3 An Overview of Refrigeration Cycles

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INTRODUCTION

Freezing is the process of removing heat to produce and maintain temperatures below initial freezing point [1]. In the food industry, freezing is the most popular long-term preservation method for food products [2]. Besides convenience to consumers, frozen foods are also popular because they continue to demonstrate a good food safety record, since freezing can effectively reduce the activity of microorganisms and enzymes, thus retarding deterioration [1,3]. In addition, crystallization of water reduces the amount of liquid water in food items and inhibits microbial growth [4].

The food-freezing process consists of three steps: (a) cooling to remove sensible heat and to reduce the product temperature to the freezing point; (b) removing the product’s latent heat of fusion and changing the water to ice crystals; and (c) continued cooling below the freezing point to reduce the product temperature to the desired frozen storage temperature [2,5,6]. The longest part of the process is the removal of the latent heat of fusion as water turns to ice crystals, which determines the freezing rate [1,3]. Generally speaking, a high freezing rate can lead to better frozen-food quality, due to the formation of small ice crystals. In recent years, extensive research has been conducted to accelerate the freezing process and to form small and evenly distributed ice crystals throughout a frozen-food product [7–10].

The freezing systems used in the food industry are generally operated based on some refrigeration cycles [11,12]. At present, the refrigeration market is dominated by electricity-powered mechanical vapor-compression units. As electricity generation produces large amounts of carbon dioxide (CO₂), contributing to global warming, many governments in the world have committed themselves to reduce the emission of greenhouse gases. A solution to the global warming problem is to develop refrigeration systems powered by waste thermal energy or solar energy [13–19]. Utilization of low-grade thermal energy has been a research topic for many decades since the energy is widely available from sources such as industrial processes, flat plate solar collectors, and exhausts from automobiles. As refrigeration is one of the economically feasible and environmentally friendly applications for harnessing low-grade thermal energy, alternative or novel refrigeration cycles powered by thermal energies have been developed to help save significant amounts of electrical energy and reduce energy consumption [20–23].

FUNDAMENTALS

The description and analysis of refrigeration cycles require knowledge of thermodynamics and heat transfer. Therefore, it is important to review some relevant fundamental principles that are important for the calculation of refrigeration cycles [24–27].

TEMPERATURE

Temperature is the most common term used with regard to food freezing. The temperature of a substance is an indication of the ability of the substance to exchange energy with another substance that is in contact with it. The temperature scale used in the food industry is normally the Celsius scale, which has two reference points that are the freezing point of water (0°C) and the boiling point of water (100°C) at atmospheric pressure. In the Celsius scale, \( t \) (°C) is normally used as the symbol for temperature. However, in calculating refrigeration cycle, the Kelvin scale or the absolute temperature scale is also used. The Kelvin scale uses the absolute zero (0 K or −273.15°C) as its reference point, and it has the same degree intervals as the Celsius scale. In the Kelvin scale, \( T \) (K) is normally used as the symbol for temperature. Therefore, temperature differences in the Celsius scale can also be stated in the Kelvin scale. In the Kelvin scale, the freezing and boiling points of water at atmospheric pressure are 273.15 K and 373.15 K, respectively.


**Heat**

In the food industry, the term “heat” is normally used to refer to the thermal energy. A change in this energy may result in a change in temperature or a change between the solid, liquid, and gaseous states. Freezing is a process of removing heat and thus reducing temperature.

Heat is a form of the internal energy of a substance and is related to the molecular structure and the degree of molecular activity. Therefore, heat consists of the kinetic and potential energies of molecules. Depending on the degree of molecular activity, heat can be classified as sensible heat and latent heat. Sensible heat is the sum of the kinetic energies of molecules; a change in sensible heat thus causes a change in temperature. On the other hand, latent heat is associated with overcoming molecular forces that bind molecules to each other and the breaking away of molecules; a change in latent heat does not cause any change in temperature but causes a change in phase (solid to liquid, liquid to gas, or vice versa).

**Enthalpy and Specific Heat**

Enthalpy $h$ is often used to calculate the change in heat in refrigeration cycles. It is defined as the sum of internal energy and flow work and is measured as kilojoules per kilogram (kJ/kg). If only steady flow is involved in a process, the flow work will not change significantly and the difference in enthalpy will be the quantity of heat gained or lost. Therefore, enthalpy covers both sensible heat and latent heat. As the value of enthalpy is always based on some arbitrarily chosen datum plane, enthalpy difference is normally used in calculations.

If a change in enthalpy can be sensed by a change in temperature, this change is expressed as the specific heat; that is, the specific heat is the change in enthalpy per degree of temperature. In other words, the specific heat of a substance is the quantity of energy required to raise the temperature of a unit mass by one degree, and is measured as kilojoules per kilogram Kelvin [kJ/(kg K)]. There are two common specific heats, one for constant volume process $c_v$, and the other for constant pressure process $c_p$. As the refrigeration process normally occurs at a constant pressure, $c_p$ is used. If a change in enthalpy does not cause any change in temperature, but a change in phase, this change is the latent heat [26,27].

**Entropy**

Entropy $s$ of a unit mass of a substance at any given condition is an expression of the total energy transferred to the material per degree to bring the substance to that condition from some arbitrarily chosen datum plane [26], and is measured as kilojoules per kilogram Kelvin [kJ/(kg K)].

In a compression or expansion process, if a gas or vapor undergoes such a process without friction and without adding or removing heat, the entropy of the substance remains constant; this is defined as the isentropic process. If a compression or expansion process is isentropic, the change in enthalpy represents the amount of work per unit mass involved in the process [25].

**Energy Conservation for Steady Flow**

The principle of energy conservation applies to every process. The principle simply states that energy cannot be created or destroyed, and the total amount of energy remains constant. Therefore, for a system in a refrigeration cycle, energy conservation can be expressed as the rate of energy with the flow stream entering the system plus the rate of heat added minus the rate of work performed and minus the rate of energy with the flow stream leaving the system which equals the rate of energy change in the system [24]. This expression can be described mathematically as

\[
m \left( h_{in} + \frac{V_{in}^2}{2} + g z_{in} \right) + Q - W - m \left( h_{out} + \frac{V_{out}^2}{2} + g z_{out} \right) = \frac{dE}{d\theta} \]  

(3.1)
In most of refrigeration systems, as the mass flow rate remains almost constant, the flow can be assumed to be steady flow; therefore, the term describing the rate of energy change in Equation 3.1 can be omitted. Furthermore, the changes in potential and kinetic energies of the flow are very small as compared with the change in enthalpy, heat transferred, or work done; their effects can be neglected. Hence, for a system in a refrigeration cycle, Equation 3.1 can be rewritten as

$$Q - W = m\Delta h = m(h_{\text{out}} - h_{\text{in}})$$ (3.2)

For a system without the involvement of work such as a condenser, evaporator, or heat exchanger, the heat transferred to or from these systems can be calculated by the change in enthalpy multiplied by the mass flow rate:

$$Q = m(h_{\text{out}} - h_{\text{in}})$$ (3.3)

On the other hand, for a system on which work is done by a compressor or a pump, the amount of heat transferred is negligible; therefore, Equation 3.2 can be simplified as

$$W = m(h_{\text{in}} - h_{\text{out}}) = m(P_{\text{in}} - P_{\text{out}})\nu$$ (3.4)

**Heat Transfer**

Heat transfers from a high-temperature body to a low-temperature body. The transfer of heat occurs by three different methods: conduction, convection, and radiation. In refrigeration systems, only heat conduction and heat convection generally occur.

Conduction transfers heat through a continuous mass or from one body touching the other. It involves the transfer of energy from the more energetic molecules of a substance to the adjacent less energetic ones due to the interactions between the molecules [28–30]. Fourier’s law is used to describe heat conduction:

$$Q = -kA \frac{dT}{dx}$$ (3.5)

Equation 3.5 indicates that heat conduction takes place in a direction of decreasing temperature, and the rate of heat conduction in a direction is proportional to the temperature gradient in that direction. The ability to conduct heat in a material is related to the characteristics of the material itself. This ability is defined as the thermal conductivity $k$ of the material and is measured as kilowatt per meter Kelvin [kW/(m K)] [31–33].

Convection is the combined effects of heat conduction and fluid flow. It transfers energy between a solid surface and the adjacent liquid or gas which is in motion. Depending on the flow velocity, convection can be free (or natural) or forced. In free convection, the fluid flow is caused by buoyancy forces that are induced by density differences due to the variation of temperature in the fluid. In contrast, in forced convection, the fluid is forced to flow by external means such as a pump or fan. Newton’s law of cooling is used to determine the rate of convection heat transfer:

$$Q = h_A (T_{\text{sur}} - T_i)$$ (3.6)

where $h$ is an experimentally determined parameter defined as the convection heat transfer coefficient measured as kilowatt per square meter Kelvin [kW/(m² K)], and its value depends on many relevant factors such as the nature of fluid flow, properties of the fluid, and surface geometry [1,34,35].

**Phase Change Processes**

The refrigerants used in most cooling systems pass between liquid and vapor states in refrigeration cycles. These refrigerants behave similarly during the changing states. Figure 3.1 shows the
phase-change process using pure water at atmospheric pressure as an example [24,27]. At a temperature below 0°C, water exists in the solid state as ice. If heat (sensible heat) is added, the temperature of the ice begins to rise until it reaches the melting point (or freezing point) at 0°C. At the melting point, if heat is further added, the ice begins to melt. The heat required to melt the ice is the latent heat which is 334 kJ/kg. During the melting process, a mixture of ice and liquid water coexists. The melting process continues at the constant melting temperature until the ice is completely changed to liquid water, which is also termed as compressed or subcooled liquid. If heat is further added, the temperature of the subcooled water starts to rise above the melting point until it reaches boiling point at 100°C (the liquid that is about to vaporize is called saturated liquid). The sensible heat required to raise the temperature of the subcooled water from 0°C to 100°C is 419 kJ/kg. If heat is further added, the liquid water begins to boil, and the boiling process occurs at a constant temperature of 100°C and latent heat must be supplied. The latent heat of boiling is 2257 kJ/kg. During boiling, a mixture of saturated liquid and saturated vapor coexists until the saturated liquid is completely changed to saturated vapor (the vapor that is about to condense is termed as saturated vapor). Then, if further heat is added, the temperature of the vapor rises above 100°C, and this vapor is defined as superheated vapor as it is not about to condense.

If the pressure in the above process is above or below atmospheric pressure, the melting and boiling points are different. With an increase in pressure, the boiling temperature increases; however, the temperature at which liquid water begins to freeze decreases.

During the phase-change process from liquid to vapor, if the refrigerant is in a saturated state (such as saturated liquid or saturated vapor), saturation pressure \( P_s \), enthalpy \( h_s \), or entropy \( s_s \) is a function of saturation temperature \( T_s \) only, and therefore if \( T_s \) is known, \( P_s, h_s, \) and \( s_s \) can be determined by the following functions:

\[
P_s = P(T_s), \quad h_s = h(T_s), \quad s_s = s(T_s)
\]  \( 3.7 \)

If the refrigerant is in any other state (such as subcooled liquid, mixture of saturated liquid and saturated vapor, or superheated vapor), enthalpy \( h \) or entropy \( s \) is a function of temperature \( T \) and pressure \( P \), and vice versa:

\[
T = T(P, h), \quad h = h(T, P), \quad P = P(T, h), \quad s = s(T, P)
\]  \( 3.8 \)

The actual expressions of Equations 3.7 and 3.8 depend on the refrigerant used. These expressions are normally a set of polynomial equations that are available from literature or can be obtained by curve-fitting to published thermodynamic data [36].

![Schematic showing the relation between temperature and enthalpy during the phase-change process of water (H2O).](image-url)
Multicomponent solution is needed for the operation of the absorption refrigeration cycle. Depending on the cooling requirement, various solutions are available [37–41]; however, the most commonly used ones are the water–lithium bromide (H₂O–LiBr) solution [17] and the ammonia–water (NH₃–H₂O) solution [13,15]. Figure 3.2 shows an equilibrium condition using the H₂O–LiBr solution as an example [25]. LiBr is a solid salt crystal that absorbs water vapor to become a liquid solution when it comes into contact with the vapor. The pressure of the water vapor in the solution vessel is a function of the temperature and concentration of the H₂O–LiBr solution. Therefore, many different combinations of temperatures and concentrations of solution can provide the same vapor pressure. If pure water is contained in another vessel, pressure of the water vapor in this vessel is a function of temperature of the pure water only. If these two vessels are connected, equilibrium will be established between the two vessels; in other words, the two vessels will have the same water vapor pressure. By changing the temperature in the vessel containing pure water, the water vapor pressure in the two connected vessels will be changed, and therefore the solution concentration can be altered depending on the solution temperature.

Obviously, in the multicomponent solution, the solution concentration \( X \) is a function of temperature \( T \) and vapor pressure \( P \) of the solution; therefore if any of the two variables are known, the third variable can be obtained:

\[
P = P(T,X), \quad T = T(P,X), \quad X = X(P,T)
\]

(3.9)

As long as the state of the solution is determined, enthalpy \( h \) or entropy \( s \) of the solution can be calculated. For example, if temperature \( T \) and concentration \( X \) of the solution are known, its enthalpy or entropy can be determined, and vice versa:

\[
h = h(T,X), \quad s = s(T,X), \quad T = T(X,h), \quad X = X(T,h)
\]

(3.10)

Again, the actual expressions of Equations 3.9 and 3.10 depend on the solution used. These expressions are available from the literature or can be obtained by curve-fitting to published thermodynamic data [13,15,17].

**Coefficient of Performance**

The efficiency of a refrigeration cycle is expressed in terms of the coefficient of performance (COP). As the purpose of refrigeration cycles is to remove heat from the refrigerated area, in order to accomplish this purpose, the cycle requires an energy input as either heat or work, depending on the operation of the actual cycle. Therefore, the COP is defined as

\[
\text{COP} = \frac{\text{desired output}}{\text{required input}} = \frac{Q_e}{E_{\text{net, in}}}
\]

(3.11)
where $Q_e$ is the useful refrigeration, and $E_{\text{net, in}}$ is the net energy input in heat or work form. These two terms must be in the same units so that COP is dimensionless. The COP value can be greater than unity, which indicates that the amount of heat removed from the refrigerated area can be greater than the amount of energy input.

**VAPOR COMPRESSION CYCLE**

The vapor compression cycle is the most frequently used refrigeration cycle in the food industry. The cycle mainly consists of four main components: a compressor, a condenser, an expansion valve, and an evaporator. The evaporator and condenser can be easily identified in a household refrigerator. The freezer compartment used to store frozen foods serves as the evaporator as the heat from the foods is removed by the refrigerant in the evaporator. The condenser is located behind the refrigerator where the coils dissipate heat to the kitchen. Simulation has been widely used for performance prediction and optimum design of the cycle. A recent review has detailed some developing simulation techniques for vapor compression cycles [42].

**CARNOT REFRIGERATION CYCLE AND ITS MODIFICATION**

The Carnot refrigeration cycle is the most efficient cycle and therefore is ideal. No other refrigeration cycles perform better than the Carnot cycle [24]. The cycle 1’-2'-3’-4’ shown in Figure 3.3 is the Carnot cycle if there is no temperature difference between $T_e$ and the load and no temperature difference between $T_c$ and the ambient. Therefore, the Carnot cycle consists of the following thermodynamically reversible processes: adiabatic compression 1’-2’, isothermal rejection of heat 2’-3’, adiabatic expansion 3’-4’, and isothermal addition of heat 4’-1’ [25]. Among them, process 4’-1’ is the refrigeration step that is the ultimate goal of the cycle as it removes heat from the load; all the other processes serve to achieve this goal by discharging the removed heat to the ambient.

In the Carnot cycle in Figure 3.3, the desired output of the cycle is the useful refrigeration which is the area beneath line 4’-1’, and the desired input to the cycle is the net work which is the area enclosed in rectangle 1’-2’-3’-4’; therefore, the COP is the ratio of the first area over the second area.

The Carnot cycle assumes no resistance in heat transfer with the load and with the ambient; in reality, no heat transfer without temperature difference can occur as the temperature difference is the driving force for heat flow. Therefore, $T_e$ should be lower than the load temperature so that the heat from the load can be transferred to the refrigerant, and $T_c$ should be higher than the ambient temperature in order for the heat from the refrigerant to be rejected to the ambient. This modification is illustrated in Figure 3.3 showing the comparison of the ideal Carnot cycle 1’-2’-3’-4’ with the modified cycle 1-2-3-4. Obviously, the modified cycle has lower COP as the rectangular area enclosed by 2’-2-3’

![FIGURE 3.3]({{ site.root }}/uploads/fig3.3.png) Carnot refrigeration cycle and its modification.
STANDARD VAPOR COMPRESSION CYCLE

The refrigeration cycle in Figure 3.3 is more often shown in the pressure–enthalpy chart, as such a chart can give information on the liquid and vapor states of the refrigerant. Figure 3.4 illustrates the standard vapor compression cycle in the pressure–enthalpy chart [43]. The flow diagram in Figure 3.4a indicates that the refrigerant vapor (1) from the evaporator is compressed by the compressor to high-pressure vapor (2), which is then condensed in the condenser to liquid. The heat of condensation is rejected to the ambient. The liquid (3) leaves the condenser, then passes through an expansion valve to undergo a pressure reduction, and finally flows into the evaporator to evaporate. The latent heat of evaporation needed is supplied by the cooling load, thus generating refrigeration. The evaporated vapor (4) leaves the evaporator and flows into the compressor to complete the cycle. This process is clearly shown in Figure 3.4b.

The standard vapor compression cycle is different from the Carnot refrigeration cycle. First, the refrigerant (1) entering the compressor is saturated vapor; after compression, the vapor becomes superheated vapor (2). This compression process is assumed to occur at constant entropy (isentropic). Second, the liquid (3) from the condenser passing through the expansion process has the same value of enthalpy after it exists to become subcooled liquid (4). This means that the expansion process (3–4) is a constant-enthalpy (isenthalpic) process and its entropy increases instead rather than the constant-entropy process as required in the Carnot cycle.

Therefore, referring to Figure 3.4b, the standard vapor compression cycle consists of the following processes: reversible and adiabatic compression from saturated vapor to the condenser

FIGURE 3.4 The standard vapor compression cycle represented as a (a) flow diagram and (b) pressure–enthalpy diagram.
pressure (1–2), desuperheating and condensation of the refrigerant and reversibly rejecting heat at the condenser pressure (2–3), irreversible expansion at constant enthalpy from saturated liquid to the evaporator pressure (3–4), and reversibly adding heat at the evaporator pressure to cause evaporation to saturated vapor (4–1) [25,43].

**Cycle Analysis**

In order to calculate the COP for the standard vapor compression cycle, thermodynamic analysis should be performed. Referring to Figure 3.4b, the cycle is characterized by the condenser and evaporator temperatures. Therefore, the refrigerant properties at states (1) and (3) can be specified as follows:

\[ T_1 = T_c, \quad P_1 = P(T_c), \quad h_1 = h(T_c), \quad s_1 = s(T_c) \]  
\[ T_3 = T_c, \quad P_3 = P(T_c), \quad h_3 = h(T_c) \]  

(3.12)  

(3.13)

The condensed fluid emerging from the condenser undergoes a pressure reduction via the expansion valve; hence,

\[ P_4 = P_1, \quad h_4 = h_3, \quad T_4 = T(P_4, h_4) \]  

(3.14)

The vapor from the evaporator is compressed to the condenser pressure by the compressor before entering the condenser. This compression process is normally assumed to be isentropic; therefore,

\[ P_2 = P_3, \quad s_2 = s_1, \quad T_2 = T(P_2, s_2), \quad h_2 = h(T_2, P_2) \]  

(3.15)

The energy balance across the evaporator and compressor to calculate COP is determined:

\[ Q_e = m(h_1 - h_4), \quad W_c = m(h_2 - h_1) \]  

(3.16)

Finally, the COP is calculated:

\[ \text{COP} = \frac{Q_e}{W_c} \]  

(3.17)

**Absorption Refrigeration Cycle**

The absorption refrigeration cycle has recently attracted much attention in research because of the possibility of using waste thermal energy or renewable energies as the power source, thus reducing the demand for electricity supply. For example, recent studies demonstrated that solar-powered absorption refrigeration technologies could be used for refrigeration and ice-making; however, due to the relatively low COP, currently most of the applications are in the stage of demonstration and prototyping [44]. The H₂O–LiBr and NH₃–H₂O systems are the most common absorption systems where the components are given as refrigerant–absorbent [13,15,17]. In order to improve the performance of absorption systems, new refrigerant–absorbent pairs have been developed [37,38,45,46]. These pairs include (H₂O–NH₃)–LiBr, CH₃OH–(LiBr–ZnBr₂), H₂O–(LiNO₃–KNO₃–NaNO₃) [39], H₂O–glycerol [40], H₂O–LiCl [41], LiNO₃–NH₃ [38], and H₂O–EMISE (i.e., 1-ethyl-3-methylimidazolium ethyl sulfate) [47].

In the H₂O–LiBr system, water is used as refrigerant; therefore, the application of such a system is limited by the freezing point of water, and H₂O–LiBr systems are normally used in the air-conditioning industry [17]. On the other hand, in the NH₃–H₂O systems, as ammonia is used as the refrigerant, these systems can be applied to food refrigeration or ice-making [13,48]. Therefore, only the NH₃–H₂O absorption cycle is discussed.


**Cycle Description**

Generally speaking, an absorption cycle consists of a generator, condenser, evaporator, absorber, pump, heat exchanger, and two expansive valves. However, for the NH$_3$–H$_2$O absorption cycle, two special components, that is, a rectifier and a dephlegmator, are occasionally needed as water is volatile. The NH$_3$–H$_2$O absorption cycle is shown in Figure 3.5 [17]. When ammonia is evaporated off the generator, it contains some water vapor, and if the mixture of ammonia and water flows into the evaporator, the water elevates the evaporating temperature; in the meantime, the water can also freeze along the pipelines. Therefore, this water must be removed as completely as possible. With the rectifier and dephlegmator, as illustrated in Figure 3.5, the vapor driven off at the generator first flows countercurrent to the incoming solution in the rectifier, the solution then passes through the dephlegmator and condenses some water-rich liquid that drains back to the rectifier. Therefore, only a small amount of water vapor may eventually escape the dephlegmator and flow from the evaporator to the absorber.

The operating principle of the NH$_3$–H$_2$O absorption cycle, as shown in Figure 3.5, is as follows. High-pressure refrigerant vapor (1) generated by the generator condenses into liquid (2) in the condenser, and the heat of condensation is rejected to the environment. The condensed liquid then enters the evaporator (3) to evaporate through a throttling valve that is used to maintain the pressure difference between the condenser and the evaporator. The heat required for evaporation is provided by the cooling load. Vapor (4) evaporated from the evaporator is absorbed by the liquid strong solution (10) coming from the generator in the absorber. The heat of absorption is rejected to the environment. The pump receives low-pressure liquid weak solution (5) from the absorber, elevates the pressure of the weak solution (6), and delivers (7) to the generator. Weak solution (strong solution) implies that the ability of the solution to absorb the refrigerant vapor is weak (strong). In the generator, heat from a high-temperature source drives off the refrigerant vapor (1) in the weak solution. The liquid strong solution (8) then returns to the absorber (9) through a throttling valve (10). The function of the throttling valve is to provide a pressure drop to maintain the pressure difference between the generator and the absorber. Therefore, for an absorption cycle, the generator

![FIGURE 3.5 The schematic of the ammonia–water (NH$_3$–H$_2$O) absorption cycle.](#)
and condenser operate at the same high-pressure level while the evaporator and absorber maintain the same low-pressure level. In order to improve cycle performance, a solution heat exchanger is normally added to the cycle as shown in Figure 3.5. This solution heat exchanger is an energy-saving device that is essential to the successful operation of the cycle.

The heat flow pattern to and from the absorption cycle is that a high temperature is required at the generator while a low temperature from the substance being refrigerated enters the evaporator. The rejection of heat from the cycle at the absorber and condenser is at temperatures higher than the atmosphere so that the heat can be rejected to the environment.

**Cycle Analysis**

As shown in Figure 3.5, the operation of the absorption cycle is characterized by the temperatures at the generator, condenser, absorber, and evaporator, and the refrigerant mass flow through the evaporator or the required refrigerating load. Therefore, the cycle can be thermodynamically analyzed as follows [13,15].

The operating pressures in the above components are determined by the saturated liquid and vapor concentrations in the condenser and evaporator, respectively, and their temperatures, that is,

\[ P_c = P_e = P(T_2, X_2), \quad P_e = P_a = P(T_4, X_4) \] (3.18)

For the strong solution at state (8) from the generator:

\[ T_8 = T_g, \quad P_8 = P_g, \quad X_8 = X(T_8, P_8), \quad h_8 = h(T_8, X_8) \] (3.19)

\[ T_1 = T_g, \quad P_1 = P_g, \quad h_1 = h(T_1, Y_1) \] (3.20)

For the weak solution at state (5) from the absorber:

\[ T_5 = T_a, \quad P_5 = P_a, \quad X_5 = X(T_5, P_5), \quad h_5 = h(T_5, X_5) \] (3.21)

Applying the mass conservation principle at the generator yields:

\[ X_7 = X_5, \quad m_7X_7 = m_8X_8 + m_1Y_1, \quad m_1 = m_7 - m_8 \] (3.22)

Therefore, the mass flow rates of the strong and weak solutions at states (8) and (7), respectively, can be found as

\[ m_8 = \frac{X_7 - Y_1}{X_8 - X_7} m_1, \quad m_5 = m_7 = \frac{X_8 - Y_1}{X_8 - X_7} m_1 \] (3.23)

The weak solution from the absorber is pumped to the solution heat exchanger. As a result, the enthalpy at (6) is increased:

\[ T_6 = T_5, \quad P_6 = P_g, \quad X_6 = X_5, \quad u_6 = u(T_6, X_6), \quad h_6 = h_5 + (P_6 - P_5)u_6 \] (3.24)

If the effectiveness of the solution heat exchanger is \( \eta \), the fluid at states (9) and (7) is derived by using the above values as follows:

\[ T_9 = \eta T_6 + (1 - \eta)T_5, \quad X_9 = X_8, \quad h_9 = h(T_9, X_9) \] (3.25)

\[ X_7 = X_6, \quad m_6 = m_5, \quad h_7 = h_6 + \frac{m_8}{m_6} (h_8 - h_9), \quad T_7 = T(X_7, h_7) \] (3.26)
Through the expansion valve, the fluid pressure is reduced from high pressure at (9) to low pressure at (10); therefore,

\[ X_{10} = X_9, \quad h_{10} = h_9, \quad T_{10} = T(X_{10}, h_{10}) \]  

(3.27)

The condensed load refrigerant at (2) from the condenser can be specified:

\[ P_2 = P_c, \quad X_2 = X_1, \quad T_2 = T(P_2, X_2), \quad h_2 = h(T_2, X_2) \]  

(3.28)

Similarly, the evaporated vapor refrigerant at (4) from the evaporator can be specified:

\[ P_4 = P_e, \quad T_4 = T_e, \quad Y_4 = Y_1, \quad h_4 = h(T_4, Y_4) \]  

(3.29)

The liquid at (2) undergoes a pressure reduction via the expansion valve before entering the evaporator; therefore,

\[ P_3 = P_e, \quad h_3 = h_2, \quad T_3 = T(h_3, X_3) \]  

(3.30)

In order to obtain the COP of the cycle, energy equilibrium at the generator and evaporator is required; thus,

\[ Q_g = m_i h_i + m_8 h_8 - m_7 h_7, \quad Q_e = m_i (h_4 - h_3), \quad W_{me} = (P_6 - P_3) v_6 \]  

(3.31)

Finally, the performance of the cycle is calculated as

\[ \text{COP} = \frac{Q_e}{Q_g + W_{me}} \]  

(3.32)

The circulation ratio of the cycle is defined as

\[ f = \frac{m_5}{m_1} \]  

(3.33)

The heat rejected by the absorber and condenser can also be calculated by applying the heat conservation principle as

\[ Q_a = m_4 h_4 + m_10 h_{10} - m_2 h_5, \quad Q_c = m_1 (h_1 - h_2) \]  

(3.34)

**Performance Characteristics**

Figure 3.6 shows the typical effect of generator temperatures on COP at various concentrations of ammonia [13,15]. The performance characteristics of the NH₃–H₂O absorption refrigeration cycle are different from those of a mechanical vapor compression cycle. The NH₃–H₂O is a binary mixture, with H₂O as the absorbent and NH₃ as the refrigerant. As discussed previously, the refrigerant is not 100% ammonia as some water is contained in it. This is caused by the vapor–liquid behavior of a binary mixture, with one as absorbent and the other as refrigerant. For a fixed concentration of ammonia, which is controlled by the rectifying process, the cycle performance varies with the generator temperature, and there exists an optimum generator temperature. Since the ammonia concentration of the weak solution entering the generator is determined by the absorber conditions, and the condenser and generator are at the same pressure level; the optimum generator temperature determines the optimum concentration of the strong solution leaving the generator, and therefore the optimum flow rate of the refrigerant vapor emerging from the generator. If the solution temperature at the generator is higher than its optimum value, some thermal energy is wasted, as a higher temperature means a higher grade of thermal energy. However, if the solution temperature
is lower than the optimum value, less refrigerant vapor is produced at the generator. Therefore, the control of generator temperature is an important issue in achieving high performance of the absorption cycle.

Figure 3.6 also shows that if the ammonia concentration in the refrigerant is lowered by the proper control of the rectifying process, COP can be improved due to the increase in refrigerant flow rate for the same amount of the weak solution entering the generator. However, in actual operation, the refrigerant with an ammonia concentration as high as possible is preferred since the water vapor in the refrigerant may freeze along the pipes. Furthermore, if water enters the evaporator, it elevates the evaporating temperature \([13,15]\).

Furthermore, Figure 3.6 shows the possible operation region for the generator as enclosed by the generator temperature envelope. Any generator temperature for the cycle within the region can produce a refrigerating effect. Figure 3.6 also indicates the minimum and maximum permitted generator temperatures; however, it should be noted that the envelope curve may change if conditions at other components vary \([13,15]\).

**Optimum Design Map**

The COP values shown in Figure 3.6 indicate the local maximum for various operating conditions. In order to provide detailed optimum operating conditions for absorption cycles, optimum design maps can be constructed as the one illustrated in Figure 3.7 \([17]\).

The optimum design map in Figure 3.7 is for the NH\(_3\)-H\(_2\)O absorption cycle with absorber temperature \(T_a\) at 30°C. The dashed lines represent constant evaporator temperatures and the solid lines represent condenser temperatures. As NH\(_3\) is the refrigerant in the absorption cycle in Figure 3.7, optimum design maps for evaporating temperatures lower than 0°C can also be constructed in a similar way. The optimum design map can be used to find the required \(T_g\) for achieving the maximum COP value under specified \(T_e\), \(T_c\), and \(T_a\). If there is a shift in one of the conditions, the cycle COP will be lower than the original maximum value, and, therefore, relevant conditions need to be readjusted according to the maps to establish new optimum conditions for the
maximum COP. For example, if $T_e = 4^\circ C$, $T_c = 30^\circ C$, and $T_a = 30^\circ C$, from the map $T_g$ can be found to be $81^\circ C$ and the corresponding COP is 0.720 which is the maximum. If $T_g$ shifts to $72^\circ C$, the COP value is lowered to 0.695 which is no longer the maximum, indicating that the new conditions are not optimized. In order to maintain optimum operation, for $T_g = 72^\circ C$, $T_c$ must be decreased to $25^\circ C$ along the dashed line of $T_e = 4^\circ C$ if the refrigerating temperature remains unchanged; in this case, the conditions are optimized again and the COP value is 0.744, which is higher than the nonoptimum value of 0.695.

Obviously, the optimum design map contains important information for designing new systems and choosing operating conditions for existing systems. If the map is stored in the control system of a cycle, automatic control of absorption systems can be realized for maintaining their optimum performance under various operating conditions [17].

**EJECTOR REFRIGERATION CYCLE**

The ejector refrigeration cycle is normally powered by heat and it can be used to harness waste thermal energy or renewable energies. The principle of ejector refrigeration cycle has been known for many decades; however, its potential economic advantages have not always been realized [49]. The ejector refrigeration units experienced their first wave of popularity in the 1930s. However, these units were later supplanted by mechanical vapor compression systems. Since an ejector refrigeration cycle has several advantages over conventional vapor compression cycles, which include no moving parts (except the pump), very little wear and susceptibility to breakdown, the use of an easily obtainable and safe refrigerant, and utilization of thermal energy as a power source, recent investigations have shown that it could be an alternative heat-powered refrigeration cycle [50–52]. Recent researches indicate that ejector refrigeration technologies not only can serve the needs for cooling requirements such as food preservation in remote areas but also can meet demand for energy conservation and environment protection. Therefore, research activities in this area continue to increase to solve crucial problems that prevent these systems from being ready to compete with the well-known vapor compression system [53]. For example, in order to improve the efficiency of the simple ejector cycle, more complex cycles have been investigated [54–56].
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Cycle Description

The layout of the ejector refrigeration cycle is shown in Figure 3.8 [57,58]. The operating principle is as follows. At the generator, the refrigerant is vaporized at high pressure and the refrigerant vapor flows to the ejector. The vapor or primary fluid then enters the primary convergent–divergent ejector nozzle and expands. This expansion causes a low-pressure region in the ejector that induces the vapor or secondary fluid from the evaporator at state point (8). The primary and secondary fluids then mix in the mixing section and enter the constant-area section of the ejector where an aerodynamic transverse shock is usually induced to create a major compression effect. The mixed stream is further compressed to the back pressure of the condenser in the diffuser section of the ejector, which then leaves the ejector and flows into the condenser and condenses there. Finally, the condensed liquid is divided into two parts. One part (4) is pumped back to the generator and the other (6) expands through a throttling valve to a low-pressure state and enters the evaporator from where it is evaporated to produce the necessary cooling effect. Details of the structure and analysis of an ejector can be found in literatures [57,59–62]. Unlike the vapor compression cycle and the absorption cycle, the ejector cycle operates at three distinct pressure levels: the generator at high pressure, the condenser at intermediate pressure, and the evaporator at low pressure.

Cycle Analysis

As indicated in Figure 3.8, the operation of an ejector refrigeration cycle is normally characterized by the temperatures at the generator, condenser, and evaporator and the total refrigerant mass flow through the generator \( m_1 \) [57,59]. Based on this, at the generator, the cycle can be thermodynamically analyzed as follows. The high-pressure vapor at state (1) before entering the ejector is given by the following functional relationships:

\[
T_1 = T_g, \quad P_1 = P(T_g), \quad h_1 = h(T_g)
\]  

(3.35)

The low-pressure vapor at state (8) is determined by the evaporator temperature:

\[
T_8 = T_e, \quad P_8 = P(T_e), \quad h_8 = h(T_e)
\]  

(3.36)
The condensed liquid at state (3) emerging from the condenser can be specified by the following equation:

\[ T_3 = T_c, \quad P_3 = P(T_c), \quad h_3 = h(T_c) \]  

(3.37)

If the properties at states (1), (8), and (3) are known, the entrainment ratio \( \omega \) of the ejector can be calculated from the following relationships:

\[ P_2 = P_3, \quad \omega = f(P_2, T_1, P_8, T_8, P_2, A_2), \quad h_2 = (h_1 + \omega h_b)/(1 + \omega), \quad T_2 = T(P_2, h_2) \]  

(3.38)

Details of the determination of \( \omega \) can be found in literature [57,59,60]. One part of the condensate is pumped back to the generator; as a result, the enthalpy at (5) is increased:

\[ P_5 = P_1, \quad h_5 = h_1 + (P_5 - P_1)\upsilon_3, \quad T_5 = T(P_5, h_5) \]  

(3.39)

The other part of the condensate flows through the expansion valve, and the fluid pressure is reduced from the condenser pressure to the evaporator pressure; therefore,

\[ P_7 = P_\delta, \quad h_7 = h_3, \quad T_7 = T(P_7, h_7) \]  

(3.40)

The mass flow continuity around the cycle yields the following:

\[ m_5 = m_1, \quad m_8 = m_7 = \omega m_1, \quad m_2 = (1 + \omega)m_1 \]  

(3.41)

In order to determine the COP of the cycle, the energy conservation principle should be applied at the generator and evaporator, which gives the following energy flow rates:

\[ Q_g = m_1(h_1 - h_5), \quad Q_e = \omega m_1(h_6 - h_7), \quad W_{me} = (P_3 - P_5)\upsilon_3 \]  

(3.42)

Therefore, COP is determined:

\[ \text{COP} = \frac{Q_e}{Q_g + W_{me}} \]  

(3.43)

**Performance Characteristics**

In the ejector refrigeration cycle, the ejector is the key component of the cycle, as the performance of the cycle mainly depends on the performance of the ejector. All the ejectors used have their fixed geometries and several distinct characteristics. Therefore, the ejector cycle demonstrates some interesting phenomena [14,60]. The performance of an ejector is measured by the ratio of the primary flow (flow coming from the generator) \( m_1 \) and secondary flow (flow coming from the evaporator) \( m_8 \). This ratio is termed as the entrainment ratio \( \omega \) of the ejector. The higher the entrainment ratio, the higher the secondary flow rate, and, consequently, the higher the coefficient of performance. Therefore, ejectors should be optimally designed to have the highest possible entrainment ratio [63].

One of the ejector characteristics is shown in Figure 3.9 [14]. It is noted that there exists an optimum value of generator temperature at which the entrainment ratio reaches the maximum. The optimum generator temperature depends on the evaporator temperature but in a very narrow range. If the generator temperature is higher than the optimum value, the entrainment ratio falls, resulting in a reduction in COP value and cooling capacity of the cycle [14,50].

Another characteristic of an ejector is its so-called constant capacity shown in Figure 3.10. The constant capacity means that the entrainment ratio is independent of the ejector back pressure, that is, the condenser pressure \( P_c \). As illustrated in Figure 3.10, if the back pressure is higher than a certain value, the entrainment ratio decreases suddenly and then falls to zero. This pressure at...
which the entrainment ratio begins to drop is defined as the critical back pressure of the ejector [14,50]. On the other hand, if the back pressure is lower than the critical pressure, the entrainment ratio remains constant. This constant capacity phenomenon is caused by the ejector’s distinct behavior of choking of the secondary flow. During the operation of an ejector, the walls formed by the expansion of the primary stream \( m_1 \) from the nozzle and by the converging mixing section make a hypothetic converging duct termed as “aerodynamic convergent nozzle” [14] for the secondary fluid \( m_8 \) to pass. The secondary flow accelerates in this aerodynamic convergent nozzle to sonic velocity at its exit plane. The sonic velocity is the condition of flow choking [14]. When the secondary

![FIGURE 3.9](image_url)  
**FIGURE 3.9** Effect of the generator temperature on the entrainment ratio of an ejector refrigeration cycle.

![FIGURE 3.10](image_url)  
**FIGURE 3.10** Effect of the condenser temperature on the entrainment ratio of an ejector refrigeration cycle.
flow is choked, that is, the flow reaches its maximum velocity which is sonic at the exit of the aerodynamic nozzle, a back pressure lower than the critical pressure cannot be sensed upstream within the aerodynamic nozzle, and therefore lowering the back pressure does not affect the flow rate. Under choked conditions, a transverse shock wave normally occurs in the constant-area section or diffuser section of the ejector. The lower the back pressure, the further downstream the shock wave will appear. However, if the back pressure is increased the shock wave tends to move upstream toward the mixing section of the ejector. If the back pressure is further increased to above the critical value, it will cause the shock wave to penetrate into the mixing section, leading to the disappearance of choking of the secondary flow and a fall in the entrainment ratio. This effect is so severe that a little further increase in the back pressure will finally force the shock wave to pass through the mixing section and reach the exit plane of the primary nozzle, causing no secondary flow and a drop in the entrainment ratio to zero. Obviously, fixed-geometry ejector refrigeration cycles should operate at these critical conditions and should avoid condenser pressures being lower or higher than these critical values [14,50].

The characteristics of the ejector shown in Figures 3.9 and 3.10 indicate that ejector refrigeration cycles cannot perform well unless they work exactly at their design conditions. However, the operating conditions may vary in the actual operation, which will cause the cycles to lose performance or waste energy. Therefore, a variable-geometry ejector should be designed in order to achieve optimum performance over a wide range of operating conditions [60].

Water is normally used in the ejector refrigeration cycle. However, if evaporating temperature lower than 0°C is to be achieved, other refrigerants such as hydrochlorofluorocarbons (HCFCs) and hydrofluorocarbons (HFCs) should be used [58].

**Optimum Control Map**

The operation of an ejector cycle will reach its optimum performance if the ejector operates under critical conditions [14,50]. If these critical pressure data are compiled for an ejector, an optimum control map can then be constructed so that the maximum performance can be obtained [14]. Figure 3.11 shows a typical example of such optimum control maps. These optimum control maps can be stored in the control system of the refrigeration cycle in order to maintain the ejector at its critical operating conditions. An example of using the control map is given below. If the cycle

![FIGURE 3.11 The optimum control map for an ejector refrigeration cycle to achieve the maximum performance.](image-url)
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is designed to operate at critical point O, that is, at an evaporator temperature of 10°C, a generator temperature of 120°C, and a critical condenser temperature of 26.2°C, the maximum entrainment ratio that the ejector can obtain is 0.43. Any other operating points along line OG will have the same performance. Assume that, due to the change in environmental conditions, the condenser temperature is lowered to a new point M; the cycle is then no longer at its critical condition. Two methods can be used to control the cycle and bring it to a new critical condition: keeping constant cooling capacity or keeping constant evaporator temperature. If a constant cooling capacity is used, the same entrainment ratio or COP should be maintained. This can be achieved by lowering the generator temperature from line OK to line NM and simultaneously lowering the evaporator temperature from line ON to line KM; then the new critical condition for the cycle is established at point M. If the priority is to maintain a constant evaporator temperature, the generator temperature should then be lowered from point O to point H. In this case, the entrainment ratio will rise from point M to point H, and the new critical condition for the cycle is established at point H. Similarly, assume, on a hot day, the condenser temperature is increased to a higher value at point M′, then the generator temperature must be increased from line OK′ to line N′M′ with M′ as the new critical condition; the entrainment ratio then drops from point M′ to point H and the evaporator temperature rises from line ON′ to line K′M′. Obviously, during the operation of the cycle, the operating condition should be kept at critical points all the time for the maximum performance of the cycle, which can be achieved by automatic control of the generator, condenser, and evaporator temperatures according to the technical data provided in the optimum control maps [14]. Similar optimum control maps may also be constructed for an ejector operating on halocarbon compounds in order to achieve refrigeration below 0°C [58].

NOVEL COMBINED CYCLES

Besides the above-discussed vapor compression, absorption, and ejector refrigeration cycles, there are other types of refrigeration cycles such as the adsorption cycle [11]. The adsorption refrigeration cycle is based on the principle of either physisorption or chemisorption. An example of the physisorption cycle is the zeolite 13X–water adsorption cycle [11,64–66], and an example of the chemisorption cycle is the metal hydride adsorption cycle [16,67,68]. However, the vapor compression, absorption, and ejector refrigeration cycles are the three most common types of refrigeration systems. In these cycles, the vapor compression systems are generally electric-powered, and hence emit large amounts of CO₂ and NOₓ. Both absorption and ejector systems can be powered by thermal energy; however, their COP values are very low as compared with the vapor compression system.

In order to bring together the advantages of the vapor compression, absorption, and ejector refrigeration systems, novel combined cycles have been developed [20–23]. One of them combines an absorption cycle with an ejector refrigeration cycle, and the other combines an ejector cycle with a vapor compression cycle. As a result, significant improvement to the system COP can be achieved with these novel combined cycles. In addition, investigations have also been conducted to develop other novel refrigeration techniques for food preservation. For example, air cycle technology has been studied for food sector applications including refrigerated warehouse and refrigerated transport [69,70], the air cycle operates in a method similar to the vapor compression cycle, but in such a cycle air is used as refrigerant and therefore no phase changes of the refrigerant are involved [11]. Another example is the Stirling cycle that can be used in many food refrigeration applications as it can operate down to cryogenic temperatures [71,72]; furthermore, a thermoacoustic Stirling refrigerator operated by using sound waves and a nonflammable mixture of inert gas in a resonator to produce cooling has been developed, which can achieve a no-load temperature of ∼65°C, a cooling capacity of about 270 W at −20°C, and a cooling capacity of 405 W at 0°C [73]. Yet another example is the prototype larger-capacity thermoelectric domestic refrigerator utilizing the Peltier effect that has been built and tested [74].
The combined ejector–absorption cycle is shown in Figure 3.12. In this combined cycle, the ejector is integrated into the absorption cycle in order to increase the refrigerant flow rate from the evaporator and to raise the cooling capacity of the cycle [20].

**Cycle Analysis**

The operating principle of the combined cycle shown in Figure 3.12 is as follows. At the generator, the absorbent–refrigerant solution is heated at the generator by heat source $Q_g$ to produce high-pressure steam refrigerant at (1). This refrigerant (primary fluid) then flows through the primary convergent–divergent nozzle of the ejector and entrains vapor (secondary fluid) evaporated from the evaporator. The primary and secondary fluids are mixed in the ejector and emerge from it at (2). The combined stream then flows to the condenser and condenses to a liquid state at (3). The heat of condensation $Q_c$ is rejected to the environment. The condensed liquid (3) expands through a throttling valve to a low-pressure state (4) in order to enter the evaporator and evaporate there to produce the necessary cooling effect $Q_e$. Some of the evaporated vapor is entrained by the ejector (13) to mix with the primary fluid (1) and the remainder (6) is absorbed by the strong solution from a weak solution (7). The heat of absorption $Q_a$ is rejected to the environment. The weak solution (7) is pumped back (8) to the generator (9) via the solution heat exchanger, to gain sensible heat from the strong solution coming from the generator to return to the absorber (12) via the throttling valve (1). Unlike an absorption cycle, the combined cycle operates at three distinct pressure levels: the generator at a high-pressure level, the absorber and condenser at an intermediate-pressure level, and the evaporator at a low-pressure level. This is due to the integration of the ejector in the cycle. If the combined ejector–absorption cycle is used to generate cooling below 0°C, refrigerants used should be able to evaporate at temperatures below 0°C; for example, NH$_3$ in the NH$_3$–H$_2$O system.
PERFORMANCE CHARACTERISTICS

Due to the additional amount of refrigerant vapor entrained by the ejector, the COP of the combined cycle is higher than that of the absorption cycle [20]. Figure 3.13 shows the predicted COP for the combined cycle as compared with an absorption cycle. It can be seen that, for the absorption cycle, the COP varies with generator temperature. The optimum generator temperature for the maximum COP is determined by evaporator temperature. As discussed before, the solution concentration entering the generator is determined by the absorber conditions. Since the condenser and absorber operate at the same pressure level, the optimum solution concentration leaving the generator is determined by the generator temperature. If the temperature at the generator is higher than its optimum value, some thermal energy is wasted. However, if the temperature is lower than the optimum value, less refrigerant vapor is produced at the generator. Figure 3.13 also shows the COP of the combined cycle. For similar reasons, optimum generator temperature for the combined cycle is also available; however, as the pressures at the generator and condenser are at different levels, the pressure at the generator can be chosen as a further controlling parameter over solution concentration.

The results in Figure 3.13 indicate that the COP values of the combined cycle are significantly higher than that of the absorption cycle. For example, for the same evaporator temperatures at 5°C and 10°C, the COP values of the combined cycle are about 20% and 40% higher than the absorption cycle, respectively. Thermodynamically, an absorption cycle can be considered to be composed of a power subcycle (PSC) and a conventional refrigeration subcycle (CRSC). The PSC receives energy in the form of heat \( Q_g \) at \( T_g \), delivers some energy \( W \) in the form of work to the CRSC, and rejects a quantity of energy \( Q_a \) in the form of heat at \( T_a \). The CRSC receives work \( W \) from the PSC to pump heat \( Q_e \) at \( T_e \) to a sink at \( T_c \), so that \( Q_c \) is rejected. Consequently, the Carnot COP of an absorption cycle can be expressed as

\[
(COP)_{abs}^C = \left( \frac{T_e - T_c}{T_g} \right)_{PSC} \left( \frac{T_c}{T_e - T_c} \right)_{CRSC}
\]

![Figure 3.13](image.png)

**FIGURE 3.13** A comparison of the coefficient of performance (COP) values between a conventional absorption cycle and the combined ejector–absorption cycle.
For the combined cycle, the addition of the ejector forms an additional ejector refrigeration subcycle (ERSC) along with the CRSC. Therefore, work $W$ from the PSC is used to drive both the CRSC and ERSC. In the combined cycle, the Carnot COP can be expressed as

$$
(COP)_{\text{com}}^C = \left(\frac{T_e - T_s}{T_e - T_s}\right)_{\text{PSC}} \left[\left(\frac{T_e}{T_e - T_c}\right)_{\text{CRSC}} + \left(\frac{T_e}{T_e - T_c}\right)_{\text{ERSC}}\right]
$$

or

$$
(COP)_{\text{com}}^C = 2(COP)_{\text{abs}}^C
$$

**COMBINED EJECTOR–VAPOR COMPRESSION CYCLE**

The combined ejector–vapor compression cycle is shown in Figure 3.14. In this combined cycle, an ERSC and a mechanical vapor compression subcycle (VCSC) are integrated into a cycle that operates in a manner similar to a vapor compression system [21–23]. The connection between the two subcycles is the intercooler, which serves as the “evaporator” for the ERSC and as “condenser” for the VCSC. The temperature at the intercooler is set between the condenser temperature and the evaporator temperature of the combined cycle. This arrangement secures the possibility of using two different refrigerants and makes full use of the advantages of each subcycle. If only one refrigerant is used in the combined cycle, the intercooler and its separating piping can be simplified.

**CYCLE ANALYSIS**

The operating principle of the combined cycle shown in Figure 3.14 is as follows. At the generator, the refrigerant is heated to produce high-pressure refrigerant vapor (1), which flows through the
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An ejector and induces vapor (6) from the intercooler. The primary vapor and the entrained vapor then mix in the ejector and combine to a single stream. After exiting the ejector, the combined vapor stream (2) flows via the heat exchanger to the condenser where it condenses into liquid. The condensed liquid is then divided into two parts. One part (7) is pumped back (8) to the generator (9) via the heat exchanger and gains sensible heat from the vapor (2) from the ejector. The other part (4) expands through the throttling valve and enters the intercooler where it is evaporated by the condensation heat from the VCSC, which is generated from the compressed vapor refrigerant (10) from the compressor into liquid (11). The condensed liquid enters the evaporator via throttling valve (11) and evaporates there to produce the necessary cooling effect $Q_c$. The vapor (13) is then compressed by the compressor and finally flows into the intercooler. The heat exchanger in the combined cycle is an energy-saving component and it can be omitted. Due to the presence of two subcycles, the combined cycle operates at four different pressure levels, that is, the generator, condenser, intercooler, and evaporator all operate at different pressures.

**Performance Characteristics**

Figure 3.15 shows the comparison between the combined cycle in Figure 3.14 and the conventional vapor compression cycle under the same operating conditions [21]. It is noticed that COP of the combined cycle can be more than 50%, depending on the condensing temperature. From the point of view of saving energy, more than 50% electric energy could be saved, and, as a consequence, more than 50% reduction in NOx, hydrocarbons, and CO2 emissions due to electricity generation may be expected. If the COP of the combined cycle is compared with the ejector cycle under the same operating conditions, the performance of the ejector in the combined cycle is significantly improved.

Comparison has also been made of the COP values when only one refrigerant is used in both the VCSC and the ERSC when two different refrigerants are used respectively in the subcycles. Research shows that the combined cycle performs better with dual refrigerants than with only one refrigerant [22].

![Figure 3.15](image-url)  
**FIGURE 3.15** A comparison of the coefficient of performance (COP) values between a conventional vapor compression cycle and the combined ejector–vapor compression cycle.
CONCLUSIONS

In the freezing of food, freezing equipment is needed. The operation of the freezing equipment is normally based on some refrigeration cycles. At present, the most commonly used refrigeration cycle is the mechanical vapor compression cycle powered by electricity. Generally speaking, electricity is generated by burning fossil fuels, which produce a large amount of CO$_2$ that is the main contributor to global warming. On the other hand, waste thermal energy is widely available; therefore, utilization of such energy is both economically feasible and environmentally friendly. For this reason, in recent years, thermal-powered refrigeration cycles, especially the absorption cycle and the ejector cycle, have been extensively studied. However, compared with the vapor compression cycle, the COP values of the absorption and ejector cycles are significantly lower. Therefore, two innovative combined cycles, that is, the combined ejector–absorption cycle and the combined ejector–vapor compression cycle, have been developed. These combined cycles bring together the advantages of the individual conventional cycles and minimize their disadvantages, and therefore significant improvement in system COP has been made. If such combined cycles are used in food refrigeration, significant saving in running cost for the food industry can be expected.

NOMENCLATURE

$A$ Area (m$^2$)  
$A_i$ Ejector area ratio  
COP Coefficient of performance  
$cp$ Specific heat (kJ/kg K)  
$E$ Total energy (kJ)  
$f$ Ratio of mass flow rate  
$g$ Gravitational acceleration (= 9.807 m/s$^2$)  
$h$ Enthalpy (kJ/kg)  
$h_t$ Heat transfer coefficient (kW/m$^2$ K)  
$k$ Thermal conductivity (kW/m K)  
$m$ Mass flow rate (kg/s)  
$P$ Pressure (kPa)  
$Q$ Heat transfer rate (kW)  
$s$ Entropy (kJ/kg K)  
$T$ Temperature (K)  
$t$ Temperature (°C)  
$V$ Velocity (m/s)  
$W$ Power (kW)  
$x$ Distance (m)  
$X$ Solution concentration, ammonia mass fraction in liquid phase  
$Y$ Ammonia mass fraction in vapor phase  
$z$ Elevation (m)

GREEK LETTERS

Δ Difference  
$\eta$ Effectiveness of heat exchanger  
$\theta$ Time (s)  
$\upsilon$ Specific volume (m$^3$/kg)  
$\omega$ Entrainment ratio
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Subscripts

a Absorber
abs Conventional absorption cycle
c Condenser, compressor
com Combined refrigeration cycle
CRSC Conventional refrigeration subcycle
e Evaporatorfluid

g Generator
i Intercooler
in Inflow
max Maximum
me Mechanical
min Minimum
net Net
out Outflow
PSC Power subcycle
s Saturation
sur Surface

Superscript

C Carnot refrigeration cycle

REFERENCES

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