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Course Handout (Lectures & Solved Problems)

Intended for students of: Bachelor Level: 3rd Year

Refrigeration Systems and Heat Pumps

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PREFACE

This course on Refrigeration Systems and Heat Pumps is designed for third-year students pursuing a Bachelor's degree in Mechanical Engineering with a specialization in Energy. At this stage of their academic development, students are expected to move beyond introductory thermodynamics and begin applying fundamental principles to real engineering systems. The objective of this course is to bridge theory and practice by providing a clear and structured understanding of refrigeration and heat pump technologies, which are essential components of modern energy systems.

Refrigeration and heat pump applications play a critical role in industry, buildings, food preservation, thermal process control, and energy efficiency. They are also central to current global challenges related to sustainable development, energy consumption reduction, and environmental protection. For energy engineers, mastering these technologies is not only a technical requirement but also a responsibility in the design of efficient and environmentally conscious systems.

This course builds on prior knowledge of thermodynamics, heat transfer, and fluid mechanics. It develops the ability to analyze vapor compression cycles, interpret T–s and P–h diagrams, apply energy balances, and evaluate system performance through the coefficient of performance. Students will also gain an understanding of practical system components, refrigerant selection criteria, and alternative refrigeration technologies such as absorption systems and air standard cycles. The integration of heat pump analysis further strengthens the connection between cooling and heating applications in energy engineering.

The pedagogical approach emphasizes conceptual clarity, structured problem solving, and engineering reasoning. While the mathematical treatment remains accessible, the focus is placed on understanding physical behavior, identifying performance limitations, and linking theoretical models to real operating conditions. Exercises and practical questions are included to encourage critical thinking and prepare students for professional practice.

By the end of this course, students should possess the analytical tools and technical vocabulary required to study, design, and evaluate refrigeration and heat pump systems. This foundation will support further coursework in thermal systems, energy efficiency, renewable energy integration, and advanced HVAC engineering, as well as future professional responsibilities in the energy sector.

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Contents

Introduction

This course provides a first, fundamental introduction to refrigeration systems and heat pumps, aimed at students who are encountering these technologies for the first time in a structured way. Its objective is to build a solid conceptual and methodological basis rather than to cover advanced design or specialized industrial applications. The emphasis is on understanding basic principles, standard technical terminology, and simple performance indicators, so that students can later approach more advanced refrigeration and heat pump topics with confidence.

The course starts with a general overview of refrigeration, including the historical development of artificial cooling and the main areas where refrigeration and heat pumps are used, such as food preservation, comfort cooling, and heating. At this stage, the focus is on qualitative understanding: what a refrigeration machine does, what a heat pump does, and why these systems are essential in modern society. The Carnot refrigeration cycle is introduced primarily as a simple ideal model that shows the basic idea of moving heat from a low temperature level to a higher one and defines the coefficient of performance (COP) as a basic measure of efficiency.

After this general introduction, the course presents the basic vapor-compression refrigeration cycle in a simple and progressive way. Students learn to identify the four main processes (compression, condensation, expansion, evaporation) and to place them on temperature–entropy (T–s) and pressure–enthalpy (P–h) diagrams at an introductory level. The diagrams are used to visualize how the refrigerant changes state in each component, without going into advanced optimization or complex off-design analysis. Basic energy balances are introduced to show how to estimate cooling capacity, compressor work, and COP, using simple examples with clear assumptions.

The main components of a vapor-compression system (compressor, condenser, evaporator, and expansion device) are then studied with an emphasis on their role in the cycle and their basic operating principles. The course introduces only the most common configurations and concepts needed at an introductory level, such as the idea of heat exchangers for condensers and evaporators and the function of a simple expansion device. The aim is for students to be able to recognize these components in real systems, understand their purpose, and relate them to the thermodynamic processes they have studied in the cycle.

To complete the general picture, the course briefly presents other types of refrigeration machines (such as absorption units and air-cycle systems) and the basic concept of heat pumps. These topics are treated qualitatively, to show that different technologies can achieve refrigeration or heating using the same fundamental thermodynamic principles. For heat pumps, the course highlights the idea of a reversible system that can provide heating in winter and cooling in summer, and introduces COP as a simple tool to compare heating and cooling performance under different operating conditions.

Throughout the course, the level of treatment is deliberately introductory: mathematical developments are kept to what is necessary for basic cycle calculations, and the technical language used corresponds to standard engineering terminology without going into specialized sub-disciplines. By the end, students are expected to be able to describe the purpose and basic operation of refrigeration machines and heat pumps, interpret simple T–s and P–h diagrams, perform straightforward COP and energy balance calculations, and identify the main components of a basic system. These outcomes provide a clear foundation for subsequent, more advanced courses in refrigeration, heat pumps, and energy engineering.

Chapter I: General Overview

1. Introduction

The first chapter provides a general overview of refrigeration machines and introduces the basic concepts that will be used throughout the course. It begins with a short historical review of the development of artificial refrigeration, highlighting the key technological milestones and the growing importance of refrigeration in food preservation, comfort applications, and industry. On this basis, the chapter presents the Carnot refrigeration cycle as an ideal reference, in order to clarify the fundamental idea of transferring heat from a low-temperature reservoir to a high-temperature reservoir. The coefficient of performance (COP) of the Carnot cycle is introduced and interpreted as a simple measure of the theoretical efficiency of a refrigeration machine.

In this introductory framework, the emphasis is placed on a clear, qualitative understanding of the processes involved, rather than on detailed calculations or advanced system design. The chapter establishes the thermodynamic language and symbols that will be used later for the study of real refrigeration cycles and heat pumps. It is intended to give students a solid first contact with the physical principles governing refrigeration, and to prepare them for the more detailed analysis of vapor-compression systems developed in the following chapters.

2. History of Refrigeration

2.1. Key Historical Milestones

For over two thousand years, humans have known that cold prevents food from deteriorating, but it is only recently (about a hundred years) that they understood why. They have therefore developed various means of preservation:

- Storage in cooler places
- Smoking
- Drying
- Dehydrating

In Ancient Rome, thousands of slaves were employed to bring snow and ice from mountain peaks for the emperors. Ice and cold water thus helped preserve food in reasonably good condition.

The food preservation industry was born and became commercially important during the 18th century, when ice cut during winter began to be sold. This led, in the 19th century, to the first insulated wooden icebox.

1824 - Michael Faraday's experiments on ammonia absorption and gas liquefaction contributed to the scientific foundations of absorption refrigeration.
1834 - Jacob Perkins filed a patent for a vapor-compression refrigeration machine.
1844 - Dr. John Gorrie developed an air-cycle refrigeration machine capable of producing ice.
1858 - Ferdinand Carré developed an ammonia–water absorption refrigeration machine.
Around 1900 - Only a few manufacturers produced early household refrigerators, which were expensive, unreliable, and required frequent maintenance.
1919 - Refrigerators were publicly exhibited and offered for sale.
1927 - General Electric introduced a hermetically sealed refrigeration unit for domestic refrigerators.
1930 - The invention and development of Freon-12 (R-12) led to a major expansion of the refrigeration industry.

2.2. Methods of refrigeration

Seven Primary Methods:

1. **Sublimation of a Solid**

   This consists of making a substance pass from the solid state to the vapor state by heat absorption. The most common case is CO_2, which at atmospheric pressure has a sublimation temperature of -78.9°C.

2. **Expansion of a Compressed Gas**

   This is based on the principle of temperature reduction of a fluid during expansion (with or without external work). However, this reduction is greater during expansion without external work (Joule-Thomson expansion: throttling through a valve).

3. **Fusion of a Solid Body**

   This occurs at constant temperature by absorption of the latent heat of fusion of the body considered. This discontinuous process, although simple, has the disadvantage of requiring prior freezing unless this state is available naturally.

4. **Thermoelectric Cooling (Peltier Effect)**

   This is used to produce very small quantities of cold. It consists of passing a direct current through a thermocouple made of conductors of different natures alternately connected by copper bridges.

5. **Dissolution of a Salt**

   The dissolution of a salt in water causes a temperature decrease of the solution. For example, a mixture of snow (4 parts) and potash (3 parts) lowers the solution temperature from 0°C to -40°C.

6. **Adiabatic Demagnetization**

   This consists of a reorganization of the electron cloud of a body, which allows obtaining very low temperatures (10^{-2} to 10^{-6} K).

7. **Vaporization of a Liquid**

   This allows cold production through heat absorption via a heat exchanger (evaporator), with the produced vapor subsequently liquefied in another exchanger (condenser). The fluid thus describes a cycle within a continuously operating machine.

   Machines using this principle can be grouped into two major families: mechanical compression machines and absorption machines.

   Vaporization of a liquid in a closed circuit remains the most widely used method for cold production.

2.3. Main Applications of Refrigeration

Refrigeration is used across a wide range of engineering applications, mainly depending on the temperature level required. Table I.1 summarizes the most common classical refrigeration applications: air conditioning (about +16 to +26°C) for human comfort; food refrigeration (0 to +10°C) for short- to medium-term preservation; food freezing (−35 to 0°C) for long-term preservation; freeze-drying (−80 to −30°C) for low-temperature drying of sensitive products; and industrial cooling processes (down to around −200°C) used in chemical applications and material thermal testing. Table I.2 shows applications of cryogenics, which deals with the production of extremely low temperatures (for example natural gas liquefaction, air separation, and hydrogen/helium liquefaction). However, cryogenic refrigeration is outside the scope of this course, and the course will focus only on the classical refrigeration systems and their practical applications.

Table I.1 Main applications of classical refrigeration

Application	Temperature Range (°C)	Purpose
Air conditioning	+16 to +26	Human comfort
Food refrigeration	0 to +10	Short/medium-term food preservation
Food freezing	-35 to 0	Long-term food preservation
Freeze-drying	-80 to -30	Low-temperature desiccation
Industrial processes	-200 to 0	Chemical applications, material thermal testing

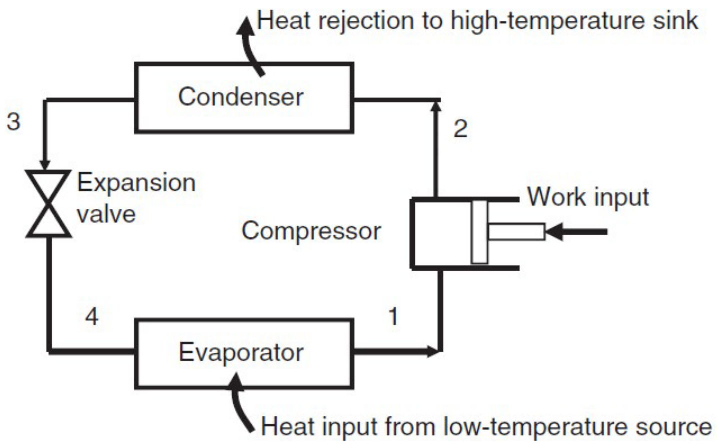
Table I.2 Main applications of cryogenics

Application	Temperature Range (K)	Purpose
Natural gas liquefaction	93 to 113	Transport as LNG
Air separation by distillation	70 to 80	Distillation
Hydrogen liquefaction	14 to 30	Rocket propulsion / aerospace
Helium liquefaction	1 to 5	Superconductivity
Magnetic methods	10^{-3} to 10^{-2}	Adiabatic demagnetization / Magnetic refrigeration

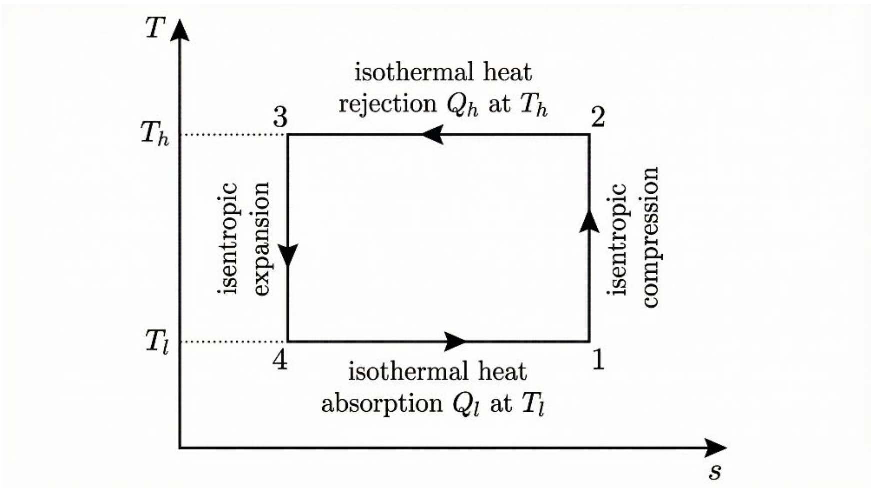
3. Carnot Refrigeration Cycle

The Carnot cycle is an ideal, fully reversible thermodynamic cycle that sets the upper theoretical limit for the performance of any refrigeration machine. It operates between two thermal reservoirs: a low-temperature reservoir at temperature T_l, from which heat is absorbed, and a high-temperature reservoir at temperature T_h, to which heat is rejected. In refrigeration mode, the Carnot cycle corresponds to the Carnot heat engine operated in reverse: external work is supplied to drive the cycle, with the net result of removing thermal energy Q_l from the cold reservoir and rejecting Q_h to the hot reservoir. The cycle is conveniently represented in the temperature–entropy (T–s) diagram, where the four ideal processes appear as two isentropic vertical lines and two isothermal horizontal lines, forming a closed loop (see Figure I.1.(a) and (b)). This representation is particularly useful to visualize the heat exchanges at constant temperature and the work requirement associated with the area enclosed by the cycle.

- Isentropic compression (1 → 2): The working fluid is compressed reversibly and adiabatically from state 1 to state 2. During this process, the pressure and temperature increase from the evaporator conditions to the condenser conditions, while the entropy remains constant. The compressor input work W is supplied in this step, and the state moves vertically upward on the T–s diagram.
- Isothermal heat rejection (2 → 3): At the high temperature T_h, the working fluid rejects heat Q_h to the hot reservoir at constant temperature in a reversible manner. Entropy decreases from state 2 to state 3, while the temperature remains equal to T_h, so the process is represented by a horizontal line to the left on the T–s diagram. This step models the ideal behavior of the condenser in contact with the high-temperature reservoir.
- Isentropic expansion (3 → 4): The fluid undergoes a reversible adiabatic expansion from state 3 to state 4. Its pressure and temperature decrease from the condenser level back to the evaporator level, while the entropy remains constant. On the T–s diagram, this transformation is again shown as a vertical line, this time downward, symmetric to the compression process.
- Isothermal heat absorption (4 → 1): At the low temperature T_l, the working fluid absorbs heat Q_l from the cold reservoir at constant temperature in a reversible manner. Entropy increases from state 4 to state 1, while the temperature remains equal to T_l, so the process appears as a horizontal line to the right on the T–s diagram. This step models the ideal behavior of the evaporator in contact with the low-temperature reservoir.



(a)



(b)

Figure I.1 (a) A schematic illustration of refrigerator and (b) T-s diagram of the Carnot refrigeration cycle.

The energy balance over one complete cycle expresses that the net work input is equal to the difference between the heat rejected and the heat absorbed:

$$W = Q_h - Q_l$$

4. Coefficient of Performance (Cop)

For the Carnot refrigeration cycle, the coefficient of performance is defined as the ratio of the useful refrigeration effect to the work input:

$$\text{COP}_{\text{Carnot}} = \frac{Q_l}{W}$$

Using the fact that, for a reversible cycle operating between two constant-temperature reservoirs,

$$\frac{Q_l}{T_l} = \frac{Q_h}{T_h}$$

one obtains the classical expression of the Carnot COP in terms of the reservoir temperatures:

$$\mathrm{COP}_{\mathrm{Carnot}} = \frac{T_l}{T_h - T_l}$$

5. Chapter Summary

Chapter I introduces the fundamental concepts of refrigeration by presenting its historical development, operating principles, and thermodynamic foundations. It begins with a brief history showing how humans progressed from natural cooling methods to modern refrigeration technologies, highlighting milestones such as early ice preservation, 19th-century mechanical systems, and the widespread adoption of household refrigerators in the 20th century. The chapter then explains the main methods of producing cold, emphasizing vaporization in closed cycles as the most widely used approach, and outlines key application areas such as air conditioning, food preservation, freezing, and industrial cooling (as summarized in the tables on pages 4–5). It introduces the Carnot refrigeration cycle as the ideal reference model, describing its four reversible processes—isentropic compression, isothermal heat rejection, isentropic expansion, and isothermal heat absorption—illustrated in the T–s diagram (Figure I.1, pages 5–6), and establishes the energy balance of the cycle. Finally, the coefficient of performance (COP) is defined as the ratio of useful cooling to work input, with its theoretical expression derived from the temperatures of the hot and cold reservoirs, providing a measure of the maximum efficiency achievable by refrigeration machines. Overall, the chapter builds a qualitative thermodynamic framework and vocabulary that prepares students for the detailed study of real refrigeration and vapor-compression systems in later chapters.

6. Solved Exercises

Things Engineers Think About

Note to students

Some questions in this section are not explicitly answered in the course document. They are intended to complement the course by encouraging application of concepts and practical engineering reasoning. The answers provided therefore extend the lecture material and help develop understanding of real refrigeration system operation.

1. Why is the Carnot refrigeration cycle used as an ideal reference even though it cannot be achieved in practice?
2. What physical meaning does the coefficient of performance (COP) have in refrigeration, and why can it be greater than one?
3. Why must external work be supplied to transfer heat from a low-temperature reservoir to a high-temperature reservoir?
4. How does the temperature difference between hot and cold reservoirs affect the maximum achievable COP?
5. Why is refrigeration considered an energy transfer process rather than an energy conversion process?
6. What are the thermodynamic limitations imposed by the second law on refrigeration machines?
7. Why is the T–s diagram particularly useful for explaining the Carnot refrigeration cycle?
8. In the Carnot cycle, why are compression and expansion assumed to be isentropic?
9. What is the practical meaning of the energy balance $W = Q_h - Q_l$ for a refrigeration cycle?
10. Why is the Carnot COP considered the upper limit for any real refrigeration system?
11. How did the historical development of refrigeration influence modern industrial and domestic applications?

12. Why is vaporization in a closed cycle the most widely used method for producing refrigeration?

13. What distinguishes refrigeration from simple cooling by natural processes such as ice melting?

14. How does understanding ideal cycles help engineers analyze and improve real refrigeration systems?

Answers

1. Because it represents the maximum theoretical efficiency possible between two temperature levels.

2. COP measures heat removed per unit of work input. It can exceed 1 because refrigeration transfers heat rather than producing it.

3. Heat cannot naturally flow from cold to hot without energy input, according to the second law of thermodynamics.

4. A larger temperature difference reduces the maximum COP; a smaller difference increases it.

5. The system moves heat from one place to another instead of converting energy into work.

6. The second law prevents perfect efficiency and requires work input for heat transfer from cold to hot.

7. It clearly shows heat transfer, entropy changes, and ideal processes in the refrigeration cycle.

8. They are assumed reversible and adiabatic to represent ideal behavior.

9. The work input equals the difference between heat rejected and heat absorbed.

10. Real systems have irreversibilities such as friction and heat losses, so they cannot reach Carnot performance.

11. It led to modern technologies used in food preservation, air conditioning, and industrial cooling.

12. Phase change absorbs large amounts of heat efficiently, making refrigeration practical.

13. Refrigeration actively removes heat using a controlled thermodynamic cycle, not passive cooling.

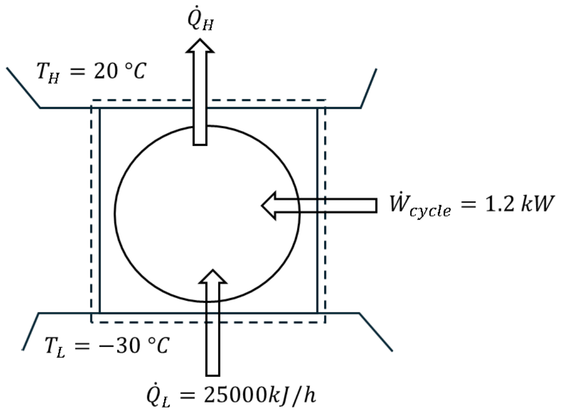
14. They provide a reference for evaluating real systems and identifying performance improvements.

Exercise 1

An inventor claims to have developed a refrigeration cycle that requires a net power input of 1.2 kW to remove 25000 kJ/h of energy by heat transfer from a reservoir at -30°C and discharge energy by heat transfer to a reservoir at 20°C. There are no other energy transfers with the surroundings. Evaluate this claim.

Solution

Schematic and given data:



The system shown in the schematic undergoes a refrigeration cycle while receiving energy by heat transfer from reservoir at T_L and discharging energy by heat transfer to a reservoir at T_H.

The COP_R of this cycle is given by:

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{cycle}} = \frac{25000/3600}{1.2} = 579$$

The maximum coefficient of performance any refrigeration cycle can have while operating between reservoirs at T_L, T_H is given as follows:

$$COP_{R,Carnot} = \frac{T_L}{T_H - T_L} = \frac{-30 + 273.15}{(20 + 273.15) - (-30 + 273.15)} = 4.86$$

Since the claimed value for COP_R exceeds $COP_{R,Carnot}$, the inventor's claim is **invalid**.

Exercise 2

A Carnot refrigerator operates between a cold reservoir at $T_l = 260\ K$ and a hot reservoir at $T_h = 310\ K$.

a. Calculate the theoretical COP of the refrigerator.

b. Explain the physical meaning of this value.

c. State two reasons why real refrigeration systems have lower COP values than the Carnot cycle.

Solution

a) For a Carnot refrigerator, the coefficient of performance is:

$$COP_{Carnot} = \frac{T_l}{T_h - T_l}$$

Substituting the given values:

$$COP_{Carnot} = \frac{260}{310 - 260} = \frac{260}{50} = 5.2$$

b) A COP of 5.2 means that for every 1 unit of work supplied to the refrigerator, the system removes **5.2 units of heat** from the cold reservoir.

This indicates the theoretical maximum efficiency achievable between these two temperature levels.

c) Real refrigeration systems always have lower COP values than the Carnot cycle due to irreversibilities such as:

- Friction and pressure losses in pipes and components
- Non-ideal compression and expansion processes
- Heat losses to the surroundings
- Temperature differences required for real heat transfer in condensers and evaporators

These factors reduce the efficiency compared with the ideal reversible Carnot cycle.

Exercise 3

A Carnot refrigeration machine operates between a cold reservoir at $T_l = 250\ K$ and a hot reservoir at $T_h = 300\ K$.

a. Calculate the theoretical coefficient of performance (COP) of the refrigerator.

b. If the refrigerator extracts $500\ kJ$ of heat from the cold reservoir per cycle, determine the work input required.

c. Determine the amount of heat rejected to the hot reservoir.

d. Explain briefly how the COP would change if the temperature of the cold reservoir decreases.

Solution

a) For a Carnot refrigerator:

$$COP_{Carnot} = \frac{T_l}{T_h - T_l}$$

$$COP_{Carnot} = \frac{250}{300 - 250} = \frac{250}{50} = 5$$

b) The COP is defined as:

$$COP = \frac{Q_l}{W}$$

Thus,

$$W = \frac{Q_l}{COP} \Rightarrow W = \frac{500}{5} = 100\,kJ$$

c) Energy balance for one cycle:

$$Q_h = Q_l + W$$

$$Q_h = 500 + 100 = 600\,kJ$$

d) From the COP expression:

$$COP = \frac{T_l}{T_h - T_l}$$

If the cold reservoir temperature decreases:

- $T_l$becomes smaller
- The denominator $(T_h - T_l)$becomes larger

Therefore, the COP decreases.

Conclusion:

When the cold temperature decreases, the refrigerator becomes less efficient and requires more work to produce the same cooling effect.

Exercise 4

A refrigeration cycleis used to keep a food department at −18 °C in an environment at 20°C. The total heat gain to the food department is estimated to be 1800 kJ/h and the heat rejection in the condenser is 3000 kJ/h. Calculate

a. the power input to the compressor in kW,

b. the COP of the refrigerator,

c. the Carnot COP of the refrigerator,

d. the minimum power input to the compressor if a reversible refrigerator was used.

Solution

a) The power input to the compressor is determined from energy balance on the refrigeration cycle:

$$W_{in} = \dot{Q}_H - \dot{Q}_L = 3000 - 1800 = 1200\, kJ/h = 1200 \times \frac{1}{3600} = 0.333\, kW$$

b) The COP of the refrigerator can be determined as

$$COP_R = \frac{\dot{Q}_L}{W_{in}} = \frac{1200/3600}{0.333} = 1.5$$

c) The Carnot or maximum COP of the refrigerator can be determined as

$$COP_{R,Carnot} = \frac{T_L}{T_H - T_L} = \frac{-18 + 273.15}{(20 + 273.15) - (-18 + 273.15)} = \frac{255.15}{38} = 6.71$$

d) rom the definition of the maximum COP of the refrigerator, the minimum power input to the compressor is determined as

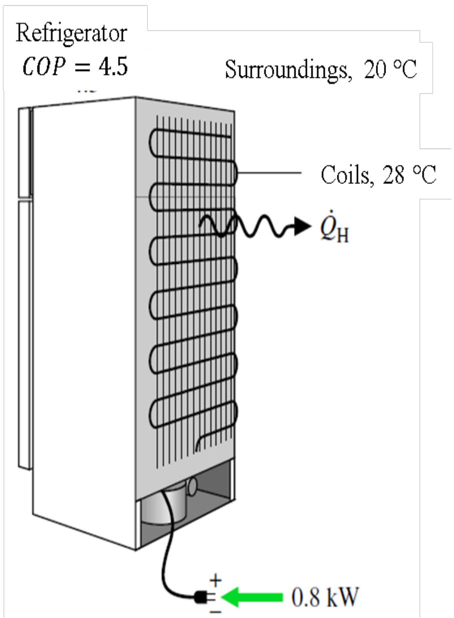
$$W_{min} = \frac{\dot{Q}_L}{COP_R} = \frac{1200/3600}{6.71} = 0.045$$

Exercise 5

The refrigerator shown here operates at steady state with a coefficient of performance of 4.5 and a power input of 0.8 kW. Energy is rejected from the refrigerator to the surroundings at 20 °C by heat transfer from metal coils whose average surface temperature is 28 °C. Determine

a. the rate energy is rejected, in kW,

b. the lowest theoretical temperature inside the refrigerator, in K,



Solution

1. An energy balance for the refrigerator reads

$$\dot{Q}_H = \dot{Q}_L + W_{in}$$

With the COP definition

$$COP_R = \frac{\dot{Q}_L}{W_{in}}$$

Collecting results

$$COP_R = \frac{\dot{Q}_H - W_{in}}{W_{in}} \implies \dot{Q}_H = (1 + COP_R)W_{in}$$

$$\dot{Q}_H = (1 + 4.5)0.8 = 4.4\ kW$$

2. We know that $COP_R \leq COP_{R,Carnot}$

Thus,

$$4.4 \leq \frac{T_L}{T_H - T_L} \implies T_L \gg 0.818T_H = 0.818 \times 293.15 = 239.8\ K$$

The lowest theoretical temperature inside the refrigerator is **239.8 K**.

Chapter II: Thermodynamic Cycle of a Vapor-Compression Refrigeration System

1. Introduction

This chapter focuses on the thermodynamic analysis and practical operation of refrigeration systems, with particular emphasis on the vapor-compression cycle, which is the most widely used technology in industrial and commercial applications. Building on the fundamental concepts introduced in the previous chapter, it aims to develop a deeper understanding of how refrigeration machines operate in real conditions and how their performance can be evaluated.

The chapter begins with the representation of the basic and practical thermodynamic cycles using temperature–entropy (T–s) and pressure–enthalpy (p–h) diagrams, which are essential tools for visualizing heat transfers, work interactions, and irreversibilities. It then examines the thermal balance of the refrigeration cycle and introduces the concept of refrigerants, their classification, and their role in heat transfer processes. Particular attention is given to performance analysis through parameters such as the coefficient of performance (COP) and system efficiency. Finally, the chapter highlights the main industrial applications of refrigeration, illustrating the importance of these systems in food preservation, air conditioning, and various engineering processes.

Overall, this chapter provides the theoretical and practical foundation required to analyze real refrigeration systems and prepares students for more advanced studies in refrigeration technology and thermal system design.

2. Representation of the Basic Thermodynamic Cycle (T–s and p–h Diagrams)

The ideal vapor-compression refrigeration cycle consists of four main thermodynamic transformations. These processes are clearly visualized on the temperature–entropy (T–s) and pressure–enthalpy (P–h) diagrams, which are standard tools for analyzing refrigeration systems. The ideal cycle consists of four main processes (see Figures II.1):

- **Process 1 → 2: Isentropic compression (compressor)**

The refrigerant enters the compressor as a low-pressure vapor coming from the evaporator and is compressed to a high pressure by mechanical work input W. During this process, both temperature and pressure increase significantly. In the ideal cycle, compression is isentropic, which appears as a vertical line on the T–s diagram, while on the P–h diagram it is represented by an increase in pressure and enthalpy.

- **Process 2 → 3: Condensation (Condenser)**

The high-pressure refrigerant vapor flows through the condenser where it releases heat $Q_H$to the surrounding environment and gradually condenses into a saturated liquid. This process occurs approximately at constant pressure. On the T–s diagram, condensation is represented by a horizontal line at high temperature, while on the P–h diagram it appears as a horizontal movement toward lower enthalpy.

- **Process 3 → 4: Expansion (Expansion valve)**

The liquid refrigerant passes through the expansion valve where it undergoes a sudden drop in pressure and temperature. This throttling process is adiabatic and highly irreversible, and the enthalpy remains approximately constant while part of the liquid vaporizes. On the T–s diagram, this transformation is shown as a downward curve with an increase in entropy, and on the P–h diagram it corresponds to a vertical line at constant enthalpy.

- **Process 4 → 1: Evaporation (Evaporator)**

The low-pressure refrigerant enters the evaporator where it absorbs heat $Q_L$from the refrigerated space and evaporates, producing the desired cooling effect. This process occurs at nearly constant pressure and results in an increase in enthalpy as the refrigerant changes from a liquid–vapor mixture to saturated vapor. On the T–s diagram, evaporation appears as a horizontal line at low temperature, while on the P–h diagram it is represented by a horizontal movement toward higher enthalpy.

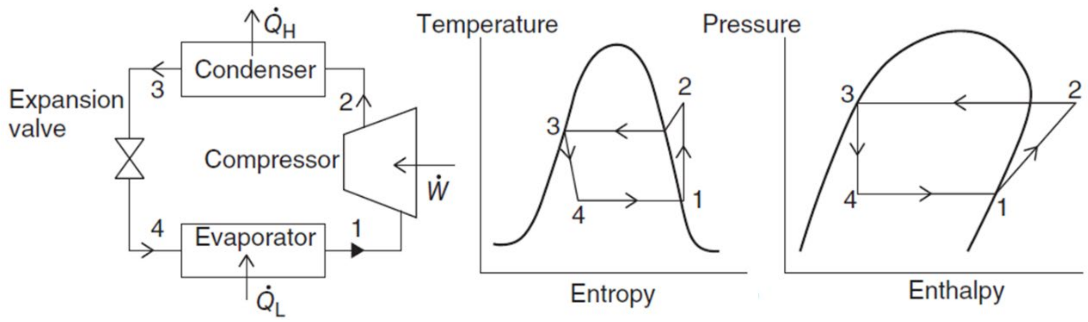


Figure II.1 (a) A basic vapor-compression refrigeration system, (b) its T–s diagram, and (c) its log P–h diagram.

3. Representation of the Practical Thermodynamic Cycle (T–s and p–h Diagrams)

In real refrigeration systems, the thermodynamic cycle differs from the ideal cycle due to irreversibilities, pressure losses, and heat exchanges with the surroundings. The practical cycle is therefore represented on the T–s and P–h diagrams with noticeable deviations from the ideal transformations, reflecting real operating conditions of compressors, condensers, expansion devices, and evaporators (see Figure II.2).

- **Process 1 → 2: Real compression (compressor)**

In practice, the refrigerant vapor entering the compressor is often slightly superheated, and the compression process is not perfectly isentropic due to mechanical friction and heat transfer with the environment. As a result, entropy increases during compression and the discharge temperature is higher than in the ideal case. On the T–s diagram, the compression curve inclines to the right instead of being vertical, while on the P–h diagram the enthalpy rise is greater, indicating higher compressor work.

- **Process 2 → 3: Real condensation (condenser)**

  The refrigerant rejects heat to the surroundings at a pressure slightly lower than the compressor discharge pressure due to pressure drops in pipes and heat exchangers. The temperature of condensation is also slightly higher than the ambient temperature to ensure heat transfer. On the T–s diagram, the condensation path is not perfectly horizontal, and on the P–h diagram a pressure decrease may be observed along the condenser.

- **Process 3 → 4: Real expansion (expansion valve)**

The expansion process remains highly irreversible and occurs with significant pressure losses. Although it is approximately isenthalpic, additional entropy is generated due to internal irreversibilities. On the T–s diagram, the curve shifts further to the right compared with the ideal case, while on the P–h diagram the vertical drop may deviate slightly because of real fluid effects.

Process 4 → 1: Real evaporation (evaporator)

Heat absorption occurs at a pressure slightly lower than the ideal evaporation pressure because of flow resistance in the evaporator and piping. The refrigerant leaving the evaporator is often superheated to ensure that no liquid enters the compressor. On the T–s diagram, the evaporation curve extends into the superheated region, and on the P–h diagram the enthalpy increases beyond the saturated vapor line.

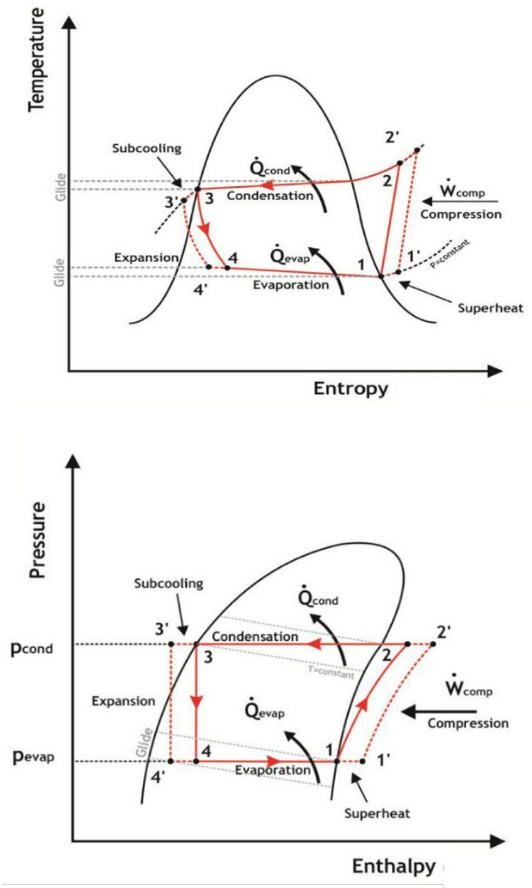


Figure II.2 An actual vapor-compression refrigeration system and its T–s, logP-h diagrams.

4. Energy Balance of the Thermodynamic Cycle

The operation of a vapor-compression refrigeration system is governed by the first law of thermodynamics applied to each component of the cycle and to the overall system. The energy balance makes it possible to determine the refrigeration capacity, compressor power, and heat rejected to the surroundings. For a steady-state refrigeration cycle, the refrigerant circulates continuously through the compressor, condenser, expansion valve, and evaporator. The net energy exchange over one complete cycle must satisfy the global energy conservation principle.

- **Compressor**

  The compressor receives mechanical work and increases the pressure and temperature of the refrigerant. The power input to the compressor corresponds to the increase in refrigerant enthalpy between the compressor inlet and outlet:

$$\dot{W}_{comp} = \dot{m}(h_2 - h_1)$$

where $\dot{m}$is the refrigerant mass flow rate and h_1, $h_2$are the specific enthalpies at the compressor inlet and outlet.

- **Condenser**

  In the condenser, the refrigerant rejects heat to the external environment at approximately constant pressure. The heat rejected is given by:

$$\dot{Q}_H = \dot{m}(h_2 - h_3)$$

This heat corresponds to the sum of the absorbed heat in the evaporator and the compressor work.

- **Expansion valve**

The expansion process is a throttling transformation in which pressure drops significantly while enthalpy remains approximately constant:

$$h_3 \approx h_4$$

No work is produced and heat transfer is usually negligible.

- **Evaporator**

  In the evaporator, the refrigerant absorbs heat from the refrigerated space, producing the desired cooling effect. The refrigeration capacity is expressed as:

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

Overall energy balance of the cycle

For the complete refrigeration cycle, the energy balance gives:

$$\dot{W}_{comp} = \dot{Q}_H - \dot{Q}_L$$

This relation shows that the compressor work input equals the difference between the heat rejected in the condenser and the heat absorbed in the evaporator.

The energy balance is essential for evaluating system performance, determining cooling capacity, and sizing refrigeration equipment for industrial and commercial applications.

5. Performance Study (Coefficient of Performance – Cop)

The performance of a refrigeration system is primarily evaluated using the coefficient of performance (COP), which expresses the effectiveness of the system in producing a refrigeration effect relative to the work supplied to the compressor. Unlike thermal efficiency in heat engines, the COP of a refrigeration system can be greater than unity because the objective is to transfer heat rather than convert it into work.

For a vapor-compression refrigeration cycle, the COP is defined as the ratio of the useful refrigeration effect to the compressor work input:

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{comp}}$$

where $\dot{Q}_L$ is the heat absorbed in the evaporator (refrigeration capacity), $\dot{W}_{comp}$ is the compressor power input.

Using enthalpy values obtained from the P–h diagram or refrigerant tables, the COP can be expressed as:

$$COP_R = \frac{h_1 - h_4}{h_2 - h_1}$$

This expression shows that the system performance depends directly on the refrigeration effect in the evaporator and the work required for compression.

In practical systems, the COP is influenced by several operating parameters, including the evaporation and condensation temperatures, compressor efficiency, pressure losses in the system, and heat transfer conditions in the evaporator and condenser. Real refrigeration systems always exhibit lower COP values than the ideal cycle due to irreversibilities such as friction, non-isentropic compression, and heat exchange with the surroundings.

To quantify the deviation of real compression from ideal isentropic compression, the **isentropic efficiency of the compressor** is defined as:

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

where $h_{2s}$is the enthalpy at the compressor outlet for an ideal isentropic compression to the same discharge pressure, and $h_2$is the actual outlet enthalpy. An isentropic efficiency lower than unity indicates additional work input due to irreversibilities, which directly reduces the COP of the refrigeration system.

6. Refrigerants

1.1 Classification of Refrigerants

Refrigerants can be classified according to their chemical composition and thermodynamic behavior. This classification helps engineers and students identify the main families of refrigerants used in refrigeration systems and understand their typical characteristics and applications. The following flowchart (Figure II. 3) presents a simplified overview of the principal categories of refrigerants, including inorganic fluids, organic fluids, refrigerant mixtures, and hydrocarbons.

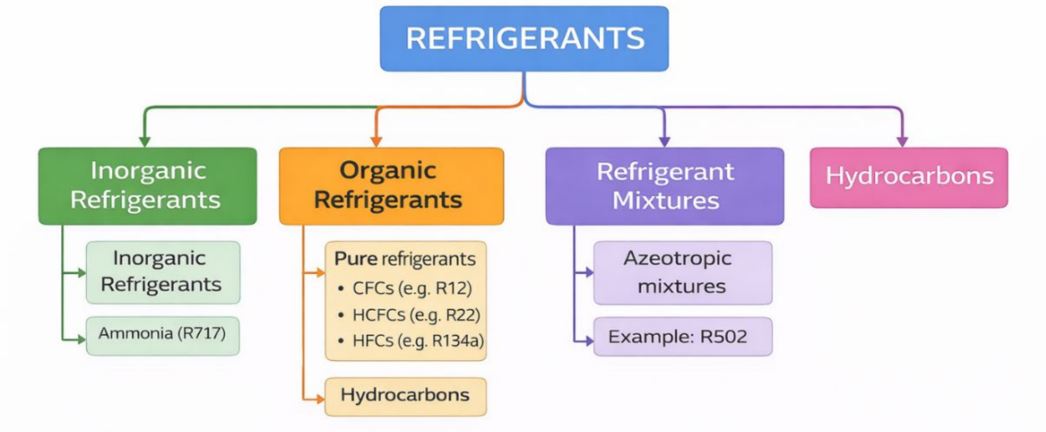


Figure II.3 Classification of Refrigerants by Chemical Family

1.2 Explanation of Refrigerant Classification

Refrigerants are classified according to their chemical composition and thermodynamic behavior.

1.2.1 Inorganic refrigerants (700 series)

These refrigerants consist of simple inorganic compounds. They are widely used in industrial refrigeration due to their excellent thermodynamic properties.

- **Ammonia (R717):** highly efficient, widely used in industry.
- **Carbon dioxide (R744):** environmentally friendly, used in modern systems.

In the 700 series:

- The first digit (7) indicates inorganic refrigerants.
- The remaining digits correspond to the molecular mass.

Example:

R717 → NH_3 (molar mass ≈ 17)

1.2.2 Organic refrigerants

These refrigerants are carbon-based compounds.

a) Pure refrigerants

They are divided into three main families:

- **CFCs:** Chlorofluorocarbons (e.g., R12)
- **HCFCs:** Hydrochlorofluorocarbons (e.g., R22)
- **HFCs:** Hydrofluorocarbons (e.g., R134a)

These fluids follow a specific international numbering system.

Naming convention for pure refrigerants

A refrigerant is designated by **R-xyz**, where:

- x = number of carbon atoms − 1
- y = number of hydrogen atoms + 1
- z = number of fluorine atoms
- Remaining bonds correspond to chlorine atoms
- Letters (a, b, etc.) indicate molecular structure variations

Illustrative Table: Refrigerant Naming Convention

Designation	Units digit (Fluorine atoms)	Tens digit (Hydrogen atoms + 1)	Hundreds digit (Carbon atoms − 1)	Number of chlorine atoms	Chemical formula
R12	2	1 → 0 hydrogen	0 → 1 carbon	2	CCl_2F_2

R22	2	$2 \rightarrow 1$ hydrogen	$0 \rightarrow 1$ carbon	1	$CHClF_2$
R134a	4	$3 \rightarrow 2$ hydrogen	$1 \rightarrow 2$ carbon	0	CH_2FCF_3

b) Refrigerant mixtures

- Azeotropic mixtures (500 series)
  - Behave like pure substances
  - Constant boiling temperature

Example: R502 (mixture of R22 and R115)

- Zeotropic mixtures (400 series)
  - Mixtures with temperature glide
  - Components evaporate at different temperatures
  - Example: R404A (mixture of R143a, R125, R134a)

c) Hydrocarbon refrigerants

Derived from petroleum and natural gas:

- R290: Propane
- R600: Butane
- R600a: Isobutane

- Advantages:
  - Excellent thermodynamic performance
  - Very low environmental impact
- Disadvantage:
  - High flammability (requires safety measures)

1.3 Conclusion

Refrigerants are classified into inorganic fluids, organic fluids, and mixtures.

Their designation follows an international numbering system based on chemical composition.

Understanding refrigerant classification and naming conventions is essential for selecting appropriate working fluids and analyzing refrigeration system performance.

7. Industrial Applications of Refrigeration

Refrigeration is an essential technology in modern industry, enabling temperature control for preservation, manufacturing processes, and human comfort. Its applications extend from domestic refrigeration to large-scale industrial and cryogenic systems. The required temperature level largely determines the type of refrigeration system and refrigerant used.

7.1 Air Conditioning and Human Comfort

Air conditioning represents one of the most widespread applications of refrigeration. It is used in residential buildings, offices, hospitals, laboratories, and industrial facilities to maintain indoor thermal comfort. Typical operating temperatures range between **16 °C and 26 °C**, ensuring suitable environmental conditions for occupants and equipment.

Beyond temperature control, air-conditioning systems regulate humidity, air circulation, and air purity. In industrial environments, such systems are essential for maintaining stable conditions required for precision manufacturing, electronic equipment, and data centers.

7.2 Food Preservation and Food Industry

The food sector is one of the largest users of refrigeration technology. Refrigeration slows down bacterial growth and chemical reactions, thereby preserving food quality and safety.

- **Food refrigeration (0 to 10 °C):**
  Used for short- and medium-term storage of fresh products such as fruits, vegetables, dairy products, and meat.
- **Food freezing (−35 to 0 °C):**
  Ensures long-term preservation by reducing microbial activity and enzymatic reactions. Frozen storage is widely used in food processing industries and distribution chains.
- **Cold chain systems:**
  Refrigeration is essential throughout transportation, storage, and distribution to maintain product quality from production to consumption.

7.3 Industrial and Chemical Applications

Many industrial processes require controlled temperature conditions to ensure proper operation and product quality. Refrigeration is used in:

- Chemical and petrochemical industries for reaction temperature control.
- Plastic and rubber manufacturing processes.
- Thermal treatment and testing of materials.
- Cooling of machinery and industrial equipment.

Industrial refrigeration systems may operate at temperatures ranging from ambient conditions down to approximately **−200 °C**, depending on process requirements.

7.4 Freeze-Drying (Lyophilization)

Freeze-drying is a specialized refrigeration application used to remove moisture from temperature-sensitive products. The process involves freezing the material and then removing ice by sublimation under low pressure.

This technique is widely used in:

- Pharmaceutical and biomedical industries
- Food preservation (instant coffee, dehydrated foods)
- Biotechnology and research laboratories

Typical operating temperatures range from **−80 °C to −30 °C**.

7.5 Cryogenic Applications

Cryogenics deals with the production and use of extremely low temperatures, generally below **−150 °C**. Refrigeration at cryogenic levels is used in advanced industrial and scientific applications such as:

- Liquefaction of natural gas for storage and transport (LNG)
- Air separation to produce oxygen, nitrogen, and argon
- Liquefaction of hydrogen and helium
- Superconductivity and aerospace applications

Cryogenic refrigeration requires specialized systems and refrigerants and often involves multi-stage or cascade refrigeration cycles.

7.6 Importance of Industrial Refrigeration

Industrial refrigeration is crucial for ensuring product quality, safety, and efficiency in modern engineering systems. It supports food security, healthcare, manufacturing, and energy industries while improving productivity and maintaining controlled environmental conditions.

The continuous development of refrigeration technologies aims to increase energy efficiency, reduce environmental impact, and meet the growing demand for reliable temperature-controlled processes across all industrial sectors.

8. Chapter Summary

This chapter presented the fundamental principles and practical aspects of vapor-compression refrigeration systems, focusing on their thermodynamic representation, performance evaluation, refrigerants, and industrial applications. It began with the description of the basic thermodynamic cycle using temperature–entropy (T–s) and pressure–enthalpy (P–h) diagrams, which are essential tools for analyzing energy transfers and transformations within refrigeration systems. The four main processes of the cycle (compression, condensation, expansion, and evaporation) were examined to illustrate how mechanical work is used to extract heat from a low-temperature region and reject it to a high-temperature environment.

The chapter then introduced the practical thermodynamic cycle, highlighting the differences between ideal and real system operation. In real systems, irreversibilities such as pressure drops, heat losses, and non-isentropic compression lead to deviations from the ideal cycle and reduce system efficiency. These effects were represented on T–s and P–h diagrams to demonstrate their impact on system performance.

An energy balance analysis of the refrigeration cycle was also presented based on the first law of thermodynamics. Expressions for compressor power, heat rejected in the condenser, and refrigeration capacity in the evaporator were established using refrigerant enthalpy values. This analysis forms the basis for evaluating and designing refrigeration systems.

System performance was evaluated using the coefficient of performance (COP), defined as the ratio of refrigeration effect to compressor work input. Factors affecting COP—such as operating temperatures, pressure losses, and compressor efficiency—were analyzed, and the concept of isentropic compression efficiency was introduced to quantify deviations from ideal compression.

The chapter further discussed refrigerants, including their classification into inorganic refrigerants, organic refrigerants, hydrocarbon refrigerants, and refrigerant mixtures. The international naming convention for refrigerants was explained, allowing identification of their chemical composition from their designation. Selection criteria such as thermodynamic properties, environmental impact, safety, and cost were also considered.

Finally, the chapter reviewed the major industrial applications of refrigeration, including air conditioning, food preservation, industrial cooling, freeze-drying, and cryogenic processes. These applications demonstrate the essential role of refrigeration in modern industry, energy systems, and daily life. Overall, the chapter provided a comprehensive understanding of refrigeration system operation and performance, forming a solid foundation for advanced study and practical engineering applications.

9. Solved Exercises

Things Engineers Think About

Note to students

Some questions in this section are not explicitly answered in the course document. They are intended to complement the course by encouraging application of concepts and practical engineering reasoning. The answers provided therefore extend the lecture material and help develop understanding of real refrigeration system operation.

1. A vapor-compression refrigeration system operates between an evaporating temperature of −5 °C and a condensing temperature of 35 °C. How would increasing the condensing temperature affect the COP and compressor work? Explain physically.

2. When analyzing a refrigeration cycle on a P–h diagram, which enthalpy differences correspond to compressor work and refrigeration effect? How can these be used to estimate system performance?

3. In practice, the vapor entering the compressor is often superheated. Why is superheating necessary, and how does it influence system efficiency and compressor reliability?

4. Compare the ideal and practical vapor-compression cycles. What are the main sources of irreversibility, and how do they affect system performance?

5. A refrigeration system shows a significant pressure drop in the evaporator. What impact will this have on refrigeration capacity and COP? Suggest engineering solutions to reduce this effect.

6. Why is the expansion valve considered an irreversible component of the refrigeration cycle? What would be the theoretical benefit of replacing it with an expansion turbine?

7. An engineer must select a refrigerant for a new industrial refrigeration plant. What criteria should be considered regarding thermodynamic performance, environmental impact, safety, and cost?

8. Explain how the first law of thermodynamics applies to each component of a vapor-compression refrigeration system. Which component consumes work and which components exchange heat?

9. How does the evaporation temperature influence refrigeration capacity and compressor power consumption? Provide a qualitative explanation.

10. In industrial applications, multi-stage or cascade refrigeration cycles are sometimes used. Why are these configurations preferred at very low temperatures?

11. A refrigeration system operates continuously in a food processing plant. What design considerations are necessary to ensure reliability, energy efficiency, and safety?

12. Compare refrigeration and heat pump operation. What is the difference in the desired effect, and how does this influence performance evaluation?

13. Why are hydrocarbon refrigerants increasingly used in modern refrigeration systems despite their flammability? Discuss advantages and limitations.

14. In cryogenic applications, extremely low temperatures are required. What technical challenges arise when designing refrigeration systems for such conditions?

15. If compressor isentropic efficiency decreases over time, what symptoms would appear in system performance, and what engineering actions should be taken?

Answers

1. Effect of increasing condensing temperature

An increase in condensing temperature raises the compressor discharge pressure, which increases compressor work and reduces COP. The system must reject heat at a higher temperature, making heat transfer more difficult and lowering efficiency.

2. Use of enthalpy differences on the P–h diagram

The enthalpy rise across the compressor $(h_2 - h_1)$represents compressor work, while the enthalpy difference across the evaporator $(h_1 - h_4)$represents the refrigeration effect. These values allow estimation of COP and cooling capacity.

3. Reason for superheating before compression

Superheating ensures that only vapor enters the compressor, preventing liquid droplets that could damage it. However, excessive superheating may reduce COP because it increases compressor work without significantly increasing refrigeration effect.

4. Ideal vs practical cycles

The ideal cycle assumes isentropic compression and no pressure losses. In real systems, irreversibilities such as friction, heat losses, and pressure drops increase entropy, require more work, and reduce COP.

5. Impact of pressure drop in the evaporator

A pressure drop lowers the evaporation temperature, reducing refrigeration capacity and COP. Engineers may reduce this effect by improving heat exchanger design, minimizing pipe length, or increasing pipe diameter.

6. Irreversibility of the expansion valve

The expansion valve produces a throttling process with entropy generation and no work recovery. A turbine could recover useful work and increase efficiency, but it is usually impractical due to cost and complexity.

7. Criteria for refrigerant selection

Engineers consider thermodynamic efficiency, environmental impact (ODP, GWP), toxicity, flammability, compatibility with materials, availability, and cost.

8. Application of the first law

The compressor consumes work, the evaporator absorbs heat from the cold space, and the condenser rejects heat to the surroundings. The overall energy balance satisfies $W = Q_H - Q_L$.

9. Effect of evaporation temperature

Higher evaporation temperatures increase COP and reduce compressor work because the pressure ratio decreases. Lower evaporation temperatures increase energy consumption.

10. Use of multi-stage or cascade cycles

These systems reduce the compression ratio per stage, improve efficiency, and allow operation at very low temperatures where single-stage systems become inefficient or impractical.

11. Design considerations for industrial systems

Engineers must ensure reliability, energy efficiency, proper refrigerant selection, safety controls, and ease of maintenance. Continuous monitoring and redundancy may be required.

12. Refrigerator vs heat pump

A refrigerator aims to remove heat from a cold space, while a heat pump aims to supply heat to a warm space. Performance is measured differently: COP_R for refrigeration and COP_{HP} for heating.

13. Use of hydrocarbon refrigerants

Hydrocarbons offer excellent thermodynamic performance and very low environmental impact. However, their flammability requires strict safety measures and limits their use to specific applications.

14. Challenges in cryogenic systems

Extremely low temperatures require multi-stage cycles, special materials, advanced insulation, and careful control of heat leaks and pressure conditions.

15. Effect of reduced compressor isentropic efficiency

Lower efficiency increases compressor work, reduces COP, raises discharge temperature, and may indicate wear or malfunction. Engineers should inspect, maintain, or replace compressor components and optimize operating conditions.

Exercise 1

Refrigerant 134a is the working fluid in an ideal vapor-compression refrigeration cycle that communicates thermally with a cold region at 0 °C and a warm region at 26 °C. Saturated vapor enters the compressor at 0 °C and saturated liquid leaves the condenser at 26°C. The mass flow rate of the refrigerant is 0.08 kg/s. Determine

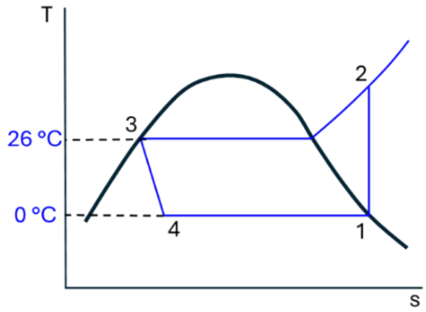
a. the compressor power, in kW,

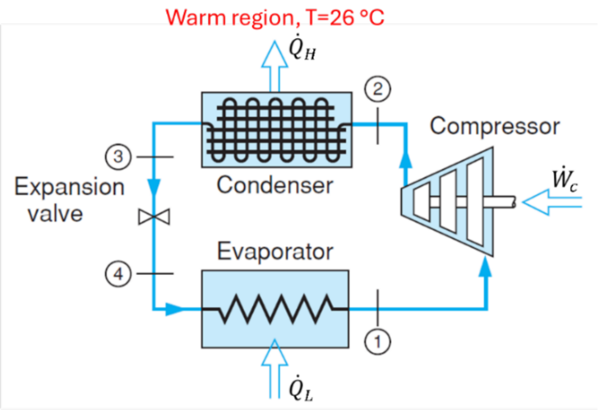
b. the refrigeration capacity, in tons,

c. the coefficient of performance,

d. the coefficient of performance of a Carnot refrigeration cycle operating between warm and cold regions at 26 and 0 °C, respectively.

Solution





From thermodynamic tables (R134a) :

- **State 1 (saturated vapor at 0 °C):**

  $h_1 = 247.23 \text{ kJ/kg}, s_1 = 0.9190 \text{ kJ/kg. K}$

- **State 2 (after isentropic compression to condenser pressure at 26 °C):**

  $p_2 = 6.853$ bar,$s_2 = s_1$

  From superheated vapor tables:

  $h_2 = 264.7 \text{ kJ/kg}$

- **State 3 (saturated liquid at 26 °C):**

  $h_3 = 85.75 \text{ kJ/kg}$

- **State 4 (after throttling valve):**

  Isenthalpic expansion:

$h_4 = h_3 = 85.75 \text{ kJ/kg}$

a) Compressor power input

$$\dot{W}_c = \dot{m}(h_2 - h_1)$$

$$\dot{W}_c = 0.08(264.7 - 247.23)$$

$$\dot{W}_c = 1.4 \text{ kW}$$

Refrigeration capacity

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

$$\dot{Q}_L = 0.08(247.23 - 85.75)$$

$$\dot{Q}_L = 12.9 \text{ kW}$$

b) Coefficient of performance

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c}$$

$$COP_R = \frac{12.9}{1.4}$$

$$COP_R = 9.2$$

c) For a **Carnot vapor refrigeration cycle** operating at $T_H = 26\,°C = 293.15\,K$ and $T_L = 0\,°C = 273.15\,K$, the coefficient of performance is

$$COP_{R,Carnot} = \frac{\mathrm{T}_L}{\mathrm{T}_H - \mathrm{T}_c} = \frac{273.15}{299.15 - 273.15}$$

$$COP_{R,Carnot} = 10.5$$

Exercise 2

Consider again Exercise 1, but with the following modifications:

Saturated vapor enters the compressor at −10 °C. Superheated vapor enters the condenser at a pressure of 9 bar. The liquid leaves the condenser at a temperature of 30 °C. After expansion, the vapor quality of the refrigerant mixture is 0.2667. The isentropic efficiency of the compressor is 80%.

Determine:

a. The compressor power input;

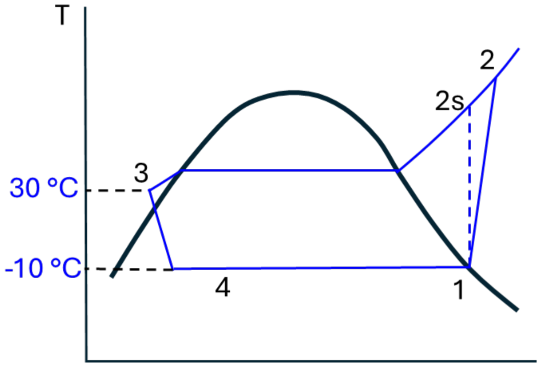
b. The refrigeration capacity;

c. The coefficient of performance (COP).

Compare the results obtained with those of Exercise 1.

Solution

The actual cycle is shown il the T-s diagram



Thermodynamic properties obtained from refrigerant tables:

State 1 (saturated vapor at −10 °C):

$h_1 = 241.35 \text{ kJ/kg}, s_1 = 0.9253 \text{ kJ/kg. K}$

State 2s (isentropic compression to $p_2 = 9$bar):

$s_{2s} = s_1$

From superheated vapor tables:

$h_{2s} = 272.35 \text{ kJ/kg}$

The compressor isentropic efficiency is:

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \Longrightarrow 0.8 = \frac{272.35 - 241.35}{h_2 - 241.35}$$

$h_2 = 280.15 \text{ kJ/kg}$

State 3 (liquid at condenser outlet):

The enthalpy is equal to that at state 4 after throttling:

$h_3 = h_4$

State 4 (after expansion valve):

Vapor quality:

$x_4 = 0.2667$

$h_4 = (1 - x)h_f + xh_g$

With:

$h_f = 36.97 \text{ kJ/kg}, h_g = 241.35 \text{ kJ/kg}$

$h_4 = 91.48 \text{ kJ/kg}$

a) Compressor power input

$$\dot{W}_c = \dot{m}(h_2 - h_1)$$

$$\dot{W}_c = 0.08(280.15 - 241.35)$$

$$\dot{W}_c = 3.1 \text{ kW}$$

b) Refrigeration capacity

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

$$\dot{Q}_L = 0.08(241.35 - 91.48)$$

$$\dot{Q}_L = 11.99 \text{ kW}$$

c) Coefficient of performance

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c}$$

$$COP_R = \frac{11.99}{3.1}$$

$$COP_R = 3.87$$

Compared with Exercise 1, the compressor work increases and the COP decreases due to non-ideal compression and modified operating conditions.

Exercise 3

Refrigerant 22 enters the compressor of an ideal vapor-compression refrigeration system as saturated vapor at -40°C with a volumetric flow rate of 15 m^3/min. The refrigerant leaves the condenser at 19°C, 9 bar. Determine

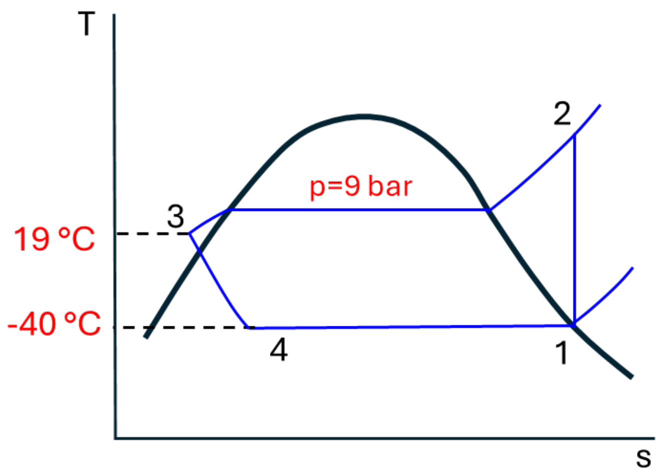
a. the compressor power, in kW.

b. the refrigerating capacity, in tons.

c. the coefficient of performance

Solution

The T-s diagram is shown below



Thermodynamic properties of R22 (from tables):

State 1 (sat. vapor at −40 °C):

$v_1 = 0.2052\ m^3/kg, h_1 \approx 233.27$ kJ/kg, $s_1 = 1.0005 \frac{kJ}{kg}.K$

Mass flow rate:

$\dot{m} = \frac{\dot{V}}{v_1} = \frac{15/60}{0.2052} = 1.22$ kg/s

State 2 (after isentropic compression to 9 bar):

From superheated tables at $p_2 = 9$bar and $s_2 = s_1$:

$h_2 = 287.54$ kJ/kg

State 3 (sat. liquid at 9 bar):

$h_3 = 67.9$ kJ/kg

State 4 (after expansion valve):

$h_4 = h_3 = 67.9$ kJ/kg

a) Compressor power

$$\dot{W}_c = \dot{m}(h_2 - h_1)$$
$$\dot{W}_c = 1.22(287.54 - 233.27)$$
$$\dot{W}_c \approx 66.21 \text{ kW}$$

---

b) Refrigerating capacity

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$
$$\dot{Q}_L = 1.22(233.27 - 67.9)$$
$$\dot{Q}_L \approx 201.75 \text{ kW}$$

Coefficient of performance

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c}$$
$$COP_R = \frac{201.75}{66.21}$$
$$COP_R = 3.05$$

Exercise 4

In a vapor-compression refrigeration cycle, ammonia exits the evaporator as saturated vapor at -22 °C. The refrigerant enters the condenser at 16 bar and 160 °C, and saturated liquid exits at 16 bar. There is no significant heat transfer between the compressor and its surroundings, and the refrigerant passes through the evaporator with a negligible change in pressure. If the refrigerating capacity is 150 kW, determine

a. the mass flow rate of the refrigerant, in kg/s,

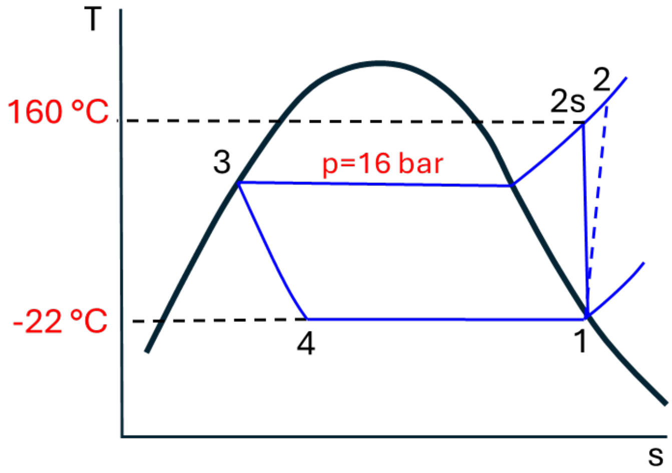
b. the power input to the compressor, in kW,

c. the coefficient of performance,

d. the isentropic compressor efficiency.

Solution

The T-s diagram is shown below



Thermodynamic properties of ammonia (from tables):

State 1 (saturated vapor at −22 °C):

$h_1 = 1415.08\ \mathrm{kJ/kg}$

State 2s (superheated vapor at $p_2 = 16\ bar$ **and** $T_2 = 160\ °C$ **):**

$h_2 = 1798.45\ \mathrm{kJ/kg}$

State 3 (saturated liquid at $p_2 = 16\ bar$**):**

$h_3 = 376.46\ \mathrm{kJ/kg}$

State 4 (after expansion valve):

$h_3 = h_4$

a) Mass flow rate of the refrigerant

Refrigerating capacity: $\dot{Q}_L = \dot{m}(h_1 - h_4)$

$$\Rightarrow \dot{m} = \frac{\dot{Q}_L}{h_1 - h_4}$$

$$\dot{m} = \frac{150}{1415.08 - 376.46} = 0.1444 \text{ kg/s}$$

$$\dot{m} = 0.1444 \text{ kg/s}$$

b) Power input to the compressor

$$\dot{W}_c = \dot{m}(h_2 - h_1)$$

$$\dot{W}_c = 0.1444(1798.45 - 1415.08)$$

$$\dot{W}_c = 55.36 \text{ kW}$$

c) Coefficient of performance

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c}$$

$$COP_R = \frac{150}{55.36}$$

$$COP_R = 2.71$$

d) isentropic compressor efficiency

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$s_{2s} = s_1 = 5.6457 \text{ kJ/kg}.K \implies h_{2s} = 1755.38 \text{ kJ/kg}$

$$\eta_c = \frac{1755.38 - 1415.08}{1798.45 - 1415.08}$$

$$\eta_c = 88.8\%$$

Exercise 5

R-134a enters the compressor of a vapor-compression refrigeration cycle at 120 kPa as a saturated vapor and leaves at 900 kPa and 75 °C. The refrigerant leaves the condenser as a saturated liquid. The rate of cooling provided by the system is 18000 Btu/h. Determine

a. the mass flow rate of R-134a,

b. the COP of the cycle,

c. the COP of the cycle if the expansion valve is replaced by an isentropic turbine (Do you recommend such a replacement for refrigeration systems?),

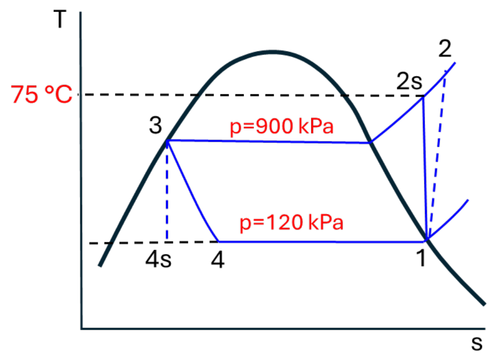
d. the COP if the evaporator pressure is 160 kPa and other values remain the same,

e. the COP if the condenser pressure is 800 kPa and other values remain the same.

Solution

Note: $1\ kW = 3412 \text{ Btu/h} \implies 1\ Btu/h = 0.000293071 \text{ kW}$

The T-s diagram is shown below



Thermodynamic properties of R134a (from tables):

State 1 (saturated vapor at 120 kPa):

$h_1 = 233.86\ \mathrm{kJ/kg}$

State 2 (superheated vapor at $p_2 = 900\ \mathrm{kPa}$ **and** $T_2 = 75\ °\mathrm{C}$ **):**

$h_2 = 309.28\ \mathrm{kJ/kg}$

State 3 (saturated liquid at $p_2 = 900\ \mathrm{kPa}$**):**

$h_3 = 99.56\ \mathrm{kJ/kg}$

State 4 (after expansion valve):

$h_3 = h_4$

State 4s (after isentropic turbine, $\boldsymbol{p_1 = 120\ \mathrm{kPa}}$**):**

$s_{4s} = s_3 = 0.3656\ \mathrm{kJ/kg}.K$

Compute quality x_{4s}

$$s_{4s} = x_{4s}s_g + (1 - x_{4s})s_f \implies x_{4s} = \frac{s_{4s} - s_f}{s_g - s_f} = \frac{0.3656 - 0.0879}{0.9354 - 0.0879} \approx 33\%$$

$$h_{4s} = x_{4s}h_{4g} + (1 - x_{4s})h_{4f}$$

$$h_{4s} = 0.33 \times 233.86 + (1 - 0.33) \times 21.32$$

$$h_{4s} = 91.46\ \mathrm{kJ/kg}$$

a) Mass flow rate of the refrigerant

Refrigerating capacity: $\dot{Q}_L = \dot{m}(h_1 - h_4)$

$$\implies \dot{m} = \frac{\dot{Q}_L}{h_1 - h_4}$$

$$\dot{m} = \frac{18000 \times 0.000293071}{233.86 - 99.56}$$

$$\dot{m} = 0.039\ \mathrm{kg/s}$$

b) COP of the cycle

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c}$$

$$\dot{W}_c = \dot{m}(h_2 - h_1)$$

$$\dot{W}_c = 0.039(309.28 - 233.86) = 2.94 \text{ kW}$$

$$COP_R = \frac{18000 \times 0.000293071}{2.94}$$

$$COP_R = 1.79$$

c) COP of the cycle if the expansion valve is replaced by an isentropic turbine

In this case, the turbine produces a power

$$\dot{W}_t = \dot{m}(h_3 - h_{4s})$$

$$\dot{W}_t = 0.039 \times (99.56 - 91.46) = 0.32 \text{ kW}$$

$$\dot{W}_{net} = \dot{W}_c - \dot{W}_t = 2.94 - 0.32 = 2.62 \text{ kW}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}} = \frac{18000 \times 0.000293071}{2.62}$$

$$COP_R = 2.01$$

The COP increases by 12.3 % by replacing the expansion valve by a turbine. This replacement makes thermodynamic sense since it decreases the work requirement and thus increases the COP. However, this is not practical for household refrigerators and most other refrigeration systems. In natural gas liquefaction plants, the liquefied natural gas is expanded by cryogenic turbines, which is proven to be feasible.

COP if the evaporator pressure is 160 kPa and other values remain the same

$h_1 = 237.97 \text{ kJ/kg}$

$h_4 = h_3 = 99.56 \text{ kJ/kg}$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_R = \frac{237.97 - 99.56}{309.28 - 237.97}$$

$$COP_R = 1.94$$

Increasing the evaporator pressure from 120 to 160 kPa (increasing the evaporating temperature from −22.3 to −15.6 °C) increases the COP from 1.79 to 1.94, an increase of 8.4 %.

d) COP if the condenser pressure is 800 kPa and other values remain the same

$h_2 = 310.75 \text{ kJ/kg}$

$h_4 = h_3 = 93.42 \text{ kJ/kg}$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_R = \frac{233.86 - 93.42}{310.75 - 233.86}$$

$$COP_R = 1.83$$

Decreasing the condenser pressure from 900 to 800 kPa (decreasing the condensing temperature from 35.5 to 31.3 °C) increases the COP from 1.79 to 1.83, an increase of 2.2%.

Exercise 6

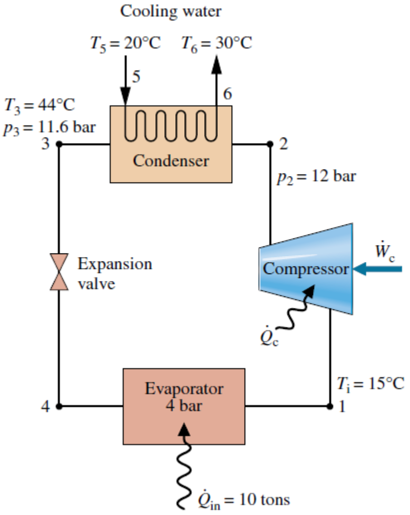
A vapor-compression refrigeration system with a capacity of 10 tons has Refrigerant 134a as the working fluid. Information and data for the cycle are provided in the Figure below. The compression process is internally reversible and can be modeled by $pv^{1.01} = constant$. The condenser is water-cooled, with water entering and leaving with a negligible change in pressure. Heat transfer from the outside of the condenser can be neglected. Determine

a. the mass flow rate of refrigerant, in kg/s,

b. the power input and the heat transfer rate for the compressor, each in kW,

c. the coefficient of performance,

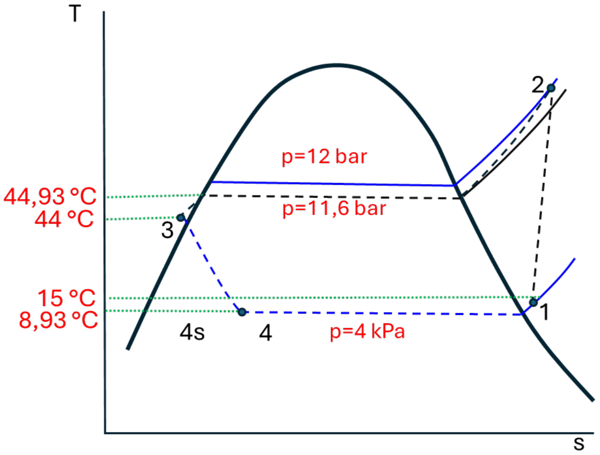
d. the mass flow rate of the cooling water, in kg/s.



Solution

Note : 1 Ton ≈ 3.517 kW

The T-s diagram is shown below



Thermodynamic properties of R134a (from tables):

State 1 (superheated vapor at 4 bar and 15°C):

$h_1 = 258.16$ kJ/kg, $v_1 = 0.05258$ m^3/kg

State 2 (superheated vapor at 12 bar**):**

$p_1 v_1^{1.01} = p_2 v_2^{1.01} \Longrightarrow v_2 = 0.01772 \text{ m}^3/\text{kg}$

$\Longrightarrow h_2 = 281.33 \text{ kJ/kg}$

State 3 (subcooled liquid at 11.6 bar and 44°C):

$h_3 = 112.22$ kJ/kg

State 4 (after expansion valve):

$h_3 = h_4$

a) Mass flow rate of refrigerant, in kg/s

Refrigerating capacity: $\dot{Q}_L = \dot{m}(h_1 - h_4)$

$$\Longrightarrow \dot{m} = \frac{\dot{Q}_L}{h_1 - h_4}$$

$$\dot{m} = \frac{10 \times 3.517}{258.16 - 112.22}$$

$$\dot{m} = 0.241 \text{ kg/s}$$

b) The power input and the heat transfer rate for the compressor, each in kW:

Applying the first law of thermodynamics to the compressor:

$$q_c - w_c = h_2 - h_1$$

$$q_c - w_c = 281.33 - 258.16 = 23.18 \text{ kJ/kg}$$

Work of compression can be evaluated as follows:

$w_c = \int_1^2 v dp$, with $pv^{1.01} = \text{const.} = p_1 v_1^{1.01} = p_2 v_2^{1.01}$

$$\Rightarrow \ w_c = -1.01 \times \int_{v_1}^{v_2} const. \frac{dv}{v^{1.01}} \ \Rightarrow w_c = -\frac{1.01}{0.01}[p_2 v_2 - p_1 v_1]$$

$$w_c = -\frac{1.01}{0.01}[12 \times 0.01772 - 4 \times 0.05258] \times 10^5$$

$$w_c = -23.432 \text{ kJ/kg}$$

the power input for the compressor is then:

$$\dot{W}_c = \dot{m}|w_c| = 0.241 \times 23.432$$

$$\dot{W}_c = 5.647 \text{ kW}$$

The heat transfer rate for the compressor is:

$$q_c - (-23.432) = 23.17 \Rightarrow q_c = -0.262 \text{ kJ/kg}$$

$$\dot{Q}_c = \dot{m} q_c = 0.241 \times 0.262 = 0.0631 \text{ kW}$$

c) The coefficient of performance

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_c} = \frac{10 \times 3.517}{5.647}$$

$$COP_R = 6.228$$

d) mass flow rate of the cooling water, in kg/s.

Evaluating the water mass flow from an energy balance:

$$\dot{m}_w C_w (T_{out} - T_{in}) = \dot{m}_R (h_2 - h_3)$$

$$\dot{m}_w = \frac{\dot{m}_R (h_2 - h_3)}{C_w (T_{out} - T_{in})}$$

$$\dot{m}_w = \frac{0.241 \times (281.33 - 112.22)}{4.184 \times (30 - 20)}$$

$$\dot{m}_w = 0.974 \text{ kg/s}$$

Chapter III: Components of a Vapor-Compression Refrigeration System

1. Introduction

This chapter introduces the main components of a vapor-compression refrigeration machine and explains how they work together to produce cooling in practical applications. Refrigeration is achieved by removing heat from a low-temperature space and transferring it to a higher-temperature environment through four basic processes: evaporation, compression, condensation, and expansion. These processes are carried out by the evaporator, compressor, condenser, and expansion device, which form the core of any refrigeration system. Each component has a specific role: the evaporator absorbs heat from the space to be cooled, the compressor raises the pressure and temperature of the refrigerant, the condenser rejects heat to the surroundings, and the expansion device reduces the refrigerant pressure to restart the cycle. Understanding how these components operate and interact is essential for energy engineering students, as it provides the fundamental basis for studying refrigeration systems, evaluating their performance, and recognizing their importance in applications such as air conditioning, food preservation, and industrial cooling.

2. Main Components of a Vapor-Compression System

As introduced in the previous chapter, the vapor-compression refrigeration cycle is based on four fundamental processes: compression, condensation, expansion, and evaporation. In practice, these thermodynamic processes are carried out by physical devices that form the main components of a refrigeration system. The operation of the system therefore depends not only on the cycle itself, but also on the performance and interaction of these components.

A typical vapor-compression system consists of four essential elements: the **compressor**, **condenser**, **expansion device**, and **evaporator**. The refrigerant circulates through these components in a closed loop, undergoing successive changes in pressure, temperature, and phase. As illustrated in Figure III.1, the cycle begins at the compressor inlet (state 1), where low-pressure refrigerant vapor is compressed to a high-pressure, high-temperature state. The refrigerant then flows into the condenser, where it rejects heat to the surrounding environment and condenses into a liquid. Next, the liquid refrigerant passes through the expansion device, where its pressure and temperature decrease significantly. Finally, the low-pressure refrigerant enters the evaporator, where it absorbs heat from the space to be cooled and vaporizes before returning to the compressor to complete the cycle.

While the thermodynamic behavior of the cycle has already been studied, the focus here is on the actual components that make this process possible. Each component has a specific role and operating characteristics that influence system performance, energy consumption, and reliability. In addition to the four main elements, refrigeration systems also include auxiliary devices (e.g., piping, receivers, and control units) that ensure proper circulation of the refrigerant and stable operation of the system. Understanding these components is essential for moving from theoretical cycle analysis to the practical study and design of refrigeration systems.

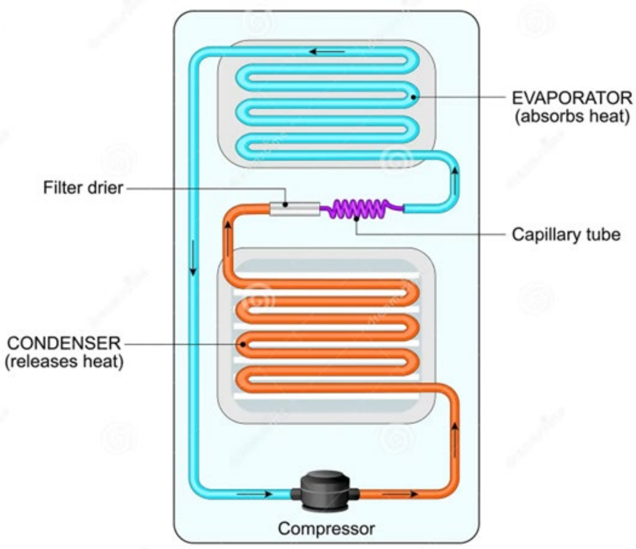


Figure III.1 Cross-sectional diagram of a domestic refrigerator illustrating the main components of a vapor-compression refrigeration system.

3. Compressors

The compressor is the key component of a vapor-compression refrigeration system and is often described as its "heart," since it ensures the continuous circulation of the refrigerant and enables the thermodynamic cycle to operate. It receives the refrigerant as a low-pressure vapor at the compressor inlet (state 1) and compresses it to a higher pressure and temperature before it enters the condenser. This pressure rise is essential for allowing heat rejection at a temperature higher than the surroundings.

3.1. Role, operation, and importance of the compressor

In a refrigeration system, the compressor performs two fundamental functions. First, it draws refrigerant vapor from the evaporator and circulates it through the system, maintaining the required low pressure for heat absorption. Second, it compresses the refrigerant vapor, increasing its pressure and temperature so that it can release heat in the condenser. During this process, the refrigerant changes from a saturated or slightly superheated vapor to a high-pressure superheated vapor, linking the low-pressure side of the system (evaporator) to the high-pressure side (condenser).

The compressor has a major influence on the overall performance of a refrigeration system. It consumes most of the system's energy, determines the refrigerant mass flow rate, and strongly affects the coefficient of performance and operating reliability. For effective operation, compressors must therefore meet several requirements, including high efficiency, reliable performance, long service life, ease of maintenance, compact size, and reasonable cost.

The selection of a suitable compressor depends on a number of practical factors such as the type of refrigerant, required cooling capacity, compression ratio, volumetric flow rate, and operating pressures and temperatures. Understanding these aspects is essential for evaluating system performance and ensuring proper operation in real refrigeration applications.

3.2. Classification of compressors

Refrigeration compressors are commonly classified into two main groups (see Figure III.2):

- **Positive displacement compressors**: increase pressure by reducing the volume of refrigerant.
- **Dynamic compressors**: increase pressure by accelerating the refrigerant using rotating elements.

Typical compressor types are illustrated in Figure III.3, which shows examples such as reciprocating, rotary vane, screw, turbine, and centrifugal compressors. These designs are selected according to system size, operating conditions, and application requirements.

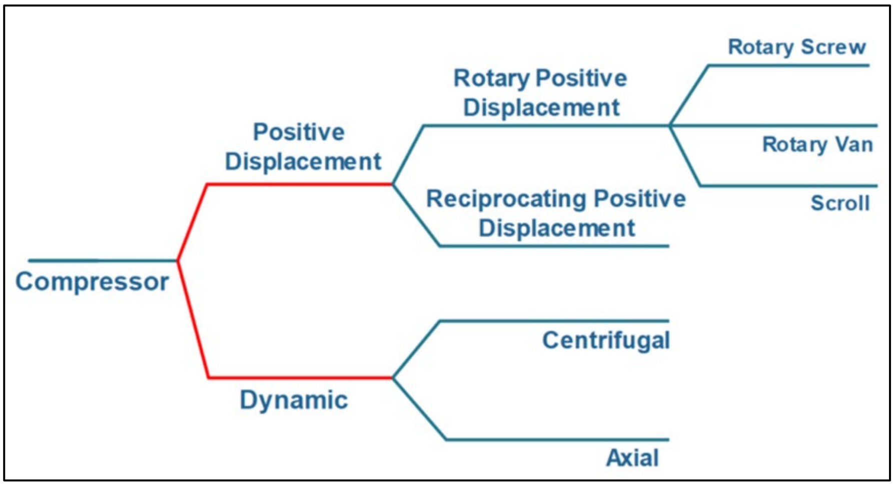


Figure III.2 Classification of refrigeration compressors.

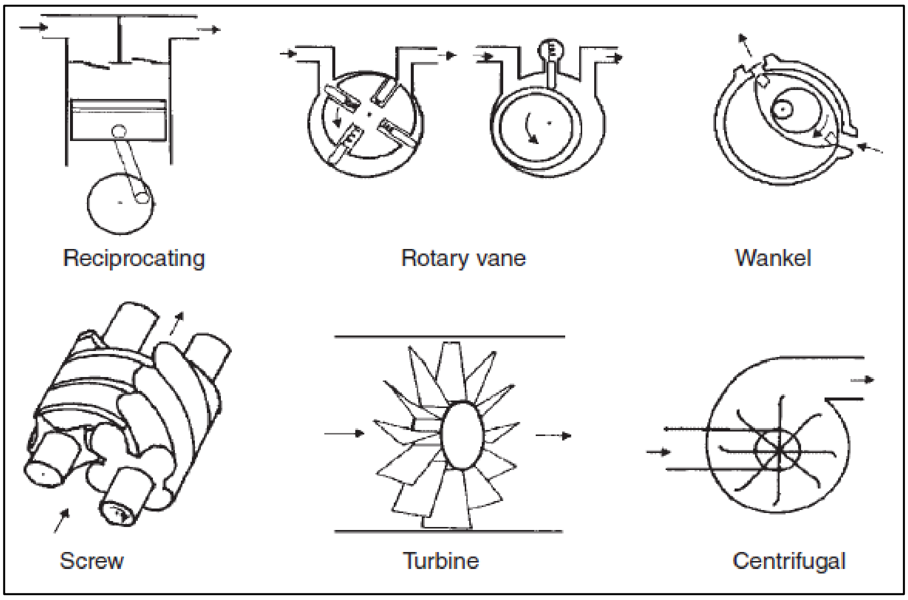


Figure III.3 Types of compressors commonly used in refrigeration systems.

3.3. Construction types

Compressors can also be categorized according to their construction:

- **Hermetic compressors**: motor and compressor enclosed in a sealed casing; widely used in domestic refrigerators and small systems (Figure III.4).

- **Semi-hermetic compressors**: enclosed but accessible for maintenance; common in commercial refrigeration (Figure III.5).
- **Open compressors**: motor and compressor separated and externally driven; used in large industrial systems (Figure III.6).

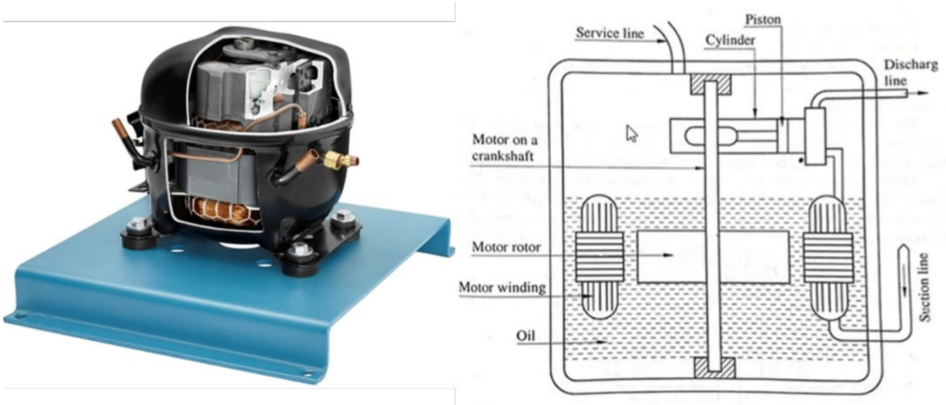


Figure III. 4 Hermetic refrigeration compressor and its internal structure.

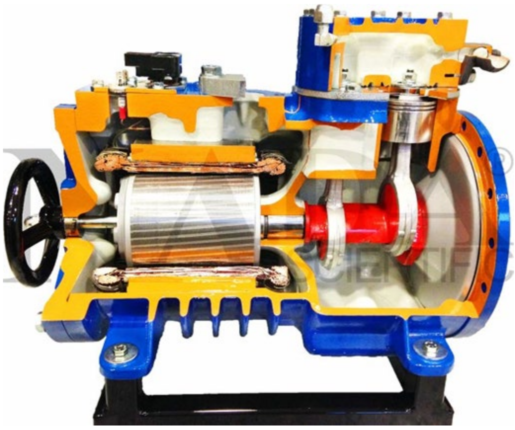


Figure III.5 Cutaway view of a semi-hermetic reciprocating compressor

Figure III.6 Open Drive reciprocating Compressor

3.4. Positive displacement compressors

Positive displacement compressors are widely used in refrigeration because they provide high pressure increases for moderate flow rates. These compressors are generally classified into two main categories: reciprocating and rotary.:

- **Reciprocating compressors** operate using a piston–cylinder mechanism and are suitable for a wide range of refrigeration capacities, from small domestic units to larger installations (Figures III.4, III.5, and III.6).
- **Rotary compressors** use rotating elements to compress the refrigerant and are commonly applied in refrigeration and air-conditioning systems. This category includes several types, such as **rotary vane compressors** (Figure III.7), **screw compressors** used in large commercial and industrial applications (Figure III.8), and **scroll compressors**, which are compact and efficient and widely used in modern air-conditioning systems (Figure III.9).

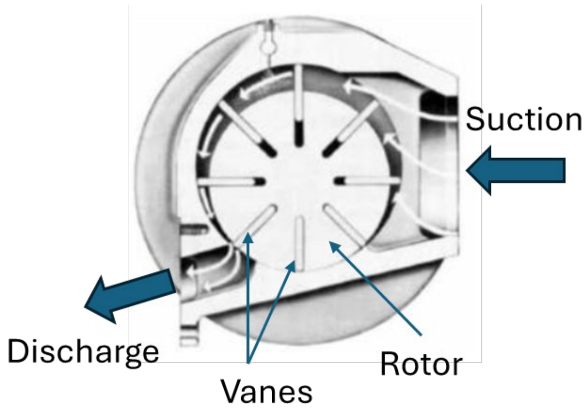


Figure III.7 Cutaway view of a rotary vane compressor

Figure III.8 Cutaway view of a rotary screw compressor

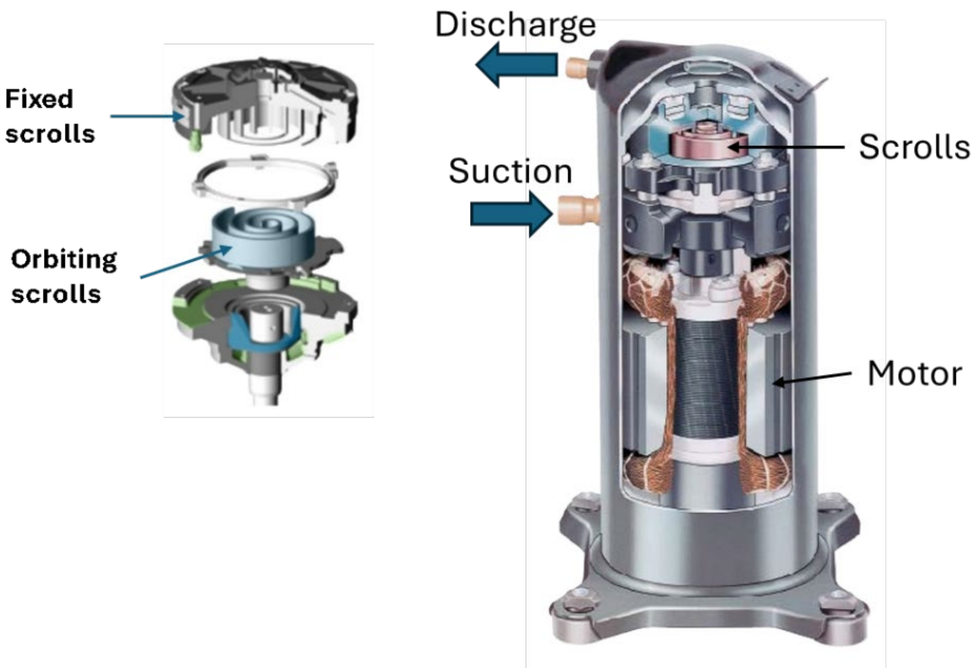


Figure III.9 cutaway view of a rotary scroll compressor

3.5. Dynamic compressors

Dynamic compressors increase pressure by converting kinetic energy into pressure energy. The main types are:

Centrifugal compressors: used in large-capacity refrigeration systems (Figure III.10).

Axial compressors: mainly used in gas turbine systems and <u>rarely in refrigeration</u>.

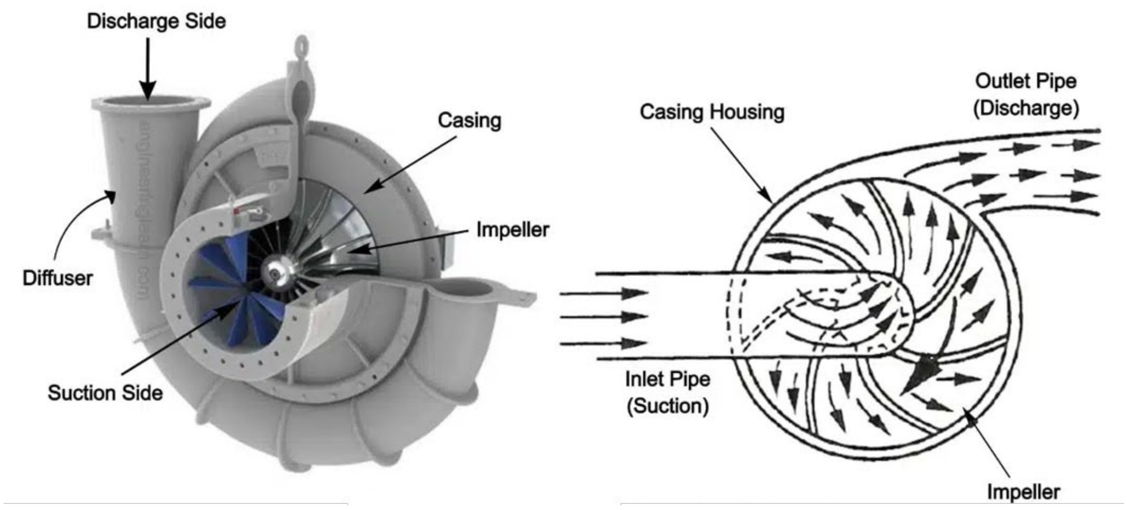


Figure III.10 cutaway view and a schematic diagram of a centrifugal compressor

3.6. COMPRESSOR PERFORMANCE CONSIDERATIONS

The performance of a compressor depends on several parameters, including suction and discharge pressures and temperatures, compression ratio, refrigerant properties, operating speed, and cooling conditions. Because of its central role and high energy consumption, the compressor is often the primary focus when improving refrigeration system efficiency and reliability.

3.6.1. Thermodynamic Analysis of the Compressor

The compressor in a vapor-compression refrigeration system operates as a steady-flow device and can be analyzed using basic thermodynamic balance equations. These equations describe how mass, energy, and entropy change during the compression process and help evaluate its performance.

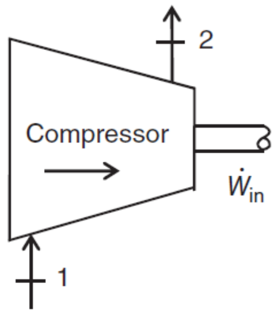


Figure III.11 A compressor considered for analysis.

- **Mass balance**

For steady operation, the mass flow rate of refrigerant entering and leaving the compressor remains constant:

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m}$$

- **Energy balance**

In a real compressor, mechanical work is supplied to the refrigerant, and there may also be heat exchange with the surroundings. The steady-flow energy equation is therefore written as:

$$\dot{Q} + \dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2$$

or

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) - \dot{Q}$$

where:

$\dot{W}_{in}$= compressor work input

$\dot{Q}$= heat transfer (usually heat loss from the compressor to the surroundings)

h_1, h_2= refrigerant enthalpies at inlet and outlet

In practice, compressors are not perfectly insulated, so a small amount of heat is often lost to the surrounding air. Although this heat transfer is usually small and sometimes neglected in basic analysis, it affects the actual work requirement and efficiency of the compressor.

- **Entropy balance**

Because of irreversibilities such as friction, turbulence, and heat transfer, entropy increases during compression:

$$\dot{m}s_1 + \dot{S}_{gen} = \dot{m}s_2$$

where $\dot{S}_{gen}$ is the entropy generation.

For an ideal isentropic compressor:

$$s_1 = s_2$$

- **Isentropic efficiency**

The performance of a real compressor is evaluated using the isentropic efficiency:

$$\eta_{comp} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

where h_{2s} is the outlet enthalpy for ideal (isentropic) compression.

- **Compression ratio**

A key operating parameter is the compression ratio:

$$CR = \frac{P_d}{P_s}$$

where P_d and P_s are the discharge and suction pressures, respectively.

This thermodynamic analysis shows that compressor performance depends on the enthalpy rise, pressure increase, heat losses, and internal irreversibilities during compression. These factors determine the work required and strongly influence the efficiency of the refrigeration system.

4. Condensers

The condenser is one of the four main components of a vapor-compression refrigeration system. After leaving the compressor as a high-pressure, high-temperature vapor, the refrigerant enters the condenser where it rejects heat to the surroundings and changes phase from vapor to liquid. This process allows the refrigeration cycle to continue and prepares the refrigerant for expansion and evaporation.

4.1. Role and basic operation

The condenser removes heat from the refrigerant, which includes:

- the heat absorbed in the evaporator, and
- the work supplied by the compressor.

Inside the condenser, the refrigerant typically undergoes three stages:

1. cooling of superheated vapor,
2. condensation at nearly constant pressure and temperature,
3. possible subcooling of the liquid before expansion.

The refrigerant must reject heat to a cooling medium (air or water) whose temperature is lower than the refrigerant condensation temperature.

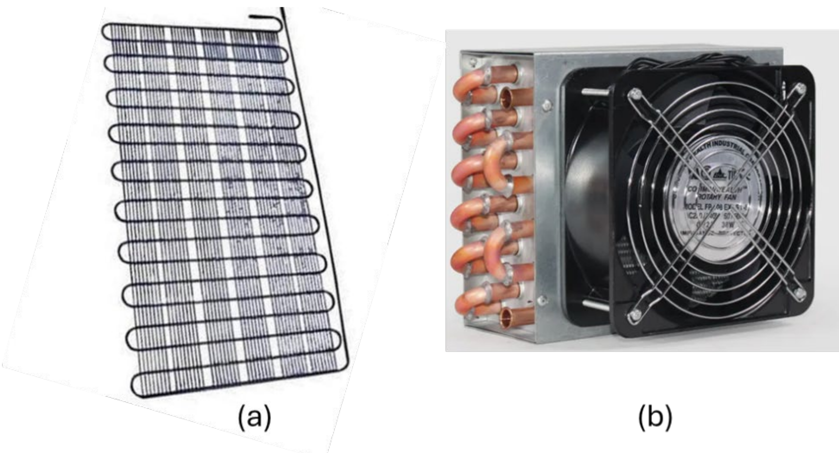
Figure III. illustrate typical condenser arrangements and the flow of refrigerant and cooling medium, showing how heat is transferred from the refrigerant to the environment.

4.2. Types of condensers

Condensers are classified according to the cooling medium used.

4.2.1. Air-cooled condensers (Figure III.12)

- Use ambient air to remove heat.
- Common in domestic refrigeration and small air-conditioning systems.
- Simple construction and easy installation.



(a) (b)

Figure III.12 Air-cooled condensers: (a) wire-and-tube air-cooled condenser; (b) forced-air condenser unit with fan and finned tubes.

4.2.2. Water-cooled condensers (Figure III.13)

- Use water as the cooling medium.
- Higher heat-transfer efficiency than air-cooled condensers.
- Used in commercial and industrial refrigeration systems.

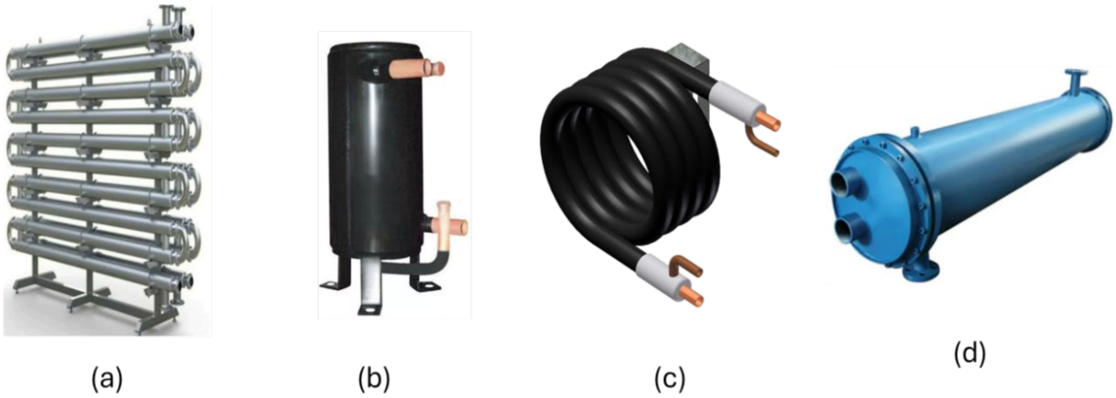


Figure III.13 Various types of water-cooled condensers: (a) Tube in Tube; (b) Shell and coil; (c) Coaxial; (d) Shell and tube.

4.2.3. Evaporative condensers (Figure III.14)

- Use both air and water for heat rejection.
- Provide improved heat-transfer performance and compact design.
- Common in industrial refrigeration installations.

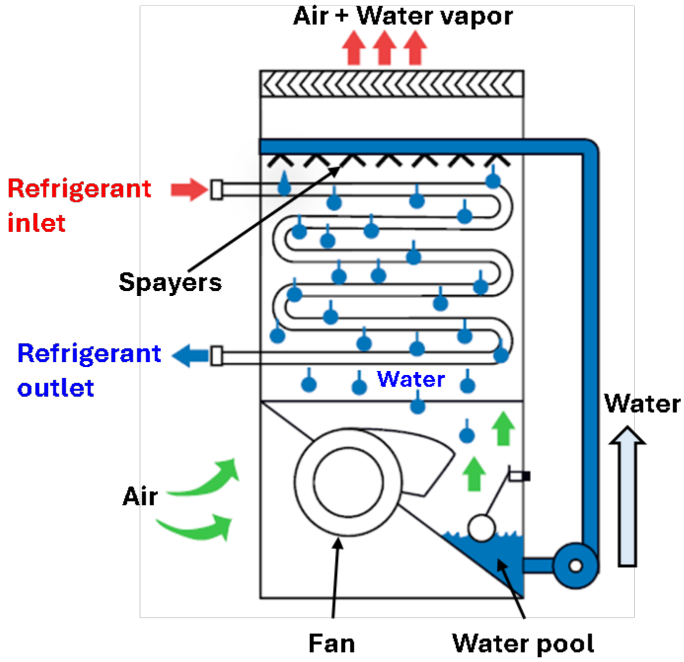


Figure III.14 Illustrative scheme of an evaporative condenser.

4.3. Thermodynamic analysis of the condenser

Condensers are used to condense the superheated refrigerant vapor leaving the compressor at essentially constant pressure, changing its phase to saturated liquid or slightly subcooled liquid and reducing its temperature to the condenser temperature. During this process, heat must be removed from the superheated vapor and rejected to a cooling medium, typically air in air-cooled condensers or water in water-cooled condensers.

In both air-cooled and water-cooled configurations, the condenser operates as steady state, steady flow devices, and its thermodynamic analysis is based on mass and energy equations.

- **Mass balance**

For an air-cooled condenser with one refrigerant inlet and one outlet, the mass balance is (Figure III.15 (a))

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

and for a water-cooled condenser (treated as a closed heat exchanger) separate balances are written for refrigerant and water (Figure III.14 (b)):

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_R, \dot{m}_3 = \dot{m}_4 = \dot{m}_w$$

where subscripts 1–2 denote refrigerant inlet and outlet, and 3–4 denote water inlet and outlet.

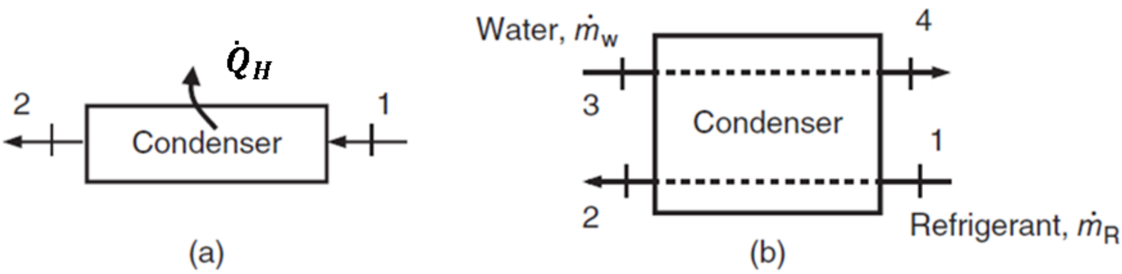


Figure III.15 (a) Air-cooled condenser and (b) water-cooled condenser for thermodynamic analysis.

- **Energy balance**

Neglecting kinetic and potential energy changes and assuming good insulation of a water-cooled unit, the energy balance for an air-cooled condenser is

$$\dot{m}\,(h_1 - h_2) = \dot{Q}_H$$

where $\dot{Q}_H$ is the heat rejection rate (condenser heat rate). For a water-cooled condenser, the energy balance becomes

$$\dot{m}_R(h_1 - h_2) = \dot{m}_w(h_4 - h_3)$$

showing that the heat rejected by the refrigerant equals the heat gained by the cooling water.

5. Evaporators

5.1. Role of the Evaporator

The evaporator is the component where the cooling effect is produced in a vapor-compression refrigeration system. Low-pressure, low-temperature refrigerant enters the evaporator and absorbs heat from the medium to be cooled (air, water, brine, product), boiling at essentially constant pressure and temperature and usually leaving as saturated or slightly superheated vapor. In practice, evaporators are broadly divided into:

Direct coolers: evaporators that cool air directly, and this cooled air then cools the product (air coolers, gas coolers).

Indirect coolers: evaporators that cool a liquid (water, brine), and this liquid then cools the product in a secondary circuit (liquid coolers, chillers).

Common evaporator groups are: **liquid coolers**, **air coolers**, and **gas coolers**.

5.2. Liquid Coolers (Chillers)

Liquid coolers are evaporators used to cool liquids such as water or brine, which may act as secondary refrigerants or be the final product itself. Typical applications include:

- Chilling potable water
- Supplying chilled water for air-conditioning coils
- Cooling milk after pasteurization
- Process cooling in food and industrial plants

Shell-and-tube heat exchangers (Figure III.16) are the most common form of liquid cooler for water-cooling and chilling applications. They are usually built with a carbon steel shell and copper tubes, and can be arranged as either flooded or direct-expansion (dry) evaporators.

Evaporators may operate as:

- flooded evaporators, where refrigerant surrounds the tubes,
- dry (direct-expansion) evaporators, where refrigerant flows inside the tubes and evaporates completely.

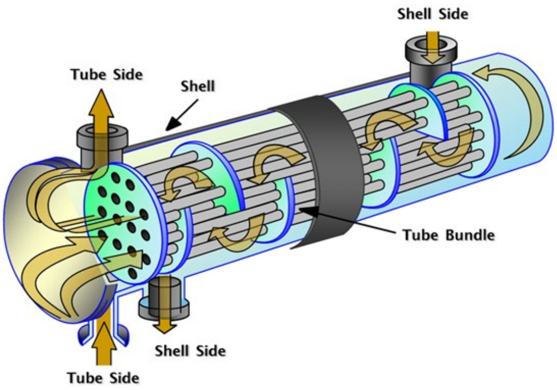


Figure III. 16 Shell-and-tube evaporator with indicated shell-side and tube-side flow paths.

5.3. Air and gas coolers

Air and gas coolers are commonly known as direct-expansion coils. They consist of finned tubes through which the refrigerant flows while air passes over the surface to be cooled (Figure III.).

These evaporators:

are widely used in refrigeration rooms and air-conditioning systems,

operate with parallel refrigerant circuits supplied from a throttling device,

may be classified as flooded or dry types.

In dry coils, refrigerant enters as a liquid–vapor mixture and leaves as superheated vapor. Flooded coils maintain a higher refrigerant level, improving heat transfer but requiring more refrigerant charge.

Figure III.17 Air cooler evaporator

5.4. Thermodynamic Analysis of Evaporators

Evaporators operate continuously and are analyzed as steady-state, steady-flow devices. The refrigerant typically enters as a saturated liquid–vapor mixture and leaves as saturated or slightly superheated vapor after absorbing heat at nearly constant pressure and temperature.

Two idealized cases are considered (See Figure III.18):

- Case (a): Refrigerant absorbs heat directly from a cooled space (air/room) – open to the environment.
- Case (b): Refrigerant absorbs heat from water in a closed heat exchanger (water-cooling evaporator).

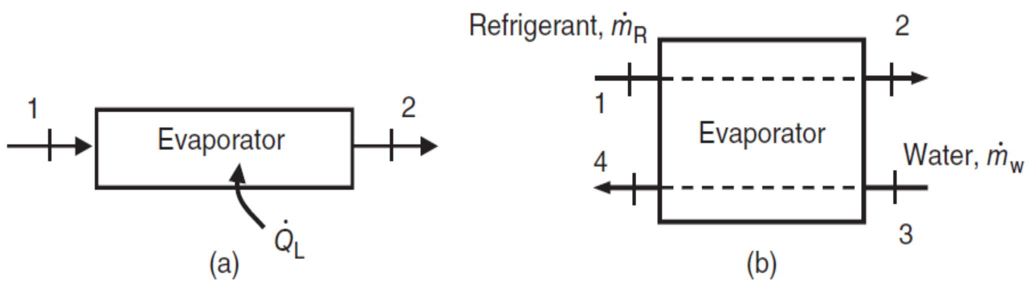


Figure III.18 Evaporators considered for thermodynamic analysis: (a) refrigerant absorbing heat from a space and (b) refrigerant absorbing heat from water.

- **Mass Balance**

For case (a), with one inlet (1) and one outlet (2) of refrigerant:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

For case (b), with separate refrigerant (subscript R) and water (subscript w) streams:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_R, \dot{m}_3 = \dot{m}_4 = \dot{m}_w$$

where 1–2 refer to refrigerant states and 3–4 to water states.

- **Energy Balance**

Neglecting kinetic and potential energy changes, and assuming the water-cooling evaporator is well insulated (no heat loss to surroundings), the energy balance for case (a) is:

$$\dot{m}(h_1 - h_2) = \dot{Q}_L$$

where $\dot{Q}_L$ is the evaporator heat rate (cooling load).

For the water-cooling evaporator (case (b)):

$$\dot{m}_R(h_1 - h_2) + \dot{m}_w(h_3 - h_4) = 0$$

or equivalently

$$\dot{m}_R(h_1 - h_2) = \dot{m}_w(h_4 - h_3)$$

showing that the heat absorbed by the refrigerant equals the heat removed from the water.

6. Throttling (Expansion) Devices

The throttling or expansion device is one of the four main components of a vapor-compression refrigeration system. Its function is to reduce the pressure of the liquid refrigerant coming from the condenser and regulate the flow of refrigerant entering the evaporator. This pressure drop produces a low-temperature liquid–vapor mixture that allows heat absorption to occur in the evaporator.

6.1. Role and basic operation

After condensation, the refrigerant leaves the condenser as a high-pressure liquid. It then passes through the expansion device, where its pressure decreases suddenly without the production of useful work. This process is known as throttling and results in:

- a large pressure drop,
- a decrease in temperature,
- partial vaporization of the refrigerant.

The refrigerant enters the evaporator as a low-pressure mixture of liquid and vapor, ready to absorb heat.

6.2. Classification of expansion devices

Expansion devices are classified according to how they control refrigerant flow into the evaporator.

6.2.1. Capillary Tubes

The capillary tube is the simplest refrigerant flow-control device and can replace an expansion valve in small systems. It is a long, small-diameter copper tube (typical internal diameter of about 0.43 mm and length around 1.55 m) through which liquid refrigerant flows from the condenser/receiver to the evaporator.

The pressure drop along the tube reduces the condensing pressure to the evaporating pressure and, for a properly selected tube length and diameter, maintains an approximately constant evaporating pressure over a range of loads. Capillary tubes are widely used in small hermetic refrigeration systems up to about 30 kW capacity, such as domestic refrigerators and freezers. Tube length is chosen to match compressor capacity, with condenser performance and evaporator size also affecting the required dimensions.

Capillary tubes can be integrated into a heat exchanger, for example soldered alongside the suction line so that warm vapor superheats and partially vaporizes the high-pressure liquid, improving system efficiency and ensuring stable expansion. They are most effective in small, factory-assembled systems with relatively stable operating conditions.

Figure III.19 Cooper capillary tube

6.2.2. Thermostatic expansion valves

A thermostatic expansion valve (Figure III.20) is essentially a reducing valve between the high-pressure liquid line and the low-pressure evaporator. It automatically maintains a nearly constant degree of superheat at the evaporator outlet by adjusting the flow of liquid refrigerant according to the evaporator load.

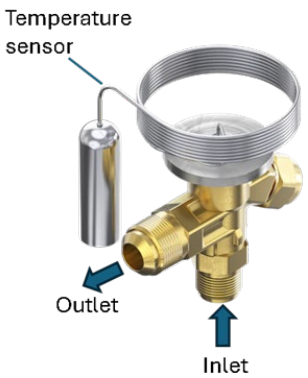


Figure III.20 Thermostatic expansion valve

The valve has three main forces acting on its diaphragm: bulb pressure (from a remote temperature-sensing bulb mounted on the suction line), evaporator pressure, and an adjustable spring force. The bulb, charged with a suitable fluid, senses the temperature of the superheated vapor leaving the evaporator; its pressure tends to open the valve, while evaporator pressure and spring force tend to close it. At equilibrium, the valve ports the amount of liquid necessary so that the measured superheat equals the design superheat setting.

When the load on the evaporator increases, the superheat at the outlet rises, bulb temperature and pressure increase, the valve opens further, and more liquid enters the evaporator. When the load decreases, superheat falls, bulb pressure drops, and the valve throttles towards closed, reducing liquid flow. In this way the TEV maintains the evaporator as full as possible of boiling refrigerant while preventing liquid from reaching the compressor.

In steady operation, the temperature just downstream of a properly functioning valve is much lower than upstream, because of the pressure drop and partial flash-off of liquid. If no temperature drop is observed, the seat and orifice are likely fouled by dirt or moisture; once correctly adjusted, the valve normally needs no further adjustment, and most problems are due to contamination of the seat. Electrically driven expansion valves, controlled by electronic thermostats and pressure sensors, are also used and follow the same principle but with a motorized actuation instead of a purely thermostatic one.

6.2.3. Automatic expansion valves

The constant-pressure, or automatic, expansion valve (Figure III.21) is an earlier form of expansion control and operates as a pressure-regulating device. It maintains a nearly constant evaporating pressure at the outlet of the valve by modulating the liquid-refrigerant flow into the evaporator based on suction pressure.

Figure III.21 Automatic expansion valve.

As the evaporator load increases and the suction pressure tends to fall, the valve opens to admit more liquid, and as the load decreases and suction pressure tends to rise, the valve closes. Thus, the refrigerant flow rate is adjusted to match the compressor capacity, keeping the evaporator pressure at the set value. Because the outlet pressure is kept essentially constant, these valves are suited only to systems with nearly constant cooling load and see limited application in modern variable-load systems.

6.2.4. Float valves

Float valves are used primarily with flooded evaporators to control the liquid refrigerant level. They are classified as high-side float valves and low-side float valves, depending on whether they are located on the high-pressure side or low-pressure side of the system.

A high-side float valve is installed on the high-pressure side and is used in systems with a single evaporator, compressor, and condenser. As liquid refrigerant accumulates in the condenser/receiver, the float rises and opens the valve, allowing liquid to flow into the evaporator until the level falls and the valve throttles, thus maintaining the desired level in the flooded cooler.

A low-side float valve is located on the low-pressure side and is often used in systems with multiple evaporators. In some arrangements, the float mechanism operates an electrical switch that controls a solenoid valve, periodically admitting liquid refrigerant and allowing the evaporator level to fluctuate within preset limits. In all cases, the purpose is to maintain adequate liquid coverage of the evaporator surface without allowing excessive carry-over of liquid to the suction line.

6.3. Thermodynamic Analysis of Throttling Valve

In a refrigeration cycle, the throttling valve operates continuously and is analyzed as a steady-state, steady-flow device with one inlet and one outlet (Figure III.22). Referring to the schematic (Figure 3.29 in the book), the mass balance for the throttling process is:

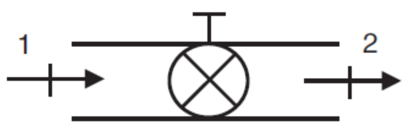


Figure III.22 A throttling valve considered for mass and energy analysis.

- **Mass Balance**

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

so the mass flow rate is constant.

The condensed refrigerant (saturated or slightly subcooled liquid) enters the valve at high pressure and exits at a lower pressure as a mixture of liquid and vapor at a reduced temperature. Neglecting changes in kinetic and potential energies and assuming no heat transfer to or from the surroundings, the energy balance becomes

$$\dot{m}(h_1 - h_2) = 0$$

or

$$h_1 = h_2$$

showing that throttling is an isenthalpic (constant-enthalpy) process.

7. Chapter Summary

This chapter presented the main components that enable a vapor-compression refrigeration system to operate in practice and showed how the thermodynamic cycle is implemented through real engineering devices. The refrigeration process is achieved through the coordinated action of four essential elements—compressor, condenser, expansion device, and evaporator—through which the refrigerant circulates in a closed loop while undergoing changes in pressure, temperature, and phase.

The compressor ensures refrigerant circulation and raises its pressure and temperature, strongly influencing energy consumption and overall system performance. The condenser rejects heat to the surroundings and converts the high-pressure vapor into liquid, while the expansion device reduces the pressure of the liquid refrigerant and controls its flow into the evaporator. The evaporator then absorbs heat from the medium to be cooled, producing the refrigeration effect and completing the cycle.

Beyond their individual roles, the performance of these components depends on thermodynamic behavior, operating conditions, and design characteristics. The chapter highlighted that compressors consume most of the system's energy, condensers determine heat rejection efficiency, evaporators control cooling capacity, and expansion devices regulate refrigerant flow and system stability. Thermodynamic analyses of each component—based on mass and energy balances—show how heat transfer, pressure changes, and irreversibilities affect system efficiency.

The classification of compressors, condensers, evaporators, and expansion devices also demonstrates the wide range of technologies available for different applications, from domestic refrigeration to industrial systems. Selecting and operating these components properly is essential for ensuring reliable performance, energy efficiency, and safe operation.

In summary, understanding the function, operation, and interaction of the components of a vapor-compression refrigeration system provides the foundation for analyzing real refrigeration installations and for designing efficient cooling systems used in air conditioning, food preservation, and industrial processes. This knowledge bridges the gap between the theoretical study of refrigeration cycles and their practical engineering implementation.

8. Solved Exercises

Things Engineers Think About

Note to students

Some questions in this section are not explicitly answered in the course document. They are intended to complement the course by encouraging application of concepts and practical engineering reasoning. The answers provided therefore extend the lecture material and help develop understanding of real refrigeration system operation.

1. Why is the compressor considered the "heart" of a refrigeration system, and what would happen to the cycle if it stopped operating?

2. How does the compression ratio affect compressor power consumption and system efficiency?

3. Why must the refrigerant enter the compressor as vapor rather than liquid? What problems can occur otherwise?

4. How do suction temperature and pressure influence compressor performance and energy use?

5. Under what conditions would an engineer choose a rotary compressor instead of a reciprocating compressor?

6. Why must the condensing temperature always be higher than the ambient air or cooling water temperature?

7. What happens to system performance if the condenser becomes dirty or airflow is reduced?

8. Why are water-cooled condensers more efficient than air-cooled condensers in large installations?

9. How does high condensing pressure affect compressor operation and system COP?

10. In what situations would an evaporative condenser be preferred over air-cooled and water-cooled condensers?

11. Why is the evaporator the component where the refrigeration effect is produced?

12. What is the difference between direct-expansion evaporators and flooded evaporators in terms of operation and efficiency?

13. How does frost formation on the evaporator surface affect heat transfer and system performance?

14. Why is it important to ensure proper refrigerant distribution inside an evaporator?

15. How does the evaporating temperature influence cooling capacity and compressor workload?

16. Why is the throttling process considered isenthalpic in refrigeration systems?

17. What is the main function of a thermostatic expansion valve compared with a capillary tube?

18. How does an expansion device influence evaporator performance and system stability?

19. Why are capillary tubes commonly used in small domestic refrigeration systems but not in large industrial systems?

20. What problems can occur if too much or too little refrigerant enters the evaporator due to improper expansion valve operation?

Answers

1. The compressor circulates the refrigerant and creates the pressure difference needed for the cycle; without it, refrigerant flow and cooling stop.

2. A higher compression ratio increases compressor work and reduces efficiency; a lower ratio improves performance.

3. Liquid entering the compressor can cause mechanical damage (liquid slugging); compressors are designed to compress vapor only.

4. Higher suction temperature and lower suction pressure increase compressor work and reduce efficiency.

5. Rotary compressors are preferred for compact systems, lower noise, and continuous operation; reciprocating compressors suit wider capacity ranges and higher pressure ratios.

6. Heat can only flow from hot to cold; the refrigerant must be hotter than the cooling medium to reject heat.

7. Heat rejection decreases, condensing pressure rises, compressor power increases, and system efficiency drops.

8. Water has higher heat-transfer capacity than air, allowing better cooling and lower condensing temperatures.

9. It increases compressor workload, energy consumption, and reduces COP.

10. When high efficiency and compact size are needed, especially in large industrial systems and warm climates.

11. Because the refrigerant absorbs heat and evaporates there, producing the cooling effect.

12. Direct-expansion evaporators use refrigerant inside tubes; flooded evaporators keep tubes immersed in liquid refrigerant and provide better heat transfer.

13. Frost acts as insulation, reducing heat transfer and lowering cooling capacity.

14. Uneven distribution reduces heat transfer efficiency and may cause poor cooling or compressor problems.

15. Lower evaporating temperature increases cooling ability but raises compressor workload; higher temperature improves efficiency.

16. Because the process occurs without work and heat transfer is negligible, enthalpy remains approximately constant.

17. A thermostatic expansion valve regulates flow based on superheat; a capillary tube provides a fixed restriction with no active control.

18. It controls refrigerant flow and pressure drop, ensuring proper evaporation and stable system operation.

19. Capillary tubes are simple and inexpensive but cannot adapt to load changes; large systems need adjustable control.

20. Too much refrigerant may return liquid to the compressor; too little reduces cooling capacity and evaporator performance.

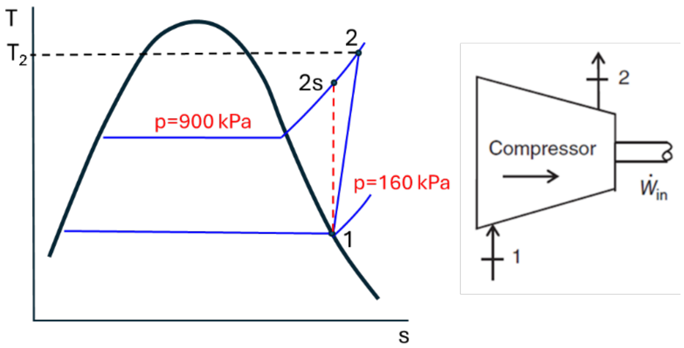
Exercise 1

R-134a enters the compressor of a refrigeration cycle at 160 kPa as a saturated vapor with a flow rate of 6.5 m^3/min and leaves at 900 kPa. The compressor isentropic efficiency is 75%. Determine

a. the temperature of R-134a at the exit of the compressor.

b. the actual work of the compressor

Solution



State 1 (saturated vapor at 160 kPa):

$h_1 = 237.97\ \mathrm{kJ/kg}$; $v_1 = 0.1229\ \mathrm{m^3/kg}$; $s_1 = 0.9295\ \mathrm{kJ/kg.K}$

State 2 (superheated vapor at $p_2 = 900\ \mathrm{kPa}$ **and** $T_2 = 75\ °\mathrm{C}$ **):**

$s_{2s} = s_1 = 0.9295\ \mathrm{kJ/kg.K}; p_2 = 900\ \mathrm{kPa}$

$\Longrightarrow h_{2s} = 273.73\ \mathrm{kJ/kg}$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \Longrightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C}$$

$$h_2 = 237.97 + \frac{273.73 - 237.97}{0.75}$$

$h_2 = 285.65 \text{ kJ/kg}$

a) Using $p_2 = 900 \text{ kPa}$ and $h_2 = 285.65 \text{ kJ/kg}$, we obtain from R134a superheated Tables $T_2 = 53°C$

b) The compressor power is calculated using the steady-flow energy equation:

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

Mass flow rate:

$$\dot{m} = \frac{\dot{V}_1}{v_1} \approx \frac{6.5/60}{0.1229} \approx 0.88 \text{ kg/s}$$

$$\dot{W}_{in} = 0.88 \times (285.65 - 237.97)$$

$$\dot{W}_{in} = 42 \text{ kW}$$

Exercise 2

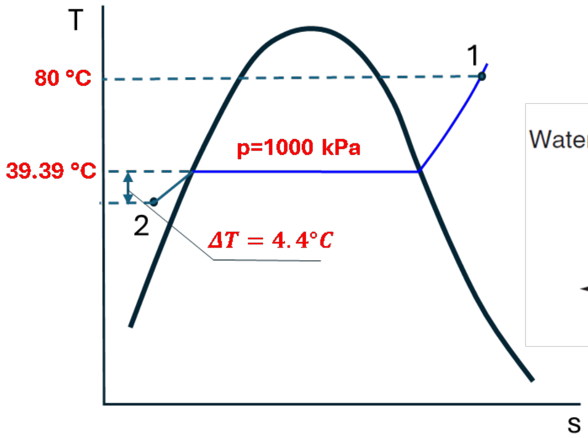
R-134a enters the condenser of a refrigeration cycle at 1000 kPa and 80 °C with a flow rate of 0.038 kg/s and leaves at the same pressure subcooled by 4.4 °C. The refrigerant is condensed by rejecting its heat to water, which experiences a temperature rise of 9 °C. Determine

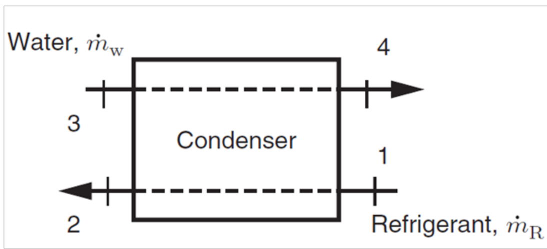
a. the rate of heat rejected in the condenser,

b. the mass flow rate of water,

c. the rate of cooling if the COP of this refrigeration cycle at these conditions is 1.4.

Solution





State 1 (superheated vapor at 1000 kPa and 80°C):

$h_1 = 313.20 \text{ kJ/kg}$

State 2 (subcooled liquid at $39.39 - 4.4 \approx 35°C$):

$h_2 = 265.93 \text{ kJ/kg}$

a) The rate of heat rejected in the condenser

$$\dot{Q}_H = \dot{m}_R(h_1 - h_2)$$
$$\dot{Q}_H = 0.038\,(313.20 - 265.93)$$
$$\dot{Q}_H \approx 1.80 \text{ kW}$$

b) the mass flow rate of water

Energy balance on the condenser (refrigerant rejects, water gains)

$$\dot{Q}_H = \dot{m}_R(h_1 - h_2) = \dot{m}_w c_{p,w} \Delta T_w$$

$$\dot{m}_w = \frac{\dot{Q}_H}{c_{p,w}\Delta T_w} \approx \frac{2.1}{4.18 \times 9}$$
$$\dot{m}_w = 0.048 \text{ kg/s}$$

c) the rate of cooling if the COP of this refrigeration cycle at these conditions is 1.4

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{\dot{Q}_L}{\dot{Q}_H - \dot{Q}_L}$$

Solve for $\dot{Q}_L$ in terms of $\dot{Q}_H$:

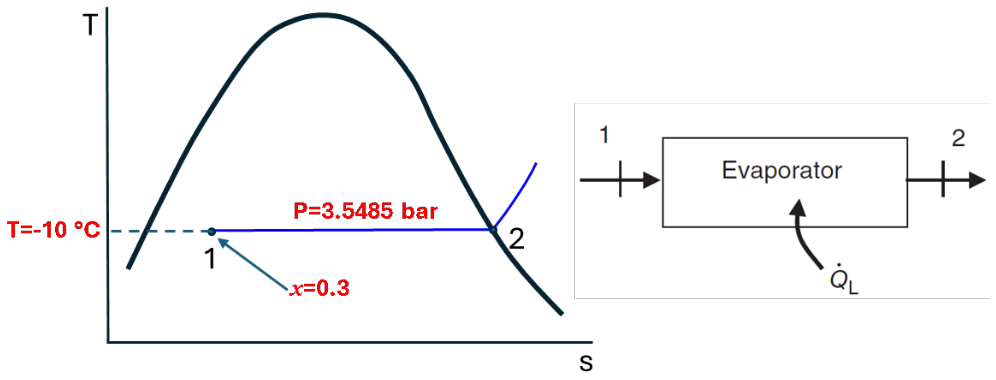
$$\text{COP} = \frac{\dot{Q}_L}{\dot{Q}_H - \dot{Q}_L} \Rightarrow \text{COP}(\dot{Q}_H - \dot{Q}_L) = \dot{Q}_L$$
$$\text{COP}\,\dot{Q}_H = \dot{Q}_L(1 + \text{COP}) \Rightarrow \dot{Q}_L = \frac{\text{COP}}{1 + \text{COP}}\,\dot{Q}_H$$

$$\dot{Q}_L = \frac{1.4}{1 + 1.4} \times 1.80 \approx 0.583 \times 2.1 \approx 1.2 \text{ kW}$$
$$\dot{Q}_L = 1.05 \text{ kW}$$

Exercise 3

Heat is absorbed from a cooled space at a rate of 320 kJ/min by R-22, which enters the evaporator at −10 °C with a quality of 0.3 and leaves as saturated vapor at the same pressure. Calculate the volume flow rates of R-22 at the evaporator inlet and outlet.

Solution



State 1 (mixture at -10 °C, x=0.3):

$h_1 = xh_g + (1-x)h_f$

$v_1 = xv_g + (1-x)v_f$

$h_1 = 0.3 \times 246.15 + (1-0.3) \times 33.54$

$v_1 = 0.3 \times 0.0652 + (1-0.3) \times 0.7606 \times 10^{-3}$

$h_1 = 97.32 \text{ kJ/kg}$

$v_1 = 0.02 \text{ kJ/kg}$

State 2 (saturated vapor at -10 °C):

$h_2 = 246.15 \text{ kJ/kg}$

$v_2 = 0.0652 \text{ kJ/kg}$

Mass flow rate of R-22

Evaporator heat transfer:

$$\dot{Q}_L = \dot{m}(h_2 - h_1)$$

$$\dot{m} = \frac{\dot{Q}_L}{(h_2 - h_1)}$$

$$\dot{m} = \frac{320/60}{246.15 - 97.32}$$

$$\dot{m} = 0.036 \text{ kg/s}$$

Volume flow rate at evaporator inlet and outlet :

$$\dot{V}_1 = \dot{m}\, v_1 = 0.036 \times 0.02$$

$$\dot{V}_1 = 0.00072 \text{ m}^3/\text{s}$$

$$\dot{V}_2 = \dot{m}\, v_2 = 0.036 \times 0.0652$$

$$\dot{V}_1 = 0.0023 \text{ m}^3/\text{s}$$

Exercise 4

R-134a enters the expansion valve of a refrigeration cycle at 900 kPa as a saturated liquid with a flow rate of 150 L/h. R-134a leaves the evaporator at 100 kPa superheated by 6.4 °C. The refrigerant is evaporated by absorbing heat from air which is cooled from 15 °C to 2°C. Determine

a. the rate of heat absorbed in the evaporator,

b. the mass flow rate of the air,

c. the COP of the cycle if the compressor work input is 72.5 kJ/kg.

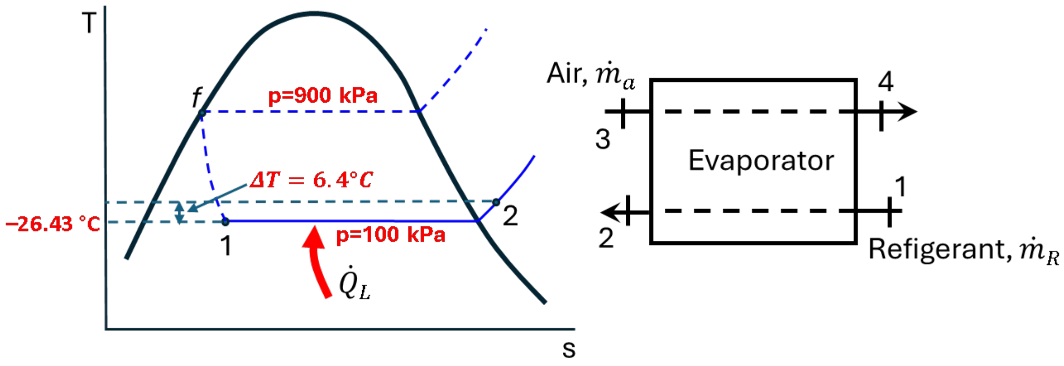
Solution

State f (expansion valve inlet), $p_f = 900$ kPa, saturated liquid:

$h_f = 99.56 \text{ kJ/kg} \,;\, v_f = 0.8576 \times 10^{-3} \text{ m}^3/\text{kg}$

State 1 (Evaporator inlet)

$h_1 = h_f = 99.56 \text{ kJ/kg}$



State 2 (Evaporator outlet) saturated vapor p=100 kPa, T≈ -20 °C:

$h_2 = 236.54 \text{ kJ/kg}$

a) the rate of heat absorbed in the evaporator:

$$\dot{Q}_L = \dot{m}_R(h_2 - h_1)$$

$$\dot{m}_R = \frac{\dot{V}}{v_f}$$

$$\dot{m}_R = \frac{150 \times 10^{-3}/3600}{0.8576 \times 10^{-3}}$$

$$\dot{m}_R = 0.049 \text{ kg/s}$$

$$\Longrightarrow \dot{Q}_L = 0.049 \times (236.54 - 99.56) = 6.7 \text{ kW}$$

b) the mass flow rate of the air:

Energy balance on evaporator:

$$\dot{Q}_L = \dot{m}_a \, c_{pa} \, \Delta T_a$$

$$\Longrightarrow \dot{m}_a = \frac{\dot{Q}_L}{\dot{m}_a \, c_{pa} \, \Delta T_a}$$

$$\dot{m}_a = \frac{6.7}{1.0035 \times (15 - 2)} = 0.51 \text{ kg/s}$$

c) the COP of the cycle if the compressor work input is 72.5 kJ/kg:

Coefficient of performance:

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_c} = \frac{\dot{Q}_L}{\dot{m}_R w_c}$$

$$\text{COP} = \frac{6.7}{0.049 \times 72.5} = 1.89$$

Chapter IV: Other Types of Refrigeration Systems

1. Introduction

In the previous chapters, the focus was placed on vapor-compression refrigeration systems, which are the most widely used in domestic, commercial, and industrial applications. However, refrigeration can also be produced using other technologies based on different operating principles and energy sources.

This chapter introduces alternative refrigeration machines that do not rely solely on mechanical compression. In particular, it examines absorption refrigeration systems and air refrigeration cycles, which operate using thermal energy, pressure variations, or gas expansion to produce cooling. These systems are especially useful in applications where electricity is limited, waste heat is available, or specific operating conditions are required.

The objective of this chapter is to present the basic principles, operation, and applications of these alternative refrigeration technologies and to highlight the main differences between them and vapor-compression systems in terms of performance, energy use, and practical implementation.

2. Absorption Refrigeration Systems

2.1. Basic Absorption Refrigeration Systems

The absorption refrigeration system is thought to be similar to the vapor-compression refrigeration cycle (using the evaporator, condenser, and throttling valve as in a basic vapor-compression refrigeration cycle), except that the compressor of the vapor-compression system is replaced by three main elements: an absorber, a solution pump, and a generator. Three steps, absorption, solution pumping, and vapor release, take place in an ARS.

In Figure IV.1, a basic absorption refrigeration system, which consists of an evaporator, a condenser, a generator, an absorber, a solution pump, and two throttling valves, is schematically shown. The strong solution (a mixture strong in refrigerant), which consists of the refrigerant and absorbent, is heated in the high-pressure portion of the system (the generator). This drives refrigerant vapor off the solution. The hot refrigerant vapor is cooled in the condenser until it condenses.

Then the refrigerant liquid passes through a throttling valve into the low-pressure portion of the system, the evaporator. The reduction in pressure through this valve facilitates the vaporization of the refrigerant, which ultimately effects the heat removal from the medium. The desired refrigeration effect is then provided accordingly. The weak solution (weak in refrigerant) flows down through a throttling valve to the absorber. After the evaporator, the cold refrigerant comes to the absorber and is absorbed by this weak solution (i.e., absorbent) because of the strong chemical affinity for each other. The strong solution is then obtained and is pumped by a solution pump to the generator, where it is again heated, and the cycle continues. It is significant to note that the system operates at high vacuum at an evaporator pressure of about 1.0 kPa; the generator and the condenser operate at about 10.0 kPa.

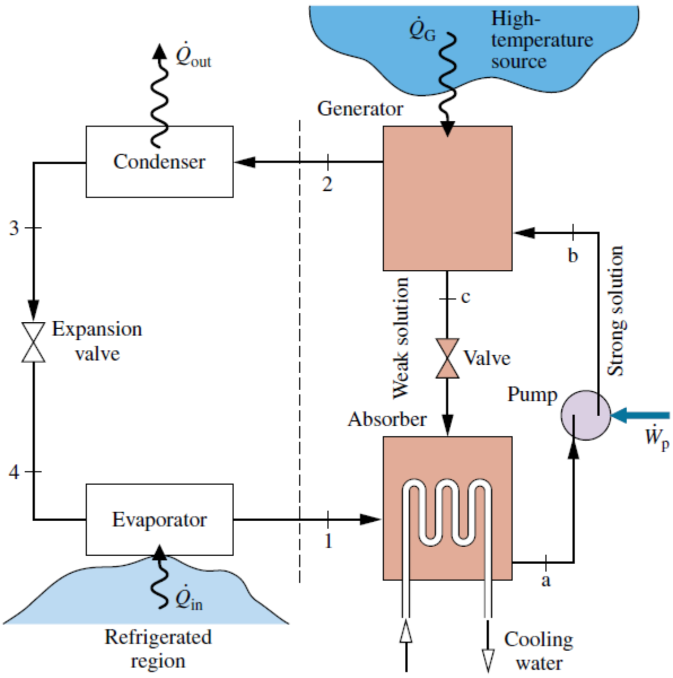


Figure IV.1 A basic absorption refrigeration system.

2.2. Ammonia–water (NH3–H2O) Absorption Refrigeration Systems

In practical systems (Figure IV.2), ammonia is used as the refrigerant and water as the absorbent. This system includes additional components:

- analyzer
- rectifier
- two heat exchangers

These components improve efficiency and ensure that nearly pure ammonia reaches the condenser.

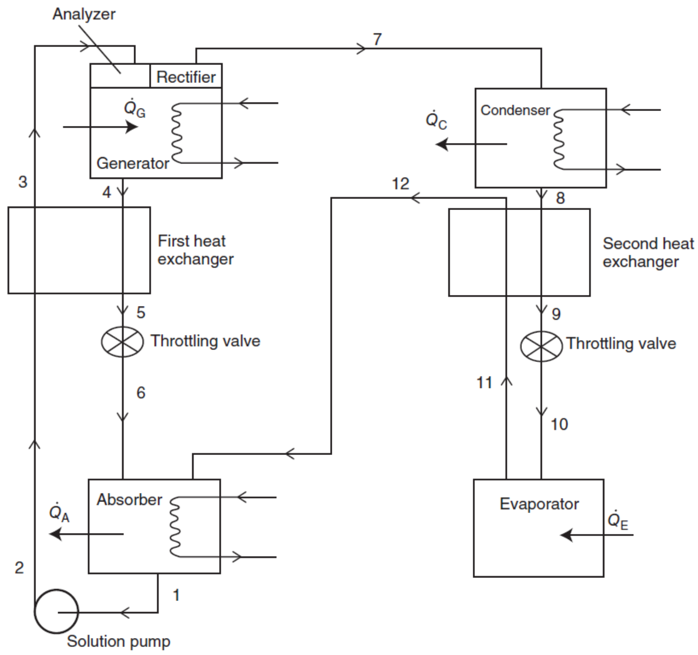


Figure IV.2 A practical ammonia–water absorption refrigeration system.

Operating principle

- In the absorber, water absorbs ammonia vapor, forming a strong solution (about 38% ammonia).
- The solution pump raises the pressure of this strong solution.
- In the generator, external heat separates ammonia vapor from water.
- The analyzer and rectifier remove water vapor from ammonia vapor to prevent contamination of the condenser.
- Ammonia condenses in the condenser, then expands through a throttling valve.
- In the evaporator, ammonia evaporates and absorbs heat (cooling effect).
- The vapor returns to the absorber, and the cycle repeats.

The absorber and condenser both reject heat to the surroundings. The pump work is very small compared to the generator heat input.

Ammonia–water systems are particularly suitable for:

- low-temperature refrigeration
- freezing applications
- industrial systems

2.3. Energy Analysis of an Absorption Refrigeration System

In the energy analysis, each component of the ammonia–water absorption refrigeration systems is treated as a steady-state, steady-flow control volume, and both energy and mass balances are written because the working fluid is a mixture whose composition changes between strong and weak solutions. Referring to the state numbering in Figure IV.2, the absorber receives weak solution (state 6) and refrigerant vapor (state 12), and delivers strong solution (state 1) while rejecting heat $\dot{Q}_A$ to the cooling medium; its balances are

$$\dot{m}_6 h_6 + \dot{m}_{12} h_{12} = \dot{m}_1 h_1 + \dot{Q}_A$$

(energy) and

$$\dot{m}_{ws} X_{ws} + \dot{m}_r = \dot{m}_{ss} X_{ss}$$

(mass), where $\dot{m}_{ws} = \dot{m}_6$, $\dot{m}_{ss} = \dot{m}_1$, $\dot{m}_r$ is the refrigerant mass flow rate, and X denotes ammonia mass fraction. The subscripts “ws” and “ss” denote "weak solution" and "strong solution", respectively.

The solution pump raises the pressure of the strong solution from absorber to generator level, so its balance is

$$\dot{m}_1 h_1 + \dot{W}_P = \dot{m}_2 h_2,$$

with compression taken as isothermal so that the pump work is relatively small. In the first solution heat exchanger, strong solution from the pump (state 2) is preheated by weak solution returning from the generator (state 4), giving

$$\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5,$$

where 3 is the hot strong-solution inlet to the generator and 5 the cooled weak-solution outlet to the throttling valve.

In the generator, strong solution (state 3) is heated by the external source $\dot{Q}_{gen}$, producing weak solution (state 4) and refrigerant vapor (state 7). The balances are

$$\dot{m}_3 h_3 + \dot{Q}_{gen} = \dot{m}_4 h_4 + \dot{m}_7 h_7$$

(energy) and

$$\dot{m}_{ws} X_{ws} + \dot{m}_r = \dot{m}_{ss} X_{ss}$$

(mass, here $\dot{m}_{ws} = \dot{m}_4$, $\dot{m}_{ss} = \dot{m}_3$).

In the condenser, the refrigerant rejects heat $\dot{Q}_H$ as it changes from vapor (state 7) to liquid (state 8), giving

$$\dot{m}_7 h_7 = \dot{m}_8 h_8 + \dot{Q}_H.$$

he second heat exchanger precools the strong solution entering the absorber and subcools the liquid refrigerant: liquid ammonia (state 8) and cold ammonia vapor (state 11) exchange heat to become states 9 and 12, respectively, with

$$\dot{m}_8 h_8 + \dot{m}_{11} h_{11} = \dot{m}_9 h_9 + \dot{m}_{12} h_{12}.$$

The two throttling (expansion) valves (one for the solution and one for the refrigerant) are modeled as isenthalpic devices:

$$\dot{m}_5 h_5 = \dot{m}_6 h_6 \;\Rightarrow\; h_5 = h_6,$$

$$\dot{m}_9 h_9 = \dot{m}_{10} h_{10} \;\Rightarrow\; h_9 = h_{10}.$$

In the evaporator, low-pressure liquid/vapor mixture (state 10) absorbs the refrigeration load $\dot{Q}_L$ and exits as vapor (state 11), giving

$$\dot{m}_{10} h_{10} + \dot{Q}_L = \dot{m}_{11} h_{11}.$$

Combining all components, and neglecting heat losses to the environment, the overall energy balance for the complete system becomes

$$\dot{W}_P + \dot{Q}_L + \dot{Q}_{gen} = \dot{Q}_A + \dot{Q}_H.$$

The coefficient of performance is then defined as

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_P + \dot{Q}_{gen}},$$

where the pump work $\dot{W}_P$ is usually much smaller than the generator heat input and can often be neglected in approximate COP calculations.

3. Air-Standard Refrigeration Cycle

3.1. Ideal cycle

Air-standard refrigeration cycles, also called reverse Brayton cycles, use air or another non-condensing gas as the working fluid instead of a refrigerant that changes phase. In these systems, refrigeration is produced by compressing and expanding the gas, rather than by evaporation and condensation as in vapor-compression systems. The cooling effect depends on the temperature change of the gas and its specific heat. Since this effect is smaller than the latent heat used in vapor-compression cycles, a larger mass flow rate of air is required to produce the same refrigeration capacity. For this reason, these systems often operate in a closed and pressurized loop to reduce equipment size.

Unlike vapor-compression systems, where expansion is achieved using a throttling valve, air-cycle refrigeration systems use an expander (expansion turbine) that produces useful work during the pressure drop. The energy required for refrigeration is supplied by the gas itself through compression and expansion processes. These systems are particularly important where lightweight equipment is needed, such as in aircraft cabin cooling.

A basic air-standard refrigeration cycle consists of four main components (Figure IV.3):

- **Compressor**

Increases the pressure and temperature of the air (process 1–2, nearly isentropic).

- **High-temperature heat exchanger**

Air rejects heat to the surroundings at nearly constant pressure (process 2–3).

- **Expander (turbine)**

Air expands, and its pressure and temperature decrease (process 3–4, nearly isentropic).

- **Low-temperature heat exchanger (cooling load)**

Air absorbs heat from the space to be cooled at constant pressure (process 4–1). This heat absorption represents the refrigeration effect.

Using air as the refrigerant becomes especially advantageous when the same air can also serve as the medium for air conditioning. For example, in aircraft systems, the air used for cooling is directly supplied to the cabin. Air-standard refrigeration cycles are also used in gas liquefaction processes and in specialized cooling applications where reliability and low weight are important.

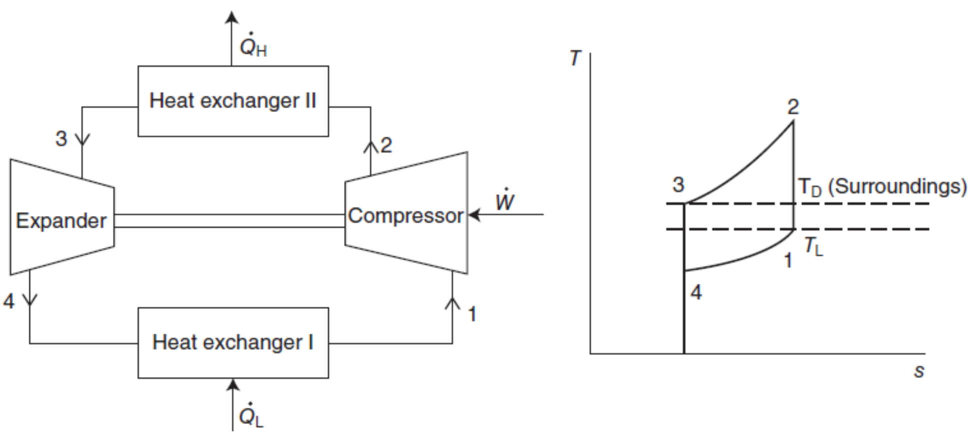


Figure IV.3 The air-standard refrigeration cycle.

Thermodynamic analysis of a basic air refrigeration cycle

In the energy analysis of a basic air-standard refrigeration cycle, as shown in Figure IV.3, we follow the same methodology that we used in the energy analysis of a vapor-compression refrigeration cycle. The only difference is that we can treat the gaseous working fluid (i.e., air) as an ideal gas. We can write the following:

For the compressor:

$$\dot{W}_c = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$

For heat exchanger II:

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = \dot{m}C_p(T_2 - T_3)$$

For the expander (turbine):

$$\dot{W}_t = \dot{m}(h_3 - h_4) = \dot{m}C_p(T_3 - T_4)$$

For heat exchanger I:

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = \dot{m}C_p(T_1 - T_4)$$

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

$$\dot{W}_c + \dot{Q}_L = \dot{W}_t + \dot{Q}_H$$

The network for the system becomes

$$\dot{W}_{net} = \dot{W}_c - \dot{W}_t = \dot{m}C_p[(T_2 - T_1) - (T_3 - T_4)]$$

The COP of the air-standard refrigeration system is

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}} = \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)}$$

For the two isentropic processes, we can write

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \text{ and } \frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$

Given that $p_1 = p_4$ and $p_2 = p_3$, then

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \therefore \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

and

$$COP_R = \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)} = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}$$

$$COP_R = \frac{T_1(1 - T_4/T_1)}{T_2(1 - T_3/T_2) - T_1(1 - T_4/T_1)} = \frac{T_1}{T_2 - T_1}$$

$$COP_R = \frac{1}{T_2/T_1 - 1} = \frac{1}{(p_2/p_1)^{\frac{\gamma-1}{\gamma}} - 1}$$

3.2. Real Cycle

In a real refrigeration cycle (Figure IV.4), the compression and expansion processes are not ideal. Because of friction, heat losses, and other irreversibilities, the actual enthalpy values at the end of compression and expansion are higher than those predicted for an ideal cycle. As a result, the amount of useful heat transferred and the overall system performance are reduced.

In addition, the work required for compression becomes significantly higher than in the ideal case. These differences arise from internal losses in the compressor and the expansion device, which introduce irreversibilities into the system.

To evaluate the performance of real machines, the isentropic efficiencies of the compressor and the expander are defined. These efficiencies compare the actual process with the ideal isentropic (reversible) process and are expressed as follows:

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$\eta_t = \frac{h_3 - h_4}{h_{4s} - h_3}$$

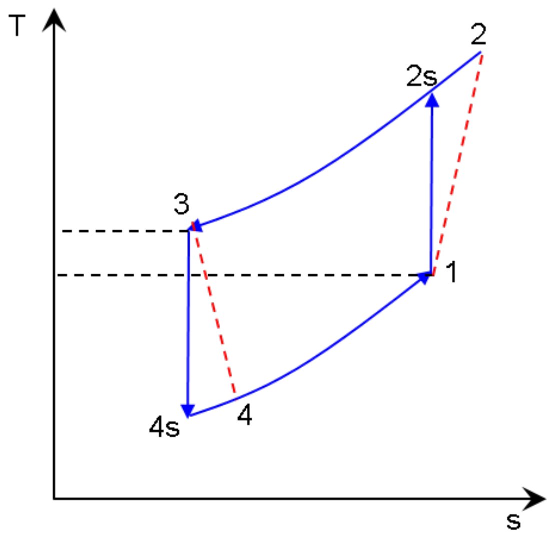


Figure IV.4 Actual air refrigeration cycle

3.3. Cycle with regeneration

Figure IV.5 shows the schematic and T–s diagram of an ideal Brayton cycle modified by introduction of a heat exchanger. The heat exchanger allows the air exiting the compressor at state 2 to cool below the warm region temperature T_H, giving a low turbine inlet temperature, T_3. Without the heat exchanger, air could be cooled only close to T_H, as represented on the figure by state a. In the subsequent expansion through the turbine, the air achieves a much lower temperature at state 4 than would have been possible without the heat exchanger. Accordingly, the refrigeration effect, achieved from state 4 to state b, occurs at a correspondingly lower average temperature.

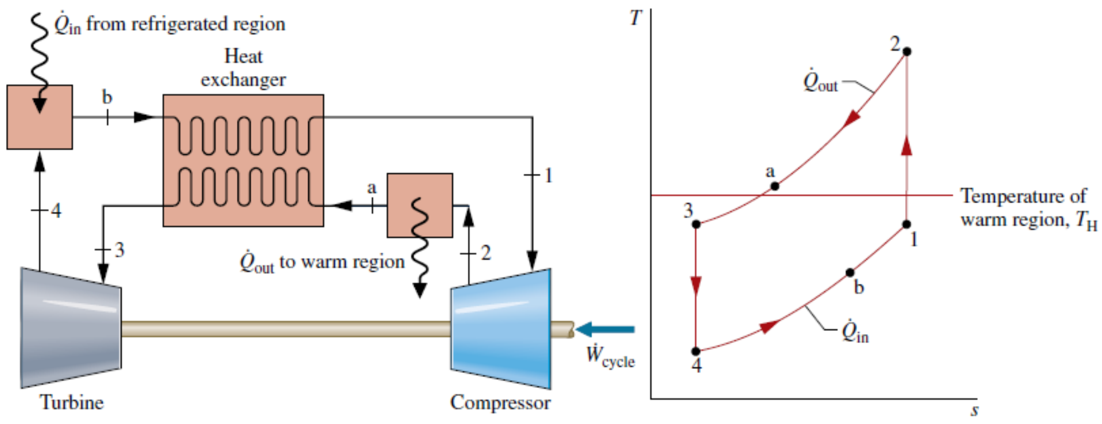


Figure IV.5 Air-Standard refrigeration cycle modified with a heat exchanger.

4. Solved Exercises

Things Engineers Think About

Note to students

Some questions in this section are not explicitly answered in the course document. They are intended to complement the course by encouraging application of concepts and practical engineering reasoning. The answers provided therefore extend the lecture material and help develop understanding of real refrigeration system operation.

1. Why can an absorption refrigeration system operate using thermal energy instead of mechanical work, and in which situations is this advantageous?

2. What practical problems could arise if heat is not effectively removed from the absorber?

3. Why is the solution pump work considered negligible compared with the generator heat input?

4. How would system performance change if the generator temperature is too low?

5. Why must nearly pure ammonia reach the condenser in NH_3–H_2O systems?

6. What is the engineering role of the analyzer and rectifier in improving system reliability and efficiency?

7. Why are absorption systems particularly suitable for applications using waste heat or solar energy?

8. Why must both mass and energy balances be applied in the analysis of absorption systems instead of energy balances alone?

9. What is the impact of heat rejection in the condenser and absorber on the overall performance of the absorption cycle?

10. Why is the COP of an absorption refrigeration system generally lower than that of a vapor-compression system?

11. Why is a large mass flow rate of air required compared with vapor-compression refrigeration systems?

12. What advantages make air refrigeration systems suitable for aircraft cooling applications?

13. How does replacing the throttling valve with an expander improve the performance of the air refrigeration cycle?

14. Why does regeneration (use of a heat exchanger) allow the air refrigeration cycle to reach lower temperatures?

15. In real air refrigeration cycles, how do irreversibilities in compression and expansion affect system efficiency and required work?

Answers

1. Because the compressor is replaced by the absorber–pump–generator combination, where heat separates the refrigerant from the solution. This allows operation using waste heat, solar energy, or steam.

2. Absorption becomes less effective, pressure rises, and system performance decreases or stops.

3. The pump handles liquid, which requires much less energy than compressing vapor in a compressor.

4. Insufficient refrigerant vapor is produced, reducing cooling capacity and system efficiency.

5. Water contamination reduces heat transfer, affects pressure levels, and lowers system efficiency.

6. They remove water vapor from ammonia, ensuring proper operation and protecting downstream components.

7. The system uses heat as its main energy source, making it ideal where thermal energy is available and electricity is limited.

8. Because the refrigerant and absorbent mix and separate, changing concentrations and flow rates.

9. Poor heat rejection raises pressures and reduces system efficiency.

10. Absorption systems rely on thermal energy, which is less efficient than mechanical compression.

11. Cooling depends on temperature change and specific heat, not latent heat, so more air is required.

12. They are lightweight, use air directly, and avoid complex refrigerant systems.

13. The expander produces useful work and improves efficiency.

14. Heat exchange between streams improves cooling before expansion, enhancing the refrigeration effect.

15. They increase required work, reduce efficiency, and decrease cooling performance.

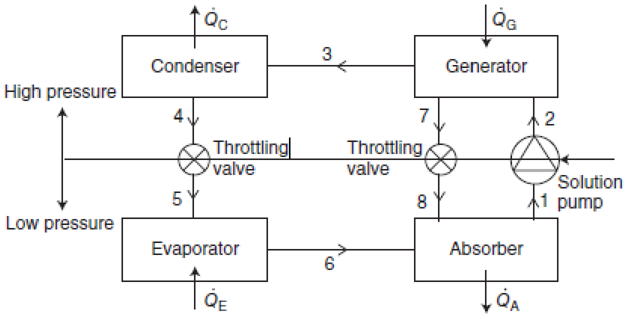
Exercise 1

Consider a basic absorption refrigeration system using ammonia–water solution, as shown in the Figure below. Pure ammonia enters the condenser at 2.5MPa and 60 °C at a rate of 0.022 kg/s. Ammonia leaves the condenser as a saturated liquid and is throttled to a pressure of 0.15MPa. Ammonia leaves the evaporator as a saturated vapor. Heat is supplied to the generator by geothermal liquid water that enters at 135 °C at a rate of 0.35 kg/s and leaves at 120 °C. Determine

a. the rate of cooling provided by the system,

b. the COP of the system.

The enthalpies of ammonia at various states of the system are given as h_3 = 1497.4 kJ/kg, h_4 = 482.5 kJ/kg, and h_6 = 1430.0 kJ/kg. Take the specific heat of water to be 4.2 kJ/kg. °C.



Solution

a) The rate of cooling provided by the system:

$$\dot{Q}_E = \dot{m}_R(h_6 - h_5)$$

$$h_5 = h_4 = 482.5 \text{ kJ/kg}$$

$$\dot{Q}_E = 0.022 \times (1430.0 - 482.5)$$

$$\dot{Q}_E = 20.85 \text{ kW}$$

b) The rate of heat input to the generator:

$$\dot{Q}_G = \dot{m}_{geo} C_{geo}(T_{in} - T_{out})$$

$$\dot{Q}_G = 0.35 \times 4.2 \times (135 - 120)$$

$$\dot{Q}_G = 22.05 \text{ kW}$$

c) the COP of the system:

$$COP_R = \frac{\dot{Q}_E}{\dot{Q}_G}$$

$$COP_R = \frac{20.85}{22.05}$$

$$COP_R = 0.95$$

Exercise 2

Consider a basic absorption refrigeration system using ammonia–water solution as shown in the figure of the previous exercise. Pure ammonia enters the condenser at 3200 kPa and 70 °C. Ammonia leaves the condenser as a saturated liquid and is throttled to a pressure of 220 kPa. Ammonia leaves the evaporator as a saturated vapor. Heat is supplied to the solution in the generator by solar energy, which is incident on the collector at a rate of 550 W/m^2. The total surface area of the collectors is 31.5 m^2 and the efficiency of the collectors is 75% (i.e., 75% of the solar energy input is transferred to the solution). If the COP of the system is estimated to be 0.8, calculate the mass flow rate of ammonia through the evaporator.

The enthalpies of ammonia at various states of the system are given as h3 = 1491.9 kJ/kg, h4 = 537.0 kJ/kg, and h6 = 1442.0 kJ/kg.

Solution

The COP of the system is given by

$$COP_R = \frac{\dot{Q}_E}{\dot{Q}_G}$$

Were,

$$\dot{Q}_E = \dot{m}_R(h_6 - h_5); \;\; h_5 = h_4$$
$$\dot{Q}_G = \eta_c I_b S$$

I_b represents the solar radiation intensity and η_c is the collector efficiency.

Combining these equations, we obtain:

$$COP_R = \frac{\dot{m}_R(h_6 - h_4)}{\eta_c I_b S}$$
$$\Longrightarrow \dot{m}_R = \frac{\eta_c I_b S COP_R}{h_6 - h_4}$$
$$\dot{m}_R = \frac{0.75 \times 550 \times 31.5 \times 0.8}{(1442.0 - 537.0) \times 10^3}$$
$$\dot{m}_R = 0.0115 \text{ kg/s}$$

Exercise 3

In an air-standard refrigeration cycle, air enters the compressor at −20 °C and 100 kPa. It undergoes isentropic compression to 500 kPa, is then cooled to 24 °C, and finally expands isentropically back to a pressure of 100 kPa.

a. Calculate the coefficient of performance (COP) of this cycle.

b. Repeat part (a) assuming an isentropic efficiency of 75% for both the compression and the expansion processes.

Solution

- $T_1 = -20^\circ C = 253.15\,K, p_1 = 100\text{kPa}$
- $P_2 = 500\text{kPa}$
- After cooling: $T_3 = 24^\circ C = 297.15\,K$
- Expansion to $p_4 = 100\text{kPa}$
- Air properties: $Cp = 1.0035 \text{ kJ/kg.K}, \gamma = 1.4$
- Pressure ratio: $r_p = \frac{p_2}{p_1} = 5$
- Temperature relation for isentropic processes:

$$T_2 = T_1 r_p^{(\gamma-1)/\gamma}$$
$$(\gamma - 1)/\gamma = 0.4/1.4 = 0.286$$
$$T_2 = 253.15 \times 5^{0.286} \approx 253.15 \times 1.585 = 401.24\ K$$

- Expansion temperature

$$T_4 = T_3 \left(\frac{1}{5}\right)^{0.286}$$
$$T_4 = 297.15 \times 0.631 = 187.50\ K$$

- Refrigeration effect

$$q_L = C_p(T_1 - T_4)$$
$$q_L = 1.0035(253.15 - 187.50)$$
$$q_L = 65.9 \text{ kJ/kg}$$

- Compressor work

$$w_c = C_p(T_2 - T_1)$$
$$w_c = 1.0035(401.24 - 253.15)$$
$$w_c = 148.6 \text{ kJ/kg}$$

- Expander work

$$w_t = C_p(T_3 - T_4)$$
$$w_t = 1.0035(297.15 - 187.50)$$
$$w_t = 110.0 \text{ kJ/kg}$$

- Net work

$$w_{net} = w_c - w_t$$
$$w_{net} = 148.7 - 110.0 = 38.6 \text{ kJ/kg}$$

a) The COP of the cycle

$$COP_R = \frac{q_L}{w_{net}}$$
$$COP = \frac{65.9}{38.6}$$
$$COP \approx 1.70$$

b) With isentropic efficiencies = 75%

- Compressor

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$
$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c}$$

$$T_2 = 253.15 + \frac{401.24 - 253.15}{0.75}$$

$$T_2 = 450.6\ K$$

- Expander

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$T_4 = T_3 - \eta_t(T_3 - T_{4s})$$

$$T_4 = 297.15 - 0.75(297.15 - 187.5)$$

$$T_4 = 214.9\ K$$

- Refrigeration effect

$$q_L = 1.0035(253.15 - 214.9)$$

$$q_L = 38.4 \text{ kJ/kg}$$

- Compressor work

$$w_c = 1.0035(450.6 - 253.15)$$

$$w_c = 198.1 \text{ kJ/kg}$$

- Expander work

$$w_t = 1.0035(297.15 - 214.9)$$

$$w_t = 82.5 \text{ kJ/kg}$$

- Net work

$$w_{net} = 198.1 - 82.5 = 115.6 \text{ kJ/kg}$$

- **COP**

$$COP = \frac{38.4}{115.6}$$

$$COP \approx 0.33$$

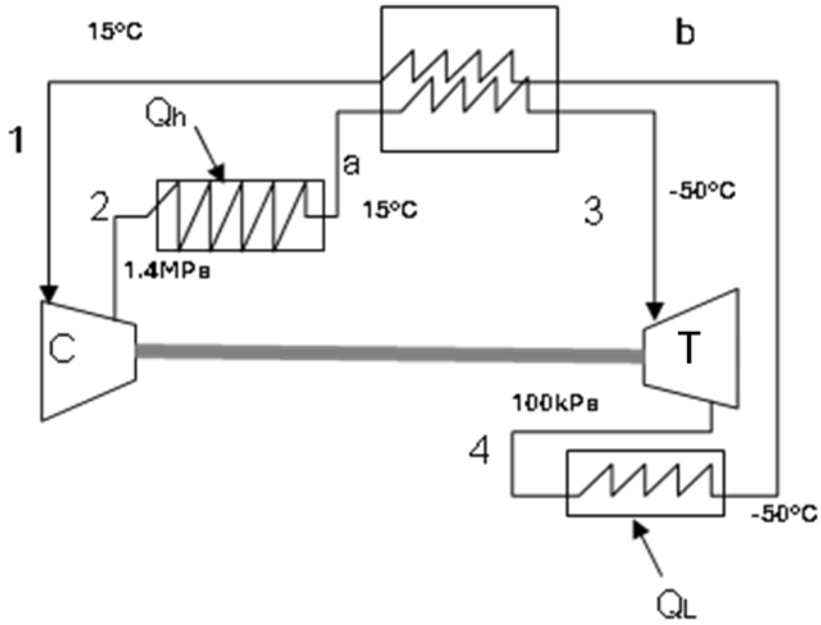
Observation:

Irreversibilities in compression and expansion significantly reduce the COP and increase the required work.

Exercise 4

a. Consider the ideal regenerative cycle illustrated in the figure below. Determine the coefficient of performance (COP) of this cycle, assuming that the compression and expansion processes are isentropic.

b. Repeat the calculation in part (a) assuming isentropic efficiencies of 75% for both the compressor and the expander.



Solution

$C_p = 1.0035$ kJ/kg ; $\gamma = 1.4$; $(\gamma-1)/\gamma = 0.4/1.4 = 0.286$
$T_1 = T_a = 288.15$ K ; $T_3 = T_b = 223.15$ K
$p_1 = 0.1\ MPa$; $p_2 = 1.4\ MPa$; $r_p = \frac{p_2}{p_1} = 14$

a)

$T_2 = T_1 r_p^{(\gamma-1)/\gamma}$; $T_2 = 612.9$ K
$T_4 = T_3 r_p^{-(\gamma-1)/\gamma}$; $T_3 = 104.9$ K

$$w_c = C_p(T_2 - T_1);\ w_c = 325.9 \text{ kJ/kg}$$
$$w_t = C_p(T_3 - T_4);\ w_t = 118.7 \text{ kJ/kg}$$
$$q_L = C_p(T_b - T_4);\ q_L = 118.7 \text{ kJ/kg}$$
$$COP_R = \frac{q_L}{w_c - w_t};\ COP_R = 0.573$$

b)

$\eta_c = \eta_c = 0.75$
$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c}\ ;\ T_2 = 721.15 \text{ K}$$
$T_4 = T_3 - \eta_t(T_3 - T_{4s})$; $T_4 = 134.5$ K

$$w_c = C_p(T_2 - T_1);\ w_c = 434.5 \text{ kJ/kg}$$
$$w_t = C_p(T_3 - T_4);\ w_t = 88.9 \text{ kJ/kg}$$
$$q_L = C_p(T_b - T_4);\ q_L = 88.9 \text{ kJ/kg}$$
$$COP_R = \frac{q_L}{w_c - w_t};\ COP_R = 0.257$$

Chapter V: Heat Pump

1. Introduction

A heat pump is a thermodynamic device designed to move heat from a low-temperature source to a high-temperature sink, using external work, and it can be viewed as a refrigeration system whose useful effect is heating rather than cooling. In contrast to the vapor-compression refrigerator studied in earlier chapters (where the objective is to remove heat from a cold space), the heat pump is arranged and evaluated so that the primary output is the heat delivered to a conditioned space or process. The basic working principle is still the vapor-compression (or, in some cases, absorption) cycle: a refrigerant evaporates at low temperature by extracting heat from an environmental source (outdoor air, ground, water), is compressed to a higher pressure and temperature, rejects heat by condensing at a higher temperature level, and finally expands back to the low-pressure state to repeat the cycle.

From a systems perspective, the heat pump chapter emphasizes that the same physical machine can supply both heating and cooling simply by reversing the direction of heat transfer or by changing which heat exchanger is considered the "useful" side. This naturally leads to the definition of the heating coefficient of performance, COPH, which compares the heat delivered to the work input and can take values significantly greater than 1, illustrating why heat pumps are often more energy-efficient than direct electric resistance heating. The chapter will introduce common configurations (air-source, water-source, and ground-source heat pumps) and discuss their operating characteristics, typical temperature levels, and practical limitations (such as frosting, source temperature variation, and auxiliary heating). It will also connect the ideal and real heat-pump cycles to the vapor-compression cycle already covered, showing how simple modifications (for example, reversing valves and defrost arrangements) allow one basic technology to serve as both a refrigerator and a space-heating device.

2. Principle of Operation and Fluid Flow Diagram

A heat pump transfers heat from a low-temperature source to a higher-temperature space by using external energy, usually supplied to a compressor. Although heat naturally flows from hot to cold, the heat pump forces heat to move in the opposite direction through a thermodynamic cycle.

The operating principle is based on the reversed vapor-compression cycle, which is similar to that used in refrigeration systems. However, the main objective here is to supply heat to a heated space rather than to produce cooling.

As illustrated in the schematic arrangement of the system components (Figure V.1), a heat pump consists of four main elements connected in a closed loop:

- evaporator,
- compressor,
- condenser,
- expansion device.

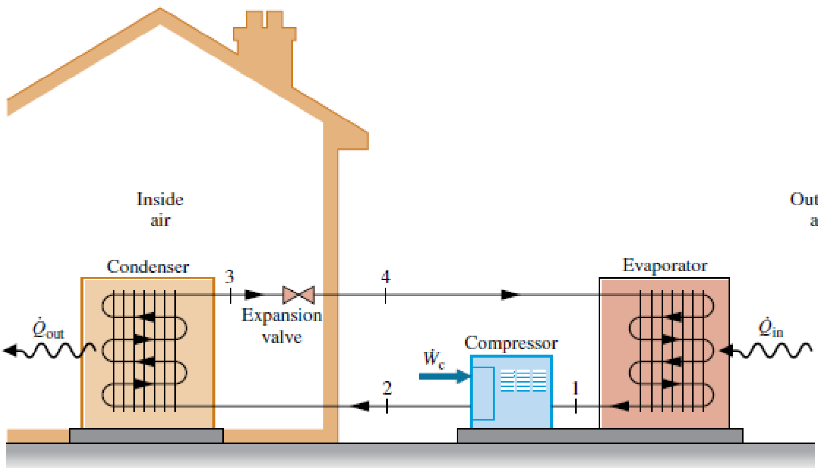


Figure IV.1 Air-source vapor-compression heat pump system

The operation of the system can be described as follows:

1- In the evaporator, the refrigerant absorbs heat from an external source (air, ground, or water) and evaporates at low pressure and temperature.

2- The vapor then enters the compressor, where its pressure and temperature increase.

3- The high-temperature refrigerant flows into the condenser, where it releases heat to the heated space and condenses into a liquid.

4- The liquid refrigerant passes through the expansion device, where its pressure decreases before returning to the evaporator to repeat the cycle.

The useful effect of the heat pump is the heat delivered at the condenser. The arrangement shown in the illustration highlights how energy supplied to the compressor enables the transfer of heat from the external environment to the indoor space.

Depending on operating conditions, the same system can operate in two modes:

- heating mode (winter), where heat is delivered indoors, and
- cooling mode (summer), where heat is removed from the indoor space.

This reversible operation makes heat pumps versatile systems widely used for heating, cooling, and domestic hot water production.

3. Reversing Valve and Reversible Heat Pumps

In the most common type of vapor-compression heat pump for space heating, the evaporator communicates thermally with the outside air. Such air-source heat pumps also can be used to provide cooling in the summer with the use of a reversing valve, as illustrated in Figure V.2. The solid lines show the flow path of the refrigerant in the heating mode, as described previously. To use the same components as an air conditioner, the valve is actuated, and the refrigerant follows the path indicated by the dashed line. In the cooling mode, the outside heat exchanger becomes the condenser, and the inside heat exchanger becomes the evaporator. Although heat pumps can be more costly to install and operate than other direct heating systems, they can be competitive when the potential for dual use is considered.

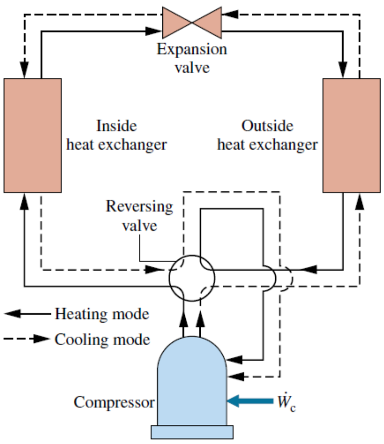


Figure V.2 Example of an air-to-air reversing heat pump.

4. Performance analysis

The performance of a heat pump is evaluated by analyzing how effectively it transfers heat from a low-temperature source to a high-temperature space. This analysis is based on energy balances and on performance indicators such as the coefficient of performance (COP). The operating conditions—particularly the temperatures of the heat source and heat sink—strongly influence system efficiency.

4.1. Performance in Heating Mode (Winter Operation)

In winter, the heat pump operates to supply heat to the indoor space. The useful effect is the heat released at the condenser.

- Heat delivered to the building:

$$Q_H$$

- Work supplied to the compressor:

$$W$$

The coefficient of performance of the heat pump in heating mode is defined as:

$$COP_{HP} = \frac{Q_H}{W}$$

Since the heat delivered at the condenser includes both the absorbed heat from the environment and the compressor work:

$$Q_H = Q_L + W$$

Therefore:

$$COP_{HP} = \frac{Q_L + W}{W}$$

A high COP means the system provides a large amount of heating for a small energy input.

- **Influence of outdoor temperature**:

The outdoor temperature has a significant influence on the performance of a heat pump. When the temperature of the heat source is higher, the compressor requires less work to transfer heat, resulting in a higher coefficient of performance (COP). Conversely, when the outdoor temperature is lower, the compressor must work harder to extract and deliver heat, which increases energy consumption and reduces the overall efficiency of the system.

4.2. Performance in Cooling Mode (Summer Operation)

In summer, the heat pump can operate as an air-conditioning system. The useful effect becomes the heat removed from the indoor space at the evaporator.

- Refrigeration effect:

$$Q_L$$

- Compressor work:

$$W$$

The coefficient of performance in cooling mode is:

$$COP_R = \frac{Q_L}{W}$$

The relationship between heating and cooling performance is:

$$COP_{HP} = COP_R + 1$$

This shows that a heat pump is generally more efficient in heating mode than in cooling mode.

- **Effect of Temperature Levels**:

The efficiency of a heat pump depends mainly on the temperature difference between the heat source and the heated space.

- **Large temperature difference**:
  - Higher compressor work
  - Lower COP
- **Small temperature difference:**
  - Reduced work input
  - Higher COP

For this reason:

- ground-source heat pumps often perform better than air-source systems,
- moderate heating temperatures improve system efficiency.

4.3. Real Performance Considerations

In practice, the actual performance of a heat pump is lower than the theoretical one due to:

- compressor inefficiencies,

- pressure drops in piping and heat exchangers,
- heat losses to the surroundings,
- frost formation in outdoor units (for air-source systems).

To evaluate real operation over an entire season, the **Seasonal Performance Factor (SPF)** is often used. It represents the average performance of the heat pump under varying climatic conditions.

5. Types of Heat Pumps

Heat pumps are classified according to the heat source from which they extract energy and the heat sink to which they deliver it. The selection of a heat pump type depends on climate conditions, available resources, installation constraints, and required heating capacity. The most common types are described below.

5.1. Air-Source Heat Pumps

Air-source heat pumps extract heat from the outdoor air and transfer it to the indoor space for heating or cooling (see Figure V.1). They are widely used in residential and small commercial buildings because they are relatively simple to install and have a lower initial cost compared with other types of heat pumps. However, their performance depends strongly on outdoor temperature: efficiency decreases in very cold conditions, and defrost cycles may be required to maintain proper operation. Despite these limitations, air-source heat pumps remain the most common type due to their accessibility, flexibility, and ease of implementation.

5.2. Ground-Source (Geothermal) Heat Pumps

Ground-source heat pumps extract heat from the ground, where the temperature remains relatively stable throughout the year. This stability allows the system to operate with higher and more consistent efficiency compared to air-source heat pumps, especially in cold climates. Heat exchange with the ground is achieved through buried pipe loops, which can be installed horizontally near the surface or vertically in boreholes (see Figure V.2). Although geothermal systems generally require a higher initial investment due to drilling or excavation, they offer improved seasonal performance and long-term energy savings.

5.3. Water-Source Heat Pumps

Water-source heat pumps use water bodies such as groundwater, lakes, rivers, or cooling towers as the heat source or heat sink (see Figure V.3). Because water temperatures are generally more stable than air temperatures, these systems can operate with relatively high efficiency and reliable performance throughout the year. They are commonly used in large buildings and industrial applications where a suitable water source is available. However, their installation depends on site conditions and may involve environmental regulations, water availability, and additional infrastructure for water circulation and heat exchange.

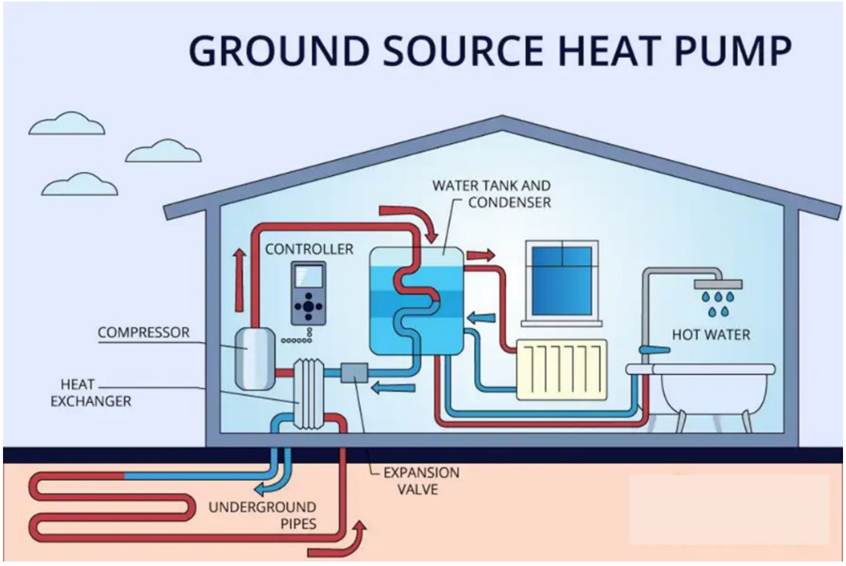


Figure V.2 Illustrative sketch of a Geothermal Heat Pump

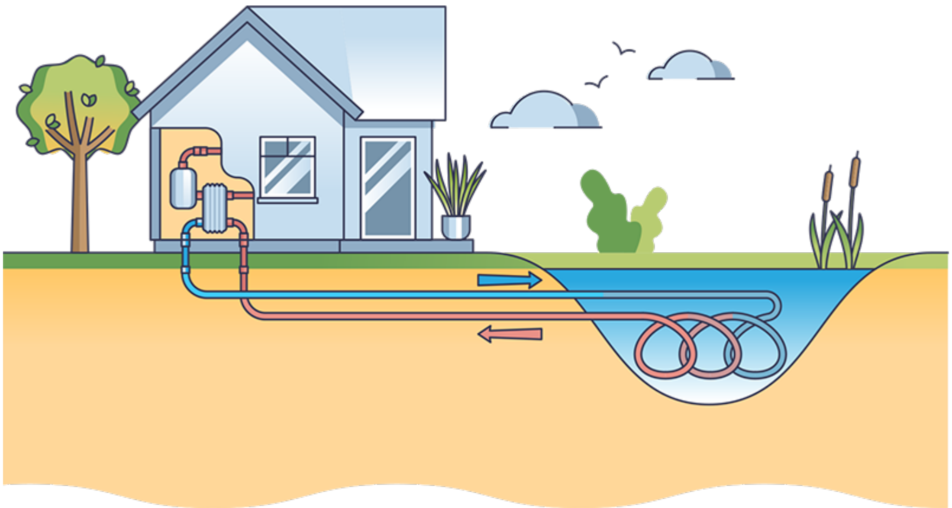


Figure V.3 Illustrative sketch of a water-source Heat Pump

6. Chapter Summary

This chapter introduced heat pumps as energy systems that transfer heat from a low-temperature source to a higher-temperature space using external energy, typically supplied to a compressor. Unlike conventional heating systems that generate heat from fuel or electricity, heat pumps move existing heat from the environment, making them highly efficient and increasingly important in modern energy applications.

The operating principle of a heat pump is based on the reversed vapor-compression cycle, in which the refrigerant circulates through the evaporator, compressor, condenser, and expansion device. The fluid flow diagram helps visualize this process and shows how heat is absorbed from a natural source and delivered to the heated space. The use of a reversing valve allows the same system to operate in both heating mode during winter and cooling mode during summer.

The chapter also presented performance analysis of heat pumps, focusing on the coefficient of performance (COP) in both heating and cooling modes. It was shown that system efficiency depends strongly on the temperature difference between the heat source and the heat sink, as well as on real operating conditions such as compressor efficiency and heat losses.

Different types of heat pumps were discussed according to the heat source used, including air-source, ground-source (geothermal), and water-source systems. Each type has specific advantages, limitations, and applications depending on climate conditions, installation requirements, and available energy resources.

Overall, heat pumps represent an important technology for energy-efficient heating and cooling in residential, commercial, and industrial applications. Understanding their operating principles, performance characteristics, and types is essential for designing and evaluating modern thermal systems and for promoting sustainable energy use.

7. Solved Exercises

Things Engineers Think About

Note to students

Some questions in this section are not explicitly answered in the course document. They are intended to complement the course by encouraging application of concepts and practical engineering reasoning. The answers provided therefore extend the lecture material and help develop understanding of real refrigeration system operation.

1. Why is a heat pump able to deliver more heat energy than the electrical energy supplied to the compressor?
2. What is the fundamental thermodynamic difference between a heat pump and an electric resistance heater?
3. Why is the reversed vapor-compression cycle suitable for heat pump operation?
4. How does the choice of refrigerant influence heat pump performance and operating limits?
5. Why is it necessary to maintain a pressure difference between the evaporator and the condenser?
6. What is the role of the fluid flow diagram in understanding heat pump operation?
7. Why is a reversing valve required for a reversible heat pump?
8. How does the direction of refrigerant flow change between heating and cooling modes?
9. What practical problems could arise if the reversing valve malfunctions?
10. Why is defrosting necessary in air-source heat pumps during winter operation?
11. Why does the COP of a heat pump decrease as outdoor temperature drops?
12. How does increasing the temperature difference between the heat source and heat sink affect compressor work?
13. Why is the COP of a heat pump in heating mode always greater than the COP of the same system operating as a refrigerator?
14. Why is seasonal performance (SPF) more representative than instantaneous COP for real installations?
15. How do irreversibilities in the compressor affect overall heat pump efficiency?
16. Why do ground-source heat pumps generally have higher and more stable performance than air-source heat pumps?

17. In what situations would a water-source heat pump be preferred over air-source or ground-source systems?

18. What engineering factors must be considered when selecting a heat source for a heat pump?

19. Why are heat pumps particularly well suited for low-temperature heating systems such as underfloor heating?

20. How do heat pumps contribute to energy savings and reduction of greenhouse gas emissions?

Answers

1. Because it transfers heat from the environment in addition to the energy supplied to the compressor.

2. A resistance heater converts electricity directly into heat, while a heat pump moves existing heat, making it more efficient.

3. It allows heat to be absorbed at low temperature and delivered at higher temperature efficiently.

4. Refrigerant properties affect pressure levels, temperature range, efficiency, and safety.

5. It enables heat absorption in the evaporator and heat release in the condenser.

6. t shows refrigerant circulation, component connections, and heat transfer locations.

7. It changes refrigerant flow direction to switch between heating and cooling modes.

8. In heating mode, heat is released indoors; in cooling mode, heat is rejected outdoors.

9. System may fail to switch modes or operate inefficiently.

10. Frost forms on the outdoor coil, reducing heat transfer and airflow.

11. The compressor must work harder to extract heat, increasing energy consumption.

12. Compressor work increases and efficiency decreases.

13. Heat delivered includes both absorbed heat and compressor work.

14. It reflects real operating conditions over time, not just instantaneous performance.

15. They increase energy consumption and reduce system efficiency.

16. Ground temperature is stable, leading to more efficient operation.

17. When a reliable water source is available and high efficiency is required.

18. Temperature stability, availability, installation cost, and environmental impact.

19. They operate efficiently when delivering moderate heating temperatures.

20. They reduce electricity and fuel consumption and rely on renewable environmental heat.

Exercise 1

An ideal vapor-compression heat pump cycle with Refrigerant 134a as the working fluid provides heating at a rate of 15 kW to maintain a building at 20 °C when the outside temperature is 5°C. Saturated vapor at 2.4 bar leaves the evaporator, and saturated liquid at 8 bar leaves the condenser. Calculate

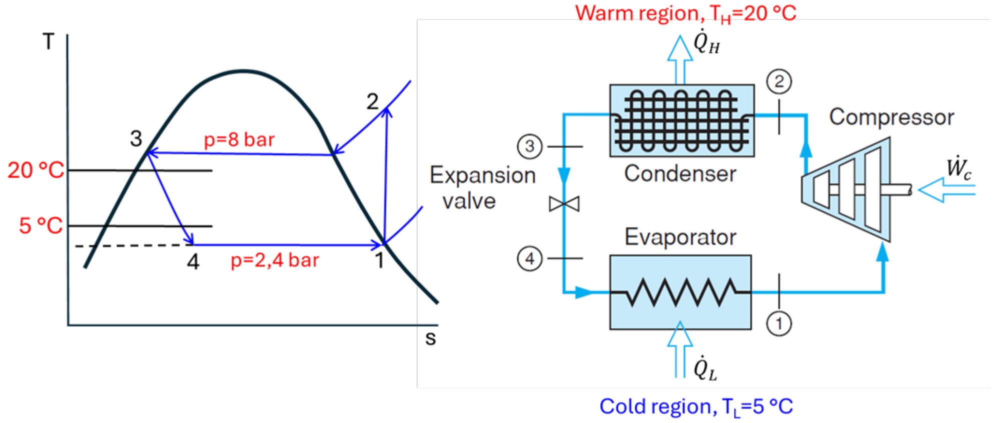
a. the power input to the compressor, in kW,

b. the coefficient of performance,

c. the coefficient of performance of a Carnot heat pump cycle operating between thermal reservoirs at 20 and 5 °C.

Solution

Schematic and T-s diagram:



State 1: $p_1 = 2.4$ bar, saturated vapor

$h_1 = 244.09$ kJ/kg ; $s_1 = s_2 = 0.9222$ kJ/kg. K

State 2: $p_2 = 8$ bar, $s_1 = s_2$; supertfeated vapor

$h_2 = 268.97$ kJ/kg

State 3: $p_3 = 8$ bar, saturated liquid

$h_3 = 93.42$ kJ/kg

State 4:

$h_4 = h_3 = 93.42 \text{ kJ/kg}$

a) the power input to the compressor, in kW:

$\dot{W}_c = \dot{m}_R(h_2 - h_1)$

$\dot{Q}_H = \dot{m}_R(h_2 - h_3) \Longrightarrow \dot{m}_R = \frac{\dot{Q}_H}{h_2 - h_3}$

$\dot{m}_R = \frac{15}{268.97 - 93.42} = 0.085 \text{ kg/s}$

$\Longrightarrow \dot{W}_c = 0.085 \times (268.97 - 244.09) = 2.11 \text{ kW}$

b) the coefficient of performance:

$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_c}$

$COP_{HP} = \frac{15}{2.11} = 7.1$

c) the coefficient of performance of a Carnot heat pump cycle:

$$COP_{Carnot} = \frac{\dot{Q}_H}{\dot{Q}_H - \dot{Q}_L} = \frac{T_H}{T_H - T_L}$$

$$COP_{Carnot} = \frac{293.15}{293.15 - 278.15} = 19.5$$

Exercise 2

A vapor-compression heat pump system uses ammonia as the working fluid. The refrigerant enters the compressor at 2.5 bar, -5 °C, with a volumetric flow rate of 0.6 m³/min. Compression is adiabatic to 14 bar, 140 °C, and saturated liquid exits the condenser at 14 bar. Determine

a. the power input to the compressor, in kW,

b. the heating capacity of the system, in kW and tons,

c. the coefficient of performance,

d. the isentropic compressor efficiency.

Solution

State 1: $p_1 = 2.5 \text{ bar}, T_1 = 5°\text{C} \Longrightarrow$ superheated vapor, because saturation temperature at 2.5 bar is -13.67°C)

$h_1 = 1447.10 \text{ kJ/kg} \,;\, s_1 = s_{2s} = 5.5989 \text{ kJ/kg.K} \,,\,;\, v_1 = 0.50180 \text{ m}^3\text{/kg}$

State 2: $p_2 = 14 \text{ bar}, T_2 = 140°\text{C};\ \text{supertfeated vapor}$

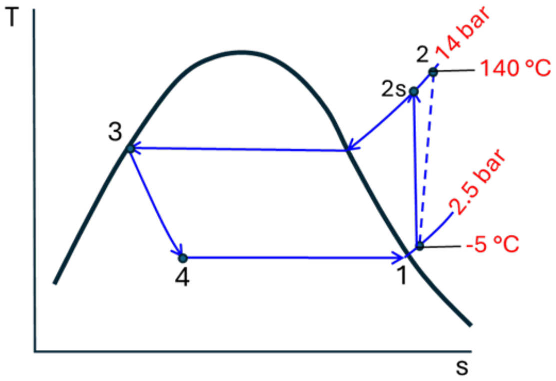
$h_2 = 1752.52 \text{ kJ/kg};\ h_{2s} = 1711.15 \text{ kJ/kg}$

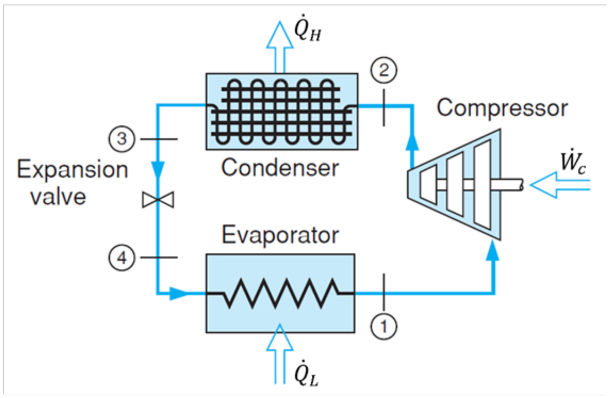
State 3: $p_3 = 14\text{ bar, saturated liquid}$

$h_3 = 352.97\text{ kJ/kg}$

State 4:

$h_4 = h_3 = 352.97\text{ kJ/kg}$





a) the power input to the compressor, in kW:

$$\dot{W}_c = \dot{m}_R(h_2 - h_1)$$

$$\dot{m}_R = \frac{\dot{V}_1}{v_1} = \frac{0.6/60}{0.50180} = 0.0199\text{ kg/s}$$

$$\Longrightarrow \dot{W}_c = 0.0199 \times (1752.52 - 1447.10) = 6.08\text{ kW}$$

b) the heating capacity of the system, in kW and tons:

$$\dot{Q}_H = \dot{m}_R(h_2 - h_3)$$

$$\dot{Q}_H = 0.0199(1752.52 - 352.97) = 27.85\text{ kW}$$

the isentropic compressor efficiency:

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$\eta_c = \frac{1711.15 - 1447.10}{1752.52 - 1447.10} = 0.865\ (86.5\%)$$

Exercise 3

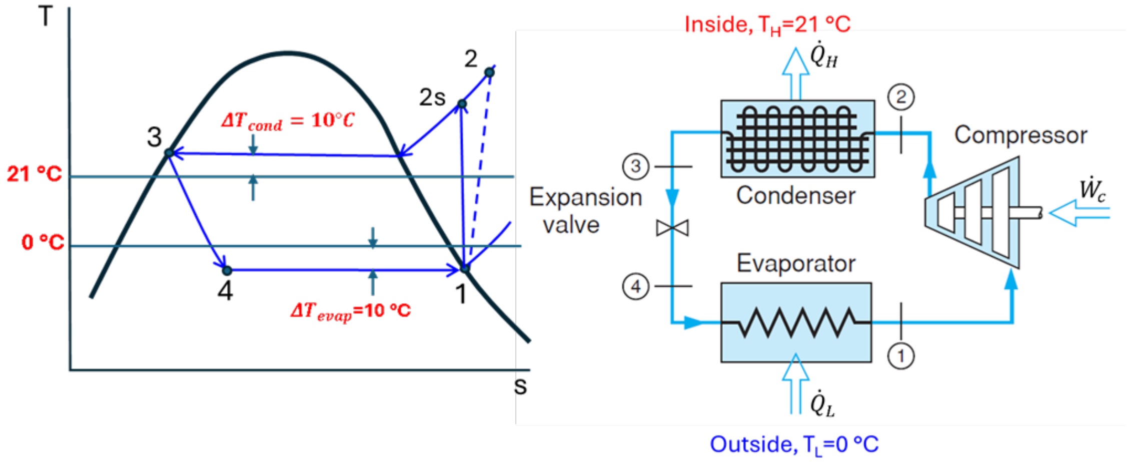
An office building requires a heat transfer rate of 20 kW to maintain the inside temperature at 21°C when the outside temperature is 0°C. A vapor-compression heat pump with Refrigerant 134a as the working fluid is to be used to provide the necessary heating. The compressor operates adiabatically with an isentropic efficiency of 82%.

Specify appropriate evaporator and condenser pressures of a cycle for this purpose assuming $\Delta T_{cond} = \Delta T_{evap} = 10°\text{C}$, as shown in the Figure below. The refrigerant exits the evaporator as saturated vapor and exits the condenser as saturated liquid at the respective pressures. Determine

a. the mass flow rate of refrigerant, in kg/s,

b. the compressor power, in kW,

c. the coefficient of performance and compare with the coefficient of performance for a Carnot heat pump cycle operating between reservoirs at the inside and outside temperatures, respectively.



Solution

The evaporator and condenser pressures must be chosen to allow for sufficient ΔT's to avoid excessive heat exchanger sizes (surface area).

For $\Delta T_{evap} = 10$ °C $\Longrightarrow$ T_1 =-10°C $\Longrightarrow$ p_1 =2.0122 bar

For $\Delta T_{cond} = 10$ °C $\Longrightarrow$ $T_3 = 31$ °C $\Longrightarrow$ $p_3 = p_2$ =7.9267 bar

State 1: $T_1 = -10\ °\mathrm{C},$ saturated vapor

$h_1 = 241.35\ \mathrm{kJ/kg}\ ;\ s_1 = s_{2s} = 0.9253\ \mathrm{kJ/kg.K}$

State 2: $p_2 = 7.9267\ \mathrm{bar}, s_{2s} = 0.9253\ \mathrm{kJ/kg.K}\ ;$ supertfeated vapor

$h_{2s} = 270.15\ \mathrm{kJ/kg}$

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_c} = 241.35 + (270.15 - 241.35)/0.82$$

$h_2 = 276.47\ \mathrm{kJ/kg}$

State 3: $T_3 = 31\ °\mathrm{C},$ saturated liquid

$h_3 = 92.94\ \mathrm{kJ/kg}$

State 4:

$h_4 = h_3 = 92.94\ \mathrm{kJ/kg}$

a) the mass flow rate of refrigerant, in kg/s :

$$\dot{Q}_H = \dot{m}_R(h_2 - h_3)$$

$$\Longrightarrow \dot{m}_R = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{20}{276.47 - 92.94} = 0.109 \text{ kg/s}$$

b) the compressor power, in kW:

$$\dot{W}_c = \dot{m}_R(h_2 - h_1)$$

$$\Longrightarrow \dot{W}_c = 0.109 \times (276.47 - 241.35) = 3.83 \text{ kW}$$

the coefficient of performance

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_c}$$

$$COP_{HP} = \frac{20}{3.83} = 5.22$$

$$COP_{Carnot} = \frac{T_H}{T_H - T_L}$$

$$COP_{Carnot} = \frac{294.15}{294.15 - 273.15} = 14.01$$

Relative deviation is evaluated as follows:

$$\text{RD} = \frac{COP_{Carnot} - COP_{HP}}{COP_{HP}} = \frac{14.01 - 5.22}{5.22} = 1.68$$

Exercise 4

A vapor-compression heat pump with a heating capacity of 500 kJ/min is driven by a power cycle with a thermal efficiency of 25%. For the heat pump, Refrigerant 134a is compressed from saturated vapor at -10 °C to the condenser pressure of 10 bar. The isentropic compressor efficiency is 80%. Liquid enters the expansion valve at 9.6 bar, 34°C. For the power cycle, 80% of the heat rejected is transferred to the heated space.

(a) Determine the power input to the heat pump compressor, in kW.

(b) Evaluate the ratio of the total rate that heat is delivered to the heated space to the rate of heat input to the power cycle. Discuss.

Solution

<u>Schematic and Given data</u>: see Figure below and T-s Diagram for the heat pump cycle.

State 1: $T_1 = -10\ °\text{C}$, saturated vapor

$h_1 = 241.35\ \text{kJ/kg}\ ;\ s_1 = s_{2s} = 0.9253\ \text{kJ/kg.K}$

State 2: $p_2 = 10\ \text{bar}, s_{2s} = 0.9253\ \text{kJ/kg.K}\ ;\ \text{supertfeated vapor}$

$h_{2s} = 274.63\ \text{kJ/kg}$

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_c} = 241.35 + (274.63 - 241.35)/0.80$$

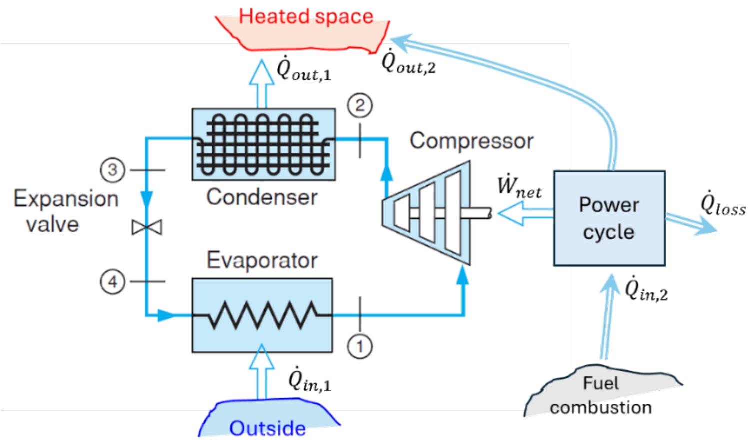
$h_2 = 282.95\ \text{kJ/kg}$

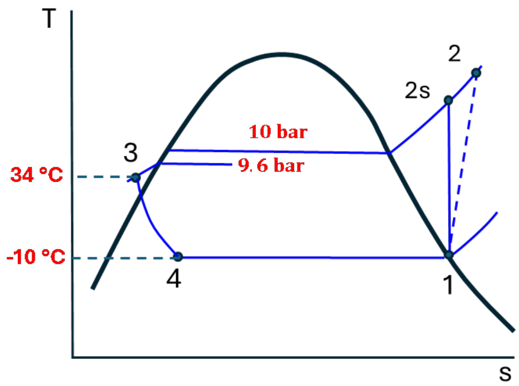
State 3: $p_3 = 9.6\ \text{bar}, T_3 = 34°\text{C} \Longrightarrow h_3 \approx h_f(34°\text{C})$

$h_3 = 97.31\ \text{kJ/kg}$

State 4:

$h_4 = h_3 = 97.31\ \text{kJ/kg}$





a) the compressor power, in kW

$\dot{W}_c = \dot{m}_R(h_2 - h_1)$

$$\dot{m}_R = \frac{\dot{Q}_{out,1}}{h_2 - h_3} = \frac{500/60}{282.95 - 97.31} = 0.04489\ \text{kg/s}$$

$\Longrightarrow \dot{W}_c = 0.04489 \times (282.95 - 241.35) = 1.87\ \text{kW}$

b) the ratio of the total rate that heat is delivered to the heated space to the rate of heat input to the power cycle

For the power cycle, $\eta = 0.25$. With $\dot{W}_{power} = \dot{W}_{HP} = 1.87 \text{ kW}$

$$\dot{Q}_{in,2} = \frac{\dot{W}_{power}}{\eta} = \frac{1.87}{0.25} = 7.48 \text{ kW}$$

The total heat rejected is, $\dot{Q}_{rej} = 7.48 - 1.87 = 5.61 \text{ kW}$

$$\Longrightarrow \dot{Q}_{out,2} = 0.8 \times \dot{Q}_{rej} = 0.8 \times 5.61 = 4.49 \text{ kW}$$

$$\frac{\dot{Q}_{out,1} + \dot{Q}_{out,2}}{\dot{Q}_{in,2}} = \frac{500/60 + 4.49}{7.48} = 1.71$$

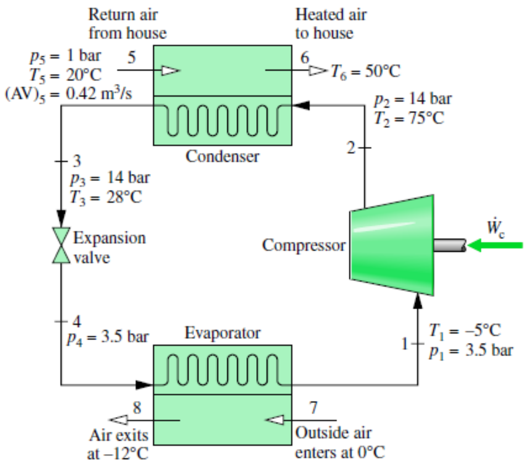
Discussion: The engine delivers more energy to the heated space than could be obtained by burning the fuel directly.

Exercise 5

A residential heat pump system operating at steady state is shown schematically in the Figure below. Refrigerant 22 circulates through the components of the system, and property data at the numbered states are given on the figure. The compressor operates adiabatically. Kinetic and potential energy changes are negligible as are changes in pressure of the streams passing through the condenser and evaporator. Determine

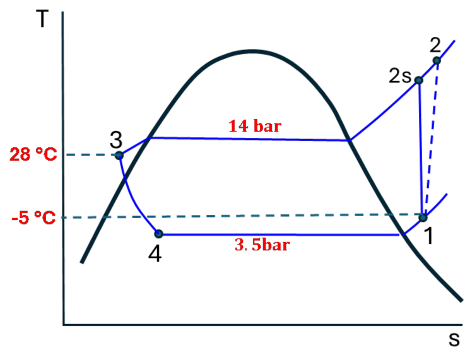
a. the power required by the compressor, in kW, and the isentropic compressor efficiency.

b. the coefficient of performance.



Solution

T-s Diagram



State 1: $p_1 = 3.5 \text{ bar}, T_1 = -5\ °\text{C}$, superheated vapor

$h_1 = 249.75 \text{ kJ/kg}$, $s_1 = s_{2s} = 0.9572 \text{ kJ/kg. K}$

State 2: $p_2 = 14 \text{ bar}$, $T_2 = 70\ °\text{CkJ/kg. K}$; supertfeated vapor

$h_{2s} = 285.58 \text{ kJ/kg}$

$h_2 = 294.18 \text{ kJ/kg}$

State 3: $p_3 = 14 \text{ bar}, T_3 = 28°\text{C (subcooled liquid)} \Rightarrow h_3 \approx h_f(28°\text{C})$

$h_3 = 79.05 \text{ kJ/kg}$

State 4:

$h_4 = h_3 = 79.05 \text{ kJ/kg}$

a) the power required by the compressor

$$\dot{W}_c = \dot{m}_R(h_2 - h_1)$$

Energy balance for the condenser

$$\dot{m}_R(h_2 - h_3) = \dot{m}_a(h_6 - h_5) = \dot{m}_a C_{pa}(T_6 - T_5)$$

$$\dot{m}_R = \frac{\dot{m}_a C_{pa}(T_6 - T_5)}{(h_2 - h_3)}$$

$$\dot{m}_a = \frac{p_a(AV)}{R_a T_5} = \frac{10^5 \times 0.42}{287 \times 293.15} = 0.5 \text{ kg/s}$$

$$\dot{m}_R = \frac{0.5 \times 1.0035 \times (50 - 20)}{(290.01 - 79.05)} = 0.071 \text{ kg/s}$$

$$\Rightarrow \dot{W}_c = 0.071 \times (294.18 - 249.75) = 3.15 \text{ kW}$$

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$\eta_c = \frac{285.58 - 249.75}{294.18 - 249.75} = 0.81 = 81\%$$

b) the coefficient of performance

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_c}$$

$\dot{Q}_H = \dot{m}_R(h_2 - h_3) = 0.071 \times (294.18 - 79.05) = 15.27\ \text{kW}$

$$COP_{HP} = \frac{15.27}{3.15} = 4.85$$

Conclusion

This course has provided a structured and progressive introduction to refrigeration systems and heat pumps, with particular emphasis on the thermodynamic principles governing their operation. Starting from the fundamental concepts of heat transfer and the Carnot refrigeration cycle, the course established the theoretical limits of performance and introduced the coefficient of performance, COP, as a central indicator of system efficiency.

The study then focused on the vapor compression refrigeration cycle, which represents the most widely used technology in industrial and commercial applications. Through the analysis of ideal and practical cycles using T–s and P–h diagrams, students developed the ability to interpret thermodynamic transformations, evaluate system performance, and understand the influence of irreversibilities such as pressure losses, non-isentropic compression, and heat exchange with the surroundings. Energy balance equations were systematically applied to each component including the compressor, condenser, expansion device, and evaporator, linking thermodynamic theory to real engineering calculations.

The course also introduced refrigerant classification and selection criteria, highlighting the importance of thermodynamic performance, environmental impact, safety considerations, and regulatory constraints. Industrial applications including air conditioning, food preservation, freeze drying, and cryogenic processes demonstrated the essential role of refrigeration in modern society and engineering practice.

Beyond mechanical compression systems, alternative technologies such as absorption refrigeration and air standard, also called reverse Brayton, cycles were presented to broaden students' understanding of different refrigeration principles. The study of heat pumps further illustrated the reversibility of thermodynamic cycles and emphasized the dual function of heating and cooling, as well as seasonal performance considerations.

Overall, this course has built a solid conceptual and methodological foundation in refrigeration and heat pump technology. Students are now able to describe the operating principles of refrigeration machines and heat pumps, interpret T–s and P–h diagrams for ideal and real cycles, perform basic energy balance and COP calculations, understand the role and selection of refrigerants, and recognize the main components and their functions in practical systems.

These competencies form a strong basis for advanced studies in thermal systems, energy engineering, sustainable refrigeration technologies, and high efficiency heat pump design.

Recommended Textbooks

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[7] William C. Whitman, William M. Johnson, John A. Tomczyk, Eugene Silberstein, Refrigeration & Air Conditioning Technology, 7th Edition, Delmar, Cengage Learning (2013).