a knowledge of the functional relationship between the heat transfer \( \delta q \) and the boundary temperature \( T \), which is usually not available.

As seen in Figure 15.4, natural gas flows through three evaporator-condenser systems in the multistage refrigeration cycle before it is fully liquefied. Thermodynamically, this three-stage heat removal from natural gas can be accomplished using three Carnot refrigerators, as seen in Figure 15.5a. The first Carnot refrigerator receives heat from the natural gas and supplies it to the heat sink at \( T_0 \) as the natural gas is cooled from \( T_1 \) to \( T_2 \). Similarly, the second Carnot refrigerator receives heat from the natural gas and supplies it to the heat sink at \( T_0 \) as the natural gas is cooled from \( T_2 \) to \( T_3 \). Finally, the third Carnot refrigerator receives heat from the natural gas and supplies it to the heat sink at \( T_0 \) as the natural gas is further cooled from \( T_3 \) to \( T_4 \), where it exits as LNG. The work rates required by the first, second, and third Carnot refrigerators respectively can be determined as:

\[ W_1 = \dot{m}_n (e_{x_1} - e_{x_2}) = \dot{m}_n [h_1 - h_2 - T_0(s_1 - s_2)] \] (15.32a)

\[ W_2 = \dot{m}_n (e_{x_2} - e_{x_3}) = \dot{m}_n [h_2 - h_3 - T_0(s_2 - s_3)] \] (15.32b)
The total power input, which expresses the minimum power input for the liquefaction process, is the sum of the above terms:

\[ W_{\text{min}} = W_1 + W_2 + W_3 = m_n (e_3 - e_4) = m_n [h_3 - h_4 - T_0 (s_3 - s_4)] \]  

(15.32c)

The total power input, which expresses the minimum power input for the liquefaction process, is the sum of the above terms:

\[ \dot{W}_{\text{min}} = \dot{W}_1 + \dot{W}_2 + \dot{W}_3 = \dot{m}_n [e_1 - e_4] = \dot{m}_n [h_1 - h_4 - T_0 (s_1 - s_4)] \]  

(15.33)

This minimum power can alternatively be obtained using a single Carnot refrigerator that receives heat from the natural gas and supplies it to a heat sink at \( T_0 \) as the natural gas is cooled from \( T_1 \) to \( T_4 \). Such a Carnot refrigerator is equivalent to the combination of three Carnot refrigerators shown in Figure 15.5b. The minimum work required for the liquefaction process depends only on the properties of the incoming and outgoing natural gas and the ambient temperature \( T_0 \).

A numerical value of the minimum work can be calculated using typical values of incoming and outgoing natural gas properties. When entering the cycle, the pressure of the natural gas is around 40 bar and the temperature is approximately the same as the ambient temperature, that is, \( T_1 = T_0 = 25^\circ \text{C} \). Natural gas leaves the cycle liquefied at about 4 bar and 150\(^\circ\)C. Since the natural gas in the cycle usually consists of more than 95% methane, it is assumed for simplicity that the thermodynamic properties of natural gas are the same as those for methane. Using these inlet and exit states, the minimum work input to produce a unit mass of LNG can be determined from Equation 15.33 as 456.8 kJ/kg. The heat removed from the natural gas during the liquefaction process is determined from:

\[ \dot{Q} = \dot{m}_n (h_1 - h_4) \]  

(15.34)

For the inlet and exit states of natural gas described above, the heat removed from the natural gas can be determined from Equation 15.34 to be 823.0 kJ/kg. That is, the removal of 823.0 kJ/kg of heat from the natural gas requires a minimum of 456.8 kJ/kg of work. Since the ratio of heat removed to work input is defined as the COP of a refrigerator, this corresponds to a COP of 1.8. This relatively low COP of the Carnot refrigerator used for natural gas liquefaction is expected due to high difference between the temperatures \( T \) and \( T_0 \) in Equation 15.31. An average value of \( T \) can be obtained from the definition of the COP for a Carnot refrigerator, which is expressed as:

\[ \text{COP}_{R, \text{rev}} = \frac{1}{T_0/T - 1} \]  

(15.35)

Using this equation with COP = 1.8 and \( T_0 = 25^\circ \text{C} \), we determine \( T = -81.3^\circ \text{C} \). This is the temperature a heat reservoir would have if a Carnot refrigerator with a COP of 1.8 operated between this reservoir and another reservoir at 25\(^\circ\)C. Note that the same result could be obtained by writing Equation 15.31 in the form:

\[ w_{\text{min}} = q \left( 1 - \frac{T_0}{T} \right) \]  

(15.36)

where \( q = 823.0 \text{ kJ/kg}, w_{\text{min}} = 456.8 \text{ kJ/kg}, \) and \( T_0 = 25^\circ \text{C} \).

We now investigate how the minimum work changes with natural gas liquefaction temperature. We take the inlet pressure of natural gas to be 40 bar, the inlet temperature to
be $T_1 = T_0 = 25^\circ\text{C}$, and the exit state to be saturated liquid at the specified temperature. The properties of methane are obtained using EES software (Klein, 2006). Using the minimum work relation in Equation 15.33 and the reversible COP relation in Equation 15.35, the plots in Figure 15.6 are obtained.

The minimum work required to liquefy a unit mass of natural gas increases almost linearly with decreasing liquefaction temperature, as shown in Figure 15.6. Obtaining LNG at $-200^\circ\text{C}$ requires three times the minimum work to obtain LNG at $-100^\circ\text{C}$. Similarly, obtaining LNG at $-150^\circ\text{C}$ requires 1.76 times the minimum work to obtain LNG at $-100^\circ\text{C}$.

The COP of the Carnot refrigerator increases almost linearly with liquefaction temperature, as shown in Figure 15.6. The COP decreases almost by half when the liquefaction temperature decreases from $-100^\circ\text{C}$ to $-200^\circ\text{C}$. These figures show that the maximum possible liquefaction temperature should be used to minimize the work input. In other words, LNG should not be liquefied to lower temperatures than needed.

For the typical natural gas inlet and exit states specified in the previous section, the minimum work is determined to be 456.8 kJ/kg of LNG. A typical actual value of work input for a cascade cycle used for natural gas liquefaction is given by Finn et al. (1999) to be 1188 kJ/kg of LNG. Then the exergetic efficiency of a typical cascade cycle can be determined from Equation 15.30 to be 38.5%. The actual work input required depends mainly on the feed and ambient conditions and the compressor efficiency.

15.3.5 Discussion

Recent advances have made it possible to replace the Joule Thompson (JT) expansion valve of the cycle with a cryogenic hydraulic turbine. The same pressure drop as in the JT valve is achieved with the turbine while producing power. Using the same typical values as before, we take the cryogenic turbine inlet state to be 40 bar and $-150^\circ\text{C}$. Assuming isentropic expansion to a pressure of 4 bar, the work output is calculated to be 8.88 kJ/kg of LNG. When this work is subtracted from the work input, this corresponds to a decrease of 2% in the minimum work input. The use of a cryogenic turbine results in 4% extra LNG production in an actual natural gas liquefaction plant (Verkoelen, 1996). Also, for the same inlet conditions, the temperature of LNG at the cryogenic turbine exit is shown to be lower than that at the expansion valve exit (Kanoglu, 2000b).

The main site of exergy destruction in the cycle is the compressors. Improvements to the exergy efficiencies of the compressors will reduce the work input for the liquefaction process. Having three-stage evaporation for each refrigerant in the cascade cycle results in a total of nine evaporation temperatures. Also, having multiple stages makes the average temperature difference between the natural gas and the refrigerants small. This results in a smaller exergy destruction in the evaporators since the greater the temperature difference, the greater the exergy destruction. As the number of evaporation stages increases, the exergy destruction decreases. However, adding more stages means additional equipment cost. More than three stages for each refrigerant is usually not justified.

15.4 Exergy Analysis of an Integrated Hydrogen Liquefaction Using Geothermal Energy

In this section, an exergy analysis is described of an advanced conceptual system for hydrogen liquefaction. Liquefaction of hydrogen is of increasing interest (Kanoglu...
et al., 2010), in part because of hydrogen’s potential as an energy carrier. The system integrates a triple effect absorption cooling system (TEACS), which is used for precooling to improve efficiency and economics, and an LH hydrogen liquefaction system.

Geothermal energy drives the process. High-temperature geothermal resources (>150°C) are generally used for power generation, while moderate-temperature (90°C–150°C) and low-temperature (<90°C) resources are best suited for direct uses such as space and process heating, cooling, aquaculture, and fish farming. Common types of geothermal power plants include dry steam, single- and double-flash, binary, and flash/binary cycles (Kanoğlu et al., 2010). Here, we consider a binary isobutene system for electricity generation using a high-temperature geothermal source; the electricity drives the LH system.

Many types of absorption cooling systems are available, including lithium bromide/water chillers, which are suitable for space cooling, and ammonia/water systems, which can provide industrial cooling to as low as −50°C (ASHRAE, 2006). Single- and double-effect absorption cooling systems are relatively well understood (Ziegler et al., 1993; Dincer and Dost, 1996), and TEACS are being investigated (Ratlamwala et al., 2010, 2011a,b).

15.4.1 System Description

The integrated system considered here is shown in simplified form in Figure 15.7a, and its three main subsystems are illustrated in Figure 15.7b–d.

In the TEACS using ammonia-water as working fluid (Figure 15.7b), thermal energy (heat) is provided to the high-temperature geothermal water (HTG) via the high-temperature geothermal water from the underground reservoir and the return geothermal water from the binary isobutene power plant. The TEACS precools hydrogen in the evaporator from 25°C to 16.4°C, at 1 atm.

The precooled hydrogen is fed to the LH liquefaction cycle (Figure 15.7c). Makeup gas is mixed with the precooled uncondensed portion of the gas from the TEACS, and the mixture at state 1 (101 kPa) is compressed to state 2 (10 MPa). Heat is removed from the compressed gas by a coolant. The high-pressure gas is cooled to state 3 in a regenerative countercflow heat exchanger (I) by the uncondensed gas, and is cooled further by flowing through two nitrogen baths (II and IV) and two regenerative heat exchangers (III and V) before being throttled to state 8, where it is a saturated liquid—vapor mixture. The liquid hydrogen product exits and the vapor is routed through the bottom half of the cycle. After, supplying heat in a regenerative heat exchanger, the gas is mixed with fresh makeup gas from the TEACS and the cycle repeats.

The power to the compressor in the LH cycle is provided by a binary geothermal power plant with isobutene as the working fluid (Figure 15.7d). The isobutene is heated and vaporized using geothermal water from the HTG at 20°C below the geothermal source temperature. This vaporized isobutene flows through the turbine, and is then condensed and pumped back to the heat exchanger. The turbine inlet and exit pressures are taken to be 3000 and 400 kPa, respectively. The temperature at the turbine inlet is taken to be 15°C below the geothermal water temperature at the heat exchanger inlet. The geothermal water exiting the isobutene heat exchanger passes to the HTG for heat recovery before it returns to the underground well.

15.4.2 Analysis

Mass and energy balances are written for the integrated system and its components, and selected energy and exergy parameters are evaluated, including binary system efficiencies, TEACS COPs, and system utilization factors.

15.4.2.1 TEACS

Heat is provided to the HTG of the absorption cooling system using geothermal water. The geothermal water loop supplies the geothermal water from the binary system back to the HTG for heat recovery. The rate of heat transfer from the geothermal water source is expressible as follows:

\[
\dot{Q}_{geo,low} = \dot{m}_{geo}(h_{geo,source} - h_{geo,HTG,exit})
\]  
(15.37)

\[
\dot{Q}_{geo,high} = \dot{m}_{geo}(h_{geo,HTG,exit} - h_{geo,binary,exit})
\]  
(15.38)

The heat supply rate to the HTG is given by:

\[
\dot{Q}_{HTG} = \dot{Q}_{geo,low} + \dot{Q}_{geo,high}
\]  
(15.39)

The exergy destruction in the HTG can be written as:

\[
\dot{E}_{x,HTG} = \dot{E}_{x,30} + \dot{E}_{x,31} - \dot{E}_{x,29}
\]  
(15.40)

and similar relations are written for the other generators. The rate of exergy into the evaporator is calculated as:

\[
\dot{E}_{x,eva} = \dot{m}_{11}((h_{10} - h_{11}) - T_0(s_{10} - s_{11}))
\]  
(15.41)

where \( T_0 \) is taken as 298.15 K. The energetic and exergetic COPs are expressed as follows:

\[
\text{COP}_{en} = \frac{\dot{Q}_{eva}}{\dot{Q}_{geo} + w_p}
\]  
(15.42)

\[
\text{COP}_{ex} = \frac{\dot{E}_{x,eva}}{\dot{E}_{x,geo} + w_p}
\]  
(15.43)
where $\dot{w}_p$ denotes pump power. The exergy efficiency of the absorption cooling system is defined as:

$$\psi_{abs} = \frac{\text{COP}_{en}}{\text{COP}_{ex}} \quad (15.44)$$

15.4.2.2 Binary Power Plant

The electrical power produced by the turbine of the binary isobutene plant is as follows:

$$\dot{W}_{turb} = \dot{m}_{iso}(h_3 - h_4) \quad (15.45)$$

The power consumed by pump in binary cycle and parasitic losses, respectively, can be expressed as:

$$\dot{W}_{p iso} = \dot{m}_{iso}(h_2 - h_1) \quad (15.46)$$

$$\dot{W}_{parasitic} = 0.2(\dot{W}_{turb} - \dot{W}_{p iso}) \quad (15.47)$$

so the net power from the binary isobutene plant can be expressed as:

$$\dot{W}_{net_{iso}} = \dot{W}_{turb} - \dot{W}_{p iso} - \dot{W}_{parasitic} \quad (15.48)$$
15.4.2.3 LH Liquefaction Cycle

The ideal specific work required to compress hydrogen from $P_1$ to $P_2$ is given as follows:

$$w_{\text{comp,ideal}} = RT_0 \ln \frac{P_2}{P_1}$$ (15.49)

However, the actual specific work input to liquefaction cycle per unit mass of hydrogen is:

$$w_{\text{comp,actual}} = \frac{w_{\text{comp,ideal}}}{\eta_{\text{comp}}}$$ (15.50)

Here, $P_1 = 101$ kPa and $P_2 = 10$ MPa, $\eta_{\text{comp}}$ is 65%, and the electricity needed for nitrogen is taken to be 5050 kJ/kg (Nandi and Sarangi, 1993).

15.4.2.4 Utilization Factors

The energetic and exergetic utilization factors of the integrated system are expressible as follows:

$$u_{\text{en}} = \frac{\dot{E}_{\text{H}_2}}{\dot{W}_p + Q_{\text{geo,high}} + \dot{w}_{\text{comp,actual}}}$$ (15.51)

where

$$\dot{E}_{\text{H}_2} = \dot{m}_{\text{H}_2,\text{liq,final}} (h_{\text{H}_2,j} - h_{\text{H}_2,0})$$ (15.52)

$$\dot{m}_{\text{H}_2,\text{liq,final}} = f_{\text{liq}} \dot{m}_{\text{H}_2}$$ (15.53)

$$u_{\text{ex}} = \frac{\dot{E}_{\text{H}_2}}{\dot{W}_p + \dot{E}_{\text{geo,high}} + \dot{w}_{\text{comp,actual}}}$$ (15.54)

$$\dot{E}_{\text{H}_2} = \dot{m}_{\text{H}_2,\text{liq,final}} \left[ (h_{\text{H}_2,j} - h_{\text{H}_2,0}) - T_0 (s_{\text{H}_2,j} - s_{\text{H}_2,0}) \right]$$ (15.55)

15.4.3 Performance

The performance is determined for the TEACS, the binary power plant, the LH liquefaction system, and the overall system. Given the importance of geothermal energy as the driver of the integrated system, the performance parameters are determined as a function of mass flow rate and temperature of the geothermal water source.
15.4.3.1 Performance Variations with Geothermal Mass Flow Rate

An increase in the mass flow rate of geothermal water ($\dot{m}_{\text{geo}}$) results in higher energy input to the TEACS. This results in a lower output from the evaporator in form of cooling load for a fixed condenser load. The energy input to the HTG increases from 185 to 370 kW as $\dot{m}_{\text{geo}}$ increases from 1.5 to 3.0 kg/s. The cooling load of the TEACS varies from 268.2 to 43.6 kW as $\dot{m}_{\text{geo}}$ increases from 1.5 to 3.0 kg/s for a varying condenser load from 300 to 340 kW and for a fixed geothermal source temperature of 473 K. As the condenser load increases, the evaporator load rises because of a higher heat rejection from the flow exiting the generator. This higher heat rejection lowers the temperature of the input flow to the evaporator and therefore increases the cooling load.

The increase in energy input to the TEACS and decrease in cooling load with increase in $\dot{m}_{\text{geo}}$ lowers the energetic and exergetic COPs of the TEACS, as shown in Figure 15.8. The energetic and exergetic COPs vary from 1.4 to 0.12 and 1.0 to 0.08, respectively, with increasing $\dot{m}_{\text{geo}}$. The decrease in the COPs is a direct result of higher energy input to the system and the lower output from the system in the form of cooling load. Also, the increase in $\dot{m}_{\text{geo}}$ results in a higher TEACS exergy efficiency and a higher net work output from the geothermal power plant. Exergy efficiency and net work output increase from 0.69 to 0.70 and 69.7 to 139.4 kW, respectively, with increasing $\dot{m}_{\text{geo}}$ (see Figure 15.9). This behavior results because an increase in $\dot{m}_{\text{geo}}$ for a fixed temperature difference results in a higher energy input to the TEACS and a higher work output from the binary power plant.

![Figure 15.8](image1.png) Effect of geothermal water mass flow rate on energetic and exergetic COP.

![Figure 15.9](image2.png) Effect of geothermal water mass flow rate on Second Law efficiency of TEACS and net work output of the binary power plant.
The energetic and exergetic utilization factors vary from 0.06 to 0.009 and from 0.006 to 0.21, respectively, with increasing $m_{\text{geo}}$ (see Figure 15.10). This trend occurs because increasing $m_{\text{geo}}$ lowers the TEACS cooling load and thus decreases the amount of hydrogen gas precooled, which ultimately decreases the amount of hydrogen gas liquefied.

### 15.4.3.2 Performance Variations with Geothermal Source Temperature

Increasing the geothermal water source temperature ($T_{\text{geo}}$) reduces the energetic and exergetic COPs of the TEACS (see Figure 15.11). The energetic and exergetic COPs vary from 0.82 to 0.67 and from 0.59 to 0.09, respectively, as $T_{\text{geo}}$ increases from 570 to 630 K for varying condenser loads of 300, 320, and 340 kW and for $m_{\text{geo}} = 2.0$ kg/s. Increasing the geothermal source water temperature raises the energy transfer to the HTG of TEACS by increasing the temperature difference. This increase in energy input to the TEACS lowers the output from the TEACS cooling load for a fixed condenser load, decreasing the TEACS COPs.

### 15.4.4 Further Discussion and Findings

The analysis confirms, in general, that renewable energy systems can be environmentally benign, efficient, and sustainable, and that the integrated system considered provides an attractive hydrogen liquefaction option in terms of energy consumption, greenhouse gas emissions, and other factors. This system can be modified for liquefying other gases. The key observations regarding the effect of geothermal parameters on the integrated system considered here include:

![Figure 15.10](image-url)  
**FIGURE 15.10** Effect of geothermal water mass flow rate on utilization factor.

![Figure 15.11](image-url)  
**FIGURE 15.11** Effect of geothermal water source temperature on energetic and exergetic COPs.
• The cooling load, energetic and exergetic COPs, amounts of hydrogen gas precooled and liquefied, and energetic and exergetic utilization factors decrease as the mass flow rate of geothermal water increases.
• The energy input to the TEACS HTG, the TEACS exergy efficiency, and the net work of the binary power system increase as the mass flow rate of geothermal water increases.
• The energetic and exergetic COPs decrease as the geothermal water source temperature increases.

15.5 CLOSING REMARKS
Exergy analyses of cryogenic systems, particularly gas liquefaction systems, have been described and studied for various integrated systems. Exergy analysis is particularly important for cryogenic and liquefaction applications, because the exergy of a cryogenic fuel becomes increasingly significant as its temperature decreases well below the environment temperature.

PROBLEMS
15.1 What is the difference between a refrigeration system and a gas liquefaction system? Which system typically involves higher exergy destruction and thus lower exergy efficiency?
15.2 Compare the following processes: (a) transportation of natural gas in the pipelines and (b) liquefying the natural gas and then transporting it in tanks.
15.3 Compare the following processes from an exergetic point of view: (a) transportation of natural gas as a gas in tanks and (b) liquefying the natural gas and then transporting it in tanks.
15.4 How can you express the COP and exergy efficiency of a gas liquefaction process? Can the COP and exergy efficiency be greater than 1?
15.5 Provide three alternative definitions of exergy efficiency for a gas liquefaction system.
15.6 Write an expression for the minimum work for the liquefaction of a gas. What is the relationship between this expression and the exergy of the heat removed from the gas during the liquefaction process?
15.7 Which is greater in a gas liquefaction system: the heat removal per unit mass of the gas or the heat removal per unit mass of liquefaction? Can one be determined from the other?
15.8 How does the exergy efficiency of a gas liquefaction system change with liquefaction temperature?
15.9 Investigate various cycles used for gas liquefaction and compare them from an exergetic perspective.
15.10 What is the importance of liquefying hydrogen for a future hydrogen economy? Can the LH process described in this chapter be used for hydrogen liquefaction? If not, what modifications would be needed to allow for hydrogen liquefaction? Which cycles are currently used for hydrogen liquefaction?
15.11 Can an absorption cooling system be used for gas liquefaction? Explain.
15.12 Compare the work required to compress a gas in a gas liquefaction cycle using (a) an isothermal compressor and (b) an isentropic compressor.
15.13 Do you favor replacing the expansion valve with a turbine in a gas liquefaction system? Explain from an exergetic perspective.
15.14 Provide some guidelines in the selection of refrigerants for use in oxygen, methane, and hydrogen liquefaction systems?
15.15 Investigate the use of cryogenic turbines in natural gas liquefaction systems.
15.16 Which components of the natural gas liquefaction system considered in this chapter involve greater exergy destructions? Provide methods for reducing or minimizing exergy losses.
15.17 Obtain a published article on exergy analysis of a gas liquefaction system. Using the operating data provided in the article, perform a detailed exergy analysis of the system and compare your results to those in the original article. Also, investigate the effect of varying important operating parameters on the system exergetic performance.
15.18 Obtain actual operating data from a gas liquefaction system and perform a detailed exergy analysis. Discuss the results and provide recommendations based on the exergy results for improving the efficiency.
15.19 Consider the system shown in Figure 15.4. Conduct detailed exergy analysis of the system to create a bar chart showing how much exergy is destroyed in each component of the system. Also, make a bar chart showing the percentage of exergy destruction in each component of the system.
15.20 Consider the integrated system shown in Figure 15.7a. Conduct detailed exergy analysis of the system to create a bar chart showing how much exergy is destroyed in each component of the system. Also, make a bar chart showing the percentage of exergy destruction in each component of the system.
15.21 For the integrated system shown in Figure 15.7a, replace the working fluid of TEACS and a binary system with lithium-bromide-water and ammonia, respectively. Conduct energy and exergy analyses and compare your results with the results provided in this chapter.