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# Energy and Exergy Analysis of Refrigeration Systems

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## Abstract

Refrigeration systems have the priority in design for residential and industrial applications. The chapter includes five major refrigeration systems: vapor-compression refrigeration; ammonia-water absorption refrigeration; gas refrigeration where standard air is the most popular refrigerant; multi-pressure refrigeration including multistage, cascade, and multipurpose refrigeration system; and heat pump systems. Energy and exergy analysis has been presented for most of the systems. The energetic and the exergetic COP for each system are presented. Renewable energy sources are also discussed including geothermal, solar, and wind energy, a with combination with refrigeration systems in different industrial and residential applications. The overall efficiency of the renewable systems is achieved to be more than 50% providing promising solutions for energy use and having a low environmental impact.

**Keywords:** refrigeration systems, absorption cooling system, heat pump, geothermal energy, solar energy, wind energy

## 1. Introduction

The primary application of refrigeration system is to transfer heat from a lower temperature region to a higher temperature one. A refrigeration cycle consists of a source at low temperature, a sink at high temperature, and a device to produce the work done to transfer heat from the source to sink. For the complete circulation, the refrigeration cycle should have an expansion device to circulate the refrigerant to the source.

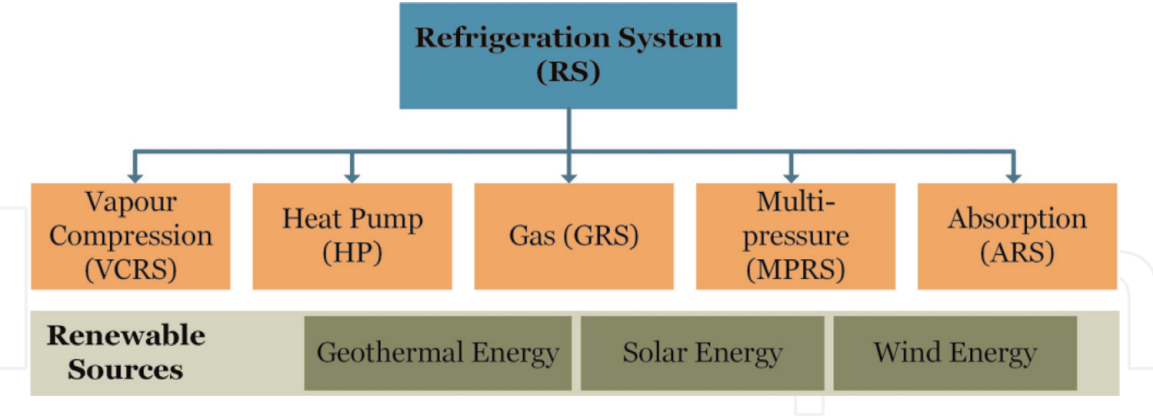
Major refrigeration systems include vapor-compression refrigeration system (VCRS), heat pump (HP), gas refrigeration system (GRS), multi-pressure refrigeration systems (MPRS), and absorption refrigeration system (ARS), as presented in **Figure 1**. These systems are combined with renewable sources, such as geothermal, solar, and wind energy sources.

## 2. Vapor-compression refrigeration system

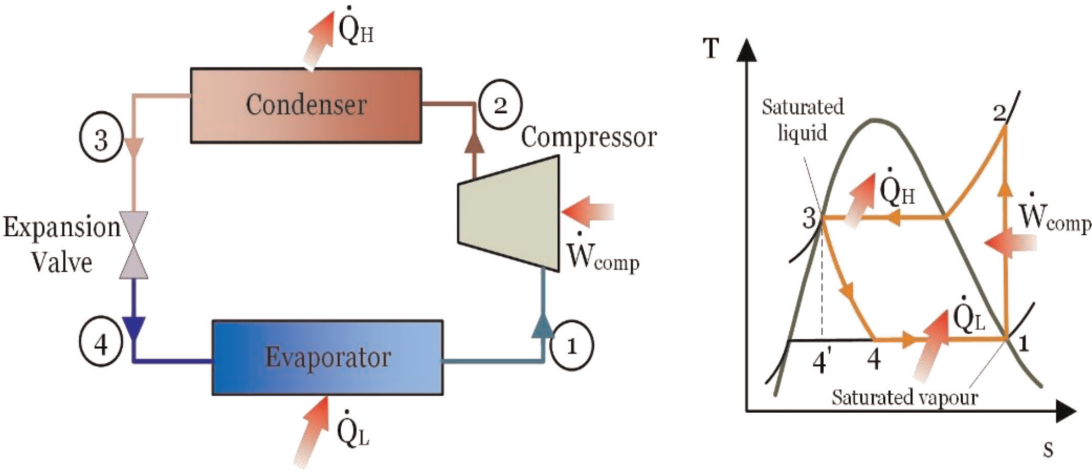
The vapor-compression refrigeration cycle (VCRS) is the most widely used cycle for refrigerators, air-conditioning systems, and heat pumps [1, 2]. It consists of four processes, as shown in **Figure 2**:

- 1-2 Isentropic compression in a compressor
- 2-3 Constant-pressure heat rejection in a condenser

- 3-4 Throttling in an expansion device
- 4-1 Constant-pressure heat absorption in an evaporator



**Figure 1.**  
*The classifications of refrigeration systems and renewable sources.*



**Figure 2.**  
*Schematic and T-s diagram for the ideal VCRS.*

The refrigerant enters the compressor from state 1 at saturated vapor to be isentropically compressed from low pressure of state 1 to high pressure and temperature of state 2, which is at the superheated region. Then, the refrigerant of state 2 enters the condenser to reject heat to the warm environment and exits at the saturated liquid as state 3. The refrigerant enters an adiabatic throttling or expansion valve to drop the pressure, which equals the pressure at the compressor inlet of state 1. The refrigerant temperature at state 1 is very low so that it absorbs heat from the refrigerated space at the evaporator and heated to be saturated vapor again. The vapor refrigeration system is a closed cycle where it starts and ends at state 1. This type of refrigeration system can be used for refrigerators, inside the air conditioners as split air conditioners, and separate as in radiant cooling systems [3, 4] and air-to-air systems [1].

2.1 Energy and exergy analysis of vapor-compression RS

The first and second law of Thermodynamics for steady-state flow is applied for each component and the whole system. They include the energy balance equation (EnBE) and exergy balance equation (ExBE) in this order. The energy balance equation considers the heat transfer and work produced or done crossing the control volume of a component or a system, while the exergy balance equation considers the irreversibilities of a process, which are described by the exergy

destruction. For the given system of **Figure 2**. The refrigerant mass flow rate is constant through the cycle and denotes as  $\dot{m}$  in kg/s,  $h$  is the specific enthalpy of each state point in kJ/kg,  $ex_i$  is the specific exergy of a state point in kJ/kg and defined as  $ex_i = (h_i - h_0) - T_0(s_i - s_0)$ , and  $s$  is the specific entropy of the refrigerant at each state point, and its unit is kJ/kg.K. The change in kinetic and potential energy is negligible for each component and the entire system. The energy balance equation (EnBE) of the evaporator considers the rate of heat removal by the evaporator,  $\dot{Q}_L$  which is released from a low-temperature environment and determined by Eq. (1) [5]. The exergy balance equation (ExBE) of the evaporator is given as Eq. (2). Also, the thermal exergy rate due to the heat transfer from the evaporator is defined as Eq. (3) [6].

$$\text{EnBE} \quad \dot{Q}_L = \dot{m}(h_1 - h_4) \quad (1)$$

$$\text{ExBE} \quad \dot{m}_4 ex_4 + \dot{Ex}_{Q, \text{evap}} = \dot{m}_1 ex_1 + \dot{Ex}_{\text{des, evap}} \quad (2)$$

$$\dot{Ex}_{Q, \text{evap}} = \dot{Q}_L \left( \frac{T_0}{T_L} - 1 \right) \quad (3)$$

The power input to the compressor,  $\dot{W}_{\text{comp}}$ , can be determined from Eq. (4), the isentropic efficiency of an adiabatic is defined as in Eq. (5) [5], and the exergy destruction of the compressor can be given as Eq. (6) [6]:

$$\text{EnBE} \quad \dot{W}_{\text{comp}} = \dot{m}(h_2 - h_1) \quad (4)$$

$$\eta_{\text{comp}} = \frac{\dot{W}_{\text{in}}}{\dot{W}} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (5)$$

$$\text{ExBE} \quad \dot{m}_1 ex_1 + \dot{W}_{\text{in}} = \dot{m}_2 ex_2 + \dot{Ex}_{\text{des, comp}} \quad (6)$$

The heat rejection rate from the condenser,  $\dot{Q}_H$ , to the environment can be written as Eq. (7) [5], while the exergy destruction and thermal exergy rate of the condenser can be given as Eqs. (8) and (9) [6]:

$$\text{EnBE} \quad \dot{Q}_H = \dot{m}(h_2 - h_3) \quad (7)$$

$$\text{ExBE} \quad \dot{m}_2 ex_2 = \dot{m}_3 ex_3 + \dot{Ex}_{Q, \text{cond}} + \dot{Ex}_{\text{des, cond}} \quad (8)$$

$$\dot{Ex}_{Q, \text{cond}} = \dot{Q}_H \left( 1 - \frac{T_0}{T_H} \right) \quad (9)$$

The energy and exergy balance equations for the expansion valve can be expressed as Eqs. (10) and (11), respectively. The expansion valves are considered to be decreasing the pressure adiabatically and isentropically, which means no heat transfer and work done in the throttling process [6]:

$$\text{EnBE} \quad \dot{m}h_3 = \dot{m}h_4 \quad (10)$$

$$\text{ExBE} \quad \dot{m}_3 ex_3 = \dot{m}_4 ex_4 + \dot{Ex}_{\text{des, exp}} \quad (11)$$

The energy balance for the entire refrigeration system can be given as [5]:

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{comp}} \quad (12)$$

The coefficient of performance (COP) of the refrigeration system is defined as the ratio of useful energy, which is the rate of heat removal by the evaporator to the

required energy, which is the power required to operate the compressor. The COP is given as below [5]:

$$COP = \frac{\dot{Q}_L}{\dot{W}_{comp}} \quad (13)$$

The Carnot or reversible COP is defined as the maximum COP of a refrigeration cycle operating between temperature limits  $T_L$  and  $T_H$ , which can be given as Eq. (14) [5]:

$$COP_{rev} = T_L / (T_H - T_L) \quad (14)$$

An actual vapor-compression refrigeration cycle differs from the ideal one because of the irreversibilities that occur in various components, such as fluid friction (causes pressure drops) and heat transfer to or from the surroundings. The aim of exergy analysis is to determine the exergy destruction in each component of the system and to determine the exergy efficiency of the entire system. Exergy destruction in a component can be evaluated based on entropy generation and an exergy balance equation using Eq. (15) [6]:

$$\dot{E}x_{des} = T_0 \dot{S}_{gen} \quad (15)$$

where  $T_0$  is the dead-state temperature or environment temperature. In a refrigerator,  $T_0$  usually equals the temperature of the high-temperature medium  $T_H$ .

The exergetic coefficient of performance ( $COP_{ex}$ ) of the refrigeration system is the second-law efficiency of the cycle. It is defined as the ratio of useful exergy rate, which is the thermal exergy of the heat removed by the evaporator, to the required exergy rate, which is the work done by the compressor. The  $COP_{ex}$  can be written as [6]:

$$COP_{ex} = \frac{\dot{E}x_{Q, evap}}{\dot{W}_{comp}} = \frac{\dot{Q}_L \left( \frac{T_0}{T_L} - 1 \right)}{\dot{W}_{comp}} \quad (16)$$

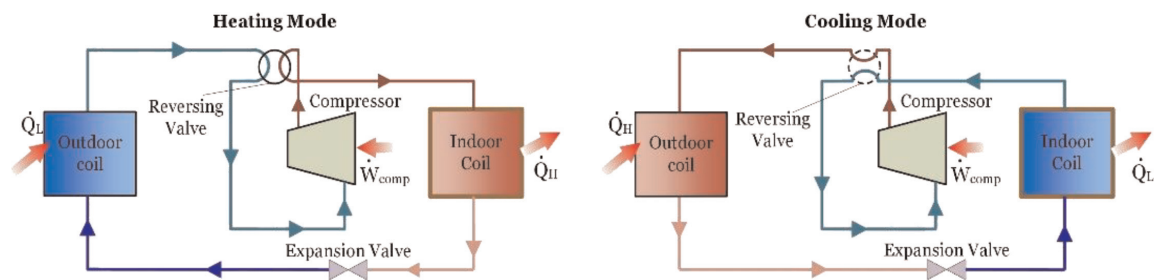
by substituting  $\dot{W}_{comp} = \dot{Q}_L / COP_{en}$ , and the  $COP_{ex}$  can be defined as the maximum COP of a refrigeration cycle operating between temperature limits  $T_L$  and  $T_H$ , which can be given as Eq. (14). Therefore, the second-law efficiency or  $COP_{ex}$  can be rewritten as Eq. (17) [6]:

$$COP_{ex} = \frac{\dot{Q}_L \left( \frac{T_0}{T_L} - 1 \right)}{\dot{W}_{comp}} = \frac{\dot{Q}_L \left( \frac{T_0}{T_L} - 1 \right)}{\dot{Q}_L / COP} = \frac{COP}{T_L / (T_H - T_L)} = \frac{COP}{COP_{rev}} \quad (17)$$

Since  $T_0 = T_H$  for a refrigeration cycle, thus, the second-law efficiency is also equal to the ratio of actual and maximum COPs for the cycles, which accounts for all irreversibilities associated within the refrigeration system.

### 3. Heat pump system

Heat pump system (HP) is similar to VCRS since it consists of a compressor, expansion valve, and outdoor and indoor coils, which operate exchangeably as



**Figure 3.**  
A heat pump can be used to heat a house in winter and to cool it in summer.

condenser and evaporator. The advantage of HP systems is the ability to provide cooling and heating for the desired space, especially for the long winter season as in Canada and north European countries. This can be achieved by adding a reversing valve, as shown in **Figure 3**. There are two essential modes: heating mode and cooling mode. The condenser and evaporator are exchanging during the cooling and heating season since the reversing valve is switching between two modes according to the weather condition.

The energy source for heat pump can be classified into air-source, water-source, and ground-source. The air-source system uses atmospheric air through the evaporator, while the water-source system uses well water of depth 80 m and operates from 5 to 18°C. The ground-source system uses long piping under the ground since the soil temperature is not affected by climate change. The capacity and efficiency of heat pump drop at low-temperature environment, and therefore, other auxiliary systems, such as heaters or furnaces, are used to provide sufficient heating load for residential buildings.

The COP of a heat pump is defined as the ratio of the heat removed for cooling mode or added for a heating mode of the indoor coil to the compressor power. Therefore, the  $COP_{heating}$  and  $COP_{cooling}$  are given in Eq. (19).  $\dot{Q}_{in}$  can be  $\dot{Q}_H$  for heating mode or  $\dot{Q}_L$  for cooling mode [6]:

$$COP = \frac{\dot{Q}_{in}}{\dot{W}_{comp}} \quad (18)$$

$$COP_{heating} = \frac{\dot{Q}_H}{\dot{W}_{comp}} \quad \text{and} \quad COP_{cooling} = \frac{\dot{Q}_L}{\dot{W}_{comp}} \quad (19)$$

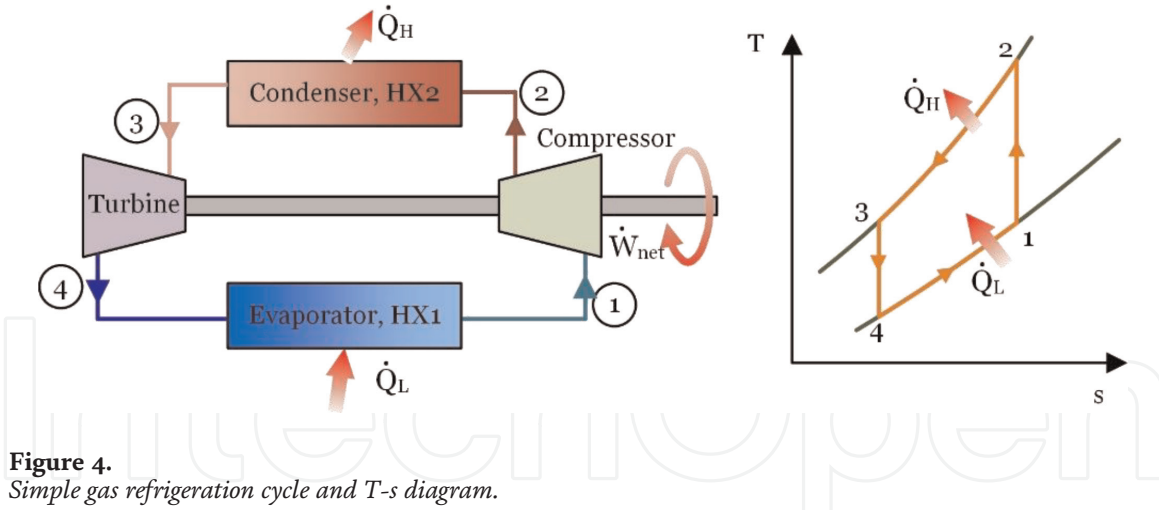
The exergetic COP is defined as the ratio of thermal exergy rate divided by the compressor power. It is also given as the ratio of COP to the reversible COP for both heating and cooling mode.  $T_{in}$  can be considered as  $T_H$  for heating mode and  $T_L$  for cooling mode [6]:

$$COP_{ex} = \frac{\dot{Q}_{in} |1 - T_0/T_{in}|}{\dot{W}_{comp}} = \frac{COP}{COP_{rev}} \quad (20)$$

#### 4. Gas refrigeration system

The VCRES is known as a modified, reverse Rankine cycle, while the gas refrigeration system (GRS) is known as a reverse Brayton cycle using a noncondensing gas such as air. The main advantage of this system is the small size for achieving the desired cooling due to the lighter weight of air than other refrigerants. This system can be used in aircraft cabin cooling.





**Figure 4.** Simple gas refrigeration cycle and T-s diagram.

As illustrated in **Figure 4**, the major elements of GRS are compressor to raise the pressure of gas from state 1 to 2, a rejecting heat exchanger (condenser), turbine or expander to decrease the gas pressure isentropically, and an absorbing heat exchanger (evaporator) to absorb the heat from the refrigerated space at constant pressure. A regenerator heat exchanger can be added to the system for heat recovery between the hot and cold paths of circulated gas. It can be located between the two heat exchangers. Air is a popular refrigerant of this system since it can be utilized as a refrigerant and air-conditioning medium in smaller equipment units as aircraft cooling systems.

#### 4.1 Energy and exergy analysis of gas refrigeration system

The energy analysis of a gas refrigeration system is similar to that of the vapor refrigeration system except that the gaseous fluid is treated as an ideal gas. Therefore, the enthalpy and entropy equations are written as [5]:

$$\Delta h = (h_e - h_i) = c_p \Delta T = c_p (T_e - T_i) \quad (21)$$

$$\Delta s = (s_e - s_i) = c_p \ln \frac{T_e}{T_i} - R \ln \frac{P_e}{P_i} \quad (22)$$

where the subscripts  $i$  and  $e$  indicate inlet and exit states, respectively. Therefore, the energy and exergy analysis for each component of **Figure 5** is listed below [5, 6].  
Compressor:

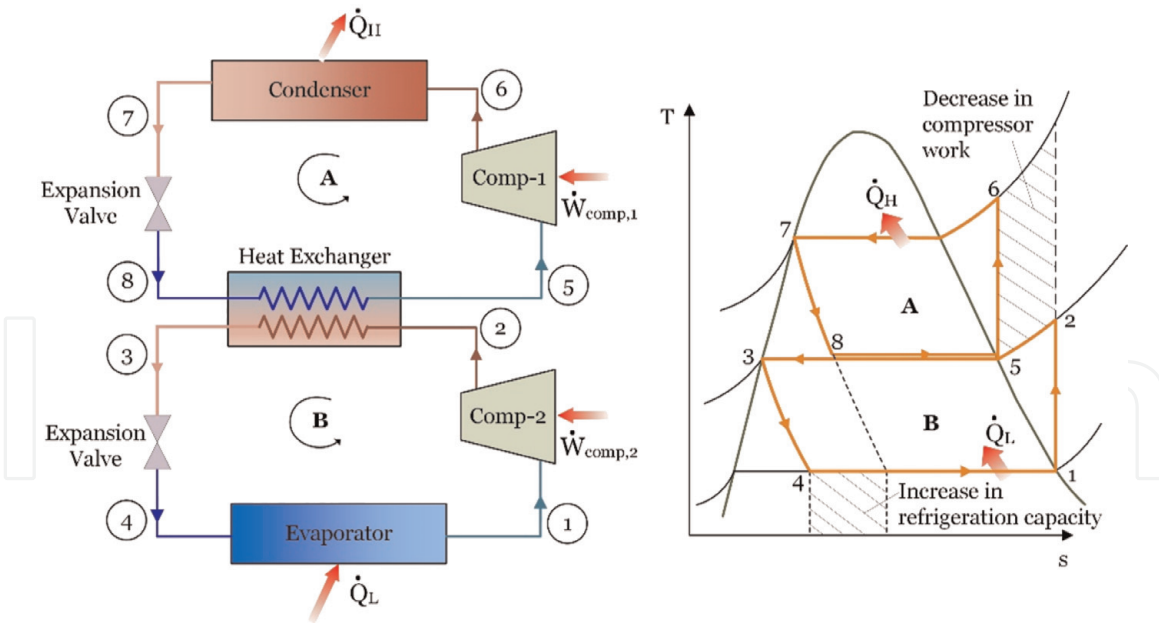
$$\text{EnBE} \quad \dot{m}h_1 + \dot{W}_{\text{Comp}} = \dot{m}h_2 \Rightarrow \dot{W}_{\text{Comp}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1) \quad (23)$$

$$\begin{aligned} \text{ExBE} \quad \dot{E}x_{\text{des, comp}} &= T_0 \dot{S}_{\text{gen, 1-2}} = \dot{m}T_0(s_2 - s_1) \\ &= \dot{m}T_0 \left( c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) \end{aligned} \quad (24)$$

Heat exchanger 2 (condenser):

$$\text{EnBE} \quad \dot{m}h_2 = \dot{Q}_H + \dot{m}h_3 \Rightarrow \dot{Q}_H = \dot{m}(h_2 - h_3) = \dot{m}c_p(T_2 - T_3) \quad (25)$$

$$\begin{aligned} \text{ExBE} \quad \dot{E}x_{\text{des, HX2}} &= T_0 \dot{S}_{\text{gen, 2-3}} = \dot{m}T_0 \left( s_3 - s_2 + \frac{q_H}{T_H} \right) \\ &= \dot{m}T_0 \left\{ \left( c_p \ln \frac{T_3}{T_2} - R \ln \frac{P_3}{P_2} \right) + \frac{q_H}{T_H} \right\} \end{aligned} \quad (26)$$



**Figure 5.**  
A two-stage cascade refrigeration system with the same refrigerant in both stages.

Turbine (expander):

$$\text{EnBE} \quad \dot{m}h_3 = \dot{W}_{turb} + \dot{m}h_4 \Rightarrow \dot{W}_{turb} = \dot{m}(h_3 - h_4) = \dot{m}c_p(T_3 - T_4) \quad (27)$$

$$\begin{aligned} \text{ExBE} \quad \dot{E}x_{des,turb} &= T_0 \dot{S}_{gen,3-4} = \dot{m}T_0(s_4 - s_3) \\ &= \dot{m}T_0 \left( c_p \ln \frac{T_4}{T_3} - R \ln \frac{P_4}{P_3} \right) \end{aligned} \quad (28)$$

Heat exchanger 1 (evaporator):

$$\text{EnBE} \quad \dot{m}h_4 + \dot{Q}_L = \dot{m}h_1 \Rightarrow \dot{Q}_L = \dot{m}(h_1 - h_4) = \dot{m}c_p(T_1 - T_4) \quad (29)$$

$$\begin{aligned} \text{ExBE} \quad \dot{E}x_{des,HX1} &= T_0 \dot{S}_{gen,4-1} = \dot{m}T_0 \left( s_1 - s_4 + \frac{q_L}{T_L} \right) \\ &= \dot{m}T_0 \left\{ \left( c_p \ln \frac{T_1}{T_4} - R \ln \frac{P_1}{P_4} \right) + \frac{q_L}{T_L} \right\} \end{aligned} \quad (30)$$

For the entire refrigeration system, the energy balance can be written as:

$$\dot{W}_{comp} + \dot{Q}_L = \dot{W}_{turb} + \dot{Q}_H \quad (31)$$

The net power for the system becomes:

$$\dot{W}_{net} = \dot{W}_{comp} - \dot{W}_{turb} \quad (32)$$

The COP of the gas refrigeration system is given as:

$$COP = \frac{\dot{Q}_L}{\dot{W}_{net}} = \frac{\dot{Q}_L}{\dot{W}_{comp} - \dot{W}_{turb}} \quad (33)$$



The total exergy destruction in the system can be calculated by adding exergy destructions of each component:

$$\dot{Ex}_{des,total} = \dot{Ex}_{des,turb} + \dot{Ex}_{des,comp} + \dot{Ex}_{des,HX1} + \dot{Ex}_{des,HX2} \quad (34)$$

It can also be expressed as:

$$\dot{Ex}_{des,total} = \dot{W}_{net} - \dot{Ex}_{Q,HX1} = \dot{W}_{net} - \dot{Q}_L \left( \frac{T_0}{T_L} - 1 \right) \quad (35)$$

Thus, the minimum power input to accomplish the required refrigeration load  $\dot{Q}_L$  is equal to the thermal exergy rate of the heat exchanger I (evaporator)  $\dot{W}_{min} = \dot{Ex}_{Q,HX1}$ . Consequently, the second-law efficiency or the exergetic COP is defined as [6]:

$$COP_{ex} = \frac{\dot{Ex}_{Q,HX1}}{\dot{W}_{net}} = 1 - \frac{\dot{Ex}_{des,total}}{\dot{W}_{net}} \quad (36)$$

## 5. Multi-pressure refrigeration system

The VCERS is the most popular refrigeration cycle because it is simple, inexpensive, and reliable. However, the industrial refrigeration systems should be efficient by providing more refrigeration load. This can be achieved by modifying the simple VCERS into multi-pressure refrigeration systems (MPRS). The MPRS can be classified into cascade RS, multi-compression RS, and multipurpose RS.

### 5.1 Cascade refrigeration systems

Some industrial applications require low temperature below  $-70^\circ\text{C}$  with substantially large pressure and temperature difference ( $-70$  to  $100^\circ\text{C}$ ). VCERS cannot achieve these applications because it can operate within a temperature range of  $+10$  to  $-30^\circ\text{C}$ . Therefore, a modification of VCERS can be performed by using multiple refrigeration cycles operating in series, the so-called cascade refrigeration systems. The refrigerants of each cycle can be different. The evaporator of the first refrigeration cycle is connected to the condenser of the next refrigeration system forming an interchange heat exchanger between the 2 cycles, as shown in **Figure 5**. Cascade refrigeration systems are mainly used for liquefaction of natural gas, hydrogen, and other gases [7–9]. The major benefit of this system is decreasing the compressor power and increasing the refrigeration load compared with a VCERS with large temperature and pressure difference, as shown in the T-s diagram of cascade system in **Figure 5**. Therefore, reducing system components can be fulfilled in an appropriate way [2].

The net compressor power can be determined by the summation of all compressor power in all cascaded refrigeration system and written as [2]:

$$\dot{W}_{net} = \dot{W}_{comp,1} + \dot{W}_{comp,2} = \dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1) \quad (37)$$

The refrigeration load can be described as:

$$\dot{Q}_L = \dot{m}_B(h_1 - h_4) \quad (38)$$

The heat exchanger that connects the 2 cycles together has an energy balance equation as follows [5]:

$$\dot{m}_B h_2 + \dot{m}_A h_8 = \dot{m}_B h_3 + \dot{m}_A h_5 \quad (39)$$

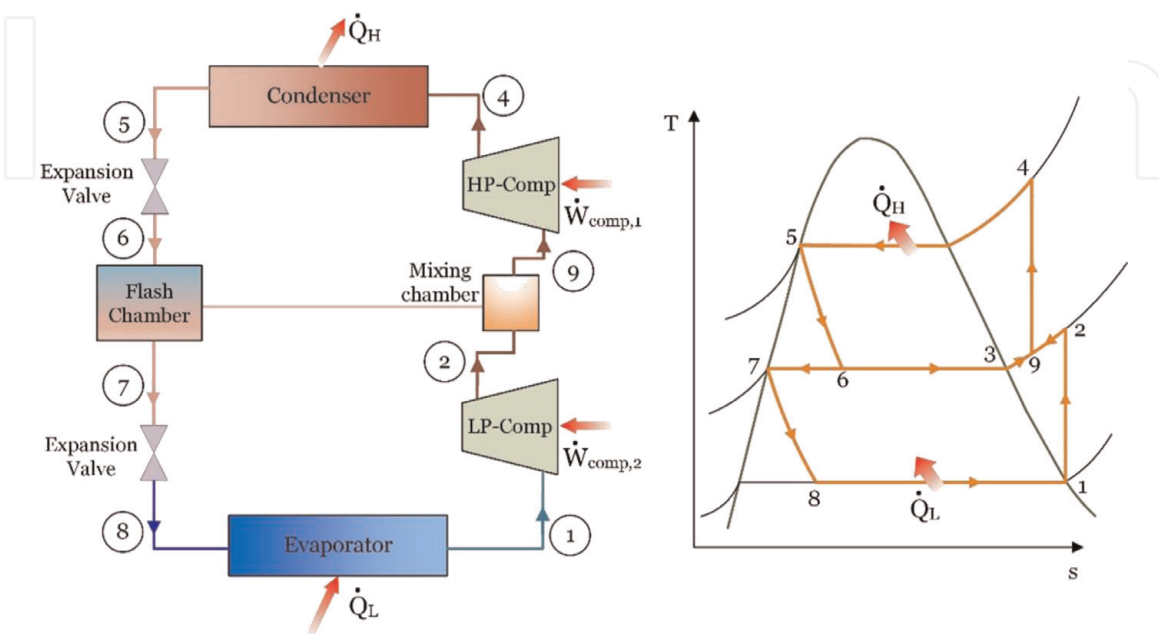
Therefore, the COP and exergetic COP of the cascade refrigeration system can be explained as the following [5, 6]:

$$COP = \frac{\dot{Q}_L}{\dot{W}_{net}} = \frac{\dot{m}_B (h_1 - h_4)}{\dot{W}_{comp,1} + \dot{W}_{comp,2}} \quad (40)$$

$$COP_{ex} = \frac{\dot{Ex}_{Q, evap}}{\dot{W}_{net}} = \frac{\dot{Q}_L \left( \frac{T_0}{T_L} - 1 \right)}{\dot{W}_{comp,1} + \dot{W}_{comp,2}} \quad (41)$$

## 5.2 Multistage compression refrigeration systems

Similar to the cascade refrigeration system, multistage compression refrigeration system is used for applications below  $-30^\circ\text{C}$ . This requires a large-pressure-ratio compressor and cannot be performed by one compressor because of the lack of efficiency and performance. Therefore, using multistage compressors connected in series can improve the performance of the refrigeration system by increasing the pressure ratio and increasing the refrigeration load. As shown in **Figure 6**, a two-stage compression refrigeration cycle consists of two compressors, a condenser, an evaporator, a flash intercooler, a mixer, and two throttling valves. The compressors. The upper compressor compresses the total refrigerant mass flow rate in a vapor form from the intermediate pressure of state 9 to the high pressure of state 4. The vapor refrigerant cools down in the condenser to saturated liquid at high pressure of state 5 and then passes through the upper expansion valve to reduce the pressure to intermediate pressure. The wet refrigerant passes through the flash intercooler to split the vapor and liquid phase. The vapor phase at state 3 enters the mixer to mix with the exit superheated refrigerant of the lower compressor at state 2. The liquid phase at state 7 is expanded by the lower throttling valve to state 8, which enters the lower pressure evaporator to absorb heat from the refrigerated space.



**Figure 6.**  
 A two-stage compression refrigeration system with a flash chamber.

The minimum temperature can be achieved by two-stage compression at  $-65^{\circ}\text{C}$ , while the three-stage compression can attain about  $-100^{\circ}\text{C}$ .

The heat transfer to the evaporator can be written, according to **Figure 6**, as [10]:

$$\dot{Q}_L = \dot{m}(1-x)(h_1 - h_8) \quad (42)$$

where  $x$  is the quality ratio of vapor mass to the total refrigerant mass flow rate at the intermediate pressure of the cycle. The term  $(1-x)$  refers to the liquid mass ratio of the cycle. The net compressor power of the cycle can be evaluated as [10]:

$$\dot{W}_{net} = \dot{W}_{comp,1} + \dot{W}_{comp,2} = \dot{m}(h_4 - h_9) + \dot{m}(1-x)(h_2 - h_1) \quad (43)$$

Therefore, the COP of this system can be determined as the following [10]:

$$COP = \frac{\dot{Q}_L}{\dot{W}_{net}} = \frac{(1-x)(h_1 - h_8)}{(h_4 - h_9) + (1-x)(h_2 - h_1)} \quad (44)$$

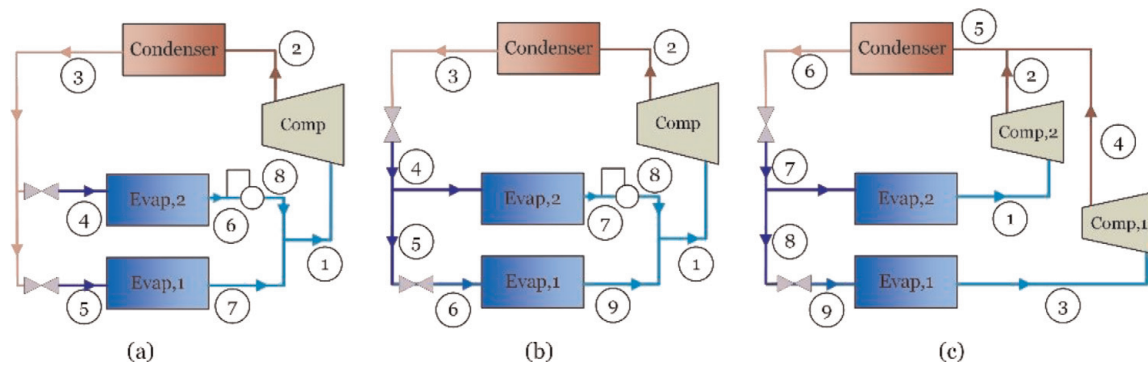
The second efficiency or the exergetic COP can be calculated as [6, 10]:

$$COP = \frac{\dot{Ex}_{Q, evap}}{\dot{W}_{net}} = \frac{\dot{Q}_L \left( \frac{T_0}{T_L} - 1 \right)}{\dot{W}_{comp,1} + \dot{W}_{comp,2}} \quad (45)$$

### 5.3 Multipurpose refrigeration systems

Multipurpose refrigeration systems are also considered as a branch of MPRS. This type of system accomplishes different refrigeration loads in one system. Therefore, a modification of VCRS can be done by using multiple evaporators at different low pressure and different refrigerant capacity. Also, this system can be operated using one compressor or multistage compressor.

There are different configurations of multipurpose refrigeration systems [2], as shown in **Figure 7**. Firstly, a system of a single compressor and individual expansion valves consists of two evaporators and single compressor with individual expansion valves for each evaporator and one condenser, as shown in **Figure 7a**. Operation under these conditions means the dropping of pressure from high-pressure evaporators through back pressure valves to ensure the compression of the vapor from the higher temperature evaporators through a pressure ratio. Secondly, a system of a single compressor with multi-expansion valves consists of two evaporators and a compressor with multiple arrangements of expansion valves, as shown in **Figure 7b**.



**Figure 7.** Multipurpose refrigeration system: (a) two evaporators with individual expansion valve, (b) two evaporators and multi-expansion valve, and (c) individual compressors and multi-expansion valve.

The only advantage of the arrangement is that the flashed vapor at the pressure of the high-temperature evaporator is not allowed to go to the lower-temperature evaporator, thus improving its efficiency. Finally, a system of individual compressors with multi-expansion valves consists of a compressor for each evaporator and multiple arrangements of expansion valves, as shown in **Figure 7c**, to reduce the total power requirement. This amounts to parallel operation of evaporators and is called sectionalizing. There may be a separate condenser for each compressor or a common condenser for the whole plant.

The heat transfer to the evaporators and the net compressor power of the multipurpose refrigeration system despite the system configuration can be evaluated as [2]:

$$\dot{Q}_{evap, total} = \sum_{i=1}^n \dot{Q}_{evap, i} = \sum_{i=1}^n \dot{m}_{evap, i} (h_{evap, ex} - h_{evap, in})_i \quad (46)$$

$$\dot{W}_{net} = \sum_{k=1}^m \dot{W}_{comp, k} = \sum_{k=1}^m \dot{m}_{comp, k} (h_{com, ex} - h_{com, in})_k \quad (47)$$

where  $i$  is the number of evaporators from 1 to  $n$ , the subscripts  $evap$ ,  $in$  and  $evap$ ,  $ex$  refer to the inlet and exit states of each evaporator  $i$ ,  $k$  is the number of compressor in the refrigeration system from 1 to  $m$ , and the subscripts  $comp$ ,  $in$  and  $comp$ ,  $ex$  refer to the inlet and exit states of each compressor  $k$ .

Therefore, the COP of this system can be determined as the following [5]:

$$COP = \frac{\dot{Q}_{evap, total}}{\dot{W}_{net}} \quad (48)$$

The second efficiency or the exergetic COP can be calculated as [6]:

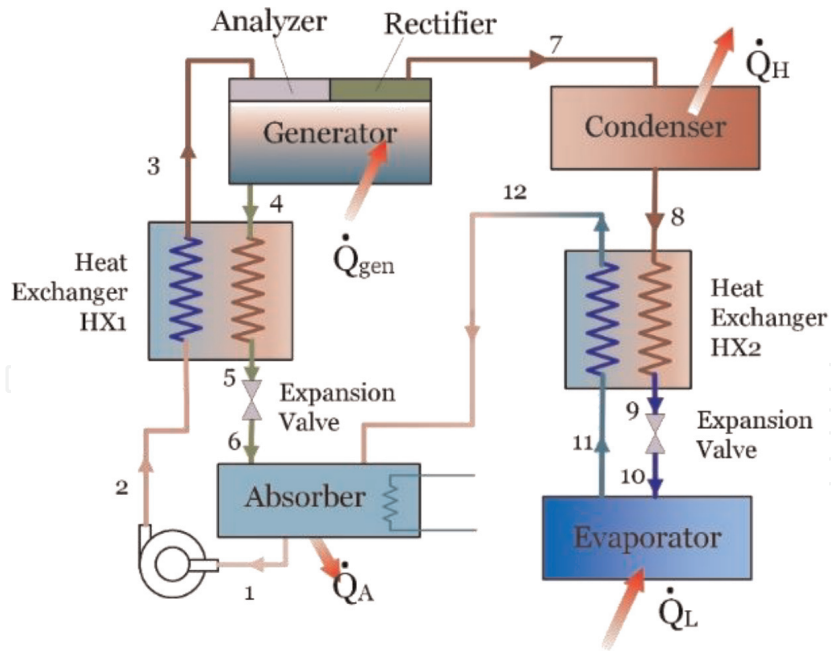
$$COP = \frac{\sum_{i=1}^n \dot{Ex}_{Q, evap}}{\dot{W}_{net}} = \frac{\sum_{i=1}^n \dot{Q}_{evap, i} \left( \frac{T_0}{T_{L, i}} - 1 \right)}{\sum_{k=1}^m \dot{W}_{comp, k}} \quad (49)$$

## 6. Absorption refrigeration system

The absorption refrigeration system (ARS) is similar to the VCRS except that the compressor of the vapor-compression system is replaced by three elements: an absorber, a solution pump, and a generator. The ABS medium is a mixture of a refrigerant and absorbent, such as ammonia-water system ( $\text{NH}_3 + \text{H}_2\text{O}$ ) and water-lithium bromide ( $\text{LiBr}_2 + \text{H}_2\text{O}$ ). The solubility of refrigerant (ammonia or lithium bromide) in the absorbent (water) is satisfactory, but the difference in boiling points is significant, which may affect the purity of vaporization. Thus, a purge unit or rectifier is used in the system. The refrigerant concentration in the mixture changes according to the pressure and temperature for each step. The ABS.

As shown in **Figure 8**, the ARS consists of a condenser, an evaporator, an absorber, a regeneration heat exchanger (HX1), heat recovery heat exchanger (HX2), a generator, two expansion valves, and a solution pump. The system includes an analyzer and a rectifier to remove the water vapor that may have formed in the generator. Thus, only ammonia vapor goes to the condenser. This system utilizes the absorbent water to release and absorb ammonia as the refrigerant. Starting from state 3, the strong solution (a high concentration of ammonia





**Figure 8.**  
*Ammonia absorption refrigeration cycle.*

refrigerant) is heated in the high-pressure generator. This produces refrigerant vapor off the solution at state 7. The hot pure ammonia vapor is cooled in the condenser at state 8 and condenses at state 9 by passing through the HX2 before entering a throttling valve into the low pressure at state 10. Then the refrigerant liquid passes through the evaporator to remove the heat from refrigerated medium and leaves at low-pressure vapor phase of state 11. The pure ammonia is heated by the HX2 to enter the absorber and mixed with the absorbent water. The weak solution (about 24% ammonia concentration) flows down from the generator at state 4 through the regeneration heat exchanger HX1 at state 5 through a throttling valve and enters the absorber at state 6. Therefore, the weak refrigerant is absorbed by the water because of the strong chemical affinity for each other. The absorber is cooled to produce a strong solution at low pressure at state 1. The strong solution is obtained and pumped by a solution pump to the generator passing through HX1, where it is again heated, and the cycle continues. Then, the water absorbs the ammonia in the absorber at the condenser temperature supplied by the circulating water or air, and hence a strong solution (about 38% ammonia concentration) occurs. For ammonia-water ARSs, the most suitable absorber is the film-type absorber because of high heat and mass transfer rates, enhanced overall performance, and large concentration rates [11].

### 6.1 Energy and exergy analysis of ammonia-water ( $\text{NH}_3\text{-H}_2\text{O}$ ) ARSs

The energy and exergy analysis for each component is presented according to **Figure 8**. The partial mass balance (PMBE) is also included to determine the concentration mass of ammonia and water in the absorber and generator. That is because the ARS has two fluids as refrigerant and absorbent and their composition at different points is different, particularly in the absorber and generator. The exergy analysis of ammonia-water ARSs is to determine the exergy destruction of each component and to determine the overall exergy efficiency based on the second law of thermodynamics. The exergy analysis (ExBE) for each component is stated below [5, 6]:

Absorber:

$$\text{EnBE} \quad \dot{m}_6 h_6 + \dot{m}_{12} h_{12} = \dot{m}_1 h_1 + \dot{Q}_{\text{Absorber}} \quad (50)$$

$$\text{PMBE} \quad \dot{m}_{ws} X_{ws} + \dot{m}_r = \dot{m}_{ss} X_{ss} \quad (51)$$

$$\text{ExBE} \quad \dot{m}_6 ex_6 + \dot{m}_{12} ex_{12} = \dot{m}_1 ex_1 + \dot{Ex}_{Q,A} + \dot{Ex}_{des,absorber} \quad (52)$$

where  $\dot{Q}_{\text{Absorber}}$  is the absorber head load in kW;  $X$  is the concentration of ammonia (refrigerant);  $\dot{m}_{ws}$  is the mass flow rate of the weak solution in kg/s, which equals to  $\dot{m}_6$ ;  $\dot{m}_{ss}$  is the mass flow rate of the strong solution in kg/s, which equals to mass flow rate exiting from the absorber at  $\dot{m}_1$ ; and  $\dot{m}_r$  is the mass flow rate of pure ammonia (refrigerant) in kg/s, which flows from the generator at state 7 to state 12;  $\dot{Ex}_{Q,A}$  is the thermal exergy rate of the absorber due to the heat transfer  $\dot{Q}_A$  to the environment, and it is calculated as  $\dot{Ex}_{Q,A} = \dot{Q}_A (1 - T_0/T_s)$ . Here, state 1 is a saturated liquid at the lowest temperature in the absorber and is determined by the temperature of the available cooling water flow or air flow.

Solution pump:

$$\text{EnBE} \quad \dot{m}_1 h_1 + \dot{W}_{\text{Pump}} = \dot{m}_2 h_2 \quad (53)$$

$$\text{ExBE} \quad \dot{m}_1 ex_1 + \dot{W}_P = \dot{m}_2 ex_2 + \dot{Ex}_{des,pump} \quad (54)$$

Regeneration heat exchanger (HX1):

$$\text{EnBE} \quad \dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5 \quad (55)$$

$$\text{ExBE} \quad \dot{m}_2 ex_2 + \dot{m}_4 ex_4 = \dot{m}_3 ex_3 + \dot{m}_5 ex_5 + \dot{Ex}_{des,HX1} \quad (56)$$

Generator:

$$\text{EnBE} \quad \dot{m}_3 h_3 + \dot{Q}_{\text{gen}} = \dot{m}_4 h_4 + \dot{m}_7 h_7 \quad (57)$$

$$\text{PMBE} \quad \dot{m}_{ws} X_{ws} + \dot{m}_r = \dot{m}_{ss} X_{ss} \quad (58)$$

$$\text{ExBE} \quad \dot{m}_3 ex_3 + \dot{Ex}_{Q,gen} = \dot{m}_4 ex_4 + \dot{m}_7 ex_7 + \dot{Ex}_{des,gen} \quad (59)$$

where  $\dot{Q}_{\text{gen}}$  is the heat input to the generator in kW;  $\dot{m}_{ws} = \dot{m}_4$  and  $\dot{m}_{ss} = \dot{m}_3$ ;  $\dot{Ex}_{Q,gen}$  is the thermal exergy rate of the generator due to the heat transfer  $\dot{Q}_{\text{gen}}$  to the environment, and it is calculated as  $\dot{Ex}_{Q,gen} = \dot{Q}_{\text{gen}} (1 - T_0/T_s)$ .

Condenser:

$$\text{EnBE} \quad \dot{m}_7 h_7 = \dot{Q}_H + \dot{m}_8 h_8 \quad (60)$$

$$\text{ExBE} \quad \dot{m}_7 ex_7 = \dot{Ex}_{Q,cond} + \dot{m}_8 ex_8 + \dot{Ex}_{des,cond} \quad (61)$$

where  $\dot{Ex}_{Q,cond}$  is the thermal exergy rate of the condenser due to the heat transfer  $\dot{Q}_H$  to warm environment and is calculated as  $\dot{Ex}_{Q,cond} = \dot{Q}_H (1 - T_0/T_s)$ .

Heat recovery heat exchanger (HX2):

$$\text{EnBE} \quad \dot{m}_8 h_8 + \dot{m}_{11} h_{11} = \dot{m}_9 h_9 + \dot{m}_{12} h_{12} \quad (62)$$

$$\text{ExBE} \quad \dot{m}_8 ex_8 + \dot{m}_{11} ex_{11} = \dot{m}_9 ex_9 + \dot{m}_{12} ex_{12} + \dot{Ex}_{des,HX2} \quad (63)$$



Expansion valves:

$$\text{EnBE} \quad \dot{m}_5 h_5 = \dot{m}_6 h_6 \Rightarrow h_5 = h_6 \quad (64)$$

$$\dot{m}_9 h_9 = \dot{m}_{10} h_{10} \Rightarrow h_9 = h_{10} \quad (65)$$

$$\text{ExBE} \quad \dot{m}_5 ex_5 = \dot{m}_6 ex_6 + \dot{Ex}_{des, EX1} \quad (66)$$

$$\dot{m}_9 ex_9 = \dot{m}_{10} ex_{10} + \dot{Ex}_{des, EX1} \quad (67)$$

Evaporator:

$$\text{EnBE} \quad \dot{m}_{10} h_{10} + \dot{Q}_L = \dot{m}_{11} h_{11} \quad (68)$$

$$\text{ExBE} \quad \dot{m}_{10} ex_{10} + \dot{Ex}_{Q, evap} = \dot{m}_{11} h_{11} + \dot{Ex}_{des, evap} \quad (69)$$

where  $\dot{Ex}_{Q, evap}$  is the thermal exergy rate of the evaporator due to the heat transfer  $\dot{Q}_L$  from refrigerated space and is calculated as  $\dot{Ex}_{Q, evap} = \dot{Q}_L (T_0/T_L - 1)$ .

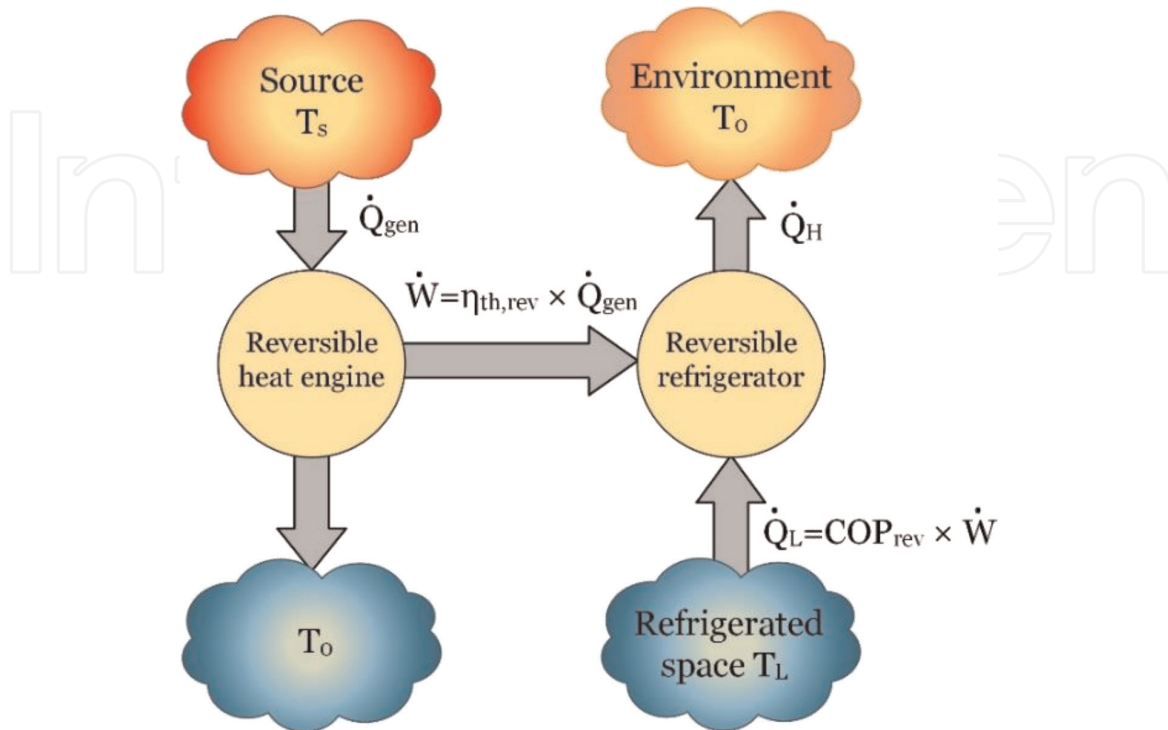
For the entire system, the overall energy balance of the complete system can be written as follows, by considering that there is negligible heat loss to the environment:

$$\dot{W}_P + \dot{Q}_{gen} + \dot{Q}_L = \dot{Q}_A + \dot{Q}_H \quad (70)$$

The COP of the system then becomes:

$$COP = \frac{\dot{Q}_L}{\dot{W}_P + \dot{Q}_{gen}} \cong \frac{\dot{Q}_L}{\dot{Q}_{gen}} \quad (71)$$

where  $\dot{W}_P$  is the pumping power requirement, and it is usually neglected in the COP calculation.



**Figure 9.**  
The maximum COP of an absorption refrigeration system.

The ARS is a heat-driven system, which requires heat pump instead or required power by a compressor. That means the ARS is a combination of a heat pump and a refrigeration cycle without a compressor. Therefore, the maximum (reversible) of an ARS can be achieved by a reversible heat engine and a reversible refrigerator, as shown in **Figure 9**. A reversible heat pump is operating by absorbing the heat from a source at and rejecting heat to an environment of to produce a work output from the heat engine. This work is defined as the reversible efficiency of the heat pump multiplied by the heat absorber from the source, which is the heat transfer from the generator in the ARS. This work output is used by the reversible refrigerator to keep a refrigerated space at  $T_L$  while rejecting heat to the environment at  $T_0$ . Therefore, the reversible COP of ARS can be obtained by the thermal efficiency of a reversible heat engine and the COP of a reversible refrigerator as in Eq. (72) [10]:

$$COP_{abs, rev} = \frac{\dot{Q}_L}{\dot{Q}_{gen}} = \frac{\dot{W}}{\dot{Q}_{gen}} \frac{\dot{Q}_L}{\dot{W}} = \eta_{th, rev} COP_{rev} = \left(1 - \frac{T_0}{T_s}\right) \left(\frac{T_L}{T_0 - T_L}\right) \quad (72)$$

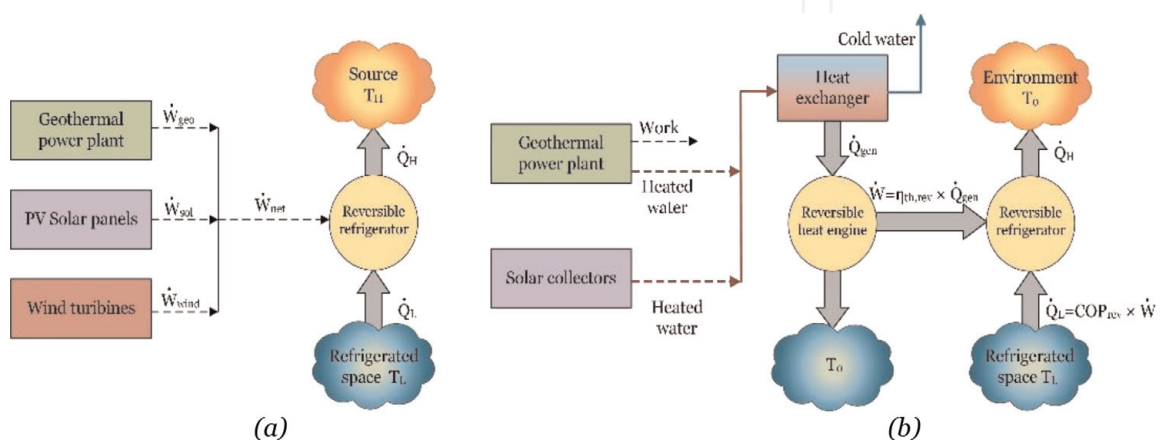
The temperature of the heat source is taken as the average temperature of geothermal water. Then the second-law efficiency of this absorption system is determined to be [10]:

$$COP_{ex, abs} = \frac{\dot{Ex}_{Q, evap}}{\dot{Ex}_{Q, gen}} = \frac{\dot{Q}_L(T_0/T_L - 1)}{\dot{Q}_{gen}(1 - T_0/T_s)} = \frac{\dot{Q}_L/\dot{Q}_{gen}}{(1 - T_0/T_s)(T_L/(T_0 - T_L))} \quad (73)$$

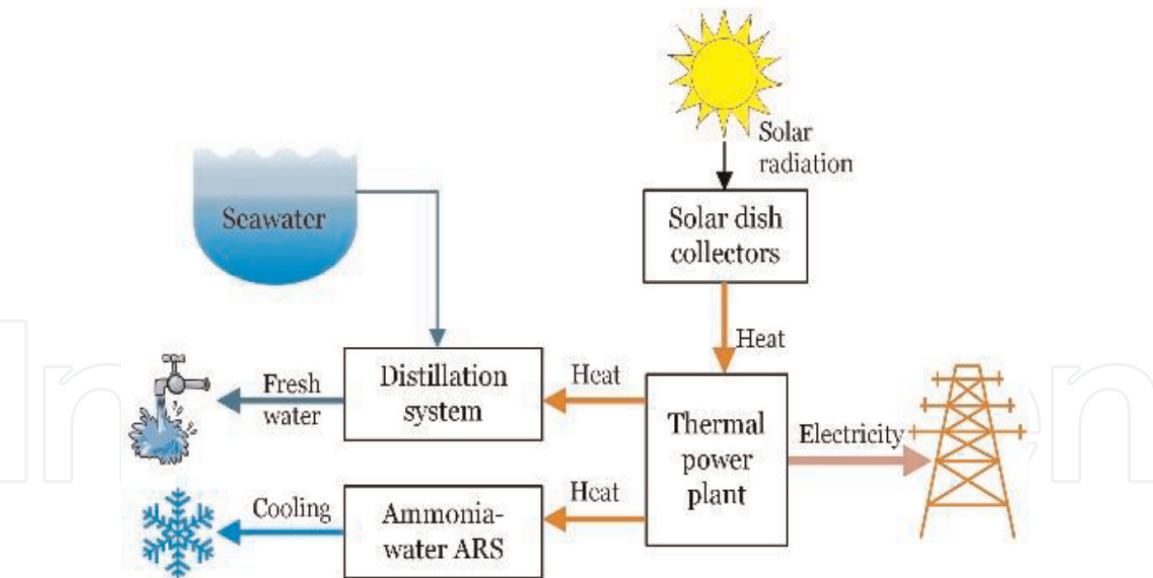
$$COP_{ex, abs} = \eta_{II, abs} = \frac{COP}{COP_{abs, rev}} \quad (74)$$

## 7. Renewable sources for refrigeration system

The refrigeration systems require an input work to release the heat from the refrigerated space to the environment, which is called as a work-driven system. The absorption refrigeration system is based on external heat transfer from an external source, which can be classified as a heat-driven system. For industrial refrigeration systems, energy demand is high and should be provided in a secure and eco-friendly approach to reduce environmental pollution. This can be executed by fossil-based fuels such as oil, natural gas, and coal, which produce substantial carbon monoxide and dioxide emissions that affect global warming and climate change.



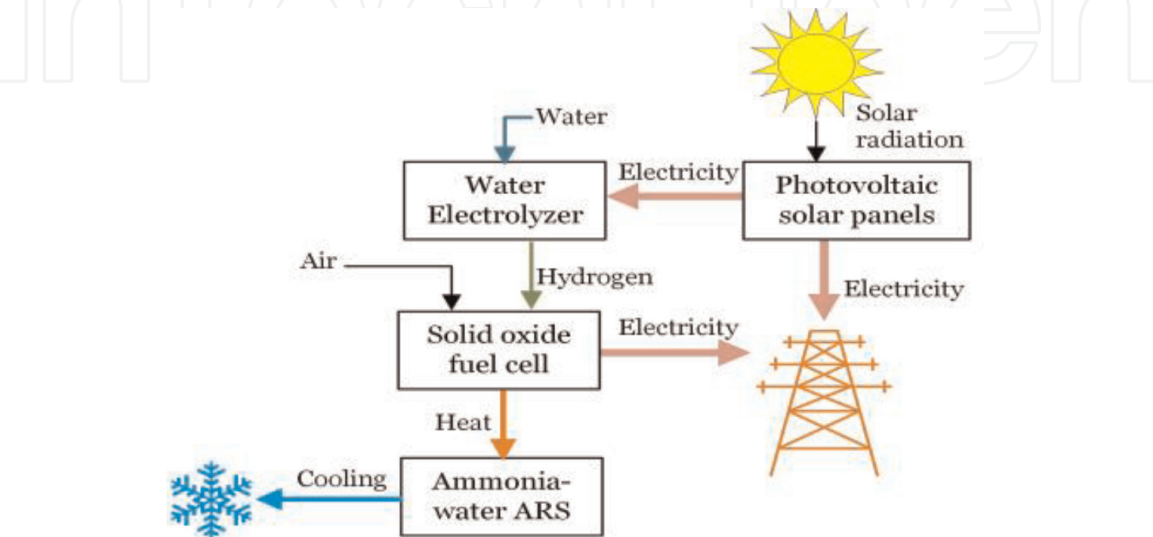
**Figure 10.** Renewable sources for refrigeration systems: (a) work-driven and (b) heat-driven source.



**Figure 11.** Schematic diagram of the integrated solar thermal power plant, absorption refrigeration system, and MED cycle (adopted from [12]).

Massive efforts point to renewable sources such as geothermal energy, solar energy, and wind energy, which promise a potential solution to provide the clean energy needed as work or heat to operate the refrigeration. Schematic diagram of **Figure 10** shows possible ways of renewable sources for work-driven and heat-driven refrigeration system.

An integrated system of a concentrated solar power plant integrated with desalination process and absorption refrigeration cycle is utilized to supply power, freshwater, and refrigeration [12]. The system, as shown in **Figure 11**, consists of concentrated solar collectors connected with steam turbine power plant, a multi-effect desalination process with a parallel feed of seawater, and a single-stage ammonia-water absorption refrigeration system. The solar collectors provide thermal energy 21,030 kW to the steam power plant to deliver an electric power of 4632 kW. The refrigeration load from the absorption cooling system is 820.8 kW. The desalination system can also provide 22.79 kg/s freshwater. This cycle has obtained overall energy and exergy efficiencies to be 80.70% and 66.05%, respectively.

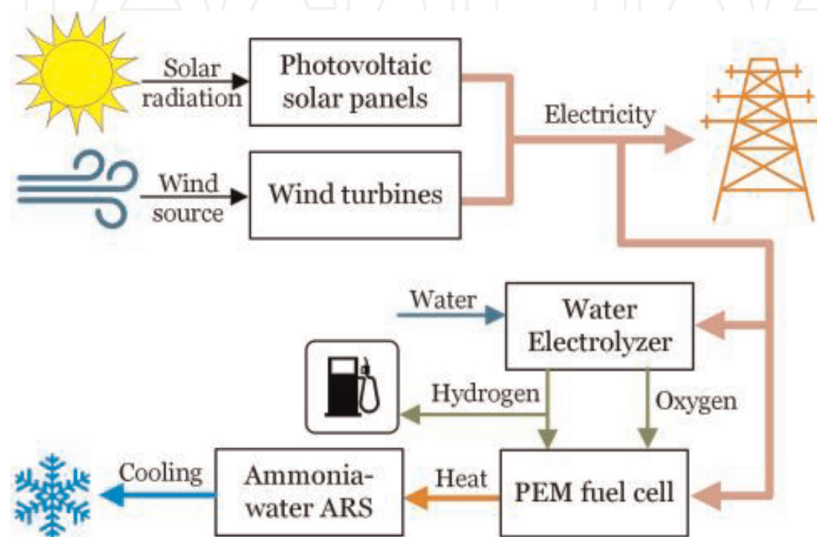


**Figure 12.** Schematic of the photovoltaic-fuel cell CHIP system for residential applications (adopted from [13]).

Another example, a small-scale system, is designed to provide an electrical load to residential buildings [13]. This system utilizes, as shown in **Figure 12**, photovoltaic solar system (PV) to provide electrical power. This electric power is used for a water electrolyzer system to split the water electrochemically to produce hydrogen and oxygen gases. The hydrogen gas enters high-temperature solid oxide fuel cells (SOFC) to produce electricity and heat. The heat is transferred to an absorption cooling system by heat recovery generator. The PV system may generate excess electricity more than the demand during off-peak hours. This system is designed for a detached house in Toronto city, Canada. The PV solar system delivers maximum power of 3.35 kW. The water electrolyzer can produce 0.792 and 0.538 kg/day of gaseous hydrogen in summer and winter seasons. The SOFC fuel cell supplies 8.43 kWh per day in summer season. The maximum energy and exergy efficiencies of the photovoltaic system are 17 and 18.3%, respectively, while the maximum total energy and exergy efficiencies are obtained to be 55.7 and 49.0%, respectively.

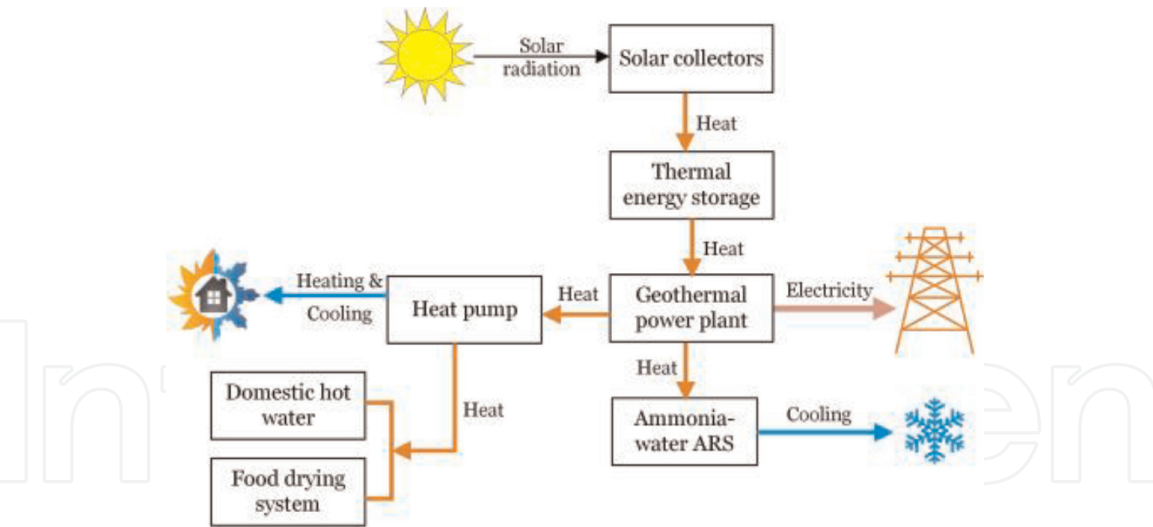
In a similar study, a hybrid renewable system was designed to produce electricity and clean fuel such as hydrogen gas and provide cooling for a residential building in two locations Egypt and Saudi Arabia in summer season [14]. The cooling loads for a house are 18.06 and 19.3 kW in Egypt and Saudi Arabia, respectively. This system, as shown in **Figure 13**, depends on the photovoltaic solar system and wind turbines to provide excess electricity more than the electric grid. The excess electricity is delivered to a water electrolyzer to produce pure oxygen and hydrogen gases stored in tanks for clean fueling services. Part of the hydrogen gas is used for a proton-exchange membrane (PEM) fuel cell that can produce heat and electricity through an electrochemical process without any mechanical parts. The heat generated from the fuel cell can be utilized by a generator of an ammonia-water ARS to provide cooling. The hybrid renewable system can operate in a significant performance with water mass flow rate of 1.8 kg/s to produce hydrogen with a mass flow rate of 0.2 kg/s and ammonia mass flow rate of about 0.2 kg/s to produce cooling load between 40 and 120 kW more than the design cooling load of one house. The energy and exergy efficiencies are obtained to be about 67 and 68%, respectively. Therefore, this hybrid system can be sufficient for more than one house.

A multigeneration system is designed by [15] and powered by geothermal energy assisted with solar energy to produce five outputs: heating air for residential building, hot domestic water, drying food, refrigeration for industry, and electricity. This multigeneration system, as shown in **Figure 14**, consists of a heat pump

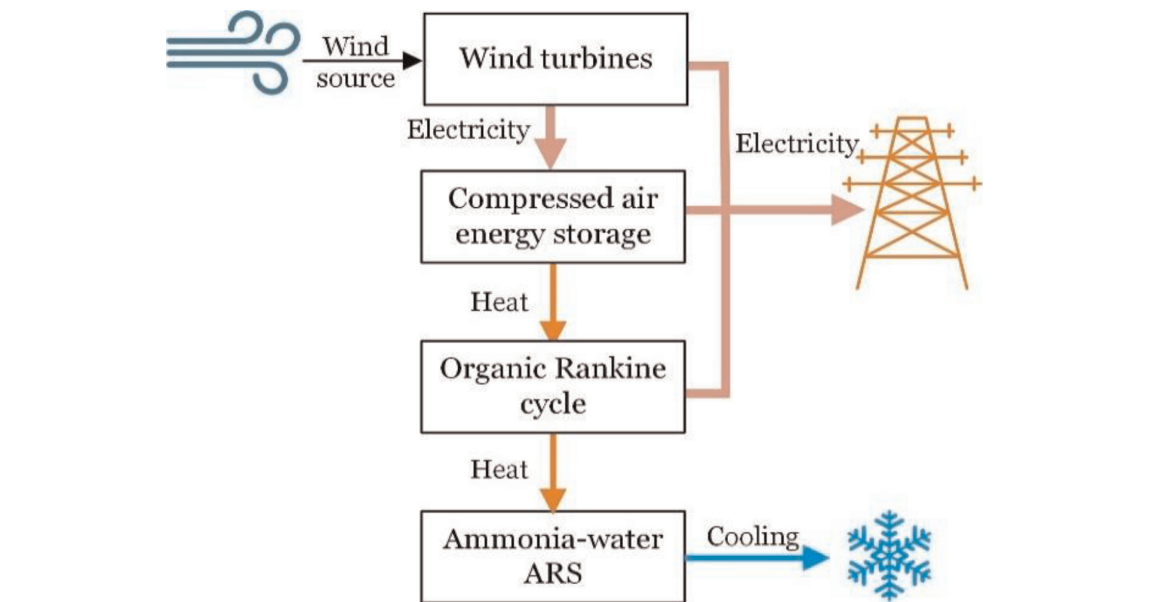


**Figure 13.**  
Schematic diagram of a hybrid renewable system (adopted from [14]).





**Figure 14.** Schematic diagram of the multigeneration system powered by the solar and geothermal energy (adopted from [15]).



**Figure 15.** Schematic diagram of gas power cycle with wind turbine, CAES, ORC, and ARS (adopted from [16]).

system, a single flash geothermal cycle, an absorption cooling system, thermal energy storage connected with auxiliary steam turbine and concentrated solar collectors, hot water system, and drying system. The system has achieved overall energy and exergy efficiencies to be 69.6 and 42.8%, respectively. The first and second steam turbines have the power of 10,043 and 9886 kW. The  $COP$  and  $COP_{ex}$  are 0.678 and 0.253 for the absorption cooling system and 2.029 and 0.1826 for heat pump system, respectively. The refrigeration load is 1787 kW. The overall energy and exergy efficiencies for the whole system are 69.6 and 42.8%, respectively.

A wind system is combined with a refrigeration system, as shown in **Figure 15**. Wind energy is coupled with compressed air energy storage (CAES) systems to store wind energy for long-term usage [16]. The integrated system consists of a combined gas power cycle, including compressors, intercooling heat exchangers, and gas turbine, an organic Rankine power cycle (ORC), and an absorption refrigeration system (ARS). The system objective is to provide electricity, domestic hot water, and cooling load. The system can generate electricity of 33.67 kW provided

by wind turbines (83.24 kWh) and fuel combustion (258.97 kWh), cooling load of 2.56 kW, and mass flow rate of hot water of 1.82 ton per day hot. The energy efficiency of the system is achieved to be 53.94%.

## 8. Conclusion


The refrigeration systems are applied in our life for preserving food, cooling air, and other industrial applications. Most refrigeration systems require external power or external heat to release the heat from the refrigerated space. Many industrial applications involve large cooling energy, which can be operated by multi-pressure refrigeration system, which requires a large amount of external power. The chapter has presented some applications with renewable sources to replace the fossil fuel-driven energy with an environmentally friendly energy source such as geothermal, solar, and wind energy so-called hybrid or integrated systems. In addition to cooling load, the hybrid systems can produce electricity, heating load, and clean fuel such as hydrogen fuel. The absorption refrigeration system is mostly-combined with hybrid system to use the heating load from solar or geothermal energy to produce cooling load.

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